An Introduction to Thermogeology: Ground Source Heating and Cooling

For Jenny 'the Bean'

An Introduction to Thermogeology: Ground Source Heating and Cooling

2nd Edition

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UK

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Main photo: Coils of heat exchange pipe can be installed in natural lakes. They can be mounted in a steel frame, rowed out, filled and sunk to the base of the lake. Photo by kind permission of Geowarmth Heat Pumps Ltd. of Newcastle-upon-Tyne.

Top inset photo: Staff of the Geological Survey of Norway carry out a thermal response test on a closed loop heat exchange borehole drilled into greenstone rocks in Trondheim. Photo by David Banks.

Bottom inset photo: An underground house in Matmata, Tunisia. The rocks store 'coolth' from winter and night-time, such that the underground is much cooler than the surface at the height of summer. Photo by David Banks.

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About the Author

David BANKS was born in Bishop Auckland in 1961. He is a hydrogeologist with 26 years experience of investigating groundwater-related issues. He started his career with the Thames Water Authority in southern England, then moved across the North Sea to the Geological Survey of Norway, where he eventually headed the Section for Geochemistry and Hydrogeology. Since returning to the United Kingdom in 1998, he has worked as a consultant from a base in Chesterfield, sandwiched between the gritstone of the Peak District National Park and the abandoned mines of Britain's largest coalfield. He has international experience from locations as diverse as Afghanistan, the Bolivian Altiplano, Somalia, Western Siberia, Darfur and Huddersfield. During the past 10 years, his attention has turned to the emerging science of thermogeology: he has worked closely with the ground source heat industry and has also enjoyed spells as a Senior Research Associate in Thermogeology at the University of Newcastle-upon-Tyne. Most recently, he was employed by Newcastle University to provide input to the European Union 'GeoTrainet' program of geothermal education.

In his spare time, Dave enjoys music. With his chum Bjørn Frengstad, he has formed almost one half of the sporadically active acoustic lo-fi stunt duo 'The Sedatives'. They have murdered songs by their musical heroes (who include Jarvis Cocker, Benny Andersen, Richard Thompson and Katherine Williams) in a variety of seedy locations.

Reviews of 'An Introduction to Thermogeology'

> '. . . it is seldom that one needs to use superlatives when talking about a book . . . this book should be a bible for all who would like to gain insight into the nature of the earth's heat, and how we can exploit it in practice'.
>
> Inga Sørensen, writing in Geologisk Nyt,
> Denmark, August 2009

Other books by the same author

With Bruce Misstear and Lewis Clark, Dave Banks has previously co-authored 'Water Wells and Boreholes', currently available from Wiley.

> 'The book is fulsome. It is a complete counterbalance to the common, but naïve, notion that if you want a new water well "you just go out and get yourself a driller." This book

explains how to do it properly. . . . It is an important achievement. I expect that it will become a "Bible" that will be on the desk or in the field with every practical hydrogeologist . . .'.

David Ball, writing in the Geological Survey of Ireland Newsletter

'. . . it far outshines most other volumes with which it might otherwise be compared. . . . I would recommend every aspiring and practising hydrogeologist to buy it and thumb it to pieces'.

Paul Younger, writing in the Quarterly Journal of Engineering Geology and Hydrogeology

Preface to the First Edition

In the late 1990s, I was working for the Norwegian Geological Survey's Section for Hydrogeology and Geochemistry. Despite the Section being choc-a-bloc with brainy research scientists, one of my most innovative colleagues was an engineer who called me, on what seemed a weekly basis, brimming with enthusiasm for some wizard new idea. One day, he started telling me all about something called *grunnvarme* or ground source heat, which was, apparently, very big in Sweden. Initially, it seemed to me to be something akin to perpetual motion – space heating from Norwegian rock at 6°C? – and in violation of the second law of thermodynamics to boot. Nevertheless, he persuaded me that it really did have a sound physical basis. In fact, my chum went on to almost single-handedly sell the concept of ground source heat to a Norwegian market that was on the brink of an energy crisis. A subsequent dry summer that pulled the plug on Norway's cheap hydroelectric supplies and sent prices soaring was the trigger that ground source heat needed to take off. So, firstly, a big thank you to Helge Skarphagen (for it was he!), who first got me interested in ground source heat.

On my return to England in 1998, I tried to bore anyone who gave the appearance of listening about the virtues of ground source heat (I was by no means the first to try this – John Sumner and Robin Curtis, among others, had been evangelists for the technology much earlier). It was not until around 2003, however, that interest in ground source heat was awakened in Britain and I was lucky enough to fall in with a group of entrepreneurs with an eye for turning it into a business. So, secondly, many thanks to GeoWarmth of Hexham (now based at Newcastle) for the pleasure of working with you, and especially to Dave Spearman, Jonathan Steven, Braid and Charlie Aitken, Nick Smith and John Withers.

Oh, and by the way, Jenny, I don't know what you've been up to while I've been locked in the attic writing this book, but normal parental service will shortly be resumed!

David Banks
Chesterfield, Derbyshire, 2007

Preface to the Second Edition

This book is written for an international audience. It aims to aid professionals in conceptualising the ground – heat exchanger – building linkages at the heart of ground-coupled heat exchange systems. Forgive me, therefore, if I focus for a few moments on my own recent British experiences in this Preface.

At the time of writing the first edition of this book, the ground source heating and cooling industry in the United Kingdom was in its infancy and growing fast. Four years later, the profession is much larger but is still regrettably immature.

British ground source heat pump meetings abound with grey-suited salesmen (and they are invariably men) warning us to be on our guard against 'cowboys', who will drag our profession into disrepute by their ignorance. Quite who these 'cowboys' are is never fully explained . . . probably due to the fact that the speaker himself is a 'cowboy', as are most of the audience. The hard truth is that almost all ground source heat practitioners (with a few honourable exceptions) in the United Kingdom are relatively new to the science and are still learning fast. Indeed, I will myself admit to being just such a 'cowboy'. But, hey, being a cowboy can be fun – cowboys are pioneers, blazing a trail in unknown terrain. Cowboys can be rough and ready and can make mistakes but, with time and experience, they will form the backbone of a new frontier community. Westward ho and wagons roll!

The UK ground source heat pump market is still reported to be the fastest growing in the world (Lund, 2010). However, I trust that the time is approaching when the cowboys are beginning to settle, to form professional communities and to get a real grasp of their tools and terrain. We should be reaching a stage where we are not merely building ground source heat systems that work, but ones that work really efficiently. At the time of writing, the industry is still reeling from the implications of a report by the Energy Saving Trust (2010), which found that the system performance factors of UK ground source heat pump systems were typically as low as between 2 and 3. Such low efficiencies risk not only that the system fails to save the owner any money in operational costs, but also that it ultimately releases more atmospheric CO_2 than a conventional mains gas boiler. This is very bad news for the industry. The UK industry has, in response, forced out new standards designed to promote significantly more efficient systems (GSHPA, 2011; MIS, 2011a,b,c).

I feel that the time is ripe for a second edition of this book. This edition will not merely cover the thermophysics of subsurface heat transfer and the conceptualisation of a ground heat exchange system. It will also address many of the key issues involved in designing efficient systems: the impact of design loop temperatures and hydraulics, the impact of client pressure to cut capital costs, the influence of building (load side) heat delivery decisions and the importance of energy storage. It will attempt to stress the importance of considering system design, not merely in terms of thermogeological variables, but also in the light of your nation's physical climate and energy/carbon economy.

David Banks

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- Dr Robin Curtis, now with Mimer Energy Ltd in Falmouth
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I am grateful to all those who have provided materials, case studies, inspiration, diagrams, photos and more – Pablo Fernández Alonso, Hannah Russell, Johan Claesson, Richard Freeborn of Kensa Engineering, Bjørn Frengstad of the fabulous 'Sedatives', the International Ground Source Heat Pump Association (IGSHPA), the Geotrainet team, Ben Lawson and Paul Younger of Newcastle University, John Lund and the Geo-Heat Centre in Oregon, Jim Martin, John Parker, Umberto Puppini of ESI Italia, Kevin Rafferty, Randi Kalskin Ramstad of Asplan VIAK, Igor Serëdkin, Chris Underwood of Northumbria University and many more. I'd like to thank John Vaughan of Chesterfield Borough Council, Doug Chapman of Annfield Plain, Les Gibson of Grindon Camping Barn and Lionel Hehir of the Hebburn Eco-Centre for permission to use their buildings as case studies. Janet Giles was heroic in assisting with administration and printing. Lastly, Ardal O'Hanlon has unwittingly provided me with the alter ego of *Thermoman*, which I have occasionally assumed while lecturing. While I have tried to contact all copyright holders of materials reproduced in this book, in one or two cases you have proved very elusive! If you are affected by any such omission, please do not hesitate to contact me, via Wiley, and we will do our utmost to ensure that the situation is rectified for any future editions of this book.

1

An Introduction

> Nature has given us illimitable sources of prepared low-grade heat. Will human organisations cooperate to provide the machine to use nature's gift?
>
> John A. Sumner (1976)

Many of you will be familiar with the term *geothermal energy*. It probably conjures mental images of volcanoes or of power stations replete with clouds of steam, deep boreholes, whistling turbines and hot saline water. This book is *not* primarily about such geothermal energy, which is typically high temperature (or high enthalpy, in technospeak) energy and is accessible only at either specific geological locations or at very great depths. This book concerns the relatively new science of *thermogeology*. Thermogeology involves the study of so-called *ground source heat*: the mundane form of heat that is stored in the ground at normal temperatures. Ground source heat is much less glamorous than high-temperature geothermal energy, and its use in space heating is often invisible to those who are not 'in the know'. It is hugely important, however, as it exists and is accessible everywhere. It genuinely offers an attractive and powerful means of delivering CO_2-efficient space heating and cooling.

Let me offer the following definition of *thermogeology*:

> Thermogeology is the study of the occurrence, movement and exploitation of low enthalpy heat in the relatively shallow geosphere.

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

By 'relatively shallow', we are typically talking of depths of down to 300 m or so. By 'low enthalpy', we are usually considering temperatures of less than 40°C.[1]

1.1 Who should read this book?

This book is designed as an introductory text for the following audience:

- graduate and postgraduate level students;
- civil and geotechnical engineers;
- buildings services and heating, ventilation and air conditioning (HVAC) engineers who are new to ground source heat;
- applied geologists, especially hydrogeologists;
- architects;
- planners and regulators;
- energy consultants.

1.2 What will this book do and not do?

This book is not a comprehensive manual for designing ground source heating and cooling systems for buildings: it is rather intended to introduce the reader to the concept of thermogeology. It is also meant to ensure that architects and engineers are aware that there is an important geological dimension to ground heat exchange schemes. The book aims to cultivate awareness of the possibilities that the geosphere offers for space heating and cooling and also of the limitations that constrain the applications of ground heat exchange. It aims to equip the reader with a *conceptual model* of how the ground functions as a heat reservoir and to make him or her aware of the important parameters that will influence the design of systems utilising this reservoir.

While this book will introduce you to design of ground source heat systems and even enable you to contribute to the design process, it is important to realise that a sustainable and successful design needs the integrated skills of a number of sectors:

- The thermogeologist
- The architect, who must ensure that the building is designed to be heated using the relatively low-temperature heating fluids (and cooled by relatively high-temperature chilled media) that are produced efficiently by most ground source heat pump/heat exchange schemes.

[1] Although in conventional geothermal science, anything up to around 90°C is still considered 'low enthalpy'!

- The buildings services/HVAC engineer, who must implement the design and must design hydraulically efficient collector and distribution networks, thus ensuring that the potential energetic benefits of ground heat exchange systems are not frittered away in pumping costs.
- The electromechanical and electronic engineer, who will be needed to install the heat pump and associated control systems
- The pipe welder and the driller, who will be responsible for installing thermally efficient, environmentally sound and non-leaky ground heat exchangers.
- The owner, who needs to appreciate that an efficient ground heat exchange system must be operated in a wholly different way to a conventional gas boiler (e.g. ground source heat pumps often run at much lower output temperatures than a gas boiler and will therefore be less thermally responsive).

If you are a geologist, you must realise that you are not equipped to design the infrastructure that delivers heat or cooling to a building. If you are an HVAC engineer, you should acknowledge that a geologist can shed light on the 'black hole' that is your ground source heat borehole or trench. In other words, you need to talk to each other and work together! For those who wish to delve into the hugely important 'grey area' where geology interfaces in detail with buildings engineering, to the extent of consideration of pipe materials and diameters, manifolds and heat exchangers, I recommend that you consult one of several excellent manuals or software packages available. In particular, I would name the following:

- the manual of Kavanaugh and Rafferty (1997) – despite its insistence on using such unfamiliar units as $\mathrm{Btu\,ft^{-1}\,{}^{\circ}F^{-1}}$, so beloved of our American cousins;
- the set of manuals issued by the International Ground Source Heating Association (IGSHPA) – IGSHPA (1988), Bose (1989), Eckhart (1991), Jones (1995), Hiller (2000), and IGSHPA (2007);
- the recent book by Ochsner (2008a);
- the newly developed Geotrainet (2011) manual, which has a specifically European perspective and has been written by some of the continent's foremost thermophysicists, thermogeologists and HVAC engineers;
- the German Engineers' Association standards (VDI, 2000, 2001a,b, 2004, 2008);
- numerous excellent booklets aimed at different national user communities, such as that of the Energy Saving Trust (2007).

1.3 Why should you read this book?

You should read this book because *thermogeology is important for the survival of planet Earth!* Although specialists may argue about the magnitude of climate change ascribable to greenhouse gases, there is a broad consensus (IPCC, 2007) that the continued emission of fossil carbon (in the form of CO_2) to our atmosphere has the potential to detrimentally alter our planet's climate and ecology. Protocols negotiated via international conferences, such as those at Rio de Janeiro (the so-called Earth Summit) in 1992 and at Kyoto in 1997, have attempted to commit nations to dramatically reducing their emissions of greenhouse gases [carbon dioxide, methane, nitrous oxide, sulphur hexafluoride, hydrofluorocarbons (HFCs) and perfluorocarbons (PFCs)] during the next decades.

Even if you do not believe in the concept of anthropogenic climate change, recent geopolitical events should have convinced us that it is unwise to be wholly dependent on fossil fuel resources located in unstable parts of the world or within nations whose interests may not coincide with ours. Demand for fossil fuels is increasingly outstripping supply: the result of this is the rise in oil prices over the last decade. This price hike is truly shocking, not least because most people seem so unconcerned by it. A mere 10 years ago, in 1999, developers of a new international oil pipeline were worrying that the investment would become uneconomic if the crude oil price fell below $15 USD per barrel. At the time of writing, Brent crude is some $105 per barrel, and peaked in 2008 at over $140 (Figure 1.1). The increasingly efficient use of the fuel resources we do have access to, and the promotion of local energy sources, must be to our long-term benefit.

I would not dare to argue that the usage of ground source heat alone will allow us to meet all these objectives. Indeed, many doubt that we will be able to adequately reduce fossil carbon emissions soon enough to significantly brake the effects of global warming. If we are to make an appreciable impact on net fossil carbon emissions, however, we will undoubtedly need to consider a wide variety of strategies, including the following:

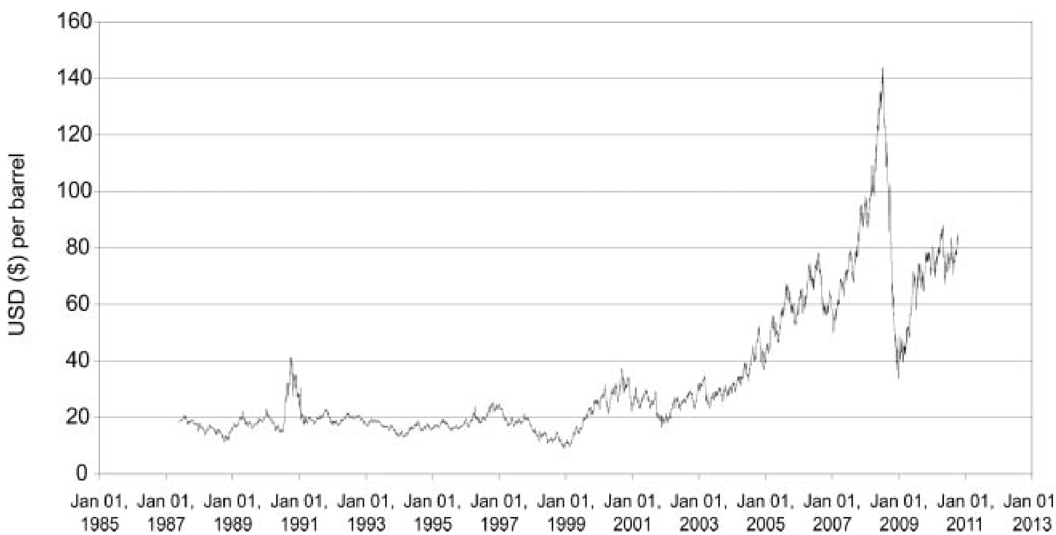


Figure 1.1 Spot prices for Brent Crude Oil in the period 1987–2010 (USD per barrel). Based on the data from the US Energy Information Administration (EIA).

1. A reduction in energy consumption, for example, by more efficient usage of our energy reserves.
2. Utilisation of energy sources not dependent on fossil carbon. The most strategically important of these non-fossil-carbon sources is probably nuclear power (although uranium resources are finite), followed by hydroelectric power. Wind, wave, biomass, geothermal and solar powers also fall in this category.
3. Alternative disposal routes for fossil carbon dioxide, other than atmospheric emission: for example, underground sequestration by injection using deep boreholes.

I will argue, however, that utilisation of *ground source heat* allows us to significantly address issues (1) and (2). Application of *ground source heat pumps* (see Chapter 4) allows us to use electrical energy highly efficiently to transport renewable environmental energy into our homes (Box 1.1).

If the environmental or macroeconomic arguments don't sway you, try this one for size: *Because the regulatory framework in my country is forcing me to install energy-efficient technologies!* The Kyoto Protocol is gradually being translated into European and national legislation, such as the British Buildings Regulations, which not only require highly thermally efficient buildings, but also low-carbon space heating and cooling technologies. Local planning authorities may demand a certain percentage of 'renewable energy' before a new development can be permitted. Ground source heating or cooling may offer an architect a means of satisfying ever more stringent building regulations. It may assist a developer in getting into the good books of the local planning committee.

BOX 1.1 Energy, Work and Power

Energy is an elusive concept. In its broadest sense, energy can be related to the ability to do *work*. Light energy can be converted, via a photovoltaic cell, to electrical energy that can be used to power an electrical motor, which can do *work*. The chemical energy locked up in coal can be converted to heat energy by combustion and thence to mechanical energy in a steam engine, allowing *work* to be done. In fact, William Thomson (Lord Kelvin) demonstrated an equivalence between energy and work. Both are measured in *joules* (J).

Work (W) can defined as the product of the *force* (F) required to move an object and the *distance* (L) it is moved. In other words,

$$W = FL$$

Force is measured in *newtons* and has a dimensionality $[M][L][T]^{-2}$. Thus, work and energy have the same dimensionality $[M][L]^2[T]^{-2}$ and $1\,\mathrm{J} = 1\,\mathrm{kg\,m^2\,s^{-2}}$.

Power is defined as the *rate* of doing work or of transferring energy. The unit of power is the *watt* (W), with dimensionality $[M][L]^2[T]^{-3}$.

$1\,\text{watt} = 1\,\text{joule per second} = 1\,\mathrm{J\,s^{-1}} = 1\,\mathrm{kg\,m^2\,s^{-3}}$.

Finally, the most powerful argument of all: *Because you can make money from ground source heat.* You may be an entrepreneur who has spotted the subsidies, grants and tax breaks that are available to those who install ground source heating schemes. You may be a consultant wanting to offer a new service to a client. You may be a drilling contractor – it is worth mentioning that, in Norway and the United Kingdom, drillers are reporting that they are now earning more from drilling ground source heat boreholes than from their traditional business of drilling water wells. You may be a property developer who has sat down and looked cool and hard at the economics of ground source heat, compared it with conventional systems and concluded that the former makes not only environmental sense, but also economic sense.

1.4 Thermogeology and hydrogeology

You don't have to be a hydrogeologist to study thermogeology, but it certainly helps. A practical hydrogeologist often tries to exploit the earth's store of groundwater by drilling wells and using some kind of pump to raise the water to the surface where it can be used. A thermogeologist exploits the earth's heat reservoir by drilling boreholes and using a ground source heat pump to raise the temperature of the heat to a useful level. The analogy does not stop here, however. There is a direct mathematical analogy between groundwater flow and subsurface heat flow.

We all know that water, left to its own devices, flows downhill or from areas of high pressure to low pressure. Strictly speaking, we say that water flows from locations of high *head* to areas of low head (Box 1.2). Head is a mathematical concept which combines both pressure and elevation into a single value. Similarly, we all know that heat tends to flow from hot objects to cold objects. In fact, a formula, known as Fourier's law, was named after the French physicist Joseph Fourier. It permits us to quantify the heat flow conducted through a block of a given material (Figure 1.2):

$$Q = -\lambda A \frac{d\theta}{dx} \tag{1.1}$$

where

Q = flow of heat in joules per second, which equals watts ($\mathrm{J\,s^{-1}} = \mathrm{W}$),

λ = thermal conductivity of the material ($\mathrm{W\,m^{-1}\,K^{-1}}$),

A = cross-sectional area of the block of material under consideration ($\mathrm{m^2}$),

θ = temperature (°C or K),

x = distance coordinate in the direction of decreasing temperature (note that heat flows in the direction of decreasing temperature: hence the negative sign in the equation),

$\frac{d\theta}{dx}$ = temperature gradient ($\mathrm{K\,m^{-1}}$).

The hydrogeologists have a similar law, Darcy's law, which describes the flow of water through a block of porous material, such as sand:

BOX 1.2 Head

We know intuitively that water tends to flow downhill (from higher to lower elevation). We also know that it tends to flow from high to low pressure. We can also intuitively feel that water elevation and pressure are somehow equivalent. In a swimming pool, water is static: it does not flow from the water surface to the base of the pool. The higher elevation of the water surface is somehow compensated by the greater pressure at the bottom of the pool.

The concept of *head* (h) combines elevation (z) and pressure (P). Pressure (with dimension $[M][L]^{-1}[T]^{-2}$) is converted to an equivalent elevation by dividing it by the water's density (ρ_w: dimension $[M][L]^{-3}$) and the acceleration due to gravity (g: dimension $[L][T]^{-2}$), giving the formula

$$h = z + \frac{P}{\rho_w g}$$

Groundwater always flows from regions of high head to regions of low head. Head is thus a measure of groundwater's potential energy: it provides the potential energy gradient along which groundwater flows according to Darcy's law.

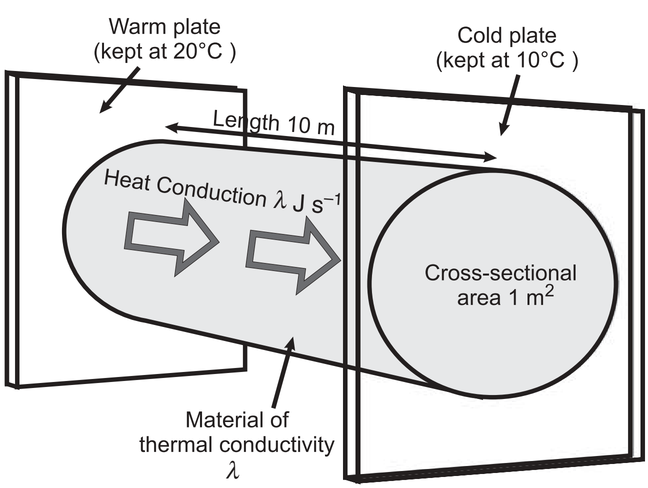


Figure 1.2 The principle of Fourier's law. Consider an insulated bar of material of cross-sectional area 1 m^2 and length 10 m. If one end is kept at 20°C and the other end at 10°C, the temperature gradient is 10 K per 10 m, or 1 K m^{-1}. Fourier's law predicts that heat will be conducted from the warm end to the cool end at a rate of λ J s^{-1}, where λ is the thermal conductivity of the material (in W m^{-1} K^{-1}). We assume that no heat is lost by convection or radiation.

$$Z = -KA\frac{dh}{dx} \tag{1.2}$$

where

Z = flow of water ($m^3 s^{-1}$),

K = hydraulic conductivity of the material ($m s^{-1}$), often referred to as the permeability of the material,

A = cross-sectional area of the block of material under consideration (m^2),

h = head (m),

x = distance coordinate in the direction of decreasing head (m),

$\frac{dh}{dx}$ = head gradient (dimensionless).

A hydrogeologist is interested in quantifying the properties of the ground to ascertain whether it is a favourable target for drilling a water well (Misstear *et al.*, 2006). Two properties are of relevance. Firstly, the permeability (or *hydraulic conductivity*)

BOX 1.3 Maslow, Geology and Human Needs

> Food is the first thing – morals follow on.
>
> Bertolt Brecht, A Threepenny Opera

Abraham Maslow (1908–1970) was an American humanist and psychologist, who studied and categorised fundamental human needs. His ideas are often summarised in some form of tiered structure – a hierarchy of needs – where the lowest levels of need must be fulfilled before a human can pursue happiness and aspire to satisfy his or her higher-level needs. The most familiar conceptualisation involves the following:

Tier 5 – Self-actualisation: includes art, morality
Tier 4 – Esteem: self-respect, respect of others, sense of achievement
Tier 3 – Belonging: friendship, family
Tier 2 – Safety: employment, resources, health, property
Tier 1 – The fundamentals: sex, respiration, food, water, homeostasis, excretion, sleep

Humble hydrogeologists, environmental geochemists and thermogeologists may not be glamorous, but they can comfort themselves with the fact that they are satisfying basic human needs in Tier 1. Hydrogeologists provide potable water and secure disposal of wastes via pit latrines and landfills; environmental geochemists ensure that our soils are fit for cultivation. Thermogeologists contribute to ensuring homeostasis – a flashy word that basically means a controlled environment (shelter), of which space heating and cooling are fundamental aspects.

For sex and sleep, the Geologist's Directory may not be able to assist you.

is an intrinsic property of the rock or sediment that describes how good that material is at allowing groundwater to flow through it. Secondly, the *storage coefficient* describes how much groundwater is released from pore spaces or fractures in a unit volume of rock, for a 1 m decline in groundwater head. A body of rock that has sufficient groundwater storage and sufficient permeability to permit economic abstraction of groundwater is called an *aquifer* (from the Latin 'water' + 'bearing').

In thermogeology, we again deal with two parameters describing how good a body of rock is at *storing* and *conducting* heat. These are the *volumetric heat capacity* (S_{VC}) and the *thermal conductivity* (λ). The former describes how much heat is released from a unit volume of rock as a result of a 1 K decline in temperature, while the latter is defined by Fourier's law (Equation 1.1). We could define an *aestifer* as a body of rock with adequate thermal conductivity and volumetric heat capacity to permit the economic extraction of heat (from the Latin *aestus*, meaning 'heat' or 'summer').[2] In reality, however, all rocks can be economically exploited (depending on the scale of the system required – see Chapter 4; Box 1.3) for their heat content, rendering the definition rather superfluous.

Table 1.1 summarises the key analogies between thermogeology and hydrogeology, to which we will return later in the book.

Table 1.1 The key analogies between the sciences of hydrogeology and thermogeology (see Banks, 2009a). Note that θ_o = average natural undisturbed temperature of an aestifer, T = transmissivity, t = time, s = drawdown and $W(\,)$ is the well function (see Theis, 1935).

	Hydrogeology	**Thermogeology**
What are we studying?	Groundwater flow	Subsurface heat flow
Key physical law	Darcy's law $Z = -KA\dfrac{dh}{dx}$	Fourier's law (conduction only) $Q = -\lambda A\dfrac{d\theta}{dx}$
Flow	Z = groundwater flow ($m^3 s^{-1}$)	Q = conductive heat flow = ($J s^{-1}$ or W) q = heat flow per metre of borehole ($W m^{-1}$)
Property of conduction	K = hydraulic conductivity ($m s^{-1}$)	λ = thermal conductivity ($W m^{-1} K^{-1}$)
Measure of potential energy	h = groundwater head (m)	θ = temperature (°C or K)
Measure of storage	S = groundwater storage (related to porosity)	S_{VC} or S_C = specific heat capacity ($J m^{-3} K^{-1}$ or $J kg^{-1} K^{-1}$)
Exploitable unit of rock	Aquifer (Lat. *aqua*: water)	Aestifer (Lat. *aestus*: heat)
Transient radial flow	Theis equation $s = \dfrac{Z}{4\pi T} W\left(\dfrac{r^2 S}{4Tt}\right)$	Carslaw's equation $\theta_o - \theta = \dfrac{q}{4\pi\lambda} W\left(\dfrac{r^2 S_{VC}}{4\lambda t}\right)$
Tool of exploitation	Well and pump	Borehole or trench and heat pump
Measure of well/borehole efficiency	Well loss = CZ^2 where C is a constant	Borehole thermal loss = $R_b q$ where R_b = borehole thermal resistance

[2] The word *aestifer* may sound like a very artificial concoction – but it has an ancient pedigree (Banks, 2009a). Virgil (in the *Georgics, Liber II*) and Marcus Cicero (in *Aratea*) used the term *aestifer* astronomically to describe (respectively) the dog-star Sirius and the constellation Cancer as the harbingers of summer's heat. Lucretius used the word in around 60 BC in his work "De Rerum Natura" to describe the heat-bearing nature of the sun's radiation (Possanza 2001).

STUDY QUESTIONS

1.1 An aquifer is composed of sand with a hydraulic conductivity of $3 \times 10^{-4}\,m\,s^{-1}$ and is 30 m thick. It is fully saturated with water, and the groundwater head declines by 8 m every 1 km from north to south. Estimate the total groundwater flow through 1 km width of the aquifer every year.

1.2 A small, insulated core of granite, with a thermal conductivity of $3.1\,W\,m^{-1}\,K^{-1}$, a diameter of 30 mm and a length of 55 mm is placed between two metal plates. One of the plates is kept at 22°C, while the other is heated to 28°C. What is the flow of heat through the core of rock?

1.3 Think about the following sentences:

A stream of water, flowing from high topographic elevation to low elevation is able to turn a water wheel, which can perform mechanical work.

We can use mechanical energy (work) to power a pump, which can lift water from a well up to a water tower.

Try to construct analogous sentences for the concept of heat flow, rather than water flow. Take a look at Sections 4.1 and 4.2 if you get into trouble.

2

Geothermal Energy

It has pressed on my mind, that essential principles of Thermo-dynamics have been overlooked by . . . geologists.

William Thomson, Lord Kelvin (1862)

2.1 Geothermal energy and ground source heat

Let us clear up this business of 'geothermics' once and for all, because high-temperature geothermal energy will not be covered later in this book. We have already stated that, in the context of this publication, we will tend to use the terms *geothermal energy* and *geothermics* to describe the high-temperature energy that

- is derived from the heat flux from the earth's deep interior;
- one finds either in very deep boreholes or in certain specific locations in the earth's crust (or both).

However, the European Union has recently announced that shallow *ground source heat* is also classed as *geothermal energy*. The EU Renewable Energy Directive 2009/28/EC states that

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

'geothermal energy' means energy stored in the form of heat beneath the surface of solid earth.

At the risk of incurring the wrath of the European Commissioners and of riling specialists in the field of geothermics, I am going to use the terms *ground source heat* and *thermogeology* in this book to describe the low-enthalpy heat that

- occurs ubiquitously at 'normal' temperatures in the relatively shallow subsurface;
- may contain a component of genuine geothermal energy from the deep-earth heat flux, but will usually be dominated by solar energy that has been absorbed and stored in the subsurface.

Let me explain my reasoning. I have selected the term *thermogeology* because I believe that the practical utilisation of ground source heat has reached a stage where it has developed a distinct theoretical substructure. Thermogeology is a highly suitable word for this theoretical framework because it invites an analogy with the science of hydrogeology. Hydrogeology is the study of the occurrence, movement and exploitation of water in the geosphere (in other words, the study of groundwater). The comparison appears fortuitous, but once we start looking more closely, the analogy is rigorous and the two sciences enjoy pleasingly parallel theoretical frameworks (see Table 1.1).

2.2 Lord Kelvin's conducting, cooling earth

Since deep mining commenced in the sixteenth and seventeenth centuries, it was known that the earth became warmer with increasing depth, while in 1740, the first geothermometric measurements were taken by de Gensanne in a French mine (Prestwich, 1885; Dickson and Fanelli, 2004). In other words, it gradually became clear that there is a geothermal gradient. Fourier's law tells us that, if there is a geothermal gradient and if rocks have some finite ability to conduct heat, then the earth must be conducting heat from its interior to its exterior:

$$Q = \lambda A \frac{d\theta}{dz}, \text{ Fourier's law} \tag{2.1}$$

where Q = heat flow (W), A = cross-sectional area (m^2), θ = temperature (°C or K), z = depth coordinate (m) and λ = thermal conductivity (W m^{-1} K^{-1}) of rocks.

From here it is a short leap to the deduction that the earth is losing heat and cooling down. It is exactly this chain of logic that William Thomson (later Lord Kelvin – Box 2.1) used in the middle of the nineteenth century to deduce the age of the earth, staking his claim to be the world's first thermogeologist (Thomson, 1864, 1868). At that time, Thomson and many other geologists suspected that the earth had been born

BOX 2.1 William Thomson, Lord Kelvin

William Thomson was born in Belfast, Ireland, in 1824 to James Thomson, an engineering professor (O'Connor and Robertson, 2003). The family subsequently moved to Scotland after James was appointed professor of mathematics at Glasgow University. The young William started as a student at the same university at the tender age of 10; while by his mid-teens, he was writing essays on the earth and studying Joseph Fourier's theories of heat. After further study and research in Cambridge and Paris, he was appointed professor of natural philosophy at Glasgow in 1846.

Thomson is probably best remembered for the hugely important theoretical underpinning that he provided for the science of thermodynamics. Among other things, he proposed (in 1848) the absolute temperature scale (the Kelvin scale) and developed the principle of the equivalence between mechanical work, energy and heat. He seems to have been the first person to propose the notion and lay the theoretical foundations for the heat pump, where mechanical energy is used to transfer heat from a low-temperature environment to a high-temperature one.

Later in his career, Thomson was created Baron Kelvin of Largs by the British Government. This title provides us with the name of the SI unit of temperature, possibly the only SI unit to be ultimately named after a Glaswegian river!

Thomson can also lay claim to being the world's first thermogeologist and was able to estimate the age of the earth from the earth's heat flux. His estimate of around 100 million years was wrong (due to the lack, at that time, of any concept of heat generation by radioactive isotopes in the earth), but his techniques were fundamentally correct.

Kelvin died in his eighties in 1907 near Largs in Scotland. It is time that his reputation is rehabilitated and that he is recognised not only as one of the fathers of the heat pump but also as the founder of the science of thermogeology.

as a globe of molten rock and had subsequently cooled. They believed that the observed geothermal gradient was due to the residual heat in the earth's interior gradually leaking away into space through a solid lithosphere. As heat was lost, the thickness of the lithosphere increased at the expense of the molten interior. Thomson combined Fourier's law with the one-dimensional equation for heat diffusion by conduction (which we will meet again in Chapter 3):

$$\frac{\partial^2\theta}{\partial z^2} = \frac{S_{VC}}{\lambda}\frac{\partial\theta}{\partial t} \tag{2.2}$$

where z = depth below the earth's surface (m), $\partial\theta/\partial z$ = geothermal gradient, S_{VC} = specific volumetric heat capacity of rocks ($\mathrm{J\,m^{-3}\,K^{-1}}$) and t = time (s).

Thomson tried to work out the age at which the earth's crust had formed, by making assumptions about the initial temperature of the earth's interior (around 7000°F hotter than the current surface temperature, or somewhat over 4100 K; Thomson, 1864; Ingersoll *et al.*, 1954; Lienhard and Lienhard, 2006) and a reasonable estimate of the thermal diffusivity of rocks. In 1862, he was able to conclude that the age of the earth, that is, the time it would take to cool down to the current observed temperature and geothermal gradient, was somewhere between 20 and 400 million years (he later homed in on 100 million years, and eventually settled on an age at the younger end of the range; Lewis, 2000).

We will not worry about the mathematics here, but combining Equations 2.1 and 2.2, followed by integration, yields the following expression (Ingersoll *et al.*, 1954; Clark, 2006):

$$t_{\text{earth}} = \frac{S_{\text{VC}}.(\theta_{\text{i}} - \theta_{\text{s}})^2}{\lambda \cdot \pi \cdot \psi^2} \tag{2.3}$$

where ψ is the current geothermal gradient near the surface, t_{earth} is the age of the earth's crust (in seconds), θ_s is the surface temperature and θ_i is the temperature of the earth's molten interior. You can try the calculation yourself by choosing 'guess-timates' of input values (see Table 3.1 for typical values of λ and S_{VC}, and try a value of $0.02\,\text{K}\,\text{m}^{-1}$ for ψ).

Thomson's estimate placed him at odds with conservative Christians, who accepted a young earth, based on tendentious genealogical calculations from the Bible. It also won him little popularity with some contemporary geologists and biologists, who thought that the lower of Thomson's age estimates was rather short to account for the observed stratigraphy of the earth and the evolution of life. Thomson's estimate ultimately proved to be a gross underestimate: we currently reckon the earth to be around 4.5 billion years old. As a result, Thomson is sometimes ridiculed by modern geologists. But his calculations were fundamentally correct, given the knowledge and conceptual model he had at the time. We now know that the earth's interior is kept hot by the continuous decay of radionuclides, chiefly isotopes of uranium (^{238}U and ^{235}U), potassium (^{40}K) and thorium (^{232}Th); hence, it cools far slower than Thomson's prediction. But Thomson could not know this: radioactivity was only discovered by Antoine Henri Becquerel in 1896, and radioactive elements were only isolated by Marie and Pierre Curie in 1902.

2.3 Geothermal gradient, heat flux and the structure of the earth

Thomson assumed, not unreasonably, that the transport of heat through the earth's lithosphere was dominated by conduction, and that it was spatially homogeneous. In other words, he assumed that the geothermal gradient and heat flux were uniform over the earth's surface. In the later part of the nineteenth century, workers such as

Joseph Prestwich and J.D. Everett (and other colleagues, including my thermogeological predecessor at the University of Newcastle, George A.L. Lebour) focussed on quantifying the magnitude of the gradient from measurements in English coal mines, Cornish tin mines and drilled boreholes (Everett *et al.*, 1876, 1877, 1880, 1882; Everett, 1882; Prestwich, 1886). Indeed, Prestwich (1885) deduced a value of 0.037°C m^{-1} from determinations in coal mines and a value of 0.042°C m^{-1} from the Cornish mines. In fact, we now know that geothermal gradient varies considerably between different locations, although typical values are in the range of 2–3.5°C per 100 m (0.02–0.035 K m^{-1}). The typical geothermal heat flux is of the order 60–100 mW m^{-2}, with a global average estimated at 87 mW m^{-2} (Pollack *et al.*, 1993; Dickson and Fanelli, 2004). Using Fourier's law (see above), we can try these values for size to derive a typical thermal conductivity of the earth's subsurface of

$$\lambda = 0.087\ \mathrm{W\ m^{-2}} / 0.0275\ \mathrm{K\ m^{-1}} = 3.2\ \mathrm{W\ m^{-1}\ K^{-1}}$$

It has also become clear that the earth has a somewhat more complicated internal structure than Kelvin's conceptual model presupposed. In terms of geochemistry and mineralogy, the earth's structure can be considered to comprise (Figure 2.1)

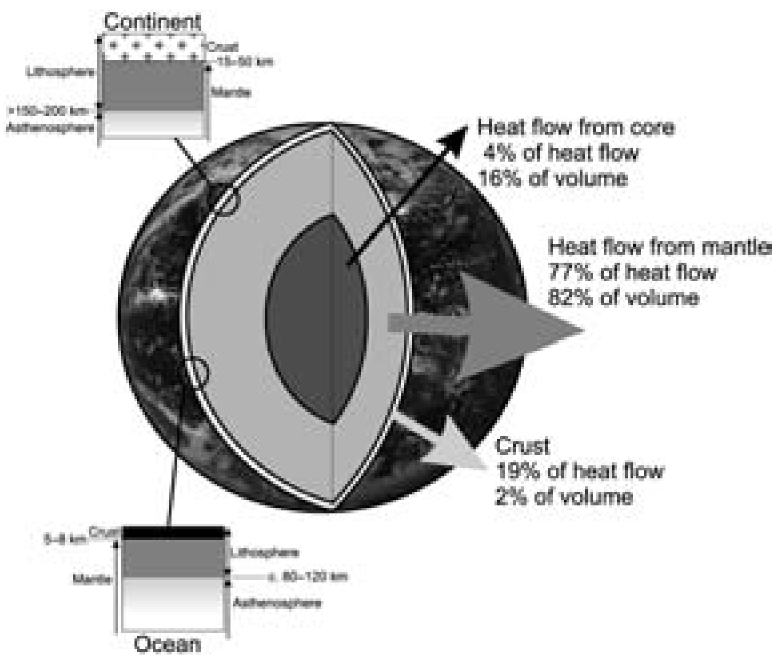


Figure 2.1 Schematic diagram of the structure of the earth, showing the percentage of geogenic heat derived from the core, mantle and crust (numbers adjacent to arrows), compared with the volume of these three portions of the earth. The figure also shows the typical structure of the lithosphere in continental and oceanic plates.

1. a solid inner core of metallic iron–nickel, of radius 1370 km;
2. a molten outer core of iron–nickel, of thickness 2100 km;
3. a mantle of ultrabasic Fe- and Mg-rich composition, of thickness 2900 km. The uppermost part of the mantle is, for example, dominantly composed of the minerals olivine and pyroxene, the constituents of a rock called peridotite;
4. a very thin crust (the boundary between crust and mantle is called the *moho*).

The oceanic crust is wholly different from the crust beneath the continents. The former is very thin (only some 5–8 km) and is predominantly composed of basic minerals and rocks (e.g. gabbro, dolerite and basalt). Continental crust is somewhat thicker (15–50 km, and even more under mountain belts; Smith, 1981) and less dense than oceanic crust. It is geochemically more acidic and 'sialic' (i.e. rich in *si*licon and *al*uminium) and contains minerals such as quartz and feldspar, which are the familiar constituents of the granites, gneisses and sedimentary rocks that we encounter during our land-based geology field trips.

The earth's radius is about 6370 km. Thus, its circumference is around 40 000 km and its surface area is around 510 million km^2 (5×10^{14} m^2). As the average geothermal heat flux from the earth is known, it can be estimated that the total heat flow from the earth is around 44 TW (Dickson and Fanelli, 2004). The proportions of geogenic heat flux from the core, mantle and crust of the earth are shown in Figure 2.1. The heat derived from the core is small, relative to the core's volume. This is due to the core being composed largely of metallic iron and nickel, impoverished in the heat-generating radioactive nuclides. The crust (and especially the continental crust) is responsible for more than its volumetrically 'fair' share of heat production, at around 19%, being relatively rich in radioactive uranium, potassium and thorium minerals.

2.4 Internal heat generation in the crust

Wheildon and Rollin (1986) pointed out that the earth's geothermal gradient changes with depth, due to radiogenic heat generation within the crust itself. If Å is the heat production per unit volume of the earth's crust (W m^{-3}) and q is the geothermal heat flux, then

$$\frac{\partial q}{\partial z} = \frac{\partial}{\partial z}\left[\lambda \frac{\partial \theta}{\partial z}\right] = -\text{Å} - S_{VCf} v_D \frac{\partial \theta}{\partial z} + S_{VC} \frac{\partial \theta}{\partial t} \tag{2.4}$$

The second term on the right-hand side relates to heat transport by convection, where S_{VCf} is the volumetric heat capacity of the convecting fluid (e.g. groundwater, gas or magma) and v_D is its vertical fluid flux rate (positive upwards). The third term on the right-hand side represents the change in heat stored in the rock with time, where S_{VC} relates to the volumetric heat capacity of the rock. Thus, if convectional heat transfer is negligible and if we consider a steady-state situation:

$$\frac{\partial q}{\partial z} = \frac{\partial}{\partial z}\left[\lambda \frac{\partial \theta}{\partial z}\right] = -\text{Å} \tag{2.5}$$

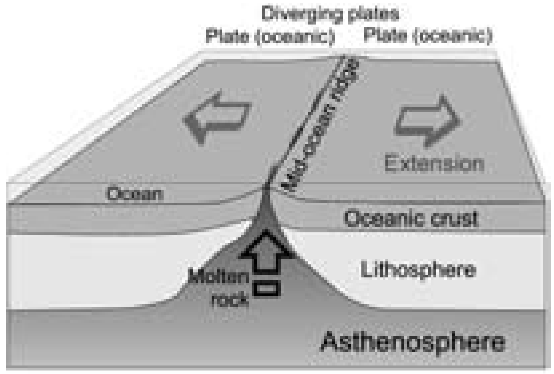
We have already encountered Fourier's law (Equation 2.1), which relates heat flow to geothermal gradient, with the caveat that the geothermal gradient may vary with depth due to internal production of heat. We now also have Equation 2.5, which is a form of Poisson's equation that relates the change in average geothermal gradient to internal heat production. This sounds temptingly simple, but we should also remember that internal heat production will be depth dependent! In fact, Å will typically decrease with increasing depth, as the crust becomes less sialic in nature and with a lesser content of radioactive minerals.

Equation 2.4 predicts that we would expect the highest geothermal heat fluxes in regions with high radiogenic heat production in the upper crust or with strong upward convection of hot fluids from depth. For example, if we consider Figure 3.10 (showing the geothermal heat flux in the United Kingdom), the highest heat fluxes are from areas underlain by granites (south-west England and Weardale). In the granitic terrain of Devon and Cornwall (south-west England), internal radiogenic heat production (Å) reaches $5\,\mu\text{W}\,\text{m}^{-3}$, while heat fluxes ($q$) in excess of $100\,\text{mW}\,\text{m}^{-2}$ are observed. Other anomalies, such as those in central England (around the Peak District), are more likely to be the result of deep convection of groundwater (see Brassington, 2007). According to Busby *et al.* (2011), the average geothermal gradient below onshore Britain is $0.028\,\text{K}\,\text{m}^{-1}$.

2.5 The convecting earth?

While Kelvin's conceptual model involved a static conducting globe, we now know (thanks to the plate tectonic paradigm shift of the 1960s) that the earth is not a rigid sphere. Over long periods of geological time and at the temperature and pressure conditions prevailing in the earth's mantle, we can envisage rocks behaving more like fluids than solids. It is widely believed by many geologists that the earth's mantle, at some scale, is subject to convection processes. Put very simply, just as a saucepan full of milk, heated from below, will begin to form roiling convection cells, the earth's interior is in constant, slow fluid motion.

We can think of the earth's tectonic plates as a kind of stiff, low-density 'scum' (or *lithosphere*) of rock floating on a deeper, fluidly deforming *asthenosphere*. The boundary between the lithosphere and asthenosphere does not coincide with the crust/mantle boundary (or moho). Rather, the lithosphere comprises the crust and a rigid portion of the underlying mantle, while the asthenosphere lies wholly in the mantle (Figure 2.1). Below the oceanic crust, the lithosphere may be around 80–120 km thick [although at mid-oceanic ridges (Figure 2.2), it may only be several kilometres thick]. Below continents, the lithosphere is believed to be considerably thicker, exceeding 200 km in places.



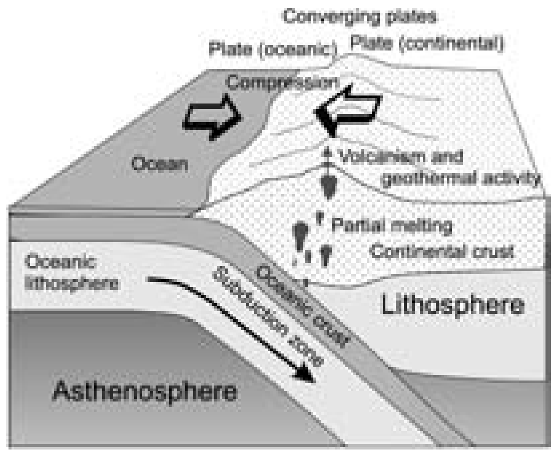


Figure 2.2 A simplified cross section of the earth's lithosphere showing both divergent (top) and convergent (bottom) plate margins. The rising, partially molten asthenosphere and the thinning of both the crust and the lithosphere at oceanic ridges result in a strongly elevated geothermal gradient and volcanic activity. At subduction zones, the presence of water in the descending oceanic lithospheric slab, coupled with prevailing temperature and pressure conditions, gives rise to partial melting along the slab. This creates bodies of magma that rise through the overlying lithosphere and eventually give rise to localised volcanism and geothermal fields in the island groups or mountain ranges located above the subduction zone.

It is widely believed that the motion of the lithosphere's tectonic plates is in some way coupled to convection cells within the mantle/asthenosphere. Tectonic plates move away from each other at mid-ocean ridges, where the lithosphere is thin and the asthenosphere rises and diverges. At subduction zones and compressive plate margins, chunks of lithosphere override each other (Figure 2.2). In fairness, most geologists agree that there are a number of 'driving forces' behind the motion of tectonic plates, including gravitational forces acting on descending slabs of lithosphere at subduction zones. Furthermore, it is also recognised that parts of the lower crust

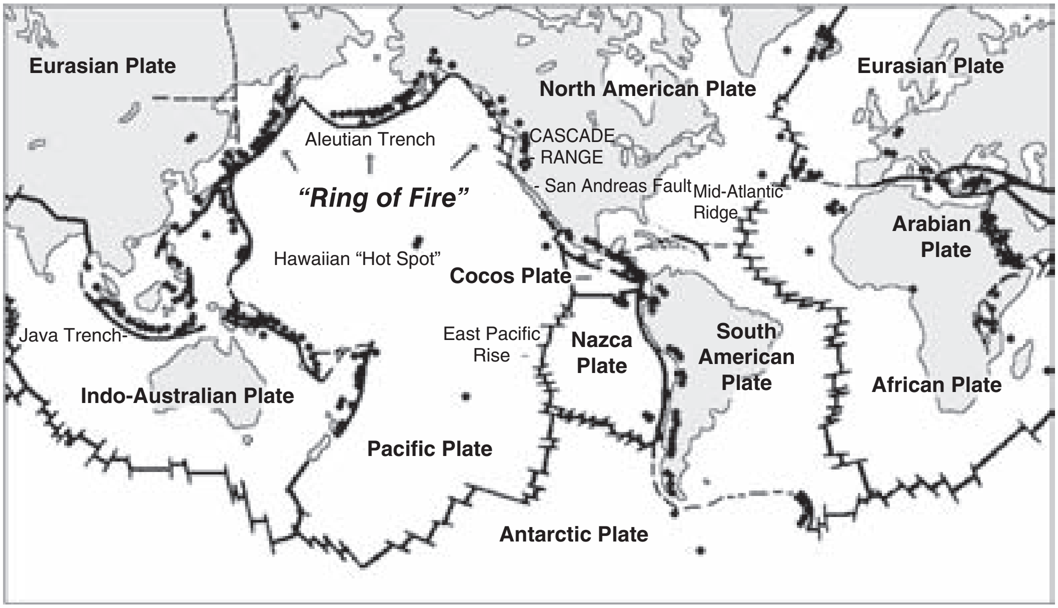


Figure 2.3 Simplified plate tectonic map of the world, showing locations of active volcanoes as dots. These tend to fall along plate boundaries. *Public domain material produced by the United States Geological Survey (USGS)/Cascades Volcano Observatory and accessed from* http://vulcan.wr.usgs.gov.

also undergo significant fluid deformation on large scales (Westaway *et al.*, 2002). Thus, mantle convection is, at best, only part of a complex picture.

Far from being a uniform, gently cooling globe, the earth is a heterogeneous (at least in its upper portions), convecting sphere. The outer shell of the earth is composed of materials of varying thermal properties and is in slow, constant motion. Volcanic and seismic activities are concentrated along tectonic plate margins (Figure 2.3). Moreover, the geothermal heat flux at these margins can average 300 mW m^{-2} (Boyle, 2004), and it should be no surprise that the earth's major geothermal resources are also concentrated along these zones.

2.6 Geothermal anomalies

In most locations on earth, direct use of true geothermal energy is not an especially attractive option. With a geothermal gradient of 0.025°C m^{-1}, we would need to drill 1.4 km to reach a temperature of 45°C (which can be regarded as necessary for low-temperature space heating). Alternatively, we could look at things another way: to utilise sustainably the earth's geothermal heat flux to heat a small house, with a peak heat demand of 10 kW, we would need to capture the entire flux (say, 87 mW m^{-2}) over an area of 115 000 m^2 (11.5 ha). Both a 1.4-km-deep hole and an 11.5-ha heat-capture field per house are rather unrealistic propositions for the average householder!

Fortunately, the earth's geothermal heat flux and temperature gradient are not uniformly distributed, and there do exist anomalous areas of the earth's surface where the heat flux is much larger than average and/or we encounter high temperatures at shallow depth. We can call these anomalies potential geothermal fields, and they can be due to a variety of geological factors.

High-temperature geothermal fields are usually related to plate tectonic features (see Figure 2.3). They typically occur at one of three tectonic locations and are often associated with current or historic volcanism:

1. Extensional plate margins: typically mid-oceanic ridges (e.g. Iceland and the Azores), or proto-rifts such as the Great Rift Valley of central and eastern Africa, and the Rhine Graben. At such extensional boundaries, the crust and lithosphere are rather thin and are being 'ripped' apart. The geothermal gradient is very high and the asthenosphere may occur at depths of only a few kilometres. Geochemically basic magma intrudes into the extensional cracks and fissures related to rifting and may overspill at the surface as volcanoes. The geothermal fields around Iceland's capital, Reykjavík, are examples of systems drawing their energy from the presence of magma at shallow depth in an extensional rifting regime (Franzson *et al.*, 1997).
2. Convergent plate margins: the presence of water in a subducting slab of oceanic crust (Figure 2.3), coupled with the particular pressure and temperature conditions at depth, can lead to partial melting along the slab. This generates bodies of magma that rise slowly through the overthrust lithospheric slab. If these 'diapirs' of magma reach the surface, the water-rich molten rock can explode as a violent volcanic eruption, such as that of Krakatoa in 1883. The accumulated volumes of magmatic material, coupled with the tectonic forces associated with subduction, usually give rise to linear mountain belts (such as the Andes) or island arcs (such as the Aleutians or Japan) above the plate margin. These margins are usually associated with geothermal or volcanic loci, such as those in the Mediterranean region (Box 2.2).
3. Below some tectonic plates, localised plumes of warm material rise from the deep mantle at 'hot spots' seemingly unrelated to the broader tectonic picture. The mechanisms of these 'mantle plumes' are still poorly understood, but these volcanic and geothermal loci can persist for geologically extended periods. The Hawaiian island chain was formed by successive volcanic eruption centres as the Pacific Plate drifted slowly across the location of a mantle plume. The Yellowstone 'supervolcano' and associated geothermal field is another example of 'plume' activity.

Away from these specific tectonic settings, more modest geothermal anomalies (either positive or negative) are related to the earth's dynamic behaviour over geological timescales, to heterogeneities within the crust, or to the effects of fluid flow in transporting heat from one location to another. For example, they may be related to the following:

BOX 2.2 Larderello – the History of Geothermal Energy in a Nutshell

The Larderello site is situated in southern Tuscany, Italy, and is usually regarded as the great-granddaddy of geothermal energy schemes. The area is renowned for its phreatic volcanic activity, that is, periodic eruptions of steam. The last such major eruption was from Lago (Lake) Vecchienna in 1282 AD, when ash and blocks of rock were also disgorged. The geothermal activity is believed to be related to the presence of a cooling body (or pluton) of granite at relatively shallow depth beneath a cover of metamorphic and sedimentary rocks (GVP, 2006).

Because they are usually deeply derived (a long time for hydro-chemical evolution) and because they are warm (enhanced chemical kinetics), geothermal waters often have high and unusual mineral contents (Albu *et al.*, 1997). The hot waters at Larderello were historically renowned for their mineral content: the Romans used the sulphur-rich waters for bathing. Later, the waters were extracted from shallow boreholes and used to produce the element boron, which they contained in abundance. The site was not known as Larderello at that time, but as Montecerboli: it was renamed after the Frenchman François de Larderel, who in 1827 first used the geothermal steam to assist in extracting boron from cauldrons of 'volcanic' mud. This drew attention to the locality's thermal, as well as hydrochemical, potential. Shortly afterwards, geothermal steam was being exploited to perform mechanical work at the boron works. In 1904, it was first used to attempt to generate electricity, followed by the construction of a power plant in 1911. The site remained the world's sole geothermal electricity producer until 1958, when New Zealand opened its first plant. It was not until 1910–1940 that the geothermal heat from the site was actually used for space heating (Dickson and Fanelli, 2004). The development of this geothermal site provides an interesting perspective on historical priorities: first mineral production, then mechanical work and finally space heating. It should serve as a reminder that it is relatively recently that the era of consequence-free, dirt-cheap fossil fuel has drawn to a close. It is only now that we are beginning to prioritise the need for sustainable, affordable, low-carbon space heating.

4. Variations in thermal conductivity of rocks. Assuming that we have a constant flux of heat from the earth's interior, Fourier's law implies that, in order to conduct this constant flux, a layer of rock with a low thermal conductivity must possess a high geothermal temperature gradient (Figure 2.4). We thus expect to find anomalously high temperatures beneath thick layers of rock with low thermal conductivity. The low-temperature Paris (Boyle, 2004) and Southampton (Box 2.3) geothermal systems are examples of reservoirs with an anomalously high temperature due to an overlying blanket of low-conductivity mudstones or limestones.

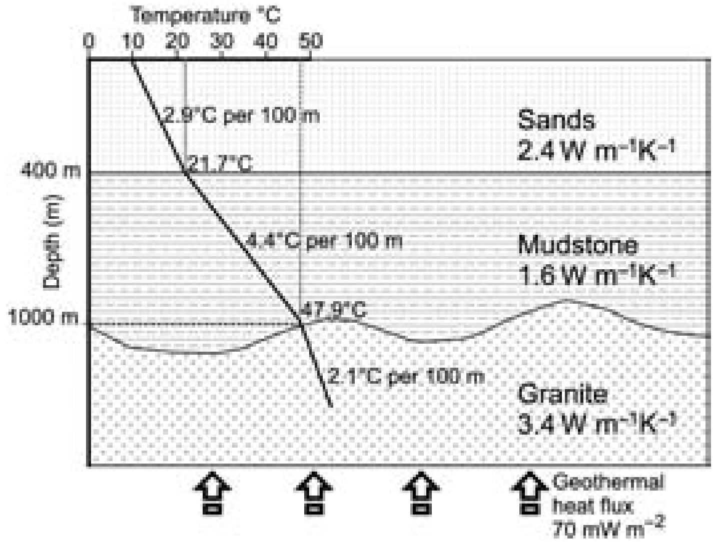


Figure 2.4 Schematic cross section through a three-layer 'sandwich' of different rock types. In order to maintain a constant geothermal heat flux of, say, 70 mW m^{-2}, the geothermal gradient in the lower conductivity mudstone layer must be higher than in the sand layer. Therefore, temperatures at the top of the granite are higher than they otherwise would have been, given the initial geothermal gradient in the top (sand) layer.

BOX 2.3 The Southampton Geothermal System

A quick glance at the map of Figure 3.10 reveals that the city of Southampton, on the English south coast, is not associated with any especially high geothermal heat flux. The fact that it is England's first and most famous functional geothermal heating system demonstrates that one does not necessarily require extraordinary geological conditions – one merely needs

- a thick sedimentary sequence of relatively low thermal conductivity (and thus high geothermal gradient), such that elevated temperatures are found at modest depth;
- an aquifer horizon at that depth.

Exploratory drilling at Southampton commenced in the early 1980s, with the first 1.8-km-deep production well being commissioned in 1987. Southampton lies in a Tertiary synclinal structure and the borehole penetrated Tertiary and Mesozoic clays, sandstones and limestones before encountering, at a depth of some 1730 m, the geothermal reservoir rock – the Triassic Sherwood Sandstone (Smith, 2000). The Sherwood Sandstone is probably the United Kingdom's most impor-

BOX 2.3 (*Continued*)

tant geothermal aquifer, with potentially exploitable deep basins in Wessex, Humberside, Worcestershire, Cheshire/Merseyside and around Larne in Northern Ireland. While many of the United Kingdom's other major aquifers are limestones (the Chalk, the Jurassic Limestones), whose fractures close up with increasing pressure and depth, the Sherwood Sandstone is relatively thick and has an intergranular porosity that can be held hydraulically open at great depth by its matrix of silica sand grains.

Southampton lies on the very edge of the Wessex basin (Downing and Gray, 1986; Holloway *et al.*, 1989) and the Sherwood Sandstone at this location has very marginal hydrogeological properties. Only the top 38 m or so of the 67-m-thick Sherwood Sandstone was found to contain water-bearing horizons, with a pumping test indicating a total transmissivity of around 4 Dm (around 6–8 m^2 day^{-1}). The Sherwood Sandstone was found to contain sodium chloride brine at 76°C, with a salinity of some 125 $g\,L^{-1}$ and a static water level around 100 m below ground level (Allen *et al.*, 1983; Allen and Holloway, 1984). Testing at a pumping rate of 19.8 $L\,s^{-1}$ caused a drawdown of around 3 $MN\,m^{-2}$ (c. 300 m).

In the operational scheme, brine is pumped from the well to the surface (the design yield was 10–15 $L\,s^{-1}$), where it passes through a heat exchanger, with associated heat pumps. Heat is thus transferred to a carrier fluid (water), which provides heat, via a network of insulated mains, to a number of properties in Southampton city centre, including homes, hotels, a college campus, a store, a stadium and numerous offices. The cooled brine (around 28°C) is discharged to the estuary of the River Test. The heat yield of the geothermal source was originally 1 MW_{th} by direct heat exchange to the carrier fluid, although this has now been increased, by the use of heat pumps, to nearer 2 MW_{th} (Barker *et al.*, 2000; Boyle, 2004).

In fact, the geothermal source is now integrated with a combined heat and power plant (CHP), including a high-efficiency 5.7 MW_e generator, supplying the district heating and cooling scheme. The CHP provides heat to the circulating carrier fluid of the district heating system, as well as electricity. The total heating capacity of the scheme is around 12 MW_{th}, and conventional fossil fuel boilers can be called on at times of peak loading.

Absorption heat pumps use the CHP's surplus heat to produce chilled water, which is circulated through a separate insulated network of pipes in a *district cooling system*, supplying customers' air conditioning needs. Cooling needs have increased dramatically since the system's conception to such a level that it is planned, during summer months, to use night-time electricity to produce ice by means of heat pumps. This ice will then be used to provide chilled water during daytime – a simple but elegant means of storing 'coolth' produced at times of surplus electrical capacity (Energie-Cités, 2001).

(*Continued*)

BOX 2.3 (*Continued*)

While the geothermal borehole now only provides around 10–20% of the total peak heating load supplied by the integrated scheme, the scheme's total impact is impressive, with an annual carbon dioxide emission saving of at least 10 000 tonnes, compared with conventional technologies (Energie-Cités, 2001; IEADHC, 2004; EST, 2005).

Recall that the Southampton scheme is developed in a hydrogeologically marginal part of the Wessex Sherwood Sandstone basin: The Sandstone is far thicker further to the west (Bournmouth and Dorchester) and is a much more promising geothermal reservoir, which has yet to be exploited.

5. The fact that some rock bodies have internal heat production (e.g. radioactive decay of uranium and potassium in granites, or chemical oxidation of sulphides in mine waste). An excellent target rock for hot dry rock geothermal systems (Section 2.12) is thus a granite with a high internal heat production, overlain by a thickness of low-conductivity sediments, thus ensuring a high geothermal gradient and high temperature at relatively shallow depth (Box 2.4).
6. Groundwater flow can transport heat rapidly by advection from one location to another (Figure 2.5). Geothermal anomalies may thus occur where faulting allows deep warm groundwater to flow up towards the surface, carrying a cargo of heat (geothermal short-circuiting). The British hot springs at Bath, Buxton and Matlock are all related to faulting that allows deep groundwater from Carboniferous limestone strata to flow to the surface (Banks, 1997; Brassington, 2007).
7. Geothermal anomalies can also occur due to the fact that earth (and its climate) is a dynamic system. The subsidence of thick sedimentary basins can 'rapidly' carry cold sediments downwards. Conversely, isostatic rebound (e.g. following the last glaciation) can raise the elevation of rocks at a rate of several centimetres per year. Furthermore, climatic cooling during the Pleistocene glaciation in the United Kingdom and northern Europe is believed to have depressed the temperature of sediments and rocks down to at least 300 m depth (Wheildon and Rollin, 1986; Šafanda and Rajver, 2001). In the slight twitches and anomalies of geothermal gradients measured in boreholes, specialists can make deductions about the past climate and about rates of crustal uplift and subsidence. For deep geothermal exploration, we must also be aware of the potential violation of one of the assumptions behind Equation 2.5: that the geological environment is in a steady state. In fact, in northern Europe, it is not – it is still recovering (thermally and isostatically) from the Pleistocene glaciations. Wheildon and Rollin (1986) suggested that ignoring this perspective may cause us to significantly underestimate our geothermal heat flows.

BOX 2.4 The Weardale Exploration Borehole

Figure 3.10 (p. 63) shows that the region of Britain with the highest geothermal heat flux is that underlain by the Hercynian granite batholiths of Devon and Cornwall, which have an internal radioactive heat production of up to $5\,\mu W\,m^{-3}$. The next most prominent feature stretches west of Sunderland from Weardale to the Lake District and does not clearly correspond with any surface geological outcrop. The anomaly is, however, known to be underlain by Caledonian (lower Devonian) granites with an estimated radiogenic heat production of $3.3–5.2\,\mu W\,m^{-3}$ (Wheildon and Rollin, 1986). The granites are overlain by a thickness of several hundred metres of Carboniferous sedimentary rocks of low thermal conductivity. We have seen (Section 2.6) that we should expect high temperatures at shallow depths where such 'hot' radioactive rocks are overlain by an 'insulating' sedimentary cover. Indeed, a team at Newcastle University found that thermochemical signatures of waters in flooded mines in the region provided tentative indications of elevated temperatures (Younger, 2000).

In 2003, a proposal was made to revive a disused cement works site at Eastgate in Weardale as a 'renewable energy village'. Hydrochemical evidence from nearby fluorite mines suggested a temperature anomaly associated with the Slitt Vein – a mineralised fault zone in the Carboniferous sedimentary rocks presumed to pass beneath Eastgate (Manning *et al.*, 2007). In summer 2004, five 50- to 60-m inclined boreholes were drilled to locate the Slitt Vein at Eastgate beneath a cover of Quaternary superficial deposits.

In August 2004, a deep exploration borehole commenced above the subcrop of the Slitt Vein, at a diameter that would allow it to be commissioned as a production well, should thermal water be encountered. Drilling proceeded through the Lower Carboniferous sedimentary rocks (including 67 m of the dolerite Whin Sill), until the granite was encountered at 272 m depth. Drilling proceeded into the granite until, at 410 m, a major fracture was encountered (the drilling bit appeared to drop through a void of some 50-cm aperture). Water entered the borehole from this fracture and its level eventually stabilised around 10 m below the ground level. The potential short-term yield of this horizon exceeded $16\,L\,s^{-1}$, comprising a hypersaline sodium–calcium–chloride brine (Paul Younger, *pers. comm.*; Manning *et al.*, 2007).

Drilling continued and, despite high bit attrition and corrosion rates, eventually terminated at 995 m. A bottom-hole temperature of 46°C was measured (compared with a predicted temperature of around 29°C assuming an average geothermal gradient of $0.02\,°C\,m^{-1}$). The water yielded by the borehole as a whole is, of course, dominated by the major fracture at 410 m, and hence has a lower temperature of around 27°C – hot enough to support a saline *kurbad* (should that

(*Continued*)

BOX 2.4 (*Continued*)

be desirable), but inadequate for direct heating purposes. Alternative options for achieving a higher temperature include the following:

- Hydraulic fracturing of the lower sections of the borehole to allow groundwater flow in the deepest, hottest part of the granite – a type of 'enhanced geothermal system' (see Section 2.12).
- Operating the Eastgate borehole as a form of 'standing column well' system (see Chapter 13). By doing this, one is sacrificing advectional heat transfer for conductive heat transfer and, while it may be possible to achieve a higher temperature, it will be at the expense of flow volume and thus of overall heat production rate.
- Utilising heat pumps to raise the temperature of 27°C to a useful space-heating value, with a high degree of efficiency.

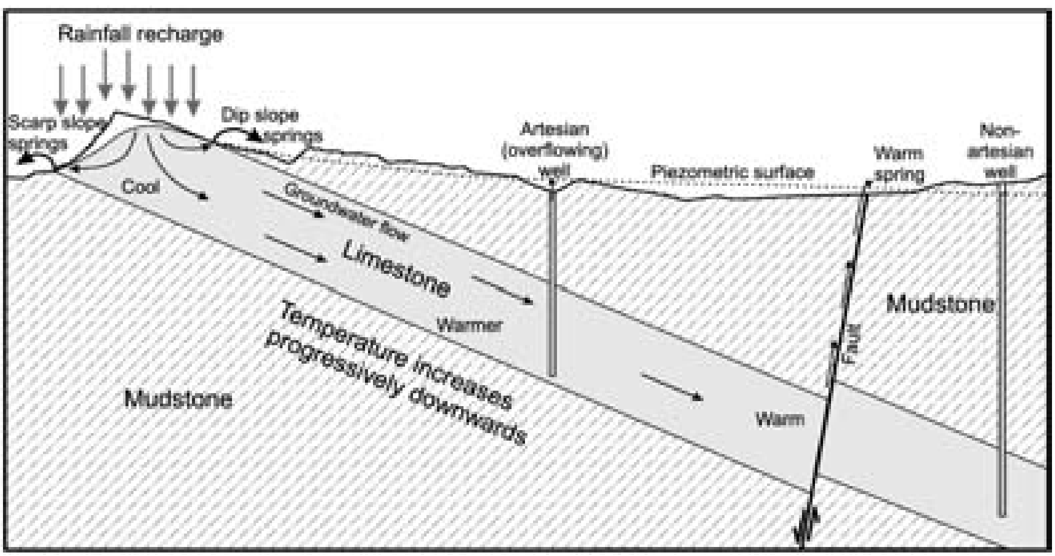


Figure 2.5 Schematic cross section through a groundwater system. Recharge falling on the limestone aquifer outcrop slowly flows down-dip, equilibrating with progressively higher temperatures with increasing depth. Small quantities of water are able to exit the aquifer system via a 'short-circuiting' fault. The ascent along a high permeability fault may be so rapid that the water does not substantially cool during its re-ascent, emerging as a warm spring. The grey shaded strata are water-saturated limestone.

2.7 Types of geothermal system

Geothermal energy systems can be classified into low-, intermediate- and high-enthalpy systems (Figure 2.6): here, the term 'enthalpy' is closely related to the temperature of the system. Various authors disagree about the boundaries between these classifications and they are, frankly, of little practical value. It is possibly better to classify geothermal systems based on their potential for use or on the characteristics of the fluids they produce (Dickson and Fanelli, 2004).

2.7.1 Water- and vapour-dominated geothermal systems

The fluid produced by wells drilled into water-dominated systems is mostly liquid water as the pressure-controlling phase, with some steam present, for example, as bubbles. The temperatures of these systems may be well above 100°C – remember that water only boils at 100°C at 1 atm pressure. In subsurface pressure conditions, water can exist as a liquid at much higher temperatures, only boiling ('flashing') when pressure is released during transit to or on arrival at the surface. Thus, water-dominated systems may produce hot water, mixtures of water and steam, wet steam or even dry steam (see Figure 2.7).

Vapour-dominated systems are rarer. Here water and steam may be present in the reservoir rocks, but steam will be the pressure-controlling phase. These systems typically produce dry or superheated steam.

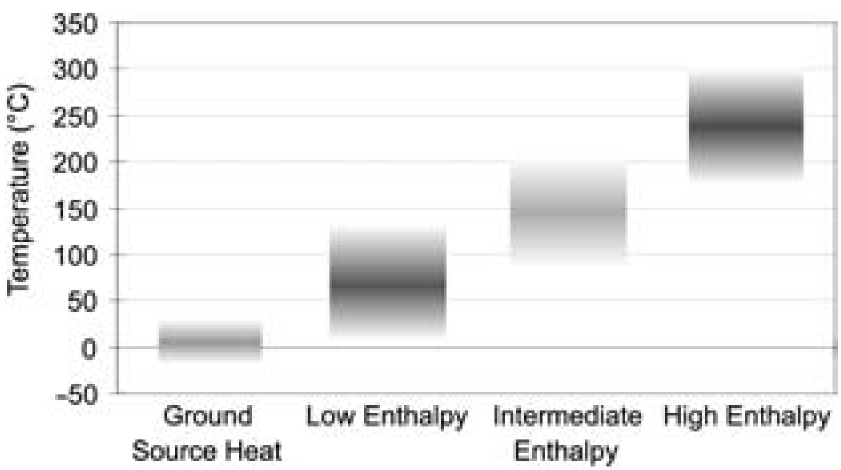


Figure 2.6 Classification of geothermal systems according to temperature (based on suggestions made by Dickson and Fanelli, 2004).

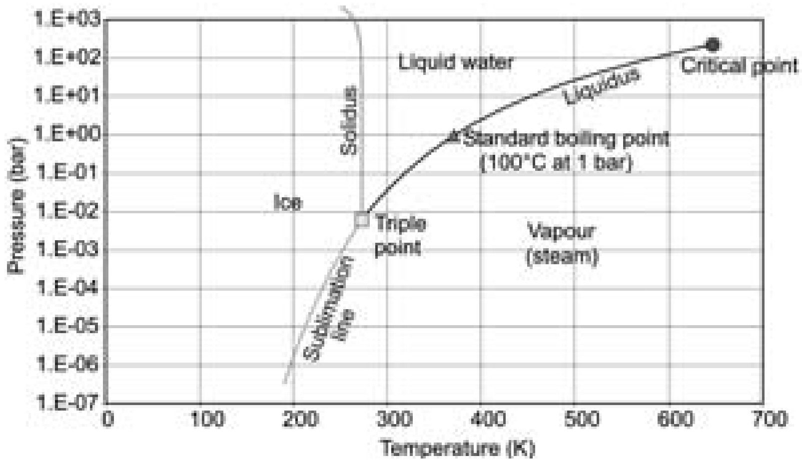


Figure 2.7 A simplified phase diagram for water, showing the phases present at various pressure and temperature conditions. The liquidus is the line dividing the liquid and steam fields; the solidus is the line dividing the ice and liquid fields. Wet steam is steam that coexists with water at the temperature of volatilisation for a given pressure (i.e. a mixture of steam and water on the liquidus); dry steam is steam without any water phase present, but still lying on the liquidus; superheated steam is steam whose temperature is above the temperature of volatilisation for a given pressure (i.e. it plots below the liquidus).

2.8 Use of geothermal energy to produce electricity by steam turbines

The use of steam turbines to directly generate electricity (Figure 2.8) typically requires geothermal systems producing fluids at temperatures over 150°C. In systems producing mixtures of water and steam or wet steam, a separator may be required to remove the water component from the steam. The efficiency of the turbine can be improved by using a condensing unit to condense the steam following passage through the turbine. If a condenser is used, it needs to be cooled via some form of heat exchanger. This will typically generate a flow of hot, secondary (coolant) fluid that can be used for space-heating purposes.

2.9 Binary systems

At lower temperatures, it is still possible to generate electricity from geothermal fluids. Here, however, we may not be able to use steam directly as the working fluid. We must use a heat exchanger to transfer heat energy to a secondary fluid that has a lower temperature of vaporisation (boiling point), such as n-pentane or butane (Boyle, 2004). It is this secondary fluid, which, having volatilised, performs mechanical work

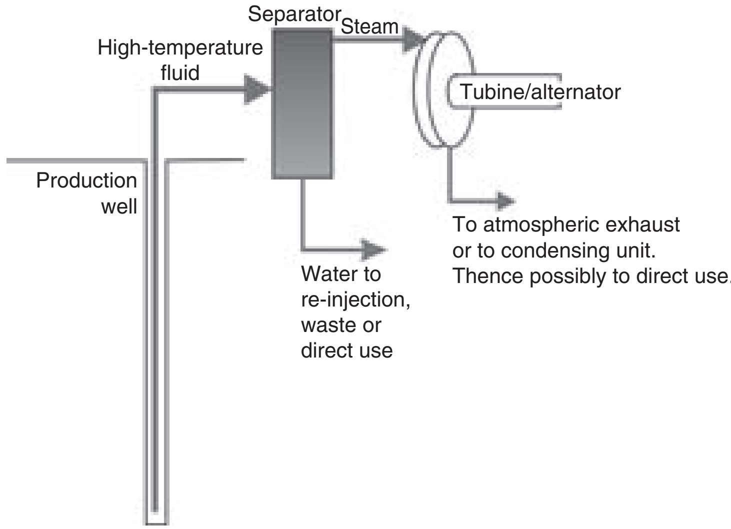


Figure 2.8 The use of a high-temperature geothermal fluid to power a steam turbine.

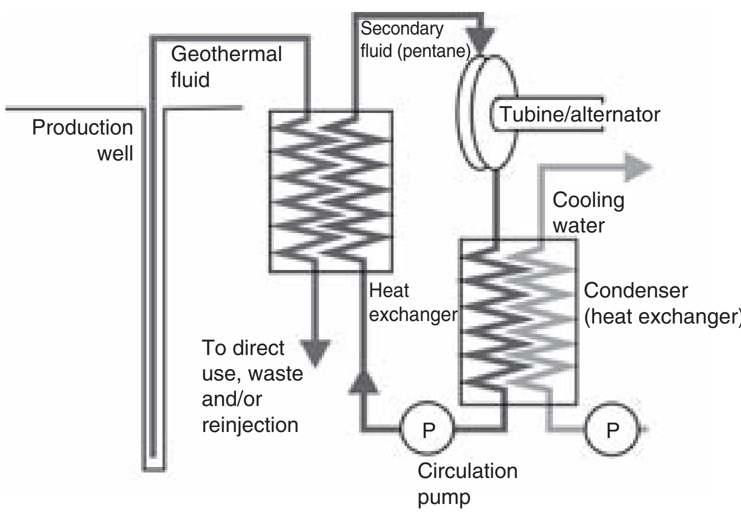


Figure 2.9 The use of an intermediate-temperature geothermal fluid to power a binary system.

in rotating a generating turbine (Figure 2.9), using what is often referred to as an organic Rankine cycle (ORC). Binary systems can generate electricity from geothermal fluids of temperatures down to 85°C (or, theoretically, even lower). Recent developments such as the ammonia-water-based Kalina cycle seem set to improve efficiencies at lower temperatures (Boyle, 2004; Dickson and Fanelli, 2004).

The technology of Stirling Engines (see Box 4.2) can also be employed to convert geothermal heat of modest temperature to mechanical work and thereafter to electricity.

2.10 Direct use

Lower temperature geothermal fluids can also be used for direct uses (Lienau, 1998), which include

- space heating;
- industrial uses;
- swimming pools;
- horticulture (especially greenhouses) and aquaculture (i.e. fish farming).

Heat pumps (Chapter 4) may or may not be employed to deliver heating to these uses. Usually, the geothermal fluid will transfer its heat, via a heat exchanger (Figure 2.10), to a delivery fluid, whose chemical composition is controlled. In this way, problems with chemical incrustation or corrosion are limited to the heat exchanger and will not affect the remainder of the heating circuit.

Geothermal fluids of moderate temperature can be used to power absorption heat pumps (see Box 4.5) to provide space cooling or to produce ice.

2.11 Cascading use

Of course, in reality, the energy of many geothermal systems is not extracted by only one means of exploitation, but by several successive 'cascading' applications. For

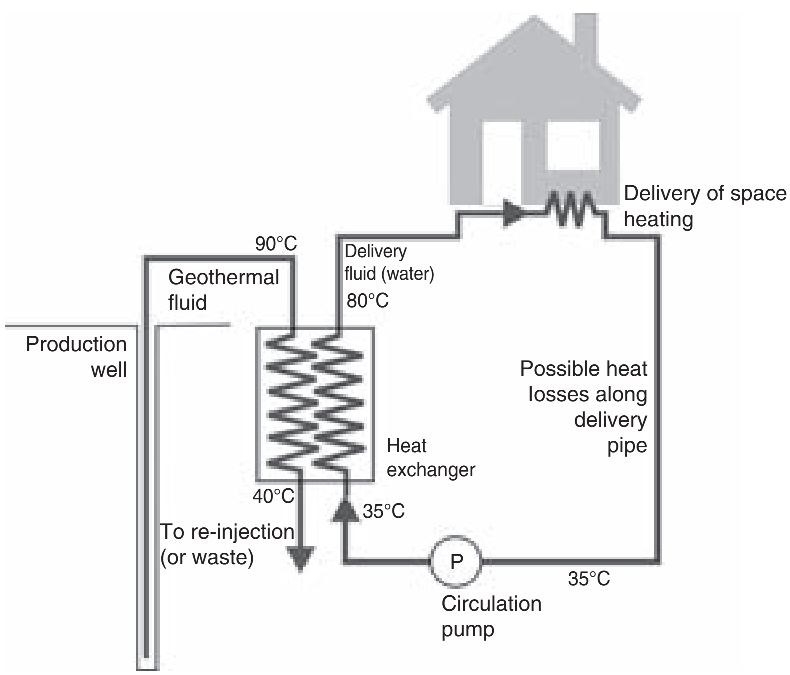


Figure 2.10 'Direct use' of a lower temperature geothermal fluid to deliver space heating.

example, a high-enthalpy geothermal system with high steam content may drive steam turbines in two steps: first using the primary steam content and second by 'flashing' the water in the separator to drive another turbine (a so-called dual-flash turbine system). Thereafter, waste heat from the turbines can be utilised for heating or cooling purposes. For example, the heat rejected by the condensers (installed after the turbines) may be extracted via heat exchangers to a secondary fluid, where it can be used directly: first, maybe, for a high-temperature industrial application or to power absorption heat pumps providing space cooling. Finally, the low temperature residual fluid may be utilised for residential space-heating purposes.

Figure 2.11a and b shows schematic diagrams for the Nesjavellir high-enthalpy geothermal field in Iceland and for the other fields contributing to the Reykjavik district-heating scheme (Lund, 2005). Note that the Nesjavellir geothermal wells produce over 1600 L s^{-1} of water at over 200°C and 14 bars. In 2005, the capacity for electricity generation by turbine at Nesjavellir was 120 MW$_e$. The waste steam from the turbines is condensed and the heat exchanged to cold groundwater from a separate wellfield, in turn raising its temperature to around 85°C. Following de-aeration to remove excess oxygen, this hot groundwater is piped around 30 km to Reykjavik,

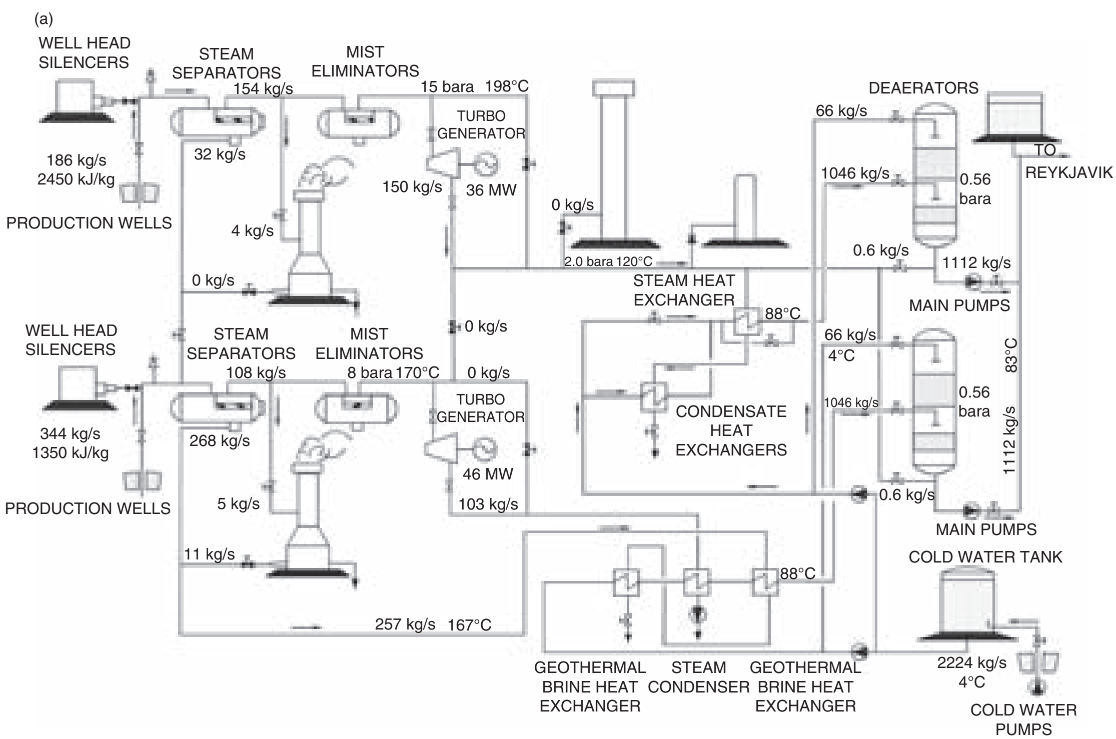


Figure 2.11 (a) Flow diagram of the Nesjavellir plant, Iceland. (b) Simplified flow diagram for the district heating in Reykjavik. *Both diagrams are after Lund (2005) and reproduced by kind permission of Dr John W. Lund and the GeoHeat Center, Klamath Falls, Oregon.*

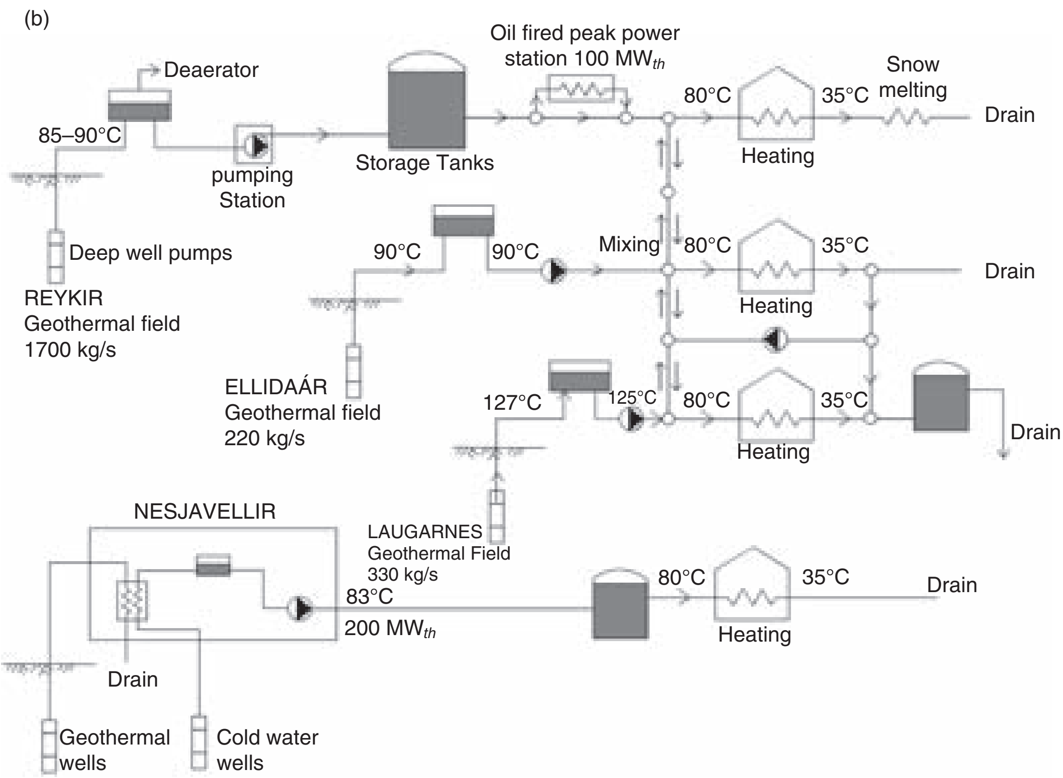


Figure 2.11 (*Continued*)

where it contributes over 200 MW$_{th}$ to the capital city's geothermal district heating scheme (which is also fed by several other geothermal fields and whose total capacity is over 800 MW$_{th}$ – see Figure 2.11b).

2.12 Hot dry rock systems [a.k.a. 'enhanced geothermal systems (EGS)']

In the discussions above, we have blithely assumed that if we drill into a geothermal reservoir, we will find a geothermal fluid (water and/or steam) that we can extract and utilise. However, some geothermal reservoirs have rather low permeability and we cannot extract large volumes of natural fluid from them. These are known as 'hot dry rock' resources, and may comprise poorly fractured hard rocks such as granites. These are attractive as they may be associated with a higher than average heat flux and geothermal gradient, due to the fact that they often contain elevated concentrations of radioactive isotopes of uranium, potassium and thorium. Such granites may represent significant geothermal anomalies, especially if buried underneath thicknesses of insulating sediments of low thermal conductivity (see Figure 2.4 and Box 2.4).

So, how do we extract heat from such a poorly permeable block of hot granite sitting in the deep subsurface? Projects such as the Cornish Hot Dry Rock project in the Carnmenellis granite of south-west England (Box 2.5) and the European trial at Soultz-sous-Forêts (Box 2.6) have discovered that we can inject cool water into deep boreholes and circulate it through artificially created fractures in the crystalline rock mass, before re-abstracting it at a higher temperature (Figure 2.12). These fractures thus effectively act as heat exchange surfaces between the rock and the water. We can create artificial fractures by using explosives down a borehole, but we can create much more controlled fractures by utilising a technique known as hydraulic fracturing or hydrofraccing (Less and Andersen, 1994; Banks and Robins, 2002; Misstear *et al*., 2006). Here, water is injected down a borehole, through a packer, at very high pressure. In fact, the pressure is so high that it exceeds the *in situ* geological stress acting on the rock mass and the rock's own inherent tensile strength and eventually creates a new fracture (or opens existing planes of weakness or joints in the rock). There is some evidence that hydrofraccing and high-pressure injection can be correlated with increased seismicity (e.g. at the Geysers in California) and such fears caused the cancellation of a European EGS project at Basel, Switzerland (Bromley and Mongillo, 2008; Basel Stadt, 2009).

BOX 2.5 The Rosemanowes EGS Project in Cornwall, UK

The reader will see (Figure 3.10) that the granites of Devon and Cornwall in south-west England provide an excellent target for geothermal exploration in an otherwise geothermally unexciting country (the United Kingdom), with a heat flux of up to 120 mW m^{-2} and a gradient of around 3.5°C per 100 m. Trials to demonstrate the 'Hot Dry Rock' concept were run between 1977 and 1991 at Rosemanowes Quarry in the Carnmenellis Granite between Falmouth and Camborne. Three deep boreholes were drilled to depths of between 2100 and 2800 m, where temperatures of around 80°C were encountered. It was demonstrated that hydraulic fracturing could be used to produce hydraulic connections between the boreholes. Although the production wells produced hot water, the depth was not great enough to produce steam for electricity generation. Some problems were also encountered due to hydrothermal short-circuiting, presumed to be caused by cool water flow along open fracture channels (with low heat exchange surface area), rather than along planar fracture features (with high surface area) – see Tenzer (2001).

The scheme showed potential to develop into a fully functional EGS scheme if the boreholes could be extended to greater depth. Lack of political and economic will at the time prevented this from being undertaken, however.

BOX 2.6 The European Hot Dry Rock Research Project: Soultz-sous-Forêts

A joint European 'Hot Dry Rock' project has been running over the past two decades at Soultz-sous-Forêts in French Alsace, not far from the border with Germany (Baldeyrou-Bailly *et al.*, 2004). The project started in 1986 and incorporated scientists from a number of existing national geothermal research projects. Several boreholes have been drilled into the crystalline basement of the Rhine Graben, an extensional tectonic structure (see Section 2.6), characterised by crustal thinning. Preliminary measurements in superficial sediments indicated high geothermal heat flows and gradients of up to 6–10°C per 100 m in places. It seems these gradients may have been exaggerated due to circulation of warm groundwater, however, as they do not persist to depth: gradients of around 2.8°C per 100 m are typical for the deep granitic basement (Tenzer, 2001). The first borehole, GPK1, was drilled to around 2 km (141°C) in 1987, and subsequently extended to 3590 m (160°C) in the early 1990s. A second hole (GPK2) was drilled to 3876 m in the first half of the 1990s, reaching a temperature of 170°C. At this stage, hydraulic fracturing successfully established hydraulic connections between the boreholes (10–25 $kg\,s^{-1}$ circulation rates seemed plausible), and steam was produced. In the late 1990s, GPK2 was extended to 5060 m, finding a temperature of 201°C (Rabemanana *et al.*, 2003). During the first decade of the twenty-first century, two further wells (GPK3 and GPK4) have been drilled to around 5 km and further hydraulic and chemical stimulation has been undertaken. Binary turbine plant has been installed at the surface to produce electricity, using GTK3 as an injection borehole, with GPK2 and GPK4 as hot extraction boreholes. The bottom-hole separation of the deviated boreholes is up to 600 m. Currently, about 35 $kg\,s^{-1}$ of fluid is being circulated, with production of 1.5 MW_e of electricity. Potential for circulation of up to 100 $kg\,s^{-1}$ and 6 MW_e may ultimately exist (Géothermie Soultz, 2010).

Although pioneering hot dry rock projects at Fenton Hill (New Mexico, USA), Rosemanowes Quarry (Box 2.5), Falkenberg, Bad Urach (both in Germany), Hijiori, Ogachi (both in Japan) and Soultz-sous-Forêts (France, Box 2.6) have demonstrated that the technology is feasible, the combination of (1) the need for a specific geological environment (which may not be in close proximity to major demand centres) and (2) the capital expenditure on research, investigation and plant, have conspired to make 'hot dry rock' applications unattractive in the current energy climate. However, in a rapidly changing global energy market where low-carbon energy sources can increasingly be sold at a premium, the day may soon arrive when other binary plants similar to that at Soultz-sous-Forêts make economic sense. Indeed, in Cornwall, UK, two new commercial hot dry rock ventures are currently being initiated (United Downs and Eden Project). A major demonstration project is underway at Coopers Basin, Australia,

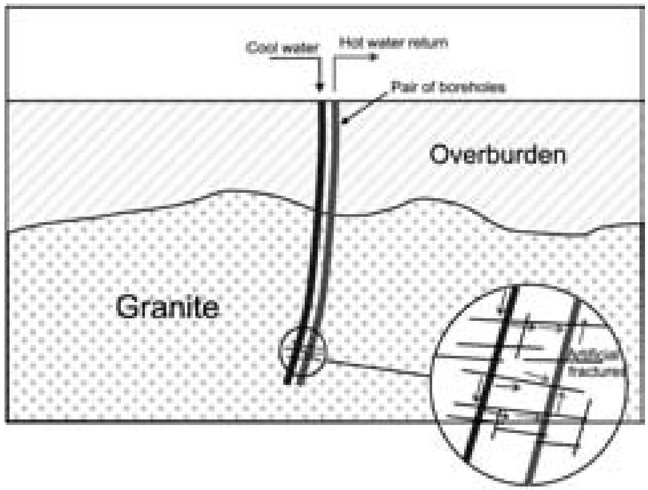


Figure 2.12 A hot dry rock (HDR) geothermal exploitation system.

while a combined heat and power plant (using 3 MW_e ORC turbines) has been operational since 2007 at Landau in the Upper Rhine Valley of Germany.

2.13 The 'sustainability' of geothermal energy and its environmental impact

When discussing the sustainability of our usage of any natural resource, it is, first, wise to be clear about our definition of the word 'sustainable' and, second, to have a clear perspective of the timescale we are considering. One of the classic definitions of sustainability is that of Gro Harlem Brundtland's commission: sustainable development meets the needs of the present without compromising the ability of future generations to meet their own needs. Stefansson and Axelsson (2003) offered a more complex definition, tailored to geothermal exploitation:

> For each geothermal system, and for each mode of production, there exists a certain level of maximum energy production, E_0, below which it will be possible to maintain constant energy production from the system for a very long time (100–300 years). If the production rate is greater than E_0, it cannot be maintained for this length of time. Geothermal energy production below, or equal to E_0, is termed *sustainable production*, while production greater than E_0 is termed *excessive production*.

The non-sustainability of our exploitation of a geothermal system may manifest itself in one of two ways (or, of course, both):

1. The temperature of the system may fall to an unusable level, because we are extracting a greater heat flux than can be naturally replenished on the timescale of the operation.

2. The supply of geothermal fluid (whether it be water or steam) may diminish, as we are abstracting the fluid at rate greater than its natural replenishment.

Thus, to assess the sustainability of a geothermal operation, we need to have a very clear understanding of both the heat budget and the water budget of the system, and the boundary conditions of these systems. When constructing our heat budget, we should remember that, in many geothermal systems, there are at least two, and maybe three, mechanisms of heat recharge to a subsurface geothermal system (Stefansson and Axelsson, 2003):

- advection of magma;
- advection of groundwater (or geothermal fluid);
- conduction.

Figure 2.13 shows an example of an approximate heat budget, on a national scale, for Iceland. As regards our fluid budget, we should understand that as we extract hot fluid from a geothermal reservoir, we may deplete the fluid resource of the system and, if there is no natural recharge of groundwater, downhole heads or pressures will drop to unusable levels. If the reservoir is open to groundwater recharge, on the other hand, abstraction may induce the additional recharge of cold groundwater to the system, maintaining fluid reserves but ultimately depleting the temperature of the reservoir and resulting in the breakthrough of cooler fluid to production wells. It has been demonstrated, both by theoretical studies (e.g. Gringarten, 1978) and empirical example, that the practice of re-injecting 'spent' geothermal fluids back into the res-

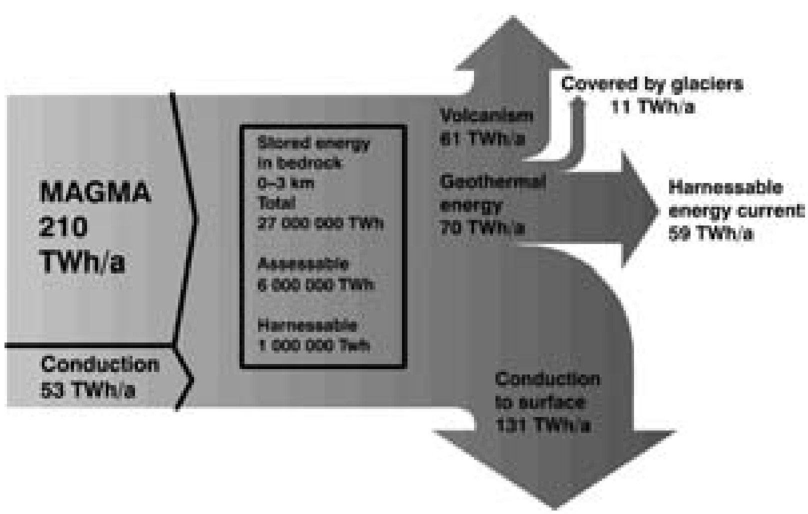


Figure 2.13 The terrestrial energy current in Iceland (after Stefansson and Axelsson, 2003). *Reproduced by kind permission of Orkustofnun and Íslenskar orkurannsóknir.*

ervoir will not only serve to maintain fluid pressures and reservoir lifetime, but can also serve to maximise the total heat extracted from the reservoir. Indeed, after years of consistently falling steam productivity at the Geysers field in California caused by paucity of natural recharge, the progressive introduction of re-injection of spent water throughout the 1980s is believed to have assisted in braking the decline by around the year 1995 (Stefansson and Axelsson, 2003).

Boyle (2004) provided some simple calculations for the Italian Tuscan geothermal fields, the Imperial Valley of California and the Krafla field of Iceland, demonstrating that the rate at which we extract heat in many geothermal operations significantly exceeds the areal rate of replenishment and argues that our exploitation of these geothermal fields is ultimately unsustainable. Rybach (2003a) also considered the sustainability of geothermal fields and indicated that we are typically considering 'lifetimes' of decades or centuries for the longevity of high-enthalpy geothermal fields and comparable periods for their recovery to natural conditions following exploitation. Stefansson and Axelsson (2003) also presented examples of geothermal operations that can be described as unsustainable on a scale of decades to centuries, including the Geysers field of California and the Hamar field of central Iceland. They also, on a somewhat more optimistic note, presented data from the Laugarnes field in Iceland, appearing to demonstrate some form of stability (= sustainability?) in production.

Finally, when considering the issue of sustainability, we would do well to remember timescale. Even though a reservoir (i.e. a set of geographically and hydrogeologically related extraction and injection wells) may turn out to have a finite lifetime of decades or centuries, the earth's tectonic and heat flow processes have a timescale of millions of years. If we have to abandon a geothermal field after 100 years, the ultimate heat source (geothermal heat flux and magmatism associated with plate tectonics) remains. Conduction, magmatic convection and groundwater convection continue to supply heat, and we can usually ultimately expect our abandoned geothermal field to recover towards its pre-exploitation temperatures on a similar timescale to that of its productive life (i.e. typically decades to centuries; Rybach, 2003a). If we are lucky enough to live in such a geologically active country as Iceland, we may be able to simply shift the centre of our operation to a new set of wells in a different geological location some tens of kilometres away and continue production while we await the recovery of our original reservoir. Such a nation may therefore be able to indefinitely sustain a national energy policy founded on geothermal energy, via a sound understanding and holistic planning of its resources of heat and geothermal fluids.

In the discussion above, we have defined 'sustainability' rather narrowly and constrained it to discussions of whether production of fluids and heat can be maintained over protracted periods. In popular perception, the discussion of sustainability is bound with the concept of pollution and environmental impact. Clearly, geothermal power stations and district heating schemes can potentially have adverse (and occasionally, beneficial) environmental impacts. Both Dickson and Fanelli (2004) and Boyle (2004) discussed these in detail, but we can here briefly list the main factors:

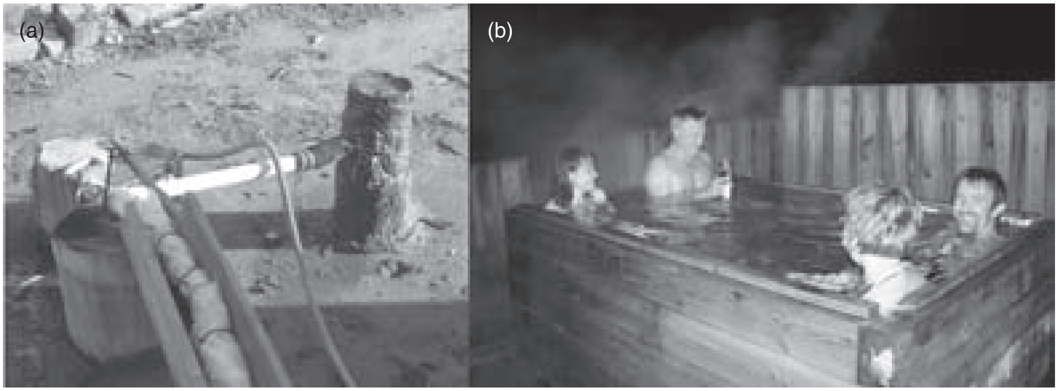


Figure 2.14 (a) A 2.6-km-deep oil exploration borehole (*photo by David Banks*) that has subsequently been converted into a 46°C geothermal well, supplying (b) hot mineral baths in the Siberian taiga (*reproduced by kind permission of Katya Komleva, Tomsk State University*). There is no significant geothermal anomaly here; the borehole simply encountered a permeable horizon at great depth. For adventurous tourists, the spa is located between Parabel and Narim (Stalin's erstwhile place of Siberian exile) on the River Ob'. **Note: If you are a health and safety officer, you are advised to read no further.** One of the bathers demonstrates the chemically reducing, H_2S- and methane-rich nature of the water, by setting fire to his own bathwater.

- Noise: from production wells and turbines. This is probably worst during the drilling, development and testing phase.
- Smell: from emission of gases such as H_2S ('rotten eggs').
- Emissions of other gases, including gases such as CO_2, from the produced geothermal fluids (see Figure 2.14). According to Boyle (2004), emissions of CO_2 from geothermal operations range from 0.004 to 0.74 kg CO_2 per kWh, with an average of 0.12 kg CO_2 per kWh. From Table 4.3, we can see that this is far from insignificant, but is much less than that from conventional electricity generation.
- Subsidence (especially where re-injection of fluid is not practiced or is inadequate) and microseismicity.
- Saline wastewaters: although these may be re-injected or run to waste in the sea. They may even be 're-branded' as tourist amenities, such as the Blue Lagoon at the Svartsengi geothermal field in Iceland (Franzson *et al.*, 1997).

In summary, therefore, although the exploitation of geothermal energy is not without some environmental drawbacks, these are widely considered to be significantly less than with conventional fossil fuel technology (Boyle, 2004).

2.14 And if we do not live in Iceland?

Meanwhile, those of us who live in such distinctly un-geothermal provinces as Henley-on-Thames and Scunthorpe are left pondering the question: What use do I

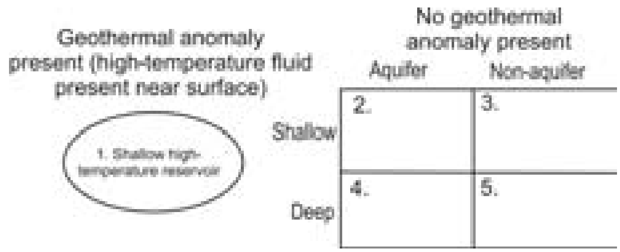


Figure 2.15 A classification scheme for geothermal systems – see study question 2.3.

have for an understanding of thermogeology? The remainder of this book is dedicated to exactly those poor souls, for the rocks beneath the Thames Valley and the Humber Estuary are blessed with the ubiquitous low-temperature reserves of heat that we will term *ground source heat*. Although Kelvin realised it 150 years ago, it has taken the rest of us over a century to recognise that we can utilise this low-temperature heat for space heating. In fact, as a strategic 'green' energy resource, it is almost certainly far more significant for the planet's future than the geothermal energy considered in this chapter.

So, let us leave conventional, high-temperature geothermal energy and geothermics behind us. Onwards! To the infinitely more exciting realm of ground source heat and thermogeology. . . .

STUDY QUESTIONS

2.1 In Box 2.3, you will note that we talk about transmissivity in units of darcy-metres (Dm) instead of the more familiar $m^2\,day^{-1}$. Drawdown is given in $MN\,m^{-2}$ instead of metres (m) of water. Why is this? Can you convert Dm and $MN\,m^{-2}$ to the more familiar units? You should know that a hydraulic conductivity for fresh water of $1\,m\,day^{-1}$ is equal to an *intrinsic permeability* (look it up in the Glossary) of about $1.18 \times 10^{-12}\,m^2$ or 1.2 D.

2.2 In Box 2.6, flow rates are cited in $kg\,s^{-1}$ and not $L\,s^{-1}$. Why is this?

2.3 Place the following schemes within the box classification scheme (Figure 2.15) for geothermal systems: Larderello, Eastgate, Dunston Innovation Centre, Rosemanowes, Soultz-sous-Forêts, Nesjavellir, Southampton, Arlanda Airport, Annfield Plain, typical open-loop ground sourced heat pump systems, typical closed-loop ground sourced heat pump systems.

3

The Subsurface as a Heat Storage Reservoir

> Yonder the harvest of cold months laid up,
> Gives a fresh coolness to the Royal Cup;
> There Ice like Christal, firm and never lost,
> Tempers hot July with December's frost . . .
> . . . Strange! that extreames should thus preserve the Snow
> High on the Alpes and in deep Caves below!
>
> Edmund Waller (1606–1687),
> on the occasional of a new royal icehouse at St James's Park, London,
> cited by Buxbaum (2002)

In its most basic form, a household storage heater is a box full of bricks that is heated by electric elements when electricity is cheap, usually at night. The heat is then slowly released during the day to keep the house warm. The storage heater uses bricks because such silicate-based media not only have a very high capacity to store heat, but also have a rather modest *thermal conductivity* – they release heat relatively slowly.

The earth's shallow subsurface can be regarded as a huge storage heater. It is warmed up by solar energy during the summer. We can also deliberately inject surplus heat into the ground when it is plentiful (during summer). We can access and extract that

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

heat during winter. Examples of surplus heat that can be stored in the *thermogeological accumulator* that the ground represents include

- natural solar radiation absorbed via the ground surface;
- solar energy deliberately harvested via solar panels and injected into the ground;
- 'waste' heat from air conditioning or dehumidification of commercial buildings or of greenhouses;
- 'waste' heat from industrial processes;
- surplus heat from combined heat and power (CHP) installations.

Most rocks are silicate-based and, like bricks, they have a huge potential to store heat. Their thermal conductivity is modest: not so high that the stored heat dissipates immediately, but not so low that we cannot draw it out of the ground through well-designed heat exchangers (typically installed vertically in boreholes or horizontally in trenches).

You have probably noticed that we are introducing two fundamental thermogeological properties here: thermal conductivity and storage. Thermal conductivity describes a material's ability to transfer heat by conduction, as described by Fourier's law (Box 3.1). We have already met this property in Equation 1.1. Already in the latter half of the nineteenth century, scientists were beginning to measure the thermal conductivity of rocks: indeed William Thomson required a value of thermal conductivity as input to his model of the cooling earth (Section 2.2). In the 1860s, Thomson, J.D. Everett and A.J. Ångström had started making determinations of the thermal diffusivity/conductivity of sediments and soils (Everett, 1860; Thomson, 1868; Rambaut, 1900). Professors A.S. Herschel and G.A. Lebour of the University of Durham College of Science (later to become the University of Newcastle-upon-Tyne) were, in the 1870s, able to present a series of determinations of thermal conductivities of rocks (Herschel and Lebour, 1877; Herschel *et al.*, 1879; Prestwich, 1885, 1886; Barratt, 1914).

The property describing storage of heat is called specific or volumetric heat capacity. It is new to us, so let us consider it in more detail.

3.1 Specific heat capacity: the ability to store heat

The ability of a medium (solid, liquid or gas) to store heat is termed its *specific heat capacity* (S_C). This is the amount of heat locked up in the medium for every degree Kelvin of temperature. It is measured in joules per Kelvin per kilogram ($\mathrm{J\,K^{-1}\,kg^{-1}}$). The specific heat capacity of water is particularly high at around $4180\,\mathrm{J\,K^{-1}\,kg^{-1}}$ at around 20°C; that of most solid rocks is around $800\,\mathrm{J\,K^{-1}\,kg^{-1}}$ (see Table 3.1). This means that if a mass (m) of 1 kg rock cools down (by an amount $\Delta\theta$) from 13 to 11°C then two degrees' worth of heat is lost, 1600 J:

BOX 3.1 Jean Baptiste Joseph Fourier

Fourier was born in 1768 into a large tailor family in Auxerre, France (O'Connor and Robertson, 1997). By the age of 13, his mathematical talents had become clear, but in his studies he was torn between following a career as a priest and indulging his love for maths. The latter eventually won. In 1790, he took a post as teacher at his former college, the École Royale Militaire of Auxerre. After a brush with revolutionary politics, which nearly terminated his life at the guillotine's blade, he went on in 1795 to the École Normale, followed by the Central School of Public Works (*École Centrale des Travaux Publiques*), where he benefited from close contact with the mathematicians, Joseph-Louis Lagrange and Pierre-Simon Laplace.

Fourier's subsequent career included positions as professor at the École Polytechnique and adviser to Napoleon during his invasion of Egypt. Later (at Napoleon's bidding), he 'accepted' the Prefecture of the Department of Isère in Grenoble, where he accomplished a number of civil engineering works. At Grenoble, he obviously had some time to pursue theoretical research; between 1804 and 1807, he completed his work *On the Propagation of Heat in Solid Bodies*, which introduced the world to his mathematical technique of expressing mathematical functions as what we now call 'Fourier series'. Fourier's theory was not universally popular: there was a tendency among some scientists to regard heat as a substance called 'caloric' and expend much effort discussing what this substance was. According to Greco (2002), Fourier's great insight was to avoid this speculation about what heat was and to focus on what it did. In other words, he treated heat flow as a mathematical process, rather than attempting to squeeze it into an ill-fitting physical conceptualisation.

Fourier died in 1830 in Paris.

$$\text{Heat lost} = m \times S_C \times \Delta\theta = 1\ \text{kg} \times 800\ \text{J K}^{-1}\ \text{kg}^{-1} \times 2\ \text{K} \tag{3.1}$$

We can also express specific heat capacity as joules per degree Kelvin per unit volume. This is termed the volumetric heat capacity (S_{VC}). The conversion is given by

$$S_{VC} = \rho S_C \tag{3.2}$$

where ρ is the density of the material in question. For water, $S_{VC} \approx 4180\,\text{J K}^{-1}\,\text{L}^{-1}$ at around 15–20°C (as its density $\approx 1\,\text{kg L}^{-1}$), whereas most rocks and saturated sediments have values of S_{VC} in the range 1.9–2.5 MJ $\text{K}^{-1}\,\text{m}^{-3}$. So, from every cubic metre of rock, we can release up to 10 MJ of energy, simply by dropping its temperature by 4 K. Conversely, we need to put a similar amount of energy back into our cubic metre of rock to raise its temperature by 4 K. The heat energy in the material is ultimately

Table 3.1 The thermal conductivity and volumetric heat capacity of selected rocks and minerals.

	Thermal conductivity ($W\,m^{-1}\,K^{-1}$)	Volumetric heat capacity ($MJ\,m^{-3}\,K^{-1}$)
Rocks		
Coal	*0.3*	*1.8*
Limestone	1.5–3.0 (*2.8, massive limestone*)	1.9–2.4 (*2.3*)
Shale	1.5–3.5 (*2.1*)	*2.3*
Basalt	1.3–2.3 (*1.7*)	*2.4*–2.6
Diorite	1.7–3.0 (*2.6*)	*2.9*–3.3
Sandstone	2.0–6.5 (*2.3*)	*2.0*–2.1
Gneiss	2.5–4.5 (*2.9*)	2.1–2.6 (*2.1*)
Arkose	2.3–3.7 (*2.9*)	*2.0*
Granite	3.0–4.0 (*3.4*)	1.6–3.1 (*2.4*)
Quartzite	5.5–7.5 (*6.0*)	1.9–2.7 (*2.1*)
Minerals		
Plagioclase	1.5–2.3	1.64–2.21
Mica	2.0–2.3	2.2–2.3
K-feldspar	2.3–2.5	1.6–1.8
Olivine	3.1–5.1	2.0–3.6
Quartz	7.7-7.8	1.9–2.0
Calcite	3.4-3.6	2.24
Pyrite	19.2–23.2	2.58
Galena	2.3–2.8	1.59
Haematite	11.3–12.4	3.19
Diamond	545	–
Halite	5.9–6.5	1.98
Other		
Air	0.024	1.29×10^{-3} at 1 atm
Glass	0.8–1.3	1.6–1.9
Concrete	0.8–1.7 (*1.6*)	*1.8*
Ice	1.7–2.0 (*2.2*)	*1.9*
Water	*0.6*	4.18
Copper	390	3.5
Freon-12[a] at 7°C (liquid)	0.073	1.3
Oak	0.1–0.4	1.4
Polypropene	0.17–0.20	1.7
Expanded polystyrene	0.035	–

[a] Dichlorodifluoromethane (CCl_2F_2).
Data from *inter alia* Halliday and Resnick (1978), Farouki (1982), Sundberg (1991), Clauser and Huenges (1995), Midttømme *et al*. (1998), Eskilson *et al*. (2000), Banks and Robins (2002), Banks *et al*. (2004a), Waples and Waples (2004a), Lienhard and Lienhard (2006) and VDI (2008). Italics show recommended values cited by Eskilson *et al*. (2000).

stored as molecular vibrational or kinetic energy; the hotter the material, the faster the molecules of a fluid whiz around, or the molecules of a crystal vibrate.

Note that, because the S_{VC} of water is exceptionally high, the S_{VC} of porous rocks, soils and sediments depends strongly on their moisture content. Very porous water-saturated sands, and microporous limestones such as the Chalk, can have volumetric heat capacities of $2.5\,MJ\,K^{-1}\,m^{-3}$ or higher. Dry sediments and soils can, conversely, have very low capacities of $<1.5\,MJ\,K^{-1}\,m^{-3}$, in some cases. Many sediments comprise solid (mineral), water and air phases: the composite volumetric heat capacity can be calculated as the volume-weighted arithmetic mean of the various components.

Volumetric heat capacity varies somewhat with temperature, partly (but not wholly) due to changes in density of the material. Figures 3.1 and 9.5 show how this affects the thermal properties of water at varying temperature (similar temperature-dependent variations also apply to solutions of antifreezes). For this reason, physicists tend to prefer using the term ρS_C is their equations, rather than S_{VC}. In this book, however, we are usually considering ground source heating/cooling systems that employ very modest temperature changes and I have decided to retain the term S_{VC} for simplicity.

We should remember that heat may also be stored or released from a substance due, not just to change in temperature (Box 3.2), but to change in phase. This stored heat is called *latent heat* (Box 3.3).

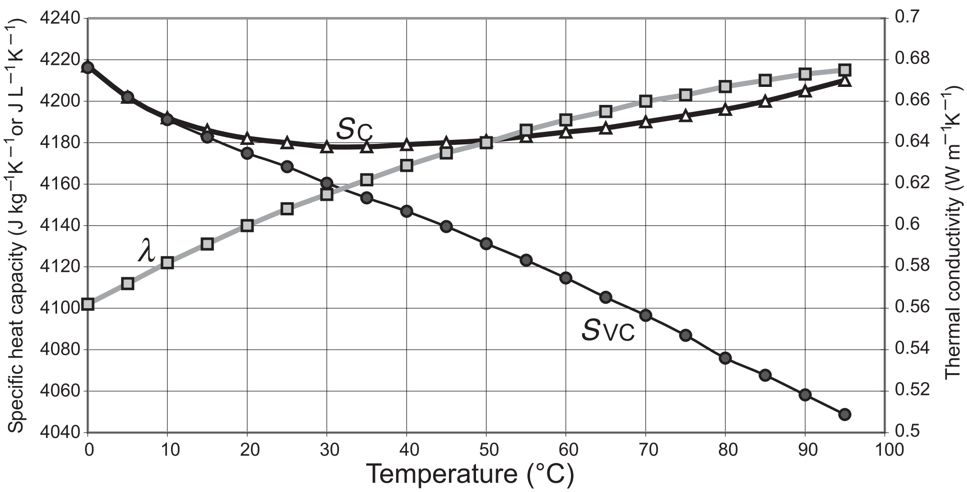


Figure 3.1 The temperature dependence of the thermal properties of water. In this diagram, S_C and S_{VC} are the specific heat capacity and volumetric specific heat capacity, while λ is the thermal conductivity. Values on the graph are derived or calculated from data provided by Eskilson *et al.* (2000).

BOX 3.2 Sensible Heat

Sensible heat is the opposite of latent heat (Box 3.3). It is heat which, when added to a solid, liquid or gas, results in an increase in temperature. The relationship between heat and temperature is encapsulated in the concept of specific heat capacity. For water at around 20°C, 4.2 kJ of heat results in a 1°C temperature change per litre. In other words, this is *sensible heat* (it can be sensed).

When heat is added to water at 100°C, it does not result in any sensible temperature change; it merely serves to convert liquid water to steam (see Box 3.3).

BOX 3.3 Latent Heat

Some energy is stored in materials simply as a property of their phase, that is, a gas, liquid or solid. For example, steam at 100°C is far more energetic than liquid water at the same temperature. To convert a liquid to a gas (i.e. to move it across the liquidus in Figure 2.7), we have to inject an additional amount of energy, without actually raising the temperature at all. The same applies if we consider the evaporation of a liquid such as water (to form water vapour) below its boiling point. For water, this *latent heat of vaporisation* is around 2.26 MJ kg^{-1} at 100°C and 2.45 MJ kg^{-1} at around 20°C (Incropera *et al.*, 2007). The uptake and loss of heat during volatilization and condensation is one of the features that makes the compression–expansion cycle at the heart of refrigerators and heat pumps function (see Chapter 4).

Similarly, when a solid melts to a liquid, it requires an uptake of heat to effect the phase transition, without any temperature rise resulting. To convert ice at 0°C to water at 0°C requires 335 kJ kg^{-1}: the *latent heat of fusion*.

3.2 Movement of heat

It is not enough to know that rocks, sediments and groundwater in the earth's subsurface store heat, we also need to understand how the subsurface absorbs and loses heat, and how we can induce heat to move to places where we can extract it. Heat is transferred by three main mechanisms:

- conduction (we have already met this in Fourier's law, Equation 1.1);
- convection;
- radiation.

In the shallow subsurface environment, conduction through minerals or pore fluids and convection via groundwater are probably the two most important mechanisms of heat flow. In some cases, radiation may also be important, so let us consider that, too.

3.2.1 Conduction

Heat conduction describes the process by which heat diffuses through a solid, liquid or gas by processes of molecular interaction. Put crudely, if we warm up one end of a chunk of granite, the molecules at that end start vibrating more strongly. These vibrating atoms or molecules cause their neighbours to start vibrating as well, such that the heat energy (and temperature) gradually diffuses throughout the piece of rock. It is this process that Fourier's law describes by means of Equation 1.1.

The thermal conductivity describes how good the medium is at conducting heat: copper is very good, rocks are less good and plastics are generally poor. Note (Tables 3.1 and 11.2) that the thermal conductivities of rocks and other geological materials tend to fall within a rather narrow range, typically between 1 and 3 $W\,m^{-1}\,K^{-1}$. Note also that, of the common rock-forming minerals, quartz has the highest thermal conductivity, at around 7 $W\,m^{-1}\,K^{-1}$. Thus, the thermal conductivity of rocks depends to a significant extent on their quartz content.

Note that the thermal conductivity of most rock-forming minerals is quite good (and especially quartz), water has a lower thermal conductivity, while air is very poor. Thus, the best geological conductors are rocks with low porosity and a dense interlocking crystal structure with a high quartz content, such as granite. Porous rocks, where the pores are saturated with water are less good conductors: water saturated clays tend to have conductivities in the range of 1.5–2 $W\,m^{-1}\,K^{-1}$, for example, while water-saturated quartz sands can have a somewhat higher conductivity. The worst conductors are dry, porous sediments, such as dry, well-sorted sands and silts (<1 $W\,m^{-1}\,K^{-1}$ in the worst cases). Remember, however, that natural sediments are seldom completely dry: although they may not be water saturated, they will usually contain some soil moisture. Fine-grained soils and sediments (clays and silts) are usually better at retaining moisture than sands and gravels, when they are located above the water table. Farouki (1982) provided useful tables of the variation of thermal conductivity with sediment compaction and moisture content.

Although we tend to treat thermal conductivity as a constant, it is, in fact, temperature-dependent to a minor degree in most materials (e.g. see Figures 3.1 and 9.5). When considering the thermogeology of ground source heating/cooling systems, we are normally working within a relatively narrow temperature range and can neglect this effect in most cases.

Thermal conductivity is also dependent on a material's phase: note, for example, that the thermal conductivity of ice is much higher than that of water (Table 3.1; its specific heat capacity is lower, however). Counter-intuitively, therefore, the thermal conductivity of soils and sediments tends to *increase* when they freeze.

3.2.2 Convection

Fluids store heat: for example, water stores around 4180 $J\,L^{-1}$ for every °C of temperature. Thus, if we move hot water from a boiler house to a bathroom, we are also moving heat. Heat transport that occurs by virtue of the motion of a fluid is termed convection. If we pump hot water from one place to another, we term the heat transport forced convection or *advection*, because the heat is flowing due to a force externally imposed on the carrier fluid. Isaac Newton proposed a simple formula for the rate at which heat is transferred to, or away from, bodies in a moving stream of fluid. This formula (Newton's Law of Cooling) can be stated in the following form:

$$q^{*} = \bar{h} \cdot (\theta_{\mathrm{body}} - \theta_{\mathrm{fl}}) \tag{3.3}$$

Where

- q^{*} is the heat transfer from body to fluid in $\mathrm{W\,m^{-2}}$ of surface area;
- $\bar{h}$ is the local coefficient of heat transfer ($\mathrm{W\,m^{-2}\,K^{-1}}$), which will depend on the nature of the fluid, its rate of flow, the body's surface properties and so on;
- θ_{body} and θ_{fl} are the temperatures of the body and the fluid, respectively.

Newton's Law of Cooling is really a working approximation rather than a law. It functions pretty well for forced convection situations where the temperature difference between the body and the fluid is not too large.

Convectional heat transfer can also take place from a hot body in a fluid that is initially static, with no externally imposed forces. Imagine switching on an electric bar fire in your lounge: the air near the bar heats up (by conduction and radiation) and it expands slightly (see Box 3.4). It thus becomes less dense than the surrounding air and starts to rise, being displaced by denser cold air. This new cold air soon warms up and rises and, before you know it, a convection cell has started within the room. The bar fire provides the heat source. The heat may ultimately be lost from the air

BOX 3.4 Boyle's Law and the Universal Gas Constant

The Irish-born scientist Robert Boyle (1627–1691) spent much time studying gases. In an appendix, written in 1662, to his work *New Experiments Physio-Mechanicall, Touching the Spring of the Air and Its Effects*, he quantified the relationship between their pressure and volume (O'Connor and Robertson, 2000). We intuitively understand that if we pressurise a given quantity of gas, we reduce its volume. Boyle found that, for a given mass of ideal gas (and most gases behave relatively ideally) at a constant temperature, which has an original volume V_1 and pressure P_1, but which is pressurised to a pressure P_2:

$$P_1 \cdot V_1 = P_2 \cdot V_2 = \text{constant}$$

where V_2 is the final volume of the gas at P_2.

Later, the temperature of the gas was found to fit neatly into this relationship:

$$\frac{P_1 \cdot V_1}{T_1} = \frac{P_2 \cdot V_2}{T_2}$$

where T_1 and T_2 are the initial and final temperatures (expressed as degrees Kelvin). In other words, when a gas is pressurised, it has a tendency to heat up

(*Continued*)

BOX 3.4 (*Continued*)

(think of a bike pump heating up as you pump up your tyres). When pressure is released, it has a tendency to cool down. You may have used a bottled gas stove when cooking baked beans on a camping trip. When you open the valve on the pressurised gas cylinder, allowing gas to flow to the burner, you are slowly decreasing the pressure in the cylinder. The remaining fluid cools and you may notice drops of condensation on the cylinder. (Note, however, that neither of these examples is strictly appropriate, as the mass of gas in the tyre and the cylinder change during these 'experiments'.)

In fact, it turns out that we can specify a universal gas constant from which we can calculate the volume of a given amount of gas at any temperature and pressure:

$$n = \frac{M}{m} = \frac{P \cdot V}{R \cdot T}$$

This is the so-called Ideal Gas Law, where n = the amount of gas (in moles – a mole is simply a chemical quantity that allows us to compare different gases):

M = mass of gas (kg);
m = molar mass of gas ($\mathrm{kg\,mol^{-1}}$);
P = pressure of gas ($\mathrm{Pa = N\,m^{-2} = kg\,m^{-1}\,s^{-2}}$);
V = volume of gas ($\mathrm{m^3}$);
T = absolute temperature of gas (K);
R = the universal gas constant = $8.3145\,\mathrm{kg\,m^2\,s^{-2}\,K^{-1}\,mol^{-1}}$.

Note (Box 1.1) that we have already seen that $1\,\mathrm{J} = 1\,\mathrm{kg\,m^2\,s^{-2}}$. Thus,

$$R = 8.3145\,\mathrm{J\,K^{-1}\,mol^{-1}}$$

The gas law helps us to predict what happens to a gas when we heat it. If we heat a closed vessel of gas, the volume is fixed and constant. Thus, as the temperature rises, the pressure increases to ensure that the quantity PV/RT remains constant. If, however, we just heat a region of air in a room, the pressure is constant (atmospheric) and the region of air must expand slightly (become less dense) for PV/RT to remain constant. This decreasing density causes that region of air to become more buoyant than the surrounding air and to rise: the beginning of free convection.

Note that the equations in this box only hold true for gases (and ideal gases, at that!).

to a ceiling, or window or external wall, but heat transport with fluid (air) flow has taken place. We call this 'free convection'.

Newton's Law of Cooling is even more difficult to apply to situations involving free convection, as the heat transfer coefficient will vary as a function of the temperature difference. In fact, Lienhard and Lienhard (2006) suggested that, for free convection,

$$\bar{h} \propto (\theta_{\mathrm{body}} - \theta_{\mathrm{fl}})^{n} \tag{3.4}$$

where n is a power, often in the range of 0.25–0.35, but sometimes reaching as high as 2 in boiling liquids.

In Section 2.5, we have already considered the huge convection cells of fluid mantle rock that are related to global plate tectonic processes and continental movements. In the applied science of thermogeology, however, it is not this type of subsurface convection that interests us. Rather, we need to know about the forced convection (advection) of heat that occurs with groundwater flow. Groundwater in the shallow subsurface flows from areas of high head to low head (Box 1.2), usually ultimately driven by gravitational forces. In doing so, it carries with it a huge cargo of heat (around $4.2\,\mathrm{kJ\,K^{-1}\,L^{-1}}$). If we drill a well and start pumping it, we locally reduce the groundwater head, forming a *cone of depression* (Figure 3.2), causing groundwater to

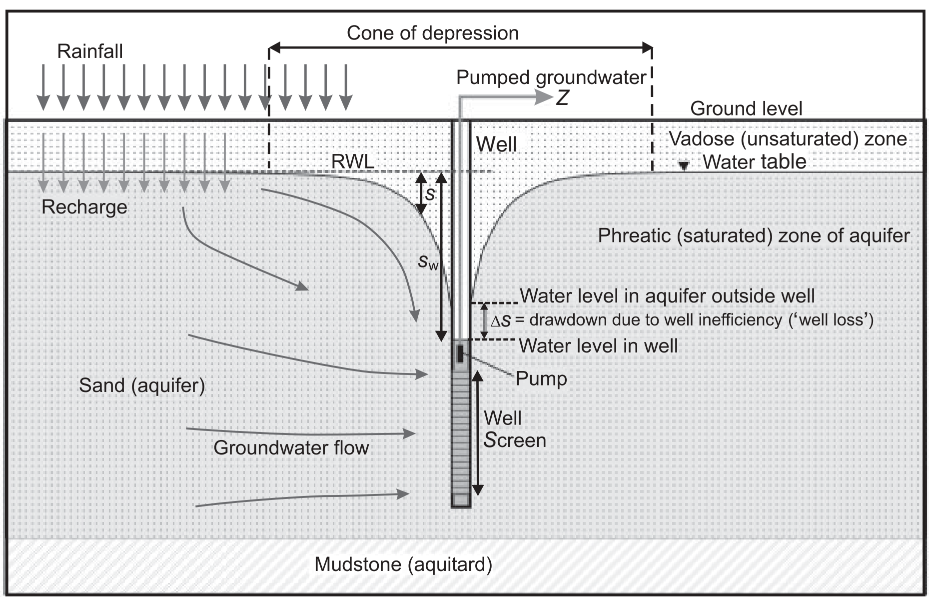


Figure 3.2 Conceptual model of a well abstracting groundwater (at a rate Z) from an unconfined aquifer. The abstracted water also carries a cargo of heat (advection) that can be utilised. Drawdown (s) is defined as the difference between rest water level (RWL) in the aquifer and the groundwater level during pumping. The drawdown in the pumped well (s_w) is the sum of the drawdown in the aquifer and additional drawdown (Δs) due to the hydraulic inefficiency of the well.

flow towards the well, where it can be abstracted. The water can be used for drinking, for household or industrial use – but we can also extract the advected cargo of heat and utilise that, too.

By the way, in thermogeology, we may also need to consider free convection cells that can establish themselves in groundwater (or even soil gas) within or around water wells or heat extraction boreholes.

3.2.3 Radiation

All bodies radiate energy in the form of electromagnetic radiation: stars, humans, lakes and the earth's cool surface. The hotter the body, the more energy it radiates. Indeed, Stefan (via experimental work in 1879) and Boltzmann (via theoretical consideration, in 1884) stated that, for an ideally radiating body (a so-called *black body*), the energy radiated (E_b) is proportional to the fourth power of the absolute temperature (θ^o, in Kelvin):

$$E_b = \sigma\left(\theta^o\right)^4 \tag{3.5}$$

where σ is the Stefan–Boltzmann constant of $5.67 \times 10^{-8}\,\mathrm{W\,m^{-2}\,K^{-4}}$.

Hot bodies radiate electromagnetic energy through a wide spectrum of wavelengths: the sun's radiation is, for example, dominated by ultraviolet, visible and infrared wavelengths. However, in general, the hotter the body, the shorter the modal wavelength (i.e. the wavelength of peak intensity) in the radiation spectrum – this is known as Wien's law. Thus, the hottest stars are bluish in hue, the cooler ones red. We know that a horseshoe in the blacksmith's furnace glows 'red hot' – it is also radiating heat in the visible spectrum.

At more familiar temperatures, such as that of our bodies or the ground that we stand on, heat is primarily radiated in the invisible, infrared part of the spectrum. The earth's surface radiates such heat energy, and an infrared camera (maybe satellite mounted) is able to sense such radiation and compile temperature maps of the earth's surface.

Very cold bodies radiate electromagnetic energy at much longer wavelengths. The universe ('space') can be regarded as having a background temperature of some 3 K above absolute zero. Throughout the universe, we can detect radiation (the so-called cosmic background) with a wavelength of some 1.9 mm (corresponding to 3 K) in the microwave region of the spectrum (even longer wavelength than visible light and infrared).

The heat that we feel on our skin or which is absorbed by the soil on a sunny day is infrared radiation emitted by the sun (along with visible light). Thus, radiation of heat and absorption of radiated energy from the sun and atmosphere are important heat transfer mechanisms at the surface of the earth (Section 3.4). Particularly when we consider the heat budget of bodies such as lakes and ponds (Chapter 12), the radiation of heat energy may be an important component in their heat budget.

3.3 The temperature of the ground

Rocks and sediments have high values of volumetric heat capacity (S_{VC}) but modest values of thermal conductivity (λ). Thus, they have rather low values of thermal diffusivity (Box 3.5). Heat pulses do not propagate very fast or far throughout the subsurface of the earth (at least, in the absence of advection by groundwater).

In summer, the surface of the earth heats up due to intensified solar radiation and elevated air temperatures. This heating effect propagates a few metres down into the earth's subsurface, but not beyond. In fact, below around 10 m depth, the temperature of the subsurface is remarkably stable, at a value approximating to the long-term annual average surface temperature. Figure 3.3 shows a hypothetical example for typical Swedish ground conditions (Rosén *et al.*, 2001) where, throughout the course of a year, the surface temperature varies by around 20°C. At 6 m depth, the amplitude of seasonal temperature variation is no more than 1°C. The seasonal temperature signal at various depths formed the basis for some of the earliest determinations of the thermal diffusivity of soils by the likes of Ångström, Thomson and Everett (see above). For a full treatment of how seasonal and diurnal temperature cycles propagate into the ground, Chapter 5 of Ingersoll *et al.* (1954) is difficult to beat.

In much of the United Kingdom, the annual average surface temperature is in the range 9–12°C and the stable subsurface temperature reflects this (Box 3.6). In other words, the earth's subsurface temperature is warmer than the air temperature in winter, but cooler in summer. The earth thus provides a convenient source of heat in winter and a source of cooling (i.e. a sink to 'dump' waste heat) in summer.

Figure 3.5 shows how the low thermal diffusivity of the earth's subsurface subdues the amplitude of the annual air temperature 'signal' and also delays it in time. The diagram is derived from a waterworks at Elverum in inland Norway, which has both a river water intake (from the River Glomma) and some groundwater wells.

BOX 3.5 Thermal Diffusivity

The ratio of thermal conductivity to volumetric specific heat capacity is known as thermal diffusivity (α), which has units m^2 s^{-1}:

$$\alpha = \frac{\lambda}{S_{VC}} = \frac{\lambda}{\rho S_C}$$

where λ = thermal conductivity (W m^{-1} K^{-1}), S_C = specific heat capacity by mass (J K^{-1} kg^{-1}), S_{VC} = specific heat capacity by volume (J K^{-1} m^{-3}) and ρ = density (kg m^{-3}).

The thermal diffusivity represents the rate and extent to which a heat signal or heat pulse is propagated throughout a medium (see Box 3.7).

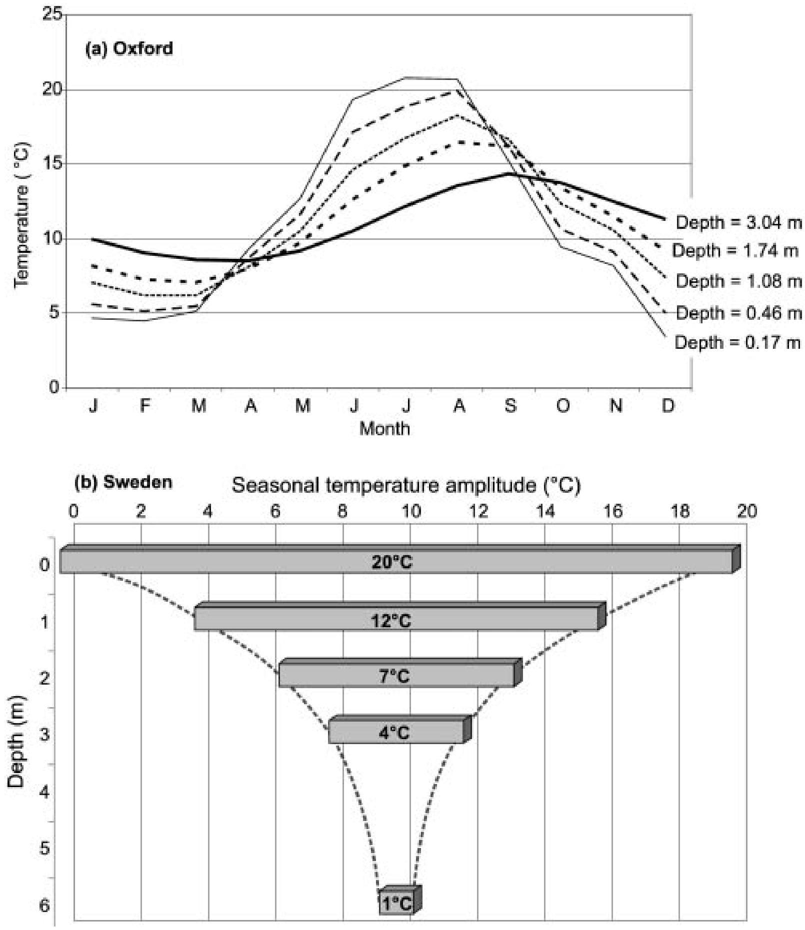


Figure 3.3 (a) The seasonal temperature fluctuation in the subsurface at various depths, as observed at Radcliffe, Oxford, UK, in 1899 by Rambaut (1900). Not only does the amplitude decrease with depth, the temperature maxima and minima become progressively delayed in time. (b) The magnitude of subsurface temperature fluctuation with depth, resulting from an annual temperature variation of 20°C at the surface, based on typical Swedish conditions, using data cited by Rosén *et al*. (2001).

Furthermore, a weather station monitors air temperature throughout the year. It can be seen that the air temperature is subject to rapid fluctuation, reaching a minimum of –24°C in winter and the positively balmy heights of +20°C in summer. The river water temperature shows a more subdued curve, with lower 'amplitude': the river is frozen (0°C) in winter and does not rise above 15°C in summer. Furthermore, the river temperature maximum (in August) is delayed relative to the maximum air temperature. The groundwater trace shows almost no temperature signal: it has a constant temperature of 5°C, just a little above the mean annual air temperature of ≈3°C (the

BOX 3.6 Annual Average Air Temperature, Soil Temperature and Subsurface Temperature: The Effect of Winter Snow Cover and Freezing

In the chapter, we have talked rather evasively about the annual average 'surface' temperature. When we start looking in detail, we find that the annual average air temperature usually differs slightly from the annual average soil temperature (and is typically a little lower). There are many factors that influence the magnitude of any such difference: for example, the radiative budget and the efficiency of heat transfer between soil and atmosphere, the aspect of the terrain (south-sloping or north-sloping; shaded?), the vegetation cover, the permeability (the ease with which rain and snowmelt can percolate into the soil), the soil moisture at different times of the year and snow cover patterns.

Snow is reflective, so if snow cover exists at times and latitudes where days may be sunny, it may serve to reflect incoming solar radiation and decrease absorption of solar energy. On the other hand, porous snow has a low thermal conductivity: it will insulate the underlying soil from extreme winter air temperatures. In northern latitudes, this effect dominates: in Sweden, Rosén *et al.* (2001) claimed that every 100 days with snow cover increases the annual average soil temperature by 1.5°C, relative to average annual air temperature.

Look at the climate data in Figure 3.6 for the town of Taiga (near Tomsk) and the city of Kemerovo in southern West Siberia. The annual average air temperature in both cases is a little below zero: we might even expect to see permafrost beginning to form here – but we do not. This part of Siberia is free from permafrost, although ice may form seasonally in the subsurface and persist well into summer before it eventually melts (Figure 3.4). Indeed, the average temperature observed in the shallow subsurface (e.g. the groundwater in shallow wells) is relatively high, typically +3–5°C (Parnachev *et al.*, 1999; Banks *et al.*, 2004b). We can speculate that at least two factors may be important for the observed differential between annual average air and subsoil temperatures:

1. The fact that a substantial snow cover persists for several months of the winter, insulating the soil from the extremely low winter temperatures.
2. The majority of the precipitation occurs in summer. A small portion of this 'warm' precipitation may infiltrate into the subsoil, elevating subsurface temperatures. The rather small amount of winter precipitation will not infiltrate the subsurface – it will remain frozen in the shallow subsurface or at the surface as snow or ice. When the accumulated snow cover eventually melts in around March–April, it will infiltrate the subsoil at 0°C.

Figure 3.4 Frozen mining spoil at Novii Berikul in south-western Siberia. Frozen ground, formed during the previous severe winter, has persisted as a layer, over 1 m thick in places, well into August 2010, when the photo was taken. The frozen ground was exposed by excavators, extracting the mining spoil for use in road construction. Remember that this is *not* permafrost – it is seasonal frozen ground, which melts throughout the summer and autumn and which is sandwiched between unfrozen ground above and below. It clearly illustrates the fact that the ground is a highly efficient *store* of winter cold. *Photo by D. Banks.*

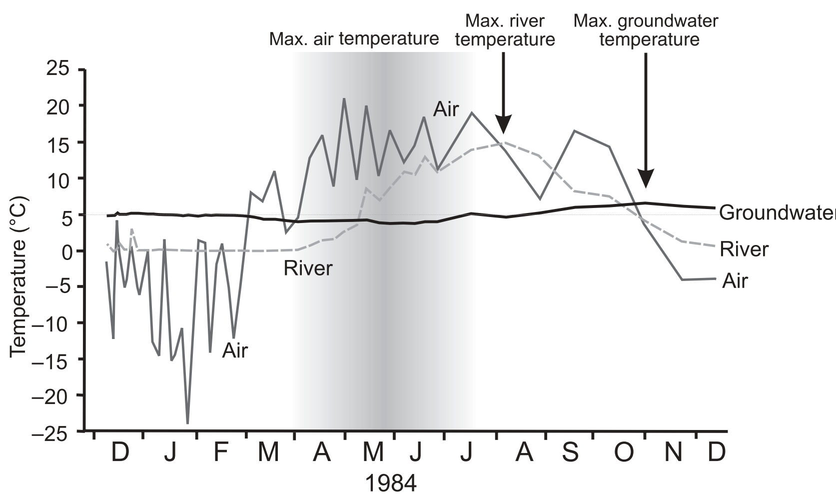


Figure 3.5 Graph showing fluctuation of temperature of the air, the River Glomma and shallow groundwater, at Elverum, inland Norway, during 1984. *Based on an original diagram by Randi Kalskin Ramstad and reproduced by kind permission of Norges geologiske undersøkelse, Trondheim, Norway.*

Norwegian Meteorological Institute, climate data for station 6600 Elverum, 1961–1990; see also Box 3.6). The imaginative eye may, however, discern a very faint amplitude, with a groundwater temperature maximum as late as November.

The temperature of the shallow subsurface (below the zone of seasonal fluctuation) varies according to climatic zone: in the United Kingdom, we have seen that it is

Table 3.2 Typical shallow (<150 m) ground temperatures in Europe (after Geotrainet, 2010), based on information provided by geological authorities in each country. Ground temperature is controlled by annual average soil temperature, which depends on annual average air temperature and factors such as aspect, seasonal distribution of precipitation and recharge and average snow cover. Annual average air temperature is controlled by latitude, regional climatic factors (proximity to sea and warm/cold ocean currents) and elevation (average air temperature decreases as elevation increases).

Country	Typical ground temperature (°C) 10–150 m depth range
Norway	2–7
Finland	2–6
Denmark	6–11
Poland	8–11
United Kingdom	9–14
France	9–15
Romania	12–16
Italy	10–15
Spain	15–19
Greece	14–20

typically 9–13°C, in Norway 4–7°C, and in parts of Russia, it can be close to 0°C (Table 3.2 and Figure 3.6). Indeed, in parts of northern and eastern Siberia, the annual average air temperature is well below zero, and so is the subsurface temperature – in fact, the ground freezes, forming permafrost. In parts of Greece, on the other hand, ground temperatures above 20°C may be common (Katsoyiannis *et al.*, 2007). In Chapter 16, we will also see how the presence of cities can affect subsurface temperatures.

3.4 Insolation and atmospheric radiation

We have seen that the temperature of the ground is largely controlled by the annual average soil temperature and thus, in turn, by annual average air temperature. In other words, solar and atmospheric energies are important components in the heat budget of a ground sourced thermal energy scheme. Let's take a closer look at these energy sources.

Insolation describes the (dominantly shortwave) solar radiation arriving at the earth's surface. The maximum solar irradiance, incident on a plane perpendicular to the sun (i.e. directly facing the sun), just outside the earth's atmosphere, is estimated to be 1366 W m^{-2}. This would be the theoretical maximum insolation on a perfectly clear day, at noon at a location on the equator, at the equinox, neglecting any effects of atmospheric absorption, reflection or backscattering!

If the incoming solar irradiance on the earth's cross section is $1366\ \mathrm{W\ m^{-2}} \times \pi r_e^2$, where r_e is the radius of the earth (6.37×10^6 m), the total incoming radiation on the side of the earth facing the sun is 1.74×10^{17} W. This amount, averaged over the earth's

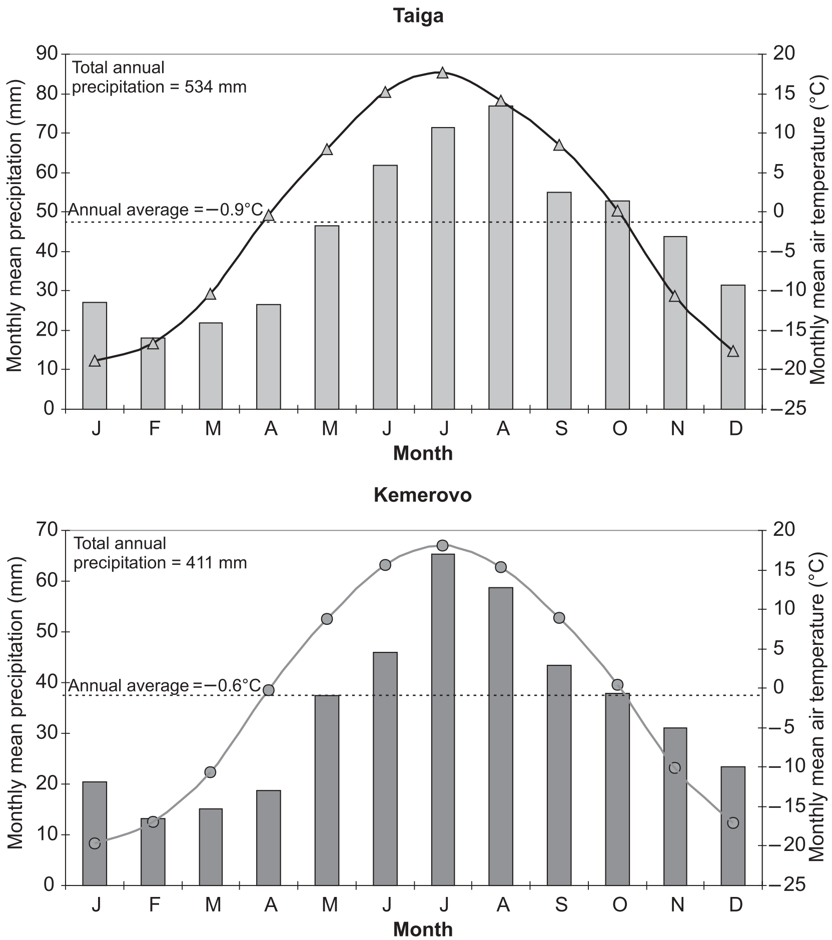


Figure 3.6 Typical annual precipitation histogram and monthly mean temperature profile for the southern Siberian town of Taiga (near Tomsk) and the city of Kemerovo, Russian Federation (based on data cited by Belyanin et al., 1969).

entire surface area ($4\pi r_e^2$), gives a mean insolation (outside the atmosphere) of 342 W m^{-2}. Of course, the actual amount of insolation at a given location will depend on latitude and time of day.

However, solar radiation will be absorbed and backscattered during its passage through the earth's atmosphere. Thus, the shortwave insolation at the earth's surface (q_{sw}) will be somewhat less (around half, as a global average) than that outside the atmosphere. In fact, Connelly (2005) estimated annual mean insolation rates of around 300 W m^{-2} in tropical regions, around 200 W m^{-2} in temperate zones, and as little as 100 W m^{-2} in northern Eurasia and North America. Moreover, the actual instantaneous insolation rate varies from season to season and throughout the day (Table 3.3). Only

Table 3.3 Average monthly and annual rates of insolation at selected European locations, after data provided by Whitlock *et al.* (2000). Figures are cited as $\mathrm{kWh\,m^{-2}\,day^{-1}}$ (and $W m^{-2}$).

	December	July	Average annual
Oslo	0.19 (*8*)	4.84 (*202*)–June	2.27 (*95*)
Edinburgh	0.32 (*13*)	4.34 (*181*)–June	2.26 (*94*)
London	0.52 (*22*)	4.74 (*198*)	2.61 (*109*)
Athens	1.63 (*68*)	7.61 (*317*)	4.56 (*190*)
Malaga	2.14 (*89*)	7.64 (*318*)	5.16 (*215*)

a relatively small proportion (<6%) of the incoming short-wave radiation is harvested by plants to drive photosynthesis (Linacre and Geerts, 1997).

We must also remember that the atmosphere is warmed up by absorption of incoming and backscattered short-wave solar radiation (and by radiation reflected and re-radiated by the earth's surface – see below). According to Stefan–Boltzmann's relation (Section 3.2.3), the atmosphere and clouds re-radiate this thermal energy as long-wave (infrared) radiation. This long-wave *downwelling* atmospheric radiation (q_{lw}) must be added to the short-wave solar radiation to give the total incoming radiation incident on the earth's surface. Formulae to estimate the long-wave radiation include

$$q_{\mathrm{lw}} = \varepsilon_{\mathrm{a}}\sigma\left(\theta_{\mathrm{a}}^{\mathrm{o}}\right)^4 \mathrm{\ W\ m^{-2}} \text{ (Kapetsky and Nath, 1997)} \tag{3.6}$$

where ε_{a} is the atmospheric emissivity, σ is the Stefan–Boltzmann constant, $\theta_{\mathrm{a}}^{\mathrm{o}}$ is the absolute effective atmospheric temperature (K). The difficulty of defining what this problematic last term actually represents can be circumvented by applying the more practical Monteith (1973) equation for clear-sky long-wave downwelling, as cited by Linacre (1992)

$$q_{\mathrm{lw}} = 208 + 6\theta_{\mathrm{sc}} \mathrm{\ W\ m^{-2}} \tag{3.7}$$

and θ_{sc} is the mean daily screen temperature in degrees Celsius. For a 15°C screen temperature, we can see that a q_{lw} of around $300\,\mathrm{W\,m^{-2}}$ might be typical. Atmospheric long-wave radiation increases with cloud cover, and Linacre (1992) suggested that his calculated flux should be multiplied by a factor ($1 + 0.0034\mathrm{Cl}^2$) where Cl is cloud coverage in oktas (eights of the sky).

It appears that atmospheric long-wave radiation is typically of a similar order of magnitude to short-wave insolation, but we must also remember that the ground itself re-radiates long-wave 'back radiation' (q_{back}) in relation to its temperature (Linacre, 1992; Hostetler, 1995):

$$q_{\mathrm{back}} = \varepsilon\sigma\left(\theta_{\mathrm{sur}}^{\mathrm{o}}\right)^4 \mathrm{\ W\ m^{-2}} \tag{3.8}$$

where ε = surface emissivity (≈0.97), σ = Stefan–Boltzmann constant and $\theta_{\mathrm{sur}}^{\mathrm{o}}$ = surface temperature in Kelvin. The back radiation is typically similar in magnitude to the incoming long-wave atmospheric radiation: a surface temperature of 12°C yields a q_{back} of 363 W m^{-2}. However, the *net* long-wave radiation budget ($q_{\text{back}} - q_{\text{lw}}$) is typically of the order of 20–90 W m^{-2} *from* the ground, for a screen temperature of 15°C and depending on cloudiness (Linacre, 1992).

We can now define a radiation budget for the earth's surface. The net incoming radiation (R_{n}) is given by

$$R_{\text{n}} = (1 - \alpha_{\text{sw}})q_{\text{sw}} + (1 - \alpha_{\text{lw}})q_{\text{lw}} - q_{\text{back}} \tag{3.9}$$

where α_{sw} and α_{lw} are the short-wave and long-wave albedos (reflectivity) of the earth's surface. The latter is usually very small. Linacre and Geerts (1997) have constructed this radiation budget for an entire clear day in a meadow in Oregon, USA (44°N). The short-wave insolation (q_{sw}) is estimated as 336 W m^{-2}, of which 24% (81 W m^{-2}) is reflected from the meadow's surface. Incoming atmospheric radiation (q_{lw}) is 289 W m^{-2}, and outgoing back radiation (q_{back}) from the ground is 376 W m^{-2}. Thus, the net incoming radiation (R_{n}) is 168 W m^{-2}. (Note that, during night-time, there would be no incoming short-wave radiation and there would be a net loss of long-wave radiation, leading to a cooling of the ground.) The annual mean net incoming radiation typically exceeds 100 W m^{-2} in the tropics, with 40–80 W m^{-2} being typical for temperate Europe and <40 W m^{-2} for northern climes (Linacre and Geerts, 1997).

What becomes of the incoming net radiation? It heats up the ground and is ultimately lost by evapotranspirative heat losses (q_{evap}) of latent heat in water vapour or convective losses (q_{conv}) of sensible heat (i.e. the ground heats up the adjacent air). We can say (neglecting any geothermal heat flux) that

$$R_{\text{n}} = q_{\text{evap}} + q_{\text{conv}} + \Delta G \tag{3.10}$$

where ΔG is change in stored heat in the ground per square metre (which should be negligible in the long term) *or* removed by other means (e.g. maybe a ground source heating scheme?).

Oh, and just to make it really complicated, we also need to remember that heat or 'coolth' may be transferred to the ground by advection, during the infiltration of rainfall or melting snow.

We can begin to see that the magnitude of net incoming radiation is at least two orders of magnitude greater than the typical geothermal heat flux (<0.1 W m^{-2}). We can thus argue that ground source heat systems are thus not truly geothermal energy systems; they merely utilise the earth's surface as a huge solar collector and storage. We can also begin to understand why so many of the design criteria (which we will meet in later chapters) boil down to specific energy extraction rates of the order of 5–20 W m^{-2} of the earth's surface – this approximates to the amount of useful solar and atmospheric radiation we can 'harvest' with a ground source heat scheme (and remember that part of this incoming radiation is reflected from the earth's surface or re-radiated by it) without unacceptably cooling it.

In fact, we could construct an energy budget for the ground–atmosphere interface in an area where we are planning a ground source heat scheme (as we will do for a lake in Chapter 12), where the components would include short-wave insolation, long-wave atmospheric radiation, long-wave back radiation, reflection, evapotranspiration, heat brought in by rainfall and convective heat transfer. We can intuit that a ground source heating scheme that lowers the temperature of the ground, albeit slightly, will tip the energy budget in favour of heat transfer from the atmosphere to the ground (e.g. by decreasing q_{back} and also by impacting convective and evaporative heat transfer).

Wow! That was complicated – particularly as we must remember that there are lots of feedback loops active between the various heat transfer processes (e.g. heat re-radiated by the earth may be re-absorbed by the earth's atmosphere; heat consumed in evapotranspiration will be re-released to the atmosphere as water vapour condenses again). If you want to read more on the complex nature of the earth's surface radiation and heat budget, try the books by Monteith (1973), Henderson-Sellers (1984) and Linacre (1992). Before we move on to examine the impact of truly geothermal heat, conducted from the earth's interior, on subsurface temperatures (Section 3.6), let's briefly consider how cyclical changes in soil or air temperature are propagated into the subsurface.

3.5 Cyclical temperature signals in the ground

Williams and Gold (1976) gave an excellent description of how we can calculate the response of ground temperatures to daily or seasonal temperature cycles. They stated that

$$\theta(z,t) = \bar{\theta} + A \cdot \exp\left(-z\sqrt{\frac{\pi}{\alpha t_o}}\right)\cos\left(\frac{2\pi t}{t_o} - z\sqrt{\frac{\pi}{\alpha t_o}}\right) \tag{3.11}$$

where
$\theta(z,t)$ is the temperature at depth z and time t;

α = thermal diffusivity = λ/S_{VC} (see Boxes 3.5 and 3.7);
$\bar{\theta}$ = the average ground temperature;
A = the amplitude of the temperature variation at the ground surface (maximum – average). If we are talking about seasonal variations, this is called the annual soil temperature *swing*;
t_o = the duration of a full cycle of temperature (24 h for a diurnal cycle, 1 year for a seasonal cycle).

In this equation, the initial *exponential* term governs the decay of the temperature swing with depth, the first half of the *cosine* term provides the cyclicity and the

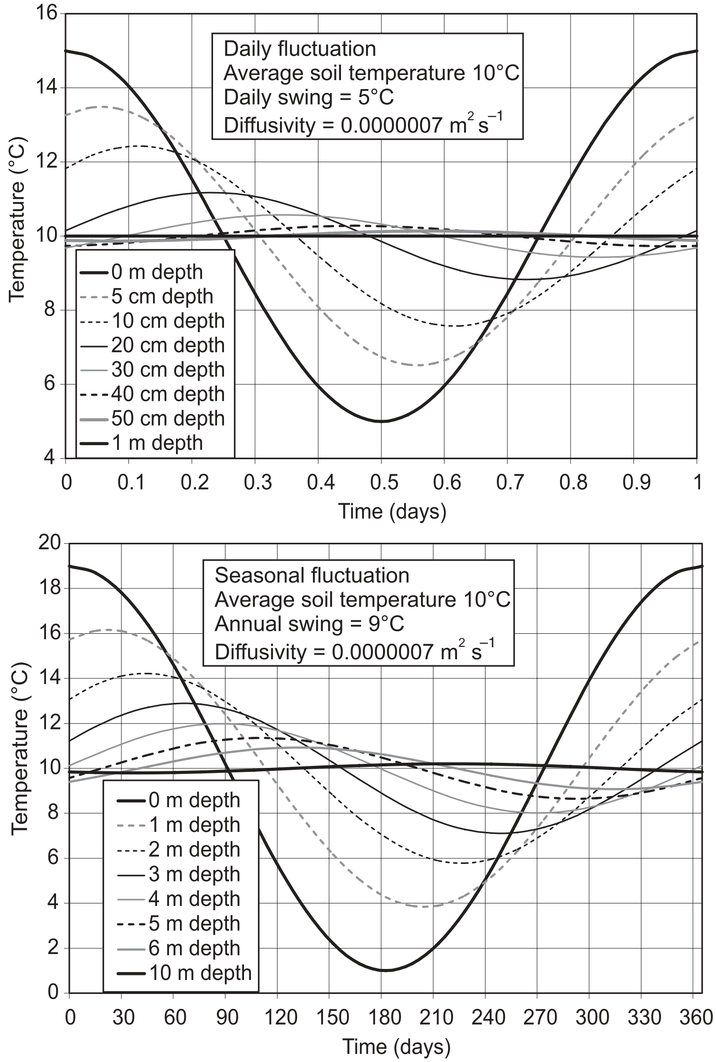


Figure 3.7 (Top) The daily variation in soil temperature caused by a daily surface temperature swing of 5°C at the top of the soil column, with a diffusivity of around $7 \times 10^{-7}\,m^2 s^{-1}$, calculated according to Equation 3.11. (Bottom) The calculated annual variation in subsoil temperature caused by a seasonal soil temperature swing of 9°C, with the same diffusivity of around $7 \times 10^{-7}\,m^2 s^{-1}$.

second half of the *cosine* term provides the increasing time lag with depth. We can try it out by assuming that a soil has a thermal conductivity $1.6\,W\,m^{-1}\,K^{-1}$ and a volumetric heat capacity of $2.3\,MJ\,K^{-1}\,m^{-3}$. It thus has a diffusivity of $7 \times 10^{-7}\,m^2 s^{-1}$. Figure 3.7 shows the results, demonstrating that diurnal fluctuations extend to around 40 cm (5°C daily swing assumed, around a mean of 10°C), while seasonal fluctuations (9°C annual swing) propagate 6–10 m.

3.6 Geothermal gradient

The temperature of the shallow subsurface is thus largely controlled by the annual average air temperature, and the heat that we extract via ground source heating schemes is dominantly ultimately derived from solar energy absorbed by the earth's surface. The earth's surface, in fact, acts as a huge solar collector.

However, there is also a minor component of true geothermal heat flux, derived from the earth's interior and migrating towards the earth's surface. This manifests itself as a geothermal temperature gradient, superimposed on the annual average surface temperature (Figures 3.8 and 3.9a). The earth's geothermal gradient, outside of anomalous or volcanically active areas, is normally in the range of 0.01–0.03°C m^{-1}, or around 1–3°C per 100 m. This represents a geothermal heat flux of some 40–100 mW m^{-2} (Figure 3.10).

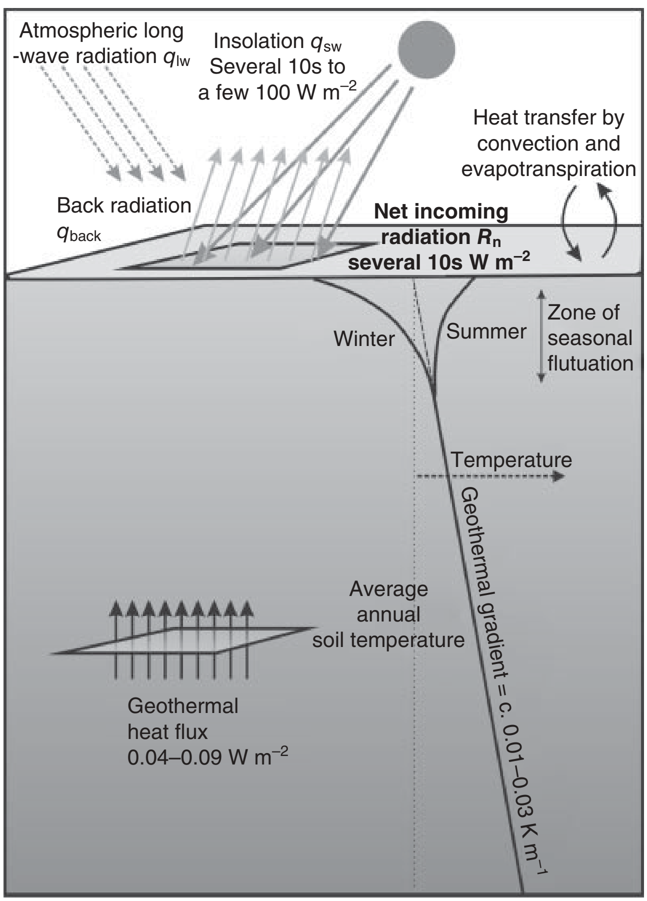


Figure 3.8 Schematic block diagram showing the downward increase in temperature in the earth due to the geothermal gradient, the seasonal zone of fluctuation in temperature and the relative magnitudes of geothermal heat flux and insolation.

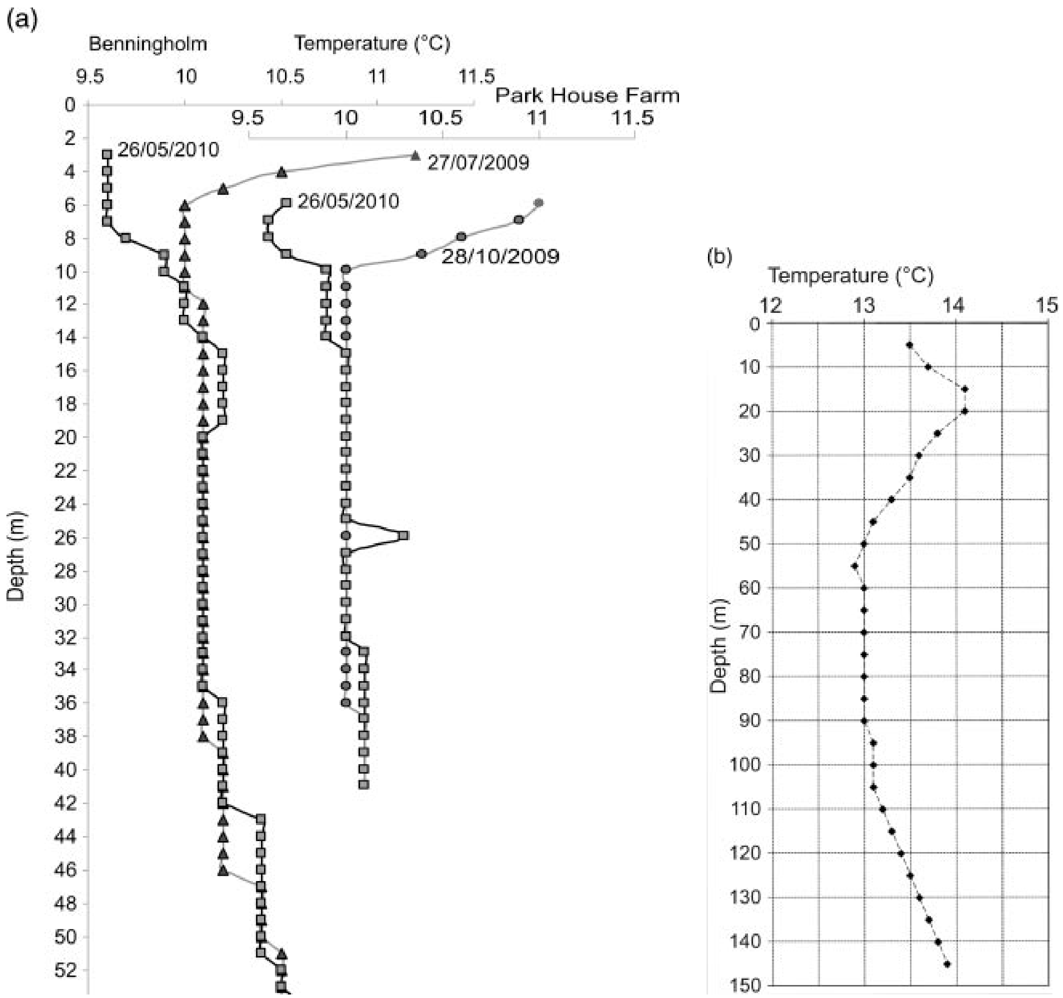


Figure 3.9 (a) 'Normal' temperature profiles from two water wells (Benningholm and Park House Farm) in the Chalk aquifer of East Yorkshire, UK, based on data from Russell (2010). Note that the top 10 m of each profile appear to be affected by seasonal temperature fluctuations (the July and October profiles show a summer 'pulse' of heat entering the top of the column, the May profiles show a residual cold 'pulse' from winter). Below 10 m, the temperature is stable and increases slowly due to the geothermal gradient. (b) A temperature profile from a closed-loop borehole in London, illustrating the effect of the urban environment in enhancing near-surface temperatures and reversing the natural geothermal gradient down to a depth of c. 50 m. *Reproduced by kind permission of Econic Ltd. and Neo-Energy (Sweden) Ltd.*

Figure 3.10 The geothermal heat flux in the United Kingdom, in $\mathrm{mW\,m^{-2}}$. *Reproduced by permission of the British Geological Survey © NERC. All rights reserved. IPR/90-20DR.*

In the United Kingdom, the measured geothermal gradient ranges from about 0.015 to $0.04\,\mathrm{K\,m^{-1}}$, while the average gradient (for onshore areas down to a depth of around 4 km) is cited by Wheildon and Rollin (1986) as $0.02\,\mathrm{K\,m^{-1}}$, and by Busby *et al.* (2011) as $0.028\,\mathrm{K\,m^{-1}}$. Thus, if we drill a borehole to 100 m depth in a part of England where the annual average surface temperature is 10°C and the geothermal gradient is

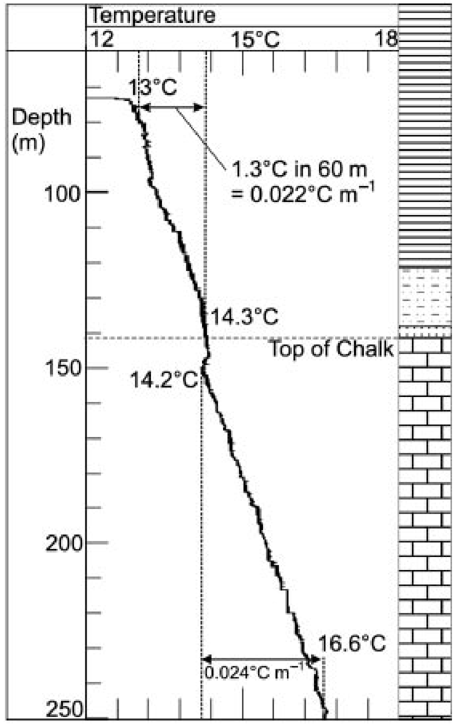


Figure 3.11 A groundwater temperature log of a water well in Surrey, southern England. The yield of the well was very poor (around 0.5 L s^{-1}), so there is very little groundwater throughflow to disturb the geothermal gradient. The borehole passes through 141 m of dominantly Tertiary clays and silts (and occasional sands) before encountering the Chalk at 141 m. The geothermal gradient is 0.024°C m^{-1} in the Chalk and 0.022°C m^{-1} in the Tertiary section. *Public domain information, provided by and reproduced with the permission of the Environment Agency of England and Wales (Thames Region).*

0.025 K m^{-1}, we would expect the temperature at the base of the borehole to be around 12.5°C, with an average temperature over the depth of the borehole of 11.25°C (see Figure 3.11). If the annual average soil temperature is θ_0, we can estimate the ground temperature θ_z at any depth z by Fourier's law:

$$\theta_z = \theta_0 + z\frac{d\theta}{dz} = \theta_0 + z\frac{Q}{A\lambda} \tag{3.12}$$

where

Q is the geothermal heat flux per unit area A (W m^{-2});

λ is the thermal conductivity of the ground (W m^{-1} K^{-1});

$d\theta/dz$ is the geothermal temperature (θ) gradient with depth (z).

The British Geological Survey has compiled maps of geothermal heat flux in $mW\,m^{-2}$ for the whole of the United Kingdom (Figure 3.10). Note particularly the high geothermal heat fluxes in Cornwall and Devon, where granites with a high content of radioactive elements outcrop or subcrop at relatively shallow depths. Note also the high fluxes in the Weardale-Lake District zone (west of Newcastle), where granite also occurs at shallow depth (see Box 2.4).

The mean geothermal heat flux measured in the United Kingdom is calculated to be $69 \pm 28\,mW\,m^{-2}$ (comparable with the mean for continental Europe of $64\,mW\,m^{-2}$). This value is believed by Wheildon and Rollin (1986) to be something of an overestimate, due to bias in the measurements towards geological formations with high heat flow. An attempt to compensate the data set for bias has resulted in a revised estimate of mean geothermal heat flux in the United Kingdom of $54 \pm 12\,mW\,m^{-2}$.

We can use Fourier's law to relate geothermal heat flux to the geothermal gradient and estimate the average thermal conductivity (λ) of the subsurface:

$$\lambda = \frac{Q}{A \cdot (d\theta/dz)} \tag{3.13}$$

If a typical geothermal gradient is $0.025\,K\,m^{-1}$ and a typical heat flux is $0.06\,W\,m^{-2}$, this yields a thermal conductivity of around $2.4\,W\,m^{-1}\,K^{-1}$, corresponding well with figures for the more lithified lithologies in Table 3.1. Thermal conductivities of some of the main British geological units are shown in Table 3.4.

3.7 Human sources of heat in the ground

We have seen above that, under natural conditions, the geothermal heat flux leaking up from the Earth's interior is manifested as a geothermal temperature gradient, with temperatures increasing slowly with depth. In Chapter 16, we will discuss some of the factors that can cause anomalies or perturbations in this gradient. Geologists are, however, occasionally surprised when they drill a borehole and encounter ground temperatures that *decline* with increasing depth – that is an apparently *reversed* geothermal gradient. A reversed gradient down to depths of 10–15 m may, of course, purely be a natural seasonal phenomenon: the input of a summer pulse of heat causes the topmost layer of ground to heat up, leading to a temporarily reversed temperature gradient persisting well into the autumn (see Figure 3.9a). Sometimes, however, the gradient is reversed to depths of many tens of metres and cannot simply be explained by natural seasonal effects. In fact, such a deep, reversed gradient is typically encountered in city environments (Figure 3.9b) and has been reported from Sweden (Hellström, 2011a); Osaka, Japan (Taniguchi and Uemura, 2005); Ireland (Allen *et al.*, 2003; Goodman *et al.*, 2004); Gateshead and London, UK (Banks *et al.*, 2009a; Figure 3.9b) and Winnipeg, Canada (Ferguson and Woodbury, 2004).

Table 3.4 Thermal conductivity data for selected lithologies in the United Kingdom, based on laboratory measurements made on samples extracted from boreholes. The thermal conductivity data are believed to represent water-saturated samples. Abstracted from Rollin (1987), based on data from [1]Bloomer (1981), and with additional measurements made by [2]Midttømme *et al.* (1998). Considerable discrepancy will be seen between Bloomer's and Midttømme *et al.*'s results, highlighting the difficulty of representatively determining the conductivity of mudstones, reproducing the saturation conditions and the necessity of considering anisotropy. [3]Determined perpendicular to bedding. [4]Determined parallel to bedding. Reproduced by permission of the British Geological Survey © NERC. All rights reserved. IPR/90-20DR. N = number of determinations.

Formation	Lithology	N	Mean thermal conductivity ($W\,m^{-1}\,K^{-1}$) ± standard error on mean
London Clay (Palaeogene)	Sandy mudstone[1]	5	2.45 ± 0.07
	Claystone with 30% silt and 10% sand[2]		0.83–0.84[3] 0.94–1.19[4]
Reading Beds (Palaeogene)	Sandy mudstone[1]	4	2.33 ± 0.04
	Mudstone[1]	10	1.63 ± 0.11
Chalk (Cretaceous)	Limestone	41	1.79 ± 0.54
Upper Greensand (Cretaceous)	Sandstone	18	2.66 ± 0.19
Gault (Cretaceous)	Sandy mudstone	32	2.32 ± 0.04
	Mudstone	4	1.67 ± 0.11
Kimmeridge Clay (Jurassic)	Mudstone[1]	58	1.51 ± 0.09
	Silty claystone[2]		1.07–1.21[4]
Oxford Clay (Jurassic)	Mudstone	27	1.56 ± 0.09
	Clay/siltstone[2]		1.11–1.29[4]
Mercia Mudstone (Triassic)	Mudstone	225	1.88 ± 0.03
Sherwood Sandstone (Permo-Triassic)	Sandstone	64	3.41 ± 0.09
Magnesian Limestone (Permian)	Dolomitic limestone	12	3.32 ± 0.17
Westphalian (Coal Measures)	Sandstone	37	3.31 ± 0.62
	Siltstone	12	2.22 ± 0.29
	Mudstone	25	1.49 ± 0.41
	Coal	8	0.31 ± 0.08
Namurian (Millstone Grit)	Sandstone	7	3.75 ± 0.16
Lower Carboniferous Limestone	Limestone	14	3.14 ± 0.13
Upper Old Red Sandstone (Devonian)	Sandstone	27	3.26 ± 0.11
Silurian slates near Selkirk	Slate	67	3.33 ± 0.05
Hercynian granites	Granite	895	3.30 ± 0.18
Basalt	Basalt	17	1.80 ± 0.11

Moreover, as we will see in Chapter 16, Ferguson and Woodbury (2004) mapped groundwater temperatures beneath the city of Winnipeg, Canada (Figure 16.1) and found that temperatures below the urban area were some 2–3°C higher, and in some cases 5°C higher, than the normal natural background temperature of the aquifer (typically less than +6°C).

In urban areas, we are thus forced to recognise that human activity has the potential to warm up the ground to considerable depths. This is partly due to the 'urban heat island' effect – local climatic warming caused by changes in reflectivity, emissivity and absorption properties of surfaces in the urban environment, together with artificial heat emissions to the atmosphere from buildings and vehicles (Linacre and Geerts, 1997; Watkins *et al*., 2002; Kolokotroni *et al*., 2006; Mayor of London, 2006). A more direct source of urban ground warming is simply due to the fact that buildings and factories leak heat through their floors and cellars as well as through their walls and roofs, as do some 'warm' buried services (railway tunnels, sewers, cables, etc.). This heat is conducted away downwards into the ground. Ferguson and Woodbury (2004) and Banks *et al*. (2009a) have made calculations of the type shown in Box 3.7 to demonstrate that an increase in soil temperatures due to urbanisation can propagate down through the geological column to result in ground warming and a reversal of the natural geothermal gradient, down to depths of 50–100 m, or more, over the course of a century or so.

BOX 3.7 An Example of Thermal Diffusivity

We have seen that thermal diffusivity (see Box 3.5) is the rate at which a heat signal is propagated into a medium. Let us, for the moment, ignore the geothermal heat flux and the geothermal gradient (see Section 3.6), and assume that the ground is at a constant temperature, which roughly corresponds with the annual average soil and air temperature – say 10°C. Let us now assume that, for some reason, whether it be climate change (Kharseh, 2011; Kharseh *et al*., 2011) or local warming due to urbanisation (Banks *et al*., 2009a), the air and soil increase suddenly in temperature by 3°C, to 13°C.

We consider the ground to be of infinite depth and constant initial temperature (10°C) – in technical terms, it is a semi-infinite solid, with the following *boundary conditions*:

1. Initially, at time $t = 0$, the ground temperature $\theta(z,t = 0) = 10$°C for all values of depth (z).
2. For all values of $t > 0$, the surface temperature $\theta(z = 0,t) = 13$°C.

Because the air and soil are now warmer than the ground, heat diffuses by conduction into the ground. It can be shown (Beltrami *et al*., 2005) that the temperature at any depth (z) and any time (t) is given by

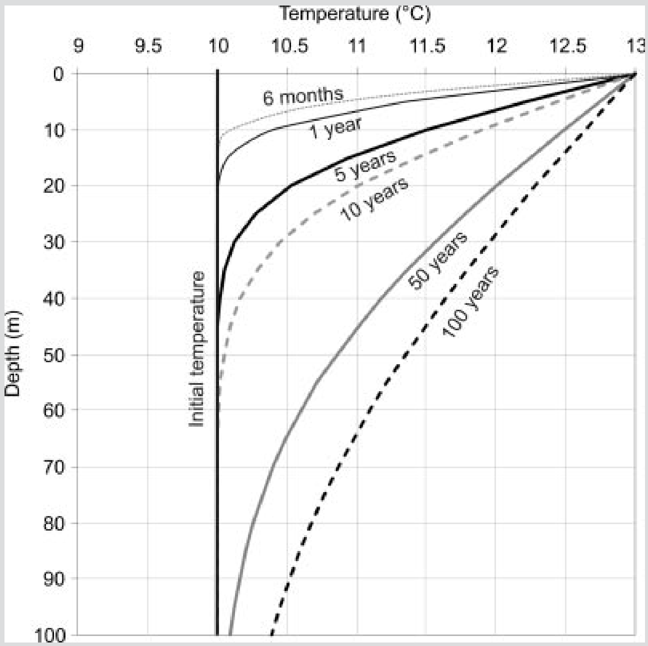
$$\Delta\theta(z,t) = \Delta\theta_0 erfc\left(\frac{z}{2\sqrt{\alpha.t}}\right)$$

where *erfc* is the complementary error function, $\Delta\theta(z,t)$ is the change in temperature at depth z and time t and $\Delta\theta_0$ is the initial change in temperature at the surface (in our example, 3°C).

(*Continued*)

BOX 3.7 (*Continued*)

Thus, if the ground has a thermal conductivity of $1.6\,W\,m^{-1}\,K^{-1}$ and a volumetric heat capacity of $2.3\,MJ\,K^{-1}\,m^{-3}$, it has a diffusivity of $7 \times 10^{-7}\,m^2\,s^{-1}$. We can calculate how the temperature signal propagates into the ground over a period of 6 months, 1 year, 10 years, 50 years and 100 years – see the figure below:



The distance z travelled into a medium by a pulse of heat during time t is proportional to $\sqrt{\alpha t}$, and a rough approximation is given by the formula (Kharseh, 2011; Kharseh *et al.*, 2011):

$$z = \pi\sqrt{\alpha t}$$

Let us return to our example, with a diffusivity of $7 \times 10^{-7}\,m^2\,s^{-1}$. During a warm day (12 h = 43 200 s), one might expect a pulse of heat (diurnal temperature signal) to penetrate 0.55 m (55 cm). During a summer (182 day = 1.57×10^7 s), one might expect a pulse of heat (seasonal temperature signal) to penetrate 10.4 m. Looks about right – particularly if we examine Figure 3.7. Remember that this only gives an approximate answer – the heat front is gradational, not sharp.

Eco-pessimists will regard this anthropogenic warming of the urban subsurface as yet another example of humankind's thermal pollution of the planet. An optimist might choose to regard it as an example of the unwitting creation of a new geothermal resource. The waste heat that has leaked out of urban buildings has not been irretrievably lost: it has been *stored* in the ground and can be re-extracted for space heating, using heat pump technology.

Before leaving this topic, we should note that downward heat leakage from buildings is very familiar to residents of towns built on permafrost. Floors of heated houses that leak heat downwards can cause the underlying permafrost to melt and the ground to subside. Permafrost dwellers thus need to ensure that their house floors are either very well insulated or artificially frozen to prevent the ground from melting (see Section 7.7). Alternatively, as in the case of the town of Longyearbyen on Spitsbergen, they build their houses on pillars or stilts, so that the warm floor of the house is not in direct contact with the permafrost.

3.8 Geochemical energy

So far, we have seen that the temperature of the shallow subsurface of the ground is controlled by the annual average air and soil temperature (solar energy), and modified by a geothermal temperature gradient (geothermal energy). In most cases, these are the two main natural energy sources that we need to consider in a ground source heat budget. In some special cases, it may be appropriate to consider internal radiogenic heat production (see Section 2.4): there may also be a component of geochemical energy affecting the heat budget of the subsurface.

Some minerals weather very rapidly in the presence of air and moisture. One such group of minerals is the sulphide minerals, such as pyrite (FeS_2), marcasite (FeS_2), sphalerite (ZnS) and galena (PbS). These minerals occur in many types of metal ore deposit and are commonly found in and around coal deposits. When these minerals are exposed to air and to water (in underground mines or in mine wastes deposited at the surface), they oxidise and release a cocktail of acid, dissolved metals and sulphate that is commonly known as acid mine drainage:

$$\mathrm{ZnS} + 2\mathrm{O}_2 = \mathrm{Zn}^{2+} + \mathrm{SO}_4^{2-} \tag{3.14}$$

$$\text{Sphalerite} + \text{oxygen} = \text{Dissolved metal} + \text{sulphate}$$

$$2\mathrm{FeS}_2 + 2\mathrm{H}_2\mathrm{O} + 7\mathrm{O}_2 = 2\mathrm{Fe}^{2+} + 4\mathrm{SO}_4^{2-} + 4\mathrm{H}^+ \tag{3.15}$$

$$\text{Pyrite} + \text{water} + \text{oxygen} = \text{Dissolved metal} + \text{sulphate} + \text{acid}$$

This is a major potential environmental pollution issue in many mining areas. It can also present a hazard, as these sulphide oxidation reactions are typically exothermic; that is, heat energy is released during oxidation. Thus, mine spoil tips can be very hot inside, sometimes in excess of 50°C. In coal stores or coal mine waste tips, the heat released by pyrite and marcasite oxidation can even lead to spontaneous combustion of the wastes. In Norwegian metals mines, sulphide oxidation can lead to tropical down-mine temperatures, and the phenomenon of *kisbrann* ('sulphide fire') was well known. Indeed, each mole of pyrite oxidised by Equation 3.15 releases 1400–1500 $kJ\,mol^{-1}$ of heat energy, while the sphalerite reaction (Equation 3.14) releases over 1700 $kJ\,mol^{-1}$ (Banks *et al.*, 2004a).

Banks *et al.* (2004a) have speculated that, if we could extract this geochemical energy from mines or mine waste dumps (maybe using heat pumps: Chapter 4) and use it for space heating, it would be an elegant means of converting an environmental liability into a sustainable energy resource.

Other exothermic reactions releasing potentially usable geochemical energy include organic degradation reactions such as might take place in landfills or manure heaps or other accumulations of organic waste. These, too, could be conceived as sources of usable energy. Indeed, Sæther *et al.* (1992) documented a Norwegian landfill, beneath and down-gradient of which groundwater temperatures are 1–2°C warmer than normal.

3.9 The heat energy budget of our subsurface reservoir

Let us consider a block of the subsurface as a heat reservoir, or *aestifer* (Figure 3.12). There will be natural components contributing to the heat budget of the block. We can refer to these as the 'boundary conditions' of our conceptual model.

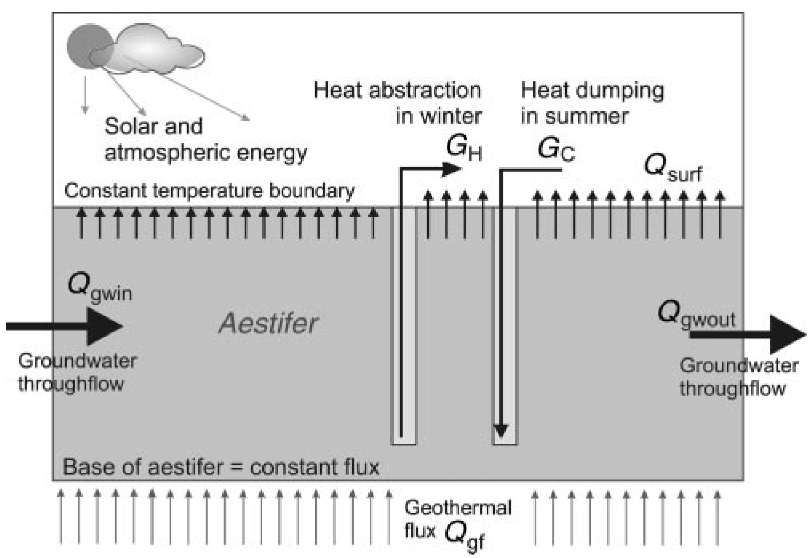


Figure 3.12 A block of the subsurface (an *aestifer*) showing possible elements of its heat budget.

1. There will be heat entering the aestifer from the geothermal heat flux. We can regard this as a constant flux. In other words, the base of our aestifer can be conceptualised as a constant heat flux boundary.
2. There may be groundwater flow passing through our aestifer, carrying with it a load of heat. In many cases, this groundwater flow can be regarded as constant over the long term, and we may be able to regard the heat being advected into the model (from the left in Figure 3.12) as another type of constant water flux/constant water temperature boundary.
3. The aestifer may be gaining or losing heat from the ground surface. Under natural conditions, there will, in the long term, usually be a net loss from the surface equal to the geothermal heat flux. If we start extracting heat from the ground, however, we will cool down the aestifer and may start to induce a flux of atmospheric (solar) energy from the surface into the ground. The long-term direction and magnitude of the heat energy flux to/from the surface depends on the magnitude of the temperature difference between the ground and the annual average surface temperature. The annual average surface temperature, at the ground–atmosphere interface, can be regarded as approximately constant in the long term. We can thus describe the top of our aestifer as a constant temperature boundary.

If there is no geochemical (or significant internal radiogenic) component to the heat budget, then we now need to consider the amount of heat that we are planning to extract from the ground (in a ground source heating scheme) or reject to the ground (in a cooling scheme). Of course, throughout a year, we may extract heat (G_H) in winter and dump it (G_C) to the ground in summer. If $G_H > G_C$ over the course of a year, we refer to our ground source heat scheme as a 'net heating' scheme in the long term, and there will be a tendency for the ground to cool down. If $G_C > G_H$, we are dealing with a 'net cooling' scheme, imparting a tendency for the ground to heat up. In some particularly happy circumstances, the heat rejected in summer balances the heat extracted in winter, resulting in minimal disturbance to the long-term heat budget of the aestifer and thus to its temperature.

The final component of the heat budget to consider is the heat stored in the ground. An increase in the heat stored in the ground is represented by ($V_{aest} \times \Delta\theta \times S_{VC}$), where V_{aest} is the volume of aestifer under consideration, $\Delta\theta$ is the average temperature change of the aestifer and S_{VC} is its specific volumetric heat capacity. Under natural conditions, over the long term,

$$Q_{gwout} + Q_{surf} = Q_{gwin} + Q_{gf} \tag{3.16}$$

An equilibrium is presumed to exist and the ground temperature does not change with time. However, if we start to interfere and extract ground source heat to warm up an office block (a net heating scheme)

$$G_H + Q_{gwout} + Q_{surf} > G_C + Q_{gwin} + Q_{gf} \tag{3.17}$$

then the heat stored in the aestifer ($V_{aest} \times \Delta\theta \times S_{VC}$) will be depleted and the temperature will fall. Hopefully, this fall in temperature will induce the following:

- Q_{surf} to decline and eventually become negative; that is, a flux of (ultimately solar) heat is induced from the surface. Q_{surf} becomes more negative in response to a greater temperature gradient between surface and subsurface in accordance with Fourier's law.
- The temperature of exiting groundwater to fall and thus Q_{gwout} to decrease.

In this way, we would hope that, in response to our extraction of ground source heat, a new equilibrium condition would ultimately be established, such that subsurface temperatures would eventually stabilise at a new, lower (but still acceptable) level. We can get it wrong, however! If G_H is too great, the temperature might continue falling until we start to freeze the ground. This might be undesirable for many reasons: geotechnical, environmental or operational.

Thus, ground source heat extraction can be genuinely sustainable, if we have a good understanding of our aestifer's heat budget. However, it is not automatically sustainable: overoptimistic ground source heat schemes may have a finite operational lifetime. (Of course, we may deliberately design our scheme with a finite lifetime, if we are dealing with a temporary construction or event that needs to be heated.)

Conversely, if we are dealing with a net cooling scheme

$$G_H + Q_{gwout} + Q_{surf} < G_C + Q_{gwin} + Q_{gf} \tag{3.18}$$

the ground will begin to heat up, Q_{surf} (i.e. heat loss from subsurface to atmosphere) will ultimately increase and Q_{gwout} may increase. Eventually, a new equilibrium may establish itself at a higher temperature and our heat rejection (net cooling) operation can be thought of as 'sustainable'. Here, if we get it wrong, temperatures in the ground may increase beyond the design limits of our scheme or our heat pump. The scheme may become progressively more inefficient and eventually inoperable.

3.10 Cyclical storage of heat

We have seen, at the start of this chapter, that the ability to store heat is hugely empowering in terms of energy strategy. Thermal storage allows us to 'buy' or produce heat at times when it is freely available or cheap and to use it at times when it is scarce or expensive.

Regular international conferences (such as the recent *EcoStock* conference in Stockholm in 2009) on the topic of thermal storage attract hundreds of delegates, all with new ideas or types of medium for storing heat. Phase change materials (PCMs) are special types of chemical (see Section 7.2.1) that have been developed to efficiently

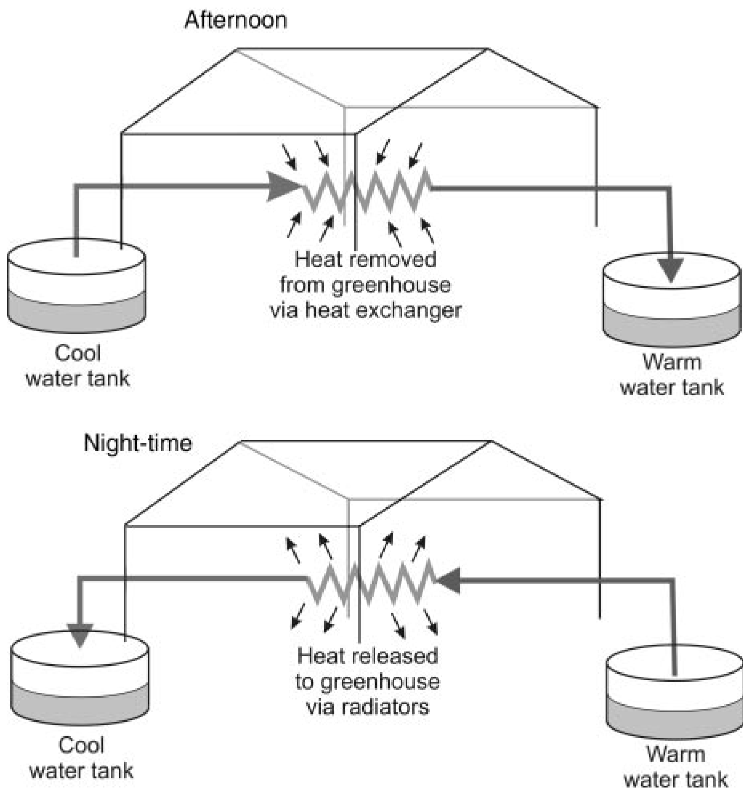


Figure 3.13 Diurnally reversible thermal energy storage at a greenhouse complex (see Section 3.10).

store heat or cold as latent heat (Box 3.3) when they change, for example, from a solid to a liquid.

Water has a very high specific heat capacity (see Section 3.1) and is thus a particularly simple and flexible medium for storing heat. Indeed, a domestic hot water cylinder that is heated by electricity at night-time (cheap electricity) and used during the morning is an example of the diurnal manipulation of heat storage.

Some industries also have cyclically varying demands for heating and cooling. A good example is the horticulture industry (Figure 3.13). On a late spring day, a greenhouse may start to overheat in the afternoon leading to a requirement for cooling. Transpiration by plants may also cause dangerously high humidity levels that could allow rot or fungal infections to thrive, especially if condensation starts to occur. 'Throw open the windows!' you might say. Unfortunately, in many modern greenhouses, this is not so simple. The CO_2 levels in the greenhouse may be carefully controlled and the owner may not want this valuable gas (for plants) to disappear through the window; moreover, the owner may not wish to risk pests entering the greenhouse from outside. In other words, the greenhouse has a need for cooling and dehumidification. This could be achieved simply by storing a large tank of cool water and circulating it via heat exchangers and dehumidification coils through the greenhouse space. As the cool water absorbs the sensible heat from the greenhouse (and the latent heat from dehumidification), it heats up. We could then store this warm water in a second tank adjacent to the greenhouse. At night-time, however, temperatures could drop to such an extent that heating is required. Fantastic! We still have our tank of warm water, storing the daytime solar heat, that can be circulated back through the greenhouse's heat exchangers (which now act as radiators) to warm the growing space. As the warm water circulates, it loses heat and cools down and can be returned to the first, cool water tank.

Many modern greenhouses really do operate in this way, cyclically storing and shunting heat and 'coolth' between large water tanks. They may also use heat pumps (Chapter 4), rather than passive heat exchange, to enhance the heat transfer between water and air. They may also employ CHP units, to generate some heat, electricity and CO_2 (a nutrient for plants, which is added as a supplement to the greenhouse's atmosphere).

In this example, we have considered the diurnal storage of heat at a greenhouse complex on a spring day. But what about seasonal storage of heat? Greenhouses will have an overwhelming demand for heat in winter months and for cooling and dehumidification in summer. Can we build tanks of water large enough to store all the surplus heat from the summer to the winter season? Probably not – they would be massive! But, we could drill boreholes into the ground and use the enormous volumes of rock and groundwater beneath our feet to store heat from season to season. This concept is termed *underground thermal energy storage* (UTES). The underlying principle is not a new one, as Figure 3.14 illustrates!

3.11 Manipulating the ground heat reservoir

In Section 3.9, we saw that the ground is very good at storing natural geothermal and solar heat that we can extract and use. In doing so, we cool the ground down and induce an increased flow of heat from the atmosphere and from the surrounding rock and soil, and hopefully arrive at a new, cooler, equilibrium situation. If we have a very intensive use of heat, however, we may run into a sustainability problem. The ground may cool down faster than natural sources can replenish it.

Let's do a brief calculation. Let's assume that a new housing development, measuring 100 m × 100 m (1 ha) is built on a rock of volumetric heat capacity 2.2 MJ m^{-3} K^{-1}. If we drill heat exchange boreholes to a depth of 100 m, we have effectively accessed a thermal 'store' of 10^6 m^3, with a thermal capacity of 2.2 TJ K^{-1}. If the housing development has a heating demand of 277 GJ $year^{-1}$ (equivalent to a peak demand of 35 kW for 2200 equivalent hours per year), we can see that annual extraction of heat is modest, compared with the thermal storage. We would need to extract heat for some 8 years before the average cooling of the ground would be expected to be 1°C

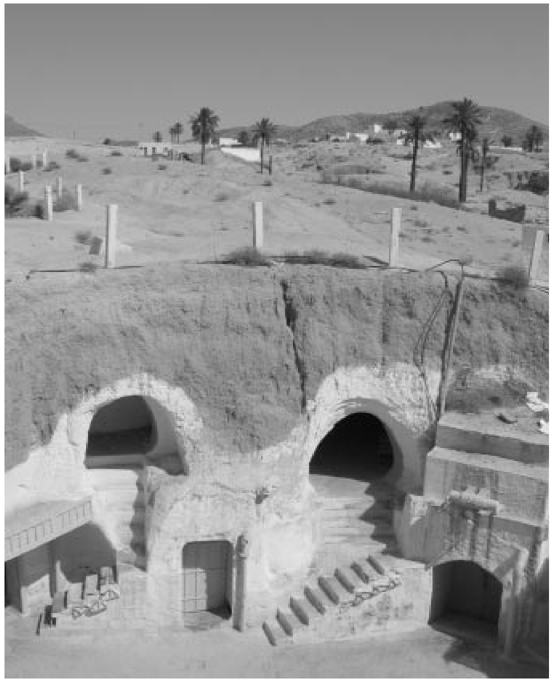


Figure 3.14 An underground house at Matmata, in the Tunisian semi-desert, which was featured in the film 'Star Wars'. The subsurface is substantially cooler than the fierce summer daytime temperatures on the surface. The Jedi certainly knew how to use the earth as a thermal store, but unfortunately Mr Skywalker was unavailable for comment. *Photo by Dave Banks.*

(2200 GJ/277 GJ year^{-1}), even without considering replenishment of the heat reservoir by transfer from the atmosphere and surrounding rocks. Thus, there is a fighting chance that the heat extraction might at least approach sustainability (we would need to check this by some more mathematics and computer modelling). If the housing development was several stories high and had a much high heat demand of, say 1.2 TJ year^{-1}, we can see that we would risk cooling down the rock thermal store by 1°C every 2 years and the operation is unlikely to be sustainable. We could get around this problem of sustainability by artificially replenishing the ground with heat during the summer, when heat is plentiful and when there is no demand for space heating in the housing. The replenished heat could be re-extracted in the winter. Possible sources of surplus heat that could be 'injected' to the ground include the following:

- Solar energy. We could collect heat in glazed or unglazed solar collectors (or other atmospheric heat exchangers such as dry coolers), mounted on roofs, and inject it to the ground.
- Cooling of the living area itself (i.e. air conditioning, removal of waste heat from the houses).

- Adjacent offices (with computers and electronic equipment that produce unwanted heat) or industries that produce surplus heat.
- Power stations, waste incinerators or CHP plants that produce more heat than can be directly utilised in summer.

If we can reintroduce this surplus summer heat into the ground at a rate approaching 1.2 TJ year^{-1}, we have re-attained a thermal balance. The heat can be extracted during winter and replenished again in summer. We have constructed a *UTES* system that is potentially indefinitely sustainable.

Stockholm's Arlanda Airport (Box 3.8) has developed a type of UTES system termed *aquifer thermal energy storage* (ATES). The system here is directly analogous to the greenhouse thermal storage system described in Section 3.10 except that, instead of using large tanks of water to store heat on a diurnal cycle, the airport uses huge natural groundwater basins (aquifers) to store heat seasonally, over an annual cycle. Another, slightly older, example of an *ATES* system can be found at Oslo's Gardermoen Airport (Box 14.2).

BOX 3.8 Passive Heating and Cooling at Arlanda Airport, Stockholm

The geology adjacent to Arlanda Airport is especially fortuitous. A very porous and permeable glaciofluvial (*esker*) deposit of sand and gravel was deposited in the geologically recent (Quaternary) glacial period in a basin formed by the ancient underlying crystalline bedrock. The glaciofluvial aquifer is divided into two 'compartments' by a naturally occurring ridge in the bedrock. The glaciofluvial sands and gravels contain groundwater, which, in Sweden, is naturally rather cool.

During summer, cool groundwater from one half of the aquifer is pumped to a large plate heat exchanger (Figure 7.9b) in a plant room. The opposite side of the plate heat exchanger is coupled to a fluid circuit that extracts heat from the terminal buildings (i.e. providing air conditioning). The waste heat from the airport buildings is thus transferred to the cold groundwater (or, if you prefer, the coolth of the groundwater is transferred to the airport terminal). The groundwater thus warms up and is re-injected to the other half of the aquifer at a temperature of some 15–20°C, by recharge wells. The recharge temperature can be increased still further by circulating the building loop fluid through buried collectors below gate aprons (which are used for ice melting in winter) to collect solar energy.

During winter, the recharge wells become abstraction wells (Figure 3.15) and the warm groundwater is pumped up again from the aquifer. The warm groundwater flows to the plate heat exchanger in the plant room, where the heat is transferred to the building circuit. The heat is used to preheat ventilation air to the terminal buildings and also to melt snow and ice on the aprons at the aircraft gates (by circulating warm fluid in buried collectors below the aprons). The cold groundwater (at around +3–5°C) is now recharged to the 'cool' side of the aquifer, for reuse the following summer.

BOX 3.8 (*Continued*)

Thus, the system has created a cold half and a warm half of the aquifer, which are used on a seasonal basis. Eleven wells, of depth between 15 and 30 m, are employed (five cold and six warm), each equipped with both submersible pumps and recharge mains. The total groundwater pumping capacity is around 720 $m^3 h^{-1}$, delivering up to 8 MW peak heating and cooling effect and up to 20 GWh annually (Andersson, 2009; Wigstrand, 2009). What is remarkable about the Arlanda system is that it does not require the use of heat pumps. All the heating and cooling is achieved 'passively', merely using natural temperature gradients. The only electrical power input is used to drive pumps in the wells and circulation pumps. The system is thus highly efficient (system seasonal performance factor of around 60). The capital investment in the scheme was around 5 million Euros, with an anticipated straight payback time of around 5 years.

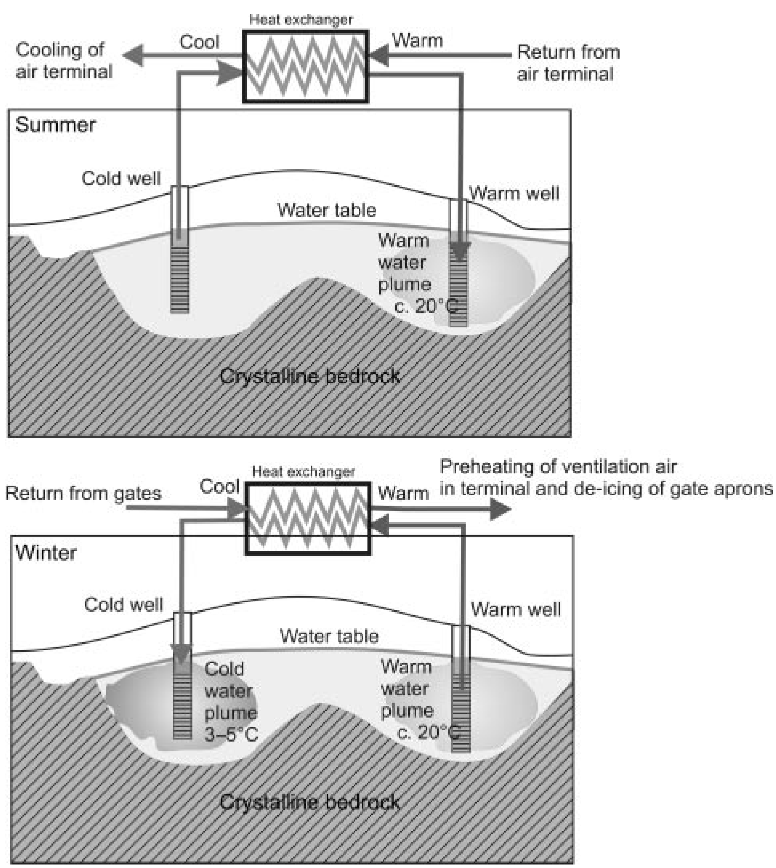


Figure 3.15 Seasonally reversible aquifer thermal energy storage at Arlanda Airport, Stockholm (schematic only; see Section 3.11 and Box 3.8).

STUDY QUESTIONS

3.1 A type of rock called dolerite has a vertical thermal conductivity of $1.8\,W\,m^{-1}\,K^{-1}$. The geothermal heat flux is $64\,mW\,m^{-2}$ and the annual average soil temperature is 9.8°C. What ground temperature do you predict at 150 m depth, assuming that geothermal heat flow is purely conductive and can be assumed to be one-dimensional and steady state?

3.2 A tank of $20\,m^3$ water-saturated sand of porosity 18% has an initial temperature of 10°C. How much heat must be extracted from the tank to freeze it? If this freezing is performed over 14 days, what is the rate of heat extraction? The saturated sand has a volumetric heat capacity of $2.5\,kJ\,L^{-1}\,K^{-1}$.

4

What Is a Heat Pump?

> Heat won't pass from a cooler to a hotter.
> You can try it if you like but you'd far better not-a
> . . . (that's entropy, man!)
>
> Michael Flanders and Donald Swann

A lot of heat is stored away in the earth's subsurface, even at normal temperatures and shallow depths. We have seen, in Chapter 3, that in the United Kingdom, the rocks, sediments and groundwater beneath our feet are typically at temperatures of 9–13°C down to depths of around 100 m (i.e. within the range of most drilling rigs). But how can we use this energy at such a low temperature? How can we heat our homes using a medium with a temperature of 12°C, when most of us enjoy a room temperature in the region of 20°C? The simple answer is – we cannot, using natural temperature gradients alone. Try as we might, we cannot get heat to flow naturally from rocks at 12°C to a living room at 20°C.

We can, however, envisage other scenarios. Let us imagine that we live in Scandinavia, where the temperature drops below 0°C for some months of the year and where there may be persistent heavy snowfall. There is thus often a need for de-icing pavements, roads or parking surfaces. In Scandinavia, the subsurface temperature of rocks and groundwater may be in the region of 4–7°C (a little higher than the annual average air temperature – Box 3.6). We could conceive of drilling a small well and pumping up groundwater at, say, 6°C in order to circulate it in a network of pipes beneath the

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

pavement to keep it ice free. We are working with a favourable temperature gradient – heat flows happily from 6 to 0°C! This is termed 'free' heating or passive heating. Box 3.8 shows how groundwater-based passive heating has been used to melt snow and ice and to preheat ventilation air at Stockholm's Arlanda Airport.

In the same way, we could imagine circulating cool groundwater around a large British office building in the summertime, through a network of pipes, beams and panels. The office owner wants to keep staff at a comfortable 20°C. If the network of panels and beams and so on is large and efficient enough, we can envisage that heat will flow naturally from the office interior, be absorbed by the circulating groundwater (at 12°C) and be carried away, keeping the building cool. This is termed passive cooling or 'free' cooling, and is one of the most environmentally friendly forms of space cooling available (Figure 4.1 and Box 3.8).

But the natural temperature of the subsurface and groundwater in the United Kingdom (10–13°C) places limits on the use of passive heating and cooling. Even with ideally efficient heat exchangers in the building, we cannot use these techniques to heat a fluid or a space above c.13°C or cool it below 10°C. Although 'high-temperature'

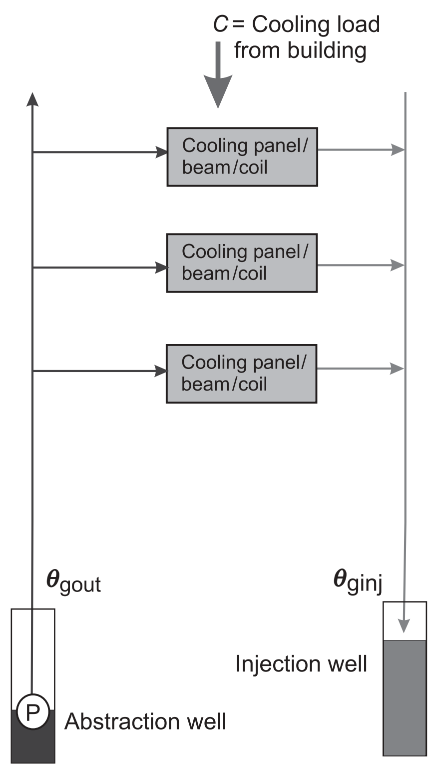


Figure 4.1 Using groundwater to perform 'passive' cooling in a British office block during the summer. θ_{gout} is the temperature of the abstracted groundwater (maybe 11°C) and θ_{ginj} is that of the waste groundwater (maybe 14°C), which in this diagram is re-injected to the aquifer via an injection well.

building cooling systems are available that operate on fluid flow temperatures of 12°C (and may thus be amenable to passive cooling), many conventional building cooling systems are based on a fluid flow temperature of 6–7°C and return temperature (after the system has absorbed heat from the building) of c. 12°C.

How can we overcome these limitations? Answer: we pump heat from a lower temperature to a higher temperature using a device called a heat pump. But before we consider the heat pump, let us look at its opposite: the heat engine.

4.1 Engines

One can define an engine as a device that converts potential energy to work (Boxes 1.1 and 4.1). To do work, we need a potential energy gradient. This may be a chemical energy gradient, a gravitational potential energy gradient, a head gradient or a temperature gradient. Let us take the example of head of water. If we have two reservoirs of water with a difference in head between them (Box 1.2), we can allow water to flow from a higher head to a lower head and perform work, perhaps by employing a water wheel or hydraulic ram. Similarly, if we have two heat reservoirs at different temperatures, we can perform mechanical work. The net effect is a flow of heat from the high-temperature source to a low-temperature exhaust.

A steam engine is a heat engine: the high-temperature source is steam at a temperature in excess of 100°C, while the low-temperature exhaust is often a water condensate. Internal combustion engines are also heat engines: the source is a hot gas (from fuel combustion), from which the heat is degraded (while doing mechanical work) to a lower-temperature exhaust. Stirling engines (Box 4.2) are another type of heat engine, which can operate on very low temperature differentials.

Sadi Carnot (1796–1832) studied the theory of ideal heat engines and was able to demonstrate that, for such an engine, the work performed was determined by the

BOX 4.1 Kilowatts and Kilowatt-Hours

A kilowatt is a unit of power: a rate of energy generation, transfer or consumption. One kilowatt (1 kW) is equal to 1000 joules per second (1000 J s^{-1}).

A kilowatt-hour (kWh) is a unit of energy. It is the amount of energy generated/consumed by an appliance rated at 1 kW left running for 1 h. In other words,

$$1\ \text{kWh} = 1000\ \text{J s}^{-1} \times 3600\ \text{s} = 3\,600\,000\ \text{J or } 3.6\ \text{MJ}$$

Similarly,

$$1\ \text{MWh} = 3.6\ \text{GJ}$$

$$1\ \text{GWh} = 3.6\ \text{TJ and so on}$$

BOX 4.2 The Stirling Engine

We have already seen (Section 4.1) that a heat engine exploits the flow of heat from a high-temperature source to a low-temperature sink. It converts some of this heat to mechanical work. The most familiar forms of heat engine include the steam engine and the internal combustion engine, where the heat sources are steam and hot combustion gases, respectively.

However, work can theoretically be extracted from any temperature gradient, however small. The Stirling engine is a device that is particularly attractive to the renewable energy sector, as it requires only a relatively low-temperature heat source. Indeed, it allows solar energy, geothermal heat or waste heat from industrial processes to be converted to useful work. It was pioneered in 1816 by Reverend Robert Stirling (1790–1878), a Scottish theologian from Perthshire. By 1843, Robert and his brother James had installed a Stirling engine to power the machinery at a Dundee iron foundry.

In its simplest form, the Stirling engine comprises a 'hot' chamber and a 'cold' chamber, containing a gaseous working fluid (e.g. air) and connected by a regenerator. The hot chamber is thermally coupled, via some form of heat exchanger, to an external heat source. The cold chamber is coupled, via another heat exchanger, to a cool external sink. The regenerator is typically a form of internal heat exchanger and thermal store which allows the gas to pass unhindered, but which efficiently retains a temperature gradient along its length, thus maintaining the temperature difference between the two chambers. A system of pistons shunts the working gas alternately from the cold chamber to the hot chamber. The phase difference between the pistons ensures a series of expansion (heating) and contraction (cooling) cycles, which drive a flywheel.

Modern Stirling engines can begin to approach an idealised Carnot efficiency. Dr James Senft has shown that it is even possible to construct Stirling engines that run on temperature differentials of less than 1 K (Kongtragool and Wongwises, 2003).

difference in temperature between the source/inlet (at θ_1) and the exhaust (at θ_2). Lord Kelvin, in 1851, further demonstrated that the maximum efficiency (E_{max} – often referred to as the *Carnot efficiency*) of an ideal heat engine, defined as the ratio between work delivered (W) and heat input at high temperature to the engine (H_{in}), could be described by the following formula (Sumner, 1948):

$$E_{max} = \frac{W}{H_{in}} = \frac{(\theta_1 - \theta_2)}{\theta_1} \tag{4.1}$$

Thus, to take Sumner's (1948) example: if we have a heat source at temperature θ_1 = 82°C (= 355 K) and a heat sink at 27°C (= 300 K), the maximum efficiency of an ideal heat engine would be $55\,K/355\,K = 0.15$ or 15%. In other words, no more than 15% of the heat flow could be converted to useful work. Eighty-five per cent of the original heat input would be discharged as exhaust heat. The concept of combined heat and power (CHP) encompasses a heat engine that provides mechanical work or electricity as an output, but where the exhaust heat is also captured and utilised (for example) for space heating (Box 4.3).

BOX 4.3 Combined Heat and Power

Many conventional heat engines (car motors, turbines at power plants, nuclear reactors) *waste* energy because they make no use of the exhaust heat. They use valuable fuels with a high concentration of useful energy (often referred to as 'exergy') and yet Equation 4.1 tells us that only a small proportion of this can be converted to useful work (mechanical output to drive turbines to produce electricity). The remainder is discharged as waste exhaust heat. Craziness! The combined heat and power (CHP) concept essentially says that we should install heat exchangers on the exhausts of heat engines, so that exhaust heat can be transferred to a secondary fluid to perform useful heating.

Thus, some greenhouse complexes have CHP generators, which produce (1) electricity for greenhouse lighting, (2) a surplus of electricity that is sold to the national grid, (3) hot water that is used for heating the greenhouses in winter *and* (4) carbon dioxide (from fuel combustion), which is introduced to the greenhouse atmosphere as a plant nutrient. Many countries, such as the United Kingdom, tend to keep their waste incinerators and power stations at a good distance from their dwellings – that is, in the middle of the countryside. In Scandinavia, however, it is possible to find urban waste incinerators that generate electricity, dispose of rubbish and produce hot water for domestic heating. In Russia and Eastern Europe, fossil fuel power stations are often located in cities and waste heat from them is piped to apartments as a form of district heating. Both of these systems are a large-scale form of CHP.

The one major problem with CHP is that people still need electricity in the summer but have little need for heat. The CHP plant risks producing heat in the summer that there is no demand for. There are two ways around this. (1) Absorption heat pumps (see Box 4.5) can be installed, which use the surplus heat to perform *cooling*! In this way, a CHP system can be used to produce heat in winter, electricity all year and cooling in summer. This is termed 'trigeneration'. Such a CHP can support a network of district heating and district cooling pipes of the type used in Southampton (see Box 2.3). Alternatively, (2) the surplus heat can be 'stored' until winter, when it can be used. A huge thermal store is needed to efficiently retain this waste heat, however. The ground or groundwater beneath our feet provides exactly such a possibility: the heat can be introduced via ground heat exchangers (see Chapter 10) or as hot water via wells (see Chapter 8) in summer and extracted again in winter (see Chapter 14).

4.2 Pumps

We have established that there is a close analogy between hydrogeology and thermogeology. That groundwater exists deep in the earth has long been known, but it has not always been accessible. Prior to 1863, the villagers of the English Chiltern village of Stoke Row (Banks, 2008) had to walk to distant springs and rivers to collect their household water, despite the fact that abundant groundwater existed in the Chalk aquifer deep beneath their feet. They had not the money to reach it, however, until the Maharajah of Benares (modern Varanasi, India) visited the village and was so shocked at their plight that he paid for a 113-m-deep well to be sunk to access the water table (Figure 4.2). The villagers used a hand-powered pump to bring the water to the surface: in reality, this was a pair of buckets on ropes, which presumably required considerable muscle power to operate. Nevertheless, the well and its 'pump' allowed the villagers to access a new resource, bringing it from a previously inaccessible depth (low *head*) to the surface where it was accessible.

For ancient miners, groundwater was a curse rather than a benefit (Younger, 2004). It flooded their mines, wet their socks and dampened their lunchtime snap boxes.

Figure 4.2 The Maharajah's Well at Stoke Row in the English Chiltern Hills (*photo by Dave Banks*).

More importantly, it prevented them from mining to deep levels, accessing the reserves of coal and metals that were presumed to exist. Miners had to make do with tedious low-capacity hand-powered or animal-powered pumps (or in some cases, water-powered pumps) until a certain Thomas Newcomen (1663–1729) developed a steam-powered, atmospheric condensing piston engine for pumping water, in the period between 1705 and 1725. This was truly one of the world's great inventions and one of the earliest efficient 'machines' (at least in our modern understanding of the term), predating James Watt's steam engine by over 40 years and revolutionising the mining industry (Figure 4.3a).

Nowadays, industrialised European nations seldom use hand-powered or steam-powered pumps to raise water from a low head to a high head. We tend to use petrol- or diesel-powered pumps or, most commonly, electrical pumps to do this job (Figure 4.3b).

A pump is thus the opposite of an engine. In a water engine, water flows down a head gradient allowing mechanical work to be 'extracted' from the system. In a water pump, electrical/mechanical work is done to move water up the head gradient (against its natural tendency), that is, from a locus of low head (low elevation or low pressure) to a locus of high head (high elevation or pressure). It allows us to transfer water from a location where it is no use to anyone (deep in an aquifer) to a useable elevation (the surface or, better still, a water tower).

If we wish to quantify the performance of a pump, we can examine the manufacturer's *pump curve* (see Chapter 9), which relates the quantity of water pumped to the mechanical/electrical power input and to the head difference up which the water flow is pumped. As a general rule, the greater the head difference over which the pump operates, the lower the flow rate *or* the greater the amount of power required by the pump.

4.3 Heat pumps

A trivial, but wholly correct, definition of a heat pump is that it is 'a device that pumps heat'. With a water pump, we can expend energy and perform mechanical work to get water to flow uphill or from low pressure to high pressure. With a heat pump, we can persuade heat to flow from a low-temperature environment to a high-temperature one – but we must perform mechanical work and expend energy to do so! A heat pump is the opposite of a heat engine. A heat pump raises the temperature of the available heat from an unusable level to a usable one (Box 4.4).

We are all familiar with water pumps, but we have conceptual difficulties with heat pumps. Nevertheless, despite our theoretical misgivings, we all happily put our trust in them. Most readers will have invested in one already – it is called a fridge. Our domestic refrigerator pumps heat from a low-temperature environment (our chilled salad compartment) into our kitchen. This will be evident if you put on some protective rubber gloves and venture – carefully – into the hidden world of lost sausages and

(a)

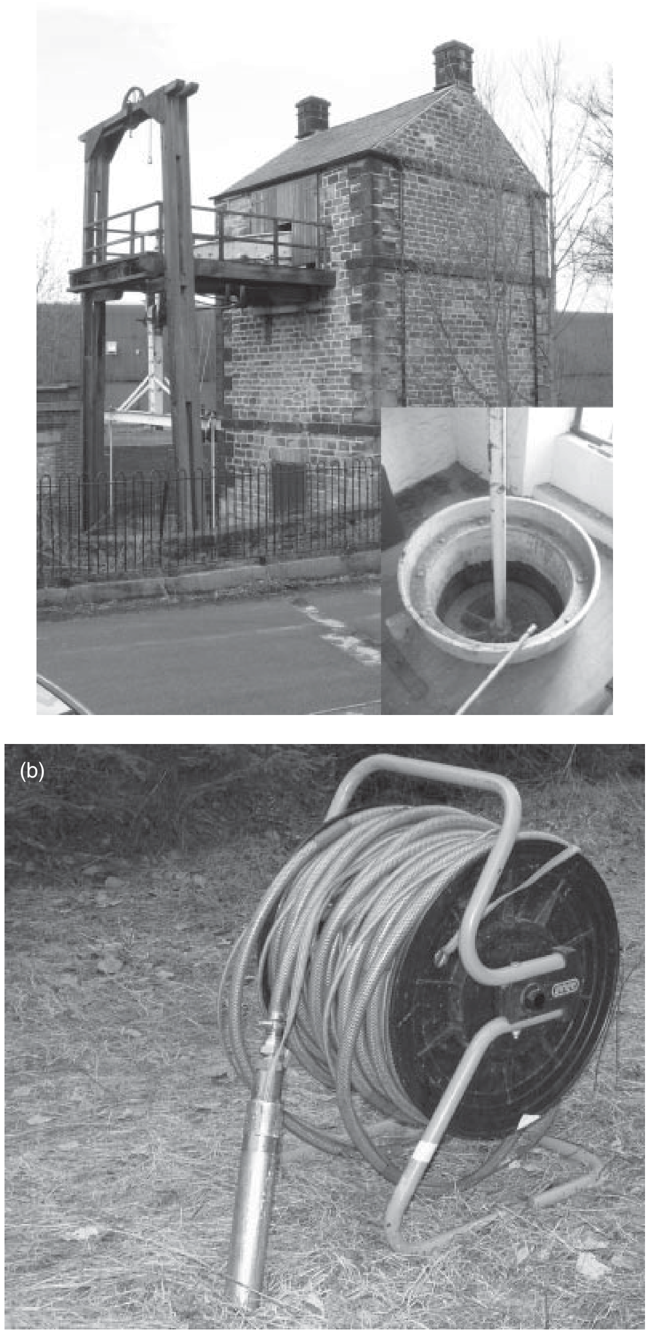


Figure 4.3 (a) The last *in situ* example of a Newcomen Beam Engine, at Elsecar Colliery, South Yorkshire, England; the inset shows the cylinder and piston (*photos by Dave Banks*). The device is both an engine and pump: the potential energy locked in coal, and the heat it produces, is converted to mechanical work. This work is used to raise water from a low head (i.e. low potential energy: deep in the mine) to high head (a drainage adit near the surface). (b) A small modern electrical submersible pump (*reproduced with the permission of Bjørn Frengstad*).

BOX 4.4 The Heat Pump at the Eco-Centre, Tyneside, Northern England

Figure 4.5 shows a schematic example of a real application of the type discussed in Section 4.9, installed at the Eco-Centre in Tyneside, northern England. Here, groundwater is pumped at a rate of around $3\,L\,s^{-1}$ from a 60-m-deep well drilled into Carboniferous sandstone strata. The temperature of the groundwater is around 10°C. The heat pump extracts some 63 kW of heat, leaving a chilled groundwater at c. 5°C to be rejected into the Tyne estuary.

The heat pump itself delivers a nominal 88 kW of heat effect. It runs on a refrigerant cycle utilising R407c (a mixture of the fluorinated hydrocarbons CH_2F_2, CHF_2CF_3 and CH_2FCF_3), powered by twin reciprocating compressors running off a 415-V, three-phase electricity supply. As the groundwater is saline (around 25 000 ppm), the heat pump is designed for marine use: the evaporator is of stainless steel and is corrosion resistant.

In a neat twist, the Eco-Centre has a large wind turbine, which generates electricity for sale to the National Grid. The Centre then repurchases electricity on a night-time tariff to power the heat pump: the National Grid is effectively being used as a 'buffer' for the wind-generated electricity. Thus, the heat pump could be regarded as being run by a green electricity source – a truly zero-carbon space-heating solution.

We have seen that ground source heat pumps are at their most efficient when they deliver heat to a low-temperature central heating system at a constant rate over prolonged periods. The Eco-Centre's heat pump delivers warm water at 45°C to an underfloor central heating system flowing at around $2.1\,L\,s^{-1}$. The system's design return temperature is 35°C. The building has a high thermal mass (i.e. it takes a long time to heat up or cool down in response to heat inputs). Thus, the heat pump is typically run during winter on cheap electricity at night. The building is thus warm when office hours commence, and the accumulated heat is retained throughout the day.

The ratio of heat delivered to electricity consumed (the coefficient of performance) was designed to be around 3.5. Accumulated experiences over the lifetime of the heat pump suggest that the actual figure is in the region of 3. One might expect a slightly better figure from a 'state-of-the-art system', but we should remember that the Eco-Centre's system was one of the earlier installations in Britain, being commissioned in 1996.

stray herrings behind your fridge. It is warm back there! In fact, there is most likely a metal radiator grid on the back of the fridge pumping heat from the fridge interior out into the kitchen.

But any heat pump requires an energy input, and the fridge is no exception. Most domestic fridges require an input of some few hundred watts of electrical energy that performs mechanical work by powering a compressor (Figure 4.4).

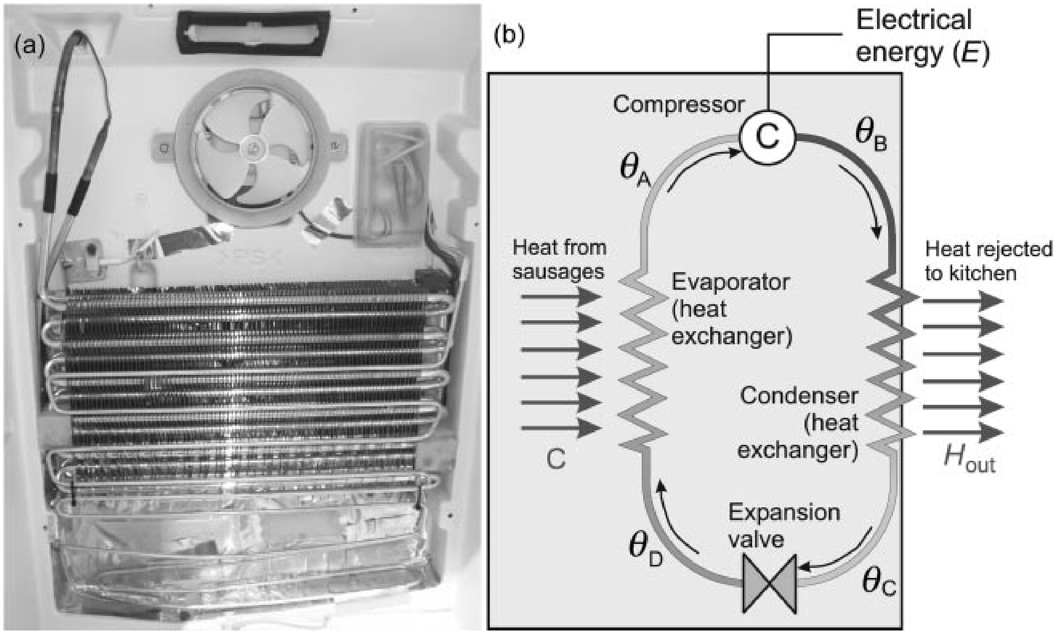


Figure 4.4 (a) The evaporator of a modern fridge, exposed by removing the fascia to reveal refrigerant pipes and heat exchange fins. (b) A schematic diagram showing how your refrigerator works.

If we wish to quantify the performance of a heat pump, we can examine the manufacturer's *specification and curves*, which relate the quantity of heat pumped to the electrical power input and to the temperature difference 'up' which the heat is pumped. As a general rule, the greater the temperature difference across which the heat pump operates, the smaller the quantity of heat pumped *or* the greater the amount of power required by the heat pump. In other words, the greater the temperature differential, the less efficient the heat pump will be.

4.4 The rude mechanics of the heat pump

In this book, we will not dwell too long on exactly how a heat pump works, but it is best to have at least a conceptual understanding. Most heat pumps (including your fridge) transfer heat by means of circulating a refrigerant fluid around a compression–expansion cycle (Figure 4.4b). Consider the inside of your fridge, which should be at around 4°C. There are four parts to the refrigerant cycle:

1. Within a network of pipes in the fridge (formally, a heat exchanger known as the evaporator, Figure 4.4a), the refrigerant fluid is circulating at a low (sub-zero) tem-

perature (θ_D). The refrigerant fluid is chosen such that it boils (under the pressure conditions in the refrigerant circuit) at a temperature below 0°C (i.e. below the target temperature of, say, 4°C in your chiller cabinet). As it boils, it absorbs a large amount of latent heat of vaporisation from the fridge's interior.

2. The refrigerant fluid, now a somewhat warmer vapour at temperature θ_A ($>\theta_D$), then passes through a compressor, powered by the electrical energy input to the fridge. We all recognise that when you compress a gas, the temperature rises – think about pumping up a bicycle tyre – remember how hot the pump can get. The pressurised gas thus emerges from the compressor at a high temperature θ_B ($\theta_B > \theta_R$, where θ_R is room temperature).
3. The refrigerant passes through another heat exchanger (the condenser: the radiator grid on the back of your fridge); heat flows from the refrigerant vapour to your kitchen, and the vapour starts to condense back to a liquid, shedding more *latent heat* as it does so. After passing through the condenser, the resultant, still pressurised, fluid now has a temperature θ_C.
4. The refrigerant completes the cycle by passing through an expansion valve, where the pressure drops abruptly. We should be familiar with the idea (Box 3.4) that expanding fluids tend to cool down – think of the condensation that forms on a butane cylinder as gas expands out of it to power your camping stove, or of how an aerosol container cools a little on use. As the refrigerant fluid passes through the expansion valve, the temperature drops significantly, back to θ_D.

A heat pump, used for heating a building, performs exactly the same cycle, although the pressures and vaporisation/condensation temperatures may differ from those of a fridge. The refrigerant fluid within a heat pump can be of many types: it should, however, be thermally stable, have a suitable specific heat capacity, have a volatility/boiling point tailored to the operating temperature and pressure of the heat pump and be environmentally benign. John Sumner's pioneering Norwich Heat Pump (Sumner, 1948 – see Box 5.3) utilised sulphur dioxide (SO_2), which has a boiling point of –1°C at 22 psi: ideal for use with near-freezing winter river water as an environmental heat source.

The earliest refrigerants used on a large scale were ammonia, sulphur dioxide and chloromethane, but these were either flammable or toxic (Heap, 1979; Table 4.1). The chlorofluorocarbons (CFCs), including Freon (a DuPont trading name), were developed in the late 1920s as low toxicity, non-flammable alternatives and R12 became commonly used in refrigeration equipment in the 1940s. CFCs were common in domestic fridges and heat pumps until relatively recently. In the late 1970s and 1980s, it became increasingly clear that the eventual release of CFCs (following disposal of the fridge) was resulting in destruction of the ozone layer. Nowadays, possible refrigerants include fluorinated hydrocarbons (see Table 4.1), hydrocarbons and ammonia (UNEP, 2003).

Many modern ground source heat pumps (GSHPs), used for producing warm air or central heating water in the range of 35–55°C from a ground temperature in the range

Table 4.1 The properties of some refrigerants (data derived from Calm and Hourahan, 2001). Boiling point is cited at atmospheric pressure; flammability = lower flammability limit (% in air by volume), HET = indicative human exposure threshold/limit in air (ppm), ODP = indicative ozone depletion potential (Freon 11 = 1), GWP = global warming potential integrated over 100 years (CO_2 = 1). Note that the human health thresholds are indicative only and should not be used in any formal risk assessment.

Number	Name		Boiling point (°C)	Flammability (%)	HET (ppm)	ODP	GWP
R11	Trichlorofluoromethane (Freon 11)	CCl_3F	23.7	None	1000	1	4600
R12	Dichlorodifluoromethane (Freon 12)	CCl_2F_2	–29.8	None	1000	0.82	10600
R32	Difluoromethane	CH_2F_2	–51.7	12.7	1000	0	550
R40	Chloromethane	CH_3Cl	–24.2	8.0	50	0.02	16
R125	Pentafluoroethane	CHF_2CF_3	–48.1	None	1000	0	3400
R134a	Tetrafluoroethane	CF_3CH_2F	–26.1	None	1000	0	1300
R143a	Trifluoroethane	CH_3CF_3	–47.2	7.0	1000	0	4300
R290	Propane	C_3H_8	–42.2	2.1	2500	0	≈20
R404a	Mixed refrigerant: 44% R125 + 52% R143a + 4% R134a		–46.6	None	–	0	3800
R407c	Mixed refrigerant: 23% R32 + 25% R125 + 52% R134a		–43.8	None	–	0	1700
R717	Ammonia	NH_3	–33.3	15	25	0	<1
R744	Carbon dioxide	CO_2	–78.4	None	5000	0	1
R764	Sulphur dioxide	SO_2	–10	None	2	0	–

of 0–15°C, typically use fluorinated hydrocarbon (hydrofluorocarbon [HFC]) refrigerants, such as R404a or R407c. Less common heat pumps, used to boost the temperature of already hot water (e.g. geothermal water, or waste water from industrial processes) up towards 90–100°C may use carbon dioxide or ammonia (Thermea, 2008; Star, 2010).

Note that the compressor is a mechanical device. It is commonly powered by electricity, but does not have to be. Indeed, refrigerators and heat pumps can be designed whose compressors are powered by diesel, steam, manual effort or even water power. Of course, if we use fossil fuels to power a heat pump compressor, we must be careful to efficiently collect the exhaust heat from the diesel/petrol motor exhaust (see Box 4.3); otherwise, this will negate the benefits that the heat pump offers.

We may ask, 'What becomes of the electrical (or other) energy used to power the heat pump's compressor?' In the action of the compressor, this electrical energy input is converted, partly to sound energy (the hum of the compressor), but mostly to heat energy, which is absorbed in the refrigerant and must be discharged at the *condenser*. This allows us to answer the tantalising question – is it sensible to leave the door of the fridge open on a hot summer's day to cool your kitchen? Although opening the fridge door might bring some temporary relief, as cool air flows out into the kitchen, the long-term answer is 'no'. The fridge's heat pump will simply be transferring energy from the kitchen in front of the fridge, via the open door, to the kitchen at the back of the fridge. And with each cycle of the heat pump, say, 100 W of electrical energy is being converted to heat energy and being added to the discharged heat load. This tempting practice will steadily lead to your kitchen heating up.

4.5 Absorption heat pumps

Finally, it should be noted that we can even design heat pumps that do not use mechanical compressors, but which use *heat* as their energy source. You may be familiar with the gas-powered fridge, where bottled gas is burned to chill a food cabinet and which is very useful in parts of the world where mains electricity is unreliable or non-existent. This gas fridge is a form of heat pump called an absorption heat pump (Box 4.5). It functions in an analogous cycle to the 'conventional' version, but the compressor is replaced with a chemical absorption reaction and the electrical energy source is replaced with a heat source (e.g. a gas burner). Thus, we can use *heat* to transfer even more heat – to produce either space heating or chilling. This is especially important in the context of 'trigeneration' (Box 4.3) – if we have surplus heat from a hot geothermal well, a CHP plant or an industrial process, we can use it to provide chilling (air conditioning, district cooling) in summer. There are now marketed ground source absorption heat pumps, for space-heating purposes, running on the combustion of mains gas and offering competitive efficiencies (Robur, 2010).

4.6 Heat pumps for space heating

Take a look at Figure 4.4, which shows a schematic diagram of a conventional fridge. A quantity of heat (C, the cooling load, in $\mathrm{J\,s^{-1}}$ or W) is absorbed from the chiller compartment and transferred via a heat pump and an external radiator grid to your kitchen. The electrical power required for the compressor is E and the total rejected heat is H_{out}. We can say (neglecting the small loss of power as acoustic noise – your humming fridge)

$$H_{out} \approx C + E \tag{4.2}$$

The temperature in the fridge is 4°C (although the temperature of the refrigerant in the evaporator will be much colder than this), and the temperature of the rejected heat may be, say, 30–40°C (the temperature of the refrigerant in the evaporator may be hotter than this).

A refrigerator is a heat pump that is extracting heat from the sausages that we place in the chiller compartment, and is using it to heat our kitchen. In fact, we can regard our fridge as a 'sausage-sourced' heat pump! It should now be possible to understand that we can take heat from any source that is thermally coupled to the evaporator, and use it to heat our house. We have already seen that the environment around us

BOX 4.5 The Absorption Heat Pump

The absorption heat pump or refrigerator is not hugely different from a vapour compression–expansion unit. The absorption cycle uses a heat source, rather than mechanical or electrical energy, as its energy input. The compressor is replaced by an 'absorber' – a reservoir of absorbent medium, in which the cycling refrigerant has a high solubility. The 'classic' combination, utilised by Carré in 1858–1859, was water as the absorbent and ammonia as the refrigerant. The ammonia may be mixed with a low-solubility carrier gas, such as hydrogen, which does not take part in the refrigeration process, but which essentially regulates pressure in the system. Other combinations could be used: historically, water (refrigerant) and sulphuric acid (absorbent) were employed, or, in more recent times, water (refrigerant) and lithium bromide (absorbent).

In an ammonia-based system, liquid ammonia volatilises in the evaporator of the heat pump, but the resulting ammonia gas is highly soluble in the water of the absorber. This results in very low partial pressures of ammonia at the evaporator and an increased tendency of the ammonia to volatilise at correspondingly low temperatures.

The ammonia-rich water in the absorbent is then heated by the heat source (steam or a gas flame, or 'waste' heat from another source) in the 'generator' unit to expel the dissolved ammonia from the water in a 'separator', resulting in a high temperature and ammonia pressure at the condenser. The cycle is completed, as in a conventional heat pump, by an expansion valve.

Historically, absorption refrigerators were very important in the large-scale production of ice. However, the toxicity of ammonia, the improvement of vapour compression machines and the increasingly reliable supply of electricity has led to a decrease in their usage. Their application may still be favoured, however, in remote regions where electricity is unreliable or unavailable. They are also attractive where low evaporator temperatures are desired or where surplus/waste heat (e.g. from industry, true geothermal wells) is available to drive the absorption–distillation unit. Their use in CHP trigeneration schemes has been mentioned in Box 4.3.

contains huge reserves of low-grade heat. By using a heat pump, we can take heat from sewage, from rivers or from the sea (Zogg, 2008). By circulating outside air over the evaporator, we can extract heat from the atmosphere and use it to heat our home. This is the principle of an *air-sourced heat pump*. By somehow coupling the ground to the evaporator, we can also extract heat from the geological environment – a *ground source heat pump*, or GSHP (Figure 4.5).

Let us assume that we are able to extract a flux of heat (Q_{env}) from some environmental reservoir with a temperature θ_{env}: this may be the ambient outdoor air on a cool spring day or a flow of groundwater pumped from a well (Box 4.4). A heat pump

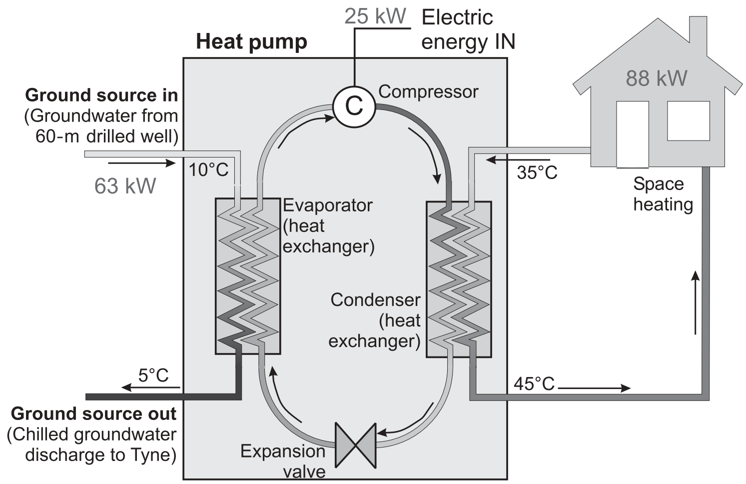


Figure 4.5 A schematic diagram of a GSHP, based on the design for the Eco-Centre at Hebburn, Tyneside (Box 4.4).

transfers this heat energy to the interior of our house and, on the way, it is upgraded to a temperature θ_{in} that is adequate to support our domestic heating system. This value of θ_{in} may be

- over 60°C if we have a ropey old conventional hot water central heating system;
- 45–55°C if we have a more modern low-temperature central heating system, with a high radiator surface area in our house;
- 35–45°C if we have underfloor water-borne central heating;
- 25–30°C if we use warm air circulation as our means of heating.

The total heating effect H is given by

$$H \approx Q_{env} + E \tag{4.3}$$

where E is the electrical energy required to power the heat pump.

4.7 The efficiency of heat pumps

We have already established that there is a theoretical limit on the efficiency of heat engines (Equation 4.1). An ideal heat pump is simply the reverse of an ideal heat engine. This, if we define the efficiency of a heat pump as the ratio of heat delivered (H) at elevated temperature (θ_1) to work performed by the compressor (W), then from Equation 4.1 we can see that the theoretical maximum heat pump efficiency, using an idealised Carnot cycle (Heap, 1979), is

$$E_{max} = \frac{H}{W} = \frac{\theta_1}{(\theta_1 - \theta_2)} \tag{4.4}$$

In modern heat pumps, the compressor is powered by electricity, so W is replaced by E (the electrical power input). The efficiency of the heat pump is usually referred to as its coefficient of performance (COP_H), where

$$COP_H = \frac{H}{E} \tag{4.5}$$

From Equation 4.4, we see that there is a theoretical maximum for COP_H, which depends on the temperature at the delivery side (θ_1) and the source side (θ_2). If $\theta_1 = 35°C$ (308 K) and $\theta_2 = 10°C$ (283 K), the maximum theoretical efficiency of an ideal heat pump would be around 1230% (i.e. the maximum COP_H would be 12.3). We also see that the efficiency of the heat pump decreases with increasing delivery temperature and decreasing source temperature. In other words, a given heat pump does not have a fixed COP_H: this will depend on the operating conditions and temperatures. As we have already noted, the greater the temperature difference between the evaporator and the condenser, the smaller the heat output (*or* the greater the amount of power required by the heat pump to pump a fixed quantity of heat).

In practice, the real COP_H of a heat pump will be much lower than the ideal, for several reasons (Heap, 1979):

- The evaporator temperature (θ_2) is usually below the environmental source temperature in order to ensure a kinetically rapid transfer of heat from the environment to the refrigerant (the ideal Equation 4.4 assumes that the evaporator temperature is very similar to the environmental source). Similarly, the condenser temperature (θ_1) will be higher than the temperature of the space to be heated.
- Real vapour compression heat pumps do not use the ideal Carnot cycle of vapour compression but often a cycle called the Rankine cycle, which is more practical but slightly less efficient.
- Compression inefficiencies and other inefficiencies in the system.

Figure 4.6 shows an example of a typical manufacturer's heat pump curve for a small space-heating GSHP. Each heat pump model will have a different performance, although not usually radically different, as most heat pumps use similar compressors and refrigerants. The manufacturer's manual will usually show a range of curves relating heat output and COP_H to factors such as source temperature, heating delivery temperature and fluid flow rates across the evaporator and condenser. We can see from Figure 4.6 that, if the source temperature is a groundwater at 10°C entering the evaporator, and if we are delivering warm water from the condenser side at 35°C to an efficient underfloor heating system, we would predict a COP_H of around 5.4 with this model (compare with the theoretical maximum of 12.3, calculated above). If, however,

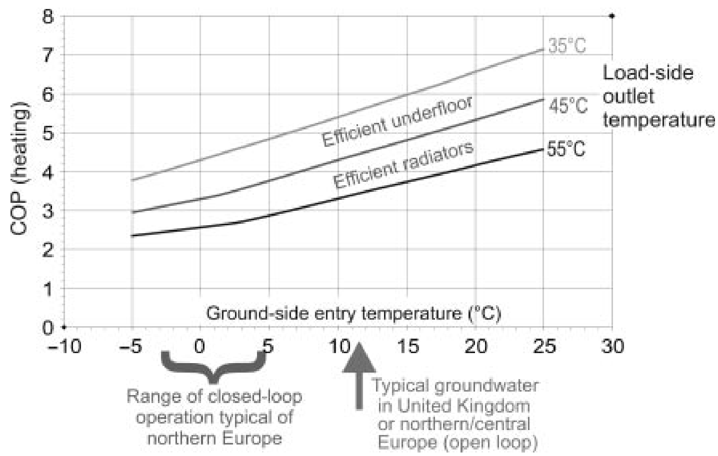


Figure 4.6 Performance curves for a typical domestic heat pump, showing coefficient of performance (COP_H) plotted as a function of evaporator inlet temperature (cold) and condenser outlet temperature (warm). NB: Based on one type of commercially marketed ground source heat pump – others will have different curves.

our source is an antifreeze solution at 0°C from a heat exchanger that has been sunk into the ground (more on exactly how we do this in Chapter 7), our COP_H would be lower (but still fairly impressive) at 4.3. If, however, the source temperature is 0°C and we deliver heat via a radiator system operating at around 55°C, the COP_H comes down to around 2.6, which may not be acceptable in some energy economies (see Sections 4.13 and 4.16). Thus, you should note the following important principle:

> To design efficient GSHP heating systems, we should supply heat via a low-temperature heating system and we should ensure that our ground-coupled source remains at an acceptably high temperature.

For most space-heating GSHPs, we would hope for a COP_H of at least 3, and probably approaching 4, under operational conditions. Table 4.2 shows the guideline values for the performance of heat pump systems in Europe. Because the efficiency of heat pumps varies depending on the source and sink temperatures, it is important that they are tested under standard conditions. In the European Union (EU), these standard conditions are currently specified in European Norm EN 14511-2 of 2007. For example, water-to-water heat pumps (i.e. the heat source is water pumped from a well or a lake and the heat is delivered by a water-borne heating system), one common test condition is

- water enters the evaporator at 10°C and leaves at 7°C, while the building's central heating circuit has a flow temperature of 35°C from the condenser and a return of 30°C (typical of an efficient underfloor heating system). This test condition is referred to as W10/W35.

Table 4.2 Target efficiencies for heat pumps and heat pump systems. The column marked ECA (2010) shows the minimum coefficient of performance in heating mode that a heat pump must attain to be eligible for an Enhanced Capital (tax) Allowance in the United Kingdom, under standard test conditions specified in the footnotes and in the European Norm EN 14511-2 of 2007. The column marked EN15450 shows the target (and minimum, in parentheses) seasonal performance factors (SPF_H) for heat pumps providing space heating and domestic hot water under actual operational conditions in Central Europe according to the European Norm EN 15450 (2007), Annex C3. *This table represents a simplification of the original documents, to which the reader is referred to compliance and design processes. Note that these values may be subject to change or modification over time.*

Heat pump environmental source	Heat pump delivery system	COP_H – ECA (2010)	SPF_H – EN15450	
			New building	Existing buildings
Air sourced	Water-borne heating	>4.0 (<20 kW)[1] >3.8 (>20 kW)[1]	3.0 (2.7)	2.8 (2.5)
Air sourced	Warm air	>3.2 (single package)[2] >3.6 (split units)[2]		
Ground sourced (closed loop)	Water-borne heating	>4.0[3]	4.0 (3.5)	3.7 (3.3)
Water sourced (including groundwater)	Water-borne heating		4.5 (3.8)	4.2 (3.5)

[1] Under standard condition: outdoor air dry bulb temperature = 7°C, warm water outlet temperature = 35°C: **A_{db}7/W35.**
[2] Under standard condition: outdoor air dry bulb temperature = 7°C, indoor air dry bulb temperature = 20°C: **A_{db}7/A_{db}20.**
[3] Under standard condition: antifreeze solution ('brine') entry temperature from ground heat exchanger to heat pump = 0°C, warm water outlet temperature = 35°C: **B0/W35.**

Another common condition is W10/W45. For GSHPs that are coupled to the ground via a buried heat exchanger through which an antifreeze solution flows, a common test condition is

- antifreeze (or 'brine' as it is sometimes referred to – a hangover from the days when antifreezes were often salt solutions) enters the evaporator at 0°C and leaves at –3°C, while the heating circuit has a flow temperature of 35°C and a return of 30°C. This test condition is referred to as B0/W35.

4.8 Air-sourced heat pumps

Air-sourced heat pumps (Figure 4.7) will generally have a slightly lower efficiency under operational conditions than GSHPs. This is because of several factors:

- the sub-zero temperatures to which air can drop on cold winter days (i.e. low source temperatures);
- inefficient transfer of heat from air to the evaporator;

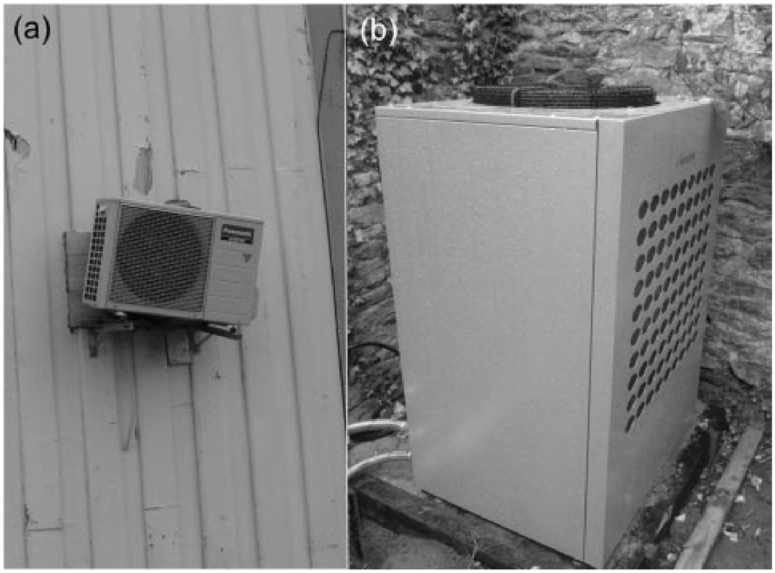


Figure 4.7 (a) A small, rather rickety and not ideally installed air-sourced heat pump on a Norwegian house (*photo by Dave Banks*). (b) A domestic 9.5-kW air-sourced heat pump showing the external air intake/heat exchanger, *reproduced by kind permission of Water and Energy Field Services Ltd.*

- energy must be expended in fanning large flows of air over the evaporator (air has a very low volumetric heat capacity, so we need lots of it!);
- the fact that energy may often be expended on heater elements to melt ice derived from moisture that has condensed and frozen on the evaporator air intake.

Air-sourced heat pumps also have a much higher visibility than GSHP, and they are seldom pretty (Figure 4.7). On the other hand, many air-sourced heat pumps on the market today employ somewhat more advanced control mechanisms and technology than many GSHPs. Many air-sourced heat pumps use variable speed compressors, rather than the single speed units that are installed in most GSHPs (for reasons that the author has yet to fully fathom). Being also cheaper to install, they can thus give GSHPs a 'run for their money'. In some situations, air-sourced heat pumps may be a very logical choice for low-carbon heat delivery.

Nevertheless, we might typically expect a coefficient of performance to be maybe 0.5 units lower than that of a corresponding well-designed GSHP. Let us consider a small house, with a peak heating load H of 6 kW, heated by an air-sourced heat pump with a $COP_H = 3$ under typical winter operating conditions. The electrical energy input (E) required will be given by Equation 4.5: $E = 6\,kW/3 = 2\,kW$. If we are generous and assume that all the electrical energy input ends up as useful heat (which may not be the case if we count the electricity we may use for de-icing or for the fan), we can use Equation 4.3 to estimate that the heat energy derived from the outside air $H_{env} \approx 6\,kW - 2\,kW = 4\,kW$. Thus, with our air-sourced heat pump, we are expending 2 kW of electricity to transfer 4 kW of free environmental energy from the outside air to our house. Of the 6-kW total heating effect delivered, 2 kW is electrically derived and 4 kW is renewable, environmental energy 'conjured' from thin air.

We should note, from Equation 4.4, that the efficiency, or COP_H, of the heat pump is greatest when the difference between θ_2 (the environmental source) and θ_1 (the heat delivery temperature) is minimised. We would expect very low heat pump efficiencies if we were trying to extract heat from cold winter's air at –5°C and deliver it to an old high-temperature radiator central heating system at 65°C. It would be far more efficient to deliver heat to a low-temperature heating system, such as warm-air circulation or underfloor central heating. It would be even better if we could extract heat from an environmental source that retains a relatively high temperature of, say, 10°C even in winter. This is where GSHPs potentially score over air-sourced heat pumps.

4.9 Ground source heat pumps

A GSHP is a heat pump where the source of environmental energy is the ground, or a medium thermally coupled to the ground. By this last phrase, we mean a medium in intimate thermal contact with the ground, such as groundwater. Some people might even class heat pumps based on deep lakes, ponds or river intakes as ground-sourced (or ground-coupled) heat pumps.

Let us consider a simple GSHP, based on groundwater being pumped from a well. We have already seen that, in the United Kingdom, groundwater might be expected to be at a temperature of around 11°C. We can thus pump groundwater from the ground at a rate Z such that it passes into our heat pump's evaporator. The heat pump will extract a heat energy flux (G) from the groundwater and the groundwater's temperature will drop. A typical magnitude for this temperature drop ($\Delta\theta$) might be around 3–5°C. If $\Delta\theta = 4°C$, this leaves us with a cool groundwater at a temperature of 11°C – 4°C = 7°C to dispose of (more of this in Chapter 8). The heat extracted is upgraded in the heat pump to a temperature θ_{in}, and is used to support a domestic heating system. Again, assuming that energy loss due to acoustic noise is negligible and that all extracted heat and heat of compression is efficiently delivered to a point of use, the total heating effect (H) is given by

$$H \approx G + E \tag{4.6}$$

where G is the ground source heat (heat extracted from the ground), and

$$COP_H = \frac{H}{E} \tag{4.5}$$

We can, however, also relate the heat extracted from the groundwater to the flow rate Z, the temperature drop across the heat pump ($\Delta\theta \approx 4°C$) and the specific heat capacity of water ($S_{VCwat} = 4180\,J\,L^{-1}\,K^{-1}$):

$$G = Z \cdot \Delta\theta \cdot S_{VCwat} \tag{4.7}$$

Thus, if $Z = 1\,L\,s^{-1}$, then

$$G = 1\ L\ s^{-1} \times 4\ K \times 4180\ J\ K^{-1}\ L^{-1} = 16\,720\ J\ s^{-1} = 16.7\ kW$$

A groundwater-based heat pump may well have a coefficient of performance COP_H of 4–5. Let's call it 4: this means that

$$\begin{aligned} G &\approx H\left(1 - \frac{1}{COP_H}\right) \\ H &= 16.7\ kW \times 4/3 = 22.3\ kW \text{ and } E = 5.6\ kW \end{aligned} \tag{4.8}$$

So, from a modest groundwater flow rate of $1\,L\,s^{-1}$, we can provide a heating effect of 22.3 kW, provided we have at least 5.6-kW electricity supply to power our heat pump. Given that $\Delta\theta =$ may vary from 3 to 5°C, we could get 13–21 kW of heat (G) from the groundwater, with a heating effect of 16–28 kW, depending on the COP_H.

4.10 Seasonal performance factor (SPF)

Note that the COP_H is an instantaneous value, which will depend on the temperatures of the heat source and sink at the time of measurement. When assessing the efficiency of a system, it may be more appropriate to take a long-term average COP_H over the entire heating season. This long-term average is often referred to as the SPF_H of the heat pump. We should also remember that, in the example above (Section 4.9), an electrical pump is being used to pump water from our well to the surface, consuming an electrical power E_{pump}. Clearly, if we wish to assess the efficiency of our entire heat pump system, we have to take this into account, together with any other power expenditure on auxiliary circulation pumps, and so on. We can define a seasonal performance factor (SPF_{Hgsub}) for our heat generation sub-system, which takes into account the total heat (H) delivered over a heating season, the total electrical energy consumed by the heat pump compressor in the same period (E), the auxiliary pumping (E_{pump}) and any backup immersion heater (E_{backup}):

$$SPF_{Hgsub} = \frac{H}{(E + E_{pump} + E_{backup} + \ldots)} \tag{4.9}$$

We could go even further and define a seasonal performance factor (SPF_{Hsys}) for the entire heating system in the building. European Norm EN15450 (2007) defines these different SPF values, although you should be aware that different countries' standards and guidelines do vary when it comes to the exact definitions of SPF and which components of the system should be included. Ensure that you understand how an SPF has been calculated and what components it includes before making claims about system efficiencies.

4.11 GSHPs for cooling

We have seen that heat pumps can extract heat from a low-temperature source (the ground) and transfer it to a high-temperature sink (a central heating system). At this point, it is necessary to mention that many models of heat pump allow the refrigerant flow to be reversed: what was the condenser (coupled to the building) now becomes the evaporator, and what was the evaporator (coupled to the ground) now becomes the condenser. Thus, heat can be sucked out of a building and transferred into the ground or groundwater. Our GSHP can provide active cooling, as a kind of ground-sourced air conditioner. A conventional air conditioner is, of course, just an air-sourced heat pump, sucking heat from the indoor air and dumping it to the outdoor air.

And, as the ground in summer is relatively cool (say, 11°C in Britain), it is far more efficient for a heat pump to dump heat to the ground than to the air (which may be at 25°C). More about using the ground for cooling in Chapter 6.

4.12 Other environmental sources of heat

Thus far, we have considered air-sourced heat pumps and GSHPs. We have noted the potential drawbacks of air-sourced heat pumps (low efficiency at certain times due to fluctuating air temperature), but should probably mention their benefits in terms of lower capital cost and ease of installation. We should also note that there are other sources of reliable environmental heat than the ground. Efficient heat pump systems can be based on abstraction of heat from deep lakes, from the sea, from fjords (below a certain depth, seawater has a relatively constant temperature) or from sewerage (Matte, 2002).

With a little imagination and an understanding of the heat flows within an industrial process, the possibilities are manifold. At dairies, there is typically a need to cool milk immediately following the early morning milking. Using heat pumps, we can chill the milk and transfer the heat to the Dairy Manager's office (a 'cow-sourced' heat pump). At other farms, root vegetables may require cool storage after they are brought in from harvest – a 'potato-sourced' heat pump can convey heat from the cool store to the other farm buildings.

4.13 The benefits of GSHPs

We can list a host of reasons why GSHPs provide attractive sources of heating and cooling. Firstly, they are visually unobtrusive: a typical water-to-water heat pump is a white box, not unlike a fridge. They can be tucked away in a cellar or a plant room and nobody need know that they are there. Conventional air-conditioning and cooling systems may require large units to be bolted on to the side of buildings or mounted on the roof (Figure 6.6). Even other environmentally friendly energy sources can have a major visual impact: consider wind turbines, solar thermal panels or photovoltaic arrays mounted on the roofs of buildings. The low visibility of GSHPs can be particularly attractive to those requiring cheap 'green' energy in a national park or other area where planning regulations restrict visual impact of new developments. The low visibility of GSHPs has a downside as well: wind turbines and solar panels advertise themselves, saying 'Look at me; I'm a low-carbon household'. GSHPs are not immediately obvious; possibly one reason for the initial resistance of the UK market to the technology.

Furthermore, a large office building that is cooled and heated by GSHPs may not need the massive roof-mounted evaporative cooling towers (Chapter 6) associated with conventional cooling systems. This may have structural implications for the building: it will not have to bear the weight of the cooling towers, possibly saving construction costs. It will also free up roof space for high-value penthouse apartments and offices.

GSHPs present a minimal fire hazard and require minimal ventilation. They are also extremely low maintenance and have a long lifetime compared with many fossil fuel boilers. GSHPs produce relatively little noise, if properly mounted in an insulated cabinet, and can be placed in a household utility room or garage with minimal disturbance. (GSHPs, like fridges, do emit some noise, however. They should probably not be placed in a room that is regularly occupied.)

Probably, the most important advantages relating to heat pumps are (1) running cost and (2) environmental impact in terms of CO_2 emissions. Before we proceed, it is important to realise that GSHPs provide a low-CO_2 source of heating, but not usually a zero-CO_2 source. GSHPs use electricity: a 6-kW unit may use 1.5- to 2-kW electricity to deliver its peak heating load. In the United Kingdom and many other countries, electricity generation results in emissions of CO_2. In fact, heating your home by electricity can be very carbon-unfriendly, due to efficiency losses during generation and transmission through the National Grid. Let us consider the United Kingdom's electricity supply (Figure 4.8). The bulk of the United Kingdom's electricity in the first decade of the twenty-first century is generated by combustion of natural gas and coal. For every kilowatt-hour of electricity consumed that is generated by gas, c. 0.4 kg of CO_2 is released. Electricity generation by coal is even worse with c. 0.9 kg CO_2 emitted for every kilowatt-hour consumed. Generation of electricity by nuclear or 'renewable' sources such as hydropower result in zero CO_2 emission at the point of generation (although we should remember all the CO_2 emitted during uranium mining and dam

	Gas	Coal	Oil	Nuclear	Renewables	Other	Average
Killip (2005)	0.44	0.95	0.84	0	0		
OFGEM (2005)	0.36	0.91		0	0	0.50	0.43
DECC (2010)	0.37	0.87		0	0	0.63	0.41

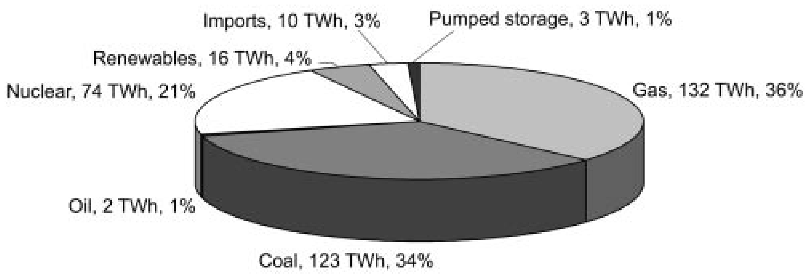


Figure 4.8 The percentage of electricity generated by various sources in the United Kingdom in 2005. The pie diagram shows the total terawatt-hours of electricity contributed to the National Grid from the various main sources, and as a percentage of a total of ≈360 TWh, according to DTI (2006). The table shows the kilogram of carbon dioxide emitted per kilowatt-hour of electricity delivered from the Grid from various sources (Killip, 2005; OFGEM, 2005; DECC, 2010). Note: 1 kg C = 3.66 kg CO_2.

construction). The carbon 'footprint' of using electricity to heat your house thus depends on the 'mix' of fuels used to generate the electricity. In the United Kingdom, the average CO_2 emission for electricity consumed is around 0.52 kg per kilowatt-hour.

Of course, elsewhere, the direct use of electricity for heating a house may result in much lower CO_2 emissions, depending on how the electricity is generated. Norway's electricity is dominated by hydroelectric power generation, and electricity consumption is very CO_2 efficient. Sweden has long been dependent on a network of nuclear power stations: these may be undesirable for other political reasons but are also CO_2 efficient.

Let us compare the costs and CO_2 emissions from using electricity to heat your house with other common domestic fuels (Table 4.3). In the United Kingdom, we can see that heating a house with electricity is both expensive and carbon inefficient. In the year 2009, according to SAP (2010), 1 kWh of electricity cost anywhere between 4.8 p and 12.8 p (depending on your specific tariff and when you use the electricity – the standard tariff was 11.5 p per kilowatt-hour) and resulted in the emission of 0.517 kg CO_2 (astonishingly, the United Kingdom's electricity has become more carbon intensive since the previous edition of SAP in 2005, when a figure of 0.422 kg CO_2 was cited). The most efficient conventional means of heating your house is with a modern condensing 'combi' gas boiler, which can be up to 90% efficient in converting the energy of combustion into useful space heating. Efficient usage of gas cost 3.10 p in 2009 and emitted only 0.198 kg CO_2 per kilowatt-hour.

Why would anybody therefore choose to run an electrically powered GSHP rather than a combi gas boiler? Because, a GSHP is not using the electricity to heat your

Table 4.3 The nominal costs per kilowatt-hour of various heating fuels, together with the approximate mass of CO_2 emitted per kilowatt-hour of energy delivered. Figures for gas, LPG, oil, coal and electricity are derived from the UK Government's standardised assessment procedure (SAP, 2010). Since publication of this document, gas and electricity costs have increased substantially: the table is thus for illustrative purposes only and should not be used for design. Note that modern condensing combi gas boilers can approach 90% efficiency. The figures for the GSHP are derived from those for electricity supply by assuming a seasonal performance factor of 3.6.

	SAP (2010) cost per kWh (British pence = £0.01)	kg CO_2 per kWh	Primary energy factor
Mains gas	3.10 p	0.198	1.02
Bottled liquid petroleum gas (LPG)	8.34 p	0.245	1.06
Heating oil	4.06 p	0.274	1.06
House coal	2.97 p	0.301	1.02
Electricity	11.46 p (standard) range of 4.78–12.82 p	0.517	2.92
GSHP ($COP_H = 3.6$)	3.18 p (standard tariff) and probably lower	0.143	

house directly: it is using electricity to transfer free, zero-CO_2 energy from the ground to your house. If we consider a GSHP with an SPF of 3.6, for every 1 kWh of heat provided to your house, the GSHP consumes only 1 kWh/3.6 = 0.28 kWh of electricity. This means that we can divide the environmental and economic costs of the electricity by 3.6, as shown in the bottom line of Table 4.3. Here, we see that the GSHP wins hands down over mains gas on CO_2 emissions with around 0.14 kg CO_2 released per kilowatt-hour heating effect. In terms of running cost, the GSHP compares very favourably with gas, too, especially if one can negotiate a cheap (e.g. night-time) tariff for electricity supply.

Thus, we have demonstrated that, while well-designed GSHPs running off mains electricity are not a zero-carbon source of heat energy, they are significantly better than any conventional alternative. And if you live in a country such as Norway, where most electricity is 'green' hydroelectricity, you can genuinely boast of having a zero-carbon heating system. Moreover, you can comfort yourself with the knowledge that you are using Norway's limited reserves of hydroelectric power three to four times more efficiently than if you had merely plugged an electric fan heater into the mains. (Norway's hydropower reserves are, of course, finite. In the late 1990s, they effectively ran out, forcing Norway to import significant quantities of energy from its neighbours for the first time, sending energy costs spiralling during dry summers.)

Given that space heating and water heating are responsible for 34% of the United Kingdom's carbon dioxide emissions (Prime and Millard, 2007), and that the domestic heating sector probably accounts for around 25% of the national total, the environmental arguments for GSHPs are unassailable. They are also beginning to look increasingly attractive from an economic point of view. So why is everyone not rushing out to buy one?

4.14 Capital cost

The fact is that the initial capital cost of a GSHP system is rather high. A 6-kW water-to-water heat pump in the United Kingdom may cost around £3000 from the manufacturer (see Figures 4.9 and 4.10). Including the cost of borehole drilling or trench installation, pipework and manifolds, these could increase the cost to over £8000. Add on the installer's profit margin and modifications to your heating system and a small domestic GSHP installation could easily cost you in excess of £12 000.

We have seen above that the running costs of a GSHP may well be lower than a conventional combi gas boiler. But are they low enough to justify that much larger capital cost (a modest combi gas boiler costs less than £1000)? A study was made of a group of around 60 similar bungalows in Nottingham, UK (Hill and Parker, 2005/06; Parker, 2007). Some of these were fitted with combi gas boilers and others with GSHPs. For the bungalows with gas boilers, and considering solely heating and hot water, the average gas bill (December 2005 tariffs) was £210–250 per year, with an annual CO_2 emission of around 2 tonnes of CO_2 per household. For the bungalows with GSHPs, the heat pump's electricity bill was £200 per year, and just over 1 tonne of CO_2 was emitted (note that the United Kingdom's electricity has become *less* carbon efficient since this study was carried out – see Table 4.3). The GSHPs recorded an average overall SPF of 3.48 [with estimated values of 4.16 when delivering low-temperature water-borne space heating and 2.47 when delivering domestic hot water (DHW) at up to 65°C]. As we have already concluded, the environmental benefits were significant, in terms of CO_2, but the cost savings were modest (compared with gas combi boilers, but substantial compared with other conventional fuels). With an annual saving in fuel bills of no more than £50, it would have taken decades to pay back the initial capital expenditure on the heat pump. We could, of course, argue that, over a c. 20-year lifetime of a heat pump, a gas combi boiler may have had to be replaced at least once. Furthermore, we could argue that the GSHP owners would not have incurred the gas boilers' hefty annual maintenance bills, and also that the GSHP households may well have received a significant government subsidy towards the cost of the heat pump. Indeed, Hill and Parker (2005/06) argued that, taken over the lifetime of the heat pump, the total costs (capital plus running costs) of a combi gas boiler and a GSHP may not be too dissimilar. Nevertheless, the economic arguments for domestic-scale GSHPs in the United Kingdom have not been overwhelming, and it begins to become clear why British citizens still tend to choose gas boilers over GSHPs! Indeed, we are forced to admit that, for domestic properties in Britain with access to mains gas, gas heating is still the cheapest form of space heating on economic grounds. In fact, those households and individuals who choose GSHPs tend to be motivated by moral, 'environmental' reasons, rather than financial ones. The exception to this observation is, of course, in rural areas of Britain where mains gas is unavailable and GSHPs are genuinely an economically attractive alternative to oil firing or bottled gas.

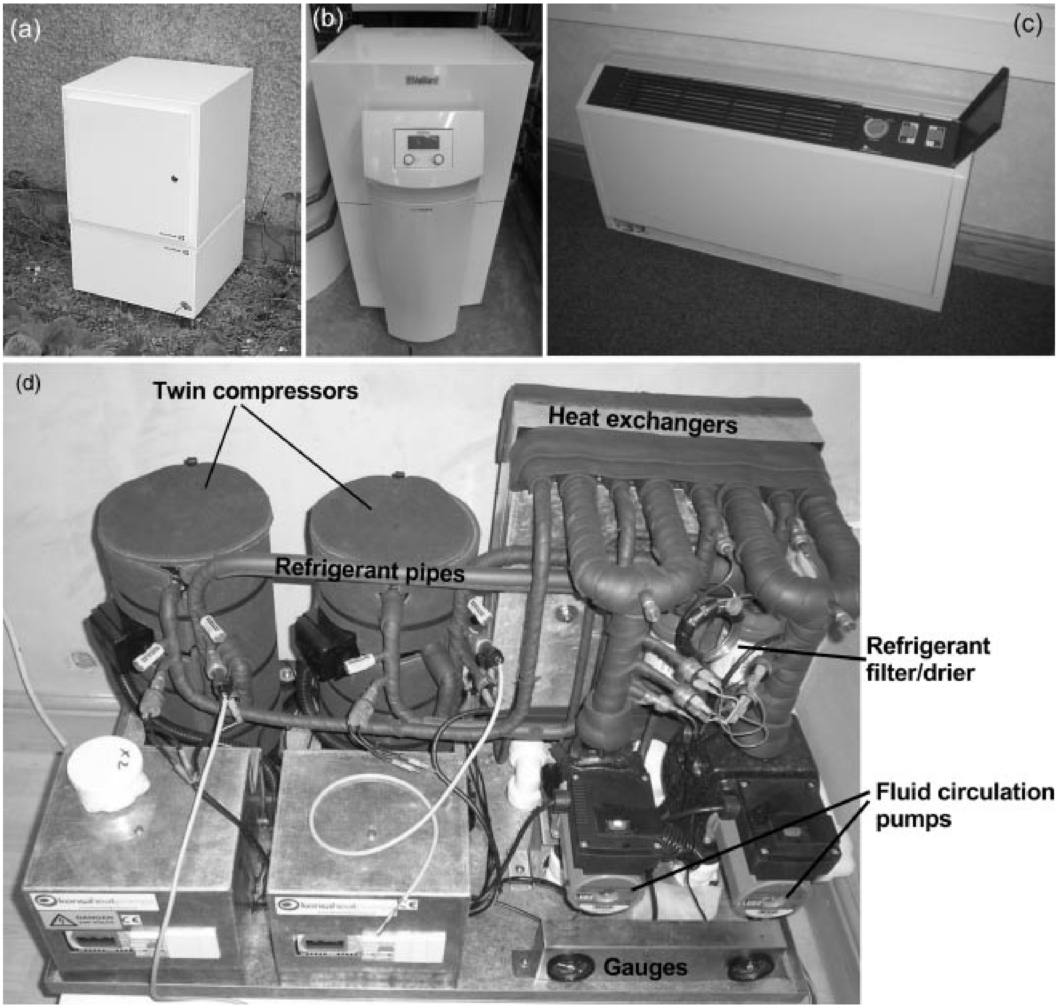


Figure 4.9 (a) A domestic 6-kW brine-to-water ground source heat pump (GSHP) that would supply a domestic water-borne central heating system or underfloor heating system, *reproduced by kind permission of Kensa Engineering Ltd.* (b) A larger 17.5-kW GSHP, *reproduced by kind permission of Water and Energy Field Services Ltd.* (c) A console-type brine-to-air GSHP: this extracts heat from a ground-coupled carrier fluid to a flow of warm air. It can be reversed in summer, to provide a current of chilled air, rejecting waste heat to the carrier fluid (*photo by David Banks*). (d) The innards of a twin-compressor brine-to-water GSHP, with integrated fluid flow circulation pumps, *reproduced by kind permission of Kensa Engineering Ltd. and Geowarmth Heat Pumps Ltd.* Note that 'brine-to-water' refers to a water-to-water heat pump installed on a closed-loop ground heat exchanger, filled with a water-based antifreeze (or 'brine').

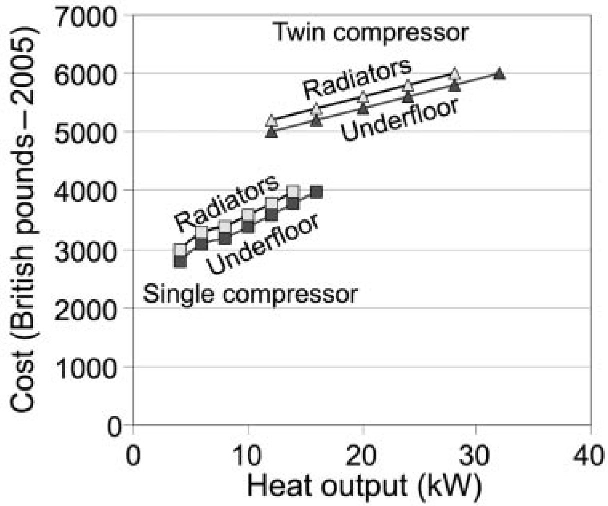


Figure 4.10 The cost of GSHP units of different heat output ratings, tailored for either underfloor or wall-mounted low-temperature radiators, supplied by a British manufacturer in 2005. Note that larger units (above 12 kW) are typically fitted with dual compressors. This reduces the peak electricity draw on start-up. There is relatively little price increase within the range of single-compressor GSHPs; the big increase in price occurs as one moves from single to dual compressor systems.

This is set to change, however, as the British government is currently introducing a 'renewable heat incentive' (RHI), which intends to pay the owner of a GSHP a subsidy per kilowatt-hour of renewable heat generated (DECC, 2011): the payment is currently set at 4.3 p per kilowatt-hour for small commercial installations. Such a tariff would potentially make GSHPs genuinely economically attractive in the long term.

Of course, in other nations, with other energy pricing structures than Britain, economic analysis will provide wholly different results – in Norway, for example, most people probably do not have access to cheap mains gas, so the choice may be much simpler: use of expensive paraffin for stoves, use of expensive electricity for resistance heaters *or* use of 70% less electricity for a heat pump.

Furthermore, large commercial and public sector organisations should note that an economy of scale comes into play. The larger the heat pump you wish to buy, the cheaper it becomes per installed kilowatt. Figure 4.10 shows the cost of a range of heat pumps supplied by one British heat pump manufacturer (2005 costs). It will be seen that while a small domestic 6-kW heat pump costs £3000 (£500 per kilowatt), a 20-kW unit costs between £5000 and £6000 (i.e. £250–300 per kilowatt). Similarly, drilling 10 boreholes at a single site will be much less expensive than 10 times the cost of drilling a single borehole, as the larger project will only need one mobilisation of equipment, one site survey, and so on, whose cost can be spread over the 10 holes. Thus, the bigger your building or project, the proportionately lower the capital cost of your GSHP system becomes per installed kilowatt and the more attractive the technology is from an economic point of view. Indeed, Banks *et al.* (2005) cited several medium-to-large projects, where a payback time of several years, rather than several decades, was calculated. A significant fall in payback times for commercial projects in recent years is also documented by Kelley (2006).

4.15 Other practical considerations

A householder considering the purchase of a GSHP system may be so overcome by considerations of cost and carbon emissions that he or she may forget some practical aspects. We would, in particular, encourage potential purchasers and installers to be mindful of the following:

- GSHPs are low-noise installations, but not no-noise. Make sure that they are installed in an insulated and vibration-proof cabinet in a separate room away from main living areas (cellar, utility room, cupboard, etc.)
- GSHPs use electricity very efficiently. A 10-kW_{th} unit may only consume 2.5 kW electrical energy – but that is still quite a lot of electricity: (11 A at 240 V). An electrician should ensure that the heat pump is powered by an appropriately rated circuit. Bear in mind that, on start-up, the compressor will draw significantly more power (maybe as much as three times more) than its normal running load. The practical implication of this is that a 240-V domestic electricity supply may not tolerate more than a 10-kW_{th} heat pump unit. For heat pumps in the range of 10- to 20-kW_{th} output, twin compressor constructions would be recommended (see Figure 4.9), such that both compressors do not start up simultaneously, limiting the current drawn at any one time. For heat pump installations in excess of 20 kW_{th}, a three-phase electricity supply may be necessary. The bottom line is 'check with your electricity supplier' – tolerable loads vary from location to location in the United Kingdom and between various utility companies.
- GSHPs operate most efficiently (highest COP_H) with a low-temperature output. Therefore, warm-air circulation or low-temperature underfloor warm-water heating systems (or even low-temperature, high-surface-area wall-mounted radiator systems) ideally complement GSHPs. GSHPs can be retrofitted to older heating systems, but they may not provide a wholly comfortable level of heat.
- A GSHP will usually be installed with some form of thermal buffer (e.g. a buffer tank of water) to provide a thermal mass against which it can work. The cut-in and cut-out of the heat pump's compressor may then be triggered by temperature sensors in the buffer tank. Such a thermal buffer prevents the heat pump cutting in and out very rapidly (so-called 'cycling'). A heat pump does *not* have to run continuously to get good results; however, very rapid cycling can cause compressor wear.

- A simple, flexible solution for ground source cooling and heating for some buildings is to circulate a fluid, chilled or heated by the heat pump, around a so-called building loop within each heating/cooling block. Fan-coil units, mounted on this loop, will distribute warm or cool air throughout the space. Larger buildings may have simultaneous heating and cooling needs and will thus require separate heating and cooling circuits (see Chapter 7).
- GSHPs can also be used to supply domestic hot water (DHW). This can be quite a challenge, however, as the efficiency of most GSHP systems decreases at the high temperatures that we tend to expect for DHW. For example, the COP_H of a heat pump delivering hot water at 55°C may not be more than 2–2.5 (K. Drage, *pers. comm. 2007* and Figure 4.6). In mitigation of this statement, it should be remembered that the overall efficiency of heating a tank of cold water up to 55°C may be significantly better than this figure, as the heat pump produces water at a much lower temperature (and much higher COP_H) during the initial part of the heating cycle, while the tank is still cool. The threat of *Legionella* infection means that many DHW systems require temperatures in excess of 60°C, at least as part of a regular sterilisation programme (see Box 7.3). Several ways of meeting the challenge of DHW have been devised and are discussed in Section 7.5.

4.16 The challenge of delivering efficient GSHP systems

We have seen (Section 4.13) that, in countries such as the United Kingdom, with 'dirty' (carbon intensive) electricity, it is of prime importance to design efficient GSHP systems. Once the SPF_H of a GSHP system begins to drop significantly below 3, a large part of the carbon advantage over mains gas has been eroded. Indeed, if the SPF_H begins to edge down towards 2, a GSHP may end up as a very costly means of delivering more carbon to the atmosphere than merely using a combi gas boiler. Unfortunately, there is an unavoidable relationship between efficiency and capital cost. An efficient system (as we shall see in Chapters 10 and 11) requires significant amounts of heat exchanger to be buried in the ground: this costs money in terms of drilling or excavation. Understandably, many GSHP customers want to save on capital costs; they are tempted to shave several tens of metres off the drilling/excavation programme, and they end up with an inefficient system that does not save them money and may not save them carbon (as compared with gas). The results of such practices can be seen in the recent Energy Saving Trust (2010) report, which documents some very poor statistics from field trials of some UK systems. It is against the background of this that recent UK standards (GSHPA, 2011; MIS, 2011a,b) specifically recommend that GSHP systems should be designed so that the temperature of the ground-coupled fluid entering the heat pump does not drop below 0°C over a 20-year design period.

Of course, in Norway or Sweden, with very low carbon electricity (hydroelectric or nuclear) and where mains gas is not usually an alternative means of space heating, the consequences of an inefficient GSHP system are arguably less severe. For example, a Norwegian heat pump with an SPF_H of 2.5 would simply be consuming more electricity (and costing Ola and Kari Nordmann more) than one running with an SPF_H of 3.5. But they would still be making savings compared with an electrical resistance panel heater (which would be a common alternative). It would take them longer to pay back the original capital cost of the system, however.

We can thus begin to see why the EU Renewable Energy Directive (2009/28/EC) states that the SPF of a heat pump should exceed:

$$SPF_H > 1.15 \times {}^{1}\!/_{\eta} \tag{4.10}$$

where η is the electricity production efficiency of the economy in question. The directive recommends using an EU average for η. However, BP (2010) suggested an efficiency of 0.38 for a typical modern thermal power station, leading us to suppose that the SPF_H of an acceptable GSHP system, which leads to a net carbon saving, should exceed 3.03.

4.17 Challenges: the future

We may envisage that future heat pumps will be sleeker and quieter, and use electrical energy more efficiently. Above all, we would hope that their capital cost will decline in line with increased demand and uptake.

We would also hope to see increased transfer of ground source heat technology to the rapidly developing, highly populated economies such as China, India, Brazil and Russia. The per capita emission of CO_2 from these nations is still modest but has the potential to increase dramatically in the forthcoming decades (Box 4.6). The outcome of the battle to retain some global control on atmospheric carbon emissions will likely be lost or won in those countries.

A less obvious, but intriguing, challenge is to investigate the potential that heat pumps can offer to deliver reliable, cheap and efficient space heating to poorer countries, maybe without a well-developed or reliable electricity grid. Electricity, which is so often regarded as being quintessentially 'modern', is actually a peculiarly energetically inefficient means of delivering space heating. Should we rather be looking at developing a new generation of non-electrical heat pumps whose compressors run on water power, biogas or diesel combustion? Or maybe even absorption heat pumps that utilise combustion of gas or wood fuel (Box 4.5)? These technologies could conceivably deliver efficient ground source heat to mountain villages in Pakistan, Scotland or Bolivia, or to remote towns in Russia or China, without the need for massive capital investment in ultimately inefficient electrical infrastructure.

Such non-electric heat pumps are, of course, not a new concept: Heap (1979) noted, however, that a precondition for the efficient operation of fuel-engine-powered heat pumps is the ability to recover the exhaust heat from combustion and deliver this as useful space heating. Sumner (1976a,b) proposed such oil-engine heat pumps, with

BOX 4.6 Energy Consumption, CO_2 Emission and Standard of Living

It has been argued that access to energy resources should be a fundamental human right, but the fact remains that abundant energy is a privilege enjoyed by the few. Energy consumption can therefore be a good indicator of standard of living and, because most energy consumption involves emitting CO_2, per capita emissions of CO_2 are also such an indicator. The UNDP (2004) has produced a fascinating chart linking energy consumption to a so-called human development index (HDI), which can be considered a surrogate index for the standard of living (Figure 4.11).

We can see that inhabitants of nations such as Mozambique and Nigeria consume less than 1 tonne oil equivalents [1 metric tonne oil equivalent (mtoe) = 12.4 MWh] and emit much less than 1 tonne CO_2 per capita per annum (Table 4.4). Russians and Italians consume around 4 toe per capita per year and emit 7–11 tonnes CO_2 per capita per year. The United States sits high on the list, with a per capita energy consumption of around 8 toe per annum and a CO_2 emission of around 19 tonnes.

Of course, there is some discrepancy between the energy consumption and CO_2 emission, depending on the source of energy. Rural Africans, depending heavily on renewable biomass (wood/charcoal/dung) for their energy needs, will have a different rate of CO_2 emission than British or Russians guzzling electricity from relatively inefficient coal- or oil-fired power stations. Icelanders enjoy abundant energy consumption of 12 toe per annum, but with a CO_2 emission rate considerably lower than the United States, owing to their reserves of high-temperature geothermal energy.

Two points are noteworthy. From the UNDP graph, a relatively high standard of living (HDI values of around 0.7) can be reached without energy consumption exceeding 2 toe per capita per annum. To achieve an HDI in excess of 0.8 requires enormously greater energy consumption – there is a marked 'kink' in the graph at this value.

Secondly, we should note that the vast populations of India, Brazil and China currently have a modest per capita annual CO_2 emission (Table 4.4). These economies are expanding rapidly, and their inhabitants understandably have expectations of commensurate increases in living standard. It is not unreasonable to suppose that the degree to which these mega-economies can achieve growth, while restricting their CO_2 emissions, will be decisive for the future of Planet Earth.

recovery of heat from combustion. Both engine-driven (Worsøe Schmidt, 1982; D'Accadia *et al.*, 2001; Lian *et al.*, 2005) and absorption cycle (Murphy and Phillips, 1984) heat pumps for space heating have been sporadically researched in recent decades, with the former even incorporating the technology of Stirling engine cycles (Angelino and Invernizzi, 1996). In fact, engine-driven heat pumps are marketed today,

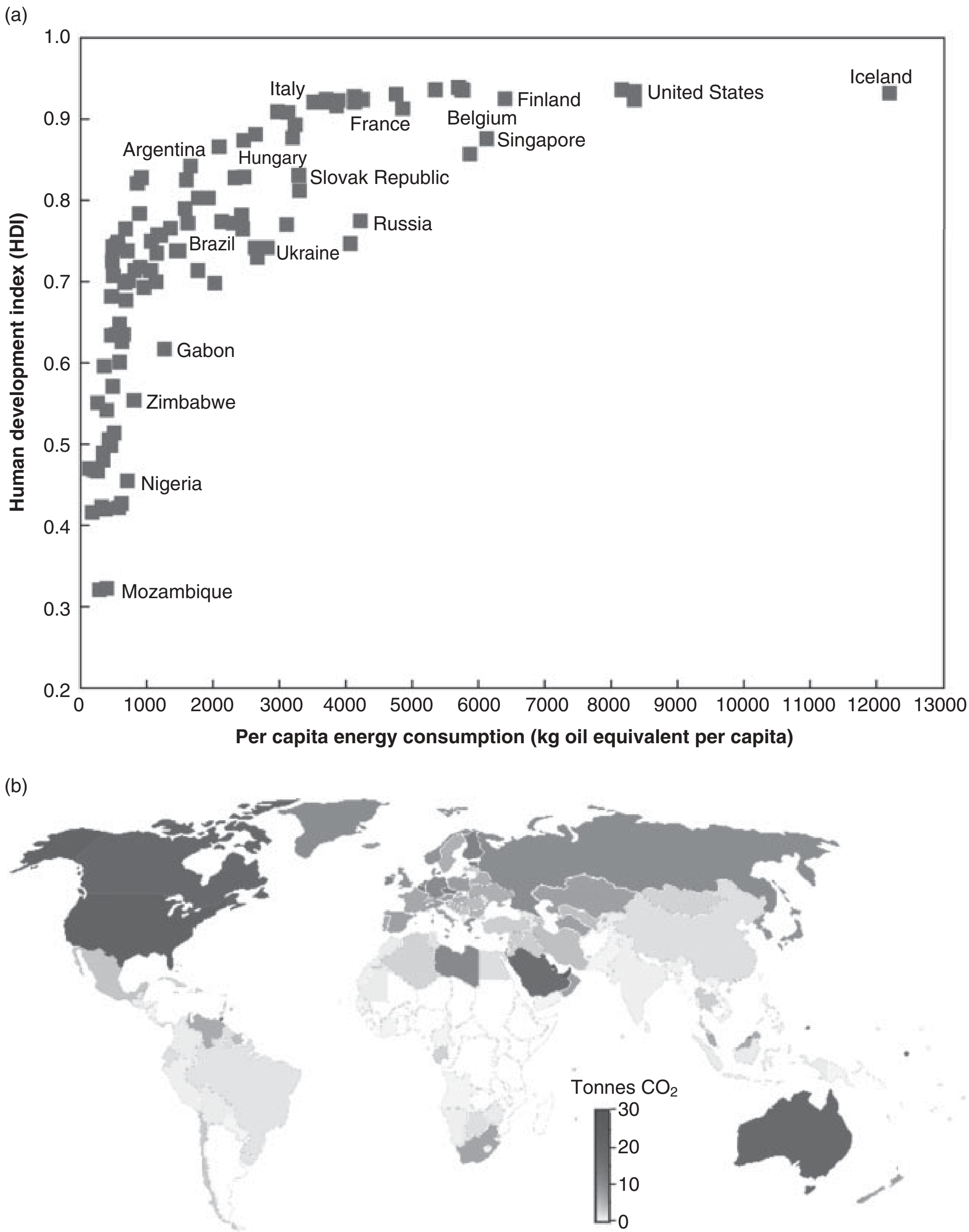


Figure 4.11 (a) The relation between per capita energy usage in kilogram oil equivalent and standard of living [human development index (HDI)] in 1999–2000. (b) The per capita annual emissions of carbon dioxide, as metric tonnes CO_2, by country. *The top figure (a) is published by UNDP (2004) and reproduced by kind permission of the United Nations Development Programme* (http://www.undp.org/energy/docs/WEAOU_full.pdf). *The map (b) appears on Wikipedia and is reproduced under the terms of the GNU Free Documentation License, Version 1.2 or any later version published by the Free Software Foundation.*

Table 4.4 Annual per capita carbon dioxide emissions by country for the year 2007, based on CDIAC (the US Department of Energy's Carbon Dioxide Information Analysis Center) data cited by the UN Statistics Division: Millennium Development Goals Indicators.

Country	Per capita annual emission of CO_2 in 2007 (tonnes CO_2)
Mozambique	0.12
Nigeria	0.64
India	1.4
Brazil	1.9
China	4.9
Sweden	5.4
Iceland	7.6
Italy	7.7
United Kingdom	8.9
Norway	9.1
Germany	9.6
Russian Federation	10.8
United States	18.9
Qatar	55.4

largely as a result of the notorious recent 'brownouts' in the power supply infrastructure along the US western coastline (Chris Underwood, *pers. comm. 2007*). To date, however, there has been no significant emphasis in the utilisation of non-electric GSHPs as 'intermediate technology'.

4.18 Summary

In this chapter, we have seen that there is a tool called a heat pump, which can be used to transfer heat from one place to another. It can be used to pump heat up a temperature gradient: for example, from the cool ground to a warm living room. It can also be used in reverse mode to pump heat down the temperature gradient faster than would have occurred under passive conditions. For example, we can use a heat pump in summer to pump waste heat from a large building to the ground, to a lake or to the outside air.

This may seem an unfamiliar concept to us, but we are all happy to use heat pumps in practice – most of us have refrigerators. Larger-scale heat pumps for heating and cooling buildings are available. They use a modest amount of electricity to transfer free, environmentally friendly energy from the outside environment into our homes (Box 4.7). A heat pump that extracts heat from, or rejects heat to, the ground (or any medium thermally coupled to the ground) is termed a GSHP.

GSHPs offer cheap, reliable, heat energy with low visual impact. They also deliver heat with a low-CO_2 emission footprint: far lower, in fact, than all conventional space-heating solutions. The main drawback is the high capital cost of GSHP systems, making them unattractive to many small-scale, domestic users. At larger scales, however, the capital cost of GSHP schemes decreases per installed kilowatt, rendering them attractive to major commercial, governmental, community, leisure and industrial developments.

BOX 4.7 Exergy

Exergy is a term, coined by the Slovenian, Zoran Rant, in 1956, describing the 'quality' of energy; that is, how useful that energy is. It is defined as the maximum useful work that can be extracted from a system before the system reaches equilibrium with a surrounding heat reservoir.

In very crude terms, petrol is a 'high exergy' fuel. We can use it to heat working fluids to very high temperatures and thus can use it as a fuel for performing lots of useful work (e.g. moving cars, aircraft, engine pistons, etc.). Ground source heat is a 'low exergy' form of energy – we can't really use it for doing much useful work (in theory, we could use heat from the ground to do a small amount of work, using a Stirling engine [Box 4.2], by coupling it to a low-temperature exhaust, such as winter air).

A rational consideration of the use of fuels would suggest that we should

1. reserve high exergy fuels (coal, oil, gas) for doing essential work (powering machines, engines and transport). We should not squander high exergy fuels merely to perform space heating.
2. use low exergy sources (e.g. ground source heat) to perform space heating. The heat pump is the tool that allows us to use low exergy energy for such purposes.

STUDY QUESTIONS

4.1 A GSHP has, under the relevant conditions of operation, a COP_H of 3.3 and a heating output of 12 kW. How much heat is being extracted from the ground and how much electricity does it use?

4.2 After 20 years' operation, the heat pump in question 4.1 is replaced by a more efficient 12-kW heat pump with COP_H of 3.6 for the same temperature conditions. How much heat is being extracted from the ground and how much electricity does it use? What implications does this have?

4.3 During the course of 1 year, a 6-kW heat pump produces 12.3 MWh of heat with an annual SPF_H of 3.3. How much of this heat is classed as 'renewable' or 'zero-carbon' heat? How many compressor running-hours per year does this heat production represent?

5

Heat Pumps and Thermogeology: A Brief History and International Perspective

> The iceman cometh . . .
>
> Eugene O'Neill (1939)

> As the world supplies of fossil fuels diminish, the value and the monetary saving by the use of the heat pump will increase proportionately. Over a century ago Lord Kelvin foresaw and warned us about a world fuel shortage and devised a machine to alleviate that shortage.
>
> John A. Sumner (1976)

The history of heat pumping is intimately connected with the history of artificial refrigeration. The development of both was stimulated first by the growth of a market for ice as a luxury item consumed by the upper classes in Europe, America and 'the colonies' and second by the needs of the marine transport industry, which was searching for means of keeping meat and food produce cool during long ocean voyages. Richard Trevithick, James Harrison and Lord Kelvin all had interests in the maritime industries, and even today, some major ground source heat pump (GSHP) manufacturers have their roots in the production of marine heat pumps (heat pumps designed for boats and ships to extract heat from or dump heat to the sea).

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

5.1 Refrigeration before the heat pump

Historically, of course, refrigeration did not entail the use of heat pumps. In fact, throughout temperate climes, the source of refrigeration was ice, collected in winter and stored until summer. Many manor houses and country estates had so-called icehouses (Buxbaum, 2002) – covered sunken chambers in the earth that were used to store ice collected from frozen lakes during winter (Figure 5.1). Such structures made use of thermogeology: they recognised that the subsurface is significantly cooler than the air during spring and summer and could be used to slow down the rate at which the ice melted. In fact, it has subsequently been shown (Weightman, 2003) that the subsurface nature of the icehouse was not the main determining factor for their satisfactory function. Other factors were more important: good insulation and, above all, the ice had to be kept dry and well drained. Most icehouses allowed meltwater to drain away from the ice blocks to a sump, where it could be extracted. Furthermore,

Figure 5.1 The ice house at Syon Park, Brentford, south-west London. This was constructed in the 1820s by order of the third Duke of Northumberland to replace earlier seventeenth/eighteenth century structures. The ice was tipped into the storage chamber via the roof and packed in straw. The store had a sump draining to waste via a sloping floor. Access to the ice house was via an 'airlock' of two doors. The ice house was not dug vertically downwards to any great extent but was subsequently covered by soil to form a landscaped mound. The ice, being collected from lakes and the River Thames, was impure and not consumed directly. It was, however, used by the aristocrats to cool drinks and produce sorbets and ice creams (by adding salt to form a freezing mixture). *Photo by David Banks*.

BOX 5.1 Applied Thermogeology in Siberia

The Siberians have a tradition of using the subsurface as a natural insulator (in winter) or coolbox (in summer). Indigenous Selkup peoples of the Ob′ valley might dig their houses partially underground to isolate the walls and floor from the winter air temperatures that can plunge beneath –40°C (Figure 5.2).

Many country cottages would have a Ледник (*lednik*) or ice well. Ice would be harvested in the harsh winter and packed away in the ice well, maybe together with fish and meat, to act as a deep freeze, from which frozen goods could be recovered in summer.

The cottage may also have had a Погреб (*pogreb*) or subsurface larder (basically a hole, several metres deep), which would simply have been used to store potatoes, jams and mushrooms. Alternatively, these may have been placed in the underfloor area of the cottage (Подпол – *podpol*). Even today, in Russian cities, it is not uncommon to find Soviet-era blocks of apartments with a central subsurface ‘bunker’ in the courtyard, where residents can store their vegetables and pickles in cool conditions.

Thanks to Igor Serëdkin of Tomsk for much of this information.

Figure 5.2 A Selkup cabin at Narim museum on the River Ob′, Siberia. Note that the cabin is partly buried underground (see Box 5.1). *Photo by David Banks.*

the icehouse may have been constructed using double courses of bricks in the subsurface (a 'cavity wall') to minimise conductive heat loss to the ground (Buxbaum, 2002). The ice may also have been packed in an insulating material such as straw or sawdust. Additionally, it was recognised that air circulation conditions in the headspace above the ice 'pile' would have been of importance.

Such solutions were not unique to the United Kingdom, of course. Similar structures are traditional in Russia, for example (Box 5.1), while the Mediterranean nations harvested ice from the Alps during winter for transport to lower elevations, to be consumed during summer.

5.2 The overseas ice trade

With the development of world trade, some northern lands realised that they had a huge asset locked up in their winter ice. Could it be exported overseas? Weightman (2003) tells the extraordinary story of Frederic Tudor, a Boston American, who dared to believe that it could. He invested, lost and eventually partially recouped astronomical sums of money in setting up a system for harvesting New England lake ice, packing it with sawdust into insulated ships' holds and transporting it to the Caribbean Isles (the first cargo was sent to Martinique in early 1806) and the southern states of the United States. Tudor quickly realised that transport was not necessarily the major problem – an equally important issue was storing the cargo efficiently on arrival. He thus set out to design and construct a network of icehouses in the ports he supplied, although he seems to have been hindered by employing a string of icehouse keepers of dubious reliability. Tudor found that it was unnecessary to construct subterranean icehouses: with his Havana icehouse, he demonstrated that a timber structure, built above ground, with cavity walls filled with sawdust or peat, and with excellent drainage and appropriate ventilation, was equally good (or better) than conventional subterranean structures.

In 1833, he pulled off the unthinkable – he transported over 100 tons of ice from Massachusetts to Calcutta, India, on board the *Tuscany*. The ship departed in May and arrived in September, and the quantity of ice lost to melting during the voyage was believed to be no more than one-third of the original total.

Once Tudor had demonstrated the ice trade to be both viable and profitable, others were keen to join in and the American ice trade expanded hugely, supplying a domestic market from New York to New Orleans and even exporting ice to England and India. The size of ice blocks harvested from the lake varied depending on the distance they had to be transported (heat transfer is related to surface area-to-volume ratio and is thus proportionately less for larger chunks): New York cubes measured 22 in. on each side while those exported to India were larger. By 1879, the American ice industry was estimated at 8 million tons harvested per year, with 5 million tons reaching the consumer (Weightman, 2003). In warm winters, onshore 'ice famines' would force the ice harvesters to chase freshwater icebergs in the Northern Atlantic and Greenland Sea.

The British market for American ice was opened up by Charles Lander in the 1840s and, subsequently, this ice was marketed as Wenham Lake Ice (a brand name originally derived from Lake Wenham, near Salem). It achieved a reputation for high purity, such that it could be consumed directly in drinks such as mint juleps and sherry cobblers. The Norwegians observed this development with interest and believed that they could supply cheaper ice to London than the Wenham Lake Ice Company – all they needed was a catchy brand name. They solved this problem in an uncharacteristically sneaky way – by renaming (at least, according to anecdote) their own unpronounceable Lake Oppegård as 'Wenham Lake' and using the same brand name to befuddle the British public! At any rate, the Norwegians became increasingly dominant in the British ice trade from around 1850, and continued exporting ice to London until well into the twentieth century. At the peak of the trade (around 1900), the United Kingdom's annual import from Norway was around 500 000 tons (Weightman, 2003).

The Norwegian ice was harvested (as in New England) by sweeping the surface of the lake free of debris and then using Nathaniel Wyeth's horse-drawn 'ice plough' to carve deep grooves into the lake surface. Blocks were then levered from these grooves and usually temporarily stored in an icehouse near the lake. In Norway, blocks of ice may have been slid down wooden 'ice railways' from the mountain lake to the harbour for export (Figure 5.3). On arrival in London docks, ice was typically loaded on barges and transported into the heart of London via the Regent's Canal. Ice wharves were located along the canal for the unloading (Figure 5.3) and storage of the ice in 'ice wells'. These phenomenal structures were huge shafts in the London Clay, several metres in diameter and several tens of metres deep. The ice importer Carlo Gatti (1817–1878) dug two ice wells, around 1857 and 1863, at Battlebridge Wharf (now the London Canal Museum in Kings Cross). They were around 12.5 m deep, 10 m in diameter and had a capacity of 750 tons ice per well. As early as 1856, Gatti had signed a contract with the entrepreneur Johan Martin Dahll of Kragerø in Norway for import of ice harvested from lakes on the Toke watercourse and possibly also from glacier falls. Gatti's more established competitor, William Leftwich, had constructed similar ice wells near the canal in Camden. His largest, at 34, Jameston Road, was built in the late 1830s and was reportedly London's deepest at 30.5 m, with a 13-m diameter and a capacity of 4000 tons. It initially received local ice from Camden ponds, but was later supplied from Norway.

As in the large American cities, the London 'Ice Kings' such as Carlo Gatti would have employed 'icemen' to operate a fleet of carts to deliver ice supplies to richer households, hotels and so on. Subscribing households would often have placed the ice in an insulated 'ice box' – a simple refrigerator that could be used to keep fresh foodstuffs cool. The iceman appears to have achieved a degree of notoriety (similar to Benny Hill's naughty milkman in a later decade) as a seducer of housewives, if we are to believe Eugene O'Neill's play *The Iceman Cometh*, set in 1912.

Figure 5.3 (a) Ice ploughing on a Norwegian Lake; (b) an ice railway conveying ice from an inland Norwegian lake to the harbour; (c) and (d) reminders of the now defunct ice trade is found in the place names along the Regent's Canal of London. [*Photos (a) and (b) were kindly provided by the London Canal Museum. Photos (c) and (d) by David Banks.*]

5.3 Artificial refrigeration: who invented the heat pump?

Even before the heat pump was invented, other techniques were employed to produce an artificial refrigerating effect. For example, it had long been known that a volatile liquid, such as ether or alcohol, produces a cooling effect when it evaporates. Think of how spilled vodka cools the skin of your hand. This is because the process of evaporation consumes a significant quantity of heat energy to effect the transition from liquid to gas (for water, this latent heat of evaporation is around 2300 $kJ\,kg^{-1}$ – Box 3.3). Traditional methods of effecting cooling and producing ice slush in the Middle East and India involved evaporation of water from shallow ponds or porous earthenware pots during low-humidity nights (Weightman, 2003). Thus, the earliest forms of artificial refrigeration did not use the compression–expansion cycle at all. As early as 1748, the Scotsman William Cullen had demonstrated refrigeration at Glasgow University, simply by allowing ethyl ether to boil away into a vacuum.

In the early nineteenth century, it was also recognised that the expansion of a pressurised gas could result in a drop in temperature (see Box 3.4). This principle, together with that of the latent heat of evaporation, forms the basis of the heat pump cycle. By the early nineteenth century, the stage was set for its gradual development.

There seems to be a general consensus that the 'grandfather of the fridge' was the American Oliver Evans (1755–1819). He was an engineer who designed many improvements to the textile industry and also an improved high-pressure steam engine. He produced a design for a refrigerator using a compression–expansion cycle in 1805, although the contraption was never built.

Patriotic Cornwalesians claim that Richard Trevithick invented refrigeration – despite all evidence to the contrary. Trevithick was born in Illogan, Cornwall, UK, in the year 1771. He worked at the Cornish tin mines (Wheal Treasury and Ding Dong mines), developing engines for raising ore. He is also famous for having constructed and run what many believe to have been the world's first steam locomotives. Another, less well-documented interest was in the marine transport industry. His claims on the heat pump are rather tenuous, but there is evidence that he became interested in the production of 'artificial cold' during the last 5 years of his life, having been struck by the magnitude of the ice-harvesting economy. In a single letter, written in June 1828, he outlined a method of producing active refrigeration by means of a steam engine and a compression–expansion sequence (in effect, a steam-powered heat pump). It is perhaps significant that the Frenchman, Sadi Carnot, had pointed out, only few years prior to Trevithick's letter, that a heat engine (see Section 4.1) could, in theory, be reversed to provide a heat pump. Trevithick died, near destitute, at Dartford, Kent, in 1833.

In 1834, an American-born resident of London, the prolific inventor Jacob Perkins (1766–1849), filed a US patent entitled *Improvement in the Apparatus and Means for Producing Ice, and in Cooling Fluid* for a refrigerator employing ether in a compression–expansion cycle, apparently using a design similar to Evans'. A prototype was built in London by John Hague (Biblioteca ETSEIB, 2003). Details can be found in Perkins English patent for the device, published in 1837.

It was left to the Scottish-born Australian, James Harrison, and the American, Alexander Twinning, to run with Perkins' ether vapour compression concept and construct commercially functioning refrigerators during the 1850s. Harrison constructed such ether-based vapour compression ice-making machines in Australia in the early 1850s and was granted a patent in 1855 (Richardson, 2003). His machine utilised a 5-m flywheel and produced some 3000 kg of ice per day. Harrison applied his machines to the meat and brewing industries and, in 1860, one was installed at the 'Glasgow and Thunder' brewery in Bendigo, Australia. By 1861, around 12 of Harrison's machines were in action, including one in a petroleum refinery at Bathgate, Scotland (Biblioteca ETSEIB, 2003). By the late 1870s, refrigeration heat pumps were being installed on ships carrying meat between Australia, the United States and Britain (Heap, 1979).

In the 1840s, the American medical doctor John Gorrie (1803–1855) had also been experimenting with the compression and expansion of air to produce ice. The ice was then used to chill the air circulating in wards for malaria and yellow fever patients. His contraption involved compressing air, resulting in a temperature increase. The compressed air was then cooled by circulating water in pipes, which served to draw heat out of the compressed air. The air was then allowed to expand, resulting in the formation of ice. He was awarded British and American patents in 1850 and 1851, respectively, and is regarded by many as the 'father of air conditioning'.

Meanwhile, in 1857, Peter Ritter von Rittinger had applied the principles of the heat pump cycle to industrial processes at the salt works of Ebensee in Austria. Green (2007) reported that water power was used to compress steam, raising its temperature for utilisation in evaporating brine for salt extraction.

The final key players in the development of refrigeration were the French Carré brothers. As early as 1850, Edmond had figured out the principles of absorption refrigeration (Boxes 4.5 and 5.2) using water and sulphuric acid. His brother, Ferdinand, perfected an ammonia–water absorption refrigerator in 1858–1859. This system was at that time particularly suited to the large-scale production of ice, although the toxicity of ammonia rendered it dangerous to use.

At around the same time, we note that William Thomson, who later became Lord Kelvin, was developing the theoretical foundations for the heat pump. In 1852, he suggested in his article *On the economy of heating or cooling of buildings by means of currents of air* that an open air-cycle heat pump concept could be used to absorb heat from the atmosphere and deliver it at a higher temperature to a building, resulting in a supply of heat greater than the mechanical energy input to the system (the building that appears to have inspired him was Queens College, Belfast, Northern Ireland). He noted that a vapour compression–expansion cycle could also do the same job (Thomson, 1852). At the time, however, there was little practical interest in this idea. The only flicker of interest came from British colonists in India, who were intrigued by the possibilities offered by heat pumps for *cooling* their public buildings (Thomson, 1852; Sumner, 1976a,b; Curtis, 2001).

5.4 The history of the GSHP

The idea of using Kelvin's concept of the heat pump to extract environmental heat lay dormant for some time. The first patent for a GSHP was granted in Switzerland

BOX 5.2 Szilárd and Einstein's Refrigerator

In 1926, Leó Szilárd (1898–1964) and Albert Einstein (1879–1955) developed a new form of absorption refrigerator. Its novelty lay in the fact that it contained no moving parts and required no external primary energy input other than heat. It also combined the concept of a sorption cycle with a more conventional refrigerant cycle. It uses three fluids: water as an absorbent, ammonia as a readily soluble sorbed component and insoluble butane as a refrigerant (this three-fluid concept had, in fact, already been pioneered by Baltzar von Platen and Carl Munters, in Stockholm in 1922). On the evaporator side, the butane evaporates and mixes with ammonia to form a mixed vapour. On the condenser/absorber side, ammonia absorbs into water. This reduces the total gas pressure and causes mixed vapour to flow from the evaporator side. As the overall pressure is maintained, while ammonia is being removed by sorption, the overall effect is to increase the *partial pressure* of butane towards the total system pressure at the condenser, where the butane can thus condense and release its load of latent heat. The external heat source regenerates and desorbs ammonia from the water. The patent was eventually sold to Electrolux. It was never mass-produced, but recently, the idea has been revived, constructed and demonstrated to be rather efficient.

In his spare time, Szilárd discovered the nuclear chain reaction and proposed the US Manhattan Project. Einstein dabbled in quantum theory and relativistic concepts of space–time and gravity.

to an engineer named Heinrich Zoelly in 1912 (Ball *et al.*, 1983; Spitler, 2005; Kelley, 2006), although his idea lay unused for years. By the late 1920s and 1930s, groundwater was being abstracted from wells on Brooklyn and Long Island and being used for air conditioning, although Kazmann and Whitehead (1980) did not clarify whether this was by the use of heat pumps or simply passive cooling.

During the mid-to-late 1920s, T.G.N. Haldane constructed an experimental heat pump installation for his home in Scotland (Heap, 1979; see also Haldane, 1930), utilising both outdoor air and mains water as a heat source (a possibility that deserves consideration by water utility companies). His system appears to have been rather sophisticated, with an electrically powered compressor (the electricity being hydroelectrically generated) and ammonia refrigerant. It supplied low-temperature waterborne central heating and domestic hot water. Bhatti (1999) noted that Haldane constructed another heat pump system for his office in London. Sumner (1948) also cited Haldane's pioneering work in Britain and draws attention to another small heat pump installation by one S.B. Jackson in 1938.

During the late 1930s, several environmental heat pump systems were installed for heating buildings in Zurich, Switzerland (Heap, 1979; Schaefer, 2000; Zogg, 2008).

These included several systems based on surface waters, such as the R-12 refrigerant heat pump system for the city hall, extracting heat from the water of the River Limmat and providing a nominal heating capacity of at least 70 kW and a cooling capacity of 55 kW (IEA, 2001a; Zogg, 2008).

The date of the first true GSHP installation is somewhat hazy. Heap (1979) claimed that 15 commercial environmental heat pump installations were in operation in the United States in 1940 – some of them using well water as the heat source. Furthermore, Arnold (2000) tells us of a building in Salem, Massachusetts, heated and cooled by well water in 1935. Zogg (2008) suggested that some of the 35 heat pumps installed in Switzerland between 1938 and 1945 were based on groundwater, as well as river/lake water and waste industrial heat.

We do know, however, that, in around 1945, Robert C. Webber of Indianapolis, USA, was recovering the heat rejected from the condenser of his cellar deep-freezer and using it to heat his house and domestic water. He was so enchanted with this idea that he constructed what may be one of the earliest direct-circulation closed-loop GSHP systems to replace his coal-fired furnace (Kelley, 2006). He built a dedicated heat pump, using circulating Freon refrigerant to extract heat from 152 m of coiled copper heat exchange pipe buried at 1–2 m depth in three trenches totalling 36 m. The system delivered warm air at a temperature of 43°C and achieved a COP_H of 3.61. It could be reversed in summer to provide active cooling (Crandall, 1946; Sanner, 2001, 2005; IGSHPA, 2008; Zogg, 2008).

At around the same time, Britain's great visionary proponent of heat pumps, John Sumner, assisted by (the probably grossly under-recognised) Miss M. Griffith, flew the flag for the technology. In 1945, Sumner (Box 5.3) constructed a full-scale ground-coupled heat pump system in Norwich, based on water from the River Wensum (Figure 5.4). In 1948, the British philanthropist millionaire, Lord Nuffield, employed Sumner as a consultant to build 12 heat pump systems using 'ground coils' for a 2-year period of testing in selected houses. These achieved a heat output of 9 kW_{th} each and a COP_H of around 3 (Sumner, 1976a; Curtis, 2001). Sumner was also involved in a project in 1951 to heat London's Royal Festival Hall using a two-stage gas-powered heat pump, based on compressors modified from Merlin aircraft engines, using the River Thames as the heat source. The 2.5-MW project is sometimes described as a bit of a failure, although it is not always made clear that the problem was that it actually delivered too much heat (Neal, 1978; Heap, 1979; Tovey, *pers. comm.*), with a COP_H of 2.5–3 and supplying hot water at up to 82°C!

In the early-to-mid 1950s, Sumner also installed a closed-loop ground source heat system at his own house (Sumner, 1976a; Neal, 1978). The ground loop was initially constructed using copper pipe buried at around 1 m depth and filled with circulating antifreeze (later supplemented with an additional lead piping coil). Sumner later experimented with black plastic pipe laid on the ground surface, to absorb solar energy. Sumner's heat pumps may have been technical triumphs, but they proved not to be a marketing breakthrough. Britons showed next to no interest in the prototypes; at that time, abundant coal seemed a cheap and limitless source of energy. In

BOX 5.3 John Sumner's Norwich Heat Pump

John Sumner was the City Electrical Engineer for Norwich, UK. He is also known as a prophet for the technology of environmental heat pumps in the United Kingdom. In 1940, the Norwich City Council Electrical Department built new premises in Duke Street, on the banks of the River Wensum. It was originally intended to use a heat pump to heat the building, but in the wartime climate, there was a lack of appetite for such an 'innovative' venture and coal boilers were installed. It was only after the Second World War, in 1945, that a decision was made to replace the coal boilers with a heat pump based on abstraction of river water from the Wensum. The original central heating system was designed for hot water at 70–80°C but, with minor modifications, was re-jigged to run at a temperature of 50–55°C. The heat pump itself was largely (by Sumner's own admission) cobbled together out of salvaged parts in the post-war 'austerity' economy, and used an SO_2 refrigerant. Even this non-optimal design, running on cold Wensum water, achieved a seasonal performance factor (SPF) of 3.45. Sumner was confident that, with a well-tailored design, this could easily be increased to >4. The pump ran at an average thermal delivery of 5 therms h^{-1} (147 kW) and a peak capacity of 8 therms h^{-1} (234 kW). Sumner compared the economy of the heat pump with the conventional coal-fired system and considered that the heat pump represented the clearly cheaper option, even taking into account the larger capital cost of the heat pump. He also considered the non-monetary benefits and recognised the lower pollution from the heat pump and its attractiveness in view of the finite resource of coal. He recognised clearly, however, that the success (or otherwise) of the heat pump technology had a political dimension, inextricably linked with the national coal economy of the time. Sumner even performed a calculation akin to that presented in Table 4.3 of this book, and concluded that, to compete efficiently with conventional heating by coal, an electrically powered heat pump needed to have a COP_H of at least 3.3. Sumner also recognised the desirability of some form of thermal storage in the system, such that off-peak electricity could be used to run the heat pump at night, with the heat being stored for use during daytime.

Sumner's findings are presented in a hugely entertaining paper read to the Institution of Mechanical Engineers (Sumner, 1948). Even more interesting is the discussion following the paper's presentation, which could be a replica of the debates still taking place today. Sumner's audience divided itself clearly into 'believers' and 'sceptics'. The 'believers' immediately understood the potential of the heat pump and proposed innovative solutions based on the United Kingdom's canal network, and on the groundwater supplied from the wells of London's Metropolitan Water Board (this suggestion was from Messrs Lupton and Westbrook). The 'sceptics' queried the economy of the heat pump compared with coal: they asked awkward (but cogent) questions such as 'Would the heat pump have been more expensive if it had been properly constructed rather than cobbled together from spare parts?' and 'Did Mr Sumner include the costs of his time in the capital cost of the heat pump?'

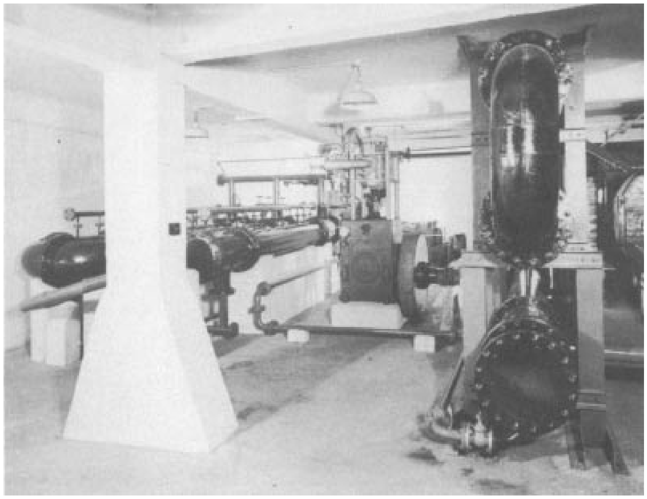


Figure 5.4 John Sumner's Norwich heat pump, from 1946 (see Box 5.3). Not so much a unit as a whole room! *After Sumner (1976a).*

particular, Sumner (1976a,b) was not quite able to contain his bitterness towards the Electricity Boards, whom he suspected of not being enthusiastic enough about a technology designed to cut their sales by 60%.

Interest in GSHPs in Britain may have been low, but elsewhere, the story was different: a large groundwater-based open-loop heating and cooling scheme was installed at an office block in Portland, Oregon, in 1948 (ASME, 1980; Arnold, 2000; Box 5.4) as a direct result of the British and Swiss experiences. Nevertheless, it took the OPEC oil crisis of the 1970s to really focus people's attention on non-fossil energy sources. The Swedes (and, later, the Swiss, Austrians and Germans), in particular, took up the GSHP torch. Switzerland's native natural energy resources were limited to hydroelectric power, and there was thus considerable motivation to use that as efficiently as possible (Sumner, 1948). Likewise, Sweden, which had very few sources of native power, was actively developing new atomic reactors and was looking for efficient means of heating homes by electricity rather than paraffin. The post-1970 period was characterised by the development especially of closed-loop systems employing polyethene, rather than metal, pipe. The first borehole-based vertical closed-loop systems were reported from Germany and Switzerland in around 1980 (Sanner, 2001, 2006) and from Norway in 1985 (Midttømme *et al.*, 2008).

5.4.1 Sweden and Switzerland

By the late 1970s, around 1000 GSHPs had been installed in Sweden (Rawlings and Sykulski, 1999). Between 1980 and 1986, 50 000 GSHPs were installed in Sweden, with

BOX 5.4 The Equitable Building Heat Pump System of Oregon, USA

The Equitable Building ground source heating and cooling scheme (ASME, 1980; Arnold, 2000) was constructed between 1946 and 1948 in Portland, Oregon. It was not only the first major installation of its kind in a commercial building in the United States, but also a genuinely ground-sourced scheme, being based on groundwater abstraction from wells (an open-loop scheme). The building in question had 14 storeys and had no opening windows. It therefore had not only a heating demand in winter, but also an air-conditioning (cooling) demand.

The Chairman of the Board of the Equitable Savings and Loan Company was the sweetly named Ralph H. Cake. He had heard of the practical application of heat pumps in the United Kingdom in around 1946 (surely this must have been via contact with John Sumner?). Cake insisted on a ground source heat pump solution at Equitable, and the air-conditioning and heating scheme was eventually designed by J. Donald Kroeker.

The four water-to-water heat pump units (using Freon 11 and Freon 113 as refrigerants) utilised groundwater pumped at 17–18°C from two wells of depth c.46 m and yields of 195–450 gpm (12–28 $L\,s^{-1}$). The used water was re-injected to a single 155-m-deep well, at a nominal temperature of c.14°C.

The building had separate heating and cooling circuits. The pumped groundwater passed over the evaporator of the heat pump array and was chilled. The chilled groundwater effected cooling via fan coil units before being bled off to the re-injection well. The heating circuit was a closed circuit, passing through the condenser of the heat pump array and utilising the heat extracted from the groundwater. Condenser units could be coupled in and out, in response to heating demand. Moreover, the system was designed to recover heat from waste ventilation exhaust air and to precondition incoming air. Thus, the heat to the building could largely be considered a 'by-product' (ASME, 1980) of the cooling load (heat recovered from cooling). No auxiliary heating was required for the building – indeed, the heat pump capacity required to provide the necessary cooling load was far in excess of the heating requirement.

As of 1980, the scheme was still running and was satisfying energy efficiency standards in force at that time. One change that had taken place since its original construction was that the direct circulation of well water into the heat pumps had been replaced by the use of prophylactic heat exchangers (see Section 7.3.2), separating the groundwater circuit from a building loop. This was found to be necessary due to scaling and corrosion damage attributable to the groundwater.

a total installed capacity of some 500 MW. In the period from the mid-1980s to mid-1990s, development slowed down due to relatively low energy prices. From 1995 onwards, the market had grown steadily, thanks in large part to a system of subsidies initiated by the Swedish government, until by the early 2000s, over 90% of all new Swedish dwellings were reportedly being installed with heat pumps, of which at least three-fourths were ground-sourced (Bouma, 2002; Lund *et al.*, 2004; Nowak, 2006). By 2008, SEA (2010) reported that around 40% of the country's total stock of detached houses was heated by some form of heat pump. Ground source heat is believed to supply up to 15% of Sweden's total 100 TWh year^{-1} heating and cooling demand (Hellström, 2006; Lund *et al.*, 2008; Bjelm *et al.*, 2010). In the mid-2000s, around 30–40 000 new GSHP systems were being installed each year. By 2008, almost 350 000 GSHP units (c. 4.2 GW_{th} installed capacity) were believed to be operational (Hellström, 2006; Lund *et al.*, 2008). The majority of such systems are small (average 10 kW), and 80% are believed to be closed-loop systems installed in boreholes to an average of 125 m depth, typically using 40-mm diameter, 6.3-bar polyethene pipe (Lund *et al.*, 2004; Hellström, 2006). The GSHP market in Sweden is relatively mature: there are recent signs that air source heat pumps are becoming increasingly popular, compared with GSHPs, partly due to their low capital cost and their applicability to increasingly well-insulated houses.

In Switzerland, in 2000, one-third of all newly built single-family homes utilised a heat pump, and of these, around 40% were ground-sourced (Rybach and Sanner, 2000). Vertical, borehole-based closed-loop systems have gained in popularity hugely in recent years and, as in Sweden, account for the majority (65–75%) of the GSHP-installed capacity, whereas horizontal closed loops account for less than 5% and groundwater-based open-loop systems 30% (Lund *et al.*, 2004). Some of the groundwater-based systems utilize drainage water from mountain tunnels. By 2008, between 30 000 and 44 000 GSHP units had been installed. In fact, there is, on average, one GSHP installation per 2 km^2, reportedly the highest density anywhere in the world (Rybach and Sanner, 2000; Lund *et al.*, 2008). One major factor in the popularity of GSHPs in Switzerland is believed to be the willingness of utility companies to develop GSHP schemes and sell space-heating energy to subscribers at a competitive rate per kilowatt-hour; in other words, the utility company bears the risk and absorbs the initial capital outlay of the scheme (Lund *et al.*, 2004).

5.4.2 North America

The heat pump market in the United States took off for different reasons: Sumner (1948) suggested that they were actively promoted by power utilities seeking new markets for consumption of excess power generation capacity. As Sumner was writing his book *Domestic Heat Pumps* in 1976, he was able to relate that there were over 1 million domestic heat pumps used for heating or cooling in the United States (admittedly mostly air-sourced or air-conditioning units), and that he had recently stayed in a hotel where 400 heat pump units could provide guests with heating or cooling at the flick of a switch.

The GSHP also expanded rapidly in the United States, with some 28 000 GSHPs being installed annually in 1994, rising to 50 or 60 000 units per year by 2008 (Bouma, 2002; Lund *et al.*, 2008). Lund *et al.* (2004) estimated that, of this total, around 46% are vertical closed loop, 38% horizontal closed loop and 15% open loop. A total of up to 750 000 units are believed to be in operation, with strong growth in the governmental and public (e.g. schools) sectors (Bouma, 2002; Lund *et al.*, 2008). Admittedly (except in the northern states), such GSHPs are often dominated by a cooling demand rather than a heating demand and are thus typically rejecting heat to the subsurface. A typical residential cooling capacity is around 10.5 kW. In order to demonstrate that GSHPs are not solely for woolly jumpered environmentalists, Lund *et al.* (2004) noted that even ex-President George W. Bush had installed a GSHP at his Texan ranch.

North of the border, in Canada, around 55 000 GSHP systems are believed to be operating, with about 5% of these in the commercial sector. The total is believed to represent around 1.1 GW_{th} capacity, with around 1% of this being derived from schemes based on water from abandoned mines. The average GSHP market growth rate is around 13% (Lund *et al.*, 2010), although annual growth rates of around 60% were recorded for the period 2006–2008. Around 15 000 GSHP units are currently being installed per annum, of which around 50% are horizontal closed loop, 35% vertical closed loop and the remaining units water-based open loop (Canadian Geoexchange Coalition, 2010).

5.4.3 Meanwhile . . . back in blighty

What was going on in Britain, which had played such a crucial role in the theoretical and practical development of heat pumps, while tens of thousands of installations were being completed elsewhere in Europe and North America? A survey in 1999 (Rawlings and Sykulski, 1999) could document a total of 10 GSHP systems in the United Kingdom! In the early 2000s, the market for GSHPs had expanded tremendously in the United Kingdom: Britain is still near the bottom of the European GSHP league table (Table 5.1), but it possesses one of the most rapidly growing markets. In 2001, for example, some 150 new GSHP systems were commissioned and the market was growing at over 100% per annum. Today, the total is not well known (as there is no obligatory form of licensing or registration of closed-loop GSHP systems), but it is widely believed that some 10 000 GSHP systems have been installed. Lund *et al.* (2010) reckoned that the United Kingdom's installed direct geothermal capacity (almost wholly GSHP) is around 190 MW_{th}.

We have seen how the development of the GSHP market differs hugely from country to country, even within Europe. In Britain, GSHP technology has had relatively low penetration but high growth. In Sweden, the market is tending towards saturation, and one can speculate what proportion of Swedish sales is simply replacing first-generation units.

Table 5.1 European GSHP market status in 2001 (after Bouma, 2002), in 2005 (after EREC, 2006) and in 2008/2009 (after EurObserv'ER, 2010). It is clear that there are some potential inconsistencies in the data: they do, however, give some impression of the diversity in GSHP uptake in Europe.

Country	Annual GSHP sales (units/ year) 2001	New installations 2005	Annual GSHP sales (units/ year) 2008	Annual GSHP sales (units/ year) 2009	Total installed GSHP 2009
United Kingdom	150	750	c. 5000	3980	14330
Ireland			2095	786	10909
Czech Republic	350	4000	2203	1959	11127
Poland	500	1465	c. 1000	c. 1000	11000
Norway	650				
Switzerland	2800				
Italy		13000 (?)			12000
France			21725	15507	139688
Germany	3600	25486	34450	29371	179634
Sweden	27000	61350	25138	27544	348231
Europe (total)	41000				
EU-25 or EU-27		140144	114452	103155	903012

The critical factors in uptake of GSHP technology from country to country are likely to be

- access to and cost of conventional fossil fuels;
- cost of electricity;
- level of subsidy provided by government or utilities companies for installation of GSHPs;
- marketing and packaged solutions offered by power utilities;
- level of awareness of GSHP technology;
- planning constraints on non-renewable energy technologies;
- prioritisation of environmental (especially CO_2 emission) concerns at both political and individual levels.

5.5 The global energy budget: how significant are GSHPs?

Statistics for global use of energy are notoriously slippery things. In addition to the obvious difficulties in collating statistics from nations as diverse as China, Nigeria and Luxembourg, it can be difficult to distinguish between figures for actual energy delivery and figures for installed potential capacity. For GSHPs and other small-scale alternative energy sources, the problems become particularly acute as, in many countries, nobody is really keeping track of GSHP installations (in the United Kingdom, closed-loop systems currently need no licence and may not require specific planning permission). Where statistics are available, it is not always clear whether they refer to GSHP *units* (i.e. heat pump modules) or *systems* (which may use multiple heat pump units). Therefore, take the following statistics with a pinch of salt and recognize that there may be significant error margins on the figures cited.

Bouma (2002) claimed that there may be as many as 100 million heat pumps (not just GSHPs) installed worldwide, with a total annual output of 1300 TWh. (Here we face our first query – to what extent does this figure include standard air-conditioning devices?) It is further claimed that this results in a global saving in annual CO_2 emissions of 0.13 Gt, in the context of a global total CO_2 emission of 22 Gt (Bouma, 2002).[1] (Here again, we must ask: if heat pumps are being used for active cooling, in what sense are they saving energy?)

In 1999, Rawlings and Sykulski estimated the total number of GSHP systems installed worldwide as no less than 400 000, while 5 years later, Lund *et al.* (2004) placed the figure at 1 100 000 (600 000 of which were in the United States and 230 000 in Sweden). Lund *et al.* (2004) estimated the worldwide total GSHP installed capacity to be ≈12 GW_{th}, of which the majority (6.3 GW_{th}) is in the United States, with 2.3 GW_{th} in Sweden.

By 2009, several estimates place the number of GSHP units installed in the European EU27 nations alone at around 1 million (representing almost 11 GW_{th} installed capacity). The European Union's (EU) annual market in sales was placed at around 100 000 GSHP units, with some 30 000 of these sales in Germany (European Geothermal Energy Council data; EurObserv'ER, 2010; see Table 5.1).

The most recent global survey by Lund *et al.* (2010) makes the following estimates of direct geothermal energy usage:

- global installed direct geothermal capacity = 50.6 GW_{th}, an increase of 79% over 2005;
- global direct geothermal usage = 122 TWh_{th} or 438 000 TJ per annum, representing a carbon saving of around 148 million tonnes of CO_2.

The figures above exclude any delivery of space cooling. Of these figures, around 70% of the installed capacity (i.e. 35.2 GW_{th}) and 49% of the usage (i.e. 215 000 TJ $year^{-1}$) is delivered by means of GSHPs. The remainder represents both 'proper' high-temperature geothermal space heating (e.g. Iceland) and lower-temperature passive heating solutions (e.g. ice melting). The number of installed GSHP units (based on a 12 kW average size) is thus cautiously estimated at 2.94 million.

Let us look at more mainstream statistics (Table 5.2), culled mainly from two sources, the UNDP (2004) World Energy Assessment and British Petroleum's Statistical Review of World Energy (BP, 2010). It would appear that the worldwide installed capacity of geothermal energy has increased by more than fourfold over the past decade. Note the difference between the delivery of heat (thermal energy = GW_{th}) from geothermal sources and delivery of electrical energy (GW_e) from turbines in power stations powered by high-temperature geothermal sources.

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[1] Henson (2006) cited a figure of 26 Gt CO_2 for 2002.

Table 5.2 Global energy status today: energy delivery as installed capacity and average usage, by source. The year for which the data apply is provided in parentheses. Data compiled from [1]BP (2010), [2]UNDP (2004), [3]Lund *et al.* (2010) and [4]Earth Policy Institute Data Centre (http://www.earth-policy.org).

	Installed capacity (2000–2001)	Usage (2001) TWh year^{-1}	Installed capacity (2009)	Usage (2009) TWh year^{-1}
GSHPs			35.2 GW_{th}[3]	60[3]
Geothermal (thermal)	11 GW_{th} (2001)[2]	55 (2001)[2]	50.6 GW_{th}[3]	122[3]
Geothermal (electricity)	8.2 GW_e (2000)[1] 8 GW_e (2001)[2]	53 (2001)[2]	10.7 GW_e[1]	
Solar photovoltaic (PV)	1.4 GW_e (2000)[1] 1.1 GW_e (2001)[2]	1 (2001)[2]	22.9 GW_e[1]	
Solar thermal	57 GW_{th} (2001)[2]	57 (2001)[2]		
Wind	18.4 GW_e (2000)[1] 23 GW_e (2001)[2]	43 (2001)[2]	160 GW_e[1]	
‘Modern’ biomass	210 GW_{th} (2001)[2] 40 GW_e (2001)[2]	1700 (2001)[2]		
Traditional biomass		10800 (2001)[2]		
Hydroelectric	715 GW_e (2001)[2]	2652 (2000)[1] 2700 (2001)[2]		3272[1]
Nuclear	349 GW_e (2000)[4]	2582 (2000)[1]	373 GW_e[4]	2698[1]
Natural gas		25300 (2000)[1] 25100 (2001)[2]		30860[1]
Coal		27200 (2000)[1] 26300 (2001)[2]		38130[1]
Oil		41400 (2000)[1] 40800 (2001)[2]		45150[1]

Table 5.2 compares these statistics with installed capacities and actual usages of other alternative and conventional energy sources. We will see that, in terms of installed capacity, GSHPs and geothermal energy are more significant than solar photovoltaic (PV) power generation, but appear to lag some way behind solar thermal and wind. If we divide the energy delivery (GWh year^{-1}) by the installed capacity (GW), we obtain the number of peak capacity operating hours per year. We can see that solar technologies clock up less than 1000 h year^{-1} (inefficient in cloudy weather and night-time), wind averages less than 2000 peak hours per year (weather-dependent), while GSHPs are assumed to run at c. 2400 peak load hours per year. Nuclear reactors run more or less continuously at more than 7000 h year^{-1}.

If we consider non-fossil-fuel energy supply, however, we see that (in contrast with a decade ago) wind, geothermal/ground source heat and solar energy are definitely significant when compared with the mainstream low-carbon sources of energy (nuclear and hydroelectric, weighing in at 680 GW generated on average). But all of these only account for a small proportion of the energy consumption represented by the three fossil fuels: gas, oil and coal, which provide energy at a combined energy consumption of c. 13 TW on average throughout a year.

5.6 Ground source heat: a competitor in energy markets?

One view of ground source heat (which Table 5.2 appears to promulgate) is that it is an alternative, renewable source of heat energy, competing in the energy market with other 'green' energy sources such as wind power and solar thermal systems. I would argue that this is not a helpful viewpoint – it's not a competition! All these forms of renewable energy technology do different things:

- Wind, hydroelectric and solar PVs are good at generating electricity.
- Solar thermal is good at delivering limited quantities of high-temperature domestic hot water.
- GSHPs excel at delivering lots of low-temperature space heating and cooling.

We should, after all, remember that GSHPs do not work by themselves. Most GSHPs require an electricity supply and should be seen as a *complementary technology* to electricity, enabling electric power to be used more efficiently. That electric power may, in itself, be generated by low-carbon or 'green' technologies, such as hydroelectric or nuclear. In a sense, one can thus regard GSHPs as an integral part of a national strategy for electricity generation by nuclear power or hydroelectric turbines, as has been the case in Sweden and Norway, respectively.

One could even conceptually envisage a GSHP being powered by an array of wind turbines or photoelectric cells although, in practical situations, the unreliability and limited capacity of these sources means that the electricity generated by them would not be used directly. Typically, a connection to a national or local electricity grid would be used to supplement the supply and to buffer weather-related and diurnal fluctuations in generating capacity. The electricity generated by the solar cells or turbines would typically be sold to a utility company, and repurchased to power the heat pump. Furthermore, we will see in the case studies of Chapters 7 and 14 that some ground source heat schemes operate in parallel with other technologies, such as solar thermal cells and conventional heating, to provide total solutions to domestic hot water and space heating. I would encourage readers not to just regard ground source heat as an alternative, 'green' technology *competing* with other conventional and renewable energy sources: rather, it is a technology that *complements* them and that can even be attractive to the most diehard petrol-head!

6

Ground Source Cooling

Keep cool and you command everybody.

Louis de Saint-Just (1767–1794)

6.1 Our cooling needs in space

In Box 1.3, we saw that *homeostasis*[1] – that is, maintaining a constant environment that is conducive to survival – is one of our most fundamental human needs. In northern Eurasia, our historical struggle for homeostasis has largely been dominated by a need for heating, in order to keep warm and survive the winter. For thousands of pensioners, this historical struggle is still bitterly relevant, and the term *fuel poverty* has been coined to describe their situation.

In southern Europe, Africa and the tropics, there may still be a heating need in some seasons and in night-time, but there may also be an overwhelming need to deal with excess heat during the day. Excess heat does have the ability to significantly affect

[1] The term *air conditioning* is usually associated with artificial cooling. Strictly speaking, the term simply describes a regulated climate with temperature and humidity varying within prescribed limits. Air conditioning thus encompasses heating and humidity control in addition to chilling. In other words, *air conditioning* means *homeostasis*.

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

human productivity, as anyone who has tried to work in a non-air-conditioned office in Iraq or Sudan will testify.

For millennia, the inhabitants of such countries have had little or no access to evaporative or mechanical cooling technologies and have had to rely on other strategies, including avoiding the heat (take a siesta), wearing sensible clothing, seeking shade and designing buildings with a high thermal mass. In extreme climates, such as the fringes of the Sahara, communities have even resorted to building extensive underground houses: the underground rock temperature is close to the annual average air temperature, *much* cooler than the temperature on a hot summer's day (Figure 3.14).

6.2 Scale effects and our cooling needs in time

Some of you may have wondered why the words 'small' and 'furry' are often found together when describing animals. An animal is a clump of cells, each cell respiring and giving off heat. The amount of heat an animal produces is thus related to its volume. The rate at which it can lose this heat is, however, largely governed by its surface area. Small animals have a small volume but a relatively large surface area – in other words, they have a high *surface area-to-volume* ratio. Their main challenge is to keep warm – they adopt survival strategies such as evolving furry, insulating coats, hibernating in winter or curling into balls (to reduce their area-to-volume ratio). Large animals, on the other hand, have the opposite problem: they have a huge volume (many respiring cells) and a relatively small surface area. Their main homeostatic challenge is losing heat and keeping cool. Their survival strategies include increasing their surface area (the elephant's ears, which serve as radiator membranes), wallowing in mud or water (the hippo) and avoiding massive furry coats (apart from the mammoth, which lived in very special climatic conditions).

As it is with animals, so it is with buildings. A mediaeval house in northern Europe, made of wattle and daub, or even a small brick nineteenth century house like my own, has an important requirement for heating. Inhabitants of such dwellings need to ensure that they have a good stock of firewood, coal, or a connection to mains gas to ensure winter survival. Even today, most single dwellings in northern and temperate Europe have no need of air conditioning in summer – it is regarded as a luxury item.

However, in the twentieth century, we have constructed ever-bigger buildings, filled with cells (rooms) that generate heat (humans respiring,[2] computers pouring out waste heat from their processors[3]). And these bigger buildings have decreasing surface area-

[2] Every adult human produces, on average, around 70–100 W of heat.

[3] Rolf Landauer deduced, in 1961, that information processing is accompanied by a mathematically calculable minimum dissipation of heat (or, strictly, an increase in entropy). Computer motherboards can produce between 25 and 140 W of heat.

Figure 6.1 Air conditioners installed on the Central Universal Department Store (TSUM) in Tomsk, Siberia. While winter temperatures can drop below –40°C, summers can be quite balmy. *Photo by David Banks.*

to-volume ratios. Their demand for artificial cooling increases. Today, many large buildings in the United Kingdom and Scandinavia will have some cooling demand, even in the depths of winter. Figure 6.1 shows a department store in the Siberian city of Tomsk, with huge batteries of air conditioners to keep it cool during summer.

The heating and cooling demands of large offices, hotels and hospitals can be quite complex. During winter, heating will need to be switched on before the working day starts, to ensure that the building is warm enough for the earliest workers when they arrive. As the day progresses, more hot humans arrive, more computers and electronic equipment are switched on, more sun shines in through the building's glass facade and, before long, a cooling need becomes dominant.

To complicate matters still further, different parts of large buildings may have different heating and cooling requirements. In a university department, a south-facing computer laboratory may contain tens of sweaty students and computers (all generating heat) and may receive full sunlight for much of the day. It may thus have a significant cooling requirement, even in spring. On the top floor, a philosophy professor may have an office on the north side of the building. He may not know how to use a computer and may spend hours sitting immobile, pondering the significance of platonic idealism in the politics of sixth century Byzantium. In other words, he might get a bit chilly and have a requirement for heating.

6.3 Traditional cooling

Without any *active* cooling, the traditional buildings of areas such as North Africa, Asia Minor and Greece show us how to keep cool by using certain construction techniques. These techniques typically involve some of the following features:

1. High thermal mass (thermal inertia). The buildings have thick walls of stone, adobe or brick. The building's structure retains the coolness of the previous night and only warms up gradually during the day. Conversely, such buildings retain the day's store of heat into the night.
2. Limited window area to constrain the entry of direct solar radiation and to maximise indoor shade.
3. In some cases, the use of air shafts or solar chimneys to encourage the circulation of fresh air through the house.

A feeling of comfort can also be obtained in a hot climate by the use of fans, whether they be handheld, manually powered (such as the Indian *punkah*) or modern electric fans. Fans do not, in themselves, reduce the temperature of the air. They do, however, promote motion of air within a space: they move warm air away from a source of heat and moisture (whether it be the human skin or a computer's CPU) and replace it with (hopefully) drier, cooler, fresher air. Thus, moving air often increases the rate of evaporation of moisture from the skin, lowering skin temperature (loss of latent heat) and increasing comfort.

Wind towers, wind catchers or *malqafs* have been a feature of architecture in the Middle East (and especially Persia) since antiquity (Aryan *et al.*, 2010). These are towers, integrated into the building, with open ports at the top to allow the entry or exit of an air current. Their function is to promote the flow of natural air through a building (and, as such, do not really cool the air but provide comfort for the inhabitants). They can function in several ways: in their simplest form, the open ports at the top of the tower face into and 'catch' the prevailing wind, channelling it through the building to some exit port. Other wind towers function as solar chimneys: they warm the air within the tower, causing it to rise and thus inducing a convection-driven draft through the house.

A final variant, which genuinely does provide a 'ground-sourced cooling' of the air, supplemented by a form of evaporative cooling (see below), involves a wind tower with the open port facing away from the prevailing wind, thus creating an underpressure. This draws outdoor air down an underground shaft, bringing it into contact with the cool waters of a groundwater-fed *qanat* (a kind of horizontal well) or water storage cistern, promoting evaporation and humidification of the air. The moist, cool air is then drawn up through another tunnel into the building, to exit via the wind tower (Bahadori, 1978; Khani *et al.*, 2009; Figure 6.2).

6.4 Dry coolers

Dry coolers are simply water-to-air heat exchangers that are mounted outside. A warm fluid (e.g. water from an industrial process or the condenser of a heat pump in cooling mode) is directed through this heat exchanger, which is often a radiator grid mounted in front of a fan (Figure 6.3). In a sense, therefore, a dry cooler is a fan-coil unit

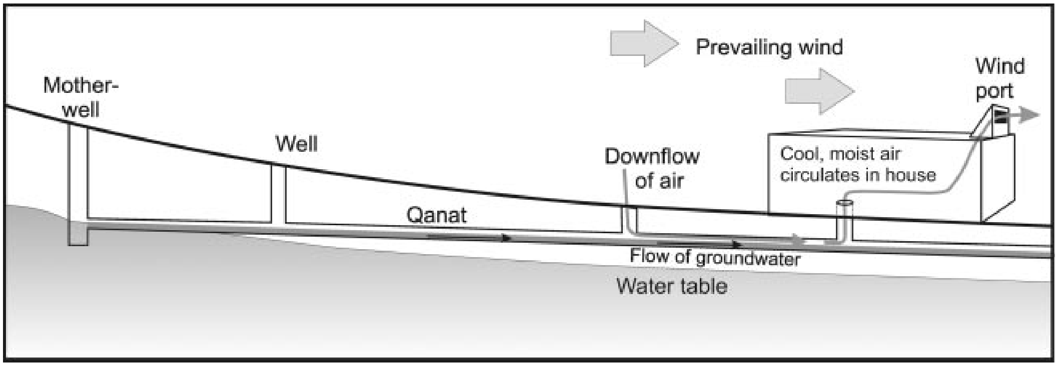


Figure 6.2 Diagram of traditional ground source cooling in the Middle East, using a qanat or well and a wind port.

Figure 6.3 An array of dry coolers in Norway. A hot fluid (which is to be cooled) circulates through a radiator grid below the horizontal fan elements. *Photo by David Banks.*

mounted outdoors. Heat is transferred from the fluid to the fan-assisted flow of cooler air by the temperature difference. Dry coolers will only work if the fluid is warmer than the air temperature. Thus, they are particularly effective at performing cooling on cold winter days. No compressor is used and they are cheap to run.

6.5 Evaporation

We have already seen that, when a liquid evaporates, it absorbs *latent heat* from the environment, producing a cooling effect. The human body uses this mechanism for thermoregulation – it produces sweat, which evaporates. In Section 5.3, we learned how inhabitants of India had a traditional method of producing 'slush' by allowing moisture to evaporate from wet straw, placed outdoors in the wind overnight.

Remember that the temperature drop that can be achieved by using an evaporative process is

$$\Delta\theta = \frac{Z_E L_V}{Z_C S_C} \tag{6.1}$$

where
Z_E is the amount of fluid evaporated ($kg\,s^{-1}$);
Z_C is the flow of circulating fluid (to be cooled) ($kg\,s^{-1}$);
L_V is the latent heat of the evaporating fluid ($J\,kg^{-1}$);
S_C is the specific heat capacity of the circulating fluid ($J\,kg^{-1}\,K^{-1}$).

6.5.1 Evaporative coolers

Evaporative cooler units are suited to hot, relatively arid climates (e.g. western United States, northern Africa). They typically consist of an air intake on the outside of the property. A fan sucks outdoor air in through a fibrous mat, which may be composed of wood-wool or some coir-like substance. The mat is kept moist by a steady trickle of water dripped on to the top of the mat. As the dry air passes through the mat, the water evaporates, absorbing latent heat from the air. The result is a stream of cool, moist air, which is delivered to the interior of the building. This form of air conditioning is relatively efficient (no compressor is required) and is very simple. It does, however, require a supply of fresh water, which may not be a trivial consideration in a country such as Sudan (Figure 6.4). Poor maintenance can result in the mat becoming a breeding ground for microbes. Moreover, the high humidity of the delivered air can cause fungal growth and damage to fittings. Indirect evaporative coolers are available, however, where evaporation takes place on the outside surface of a heat exchanger and the airflow passes through the interior, thus minimising humidity problems.

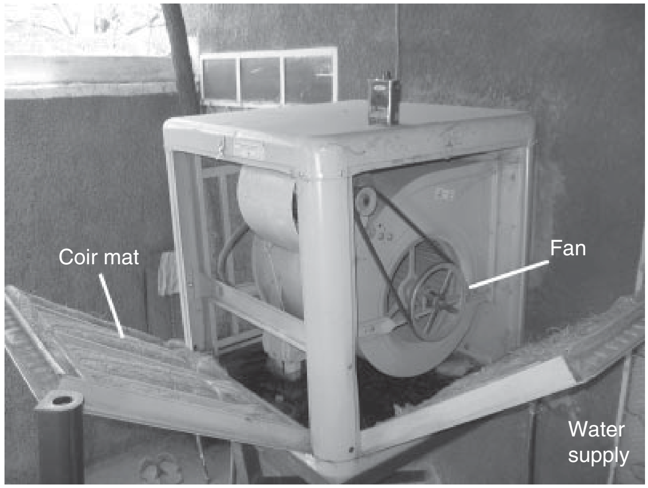


Figure 6.4 An evaporative cooler installed on a house in Nyala, Sudan. A trickle of water soaks into the coir matting in front of a fan unit. The water evaporates, cooling and humidifying the air drawn into the house. *Photo by David Banks.*

6.5.2 Evaporative cooling towers

Evaporative (or wet) cooling towers are used for removing heat from a flow of water. One of their familiar applications is to remove heat from the water stream that is used to cool the condensers associated with the steam turbines at coal-fired or nuclear power stations (Figure 6.5).

In such large industrial cooling towers, the warm water (which is to be cooled) enters near the top of the tower via spray nozzles and falls downwards through the tower. Air enters through the base of the tower and, being warmed by the cascading water, rises up through the inside of the tower – a *counter-flow* to the water. A small proportion of the water evaporates and exits with the air leaving the top of the tower, taking with it a cargo of latent heat and thus cooling the cascading water, which can then be collected and returned to the turbine condenser. This process allows the bulk of the water flow to be reused (around 5% of the water flux may be lost by evaporation for each circuit through a typical power station). Thus, the consumptive use of water resources is minimised.

Evaporative cooling towers are also manufactured in much smaller sizes and can be used to perform cooling at factories or can be mounted on the roofs of office buildings and utilised as sinks for waste heat (i.e. sources of cooling) from their air-conditioning systems. Such cooling towers have structural implications for buildings (they are heavy) and occupy valuable top-floor/roof area, which could be used more profitably.

It is not uncommon for evaporative cooling towers to be combined with compressor-driven chillers (heat pumps in cooling mode) in large buildings. In such cases, the heat

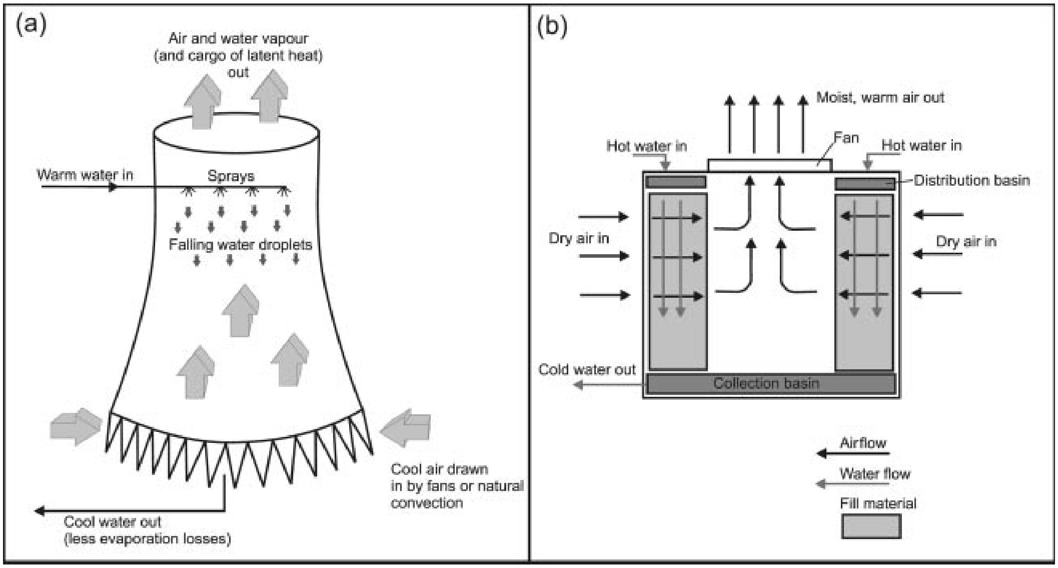


Figure 6.5 (a) A diagram of a counter-flow cooling tower. (b) An induced draft, cross-flow evaporative cooling tower, modified from a public domain file created by Eric Dreher and distributed (via Wikimedia Commons) under the terms of the GNU Free Documentation Licence and Creative Commons Attribution ShareAlike 3.0 Licence. *(In short, you are free to share and make derivative works of the work under the conditions that you appropriately attribute it, and that you distribute it only under a licence identical to this one.)*

pump does not reject heat from its condenser directly to the outside air, but to a stream of water. The water is pumped to the roof-mounted evaporative cooler, which then efficiently transfers the waste heat to the atmosphere. This practice of using a water-cooled heat pump and an evaporative cooling tower is typically more efficient than an air-cooled heat pump.

Evaporative cooling towers often contain some form of fill, of high surface area, which causes the stream of water to splash and break up into droplets, maximising contact with the airflow. The airflow may be in the opposite direction to the water flow (*counter-flow*) or may enter from the side of the tower (*cross-flow*). The airflow may simply be via natural convection (*natural draft*), or it may be assisted by fans (*mechanical draft*).

Potential problems associated with evaporative coolers include the following:

- risk of *Legionella* colonisation (see Box 7.3);
- risk of poor performance and freezing on very cold days;
- Any natural water contains small quantities of dissolved minerals. There is thus a risk of progressive build-up of salt concentrations in the water through several cycles of evaporation. If allowed to progress unchecked, this may lead to problems of mineral scaling. To prevent this, a certain quantity of water (*draw-off water*) is

removed from the circuit and replaced with fresh *make-up* water (which replaces the evaporated water and the draw-off).

If C_{SM} is the concentration of a certain conservative solute in the make-up water ($mg\,L^{-1}$), and

- C_{SC} is the solute concentration in the circulating water ($mg\,L^{-1}$),
- M is the quantity of make-up ($L\,s^{-1}$) and
- D is the quantity of draw off water, plus any other water lost to the cycle via leakage or windage ($L\,s^{-1}$),

$$D \cdot C_{SC} = M \cdot C_{SM}, \text{ thus } D = M \cdot \frac{C_{SM}}{C_{SC}} \tag{6.2}$$

6.6 Chillers/heat pumps

The use of heat pumps to perform cooling is termed mechanical or compressor-driven cooling (and the heat pump, if not reversible, is often termed a chiller). In this case, the evaporator of the heat pump is thermally coupled to the building and the condenser to the outside environment.

Remember that many modern heat pumps are reversible, so that, essentially by flicking a switch, the direction of refrigerant through the cycle (Figure 7.22) is reversed. What was the condenser in heating mode now becomes the evaporator and vice versa.

The heat pump evaporator may produce chilled air directly (water-to-air or air-to-air heat pumps), and it may produce chilled water (water-to-water or air-to-water), which will then be passed through fan coil units or chilled structural elements of a building.

The heat pump condenser may be coupled to the outside air (e.g. via a fan unit or a dry cooler), to an evaporative cooling tower, to a river or fjord, to a flux of groundwater or to a closed-loop ground heat exchanger. The waste heat from the condenser is thus dumped to the air, the water or to the rocks and sediments beneath the building.

Of course, if some spaces within the building (or within adjacent buildings) have a need for heat, the condenser can be thermally coupled to these, such that the heat extracted from the cooled space is not wasted. Using the waste heat from the condenser of a heat pump in cooling mode to preheat domestic hot water (for which there will be a demand even in summer) is an excellent use of resources and minimises the thermal load that must be disposed to the environment.

An air-to-air heat pump used for chilling is often simply referred to as an air conditioner (Figures 6.6 and 6.7). A unitary or through-wall air conditioner is typically a large box mounted in a hole in the exterior wall. A packaged-terminal or 'split' system comprises two separate units: an outdoor-mounted compressor, condenser and fan and

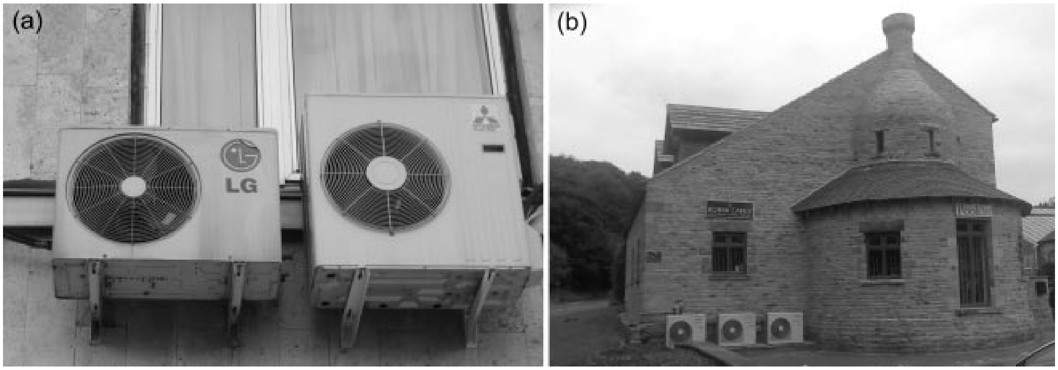


Figure 6.6 Air conditioners and air-sourced heat pumps. They may be clever – but are they pretty? Examples from (a) Moscow (Russia) and (b) Silkstone, Yorkshire (UK). *Photos by David Banks.*

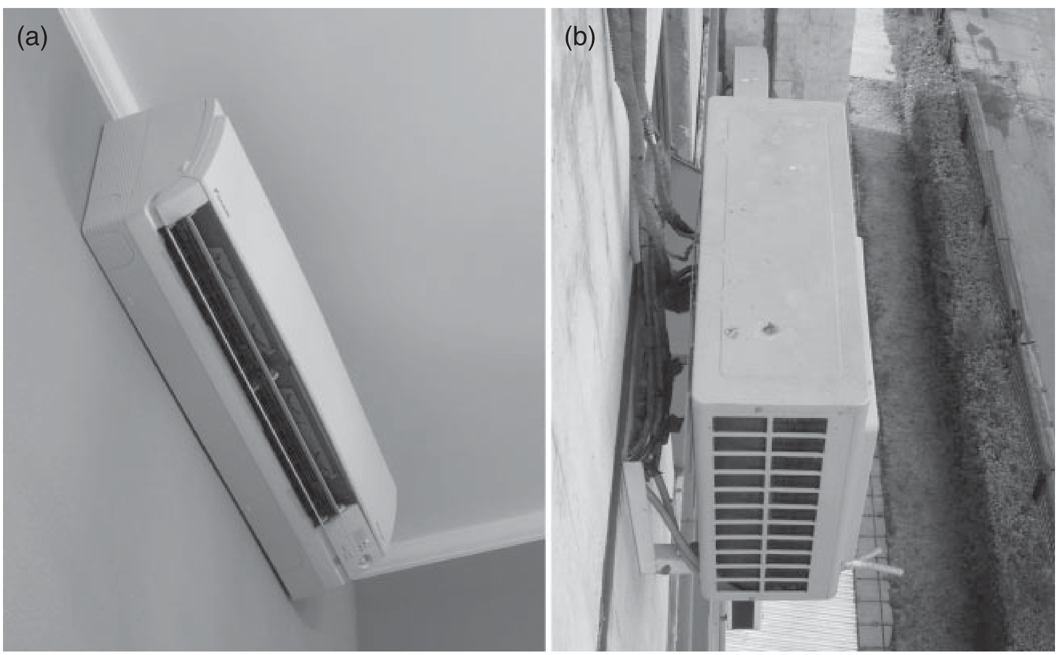


Figure 6.7 A split air conditioner, showing the interior (a) and exterior (b) units. *Photos by Dave Banks.*

in indoor-mounted evaporator and fan console unit. In either case, the indoor fan blows air over the cool evaporator while the outdoor fan blows air over the condenser, discharging waste heat to the atmosphere. Air-conditioning units may also perform a dehumidification function (see below).

We can define the coefficient of performance of a heat pump in cooling mode (COP_C) as

$$\mathrm{COP}_C = \frac{C}{E} \tag{6.3}$$

where C is the cooling effect delivered (kW), that is the amount of heat removed from the building (kW), and E is the electrical power consumed (kW). COP_C is sometimes also termed the energy efficiency ratio (EER).

Remember that, in heating mode, the power (electrical energy, E) turns up as useful heating effect (H) delivered to a building:

$$H = H_{\mathrm{env}} + E \text{ and } H_{\mathrm{env}} = H\left(1 - \frac{1}{\mathrm{COP}_H}\right) \tag{6.4}$$

In *cooling* mode, however, the electrical power used to run the compressor turns up as waste heat that must be disposed of (unless it can be used elsewhere in the building). Thus,

$$H_{\mathrm{env}} = C + E = C\left(1 + \frac{1}{\mathrm{COP}_C}\right) \tag{6.5}$$

where H_{env} = heat transferred to or from the environment.

When we are doing compressor-assisted chilling, we should remember that we may simply be using high-quality electrical energy to throw heat away to the environment. We can thus question whether the use of any form of heat pump or compressor-driven 'active' cooling is 'environmentally friendly'. The answer must be 'not really'.

We could, of course, redeem ourselves, if we could somehow utilise this waste heat elsewhere, or even store it until winter. We should be examining if there are locations in our building or in adjacent buildings that can make use of the waste heat. Failing that, we should consider whether we can store the waste heat until some time when there may be demand (daytime to night-time storage, or even seasonal storage from summer to winter). Large tanks of water (or some other medium with a high heat capacity) are ideal for storing heat or 'coolth' on a short-term or diurnal basis. For seasonal heat storage, however, the ground itself is ideally suited. We could, for example, dump waste heat to the ground during summer and then recover it again (via our GSHP in heating mode) during the winter months. This is termed underground thermal energy storage or UTES (more of this in Chapter 14).

6.7 Absorption heat pumps

We have already seen that, if we have a surplus of waste heat: for example, from an industrial process, from an electricity generating plant or from a high-temperature geothermal well, we can use it to power an *absorption heat pump* to produce chilling. The principle of this is described in Box 4.5.

6.8 Delivery of cooling in large buildings

We have seen that low-temperature underfloor central heating systems are ideal for the delivery of heat from ground source heat pumps. Because of the thermodynamics of heat pumps, we would expect relatively high temperature (that is, operating at +10 to +12°C, rather than +5–6°C), cooling systems to perform best, as the temperature differential between the cooling fluid and the ground is minimised.

If we have a reversible domestic heat pump, we could simply circulate the cold fluid (produced in cooling mode) through our normal underfloor or radiator-based heating system. This would, in fact, produce a cooling effect – but it would be far from ideal. Firstly, while hot air rises from radiators and underfloor systems, cold does not, so the cooling effect would not be efficiently distributed in the room. Secondly, there is a risk that moisture from the air would condense on the underfloor pipes or radiators and lead to problems with dampness. Thus, common means for distributing 'coolth' from heat pumps to buildings include the following:

- Ground-to-air or air-to-air heat pumps, which produce chilled air directly and avoid the need for the heat pump to produce a chilled liquid circulating through the building.

Often, however, a heat pump plant room will produce chilled water, which is then circulated through the building in a network of pipes called a *building loop*, in which case, the following methods may be used:

- Chilled structural elements – chilled beams, ceilings or panels. These building elements contain networks of heat exchange pipes through which the chilled fluid produced by a heat pump is circulated: again, care must be taken to avoid condensation issues.
- Mounting *fan coil units* on a building loop of chilled fluid. These are essentially small heat exchange grids mounted in front of fans, which force air across them, exchanging heat or 'coolth' between the building loop fluid and an air current. These are a popular choice as they work both in heating mode (hot building loop fluid produced by heat pump) and cooling mode.

Many conventional large building cooling systems produce a fluid, which enters the building loop at around 6°C (the *flow temperature*), acquires waste heat from the building and returns to the plant room at around 12°C (the *return temperature*). However, it is possible to construct modern, high-temperature cooling systems, which operate with flow temperatures of 10–12°C. Because the temperature differential between the optimum room temperature (maybe 20°C) and the building loop is lower, Fourier's law (Equation 1.1) states that we require a higher surface area or more efficient type of heat exchanger installed in the rooms to be cooled. This normally translates into a higher capital cost. However, because the heat pump is producing a higher-temperature fluid at the evaporator, we would expect better efficiency and lower running costs.

6.9 Dehumidification

Humans tend to feel most comfortable in an environment with a relative humidity of 40–50%. Air conditioning and climate control may also involve the regulation of humidity in an interior space. We have already seen how evaporative coolers can humidify the air in arid climates.

In temperature and tropical climates, we will often wish to cool the air in an interior space during summer. However, as we cool the air, its relative humidity increases (the saturated vapour pressure of water in air decreases as temperature decreases), and the risk of condensation increases. Thus, it is often necessary to carry out some form of dehumidification in parallel with chilling. This may be particularly important in, for example, horticultural greenhouses, where many plants are very sensitive to humidity and intolerant of condensation.

In old houses, where walls are inadequately weatherproofed, or where soil moisture seeps into cellars, dehumidification may also be desirable to remove excess moisture.

Dehumidification is carried out by providing some form of chilled surface, which lowers the temperature of the air below its dew point. A common way of doing this is simply to use a heat pump (chiller), with the evaporator coupled to the condensing surface. The evaporator removes latent heat from the water vapour in the air and causes it to condense and collect in a container. If the dehumidification is coupled to a space-cooling operation, it may be desirable to place the heat pump condenser outside and vent the waste latent heat to outdoor air. Alternatively, the condenser can be coupled to the ground (ground-coupled dehumidification) or to some locus in the building that has a demand for heat.

In our old, damp house, we may simply choose to feed our airflow through the dehumidifying surface (coupled to the heat pump evaporator) to remove the moisture. The dehumidified air might then be forced through a radiator coupled to the heat pump condenser, to heat the airflow. The product in this case is warm, dehumidified air. This is the design of many small portable dehumidifiers, which have the heat pump evaporator and condenser within the same physical space.

6.10 Passive cooling using the ground

We have already seen (Figure 4.1) that we could simply circulate the cool groundwater from a well through a system of chilled structural elements or fan coil units in order to cool a building. No heat pump would be necessary as the groundwater (at maybe 7°C in Sweden, or 10–11°C in the United Kingdom) is cooler than our building interior (20°C, maybe).

If no groundwater was available, we could also conceive of circulating the fluid from the building loop down into a large array of heat exchangers installed in deep boreholes near the site (so-called *closed-loop boreholes*; Figure 6.8 – see Chapter 10). The building fluid may enter the boreholes at, say, 15°C. As the fluid is warmer than

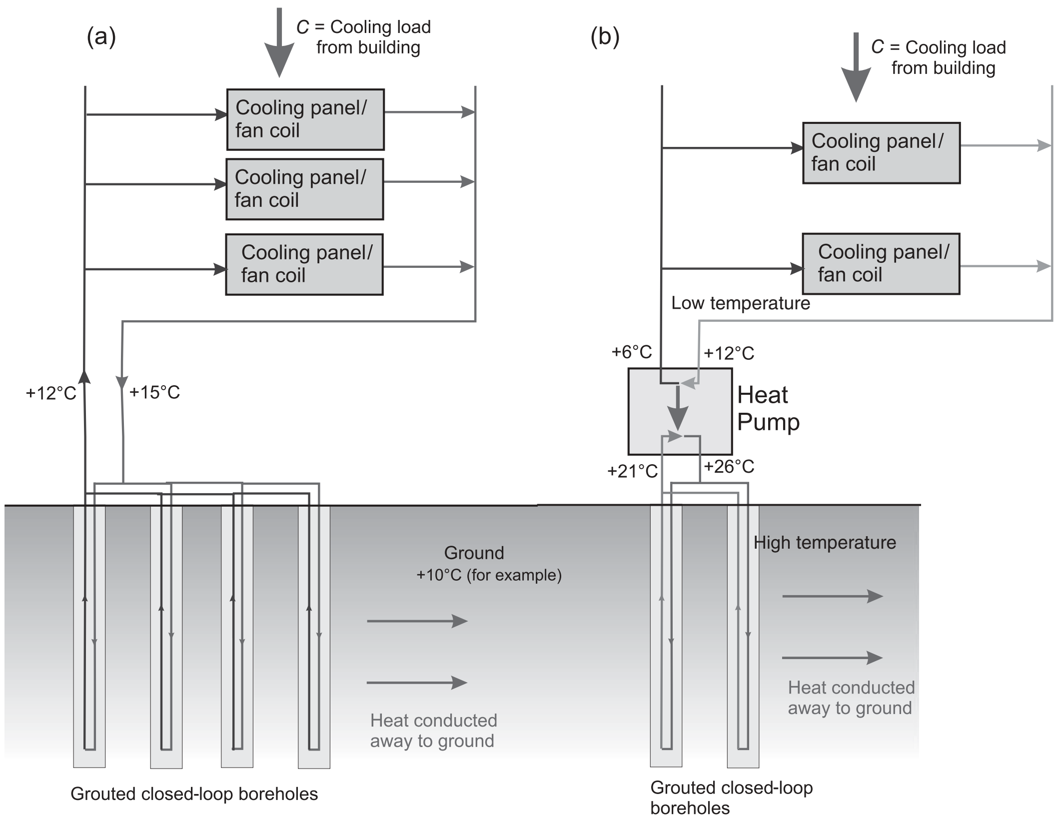


Figure 6.8 (a) Passive ground-sourced cooling and (b) 'active' ground-sourced cooling using a GSHP in cooling mode. In this diagram, a 'closed-loop' (see Chapter 10) solution is shown where heat is conducted from the ground loop away into the ground via grouted boreholes (although similar 'open-loop' solutions would also be feasible, where heat is rejected to a groundwater flux).

the ground (10–11°C in the United Kingdom), heat would be lost to the ground by conduction and the building loop fluid would be cooled, re-emerging from the boreholes at, say, 12°C. The cooled fluid re-enters the building to collect a new cargo of waste heat.

Such *passive cooling* or *free cooling* systems, which do not use a heat pump, are highly efficient in terms of carbon and running costs as they only require a modest amount of electricity to power circulation pumps.

On the other hand, they operate with low-temperature differentials (20°C to 12°C in the building and 15°C to 10°C in the ground). They thus require

1. highly efficient and/or a large area of heat exchangers in the building;
2. many metres of drilled borehole or large quantities of groundwater.

Thus, the capital costs of passive cooling schemes can be high.

6.11 Active ground source cooling

We have already seen, in Section 6.6, that heat pumps can be switched into reverse: a standard air-conditioning unit is a heat pump that pumps heat from the inside of a building to outside air. Such air-conditioning units are fine (although quite ugly), but they may operate relatively inefficiently on a hot summer's day, because they are striving to reject heat to a 'sink' (the outside air) that is at a relatively high temperature (25–30°C). A heat pump would operate significantly more efficiently if it could reject heat to a cool reservoir, such as the ground or groundwater at 11°C.

As in heating mode, a heat pump manual will usually provide data, or graphs, showing the heat pump performance in cooling mode (Figure 6.9). We will see that, if a groundwater at 10°C is used as the heat sink, we may get rather good COP_C values of around 5.5 to over 6, if the heat pump provides a temperature of c. 6–7°C to a building cooling circuit. If the building circuit operates at a higher temperature (see Section 6.8), even more favourable COP_C values can be obtained.

Compared with passive cooling (Section 6.10), a heat pump allows us to achieve greater temperature differentials:

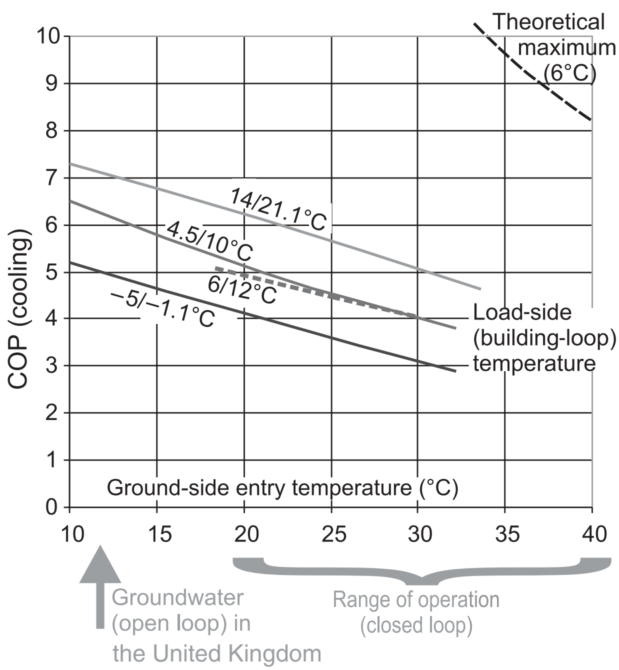


Figure 6.9 Curves showing coefficient of performance in cooling mode (COP_C) derived from two real GSHP models. Note that the COP depends both on the ground (heat sink) temperature and the building loop (heat source) temperature. The designation 6/12°C means that the fluid in the building loop leaves the heat pump at +6°C and returns at +12°C. NB: based on two types of commercial GSHP – others will have different curves.

- The evaporator can generate a building loop flow temperature of maybe 6°C. This much lower temperature allows us to save capital costs on the area or amount of heat exchange equipment incorporated into the building.
- The condenser can generate a relatively high temperature of 20–30°C (or even higher), allowing the meterage of heat exchange borehole installed in the ground to be reduced (or the amount of groundwater to be reduced in a groundwater-based system).

The price we pay for this potential decrease in capital cost is the increase in running costs (and carbon inefficiency) due to the electricity consumed by the heat pump compressor. *There is thus an inbuilt conflict between capital cost on infrastructure and marginal/carbon cost on running the system.* Although compressor-driven ground source cooling is less efficient than passive ground source cooling, it may be 20–40% more efficient (Kelley, 2006) than conventional active cooling solutions (e.g. conventional chillers and air conditioners).

We will see from Figure 6.9 that the higher the condenser (ground) temperature and the lower the evaporator (building) temperature, the less the heat pump efficiency. For example, if heat is being rejected to the borehole array at around 25°C and the heat pump is producing a chilled fluid at 6°C, we might expect a COP_C of around 4.5. If the chilled fluid in the building loop has a flow temperature of 10°C, the COP_C may well be greater than 5.

6.12 An example of open-loop groundwater cooling

Let us assume that a well provides a flow Z ($\mathrm{L\,s^{-1}}$) of groundwater at a temperature of 10°C. This could conceivably be used to provide passive cooling of a building (see Figure 4.1). Let us assume that, having travelled through a building's heat exchange elements, the groundwater has acquired 3°C of temperature (this is a modest amount, due to the small temperature difference between the ideal building temperature and the groundwater temperature).

The amount of heat removed from the building (the cooling effect, C) is given by

$$C = Z \cdot \Delta\theta \cdot S_{\mathrm{VCwat}} \tag{6.6}$$

If $Z = 1\,\mathrm{L\,s^{-1}}$, then

$$C = 1\ \mathrm{L\ s^{-1}} \times 3\ \mathrm{K} \times 4180\ \mathrm{J\ K^{-1}\ L^{-1}} = 12.5\ \mathrm{kW\ of\ cooling\ effect} \tag{6.7}$$

If we use a heat pump to enhance the heat transfer, with the heat pump now using a 6°C(flow)/12°C(return) temperature on the building loop, we may be able to obtain a greater cooling effect and/or reduce the amount of heat exchange surface installed within the building to gain the same effect. Let us say that, now, the groundwater increases in temperature by 4–6°C during its passage through the heat pump and let us say (Figure 6.9) that the COP_C of the heat pump is 5.5. The cooling effect can be calculated, from Equation 6.5, as

$$C = \frac{Z \cdot \Delta\theta \cdot S_{\mathrm{VCwat}}}{\left(1 + \frac{1}{COP_C}\right)} \tag{6.8}$$

$$C = 1\ \mathrm{L\ s^{-1}} \times 5\ \mathrm{K} \times 4180\ \mathrm{J\ K^{-1}\ L^{-1}} / 1.182 = 18\ \mathrm{kW} \tag{6.9}$$

The same $1\,\mathrm{L\,s^{-1}}$ of water may now be able to deliver 14–21 kW cooling, while consuming 2.5–4 kW electricity.

Finally, we could now arrange the system so that the heat pump extracts heat from the building circuit at 6°C(flow)/12°C(return) to an intermediate loop of fluid at around 25°C. This intermediate loop of fluid then passes through a heat exchanger and dumps its heat to the groundwater, raising the groundwater's temperature by 10°C, to 20°C. In this case, the COP_C will be lower (around 4.5 according to Figure 6.9), but the cooling effect would be

$$C = 1\ \mathrm{L\ s^{-1}} \times 10\ \mathrm{K} \times 4180\ \mathrm{J\ K^{-1}\ L^{-1}} / 1.222 = 34\ \mathrm{kW} \tag{6.10}$$

These calculations demonstrate that, the more 'passive' the cooling is, the more groundwater is required and the more heat exchange infrastructure is required in the building. The energy efficiency of the system is high. The more 'active' the cooling is, the less groundwater is required and/or the less heat exchange infrastructure is required in the building per installed kilowatt. The energy efficiency of the system is lower, however.

STUDY QUESTIONS

6.1 A ground source heat pump produces 21 kW of cooling effect. Under the appropriate operational conditions (source/sink temperature, etc.), the manual suggests you would achieve a COP_C of 4.2. How much heat would you be dumping to the ground and how much electricity would the compressor use?

6.2 If 100 kW heat is 'dumped' into a flux of $5\,\mathrm{L\,s^{-1}}$ groundwater, originally at 12°C, what would the resulting groundwater temperature be?

6.3 A closed-loop borehole system (of the type shown in Figure 6.8b) is designed to run at a maximum temperature of 40°C (note that this does not necessarily constitute a recommendation!). Using a heat pump corresponding to the curves in Figure 6.9, what COP might you expect, if the building loop is running at 6°C (flow)?

7

Options and Applications for Ground Source Heat Pumps

> A statement made by a university engineering lecturer that the description [of a heat pump] given by Lord Kelvin is impossible to accept, or by another lecturer that [it] offends thermodynamic law, is a less pleasant feature [of life].
>
> John A. Sumner (1976)

This chapter considers the various ways in which we can use ground source heat pumps (GSHPs), and includes a short discussion of design parameters and cost. After examining briefly how we can assess what size of heat pump scheme we will require, we will continue by looking at the two fundamental options for thermally coupling a heat pump to the ground: the *open-loop* and *closed-loop* systems. We will then go on to consider whether, in addition to space heating, GSHPs can also provide domestic hot water (DHW). Finally, we will examine the ways in which we can deliver flexible solutions to buildings that may have simultaneous heating and cooling needs.

7.1 How much heat do I need?

A building during winter can be considered as a warm box placed in the middle of a cold environment. If the temperature outside is colder than the temperature inside,

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

then the building will lose heat. The greater the temperature difference ($\Delta\theta$), the faster the building will lose heat. The better the standard of insulation of the building, the lower its thermal conductance (U) and the lower the rate of heat loss. As a very coarse simplification, we can say that the rate of heat loss from the building (Q in watts) is given by

$$Q \approx \Delta\theta \times U \tag{7.1}$$

7.1.1 Degree days

If we consider a period of time t (say, a whole month), we can estimate the total conductive heat loss (which will give us a first estimate of the heat demand required to maintain a comfortable indoor temperature) as follows:

$$\text{Conductive heat loss} \approx U\int_0^t \Delta\theta \mathrm{d}t \tag{7.2}$$

Of course, temperature varies with time, so to calculate this value, we need to consider the entire period of time that the outside temperature falls beneath a critical 'baseline' value (i.e. the comfortable indoor living temperature). We do this (Figure 7.1) by calculating the total area between the real temperature curve and the baseline value for the period in question. The value is an expression of both the severity and duration of cold weather and is expressed in the so-called *degree days concept* (Carbon Trust, 2006b).

The 'baseline temperature' is essentially the outdoor air temperature below which a building will require some active heating. It will, of course depend on the building,

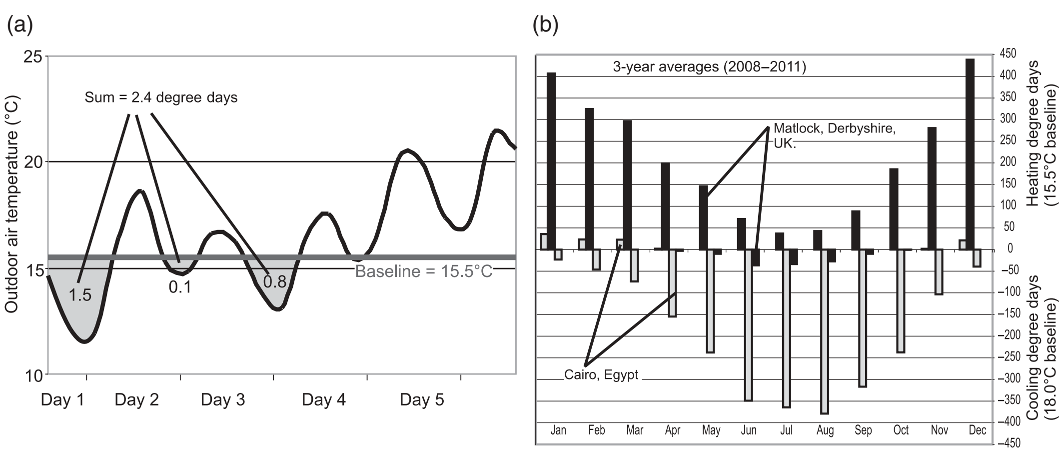


Figure 7.1 (a) Calculation of degree days for a 5-day period. The area of the shaded region gives the total number of degree days of heating demand in the 5 days; (b) average heating (baseline = 15.5°C) and cooling (baseline = 18°C) degree days per month for the English town of Matlock and for Cairo, Egypt.

its use and the sources of heat (humans, computers, etc.) that are usually present in the building. In Britain, however, it is common to take the 'baseline' temperature as around 15.5°C. At the 'baseline temperature', the internal heat sources (e.g. electronic equipment and respiring human bodies) approximately balance the heat losses from the building, such that a comfortable living environment can be maintained. If we experienced 48 continuous hours of an outside temperature of +1°C, this would be equivalent to 2 days × 14.5°C = 29 *heating degree days*. On average during the past 20 years, for example, the Thames Valley area has experienced 114 degree days in the month of October. In the harsher climate of north-east Scotland, 195 degree days is the norm in October (Carbon Trust, 2006a).

Note that, in order to determine a building's cooling demand, we can also talk about *cooling degree days* (Figure 7.1): these are related to the duration during which the outside air temperature exceeds a certain baseline value (which *may* be different to the baseline value used to calculate heating degree days).

7.1.2 Thermal resistance

If we continue to accept our rather simplified relationship that

$$\text{Conductive heat loss} \approx U\int_0^t \Delta\theta \, \mathrm{d}t = U \times (\textit{degree days}) \tag{7.3}$$

then we now need to approximate the thermal conductance of our house. This is normally done by considering our house as consisting of a number of thermal conductances – walls, roof, doors, windows, floor – coupled in parallel. For each of these elements, a 'U-value' can be calculated (Box 7.1).

Thus, if we know the outdoor temperature pattern for a given period (in degree days) and the thermal performance of our house, we can calculate how much heat we will require to keep our house warm. (Actually, reality is a bit more complex: our house loses heat not only by conduction, but also by convection – that is draughts – through openings such as chimneys, door jambs and vents, so we need to quantify these losses as well.) We can plot the actual amount of heat used against the theoretical heat demand to produce a thermal response plot, as in Figure 7.2. If the points are widely scattered around a trend, it means that our building is not being responsively managed (too much or too little heat is often supplied). If the points fit well to a linear trend (as in Figure 7.2), the building is being well managed. Ideally, the plot should pass through the origin: however, if the intercept on the 'heat supplied' axis is positive (as in Figure 7.2), it means we require heat even when the outside air temperature is at the baseline condition. This may mean that we have overestimated the building's internal heat generation (respiration, electrical equipment), and we should maybe select another baseline temperature. If the intercept on the 'heat supplied' axis is negative, it likely means we have underestimated the building's internal heat generation (Carbon Trust, 2006b).

BOX 7.1 Thermal Conductance, Thermal Resistance and ***U***-Values

Thermal conductivity (λ) is an intrinsic property of a material (albeit one that varies a little with temperature). Its inverse is thermal resistivity = $1/\lambda$ in $\mathrm{K\,m\,W^{-1}}$.

We can also talk about an extrinsic property called thermal conductance. It is extrinsic as it depends on the dimensions (specifically, the thickness) of the body in question. Let us consider an expanded polystyrene (EPS) sheet of thickness 10 cm. The thermal conductivity of EPS is around $0.035\,\mathrm{W\,m^{-1}\,K^{-1}}$. The thermal conductance (Λ) of the sheet is given by

$$\Lambda = \lambda/L = 0.035\ \mathrm{W\ m^{-1}\ K^{-1}}/0.1\ \mathrm{m} = 0.35\ \mathrm{W\ m^{-2}\ K^{-1}}$$

In other words, our sheet of EPS will conduct 0.35 W of heat per square metre area for every degree Kelvin temperature difference across it:

$$\text{Thermal resistance } (R) = \frac{1}{\Lambda} = \frac{L}{\lambda} = 2.86\ \mathrm{m^2\ K\ W^{-1}}\ \text{for our EPS sheet}$$

Crucially, thermal resistance is an additive property. Thus, if we have a wall comprising a course of 125-mm bricks, an EPS-filled cavity of 75 mm and another brick course 125 mm thick, we can calculate the total thermal resistance (the R-value) of the wall (assuming the thermal conductivity of our bricks is $0.4\,\mathrm{W\,m^{-1}\,K^{-1}}$):

$$R_{\text{total}} = R_{\text{brick}} + R_{\text{EPS}} + R_{\text{brick}} = 2\times\frac{0.125}{0.4} + \frac{0.075}{0.035} = 2.8\ \mathrm{m^2\ K\ W^{-1}}$$

Thus, the total thermal conductance for our composite wall is $0.36\,\mathrm{W\,m^{-2}\,K^{-1}}$. If it is 20°C indoors and –5°C outside, we can thus estimate that our house is losing 9 W of heat through every square metre of wall (neglecting convection effects at the skin of the wall).

The U-values cited in building regulations for various constructions of walls, windows and roofs are based on exactly this concept of thermal conductance as the inverse of the sum of the thermal resistances (R) of the various layers in a composite structure. U-values are sometimes referred to as *composite thermal conductance* or thermal transmittance. They may also factor in components of heat loss due to convection (skin effects) and radiation, as well as simply conduction, and are referred to as *overall heat transfer coefficients.* The British Standard Assessment Procedure (SAP, 2010) currently sets reference U-values for dwelling walls at $0.35\,\mathrm{W\,m^{-2}\,K^{-1}}$ and for roofs at $0.16\,\mathrm{W\,m^{-2}\,K^{-1}}$.

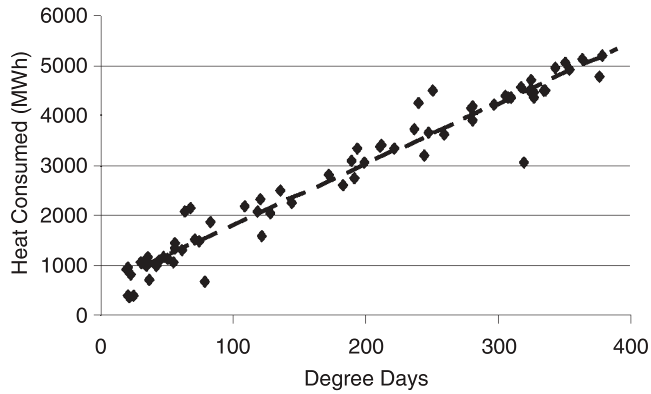


Figure 7.2 Thermal response plot for relatively modern buildings in Norwich, England. The low scatter of points around a linear trend line indicates a well-managed building – that is, heat is released in response to demand (*modified after Tovey, 2006*). *Reproduced by kind permission of Keith Tovey and the University of East Anglia.*

7.1.3 *What does all this mean for us?*

For a given climatic zone and a given standard of building insulation, we would thus expect that the amount of heating required by a house would be roughly related to the external surface area (walls, windows and roof) of the building. We would thus also intuitively expect some relationship with the size of the building. Such a relationship exists but it is not as simple as one might expect because, broadly speaking, heat losses increase with the surface area of the building, while internal heat generation (by respiration and electrical equipment) roughly increases with the volume of the building. Thus, the bigger the building, the lower the surface area-to-volume ratio and the smaller the need for additional sources of heating. Indeed, in very large buildings, there may even be a net cooling need for much of the year – that is, a need to get rid of waste heat, rather than to provide heat.

However, for modestly sized buildings, we can roughly relate peak heating demand to the size of a building. Sumner (1976b) stated that, for a reasonably well-insulated house, we can base our estimate on a calculation of the total living volume of the house (floor area × height for each room). For a comfortable living temperature of 20°C and an outdoor temperature of 0°C (which is not unreasonable for Britain), he estimated a demand of $36\,W\,m^{-3}$. Sumner reckoned that this would be adequate to cover the heat demand for 95% of the heating season. He furthermore estimated that the *average* demand throughout the heating season would be 65% of this figure. If we assume that an average room has a height of 2.4 m, we can reduce Sumner's (1976b) relationship to an effective peak heat demand of $87\,W\,m^{-2}$. It is perhaps an indication of how far the technology of house construction and insulation has progressed since Sumner was writing, that engineers in Britain today tend to target a figure of around $50\,W\,m^{-2}$ floor space for new buildings. All of this means that a typical new small-to-medium-sized domestic house will usually have an effective peak heating demand of 6–10 kW.

Remember that the above is only a potted summary – the estimation of heating and cooling demands is a specialised task (and one which should not be undertaken by geologists). A number of standardised tools and models are available to assist the building services engineer – we have already met one of these – the *Standard Assessment Procedure* (SAP, 2010) developed in the United Kingdom.

7.1.4 The heat demand of a real house

Figure 7.3 shows how we can learn something about our heating requirements from a utilities bill. The diagram shows the gas used by a small (≈90-m^2 floor area, three storey) semi-detached Victorian property in the East Midlands of the United Kingdom. A non-condensing combi gas boiler provides central heating via wall-mounted radiators in the heating season and DHW all year round.

During the peak quarter of the heating season, we can see that between 6000 and 7000 kWh of gas is consumed for space heating, equating to an average of around 3 kW. If we bear in mind Sumner's 'rule of thumb' that average heating demand in the heating season may be around 65% of peak demand, we can estimate that peak demand could be as high as 5 kW. For a floor area of 90 m^2, this equates to an estimated peak demand of 56 W m^{-2}. Not bad for a poorly insulated Victorian house – but we should remember that patterns of heat usage also come into play. In fact, the building in question is often unoccupied for much of the day and the occupants typically

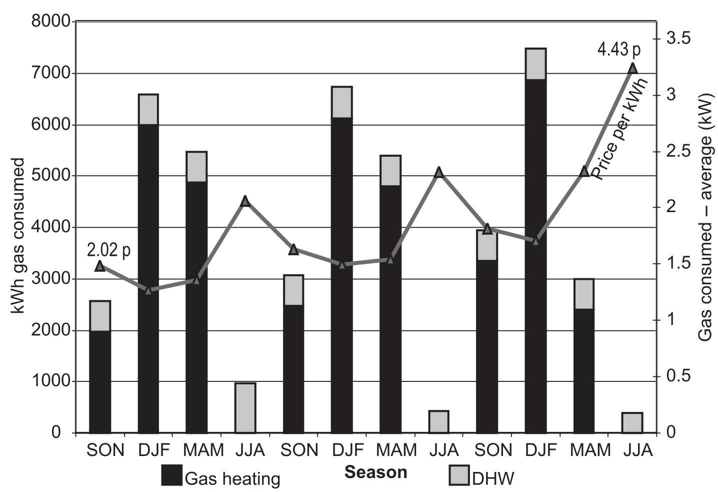


Figure 7.3 The heating supplied by a non-condensing gas combi boiler to a poorly insulated three-storey, small, Victorian, semi-detached house in the East Midlands of England, over the period 2003–2006. The line shows the trend in gas prices in pence per kilowatt-hour during that period (1 p = £0.01).

actively utilise the heating for around 6–8 h day^{-1}. This may lead us to revise our estimate of the peak heat demand to somewhere nearer 6–9 kW.

If we average the gas demand throughout the year, we find an average consumption of 13 000 kWh year^{-1} or 1.5 kW. Thus, the average heat demand over the course of a year in such a building may be less than 25–30% of the peak demand.

Of course, the estimates above are rather simplified – we have not taken into account the efficiency of the gas boiler in converting gas to heat, and we have rashly assumed that gas consumed for DHW will be the same throughout the year and can be estimated from summertime gas usage (it will, in fact, be larger during winter) – but they do hopefully render the heat budget of your home a little less obscure.

Finally, note the price of gas throughout the 3-year period of Figure 7.3. It is most expensive when purchased in small quantities (summer), at least in the United Kingdom! Furthermore, the peak yearly price has risen from 2.8 p kWh^{-1} in summer 2004 to 4.4 p kWh^{-1} in summer 2006, reflecting a drastic trend in energy prices throughout the United Kingdom. Given this, you might think that the house's occupants would be considering buying a GSHP to cut the costs of their energy bills. In fact, this may be a case where a GSHP is not an ideal solution. Technically, of course, it could be done, but

1. it may mean replacing the old high-temperature wall-mounted central heating system with a low-temperature one, or even underfloor heating. This is another significant capital investment on top of a borehole and heat pump (the garden is not big enough for a trenched system).
2. the building likely has a low thermal mass – it heats up relatively rapidly and cools down rapidly.
3. the building is rather poorly insulated (and capital may be better spent on improved insulation, rather than on a GSHP).
4. the building is only occupied for limited periods during the day. The more a heat pump is used, the quicker savings accumulate and the sooner the capital expenditure is down-paid.

A GSHP runs most efficiently at low temperatures and when it can take advantage of low electricity tariffs. It is thus an excellent solution in houses with a suitable low-temperature heating system and good standard of insulation, with a high thermal mass that can retain, for example, heat released at night. In the case of Figure 7.3, with the current pattern of occupancy, a combi gas boiler, rather than a GSHP, may be the most sensible solution.

7.2 Sizing a GSHP

We have seen in Section 7.1 how we can examine climate statistics and estimate our peak heating requirement on the coldest days of a typical year. If our peak heat demand is 8 kW, should we simply rush out and buy an 8-kW heat pump?

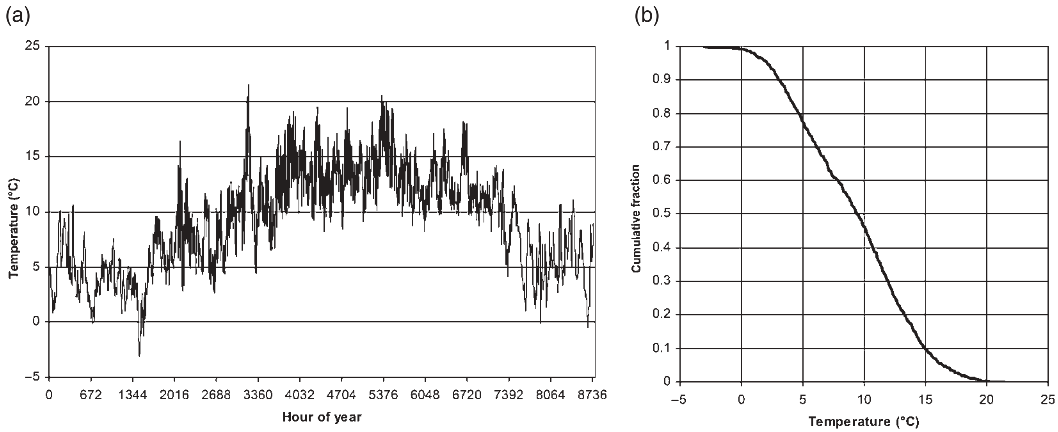


Figure 7.4 (a) Air temperature variations throughout a year (6-h average) for central Britain. It will be seen that the peak temperatures (and hence peak heating and cooling demands) occur for a tiny proportion of the time. (b) The cumulative frequency diagram shows that the 6-h average air temperature is <0°C for around 1% of the year but >3°C for 90% of the year. *Figure derived from generic temperature data for mid-Britain, sourced from NCEP Reanalysis data provided by the NOAA/OAR/ESRL PSD, Boulder, Colorado, USA, from their web site at* http://www.cru.uea.ac.uk/cru/data/ncep/.

If we consider the distribution of winter temperature (Figure 7.4), we typically find that the coldest temperatures (and the peak heating demand) occur on a very few days each winter. For example, in southern United Kingdom, the temperature in January may plunge to −5°C on a couple of days, but otherwise have a daily minimum somewhere close to 0°C. Given that the capital cost of a GSHP system is related to its capacity, it may make sense to choose a heat pump that is not dimensioned to supply the maximum heat demand (on the very few nights when the temperature is −5°C), but one capable of satisfying the demand on the majority of January days when the temperature does not fall much below 0°C. On the very coldest days, we could have a supplementary source of heat – a coal or wood fire or paraffin stove – to top up the heat provided by the 'undersized' GSHP.

In fact, Rosén *et al.* (2001) demonstrated that a GSHP with a rated output of around 60% of the calculated peak heat demand of a Swedish house can supply over 90% of the heat energy required by the house over the course of a heating season (Figure 7.5), assuming that the heat demand of the house is directly related to the outdoor temperature. We can express this relationship by defining two parameters:

$$\text{Effect coverage (\%)} = \frac{\text{Rated output of heat pump (kW)}}{\text{Peak heat demand of building (kW)}} \times 100\% \qquad (7.4)$$

$$\text{Energy coverage (\%)} = \frac{\text{Total heat supplied by GSHP (kWh)}}{\text{Total heat required by building over heating season (kWh)}} \times 100\% \qquad (7.5)$$

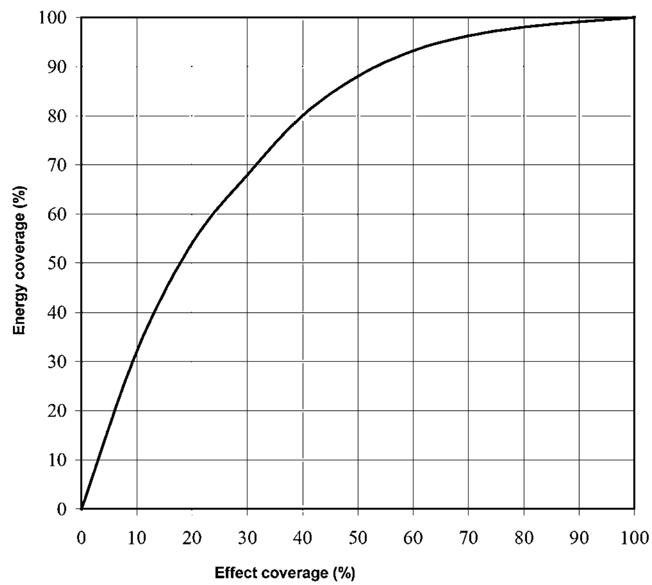


Figure 7.5 The relationship between Energy Coverage and Effect Coverage (see text) for a GSHP supplying a typical Swedish residence (prepared by G. Hellström for the publication by Rosén *et al.*, 2001). *Reproduced by kind permission of Göran Hellström.*

A GSHP operating against a high thermal mass, while energetically efficient, is not especially responsive to short-term temperature fluctuations. The combination of a base load provided by a GSHP and heat demand peaks being satisfied by a more conventional source does improve the degree of system responsiveness that can be achieved.

A heating system using a heat pump to supply a regular base load, with some form of back-up heating to supply a peak load, is often termed *bivalent*. Be very wary of using conventional electric heaters as the back-up, especially if you live in a country where electrical energy is very carbon intensive (Table 4.3). The use of a few hours of conventional electrical back-up can erode many hours of carbon gains accumulated by use of the heat pump. The alternative arrangement is a *monovalent* system, where the heat pump is sized to provide the peak load (after the effects of thermal storage have been accounted for – see Section 7.2.1) and no back-up is necessary. The advantage of such a system is that it avoids the capital cost and infrastructural complexity of investing in a secondary technology (e.g. a wood-burning stove or gas heater).

The typical running time (i.e. equivalent load hours) of a GSHP will depend on the Effect Coverage and on the duration of the heating season (and thus on the climate). In Sweden, running times of 3200–4000 h $year^{-1}$ are common for domestic heat pumps, dropping to 2200 h $year^{-1}$ in Switzerland and 1800 h $year^{-1}$ in Central Europe (Rosén *et al.*, 2001).

Skarphagen (2006) noted that it has formerly been common practice in Scandinavia to select GSHPs rated at around 60% of peak demand, partly because this supplies the majority of property's energy needs, while minimising the size (and cost) of heat pump system required and partly because it means that the heat pump is usually running against a demand (i.e. protracted running time).

Skarphagen (2006) went on to note, however, that Effect Coverage has been creeping up in recent years and that it is now common practice in Norway to select a heat pump rated at 80% or more of the peak demand. The reasons for this are as follows:

- There is relatively little difference in capital cost (e.g. of an 8-kW heat pump relative to a 6-kW heat pump) – see Figure 4.10.
- Control of heat pump output has become increasingly sophisticated in recent years. Older, simpler heat pumps have compressors that run at a single speed (either 'on' or 'off'), but inverter-driven heat pumps are now becoming available, the speed of whose compressors can be varied in response to demand.
- A higher Effect Coverage is increasingly economically attractive due to the increasingly sophisticated methods that utility companies use to monitor (and charge) electricity consumption (i.e. prices may increase at times of peak demand).

Note that much of the above refers to dimensioning for domestic houses in Scandinavia. Although the same general principles may apply further south in Europe, the exact figures may not. Furthermore, different countries have different cultures for providing heat: most Scandinavian homes will have more than one heat source, and it will be very common for a house to have either a paraffin stove or wood fire to provide supplementary heat on the coldest days. This may not be the case in Britain (and indeed, some UK heat pump manufacturers are recommending selecting a heat pump on the basis of peak heating demand – i.e., monovalent systems). Other dimensioning practices may be applicable to larger commercial or public buildings depending on their specific needs.

At the other extreme, it is quite common in much of the central and southern United States to use GSHPs to provide domestic air conditioning (i.e. active chilling) as well as a small amount of winter heating (domestic air conditioning is uncommon in most of central and northern Europe). In much of the United States, therefore, it is common practice to select a heat pump on the basis of summertime cooling load rather than heat demand.

7.2.1 Thermal storage

If you receive a water supply from a well, the groundwater will usually be pumped to a header tank or water tower, or to a pressure tank. There will be an automatic switch that cuts in the pump in the well when the level or pressure in the tank falls below a certain level. The well pump will operate, pumping water into the tank, until the level or pressure reaches a cut-out threshold. The cut-out pressure or level is somewhat higher than the cut-in level, ensuring that the well's pump operates for a reasonable amount of time at infrequent intervals, rather than cutting in and out continuously. It also means that the very peak demands of water usage are met from the storage tank and thus that the capacity of the well's pump does not have to exactly meet the instantaneous peak demand.

For the same reason, it is usually desirable to have some form of thermal storage within a GSHP system, often simply in the form of a 'buffer tank' or 'accumulator tank' – a tank of warm water that the heat pump feeds. The heat pump cuts in and out in response to the temperature of the tank, thus ensuring that, even in a monovalent system (see Section 7.2 above), the heat pump is running against a significant load. Any sudden, short-term demand for heat from the building can be supplied by the thermal store. The heat pump can then operate for a protracted period to replenish the thermal store when its temperature falls below a certain value. The heat pump cuts out again when the buffer tank reaches another, higher temperature. This prevents the heat pump compressor cutting in and out at very frequent intervals (so-called 'cycling'), which, if excessive, can lead to wear on the compressor. Furthermore, the incorporation of some thermal storage in the system mean that the worst extremes of heat demand are somewhat 'smoothed out', which will have (hopefully beneficial) implications for the sizing of the heat pump. Buffer tanks can be installed on the flow or return lines from/to the heat pump and some standard circuit diagrams are shown in Wemhöner and Afjei (2003) and EN 15450 (2007).

Other forms of thermal storage against which the heat pump operates can include the thermal mass of the building itself, which will retain heat and release it gradually, damping out externally imposed fluctuations in demand. The thermal mass or storage of the building may be enhanced by, for example, laying the underfloor heating pipes in a thick screed or concrete bed. This acts like a storage heater, enabling the GSHP to be run on a cheap electricity tariff at night to heat up the floor slab, which will then release heat during the working day, as at the Hebburn Eco-Centre (Box 4.4).

A similar principle can be used in cooling schemes, where we may wish to use the heat pump on a night-time tariff and store up 'coolth' that can be used for air conditioning during the following day (DoE, 2000). In this case, we can store 'coolth' simply by freezing water to form chunks of ice or an ice 'slush' (as in the Southampton scheme – Box 2.3). This is especially effective as we are using the latent heat released during the phase change from water to ice to store 'coolth'. More sophisticated 'phase-change' storage media are now under development – alternative materials include salt hydrides and organic compounds such as esters, fatty acids or hydrocarbons.

We have now gained some overview of how we might tailor a heat pump to the needs of our building. Let us now introduce the two main means by which we can convey heat from the ground to our heat pump: open-loop and closed-loop systems. We will examine these in considerably more detail in Chapters 8–11.

7.3 Open-loop ground source heat systems

Open-loop systems are those where we physically abstract water from a source: this can be a river, the sea or a lake. In the context of thermogeology, however, we are primarily concerned with groundwater abstracted (usually pumped) from springs, dug wells, drilled boreholes or flooded mines. Heat is extracted from this flux of pumped water or, in cooling mode, dumped into it. In cooling mode, we do not necessarily need to use a heat pump: cool groundwater at 11°C can be circulated through a network of heat exchange elements in a building to provide 'free' or 'passive' cooling (Figure 4.1). Often, however, we will use a heat pump to provide space heating or active cooling. The amount of heat (G) that we can extract from a flow of water is given by Equations 4.7 and 6.6:

$$G = Z \cdot \Delta\theta \cdot S_{\text{VCwat}} \tag{7.6}$$

where Z = flow of water in L s^{-1}; $\Delta\theta$ = the temperature drop (or rise, in cooling mode) of the water flow in Kelvin; and S_{VCwat} = specific heat capacity of water c. $4180\,\text{J L}^{-1}\,\text{K}^{-1}$.

If we use a heat pump with a coefficient of performance COP_H to extract heat from this water, the total heat load (H) delivered for space heating can be derived by combining Equations 4.7 and 4.8:

$$H \approx G + \frac{H}{\text{COP}_H} = \frac{Z \cdot \Delta\theta \cdot S_{\text{VCwat}}}{1 - (1/\text{COP}_H)} \tag{7.7}$$

If we are dumping heat from passive cooling (Figure 4.1) to our water flow, then the cooling load (C) that can be effected is simply given by

$$C = Z \cdot \Delta\theta \cdot S_{\text{VCwat}} \tag{7.8}$$

If we are using a heat pump, with efficiency COP_C, to perform active cooling, the cooling effect is given by

$$C \approx \frac{Z \cdot \Delta\theta \cdot S_{\text{VCwat}}}{1 + (1/\text{COP}_C)} \tag{7.9}$$

Let us consider a GSHP that is able to achieve a $\text{COP}_H = 4$ for heating and let us assume that it transfers heat, resulting in a temperature decrease $\Delta\theta = 4°\text{C}$ in the groundwater (3–5°C would be typical for many heat pump systems). If we pump groundwater at 11°C from a drilled well at a constant rate of $1\,\text{L s}^{-1}$ through our heat pump, it could effect

$$\text{a heating effect } H \approx \frac{1 \times 4 \times 4180}{3/4}\,\text{J s}^{-1} = 22.3\,\text{kW} \tag{7.10}$$

of which 16.7 kW comes from the groundwater and 5.6 kW from the heat pump's compressor.

In cooling mode, we might expect a better COP_C (see Figure 6.9), as the temperature differential is more favourable – the corresponding calculation is given in Section 6.12, using Equation 7.9.

Note that calculations such as these use a COP representing the instantaneous performance of the heat pump under given temperature conditions. The equations also neglect any energy expenditure on submersible pumps in our groundwater well, or circulation pumps in the building. Strictly speaking, if we wished to include these factors and to average over an entire heating or cooling season, we should use a system seasonal performance factor rather than COP (Section 4.10).

7.3.1 The abstraction well

We cannot construct a groundwater-based open-loop heating or cooling system wherever we want. The fundamental criterion is that we need an aquifer – a permeable body of rock or sediment in the subsurface that has adequate transmissivity and storage properties to yield a usable and constant flux of groundwater. We thus need specialist knowledge in order to ascertain the existence and properties of an aquifer in the subsurface. We will touch on this further in Chapter 8, but it is not the purpose of this book to delve too deeply into the science of hydrogeology. To obtain good advice on this topic, you can usually contact your national Geological Survey, Ministry of Water or Environment Agency. You are also likely to require advice, however, from a professional hydrogeological consultant.

Having ascertained the existence of an aquifer and assessed its potential to yield water, you will need to ask your tame hydrogeologist or groundwater engineer to assist in designing abstraction (and, if necessary, re-injection) wells. The fundamental types of information required to design a well are as follows:

- The design depth of well: this will depend on the depth of the aquifer stratum, the groundwater level in the aquifer and, to some extent, the hydraulic conductivity of the aquifer. Wells may be a few metres deep in shallow alluvial deposits, or may extend to a depth of several hundred metres.
- The design diameter of the well: this will ultimately depend on the yield of the well, which will in turn affect the diameter of the required pump (which must fit comfortably within the well).
- The design yield of the well (see above): this will be constrained by the hydraulic properties of the aquifer and by the desired heating/cooling load.
- Aquifer lithology: this will govern the type of well required, the drilling method and hence the cost.

It is, of course, possible to dig a well by hand, but nowadays, the job is usually done by a drilling rig to produce a narrow diameter 'drilled well' or 'borehole'. In hard,

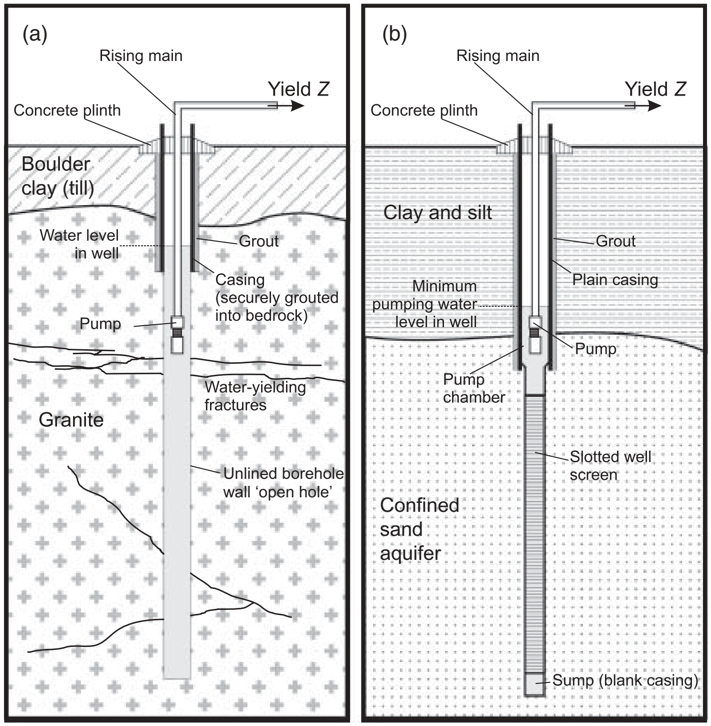


Figure 7.6 A schematic diagram of (a) a well that has been drilled 'open hole' in a hard lithology such as limestone or granite and (b) a well that has been cased and screened in a poorly competent rock or sediment.

lithified aquifers, such as granite, limestone or chalk, it maybe adequate to drill an 'open-hole' well. Here (Figure 7.6), the well is installed with a relatively short length of plain casing (usually, a steel tube) at the top of the well to support any loose or crumbly strata near the surface and to shut out any poor-quality surficial water and prevent it from entering and contaminating the aquifer. The casing should normally be grouted in with a bentonite–cement mix to provide a good seal in the annulus (again to prevent surface water from entering the aquifer). The remainder of the well is drilled unlined or 'open hole'. Groundwater enters via natural cracks, fractures and fissures in the borehole wall (see Figure 8.1).

In loose or poorly cemented lithologies, such as sands, gravels and many sandstones, we cannot leave our well unlined – the sediments would simply crumble into it. We need a means of supporting the wall of the well, shutting out the sediments, but allowing the groundwater to seep in. The solution here is to use a well screen (Figure 7.6). In its most primitive form, this may simply be a length of steel casing with slots cut into it. In its most developed form, it will comprise stainless steel wire wrapped around a framework of struts (Figure 7.7). This wire-wrapped screen can have an open area as high as 30–50%. Clearly, the size of the slots in our screen will be related to

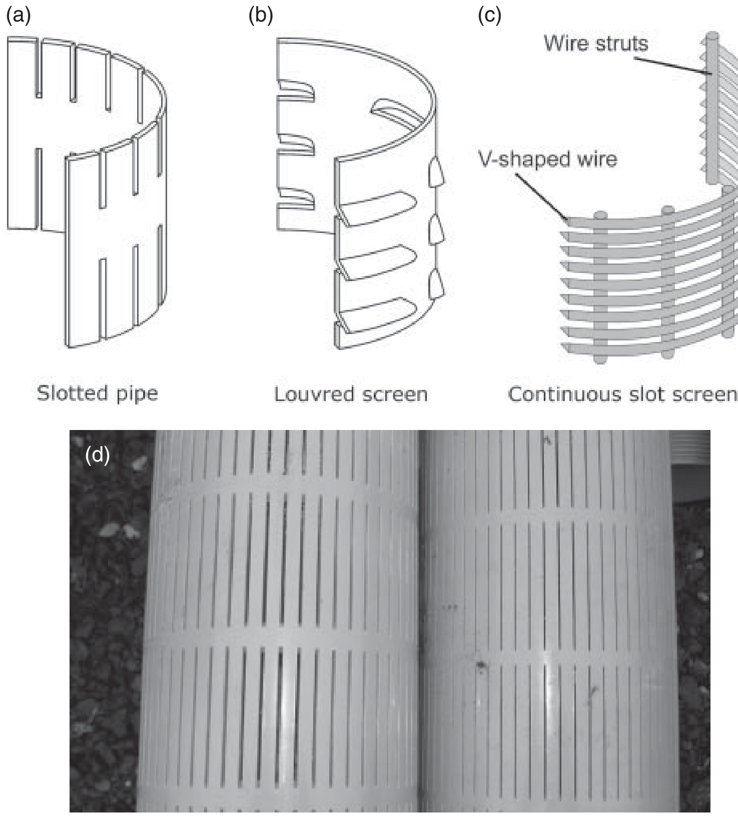


Figure 7.7 Types of well screen. (a) and (d) Vertically slotted screen; (b) louvred screen; (c) continuous wire-wrapped screen. *(a), (b) and (d) after Misstear et al. (2006) and reproduced by kind permission of John Wiley and Sons Ltd.*

the size of the grains in our sediment or rock: the larger the grains, the bigger our slots can be. In fine-grained or very uniform sediments, we may choose to install a gravel pack (or filter pack) outside the well screen. This acts as a filter medium, keeping fine particles out of the well. It allows us to have a larger slot size in our well screen than would otherwise have been the case, and thus to improve the hydraulic efficiency of the well.

Well design and drilling is a specialist business. Obtain advice from a consultant hydrogeologist or groundwater engineer and use a driller with experience of drilling wells for water supply. It is possible to make serious and expensive mistakes (of which Figure 16.3 shows a selection), which can incur the wrath of regulatory or environmental agencies and which will be discussed further in Chapter 16.

In most countries, the construction of a well for groundwater supply will normally require a permit or licence from the relevant regulatory authority. This authority may specify a number of conditions for granting the licence:

- an environmental impact assessment or water features survey;
- a pumping test to demonstrate that the required volumes of water can be obtained without unacceptable environmental impact and without detriment to other aquifer users;
- evidence that the abstracted water will be utilised appropriately or efficiently.

The regulatory authority will usually make a charge for administering the licence and may levy an annual charge for the water abstracted.

7.3.2 *Hydrochemical compatibility and prophylactic heat exchangers*

The subsurface geological environment is chemically very different from the atmosphere. It is basic and reducing, whereas our atmosphere is acidic and oxidising. As groundwater is brought to the surface, it may experience

- degassing of CO_2, which can increase the water's pH and lead to precipitation of carbonate minerals such as calcite ($CaCO_3$);
- exposure to and dissolution of oxygen. This may lead to oxidation of dissolved metals such as manganese (Mn^{II}) and ferrous iron (Fe^{II}), resulting in the formation of poorly soluble precipitates of manganese and ferric oxides or oxyhydroxides.

If we circulate our pumped groundwater directly through the evaporator or condenser of a heat pump, we are taking a number of risks:

- that any particles in the water may clog or abrade the heat pump's pipework;
- that minerals such as calcite or iron oxyhydroxide may precipitate within the evaporator of the heat pump;
- that, if the groundwater is saline enough, reducing enough or contains sufficient dissolved gases (CO_2, H_2S), it may promote corrosion. Note that the heat pump at the Hebburn Eco-Centre (Box 4.4) has a marine-grade corrosion-resistant evaporator. The Centre has, however, experienced corrosion of associated pipework due to the saline nature of the pumped groundwater;
- that the groundwater circulation may promote the formation of biofilms: slimes of non-pathogenic bacteria, such as *Gallionella*, that are commonly found in the geological environment. These biofilms can clog up well screens, pipes or heat exchange elements in the heat pump.

In order to avoid our groundwater causing such problems in our new, expensive heat pump array or in heat exchange elements within the building, we may choose not to allow the groundwater to enter the heat pump at all. We may choose to place a 'prophylactic' heat exchanger (see Box 7.2 and Figures 7.8 and 7.9) between the groundwater flow and a separate 'intermediate' loop of circulating heat carrier fluid. Such heat exchangers are typically parallel-plate heat exchangers of the type shown

BOX 7.2 Heat Exchangers

Heat exchangers are devices that efficiently transfer heat between two fluids. A car radiator, an elephant's ear and the grid on the back of a refrigerator are all heat exchangers. The most common forms of heat exchanger have the two fluids circulating on either side of a dividing wall. In an efficient heat exchanger, the dividing wall will have as large a contact area for heat transfer as possible, and as low a thermal resistance as possible. Turbulence in the fluid flow also aids heat exchange. Arguably the simplest form of heat exchanger is the single-pass counter-flow exchanger (Figure 7.8a). As the two fluids flow past each other, a heat flow rate Q passes from the warm to the cool flux and the temperatures of the fluids change as shown in Figure 7.8a. The temperature difference $\Delta\theta_a$ is known as the *approach temperature.*

If the heat exchanger has no external losses, the heat gained by the cool stream (fluid 2) should equal the heat lost by the warm stream (fluid 1):

$$Q = (\theta_{1i} - \theta_{1o}) \cdot S_{C1} \cdot \dot{m}_1 = (\theta_{2o} - \theta_{2i}) \cdot S_{C2} \cdot \dot{m}_2$$

where θ = temperature, S_C = specific heat capacity ($\mathrm{J\,kg^{-1}\,K^{-1}}$), $\dot{m}$ = mass flux of fluid ($\mathrm{kg\,s^{-1}}$). The subscripts 1 and 2 refer to the warm and cool fluids, while the subscripts o and i refer to outflow and inflow temperatures.

We can define an overall heat transfer coefficient U ($\mathrm{W\,m^{-2}\,K^{-1}}$) for the heat exchanger, such that

$$Q = U \cdot A \cdot \Delta\theta_{mean}$$

where A is the exchange area and $\Delta\theta_{mean}$ is some form of measure of the mean difference in temperature between the two fluids. For a simple parallel-flow or counter-flow heat exchanger of the type considered above, it can be shown that this is best expressed as the logarithmic mean temperature difference (LMTD) – see Figure 7.8a:

$$\Delta\theta_{mean} = \mathrm{LMTD} = \frac{\Delta\theta_a - \Delta\theta_b}{\ln(\Delta\theta_a / \Delta\theta_b)}$$

in Figure 7.9. This intermediate loop absorbs heat from the groundwater via the heat exchanger and carries it to the heat pump (Figure 7.10). A modern heat exchanger can be highly efficient, such that minimal heat loss occurs to the system as a result of using it. Of course, there is still a risk that the prophylactic heat exchanger may become fouled or corroded. It is usually cheaper, however, to temporarily decommission and clean (or replace) such a heat exchanger than a heat pump. The risk of particulate clogging can be reduced by installing removable filters prior to the heat

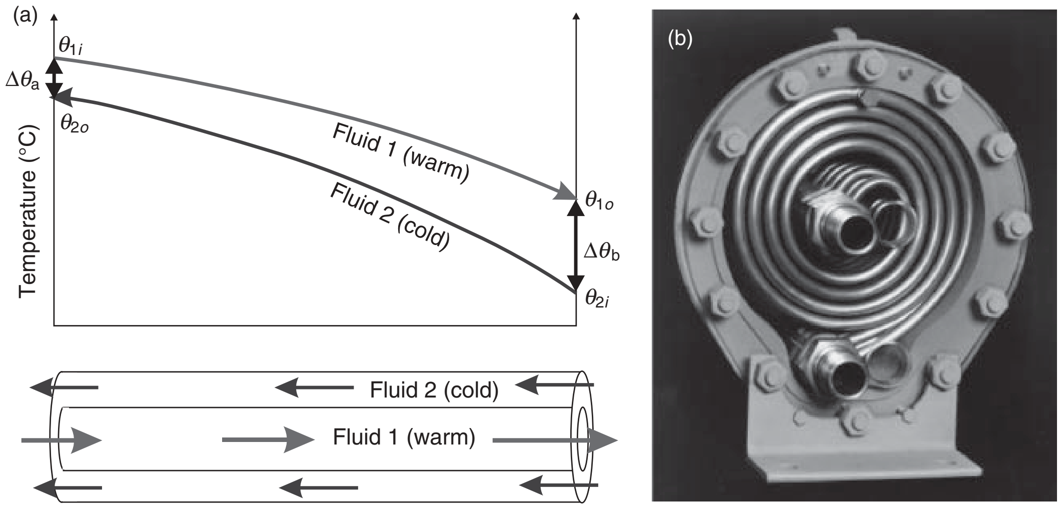


Figure 7.8 (a) Schematic diagram of a single pass, coaxial, counter-flow heat exchanger (see Box 7.2); (b) helical heat exchanger (Copyright 1999 from *Graham Corporation – Evolution of a Heat Transfer Company by R.E. Athey. Reproduced by permission of Taylor & Francis Group, LLC,* http://www.taylorandfrancis.com).

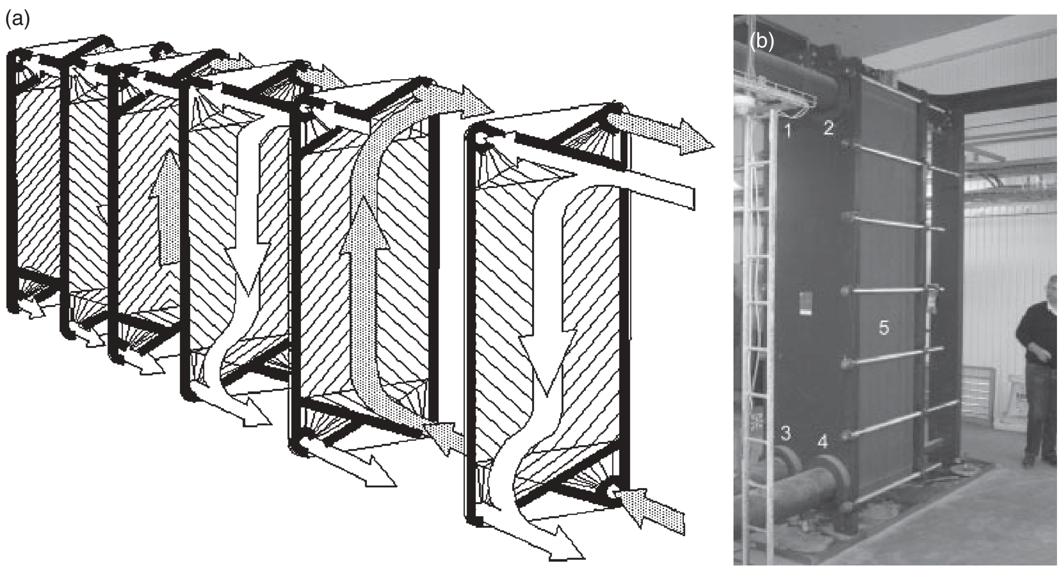


Figure 7.9 (a) A parallel-plate heat exchanger, *after Rafferty and Culver* (*1998*) *and reproduced by kind permission of the GeoHeat Center, Klamath Falls, Oregon*; (b) a parallel-plate heat exchanger at Arlanda Airport, Stockholm, which exchanges heat between a flux of groundwater and the airport terminal heating/cooling circuit (see Box 3.8). The four fluid pipes (groundwater flow/return and building flow/return) are marked 1–4 and the pack of parallel plates is labelled 5. *Photo by David Banks.*

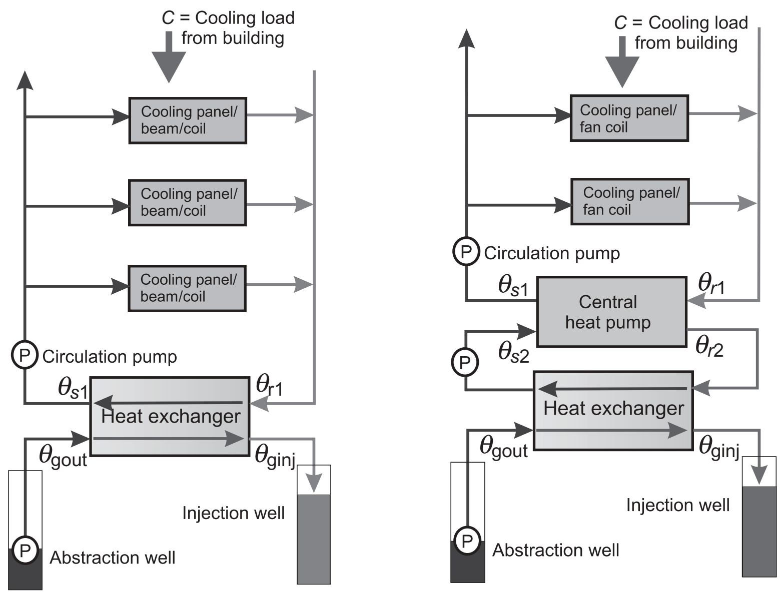


Figure 7.10 Two schematic diagrams showing how a 'prophylactic' heat exchanger could function to provide free cooling (compare with Figure 4.1) and active cooling. Note that the use of such a heat exchanger results in the need for additional circulation pumps. Note also that in a real situation, there would typically be some form of hydraulic storage (a tank) on the groundwater circuit, prior to the heat exchanger, to 'buffer' variations in flow.

exchanger. The risk of chemical or biological fouling of the heat exchanger can be reduced by the following:

1. Maintaining a high pressure within the groundwater circuit to prevent degassing of CO_2 within the exchanger (more on this in Chapter 8).
2. Preventing contact between the groundwater and oxygen in the atmosphere (i.e. closed systems).
3. Addition of small amounts of biocidal chemicals or reducing chemicals (e.g. sodium thiosulphate; Dudeney *et al.*, 2002) to prevent the formation of biofilms and the oxidation of ferrous iron, respectively (although this must be acceptable to the environmental authorities). VDI (2001b) also discussed the addition of carbonic acid to prevent the precipitation of calcite in heat exchangers, especially at high temperatures. Carbonic acid is merely a weak solution of CO_2 in water and *may* be regarded as a 'natural' additive by regulators.
4. Regular maintenance. This might involve flushing of the exchanger with acid or proprietary detergents or reagents to remove build-up of calcite or manganese/iron oxyhydroxide deposits. If a system is likely to be high maintenance, it may also be wise to select a heat exchanger that is designed for cleaning and/or disassembly.

One further advantage of a prophylactic heat exchanger and intermediate carrier fluid loop is that it allows us an added degree of flexibility with the flow rate and temperature of the groundwater discharge. Let us consider the active cooling system depicted on the right-hand side of Figure 7.10. From Box 7.2, we can see that the heat transfer rate (Q) across the prophylactic heat exchanger is given by

$$Q = (\theta_{r2} - \theta_{s2}) \cdot S_{\mathrm{VCcar}} \cdot F = (\theta_{\mathrm{ginj}} - \theta_{\mathrm{gout}}) \cdot S_{\mathrm{VCwat}} \cdot Z \tag{7.11}$$

where S_{VCcar} = volumetric heat capacity of the carrier fluid ($\mathrm{J\,L^{-1}\,K^{-1}}$), S_{VCwat} = volumetric heat capacity of groundwater ($\mathrm{J\,L^{-1}\,K^{-1}}$), F = flow rate of carrier fluid in the intermediate loop ($\mathrm{L\,s^{-1}}$), Z = groundwater flow rate ($\mathrm{L\,s^{-1}}$), and θ_{gout} and θ_{ginj} are the abstraction and waste (injection) temperatures of groundwater (K). We can thus see that we have the opportunity, with a well-designed choice of heat exchanger, to trade off groundwater rejection temperature against groundwater flow rate. If the carrier fluid is water based, Equation 7.11 reduces to

$$(\theta_{r2} - \theta_{s2}) \cdot F \approx (\theta_{\mathrm{ginj}} - \theta_{\mathrm{gout}}) \cdot Z \tag{7.12}$$

Thus, if our well yield or aquifer resources are limited, we can reduce Z. The price we pay is that θ_{ginj} increases, and a groundwater rejection temperature that is too high may be regarded as unacceptable by many environmental authorities. If a regulator imposes a stringent upper limit on θ_{ginj}, we can increase the pumped yield from our abstraction well. Note, however, that θ_{ginj} can never be higher than θ_{r2} and will usually be *at least* c. 2°C below it (in cooling mode). For an example of this type of calculation, see Section 6.12.

Taking the right-hand side of Figure 7.10 as an example, let us assume that the temperature change at the heat pump condenser is around 4°C. If the groundwater flow rate is similar to the fluid flow in the intermediate loop, the change in temperature of the groundwater will also be c. 4°C. If the abstracted groundwater temperature θ_{gout} is 11°C, then θ_{ginj} will be approximately 15°C. Depending on the efficiency of the heat exchanger, θ_{s2} may be around 14°C and θ_{r2} around 18°C. We may, however, be able to design the heat exchanger such that the groundwater flow rate could be reduced by 50%. In this case, the groundwater discharge temperature would rise to 19°C and the temperatures in the intermediate circuit might rise to, say θ_{s2} = c. 18°C and θ_{r2} around 22°C. The heat pump would operate a little less efficiently, as it is now discharging heat to a warmer fluid in the intermediate loop.

A heat exchanger should, of course, be tailored to the temperature and flow regimes that are specified for the heating/cooling system. Broadly speaking, as the efficiency of the heat exchanger increases (the U-value increases) and the approach temperature drops, the more the exchanger will cost. Kavanaugh and Rafferty (1997) contended that it is perfectly feasible to design modern plate heat exchangers with approach temperatures as low as 1.5–2°C.

7.3.3 Disposal of wastewater

Having passed our groundwater through the heat pump or heat exchanger, we are left with the problem of how to dispose of it. If the groundwater had an original water temperature of 11°C and we have used it for heating, we may now have a water whose temperature is as low as 6–7°C. If we have used it for cooling, the wastewater will be warm. Note that in some countries, two sets of permits – abstraction licences and discharge consents – may be required for an operation involving abstraction and disposal of groundwater. We will usually have the following options for water disposal:

Disposal to sewer: obviously, this depends on having a sewer or storm drain to hand. It must have the excess capacity to accept our waste flow. We will require the permission of the utility that owns the sewer and will usually have to pay a significant charge.

Disposal to a surface water body (e.g. a river): in order to do this, we will usually require the permission of a national or regional authority with responsibility for regulating the water environment. We may have to pay a discharge fee. We will usually have to perform some form of risk assessment, which will address questions such as the following:

- How will the temperature (warm or cold) of the discharged water affect the ecology of the surface water body, or its utility value to other users? Remember that waste heat (or cold) may be regarded as a potential pollutant by many environmental authorities. Warm water will usually be more environmentally problematic to discharge than cold water, as the solubility of dissolved gases, such as oxygen, decreases as temperature increases. Some guidance is available on the temperature distribution of surface waters (e.g. Orr *et al.*, 2010) and on the tolerance of different aquatic species to temperature changes (e.g. Turnpenny and Liney, 2007).
- Is the discharged groundwater geochemically compatible with the water in the recipient? Groundwater may, for example, be poor in dissolved oxygen, or rich in dissolved iron, or somewhat saline. All these factors could significantly impact the life in a watercourse.
- Will the additional discharge of groundwater to the surface water increase the flooding risk in the watercourse?

If the discharge is to the sea or to an estuary, the discharge of groundwater from a heat pump will in some cases be less tightly regulated.

Re-injection to the abstracted aquifer: a further option is to re-inject our waste groundwater back to the same aquifer from which it was abstracted. This is attractive because it has few or no water resource implications (there is no net abstraction of water from the aquifer) and will often minimise any risk of ground settlement that can occur in some soils due to prolonged net abstraction of groundwater. This solution may thus be attractive to environmental regulatory agencies charged with ensuring that water resources are protected. In some countries, such an abstraction/

re-injection operation may not require licensing. In other countries, such a licence may be less complex than for purely consumptive water use (i.e. abstraction only), and may attract lower charges.

The re-injection of limited quantities of water may be feasible via a soakaway structure. Larger quantities will usually require the use of re-injection wells. Their construction and operation is a specialist task; injection wells may need special well screens. The injected water may need to be sterilised to hinder bacterial clogging of the well screen or of the aquifer; it will need to have a very low particle content and the water pressure and its gas content will need to be controlled such that exsolving gas bubbles do not clog the aquifer and decrease its permeability. Contact with atmospheric gases should be avoided and pressure regulation will also be required to minimise the risk of mineral precipitation (see Banks *et al.*, 2009b).

Finally, we need to take care that we do not re-inject our groundwater so close to our abstraction well that we start to get 'short-circuiting' – our cold (or warm, if it is a cooling scheme) re-injected groundwater breaking through into our abstraction well. If this occurs, the temperature of our abstraction well water will start to drift downwards (or upwards) with time, compromising the efficiency of the system. This will be dealt with further in Chapter 8.

Disposal to another aquifer: if there is more than one aquifer below the site, we can abstract groundwater from one aquifer horizon and dispose of it (after passage through the heat pump system) by re-injection or re-infiltration to another. If we do this, we are consuming water from one body of water and re-injecting it to another hydraulically distinct body. A regulatory authority will regard this in much the same light as disposal to surface water and will need to be satisfied that abstraction from the first aquifer does not unacceptably compromise its resources or impact negatively on other users or on the environmental features supported by it. Likewise, the regulators will need to be satisfied that

- the injection of water will not cause an unacceptable rise in groundwater levels in the second aquifer;
- the injected water is geochemically compatible with the natural water of the second aquifer;
- any heat plume (i.e. body of warm or cold injected groundwater that is migrating with natural groundwater flow) does not constitute unacceptable heat pollution and will not adversely impact any users.

Disposal of wastewater to the abstraction well: in some smaller schemes, a proportion of the waste groundwater, having served our heating or cooling scheme, may be re-injected back into the upper part of our abstraction well. The thinking here is that the cool re-injected water (if it is a heating scheme) takes a finite time to flow down the well to the pump, and that its temperature will re-equilibrate on the way down by conductive transfer of heat from the borehole walls. Furthermore, in the well, it will be mixed with a percentage of 'new' groundwater entering the well from the aquifer such that, eventually, an equilibrium situation will be established. Such an arrangement is called a standing column well (see Chapter 13 for further information).

7.3.4 Disadvantages of open-loop heat pump systems

Disadvantages of groundwater-based open-loop systems can be listed as follows. Such systems

- are geology dependent. They require the site to be underlain by an aquifer, capable of providing an adequate yield;
- require a significant degree of design input from a hydrogeologist or groundwater engineer;
- need one or several properly constructed, durable (i.e. expensive) water wells, with pump installations, monitoring and control mechanisms;
- incur pumping costs associated with abstracting the groundwater from the well (as a rule of thumb, power consumption by water/circulation pumps and other auxiliary gear should be <10% of the total electrical energy budget of a GSHP scheme);
- generate a used water flow that must be legally disposed of;
- will usually require formal consent from a regulatory authority to abstract groundwater and to discharge the used water to a recipient. A fee may also be levied;
- will usually carry some risk of fouling/clogging of the heat exchanger, recharge well or re-injected aquifer by particulate matter, bubbles of exsolved gas, mineral precipitates or biofilms. The turbidity, chemistry, microbiology and pressure regime of the groundwater circuit will often need to be monitored or regulated to minimise the risks of fouling and clogging.

7.3.5 Advantages of open-loop heat pump systems

On the other hand, open-loop groundwater-based heating and cooling systems have a number of persuasive advantages:

- They utilise a natural medium (groundwater) that occurs at a constant temperature in the subsurface and has a huge specific heat capacity ($4180\,\mathrm{J\,L^{-1}\,K^{-1}}$).
- They extract heat by forced convection of groundwater rather than by subsurface conduction. They thus tend to extract more heat per borehole/well than closed-loop systems do (Figure 7.11).
- The abstraction well may provide a resource of potable water, as well as a heat resource. There is no reason why we cannot utilise good quality groundwater, having passed it through at heat pump scheme, instead of running it to waste. Even if the groundwater is of poorer quality, it may still be used as 'grey' water (i.e. for flushing and washing purposes) in a building.
- For use in heating and cooling schemes, the quality of the abstracted groundwater is not necessarily an important issue. Thus, open-loop heating and cooling schemes

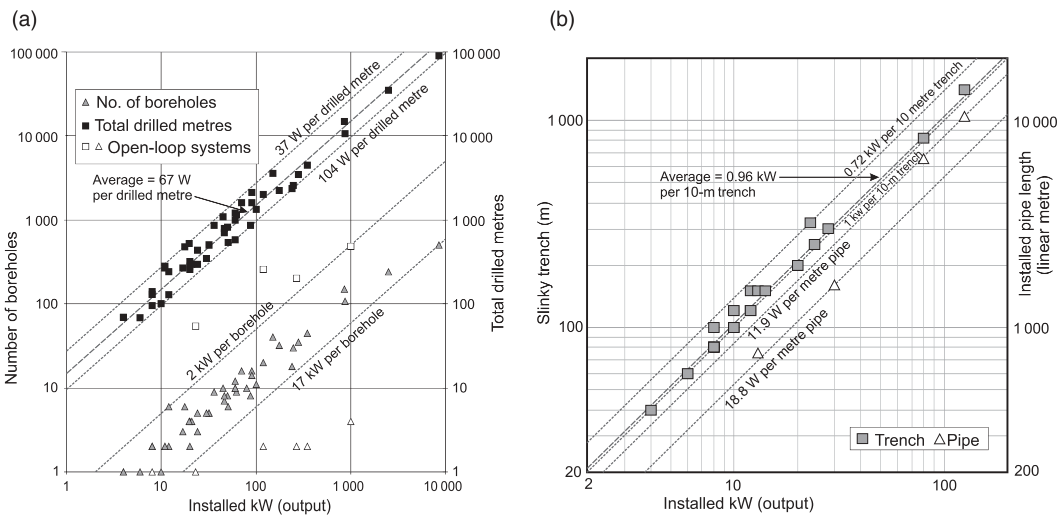


Figure 7.11 (a) The number of boreholes drilled and the total drilled metres for a variety of vertical closed-loop GSHP schemes (and a small number of open-loop schemes), related to installed heat pump kilowatt delivery. (b) The installed meterage of slinky pipe (as linear metres) *or* the installed metres of trench for a number of slinky-based horizontal closed-loop schemes. For both (a) and (b), the majority of schemes are heating dominated, though some of the larger (especially >60 kW) schemes are cooling dominated or provide both cooling and heating. Most of the schemes are in the United Kingdom and are either actually constructed or designed to a relatively detailed stage. Be careful, however, just because a GSHP scheme has been designed by a consultant, doesn't mean that it has been designed especially efficiently! These data are sourced from case studies documented by GeoWarmth Ltd., Earth Energy Ltd., Kensa Engineering Ltd., Holymoor Consultancy Ltd. and Drage (2006).

can be based on groundwater from flooded mines (Banks *et al.*, 2009b), from dewatered mines/excavations or even from contaminated sites. If groundwater is being pumped in connection with the remediation of a contaminated site ('pump and treat' schemes), there is no reason why useful heat could not be gained from it. Open-loop systems can also be based on natural saline groundwaters – for example, in coastal areas (see Box 4.4).

From Figure 7.11, we see that very often the lesser number of boreholes/wells required for open-loop schemes tends to balance the greater cost of their design, construction and licensing, such that the capital cost per installed kilowatt turns out to be comparable with closed-loop schemes (Figure 7.12).

7.4 Closed-loop systems

Hitherto, we have considered heating and cooling schemes that are based on extraction of water from a well (or spring, flooded mine, river or lake). But aquifers and

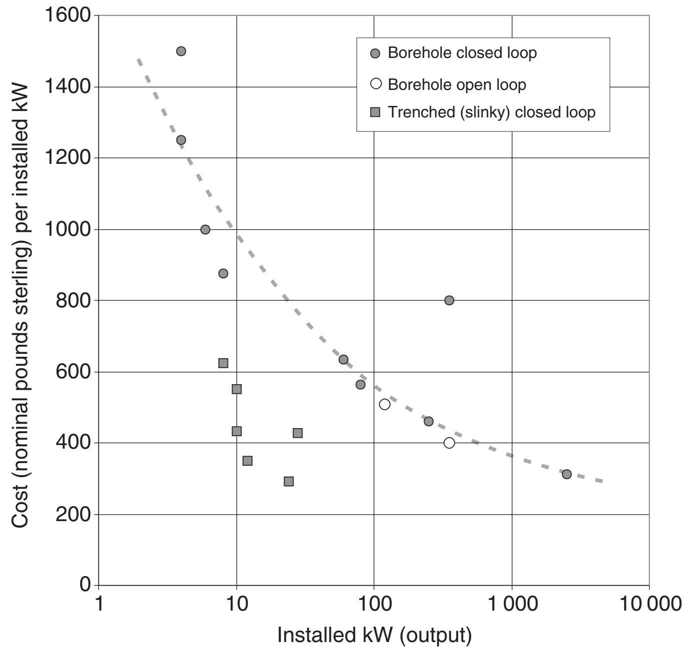


Figure 7.12 The cost, per installed kilowatt, of the schemes shown in Figure 7.11. The cost is shown in 'nominal' pounds sterling and many of the schemes are 3–7 years old. The prices likely reflect the capital cost of the trenches, boreholes, heat pump and hardware. They may not always reflect consultants' fees, taxes and other additional costs. The prices are likely to underestimate the total cost of commissioning a GSHP installation by around 50%.

surface water bodies are not ubiquitous. Fortunately, there is another way to extract heat from the ground that does not require any water to be abstracted or re-injected at all. Such schemes are called closed-loop schemes, and they can be constructed practically anywhere: in granites, clays, waste tips, permafrost or abandoned mines. Closed-loop schemes are of two types: direct circulation and indirect circulation. Direct circulation (also termed 'direct expansion' or DX) schemes were more common in the early years of GSHP systems, although they are still being installed by some companies today. However, indirect circulation schemes have become far more widespread than DX schemes in today's European GSHP market.

7.4.1 Direct circulation systems

In DX schemes (Figure 7.13), the heat pump's refrigerant is circulated into the subsurface in a closed loop of tube. As many refrigerants are organic solvents (e.g. fluorinated hydrocarbons), the tube is made of metal, typically copper. The tube may be buried in the subsurface in a shallow trench or installed in a vertical borehole. In effect, the subsurface ground loop acts as the evaporator of the heat pump. The chilled

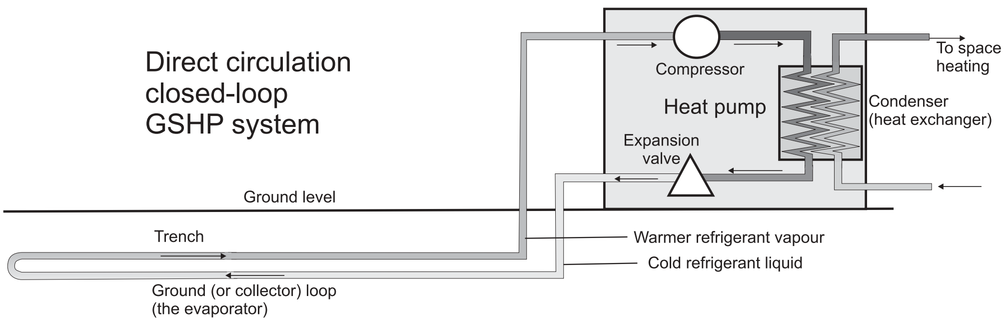


Figure 7.13 A schematic diagram of a direct circulation closed-loop scheme, installed in a horizontal trench.

refrigerant from the heat pump enters the subsurface in the loop, absorbs heat from the relatively warm earth and returns as vapour to the compressor of the heat pump, for the temperature of this heat to be boosted.

The major advantage of DX systems is the efficient transfer of heat from the ground to the refrigerant (Halozan and Rieberer, 2003), due to

- the fact that the ground loop *can* operate at rather low sub-zero temperatures (as low as –10°C or below). A bigger temperature differential means more heat absorbed per metre of ground loop, although, as in all heat pumps, there is a penalty to be paid as regards COP with very low evaporator temperatures;
- only one heat exchange step, directly from the ground to the refrigerant;
- no parasitic circulation pumps on the ground loop;
- (to a lesser extent, the high thermal conductivity of copper pipe).

Such DX heat pumps have fallen somewhat out of favour in some countries, however. This is not because DX systems do not work well, but rather because some environmental regulators get nervous about the idea of circulating refrigerant substances through the ground in a copper tube, which may be susceptible to mechanical damage and which, under some geochemical conditions, may be liable to corrosion. Refrigerants, such as the commonly used R407c, are fluorocarbons: environmental regulators usually regard such halogenated hydrocarbons as potential groundwater contaminants. It is feared by some that any future refrigerant leakages may thus be regarded as a pollution incident by regulators (regardless of the fact that the practical consequences of such a small leakage could be argued to be minimal). DX systems have also had some practical problems to overcome. In the early days of GSHPs, Sumner (1976a) drew attention to the potential for the refrigerant flow to become blocked in long ground loops by accumulations of oil from the compressor. DX systems are typically installed on horizontal ground loops or relatively shallow vertical ground loops, due to problems of guaranteeing efficient refrigerant circulation in deep vertical installations.

In other countries, there is a more upbeat attitude towards DX systems. Halozan and Rieberer (2003) described the healthy market for them in Austria and also documented new developments using CO_2 as a refrigerant fluid within stainless steel pipes, thus negating regulatory concerns regarding halogenated organics circulating in the subsurface.

7.4.2 *Indirect circulation systems*

We can also avoid circulating the refrigerant fluid directly into the ground by using an 'indirect circulation' closed-loop scheme. Here, a water-based carrier fluid circulates in a closed loop of pipe, which passes into the subsurface, either via a borehole drilled vertically into the earth (Figure 7.14), or via a horizontal coil or trench. In a heating scheme, the chilled carrier fluid exiting the heat pump absorbs heat (by conduction) from the subsurface and conveys it back to the heat pump, where the heat is extracted. The carrier fluid is thus chilled again and ready to start its next circuit through the earth. In a cooling scheme, the opposite applies: the warm carrier fluid from the reversed heat pump (or from a free cooling scheme in a building) descends into the subsurface and conducts a portion of its heat load to the relatively cooler

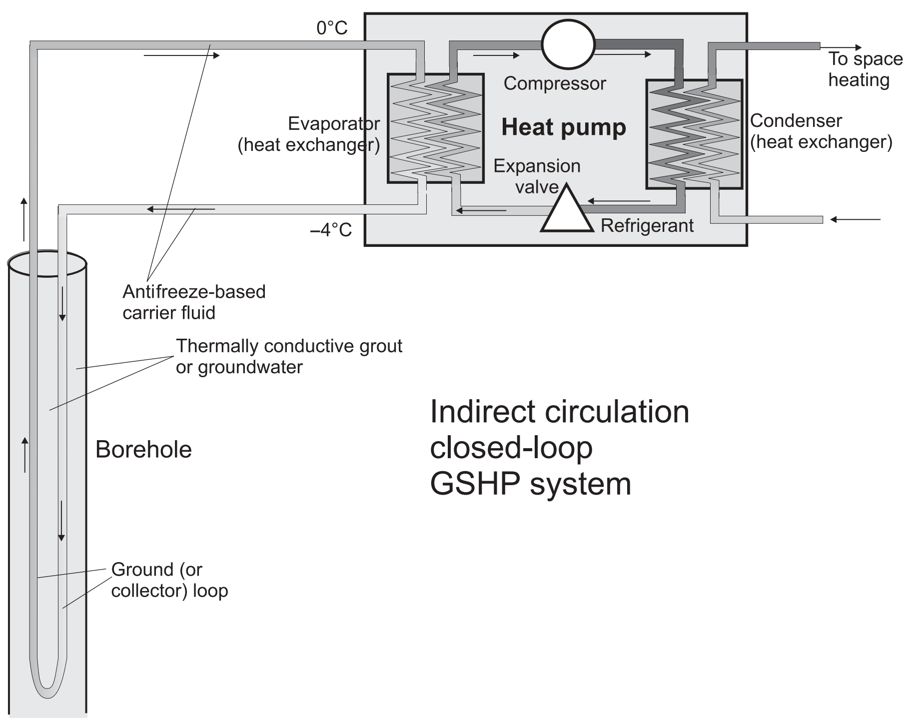


Figure 7.14 A schematic diagram of an indirect circulation closed-loop scheme, installed in a borehole.

earth. It emerges cooled to re-enter the heat pump or the building's cooling network. The closed loop thus functions as a 'subsurface heat exchanger'. Some practitioners refer to such solutions as 'geoexchange' of heat.

Sumner (1976a) initially used a copper pipe in his pioneering British closed-loop installations, but concluded that the thermal resistance of the pipe was probably of little significance to the system performance as a whole. Thus, nowadays, we typically use polyethene (PE) pipe, which has a lower thermal conductivity than copper but is much cheaper, more durable and corrosion resistant. The pipe is typically of 26- to 40-mm outer diameter (OD), although pipe diameters as high as 50 mm have been utilised in Scandinavia (Skarphagen, 2006). The pipe material is usually rated to withstand fluid pressures of 10–16 bars, although during operation, the carrier fluid is pressurised at up to around 2–3 bars (C. Aitken, *pers. comm. 2007*; GSHPA, 2011), as measured at the surface.

The carrier fluid is usually water based and is typically an antifreeze solution, allowing the fluid to be chilled to temperatures below 0°C, if necessary. The antifreeze may be a solution of ethylene glycol, propylene glycol, ethanol or a salt (among others – see Section 9.8). Carrier fluid freezing points of between –10 and –20°C are typical; monoethylene glycol has a freezing point of –11°C at 25% strength and –18°C at 33%. Regulatory authorities tend to be somewhat less concerned with the possibility of leakages of glycols and alcohols than they are of halogenated organic refrigerants. Glycols and alcohols tend to biodegrade rapidly in the environment, whereas propylene glycol and ethanol have rather low toxicity. Ethylene glycol has a somewhat higher toxicity, however, and is a concern to some environmental regulators. The fluid flow rate and pipe diameter are selected such that

- transient–turbulent flow conditions are achieved in the subsurface closed loop (turbulent conditions facilitate heat transfer from the ground to the fluid);
- hydraulic head losses (and thus energy expended on circulation pumping) are kept acceptably low;
- the fluid flow rate is compatible with the heat pump rating and can convey the required amounts of heat.

Flow rates of 3–3.5 $L\,min^{-1}$ per kilowatt of heat transfer are typical. Under typical peak operating conditions in heating mode, one might aim to achieve a *minimum* average carrier fluid temperature of 0 to –2°C. For example, the downhole flow temperature might be –3 to –4°C and the return temperature from the borehole might be around 0°C. Note that the fluid viscosity (and thus the threshold for turbulent flow) will depend both on antifreeze type, its concentration and its temperature: this may be an important consideration in designing carrier fluid antifreezes and flow rates. More detailed consideration of such factors can be found in Chapter 9.

Note that there is always a trade-off between loop operating temperature and heat pump efficiency. In heating mode, low loop temperatures increase conductive heat transfer from the ground (more heat transfer per metre of pipe), but will result in a lower heat pump COP_H. This will thus ultimately result in a trade-off between capital cost (metres of heat exchanger installed in the ground) and running cost/carbon efficiency.

7.4.3 Horizontal closed loops

One of the cheapest forms of indirect closed-loop scheme is the horizontal closed loop, installed in a trench. The optimal depth for such a trench is regarded as c. 1.2–2 m (although some will argue that a depth as shallow as 1 m is adequate). This depth is

- one that can be practically excavated using a mechanical excavator;
- deep enough to provide a sufficient (though modest) thermal storage to support a heating scheme during winter and a reasonable soil moisture content, and to isolate the loop from the worst winter frosts;
- shallow enough to allow solar and atmospheric heat to penetrate and replenish the thermal storage around the loop during the summer months.

Horizontal loops thus function largely as subsurface solar collectors.

As a rule of thumb, a single, straight PE pipe installed in a trench is judged (Rosén *et al.*, 2001) capable of supporting an installed heat pump capacity of 15–30 W m^{-1} of trench in a heating scheme (i.e. 33–67 m per installed kilowatt), depending on soil type, moisture content and length of heating season. The drier the soil, the more pipe is required. It is worth noting, however, that Sumner's (1976a) colleague, Miss M. Griffith, obtained impressively high steady-state heat absorption rates of 29–58 W per linear metre for pipes buried in London Clay. The recent MIS (2011a,b) British guidelines recommend specific peak heat extraction rates (i.e. heat absorbed from the ground) of around 7–16 W m^{-1} for typical British soils and 1800–2400 h year^{-1} full load equivalent (FLEQ) running hours: these rather modest figures reflect the desire for highly efficient operation and with the carrier fluid entry temperature to the heat pump not falling below 0°C.

To increase this output, it is common practice to bury not simply a single PE pipe in our trench, but overlapping coils of pipe. This arrangement goes by the name of a 'slinky' (Figure 7.15). Such slinkies are formed from pre-coiled PE pipe and are typically installed in relatively wide trenches, with a coil diameter of c. 0.6–1 m. Different sources claim rather different outputs for slinkies, although they typically cite figures of around 10–15 m of slinky-filled trench to support each 1 kW of installed heat pump. Figure 7.11b suggests that recent UK practice results in between 10 and 14 m of trench being excavated per installed kilowatt output, with an average of 10.5 m. These figures are dominantly based on heating only schemes. The recent MIS (2011a,b) British guidelines are again somewhat more conservative (see above), recommending specific peak heat extraction rates (i.e. heat absorbed from the ground) of around 20–50 W per metre of trench for typical British soils and 1800–2400 h year^{-1} full load equivalent

Figure 7.15 Two photos showing the installation of slinky-based horizontal closed-loop systems in trenches. The right-hand photo shows the installation of slinky-based horizontal closed loops in trenches on a hillside, spaced at about 4–5 m from each other. *Photos reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

running hours. Where more than one slinky trench is installed, parallel trenches should be at least 3 m, and preferably 5 m, apart. Chapter 11 contains more detailed consideration of the reasons why such 'rules of thumb' appear to exist and the different configurations of horizontal closed loop that are practicable.

Where the horizontal loop is used predominantly for cooling rather than heating, a more cautious approach may be required, because (1) the buried pipe must be adequate to shed the heat from the compressor as well as the heat from the building and (2) the dumping of waste heat may tend to dry out the soil, reducing its thermal conductivity and specific heat capacity.

7.4.4 Pond and lake loops

Coils of PE heat exchange pipe can also be installed in deep ponds, rivers, canals or lakes. For this to be an appropriate solution, the lake should ideally be at least 3 m deep, to ensure that natural temperature variations at its base are low. The lake should also be large enough, so that the heat extracted (or dumped) by the heat pump does not change the temperature of the lake water by an unacceptable amount. Remember that some aquatic species may be sensitive to large temperature variations for several reasons: (1) they may be directly sensitive to temperature (see Turnpenny and Liney, 2007), (2) increasing temperature will reduce the solubility of gases such as O_2 and (3) temperature changes may affect the productivity of the water (e.g. reproduction of algae). Pond loops will be considered in greater detail in Chapter 12.

The use of a restored, flooded gravel pit or opencast mine as a source of heating and cooling can greatly assist in gaining permission for quarrying activities and can potentially convert a potential environmental liability to a redevelopment asset.

7.4.5 Vertical closed-loop arrays

If we have a large amount of available space at our development site, horizontal trenched installations may be the cheapest means of installing a ground loop. At many sites, however, ground area is at a premium. A far more space-efficient means of installing geoexchange pipe is via vertically drilled boreholes.

Such boreholes are not nearly as sophisticated as the water wells discussed in Section 7.3. Although a short length of permanent casing may sometimes be installed and grouted in the uppermost section of the borehole, to prevent surface contamination entering the subsurface, such boreholes are typically drilled either 'open hole', or using only temporary casing in loose rocks or sediments.

Drilling techniques

Several different drilling techniques are available for drilling closed-loop boreholes. The drilling method of choice will depend on lithology. In Sweden and Norway, where very hard, crystalline rock is overlain by a relatively thin veneer of loose Quaternary sediments, a drilling method known as 'down-the-hole hammer' (DTH) can be used in almost all circumstances. This is essentially a drilling bit, embedded with tungsten carbide buttons, mounted on a slowly rotating compressed air hammer. The hammer is powered by a stream of compressed air pumped down the drilling stem. This airflow returns up the annulus of the borehole, removing drilling cuttings (Figure 7.16). The method is good for many different rock types and is capable of drilling a 100-m borehole in granite in less than 1–2 days. The method is not especially good in heavy clay strata or strata containing large boulders or fragments. Because the geology in FennoScandia is more uniform (in terms of geotechnical properties) than the United Kingdom, many Nordic firms operate a single (DTH) drilling method to a highly standardised procedure, leading to relatively cheap drilling rates.

In more geologically varied countries, such as the United Kingdom, the driller will usually need to maintain a greater range of drilling rigs. The greater prevalence of poorly consolidated rock means that temporary casing may be needed more often. The cost of drilling is thus generally higher, and more variable, than in Scandinavia. While DTH techniques may commonly be applicable, other techniques may be preferred by drillers. These include 'conventional' rotary drilling (using mud, foam or air as drilling fluids) and even percussion (cable tool) drilling. In this latter technique, a large chisel-shaped bit is repeatedly dropped on the end of a cable: it sounds primitive but can achieve reasonable results in certain lithologies, such as the British Chalk.

Other techniques that are sometimes used include 'push' techniques, where a string of hollow steel drill pipe is pushed down into soft clays and silts. The PE heat exchange pipe is then placed within this and the hollow steel pipe pulled back out of

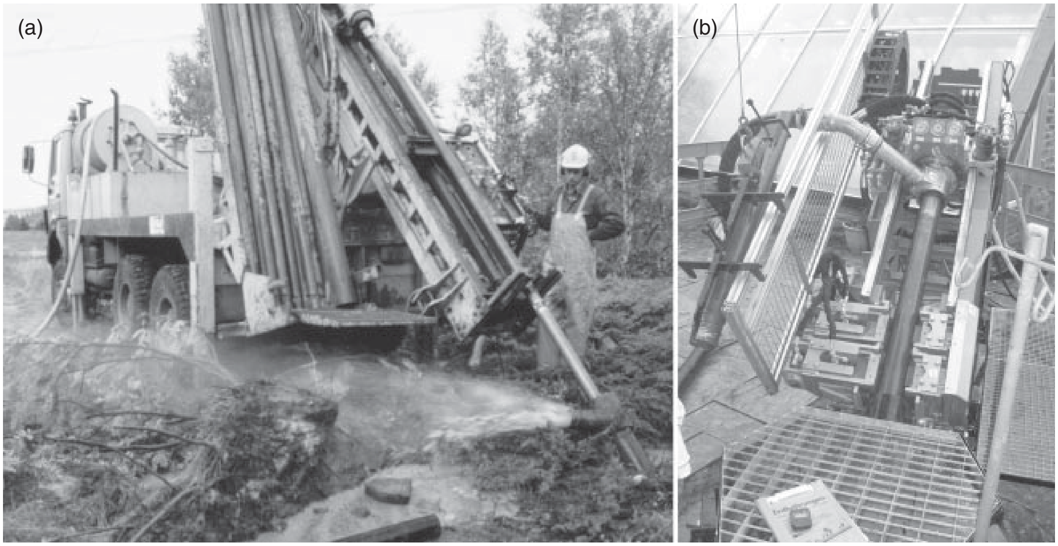


Figure 7.16 (a) A down-the-hole hammer (DTH) drilling rig, operating in Flatanger, Norway. Note that many such rigs can drill at a substantial angle from the vertical. The cuttings and groundwater are blown back up the hole by the compressed air and can be seen gushing out to the left of the borehole; (b) a modern, compact DTH rig drilling a highly angled ground source heat borehole (*photos by David Banks*).

the ground, allowing the natural clay or silt to collapse onto the heat exchange pipe. Commonly, a sacrificial cone-shaped tip will be used on the base of the steel pipe – this is left in the ground after the steel pipe is withdrawn. In some cases, a fluid grout is pumped into the base of the borehole as the steel pipe is withdrawn (to provide a seal between the PE pipe and the borehole wall). Alternatively, the natural ground may simply be allowed to collapse against the walls of the heat exchange pipe as the steel pipe is withdrawn, although in this case, it is wise to use coaxial heat exchange pipe, which (unlike 'U'-tubes) presents a regular, circular cross-sectional profile, against which the natural ground can snugly collapse.

In one variant of the 'push' technique – so-called 'sonic' drilling – a high-frequency vibration is applied to the drill pipe, leading to very rapid penetration rates.

It is probably worth noting that the drilling industry should be tending in the following directions if it intends to service the ground source heat market:

- track- or trailer-mounted drilling rigs with a small footprint that can fit into domestic properties (e.g. small DTH units with a separate compressor that can be parked at some distance from the well head);
- high turnover of specialised GSHP drilling by dedicated crews;
- flexible drilling rigs that can offer a range of techniques. Such trailer-mounted rigs are now available offering DTH compressed air drilling, in combination with conventional rotary drilling using air, foam or mud flush;

- pre-agreed pricing and guaranteed installation of U-tubes. Drilling firms may have to include geological risk and drilling failure as a component in their pricing strategy, rather than simply offloading all risk onto the customer;
- punctual delivery of services, to fit in with a rigorous building completion schedule.

Number and depth of boreholes

We have seen earlier in this chapter that, for open-loop systems, there is a proportionality between rate of groundwater flow and heating load serviced (Equations 7.6–7.9). We have also seen that there is a relationship between the length of trench required and the heating load in horizontal closed-loop systems (Section 7.4.3). We would intuitively expect the number of drilled borehole metres in vertical closed-loop schemes to also be in rough proportion with the heating load delivered. Figure 7.11a indeed demonstrates that there is a relationship between the number of closed-loop boreholes per scheme and the installed heat pump capacity in kilowatts. Indeed, the installed capacity per borehole ranges from 2 to 17 kW. The smallest installed capacities are related to the shallowest boreholes (40 m) and the highest to the deepest boreholes (180 m). If we divide the installed heat pump capacity of the scheme by the total number of drilled borehole metres, we obtain a much more linear relationship, with specific installed thermal outputs of between 37 and 104 W per drilled metre. The average value for the schemes considered in Figure 7.11b is $67\,W\,m^{-1}$, which corresponds to a peak heat extraction rate (from the ground) of $47\,W\,m^{-1}$ for a heat pump COP_H of 3.4. Note that the systems depicted in Figures 7.11 and 7.12 are mostly heating dominated or heating-only schemes, although a few of the larger ones are cooling dominated.

This ballpark figure of c. $50\,W\,m^{-1}$ specific peak heat extraction rate (c. $70\,W\,m^{-1}$ installed peak heat output) is a commonly used rule of thumb in the European ground source heat industry. However, this may not result in terribly efficient systems. The recent MIS (2011a,b) British guidelines recommend specific peak heat extraction rates of around $25–40\,W\,m^{-1}$ thermally active borehole for rock thermal conductivities of $1.5–2.5\,W\,m^{-1}\,K^{-1}$, small 'heating only' systems and $1800–2400\,h\,year^{-1}$ full load equivalent running hours: as mentioned above, these figures reflect the desire for highly efficient operation and with the carrier fluid entry temperature to the heat pump not falling below 0°C over a 20-year design life (see Section 10.11). By all means, use such rules of thumb to get a first 'guestimate' of the amount of closed-loop borehole required for smaller systems, or as a reality check. Do *not* use such rules of thumb for detailed design work – the actual amount of borehole heat exchanger required will depend on the following:

- The ground's thermal properties (undisturbed temperature, conductivity and volumetric heat capacity).
- The duration of the heating/cooling season: a greater number of borehole metres will be necessary to support a 12-kW heat pump with 3200 operational hours per year than one with 500 h.

- The balance between heating and cooling over a year: if heating and cooling loads are approximately balanced, the ground will have no long-term tendency to heat up or cool down over time, leading to less extreme ground temperatures.
- The borehole spacing: where many boreholes are closely spaced in a limited area, they will tend to thermally interfere with each other and 'steal' each other's heat.
- The ground loop's operating temperatures: a warmer carrier fluid (in heating mode) means a smaller temperature differential between the loop and the ground, and thus less conductive heat transfer from the ground (but probably a higher heat pump COP_H). A colder carrier fluid means a greater rate of heat transfer per metre, but a lower COP_H.
- Building load complexity: in large, complex buildings, the total installed heat pump capacity may not be identical to peak heating load: it may be greater, because the building may be divided into different zones, whose peak heating demands occur at different times of day (Kavanaugh and Rafferty, 1997).

For example, the lowest specific installed thermal outputs in Figure 7.11a are typically from relatively large schemes, where the output is strongly dominated by heating or cooling and where the boreholes are rather closely spaced. The largest specific installed thermal outputs are either from spacious sites where the rock has a high thermal conductivity or from schemes where there is a pleasing balance between heating and cooling loads.

If we look at Table 3.4, we see that the thermal conductivity of most British rocks ranges over a factor of around 2–3. It is not wholly coincidental that the specific installed thermal outputs in Figure 7.11a exhibit a similar range. In fact, the only reason why rules of thumb such as '40–100 $W\,m^{-1}$ of installed GSHP capacity in heating mode' appear to have some value is because most rocks have rather consistent thermal properties and their behaviour is not strongly lithology dependent.

Thus, there is no hard and fast rule about how deep a ground source heat borehole should be. In other words, two boreholes to 50 m depth (provided they are situated sufficiently far apart) should support an approximately equivalent heating or cooling load to one borehole to 100 m. However, it costs money to move a drilling rig from one site to another and to complete every new borehole (surface casing, grouting, manifolds, couplings, etc.). Moreover, two boreholes have a greater areal space requirement at a site than a single borehole. These factors may argue for fewer, but deeper, boreholes. On the other hand, drilling penetration rates become slower with depth and drilling becomes more expensive. There may be other reasons related to grouting or to hydraulics why very deep boreholes eventually become undesirable. Thus, there will come a cut-off point where it becomes cheaper and more practical to commence a new borehole than to continue drilling ever deeper in a single borehole. In practice, unless there are overriding considerations of space, ground source heat boreholes tend to be drilled to between 70 and 130 m depth (although, in Scandinavia, greater depths are becoming more frequent).

Emplacement of heat exchange tubes

Once the drilling of a borehole for the closed-loop scheme has been completed, a 'U-shaped' closed loop (or 'U-tube') is usually emplaced down the length of the borehole (although, other configurations of exchange tube are possible – Chapters 9 and 10). U-tubes are usually made of high-density polyethene (HDPE) tubing of 32- to 40-mm OD. A shank spacing (distance between the centres of the uphole and downhole tubes) of around 50–70 mm is typical. Thus, the U-tube installation will have a width of 90–110 mm (Figure 7.17). The diameter of boreholes drilled for closed-loop ground source heat schemes is typically around 120–150 mm. To prevent thermal short-circuiting of heat between the upflow and downflow tubes, shank spacers or clips can be placed down the length of the U-tube to maintain an acceptable shank spacing (Hellström, 2011b). The 'U' on the base of the tube should be a specialist prefabricated element. Indeed, the entire U-tube should be constructed in a quality-controlled environment from a minimum number of lengths of virgin PE and delivered as a complete product on a spool (Figure 7.18), in order to minimise the risk of damage to the tube and thus the risk of eventual leakages. In a water- or mud-filled borehole, there will be buoyancy effects to overcome when emplacing the tube: the tube will be filled with water and pressurised during emplacement, but the 'U'-piece will also usually be weighted, to assist with emplacement.

Once the U-tube is in place, it is common practice (though not the only way of completing a borehole) to grout the space between the U-tube and the borehole wall with some form of grout. The grout should be pumped down to the base of the bore-

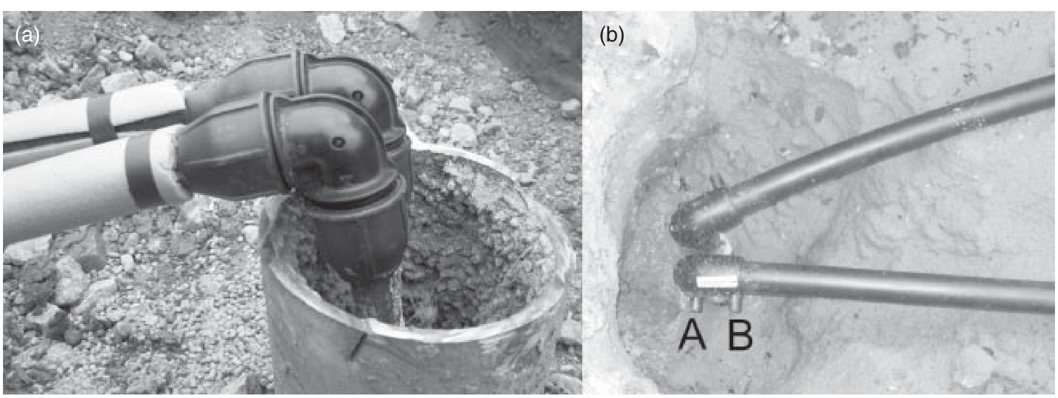


Figure 7.17 Two photos of newly installed polyethene U-tube in grouted boreholes. (a) The two shanks of the tube can be seen, as can the top of a length of steel casing (which was employed in this rather unstable, mined ground). The connections employed here are only temporary compression-type fittings, while the hole was subject to a thermal response test (see Chapter 15). Subsequently, the borehole would typically be permanently completed, with fusion couplings, below ground level. *Reproduced by kind permission of Pablo Fernández Alonso.* (b) A U-tube completed with electrofusion welded couplings to header pipes. The electrofusion terminals are marked A, B. *Reproduced by kind permission of Water and Energy Field Services Ltd.*

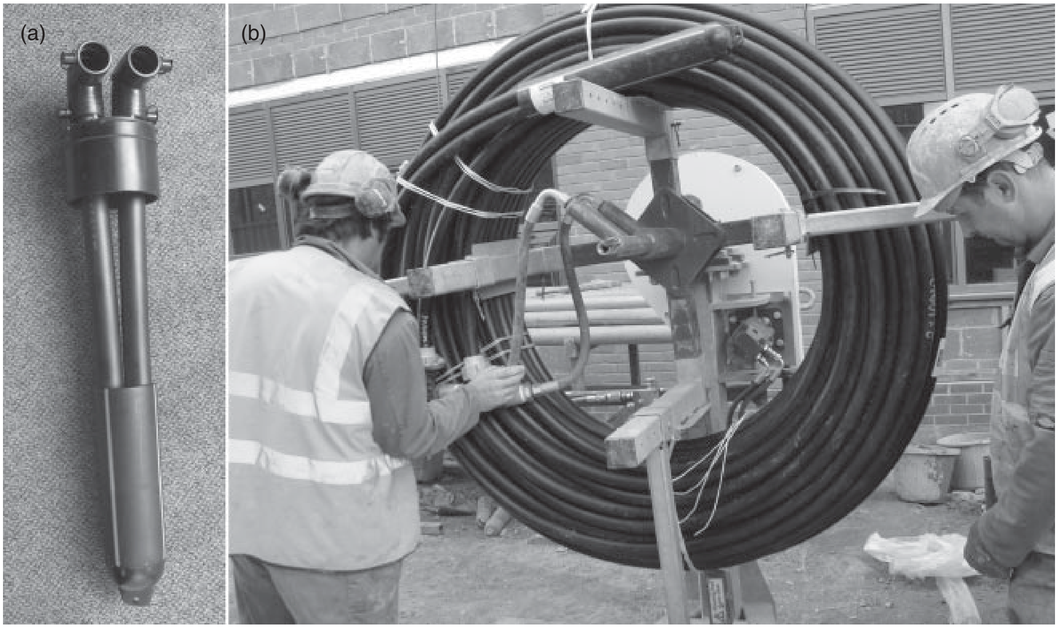


Figure 7.18 (a) A prefabricated U-tube end piece (*photo by D. Banks*); (b) a prefabricated U-tube, delivered to the drilling site on a spool, ready for installation in a borehole (the U-tube end piece can also be seen). *Reproduced by kind permission of Water and Energy Field Services Ltd.*

hole using a 'tremie pipe' – the tremie pipe is gradually withdrawn as the borehole fills up with grout (Figure 7.19). When grouting the U-tube in place, be aware that grout is denser than water and liquid grout may thus exert a significant over-pressure on the U-tube at the base of a deep hole. This grout pressure should be calculated and be taken into account when deciding how to fill and pressurise the loop during installation. Eugster (2011) provided a detailed discussion of grout pressures, fluid pressures and pipe strength. Other methods of backfilling are discussed below.

The ground loop should be connected to the near-surface header pipes (leading back, via a manifold, to the heat pumps) using a quality-controlled PE fusion method (see EN 13244, 2002; UKWIR, 2002). The most common such method is electrofusion, whereby an electric current is applied to two electrodes on a prefabricated electrofusion joint, melting the plastic and producing a robust seal. Other fusion methods may be acceptable, but compression-type fittings should not be used (Figure 7.17).

Alternatives to U-tubes

Other geometries of closed-loop heat exchanger may be installed in vertical boreholes, with the objective of increasing heat exchange area and improving the overall heat transfer efficiency (and thus potentially reducing the required drilled length for a given thermal output).

Figure 7.19 (a) Persuading a U-tube to descend a fluid-filled borehole can be trickier than it sounds, and it can also be messy. (b) Pumping grout into a closed-loop borehole. The reel of the tremie pipe and the tank containing the grout are mounted on the lorry. *Photos reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

Rather than using a single 'U'-tube in a borehole, it is possible to install a double 'U'-tube (two downflow pipes and two upflow pipes, plumbed in parallel), or even a triple 'U'-tube. Such multiple 'U'-tubes have the advantages of increased heat exchange surface area and a lower hydraulic resistance to carrier fluid flow. On the negative side, they can be tricky to install and grout in a narrow borehole. Moreover, quite large fluid flows are required to achieve transient-turbulent conditions in double/triple pipes and the sheer volume of carrier fluid required to fill them can be a significant cost factor. The decision whether to use a double 'U'-tube will often depend as much on hydraulic as heat transfer factors; this will be discussed in more detail in Chapters 9 and 10.

The use of coaxial heat exchange pipe (a narrow-diameter pipe within a larger-diameter outer pipe) has been mentioned above. The main advantage of this heat exchanger is that it presents a narrow, circular cross section: it can be installed in narrow boreholes, maybe using 'push' techniques. Soft ground can simply be allowed to collapse or squeeze against the pipe walls, removing the necessity for grouting. It is worth noting that some environmental authorities have concerns regarding non-grouted installations: if the borehole walls do not squeeze tightly against the coaxial pipe walls, a pathway for surface pollution into the ground may remain.

Finally, Rosén *et al.* (2001) reported the use of 'vertical slinky' heat exchange pipes (also known as 'Svec' coils). These simply comprise helical coils of tubing, emplaced in somewhat larger-diameter drilled boreholes.

Completion of boreholes

Several options exist for borehole completion (Figure 7.20):

Open, water-filled well: the U-tube may be suspended in a groundwater-filled borehole: the water provides a thermal contact between the rock and the U-tube. This technique is only practical and cost-effective (1) in competent lithologies, where the borehole walls are self-supporting and (2) where the water table is near the surface, such that the entire U-tube is thermally 'active'. Although water only has a modest thermal conductivity, the efficiency of heat transfer can be dramatically improved by (1) the formation of free convection cells in the groundwater column of the borehole and in the surrounding formation (the so-called thermosiphon effect), and (2) forced convection of heat with natural groundwater flow through the borehole array or (if vertical hydraulic head gradients are present) along the borehole axis. In heating mode, it is possible that a limited amount of ice may form around the heat exchange pipe (a so-called 'ice sausage'). A modest amount of ice formation is not necessarily a problem – in fact, ice has a higher thermal conductivity than water (Table 3.1).

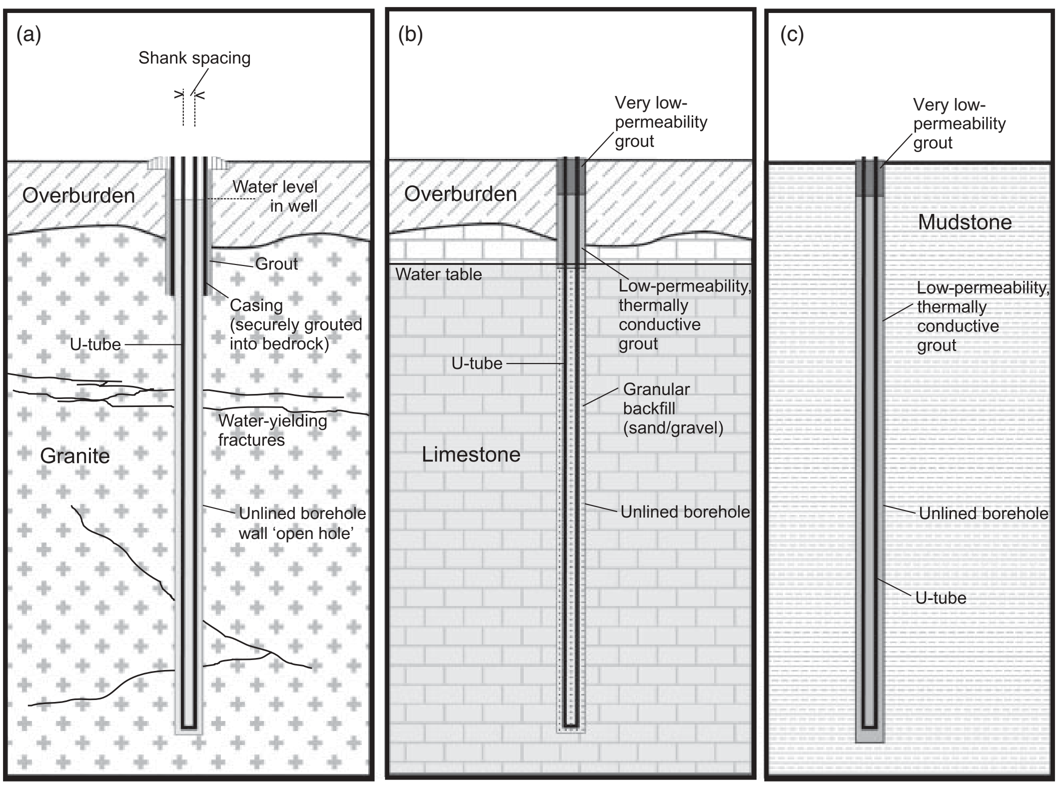


Figure 7.20 Schematic diagrams illustrating possible options for the installation of a U-tube in a borehole: (a) suspended in a groundwater-filled well, (b) backfilled with sand/gravel and (c) backfilled with a thermally efficient grout. For (a) and (b) to function well, the groundwater level must be high.

Excessive ice formation is undesirable, however, as it can lead to problems with (1) buoyancy of the loop, (2) obstruction of free groundwater flow along the open borehole, (3) compression/damage of the plastic pipe. This 'open well' borehole completion technique has been favoured in the crystalline rock terrain of Sweden and Norway. Needless to say, you should check if this type of completion is permitted in your country (e.g. EA, 2011a). You should always seek hydrogeological advice and construct such boreholes responsibly, as proper water wells. This type of completion should only be attempted in single, non-artesian, aquifers. The well should be completed with watertight headworks and grouted surficial casing to prevent contamination of groundwater by shallow or surficial water run-in (see EA, 2000).

Porous backfill: In this case, the section of borehole below the water table might be filled with a quartz-rich clean gravel or sand. The upper portion would be sealed with a low-permeability (and, preferably, high thermal conductivity) grout, which would prevent ingress of any surficial pollution to the borehole. This construction allows any temporary casing to be extracted, if the borehole was drilled in poorly lithified strata. Both the quartz grain matrix (remember quartz's high thermal conductivity) and the mobile groundwater filling the pore spaces provide efficient heat transfer between the borehole wall and the U-tube. Again, this type of completion should only be attempted in non-artesian aquifers and never in multiple aquifers. Check that this construction is permitted by the environmental regulator and seek hydrogeological advice (EA, 2011a).

Grout backfill: The most common form of borehole completion is to seal the heat exchange pipe into the borehole with a grout (Hiller, 2000). The grout serves several purposes:

- It protects the heat exchange pipe.
- It provides a thermal contact between the heat exchange pipe and the natural ground in the borehole wall.
- It provides a low permeability seal along the borehole axis, preventing leakage of contaminated surface water or shallow groundwater down into the deeper groundwater environment, and preventing leakage of groundwater along the borehole axis from one aquifer horizon to another.
- It may serve to hinder leakage of antifreeze into the groundwater in the event of damage to the heat exchange pipe.

Thus, the grout should ideally have a high thermal conductivity (to facilitate the transfer of heat) and a low hydraulic conductivity (to prevent fluid leakage). Most international standards (IGSHPA, 2007; EA, 2011a) recommend a hydraulic conductivity of 10^{-9} m s^{-1} or less. The grout should also be inert (when set), non-contaminating, non-shrinking and (if the ground loop is likely to run at sub-zero temperatures) frost resistant. The thermal conductivity can be provided by a high quartz content (Table 3.1), while the low hydraulic conductivity can be provided by a clay matrix, such as bentonite. In fact, one of the most common types of thermally enhanced grouts comprises a suspension (typically in a ratio of around 6:1) of fine quartz sand and bentonite clay in water. Swedish researchers (Rosén *et al.*, 2001) have cast some doubt on the use of bentonite in grouts, suggesting that water molecules, trapped within the bentonite structure, might form discrete pockets of ice if the grout freezes, compromising the grout integrity and imposing stresses on the U-tube. It is claimed that thermally enhanced cement-based grouts provide an alternative option, with quite a reasonable thermal conductivity. Such cement-based options include (1) a suspension of HOZ (blast furnace cement) cement, bentonite and fine silica sand (VDI, 2001a) or (2) a cement-based grout comprising sand, cement and a superplasticising agent, sometimes with a content of bentonite or fly ash.

The *open-hole* and *porous backfill* options require that the borehole be largely filled with groundwater in order to obtain good thermal contact. *Grout backfill* is the only realistic option of providing a good thermal contact for a 'dry' borehole or a borehole above the water table. The grout backfill option may often be preferred by environmental authorities (and may even be obligatory), for the pollution prevention and water resources reasons outlined above.

Capital cost

Figure 7.12 shows the reported capital cost for a number of real or projected GSHP schemes, mostly in the United Kingdom. We can note that capital costs per installed kilowatt ranged from around £400 to £1500, although the exact figures should be taken with a 'pinch of salt' for several reasons: (1) they are out of date (some of the cited schemes were developed over 8 years ago) and (2) they are largely reported by consultants or installers. We do not know whether the figures included, for example, installers' mark-ups, internal building-side works or consultancy services. In fact, it is estimated that the cost of commissioning works from a typical British installer may currently be at least 50% greater than the values indicated by Figure 7.12. More importantly, we should note the following:

- The capital costs of horizontal, trenched systems are considerably less than borehole-based systems (no drilling involved).
- The capital cost per installed kilowatt is scale dependent, declining with increasing size of scheme. These data support our assertion that ground source heating and cooling becomes increasingly economically attractive the larger the scheme (Section 4.14). This is partly due to the decreasing cost of the heat pump itself (per installed kilowatt – Figure 4.10), but is also due to design/consultancy costs and the mobilisation costs of a drilling rig being spread over many holes on a site in a large scheme.
- The capital costs per installed kilowatt of open-loop, groundwater-based schemes are not too dissimilar to closed-loop, borehole-based schemes, although the costs of drilling water wells for the open loop scheme will be heavily dependent on the aquifer lithology.

According to one British firm, the breakdown of a typical large, closed-loop contract worth £150 000 might comprise 60% drilling; 20% heat pump supply; 10% labour; 7.5% manifolds, headers and antifreeze; and 2.5% design consultancy.

7.4.6 Energy piles

Some engineers have noticed that, in many geotechnical situations, large buildings require foundations composed of drilled reinforced concrete pile structures. If it is necessary to drill such holes for purely structural reasons, why not 'kill two birds with one stone' and install a closed-loop system within the pile structure? Such 'energy piles' or 'geo-piles' will often be 15–40 m deep and may be over 1 m in diameter. One would typically install multiple U-tubes within a single pile and ensure a large shank spacing between the individual pipes. In principle, however, they function in exactly the same manner as purpose-drilled vertical closed-loop borehole schemes. When considering the use of energy piles, we should remember that the pile's primary function is structural – it supports a building! We should be very sure that the extraction or disposal of heat via a pile does not

- result in ice formation or frost heave in the ground;
- cause significant thermal expansion of the ground;
- change the bearing capacity of the pile, due to temperature-related changes in moisture content, geotechnical properties or ice formation around the pile. It is likely that frictional piles (i.e. piles that rely solely on frictional resistance between the pile surface and the surrounding sediments for their bearing capacity) will be more vulnerable to such effects than end piles (piles that fully penetrate unstable ground, such that their lower ends are founded on rockhead).

One would probably be much more cautious of extreme loop operating temperatures in an energy pile than in a closed-loop borehole that performs no structural function.

7.4.7 Heat pipes

A final form of vertical, borehole-based heat exchanger that is becoming fashionable is the heat pipe (Figure 7.21; Vasil'ev, 1987). This is effectively a deep cylinder, installed in a borehole and containing a refrigerant fluid. The cylinder is capped by a chilled heat exchange block that is coupled to the evaporator of the heat pump. The refrigerant liquid at the base of the cylinder, deep in the warm earth, volatilises and rises as a vapour (with a cargo of latent heat) to the top of the heat pipe. It comes into contact with the chilled cap, condenses, sheds its cargo of latent heat to the heat pump's refrigerant fluid and runs back down the walls of the pipe to collect at the base and re-volatilise.

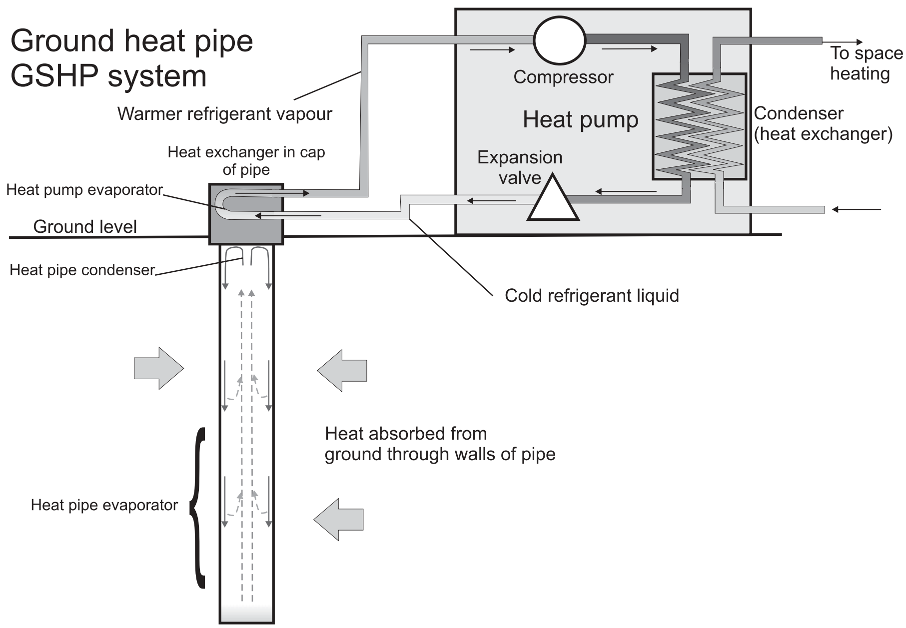


Figure 7.21 Schematic diagram of a vertical heat pipe, coupled to the evaporator of a heat pump.

The advantages of heat pipes include their lack of any subsurface moving parts and the fact that the refrigerant used can be environmentally benign (e.g. CO_2; Ochsner, 2008b). Such ground-coupled heat pipes can, however, only run in heating mode. They cannot be used to provide cooling, as the heat pipe evaporation–convection–condensation cycle cannot be reversed.

7.5 Domestic hot water by ground source heat pumps?

We have seen in Chapter 4 that GSHPs are particularly good at producing warm fluids (water or air) at temperatures of up to 50°C, which can be used for space heating. We have also learned that they can produce temperatures higher than this, but that the efficiency of the heat pump declines significantly with increasing temperature of delivery.

Because of the risk of legionellosis, caused by the proliferation of *Legionella* bacteria in warm (20–45°C) water systems, many countries will insist that DHW, at least for certain industrial or public service sectors, is stored or regularly sterilised at a temperature in excess of 60°C, above which the bacterium is killed (Box 7.3). We could thus use a GSHP simply to pre-heat DHW and use a secondary heat source to raise it to the final desired temperature.

BOX 7.3 *Legionella*

Legionellosis (including the forms known as Legionnaire's disease, Pontiac and Lochgoilhead fever) is a lung infection caused by the genus of bacterium *Legionella.* The bacterium is common in nature and exposure to it need not necessarily cause any symptoms. However, it can proliferate in water stored at elevated temperatures (20–45°C), and people may be at particular risk where high concentrations are transmitted in aerosol form, sometimes over large distances, and subsequently inhaled. Thus, the types of installation that can lead to a risk of legionellosis include hot water storage and distribution systems and some components of large-scale water-borne cooling systems, such as wet cooling towers or evaporative condensers. The disease can usually be controlled by antibiotics.

A combination of several strategies is often recommended for controlling the risk from legionellosis, including system maintenance, thermal or chemical disinfection and regular cleansing or replacement of any filters or strainers. As *Legionella* cannot survive in water at temperatures in excess of 55°C, one element of a risk-control programme is to ensure that any store of domestic hot water is maintained or regularly sterilised at a temperature in excess of this value. Indeed, at temperatures greater than 60°C, the bacterium is killed in less than 30 min. In many countries, there will even be legal requirements to assess risks and to enforce a *Legionella* control policy (HSE, 2000; Makin, 2009).

Legionella is named after a fatal outbreak of the disease at a convention of the American Legion in Philadelphia in 1976.

If we wish to use a GSHP to maintain a DHW tank above 60°C all the time, we will struggle and we may end up with a rather inefficient system. We should remember, however, the following:

- We probably don't need our DHW to be at 60°C. A temperature of 45°C is sufficient for most purposes – indeed, >50°C becomes uncomfortable to touch and bathe in.
- It may not be necessary to store DHW continuously at 60°C to obviate the *Legionella* risk. Indeed, it may be satisfactory to simply lift the temperature of the tank above 60°C for a certain period at regular intervals (Makin, 2009, suggested 1 h >60°C every day). This could simply be done by a supplementary immersion heater or other back-up heater, but care must be taken that the entire storage system be elevated to the appropriate temperature, which may involve some form of recirculation. Check the legislation on *Legionella* in your own country.
- Remember that, as the DHW tank is exhausted, it is replenished by cold mains water. We may only be rejecting the heat from the compressor at a high temperature at the very end of the heating cycle. At the start of the heating cycle, the tank will contain cold mains water, and we may start off rejecting heat to a tank at 10–20°C.

The GSHP efficiency will decrease as the tank heats up, but the overall efficiency for the entire heating cycle may still be acceptable.

If we have already chosen to use a GSHP to provide space heating, several strategies can be used to provide DHW.

7.5.1 Heat pumps that deliver DHW – the desuperheater

The amount of heat that we typically use for DHW (Figure 7.3) is often substantially less than that used for space heating (although, as buildings become far better insulated, the proportion will increase). Thus, it is possible to buy a heat pump with a gadget called a 'desuperheater' (Figure 7.22). This is a small heat exchanger, located between the compressor and the condenser, where a small amount of high-temperature heat is skimmed off from the hot refrigerant gas and transferred to the DHW system.

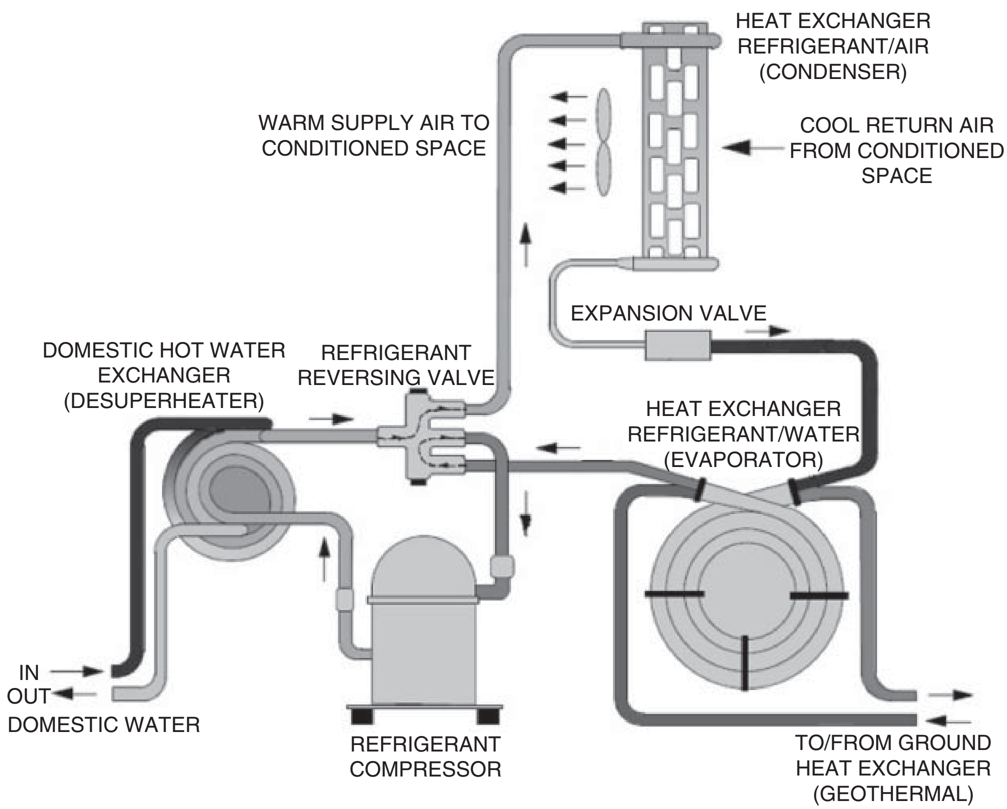


Figure 7.22 A heat pump with a 'desuperheater'. Note that the heat pump is reversible and can be switched from heating to cooling mode via the 'refrigerant reversing valve'. *After IGSHPA (1988) and reproduced by kind permission of the © International Ground Source Heat Pump Association/ Oklahoma State University.*

This provides householders with a total solution to space heating and DHW, although it introduces another layer of complexity to the system and, inevitably, the coefficient of performance of the heat pump system suffers.

7.5.2 A two-stage approach

Imagine that we use a conventional GSHP to transfer heat to a building loop of fluid at, say, 40°C that delivers space heating via an underfloor heating circuit. We could envisage a spur of the building loop that supplies warm fluid to a second heat pump, which extracts a small amount of heat from the building loop at 40°C and transfers it to a DHW circuit at 55–60°C. Because the temperature 'step' is relatively modest, this secondary heat pump performs relatively efficiently. However, again, the total system performance will inevitably still be lower than for a system solely providing space heating at 40°C. We should also note that some of the refrigerants that performed best in vapour compression heat pumps with high evaporator temperatures have now been restricted as being ozone unfriendly. Most modern environmentally benign refrigerants prefer evaporator temperatures up to 20°C. However, Rakhesh *et al.* (2003) have demonstrated that, by careful choice of refrigerant, efficient systems can be constructed to operate at high evaporator temperatures.

A two-stage system, as described above, sounds a little untidy. We can, however, purchase a single machine that effectively performs the two-stage operation within a single box. Figure 7.23 shows such a two-stage heat pump schematically. It naturally possesses two compressors and two condensers, each operating at a different tempera-

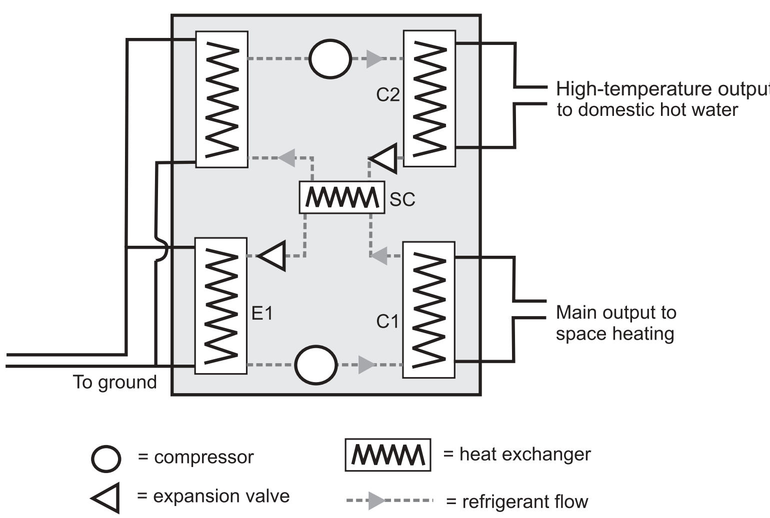


Figure 7.23 A schematic diagram of a two-stage heat pump, with a larger, lower-temperature condenser (C1) for space heating and a smaller, higher-temperature condenser (C2) for domestic hot water. E1 is the main evaporator and SC is the sub-cooler heat exchanger.

ture. It also contains an internal heat exchanger (the sub-cooler), which recovers some heat from the condensate of the primary refrigerant circuit and transfers it to the DHW circuit.

7.5.3 Tank-based solutions

Figure 7.24a shows a relatively typical circuit for a GSHP, incorporating a buffer tank as a thermal store, provision of space heating and provision of DHW. The DHW is effectively produced by passing the warm fluid from the heat pump through a heat exchange coil within a DHW storage tank. A secondary heat source (such as an electric immersion heater) can be used to periodically raise the water temperature over the required *Legionella* threshold. An alternative eco-friendly heat source, such as solar thermal panels, can also be coupled into the circuit.

Of course, if our GSHP is dominated by cooling in the summer, DHW can be produced as a free by-product of the cooling, by connecting the DHW to the condenser (ground side) of the heat pump. Any surplus waste heat is rejected to the ground (Figure 7.24b).

Finally, we could use our heat pump to produce a large thermal store of moderately hot water at, say, 55°C. As the store is a sealed system and not drawn upon directly to produce DHW, exposure to *Legionella* is minimised (and biocidal agents could also be added, if necessary). Our DHW supply is produced by simply circulating cold mains water through a heat exchange coil within the thermal store (Figure 7.24c). This is a type of 'combi' solution, without any storage of DHW and thus with low *Legionella* risk (do check your national legislation however). The drawback is that it will take a certain time for the water to run hot, however.

7.6 Heating and cooling delivery in complex systems

7.6.1 GSHPs to provide cooling – a summary

We have already established (Chapter 6) that GSHPs can be reversed to provide for active cooling. The larger the building, the smaller the surface area-to-volume ratio (typically) and the more likely the building is to have a cooling requirement for prolonged periods of the year. We have seen that promoting active ground-sourced cooling as an 'environmentally friendly means of cooling' is a slightly dubious practice: true, using GSHPs for cooling may be 20–40% more efficient (Kelley, 2006) than conventional cooling (disposing waste heat to air) and have a lower visual impact. Nevertheless, we are still essentially using energy to throw heat away! Cooling solutions by means of 'passive' or 'free' cooling, which utilise the natural temperature gradient between the building and the earth are preferable in terms of environmental impact, but may have a greater capital cost.

The situation where active cooling (using a heat pump to dispose of heat to the ground) begins to look attractive is where we have approximately balanced heating

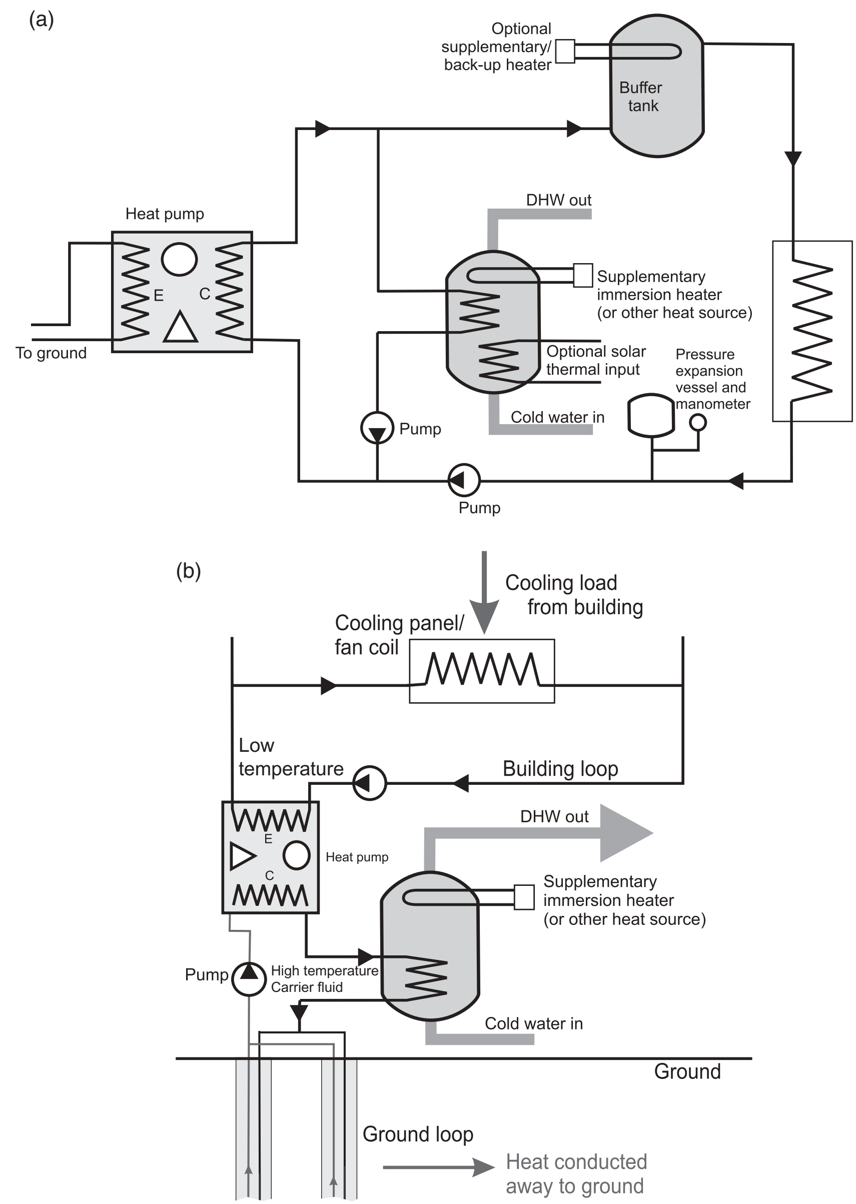


Figure 7.24 (a) A standard heat pump circuit, providing both domestic hot water and space heating and incorporating a buffer tank on the flow line; (b) a ground-sourced active cooling system, providing DHW from the condenser. The condenser is also coupled to an indirect closed-loop ground heat exchanger, although an open-loop groundwater-coupled option would also be plausible; (c) a GSHP supporting a sealed thermal store, through which domestic hot water is heated via a combi-type heat exchange coil. Note: the circuits shown are schematic only and omit many valves and other components.

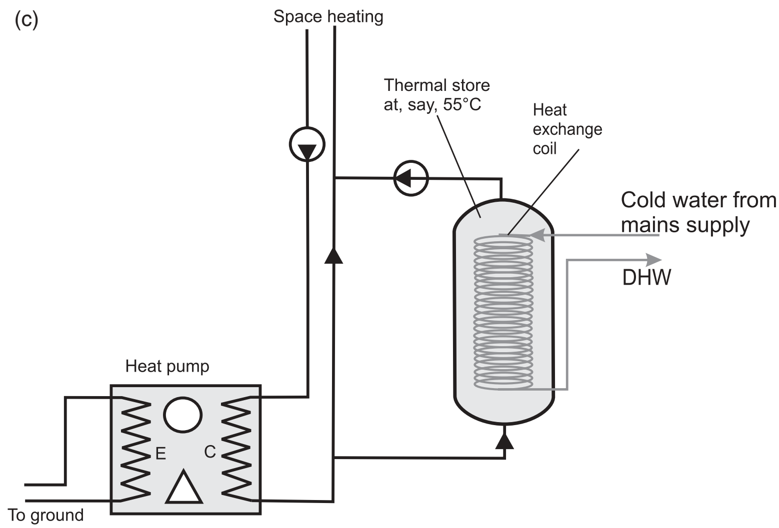


Figure 7.24 (*Continued*)

and cooling loads to the ground. In this case, the subsurface can be used to store the summer's waste heat, such that it can be re-extracted during the winter. Such underground thermal energy storage (UTES) schemes will be discussed further in Chapter 14. If (as is often the case for large buildings in temperate or southern Europe) the waste heat from cooling is larger than the heat extracted for space heating (i.e. a net cooling scheme), we can bring the heat fluxes to and from the ground back into balance by dumping a proportion of the waste heat to air via conventional cooling towers. Such hybrid schemes increase the likelihood of obtaining the subsurface thermal balance that is important for the long-term sustainability of UTES concepts (Spitler, 2005).

7.6.2 Centralised systems

A 'centralised' GSHP scheme is one that has a centralised plant room containing the heat pump(s). The heat pumps are typically either in heating or in cooling mode and provide space heating or cooling to the entire building or a specific zone of a building. A domestic heat pump scheme is an example of a small-scale centralised scheme.

Space heating can be provided in centralised schemes by means of warm air circulation (in a water-to-air or brine-to-air heat pump) or by means of a building loop containing warm water and feeding an underfloor heating system, a wall-mounted radiator system or fan coil units. In reverse (cooling) mode, such underfloor and wall-mounted radiators, through which a cool fluid is circulating, are not especially efficient at delivering a large cooling effect (furthermore, they may lead to problems of condensation). Cooling is much more efficiently achieved by means of fan coils or high-level structural elements such as chilled beams or panels.

We can see that, for buildings where there are heating and cooling demands at different times, the degree of overlap between the delivery systems is limited. Both heating and cooling can be delivered by air, however. Thus, an efficient solution can be for a heat pump to produce warm or cold water, which is fed, via a network of pipes, to fan-driven water-to-air heat exchangers in different parts of the building, where the water-borne heat or coolth is exchanged with air. Fan coil units are perhaps the most common form of such a heat exchanger.

7.6.3 Simultaneous heating and cooling

In large buildings, we must recognise that different parts of the building may have different heating and cooling requirements at the same time. We have already met, in Section 6.2, the example of a university building containing a computer lab of sweaty students on its south side (cooling demand) and a lonely professor in a north-facing garret with a heating demand. How can a GSHP system supply both heating and cooling at the same time? One solution is, of course, to divide the building into separate zones and have a heat pump array (in either heating or cooling mode) supply each zone independently.

More elegant solutions are available to us: even with a single, centralised heat pump array, we can provide both heating and cooling. We simply need to remember that every heat pump has a 'hot' and a 'cold' side.

Let us imagine that, in winter, our building has a net heating demand, but the computer lab has a cooling demand. The ground loop and the computer lab would be coupled to the evaporator of the heat pump. The cold ground-coupled carrier fluid would absorb heat from the computer lab (i.e. provide cooling, maybe via fan coil units) and also from the ground. The sum of this heat would be transferred, via the heat pump condenser, to the building's heating circuit (and a modest additional amount of heat would be added by the heat pump's compressor) (Figure 7.25a).

Let us assume (for the sake of easy maths) that the heat pump's COP_H is 4 and that the heat demand is 100 kW. This means that 25 kW comes from the compressor. Of the other 75 kW, maybe 35 kW (for example) is supplied by the ground and 40 kW by heat removed from the computer lab. Some installers would argue that the overall COP of the system is given by total effect (heating effect plus cooling effect), divided by the electricity input to the compressor. Thus, they would argue that the system COP is (100 kW + 40 kW)/25 kW = 5.6, rather than 4. Such installers have a point, such a system is very efficient . . . but it is very important to be clear about how the COP is being defined, in order to avoid double-accounting misunderstandings! Note that, as much of the heating load is being sourced from a cooling demand, rather than from the ground, this may allow potential capital savings in terms of drilled metres of borehole.

With a little imagination, we can likewise see that, for a building with a net cooling demand, heating can be provided to some zones by connecting them to the 'hot', ground-coupled side of the heat pump (Figure 7.25b). This is essentially the same

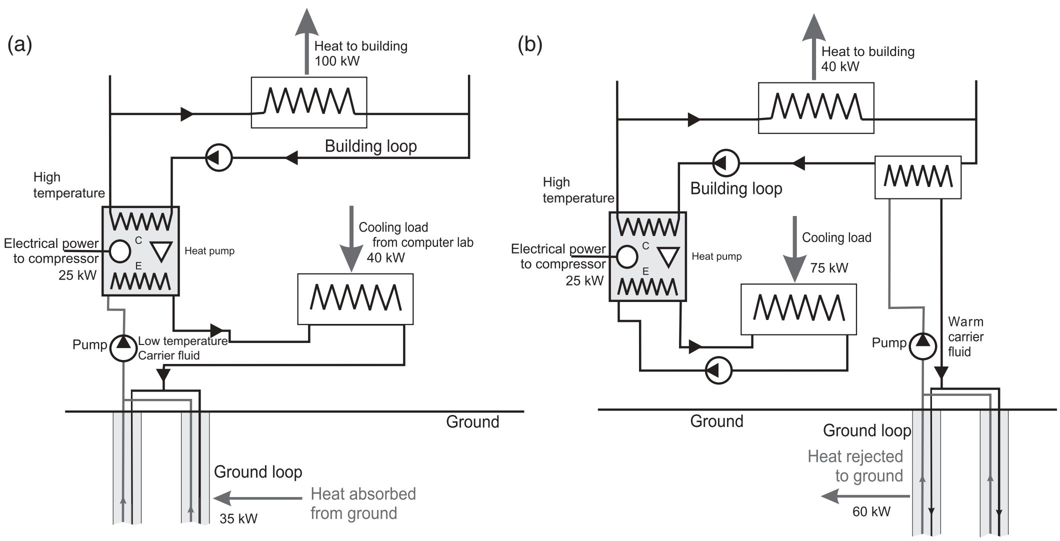


Figure 7.25 A schematic diagram of a centralised heat pump, providing simultaneous heating and cooling. (a) This figure shows the situation when heating dominates: heat is sourced from the building's cooling demand, with the surplus being extracted from the ground via closed-loop boreholes. (b) This figure shows the situation where cooling dominates: the heating circuit, coupled to the condenser, either passes directly through the ground, or is coupled to the ground via a heat exchanger, such that excess heat can be shed. Comparable open-loop groundwater-based alternatives would also be possible.

principle as that shown in Figure 7.24b for the production of hot water in a building with a net cooling demand.

7.6.4 *Distributed systems*

Another elegant solution to fluctuating heating and cooling demands in different zones of the same building is to install a so-called distributed system. Here, the ground loop does not simply enter a heat pump array in a centralised plant room. Rather, the ground loop actually becomes the building loop: it circulates around the building and feeds a network of small individual heat pumps in each zone or even each room. Each heat pump can be controlled either by thermostat or by the occupants of the room directly. On cold days, the individual heat pumps can be set to heating mode. They extract heat from the building/ground loop's fluid and deliver it to the room (the heat pump may be a small wall-mounted water-to-air console unit of the type shown in Figure 4.9c). On hot days, the heat pump extracts heat from the room and dumps it back into the building/ground loop. On some days, heat pumps in different parts of the building may be either dumping heat to or extracting heat from the building/

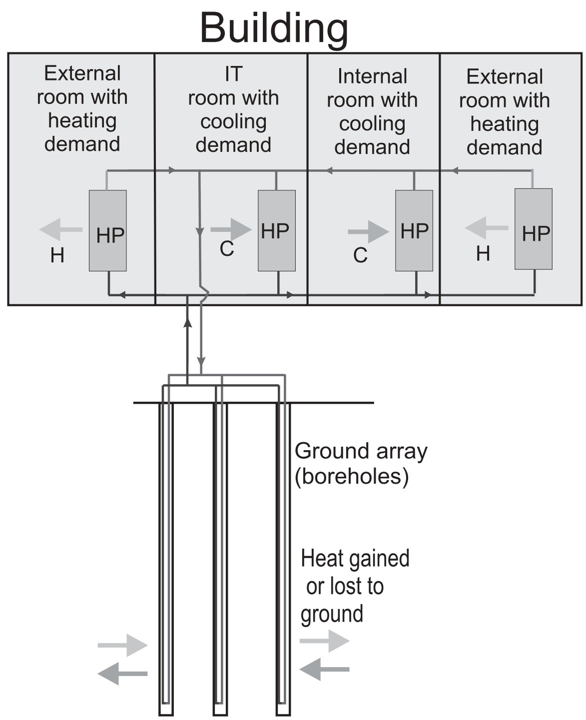


Figure 7.26 A schematic diagram of a distributed heat pump scheme, providing simultaneous heating and cooling. IT = information technology (lots of computers!); HP = heat pump; C = cooling effect; H = heating effect.

ground loop, according to demand. Only the net imbalance between heating and cooling requirements needs to be rejected to or absorbed from the ground (Figure 7.26).

We can use some basic equations to assess the amount of net heat transfer from such a loop to the ground:

$$\text{Net waste heat from building loop} \approx C\left(1+\frac{1}{\mathrm{COP}_C}\right)-H\left(1-\frac{1}{\mathrm{COP}_H}\right) \qquad (7.13)$$

where H and C are the heating and cooling loads delivered to the building, and COP_H and COP_C are the coefficients of performance of the heat pumps in heating and cooling mode, respectively. The electrical energy (E) required by the heat pumps is given by

$$E=\frac{C}{\mathrm{COP}_C}+\frac{H}{\mathrm{COP}_H} \qquad (7.14)$$

Note that both these equations neglect any energy consumption by circulation pumps and other inefficiencies in the system.

7.7 Heat from ice

Heat pumps can be utilised in extreme climatic conditions – IEA (2001b) documented a scheme in Norwegian Lappland, at Kautokeino, where 16 deep coaxial closed-loop boreholes, to depths of up to 145 m, support 290 kW of installed heat pump capacity to heat a health centre at a location where undisturbed shallow ground temperatures may be as low as 2°C.

But what about the truly extreme corners of the planet, where the ground is permanently frozen? Here, heat pumps can be used to deliver a solution to an unusual geological problem. Significant parts of northern Asia and America are underlain by permafrost; that is, frozen ground, where the average annual air temperature is so low that the upper tens or hundreds of metres of the geological column are permanently below 0°C. Pore space in rocks and sediments is saturated not with groundwater but with ice. Constructing viable dwellings in such terrain is not easy: they need to be heated, of course. But if a paraffin stove or electric heating system is installed in a house on permafrost, the heat generated tends to melt the permafrost beneath the house and to cause subsidence. One way around this is to have a very well insulated floor, such that heat exchange between the lounge and the ground is minimised. The inhabitants of Longyearbyen, on Svalbard, build their houses on pillars in order to avoid contact between the house and the ground.

An alternative solution, described from Yukon and eastern Siberia (Perl'shtein *et al.*, 2000; Environment Canada, 2002), is to install a GSHP with the closed loop beneath the building. Heat is thus extracted from the permafrost beneath the house, keeping the ground frozen and stable, and delivered as usable space-heating energy to the building. Bingo . . . heat from ice!

STUDY QUESTIONS

7.1 A water-based fluid (volumetric heat capacity 4.18 kJ L^{-1} K^{-1}) flows at a rate of 1.2 L s^{-1} across the warm side of an ideally efficient heat exchanger. Its temperature loss on passing the heat exchanger is 3.3 K. A carrier fluid, with a volumetric heat capacity of 3.65 kJ L^{-1} K^{-1} flows through the other side of the heat exchanger at a rate of 1.6 L s^{-1}. How much heat is transferred between the fluid and what is the temperature increase of the cold fluid (see Box 7.2)?

7.2 What is the vertical thermal conductance of a 50 m thickness of ground comprising 10 m of moist sand of thermal conductivity (1.3 W m^{-1} K^{-1}), 20 m of saturated sand of thermal conductivity (2.3 W m^{-1} K^{-1}) and 20 m of saturated clay of thermal conductivity (1.6 W m^{-1} K^{-1}) (see Box 7.1)? What is the equivalent composite vertical thermal conductivity of the sequence?

8

The Design of Groundwater-Based Open-Loop Systems

> Darcy's Law governs the motion of ground-water under natural conditions and . . . is analogous to the law of the flow of heat by conduction, hydraulic pressure being analogous to temperature, pressure gradient to thermal gradient, permeability to thermal conductivity, and specific yield to specific heat.
>
> Charles V. Theis (1935)

The establishment of an open-loop heating or cooling system using groundwater depends on the existence of an aquifer. An aquifer is a body or stratum of rock or sediment that has adequate hydraulic properties (hydraulic conductivity and storage) to permit the economic exploitation of groundwater. In our specific case, the aquifer must yield enough water to support the required heating or cooling load of our system.

We have already encountered open-loop systems in Chapter 7. Here we will consider them in greater detail. We will look briefly at the different types of aquifer that exist in nature. We will look at conceptual designs of wellfields in various aquifers and at means of estimating the likely yields of wells. Means of obtaining information about aquifer properties are briefly covered. Thereafter, the sustainability of the wellfield will be examined – how long will it take water to travel between an injection and an abstraction well? How does a heat signal migrate when it enters a groundwater flow field? What can we do to improve the sustainability and design life of open-loop systems?

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

Firstly, let us look at some of the common design flaws in open-loop groundwater-based systems.

8.1 Common design flaws of open-loop groundwater systems

- Lack of specialist design input from a hydrogeologist or groundwater engineer.
- Over-optimism regarding the hydraulic properties of aquifers. In particular, the common misconception is that, if one well yields $5\,L\,s^{-1}$, a wellfield of 10 wells will yield $50\,L\,s^{-1}$. This is not necessarily the case. If no re-injection occurs, the total yield will typically be significantly less than simple multiplication would suggest. On the other hand, if spent water is re-injected, it will serve to support the abstraction and it may be possible to abstract more water than would be the case with an 'abstraction-only' scheme.
- Lack of consideration given to disposal of thermally 'spent' (waste) water.
- Lack of appreciation that injection wells require specialist design, may underperform compared with abstraction wells, and will need careful operation and maintenance to ensure a long life.
- Lack of appreciation that water chemistry and microbiology may affect the long-term performance of the system (Bakema, 2001).
- Lack of management of pressure and dissolved gas conditions within a re-injection system.
- Lack of consideration of hydraulic breakthrough: that is, the possibility that warm (or cold) wastewater may flow back into the abstraction well, compromising the efficiency of the system (or even its long-term sustainability).

8.2 Aquifers, aquitards and fractures

Aquifers (from Latin: *water-bearing*) are strata or bodies of rock or sediment that yield economically useful quantities of groundwater. The critical hydraulic properties of an aquifer are the hydraulic conductivity (K) and the storage (S) of the material of which it is composed. We have already met hydraulic conductivity in Darcy's law (Chapter 1; Equation 1.2). Storage refers to the amount of water (in cubic metres) that a unit of aquifer material can release (or take up) in response to a 1-m decline (or increase) in head. Sediments such as sands and gravels, and rocks such as porous sandstones, are usually quite good aquifers, and are termed *porous medium* or *intergranular flow* aquifers. They have a high K (Table 8.1) and a respectable porosity, implying a relatively high value of S. In such aquifers, water flows through the pore spaces between the grains. The wider the necks between the pores, the higher will be the value of K.

Fine-grained sediments, such as silts and clays, may have a high porosity, but the pore spaces are only very small. As K exhibits a very strong dependence on the aperture of the pore spaces (or, strictly speaking, the necks connecting the pores), the

Table 8.1 A summary of the typical hydraulic properties of geological formations (based on Domenico and Schwartz, 1990; Allen *et al.*, 1997; Fetter, 2001; Olofsson, 2002; Misstear *et al.*, 2006).

	Porosity (n) %	Specific yield (S_Y) %	Hydraulic conductivity (K) m s^{-1}
Clay	30–60	1–10	10^{-12}–10^{-8}
Silt	35–50	5–30	10^{-9}–10^{-5}
Sands	25–50	10–30	10^{-7}–10^{-3}
Gravel	20–40	10–25	10^{-4}–10^{-1}
Sandstone	5–30	5–25	10^{-9}–10^{-4}
Most unweathered crystalline	<1	<1	10^{-13}–10^{-5}
Silicate rocks (granites, Schists, gneisses)	0.1[b] <0.05[a]		Depending on degree of fracturing
Basalt	<1–50	<1–30	10^{-13}–10^{-2}
British Chalk	10–45	0.5–5	10^{-10}–10^{-6}[c] (median 7×10^{-9})[c]

[a] Olofsson (2002) cited an effective (kinematic) porosity of crystalline rock aquifers of <0.05%.

[b] Domenico and Schwartz (1990) cited values as low as 0.1% for porosity and 0.0005% for effective porosity in granite.

[c] These figures refer to Chalk matrix hydraulic conductivities. Bulk transmissivities of the Chalk aquifer are typically in the range of 10–10 000 m^2 day^{-1} (10^{-4}–10^{-1} m^2 s^{-1}), which provides some impression of the importance of fracture flow in the Chalk (Allen *et al.*, 1997).

hydraulic conductivities of such sediments are low: wells yield poorly and the sediment is referred to as an *aquitard* (from Latin: *water* + *slow/late*).

Some rocks – in particular, crystalline rocks, such as granites, slates or gneisses – consist of tightly interlocked silicate crystals, with practically no intergranular pore space. One might think that such rocks would be wholly impermeable, but in fact, wells drilled in such aquifers commonly yield several hundred litres of water per hour, and sometimes several thousands per hour. Such rocks have been subject to the imposition and release of enormous stresses throughout geological time, which have caused them to fracture and crack. The resulting fractures and joints are able to transmit groundwater. If a well intersects such interconnected fractures, it will yield a supply of groundwater. Such aquifers are termed *fracture-flow* aquifers.

Finally, some types of aquifer, such as limestone and dolomite, typically have a rather low intergranular permeability. Limestones often consist of interlocking carbonate crystals with minimal intergranular pore space, or tiny fossilised organisms (Chalk, e.g., is made of tiny *coccolith* microfossils) providing a large overall porosity but with very small pore apertures (Table 8.1). In these lithologies, too, most groundwater flow takes place via fractures and joints. However, unlike silicate rocks, the calcite or dolomite minerals constituting these lithologies are rather soluble in groundwater. Thus, over a period of thousands of years, the fractures become wider and more permeable. They can reach several millimetres or centimetres in aperture and transmit huge amounts of groundwater flow (Figure 8.1). In extreme cases, cave systems result, and the flow regime is typically referred to as karstic. Such fissure and cave development was particularly intense in periglacial conditions towards the end

Figure 8.1 A photograph taken in a well, showing groundwater cascading out of a solution cavity (fissure) in a pumped borehole in the Chalk aquifer of southern England. The solution pipe is probably around 4 cm across. *Public domain information provided by and reproduced with the permission of the Environment Agency of England and Wales (Thames Region).*

of the last ice age, as CO_2 is more soluble in water at cold temperatures, and it is this carbon dioxide that is responsible for carbonate dissolution:

$$CaCO_3 + CO_2 + H_2O = Ca^{2+} + 2HCO_3^- \tag{8.1}$$

$$\text{Limestone} + \text{Carbon dioxide} + \text{Water} = \text{Dissolved calcium and bicarbonate}$$

Aquifers such as limestones or Chalk, where groundwater flow has enhanced hydraulic conductivity by dissolution of fractures, are often referred to as *karstified* aquifers (Banks *et al.*, 1995; Waters and Banks, 1997), much to the disgust of many speleologists who wish to retain this term to describe very specific landforms and geomorphologies!

8.3 Transmissivity

We need one further term to complete our description of the hydraulic properties of aquifers. By considering Darcy's law (Equation 1.2), it should be possible to see that an aquifer comprising a layer of sand 5 m thick will be able to transmit only half as much groundwater (per unit width) as a similar layer 10 m thick, assuming the same head gradient in both cases. Whereas hydraulic conductivity (K) is an intrinsic property of the material comprising the aquifer, the transmissivity (T) is an extrinsic property. Transmissivity describes the ability of the aquifer, as a geological unit, to transmit groundwater flow; it is defined as the product of the hydraulic conductivity (K) and the thickness (D) of the aquifer:

$$T = K \cdot D \tag{8.2}$$

Its units are $m^2\ day^{-1}$ or $m^2\,s^{-1}$. Consider Darcy's law (Equation 1.2):

$$Z = -K \cdot A \cdot \frac{dh}{dx} \tag{8.3}$$

where Z = flow of groundwater ($m^3\,s^{-1}$), $A = Dw$ = cross-sectional area of the block of material under consideration (m^2), w = width of aquifer under consideration (m), h = head (m), x = distance coordinate in the direction of decreasing head (m) and dh/dx = head gradient (dimensionless).

By substituting our expression for transmissivity, we obtain

$$Z = -T \cdot w \cdot \frac{dh}{dx} \tag{8.4}$$

As a point of interest, we can also define the transmissivity of a fracture T_f (i.e. its ability to transmit water under a given head gradient). This value depends, of course, not only on the aperture of the fracture, but also on its roughness and its tortuosity. However, for an ideal fracture: smooth-sided, plane-parallel and of constant aperture b_a (Snow, 1969; Walsh, 1981; Misstear *et al.*, 2006):

$$T_f = \frac{\rho_w g b_a^3}{12\mu_w} \approx 629\,000 b_a^3 \tag{8.5}$$

where T_f is in $m^2\,s^{-1}$ and b_a is in metres, ρ_w is the density of water ($\approx 1000\,kg\,m^{-3}$), g is the acceleration due to gravity ($9.81\,m\,s^{-2}$) and μ_w is the dynamic viscosity of water ($\sim 0.0013\,kg\,s^{-1}\,m^{-1}$). Hence, we can appreciate the huge dependence of fracture transmissivity on aperture.

8.4 Confined and unconfined aquifers

A confined aquifer is a transmissive stratum of sediment or rock that is overlain by a low-permeability aquitard and where the groundwater head is higher than the top of the aquifer. An example would be a layer of sand sandwiched between two layers of clay (Figures 2.5 and 8.2). The groundwater wholly saturates the aquifer and is, in a sense, under excess pressure. Thus, when a borehole is drilled into the aquifer, water

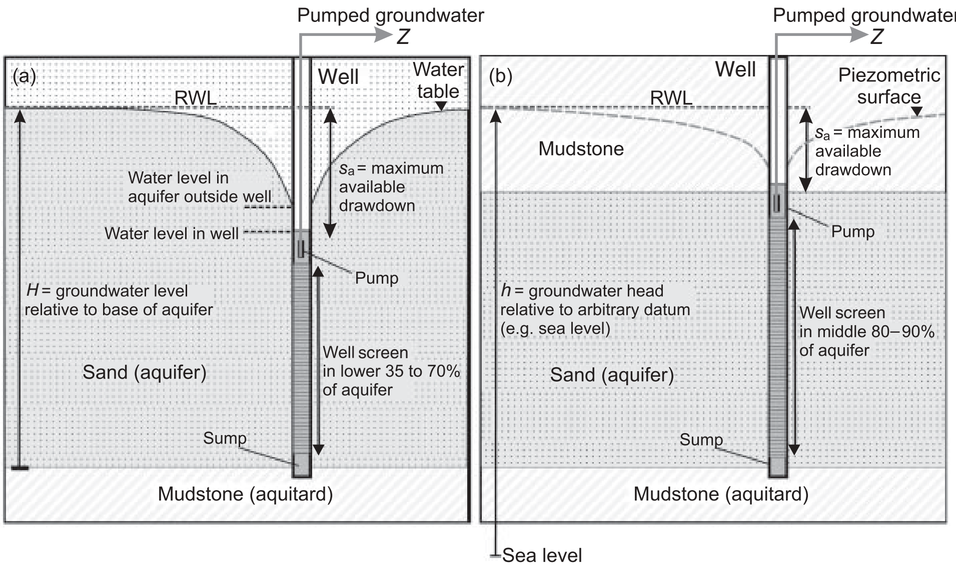


Figure 8.2 Schematic diagrams of (a) an unconfined and (b) a confined aquifer, illustrating typical placements of well screens. s_a = the available drawdown; RWL = rest water level.

rises up the borehole to a level above the top of the aquifer – a level corresponding to the groundwater head in the aquifer. We can imagine a surface, called the piezometric surface, describing the locus of the head (h) at any point in the aquifer. The gradient of this surface is the hydraulic gradient or head gradient. This, of course, controls the direction and rate at which groundwater flows (down the hydraulic gradient), according to Darcy's law.

If the piezometric surface is above ground level, a borehole drilled into the aquifer will be artesian – the water will overflow under its own pressure at ground level (Figures 2.5 and 16.4).

An unconfined aquifer is not held under pressure by a low-permeability cap (Figures 3.2 and 8.2). Its upper boundary is a free water surface – the water table. Rainfall (or other recharge) can usually enter from the top of the aquifer. The unsaturated portion of strata is called the vadose zone, while the saturated portion of the aquifer is the phreatic zone. The transmissivity of the unconfined aquifer is given by Equation 8.2, but be careful! The thickness of the aquifer (D) can change as the water table rises and falls with the seasons. It can also change if we start abstracting water from the aquifer and drawing down the water table. Thus, whereas the transmissivity of a confined aquifer is usually constant, the transmissivity of an unconfined aquifer can vary seasonally and with abstraction. Especially in an unconfined aquifer such as the Chalk (Owen and Robinson, 1978), many of the most transmissive fissures are situated in the shallow part of the aquifer, around the water table. Thus, as the water table declines, these transmissive fissures become dewatered, and the transmissivity of the aquifer can drop dramatically.

Finally, the storage coefficient of an unconfined aquifer is termed the specific yield (S_Y). If the water table falls by 1 m, a quantity of water will be released from storage that will be related to the porosity (n), but will be somewhat less, as we can never fully drain a geological material by gravity. There will always be some small amount of water that is retained – adhering to mineral grains or tucked away in 'blind' pore spaces. If we consider 1 m^2 of aquifer, we can say that

$$S_Y < n \tag{8.6}$$

The units of S_Y are cubic metre of water per square metre aquifer area per metre decline in head – that is, dimensionless. In granular porous media, S_Y is often a few percent (i.e. >1% or >0.01) and may exceed 10% (0.1 as a fraction). In fissured or fractured media, porosity (and hence S_Y) is very low. Indeed, porosities of <1% (and usually <0.1%) are likely in fresh crystalline silicate rocks (Table 8.1).

In confined aquifers, the amount of water released from every cubic metre of aquifer as head declines by 1 m is termed the specific storage (S_S). Here, the aquifer is not actually dewatered as the head drops; water is merely released because of the very small elastic responses in the water and the aquifer matrix. S_S in confined aquifers is thus very small (of the order 10^{-5} to 10^{-6} m^{-1}). Its units are cubic metre water per cubic metre aquifer per metre decline in head, or m^{-1}. We can integrate S_S over the entire confined aquifer thickness to get an overall aquifer storage coefficient or storativity (S) – this is the amount of water in cubic metre released per square metre area of aquifer per metre decline in head (i.e. dimensionless). Whereas S_S does not depend heavily on aquifer lithology, S is dependent on aquifer thickness and is given by

$$S = S_S \cdot D \tag{8.7}$$

8.5 Abstraction well design in confined and unconfined aquifers

In this brief section, we will consider some very basic elements of design of groundwater abstraction wells. For more detailed information, see Misstear *et al*. (2006). Re-injection wells can be similar to abstraction wells, but are often a rather different kettle of fish – see Section 8.9.3.

8.5.1 Confined aquifers

In a confined aquifer comprising poorly lithified porous materials (sands, gravels, poorly cemented sandstone), it is wise to screen[1] as much of our aquifer as possible to

[1] Remember, a well *screen* is a kind of cylindrical sieve, designed to keep sediment out but to let water in.

maximise the water yield from an abstraction well (Figure 8.2). Indeed, Driscoll (1986) suggested screening around 90% of the aquifer thickness, leaving small amounts of blank casing overlapping at the top and bottom of the aquifer to ensure that no fine particles from the overlying and underlying clayey aquitards migrate through the pore spaces of the aquifer to enter the well. Of course, if our required yield is small and our aquifer is very transmissive and relatively thick (>100 m, say), we may not choose to drill to the base of the aquifer, but to place a well screen in the upper portion of the aquifer only. This will reduce the hydraulic efficiency of the well, but this may be a price worth paying to save on the capital cost of a deep well to the base of the aquifer.

If the confined aquifer is well lithified (e.g. a lithified limestone, a well-cemented sandstone, a granite; Figure 7.6a), the well may be completed 'open hole' (no well screen) within the aquifer itself. A section of blank casing would usually be installed in the upper part of the well (corresponding to the confining aquitard).

When drilling into a confined aquifer, especially if there is a suspicion that it may be artesian, and if you are not using a dense drilling mud that can counteract the expected artesian head, it is good practice to securely grout a string of casing into the confining stratum (see Section 16.3) before the aquifer is encountered. Drilling continues through the grout plug in the base of the cased section of borehole at a narrower diameter. This practice ensures that there is no hydraulic communication between the surface (or any overlying aquifer) and the target confined aquifer. It also allows any artesian overflow to be controlled. If a strong artesian overflow is encountered and a string of casing has not been securely grouted into the overlying aquitard, it can be very difficult to bring the overflow under control (Figure 16.4).

When pumping a well in a confined aquifer, it is generally recommended as good practice that the confined aquifer is not dewatered. In other words, the pumping water level in the abstraction well should remain above the top of the aquifer or top of the well screen. If we start to dewater the aquifer, we introduce air into a previously anaerobic environment. This can bring about all sorts of unwanted chemical reactions, such as the oxidation of sulphide minerals, oxidation and precipitation of iron and stimulation of bacterial growth. We can thus say that the available drawdown (s_a) is the difference between the undisturbed piezometric surface and the top of the aquifer (or top of the pump, whichever is higher). In all fairness, it should be stated that this rule is often 'bent' by practitioners, especially in the case of 'open' boreholes drilled into lithified confined aquifers. We break it at our own risk, however, in the awareness that doing so may shorten the life of our well and/or impair its performance.

8.5.2 Unconfined aquifers

In an unconfined aquifer, we will draw down the water table as a result of a groundwater abstraction (Figure 8.2). It is a general guideline that this drawdown will seldom exceed 50% of the aquifer thickness, but may be in the range of 30–65% of aquifer thickness.

In designing a well screen for an abstraction well in an unconsolidated aquifer comprising poorly lithified, relatively homogeneous sediments or rocks, there are two schools of thought. Some will recommend placing a well screen in the bottom 35–70% of the aquifer, and operating the well such that the water level never falls below the top of the well screen. This means that the well screen always remains under water and is not exposed to oxygen that can promote corrosion, bacterial fouling and chemical incrustation. In this case, the available drawdown is the difference between the rest water table (in its undisturbed state) and the top of the well screen. Others will simply install a well screen in the entire saturated section of the aquifer and accept that pumping will cause the uppermost section of the screen to become alternately dewatered and re-saturated. They will accept that the performance of this upper screen section may deteriorate over time. The former philosophy makes sense if the cost of well screen is much greater than that of plain casing; the latter pragmatic philosophy makes sense if the costs are not too dissimilar.

If the aquifer is not homogeneous, but contains specific, high-transmissivity flow horizons, the zones that are installed with well screen will be selected to coincide with those horizons. Furthermore, as in the case of the confined aquifer (Section 8.5.1), if our required yield is small and the aquifer is very transmissive (or if the aquifer is very thick), we may choose not to drill to the aquifer's base.

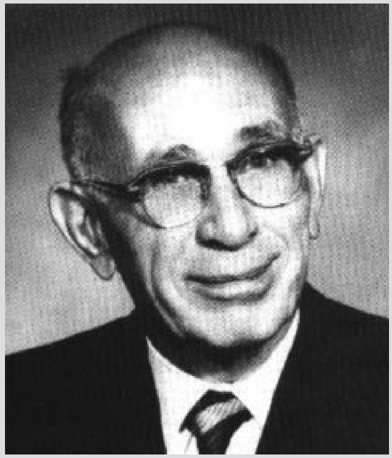
In a lithified, well-cemented or crystalline aquifer, we will typically construct our well 'open hole'. A section of blank casing would usually be installed and grouted into the upper part of the well to exclude any superficial materials, badly weathered, unstable strata and potentially contaminated shallow groundwater. In the particular example of limestones, such as the Chalk, where transmissivity is concentrated at specific horizons (in the few metres or tens of metres below the water table in the Chalk), we will often aim not to pump at such a high rate that we wholly dewater these especially transmissive horizons. This consideration may restrict still further our available drawdown.

8.6 Design yield, depth and drawdown

Using the formulae discussed in Section 6.12, Chapter 7 and Equations 4.7–4.8 and 7.6–7.9, we will usually have defined a groundwater yield that we require to satisfy a given load of ground source heating or cooling. When we pump a well in a confined aquifer (Figure 8.2), we draw down the piezometric surface around the abstraction well, creating a radial flow field (and hydraulic gradient) towards the pumping well. The area where the groundwater levels are drawn down is called the *cone of depression*, and the vertical difference between the original piezometric surface and the new piezometric surface as a result of pumping is termed the *drawdown.*

The American hydrogeologist Charles V. Theis (one of the key figures of modern hydrogeology) deduced in 1935 a formula for the transient (time-dependent) radial flow of groundwater towards a pumped well (Box 8.1). The formula provides the relationship between aquifer transmissivity (T), groundwater abstraction rate (Z) from the

BOX 8.1 Charles V. Theis



C.V. Theis. From public domain USGS document by White and Clebsch (1994).

The American hydrogeologist Charles Vernon Theis was born in Kentucky in March 1900. He studied civil engineering at the University of Cincinnati in 1917 and subsequently gained a post in the University's Geology Department. Later, in 1927, Theis became a junior geologist at the United States Geological Survey (USGS), based in Moab, Utah. In 1929, he completed his doctoral degree (in geology) from the University of Cincinnati. He joined the Ground Water Division of the USGS in 1930 and stayed there for the rest of his career. Much of his early work was based on the assessment of groundwater abstractions for irrigation in the arid region of New Mexico. He was astute enough to realise that existing equations (such as that of Thiem) were not wholly adequate for describing the impact of an abstraction on regional groundwater levels. Indeed, he realised he needed a good, time-variant solution to the fundamental groundwater flow equations (Darcy's law and the conservation of mass). It would seem that Theis's maths was not up to the job of solving this puzzle. Fortunately, he was not too proud to ask for advice, and he sought it from his old University friend, Clarence Lubin. Lubin steered Theis in the direction of some earlier work by the physicist H.S. Carslaw (published in the 1921 book *Introduction to the Mathematical Theory of the Conduction of Heat in Solids*). Lubin and Theis figured that a non-equilibrium solution to groundwater flow to a well would probably be analogous to heat conduction through a solid towards a thermal sink. Theis published the non-equilibrium groundwater solution in his 1935 paper (of which Lubin was reputedly offered co-authorship, but modestly refused). In this paper, Theis explicitly drew attention to the analogy between heat conduction and groundwater flow (see Table 1.1).

Theis enjoyed many further years with the USGS, and later worked on issues as diverse as artificial recharge, aquifer inhomogeneity and anisotropy, and radioactive waste disposal (White and Clebsch, 1994).

well, dimensionless storage (S) and drawdown (s) at a given radial distance (r) from the pumping well at any given time (t) after pumping started:

$$s = \frac{Z}{4\pi T} W(u) \tag{8.8}$$

where $W(u)$ is a function known as the Theis well function (or *exponential integral* function):

$$W(u) = -0.5772 - \ln u + u - \frac{u^2}{2.2!} + \frac{u^3}{3.3!} - \frac{u^4}{4.4!} + \frac{u^5}{5.5!} - \cdots \tag{8.9}$$

and

$$u = \frac{r^2 S}{4Tt} \tag{8.10}$$

The polynomial term $W(u)$ converges quickly and can usually be calculated by considering the first, say, five power terms (after checking that adequate convergence has occurred), allowing manual calculation of drawdown for a given r, T, S and t. We can already see from this, admittedly complex, set of formulae that

- well yield (Z) is approximately proportional to transmissivity (T);
- drawdown (s) is approximately proportional to abstraction rate (Z);
- drawdown is inversely proportional to transmissivity;
- drawdown increases with time of pumping;
- drawdown decreases with distance from abstraction well.

Cooper and Jacob (1946) realised that Theis's equation could be dramatically simplified if we assume that u is small (less than 0.01 or, some would say, less than 0.05). In this case, all the higher terms of the polynomial expansion become negligibly small and we are left with

$$s = \frac{Z}{4\pi T}\left[-0.5772 - \ln\left(\frac{r^2 S}{4Tt}\right)\right] = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25Tt}{r^2 S}\right) \tag{8.11}$$

or

$$s = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25T}{r^2 S}\right) + \frac{2.30Z}{4\pi T}\log_{10} t \tag{8.12}$$

Here we see that

- when u is small (r is small or t is large), drawdown increases in proportion to $\log_{10}(t)$;
- when u is small, drawdown decreases in proportion to $\log_{10}(r)$.

In the hypothetical, infinite, homogeneous aquifer considered by Theis, the cone of drawdown continues to expand forever. In a real aquifer, it continues to expand (Theis, 1940; Bredehoeft *et al.*, 1982) until it has

- induced sufficient recharge (i.e. induced vertical head gradients causing water to flow from wetlands, streams or lakes into the aquifer) or
- captured sufficient groundwater discharge (i.e. decreased the amount of spring flow overflowing as excess water from the aquifer)

to balance the abstracted quantity, at which point the aquifer stabilises to a steady-state condition.

Many years before Theis, Adolph Thiem (1887) had found a steady-state solution for radial groundwater flow towards an abstraction well. The following differential equation (derived from Darcy's law in radial coordinates) describes this situation:

$$Z = -2\pi rT\frac{dh}{dr} \tag{8.13}$$

where h = groundwater head and r = radial distance from the abstraction well. We can integrate between two radial distances – r_1 and r_2:

$$h_2 - h_1 = s_1 - s_2 = \frac{Z}{2\pi T}\ln\frac{r_2}{r_1} \tag{8.14}$$

where h_1 and h_2 are the groundwater heads at these radii, and s_1 and s_2 are the drawdowns. If we consider our groundwater abstraction well as one of these two points, we can say that the expected drawdown in this well (for our yield Z) is s_w, where r_w is the radius of the abstraction well:

$$s_w - s_2 = \frac{Z}{2\pi T}\ln\frac{r_2}{r_w} \tag{8.15}$$

Logan (1964) devised the most famous approximation in hydrogeology when he dared to set r_2 to a certain distance r_e from the abstraction well, where the drawdown s_2 is effectively zero. Moreover, he said that (r_e/r_w) is often approximately equal to 2000. Thus, he obtained the Logan approximation relating abstraction well yield (Z) to drawdown (s_w) in a pumped well:

$$T = \frac{Z}{2\pi s_w}\ln\frac{r_e}{r_w} = 1.22\frac{Z}{s_w} \tag{8.16}$$

We should be rightly suspicious of the chain of dodgy assumptions leading to the Logan approximation: Adolph Thiem assumed a steady state that Charles Theis demonstrated would never be reached in an ideal aquifer; Logan plucked a figure of 2000 from thin air. Yet, in fact, the Logan formula can prove a useful 'back of the envelope' first estimate of the relationship between transmissivity and well yield.

What does this mean for well design? If we know the transmissivity of the aquifer, and we have defined a maximum drawdown that is allowable in our well (e.g. to the top of the well screen), we can estimate the maximum yield we can expect from the well. Conversely, if our aquifer is thick and quite transmissive, we may wish to estimate how deep (D) we need to drill into our aquifer to achieve a given yield (Z). As transmissivity is the product of hydraulic conductivity (K) and effective aquifer thick-

BOX 8.2 Assessing the Maximum Yield of a Drilled Well

We can use the Theis (Equation 8.8) or Cooper–Jacob (Equation 8.11) approaches to predict the drawdown (s) for any given yield (Z) at a given time (t) after pumping starts, provided we know the aquifer transmissivity (T) and storage (S). But what value of t should we use? Let us consider a proposed well, where the top of the well screen is at 27 m below ground level (bgl) and the pump is just below 25 m bgl. The natural groundwater level is 15 m bgl. Thus, we have an available drawdown of around 10 m. Let us assume that our aquifer has a transmissivity of 500 m^2 day^{-1} and a storage coefficient of 0.08.

If our well, of radius 0.1 m, is going to be pumping at a rate of 10 L s^{-1} (= 864 m^3 day^{-1}) for 8 h day^{-1} throughout a 5-month winter heating season, it would probably be sensible to solve the Theis or Cooper–Jacob equation twice. Firstly, we might set Z equal to the average winter abstraction rate of 3.3 L s^{-1} = 288 m^3 day^{-1} and use a value of t = 150 days. We apply Equation 8.11:

$$s_w = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25Tt}{r_w^2 S}\right) = \frac{2.30\times 288}{4\pi\times 500}\log_{10}\left(\frac{2.25\times 500\times 150}{0.1^2\times 0.08}\right) = 0.9\text{ m}$$

Secondly, we would set Z = 864 m^3 day^{-1} and t = 8 h = 0.33 days in order to simulate the peak abstraction condition on a single day:

$$s_w = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25Tt}{r_w^2 S}\right) = \frac{2.30\times 864}{4\pi\times 500}\log_{10}\left(\frac{2.25\times 500\times 0.33}{0.1^2\times 0.08}\right) = 1.8\text{ m}$$

Thus, using the average abstraction rate over an entire winter season, we would not expect the drawdown to exceed 0.9 m. Even if we superimpose a pulse of peak abstraction onto this long-term drawdown, we would not expect drawdown to exceed 3 m in total. This is well within our available drawdown of 10 m, even allowing for hydraulic inefficiency.

If we apply the Logan approximation (Equation 8.16) to the problem, we obtain 2.1 m drawdown for the peak condition and 0.7 m for the long-term average drawdown. Not bad for a dodgy approximation!

ness, and assuming that our well will be hydraulically efficient (see Section 8.7.1), we could say that

$$KD = 1.22\frac{Z}{s_w}, \text{ thus } D = 1.22\frac{Z}{Ks_w} \tag{8.17}$$

In this case, our well of reduced depth would not perform as hydraulically efficiently as a well through the entire aquifer thickness – the drawdown (and hence the pumping costs) for a given yield would be greater. We are thus sacrificing running costs (electricity or fuel for the pump) for a lower capital cost (shallower well).

8.6.1 A more sophisticated approach

If the Logan approximation seems a little bit too primitive (and it should really only be used for an initial estimate of well yield), we can always utilise the Theis or Cooper–Jacob equations, if we can provide estimates of transmissivity and storage. To predict the drawdown for a given yield in our abstraction well, we would set r to the radius of abstraction well (r_w) and then solve the equation for a value of pumping duration (t). A worked example is given in Box 8.2.

8.7 Real wells and real aquifers

All of the above equations presuppose that our abstraction well is 100% hydraulically efficient and that our aquifer is 'ideal' – that is, confined, infinite, homogeneous, isotropic and generally well-behaved in the way that real geology is not.

8.7.1 Real wells

No well is 100% hydraulically efficient. In a real well, turbulent flow, hydraulic resistance caused by the well screen/filter pack and several other factors all contribute to additional losses in head called well losses (Figure 8.3). These generally increase as abstraction rate (Z) increases, but in a non-linear way. While the Logan approximation (Equation 8.16) predicts a simple proportionality between drawdown and yield, a more realistic equation might be

$$s_w = BZ + CZ^n = \left(\frac{1.22}{T} + B'\right)Z + CZ^n \tag{8.18}$$

where B represents linear head losses. These are dominantly natural head losses in the aquifer due to the hydraulic resistivity of the aquifer, but they may include a minor component of linear well losses, B'. C represents non-linear losses (typically well losses). n is a power law coefficient, often taken to be around 2 (Bierschenk, 1963; Hantush, 1964).

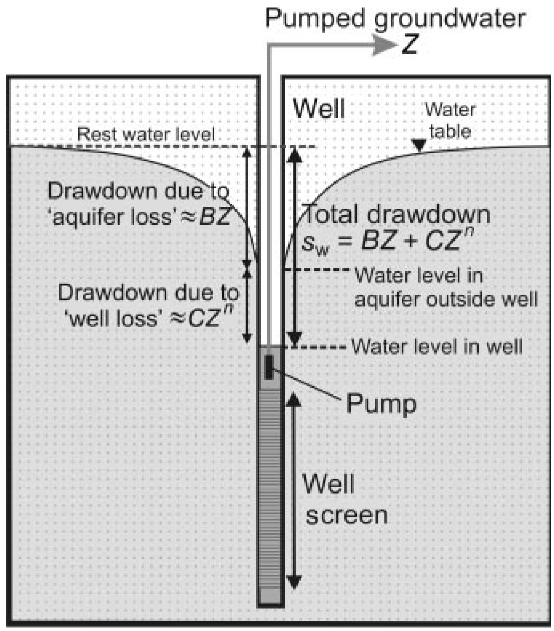


Figure 8.3 A real abstraction well, showing components of aquifer loss and well loss.

We have made so many assumptions already, what harm can another do? In fact, to make a first estimate of expected well yield, it is common practice (Misstear *et al*., 2006) to use a modified version of Logan's approximation, with the coefficient 1.22 increased (rather arbitrarily) to 2, to take into account possible well losses:

$$Z = \frac{Ts_{\mathrm{w}}}{2} \tag{8.19}$$

Remember, Logan's approach is an approximation – use it only to gain a first estimate of well yield and drawdown, not for detailed design.

8.7.2 Unconfined aquifers

The equations derived hitherto have assumed that we are dealing with a confined aquifer. In fact, we can usually use them with a good conscience in unconfined aquifers, too, provided the drawdown is small in comparison with the total aquifer thickness. If this is not the case, the transmissivity of the aquifer may reduce significantly due to the decrease in its saturated thickness caused by drawdown. We thus run the risk of underestimating our expected drawdown or overestimating our yield. Modified versions of some of the standard equations may exist for unconfined aquifers. For example, the analogue to the Thiem equation (Equation 8.14) for an unconfined aquifer is as follows:

$$H_2^2 - H_1^2 = \frac{Z}{\pi K} \ln\left(\frac{r_2}{r_1}\right) \tag{8.20}$$

where H_1 and H_2 are the water table elevations at radii r_1 and r_2 relative to the base of the aquifer (and *not* to some arbitrary datum as is normally the case when we are considering groundwater head – Figure 8.2).

8.8 Sources of information

The formulae we have looked at are very fine, but they presuppose some knowledge of the aquifer properties. The best way of making a prognosis of a new well's likely yield is to examine the performance of any nearby wells in the same aquifer. Most countries will have a database of wells and boreholes: the host organisation may be the Ministry of Water, the Geological Survey or the Environmental Authority. In the United Kingdom, the body charged with maintaining the database is the British Geological Survey (BGS). It is usually possible, on request or via the Internet, to obtain copies of drilling logs and pumping tests from boreholes and wells within, say, a 1-km radius of our proposed drilling location.

If there are no existing wells in the immediate vicinity upon which to base our prognosis, we may be able to deduce aquifer properties from more distant sources of information. In many aquifers, especially sedimentary rock aquifers, properties will often vary gradually in space, such that maps can be made showing how properties such as transmissivity vary. These spatial variations may be related to systematic changes in grain size or sedimentological facies, or they may be more related to structural geological factors or even to recent weathering or dissolution history (the latter being relevant for limestone aquifers). In the United Kingdom, the Environment Agency and BGS (Allen *et al*., 1997; Jones *et al*., 2000) have published comprehensive summaries of data for the major and minor British aquifers. These publications provide an invaluable starting point for the prospective well owner: they contain maps of hydraulic properties, statistical distributions of well and aquifer properties and details of pumping tests. If we can estimate transmissivity from such maps and data sources, we can start to make prognoses for the yield from our proposed well from the equations discussed above.

If the available published information is inadequate and/or if we are planning a very expensive wellfield comprising a number of wells, we may wish to drill a pilot borehole, upon which we can carry out a pumping test. This pilot borehole will provide us with site-specific information on the aquifer and, if successful, can be incorporated into the final ground source heating or cooling scheme.

8.8.1 Pumping tests

Whole books have been written on the subject of pumping tests, of which the definitive volume is by Kruseman *et al*. (1990). A concise summary of test pumping is given in many hydrogeological textbooks, such as Misstear *et al*. (2006). In a pumping test, we typically pump our well at a constant rate (Z), and measure the drawdown (s) as it evolves with time (t), either in the abstraction well itself or in some observation borehole at a distance (r) from the abstraction well. With these data, we can perform an inverse solution of one of the equations described above to derive values for transmissivity (T) and storage (S), for example:

- the Theis equation – Equations 8.8–8.10;
- the Cooper–Jacob approximation – Equation 8.12;
- the Hantush (1964) and Bierschenk (1963) method – Equation 8.18, assuming $n = 2$;
- the Logan approximation – Equations 8.16 or 8.19;
- or some more complex variant of the Theis equation, taking into account deviations from Theis's ideal confined aquifer.

A typical pumping test on a well will normally comprise several phases (Misstear *et al*., 2006):

1. Step testing: where the well is pumped at a low rate (Z_1) for a short period (typically 2 h) and the drawdown (s_{w1}) measured in the abstraction well. After this first step, the groundwater is allowed to recover to the rest condition and a second step is commenced at a higher rate (Z_2). The new drawdown after 2 h for step 2 (s_{w2}) is measured. This process is typically repeated for four 'steps', the rates of pumping bracketing our design yield (Z_d), such that rates Z_1, Z_2, Z_3 and Z_4 will typically be equal to $0.33 \times Z_d$, $0.67 \times Z_d$, Z_d and $1.33 \times Z_d$. Alternatively, we can perform 'continuous' step testing, where the pumping rate is simply increased to the next rate at the end of the previous step, with no period of recovery. This cuts down the time required for step testing, but the results are slightly more difficult to interpret. In either case, we end up with four well yields $Z_1 \ldots Z_4$, with corresponding drawdowns $S_{w1} \ldots S_{w4}$, enabling us to construct a curve of Z versus s_w (Figure 8.4) and allowing us to predict the 2-h drawdown for any yield. Note that the shape of the curve is often convex upwards, implying that for ever higher pumping rates, we develop disproportionately high drawdowns. This implies that well efficiency decreases as pumping rate increases: this non-linear relationship is predicted by Equation 8.18. Hantush (1964) and Bierschenk (1963) proposed a simple and elegant method for estimating well efficiency from such data. Remember that, in thin unconfined aquifers, a non-linear, convex-upwards curve may also be due to the declining transmissivity of the aquifer as it is dewatered due to increasing pumping rate.
2. Constant rate testing: here, the borehole is pumped at a constant rate (usually Z_d) for a period of typically between 24 and 72 h. The evolution of drawdown is monitored either in the abstraction well or an observation borehole (or both). From this data set, we are able (through inverse solution of the Theis or Cooper–Jacob equations) to derive a value of transmissivity. We can also mathematically derive a

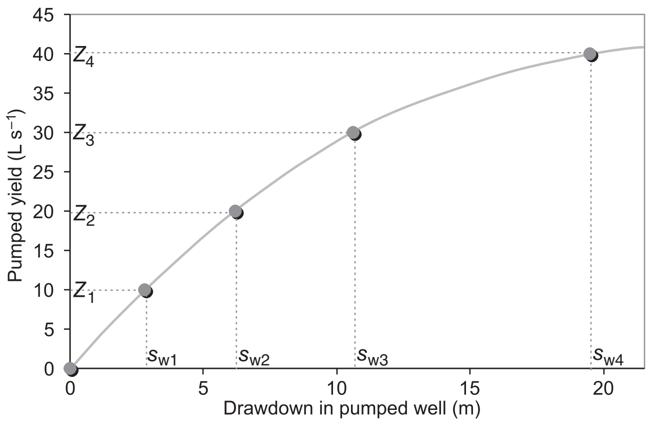


Figure 8.4 Plot of yield versus drawdown for step testing. For any yield Z, we can predict a corresponding drawdown (s_w) after 2 h of pumping.

value of storage coefficient (S), although this is seldom reliable if the drawdown data are only measured in the abstraction well rather than in a dedicated observation borehole.

3. We may then carry out an 'environmental' or 'sustainability' test. This is typically a longer-term test, running over periods of several days to several months, depending on the size and importance of the wellfield scheme. If the heating/cooling scheme is designed as a doublet (with abstraction and re-injection wells), we will probably choose to re-inject the abstracted water during this phase of testing, to simulate operational conditions. During this test, one will ensure that the design yield can be sustained over longer periods, and that there are no adverse impacts of drawdown caused by the abstraction (or of groundwater mounding caused by re-injection) on other users of the aquifer, or on environmental features that are supported by groundwater. Such environmental features may include groundwater-fed rivers and streams, lakes, wetlands and springs. In the context of ground source heating and cooling schemes, we may also use this period of testing to ensure that thermal breakthrough of a heat signature is not occurring from our re-injection point to our abstraction well (i.e. we monitor temperatures). We could also monitor temperatures in any nearby rivers/streams, adjacent wells or observation boreholes to ensure that we are not causing any unacceptable heat pollution to these features. In addition, we might also monitor the ground surface around any buildings or structures for signs of ground movement related to groundwater abstraction or re-injection.

Throughout the programme of test pumping, we would typically also take samples of groundwater for physical, chemical and microbiological analysis. The following typical set of analytical parameters would be considered a minimum for hydrogeochemical interpretation and for assessing the potential that the groundwater has for corrosion, incrustation and biofouling:

To be determined in the field, using portable meters or kits
pH, temperature, electrical conductivity, dissolved oxygen (or redox potential, Eh), alkalinity, turbidity, H_2S (possibly also cold acidity or dissolved CO_2).

To be determined in the laboratory
Major cations: calcium, magnesium, potassium, sodium, ammonium.
Major anions: chloride, sulphate, nitrate, alkalinity (bicarbonate).
Other metals: iron, manganese, aluminium (these three should be sampled and analysed as both total and dissolved metals), barium.
Microbiology: faecal coliforms, total heterotrophic plate counts.
Other: total organic carbon, turbidity, colour, total suspended solids.

Other analytical parameters
Environmental authorities may insist that other parameters are also analysed, especially if wastewater is being discharged to a sensitive recipient, or if the site is potentially contaminated. Sampling and analysis should be undertaken according to strict protocols, which are discussed by USGS (2004) and Misstear *et al*. (2006).

Analysis of dissolved gases
Increasingly, the importance of dissolved gas management is being recognised in the context of schemes where aquifer re-injection is practised, because

- small bubbles of exsolved gas can lodge in aquifer pore spaces around an injection well, reducing the permeability and causing aquifer clogging;
- exposure to atmospheric oxygen can cause precipitation of iron and manganese oxyhydroxides, resulting in clogging of heat exchangers, injection wells and the aquifer (Banks *et al*., 2009b);
- exsolution and degassing of carbon dioxide can cause an increase in pH, facilitating precipitation of carbonates (e.g. calcite) and oxyhydroxides, resulting in clogging of heat exchangers, injection wells and the aquifer.

In such cases, it is becoming more usual to analyse groundwater for its content of dissolved gases such as nitrogen, oxygen and carbon dioxide. Guidance on sampling methodologies is given by USGS (2006) and Jahangir *et al*. (2010).

8.8.2 Statistical data for hard rock aquifers

In some aquifers, especially crystalline rock aquifers (e.g. slates, granites, quartzites, gneisses and some limestones and marbles), hydraulic properties do not vary geographically in a systematic or predictable manner. This is because the hydraulic conductivity depends on secondary structures (fractures and joints) rather than

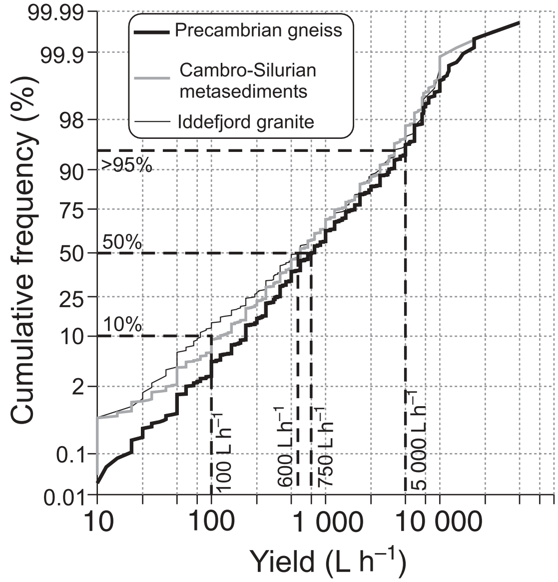


Figure 8.5 Cumulative frequency distribution curve showing distribution of well yields in three lithologies (Iddefjord Granite, Precambrian gneisses and Caledonian metasediments/metavolcanics) in Norway. The diagram enables us to predict that we have a 50% chance of obtaining a short-term yield of 600 L h^{-1} from the granite and 750 L h^{-1} from the gneisses, but only a <5% chance of obtaining 5000 L h^{-1} from the aquifers. We typically have ≈90% chance of obtaining >100 L h^{-1}. *Modified after Banks and Robins (2002) and reproduced by kind permission of Norges geologiske undersøkelse, Trondheim.*

primary ones. We cannot predict either yield or water quality in a deterministic manner in such aquifers. We can, however, use large data sets from such aquifers to describe the distributions of yield and water quality statistically (Banks and Robins, 2002; Banks *et al.*, 2005, 2010). For example, Figure 8.5 shows the distribution of well yields in three different aquifer lithologies in Norway. This diagram enables us to say that the median yield (50% chance of achieving this yield) is around 600 L h^{-1} in the Iddefjord granite and nearer 750 L h^{-1} in the Precambrian gneisses. It also enables us to say that, if we wish to achieve a yield of 5000 L h^{-1} in the Caledonian metasediments, we have a less than 5% chance of doing this with a single randomly drilled borehole (5000 L h^{-1} corresponds to >95 percentile on the cumulative frequency diagram). Be aware that the power of this technique depends on the quality and representativity of the underlying data set. Note also that yields cited in such data sets are often short-term yields reported by drillers. Long-term, 'sustainable' yields may be somewhat less.

Banks (1998) and Gustafson (2002) have also demonstrated methods of using such statistical data from single wells to predict the probability of achieving a given total yield from a wellfield comprising multiple wells.

8.9 Multiple wells in a wellfield

While some open-loop ground source heating and cooling schemes are based on a single well, many will require more than one well to achieve the required yield. It is a common fallacy among engineers that, if one well yields $5\,\mathrm{L\,s^{-1}}$, then 10 wells will provide a yield of $50\,\mathrm{L\,s^{-1}}$. This may not be the case (especially if re-injection is *not* practised) for several interrelated reasons:

- the aquifer may simply not receive enough recharge to support such a large total abstraction;
- the hydraulic properties of the aquifer may not be adequate to transmit a total flow of $50\,\mathrm{L\,s^{-1}}$ to a compact wellfield;
- the wells within the wellfield will hydraulically interfere with each other.

On the other hand, if the scheme utilises injection wells to return the wastewater to the original aquifer, the achievable, sustainable yield will usually be significantly higher than it would have been with an 'abstraction-only' scheme.

8.9.1 Multiple abstraction wells

Let us imagine an abstraction wellfield comprising three identical wells (A, B and C) of radius r_w in a line at spacing 30 m. For simplicity, we will assume no injection wells. Drawdown is an additive property: thus, the total drawdown in the middle well (B) at any time is *not* merely the drawdown predicted by the Theis or Cooper–Jacob equations, using the yield from the middle well (Z_B) and its radius (r_w). We must add to this the drawdowns caused by the abstraction from neighbouring wells A and C. Thus, the total drawdown (s_{wB}) in well B at a time t after pumping commences is given by

$$s_{wB} = \frac{Z_A}{4\pi T}W(u_A) + \frac{Z_B}{4\pi T}W(u_B) + \frac{Z_C}{4\pi T}W(u_C) \qquad (8.21)$$

where $u_A = u_C = r^2S/4Tt = (30\,\mathrm{m})^2S/4Tt$ and $U_B = r_w^2S/4Tt$.

If the values of u are small, we can apply the Cooper–Jacob approximation, resulting in

$$\begin{aligned} s_{wB} = {} & \frac{2.30Z_A}{4\pi T}\log_{10}\left(\frac{2.25T}{(30\ \mathrm{m})^2 S}\right) + \frac{2.30Z_C}{4\pi T}\log_{10}\left(\frac{2.25T}{(30\ \mathrm{m})^2 S}\right) + \\ & \frac{2.30Z_B}{4\pi T}\log_{10}\left(\frac{2.25T}{r_w^2 S}\right) + \frac{2.30(Z_A + Z_B + Z_C)}{4\pi T}\log_{10}t \end{aligned} \qquad (8.22)$$

Similarly, for the drawdown in well A,

$$s_{wA} = \frac{2.30Z_B}{4\pi T}\log_{10}\left(\frac{2.25T}{(30\ \mathrm{m})^2 S}\right) + \frac{2.30Z_C}{4\pi T}\log_{10}\left(\frac{2.25T}{(60\ \mathrm{m})^2 S}\right) + \frac{2.30Z_A}{4\pi T}\log_{10}\left(\frac{2.25T}{r_w^2 S}\right) + \frac{2.30(Z_A + Z_B + Z_C)}{4\pi T}\log_{10}t \quad (8.23)$$

In other words, the drawdown in any given well in a wellfield comprising multiple abstraction wells is greater than it would be for the same well, pumping at the same rate, on its own, *if no re-injection is practised*. Thus, the yield from each well in a multiple wellfield, for a given design drawdown, will be less than if the well was alone. Note that the above equations assume ideally efficient wells: real drawdowns may be larger due to well inefficiency (see Section 8.7.1).

8.9.2 Abstraction and injection wells

Injection wells (Figure 8.6) can be considered as 'negative' abstraction wells, where a negative groundwater pumping rate $-Z$ can be substituted into the Theis, Cooper–Jacob or Logan equations. This results in a negative drawdown, or 'upconing', of the water table or piezometric surface. In a wellfield where re-injection is practised, comprising both abstraction and injection wells, the upconing around the injection wells will partly cancel out the drawdown around the abstraction wells. The result of this is that drawdown and upconing are much less (and the yield for a given drawdown is greater) than they otherwise would have been, and the wellfield's sustainability is improved.

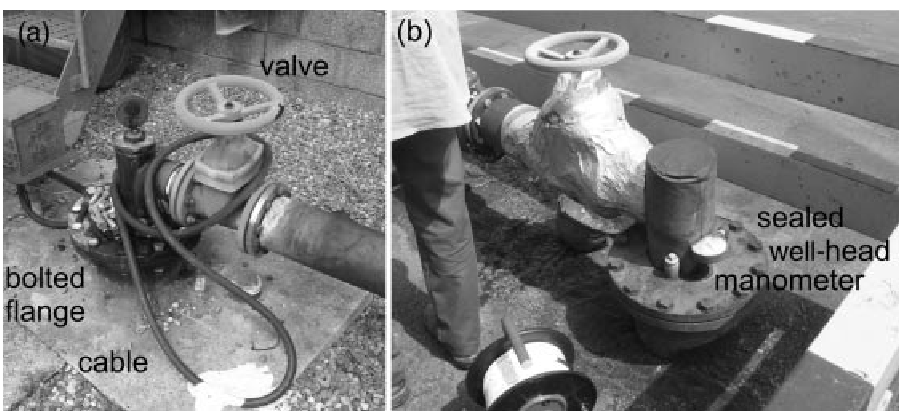


Figure 8.6 (a) The abstraction well of a well doublet, used for industrial cooling, in the Sherwood Sandstone of Yorkshire, UK. Note the bolted well-head flange, the electric cable leading to the submersible pump and the control valve on the rising main. Off to the right (not seen) is a sampling line for taking water samples; (b) the recharge well of the same well doublet. This well head is sealed and operated under pressure (the aquifer is confined by low permeability clays and silts). The well-head is fitted with a manometer to monitor pressure.

The same technique can be applied as in Section 8.9.1, but this time subtracting the terms relating to the injection wells. Let us consider a simple *well doublet* arrangement, with one abstraction well (rate Z) and a single injection well (rate $-Z$) at a distance L apart, both with radius r_w. The relevant equation for drawdown in the abstraction well (s_w), for small values of u (i.e. large values of t), is

$$s_w = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25T}{r_w^2 S}\right) - \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25T}{L^2 S}\right) + \frac{2.30(Z-Z)}{4\pi T}\log_{10}t \qquad (8.24)$$

Note here that the time-dependent term becomes zero and a steady-state drawdown and flow field should arise. The steady-state drawdown in the abstraction well in the doublet is lower than it would have been in case of a single abstraction well, and is given by Gringarten (1978):

$$s_w = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{L^2}{r_w^2}\right) = \frac{Z}{2\pi T}\ln\left(\frac{L}{r_w}\right) \qquad (8.25)$$

The upconing in the injection well should also be $-s_w$, if our injection well is 100% hydraulically efficient. Reality is never so obliging.

8.9.3 Injection wells

In theory, injection wells are the negative image of abstraction wells. Re-injection of water causes an upconing of groundwater levels instead of a drawdown. The available upconing may prove to be a major constraint on the system: the available upconing is the difference between the rest water level and the ground surface (Figure 8.8). If we wish to inject quantities of water that will result in an upconing greater than this value, then we must either inject water under excess pressure into a sealed well head (which is feasible, and may even be desirable, *if* the aquifer is confined – see Figures 8.6, 8.7 and Section 8.9.4) or use a larger number of conventional injection wells.

Injection wells require specialist design and construction. As a trivial example, wire-wrapped well screens in abstraction wells are designed to prevent clogging by particles by using a V-shaped aperture (Figure 7.7c). If we choose to use such a well screen in an injection well, the groundwater will flow into the 'V' rather than out of it, an ideal situation to promote clogging if particles are present in the water (this does not mean that we cannot use such well screens in injection wells – merely that we should be careful).

Injection wells will always be more susceptible to clogging and degradation of performance than abstraction wells. With abstraction wells, we are sucking water out of the aquifer, and there will be some self-cleaning tendency. With injection wells, we are forcing water into the aquifer (and any well screen); if the re-injected water contains any particles or chemical precipitates, these can lodge in the well screen or aquifer pore spaces, reducing their permeability. Even tiny gas bubbles can clog aquifer

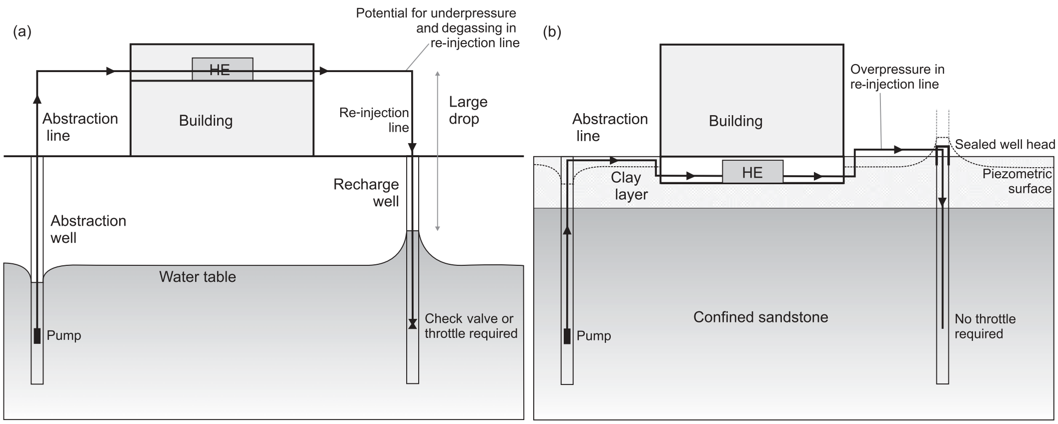


Figure 8.7 (a) A situation where there is a high risk of underpressure in the re-injection pipeline of a well doublet (deep groundwater level and high plant room); (b) a lower risk situation (basement plant room and sealed, pressurised well head). HE = heat exchanger (or heat pump).

pore spaces as effectively as particles or biofilms, if our management of dissolved gases is poor. Putting it very bluntly, we should usually expect to drill more re-injection wells (or metres of injection well) to accept a given flux of groundwater back into an aquifer, than we would require to abstract it.

Injection wells require special operation and monitoring. The re-injected water must be particle-free to prevent clogging of the well screen or aquifer. It should also be microbiologically inactive to prevent bacterial biofilm growth on the well screen or borehole wall: it is common practice to apply UV disinfection to injected water. The re-injected water should not contain gas bubbles or concentrations of dissolved gas that are likely to exsolve in the aquifer. Finally, one must also be aware of the possibility of chemical precipitation and clogging of the injection well or heat exchanger/heat pump. Contact between water and atmospheric oxygen may increase the risk of precipitation of iron and manganese oxyhydroxides. Degassing of excess carbon dioxide, especially in ground source cooling schemes where the re-injected water has been heated, may promote precipitation of calcite. Chemical analyses of the groundwater are useful to find out if it contains problematic concentrations of iron and manganese, or is particularly hard. Abstraction and (especially) injection wells will usually require a pre-planned programme of cleaning and maintenance to keep them in operating efficiently.

8.9.4 Management of dissolved gases

Groundwater usually contains dissolved gases. Its content of relatively inert nitrogen (N_2) will usually be similar to atmospheric, or slightly above. Its content of oxygen (O_2) will usually be lower than atmospheric, as oxygen reacts with many subsurface minerals and organic carbon. In deep/confined aquifers, groundwater may be almost oxygen-free. In such chemically reducing conditions, large concentrations of iron and manganese can dissolve in groundwater – but, as soon as the water is re-exposed to oxygen, these elements oxidise to insoluble forms and precipitate out as sticky oxy-hydroxides. The partial pressure of carbon dioxide (CO_2) in groundwater can be higher than atmospheric: CO_2 is released to groundwater by the respiration of soil bacteria, the decomposition of organic matter and the action of acids on carbonate minerals. For all these reasons, when we pump deep groundwater up to the surface, there is risk of several things happening:

- Degassing of excess concentrations of gases such as N_2 and CO_2. They can form small bubbles that clog the aquifer pore spaces around re-injection wells. Because CO_2 is a mildly acidic gas, when it is degassed from groundwater, the groundwater becomes more alkaline, thereby facilitating the precipitation of minerals such carbonates and oxyhydroxides, that can cause incrustation and clogging.
- Contact of deep groundwater with atmospheric oxygen causes oxidation and precipitation of iron and manganese (if present) as oxyhydroxides, that can cause incrustation and clogging (Banks *et al.*, 2009b).

When operating an abstraction-heat exchange-re-injection scheme (e.g. a well doublet), it very important to have good-quality analyses of the water's chemistry (especially parameters such as pH, alkalinity, iron, manganese, calcium) and its dissolved gas content (see Section 8.8.1 for advice on analysis). Use of hydrochemical modelling programmes can assist us in predicting whether there is a risk of degassing or mineral precipitation (Banks *et al.*, 2009b).

Contact between the groundwater and the atmosphere should be avoided. The abstraction-heat exchange-re-injection circuit should be a sealed line and re-injection should take place well below the water level in the re-injection well (*not* allowed to cascade freely into the re-injection well head).

It is also very important to *manage* the pressure within the abstraction-heat exchange-re-injection line. Simple pipe flow models can assist in calculating the pressure conditions at various points in the circuit. The overall objective is that there should be no locations of *underpressure* within the circuit and preferably a significant overpressure (say, >0.5 bar above atmospheric). One should avoid large vertical drops on the re-injection line (e.g. if the heat exchanger plant room is on the top floor of a building and the water level in the re-injection well is 20 m below ground level). In such a case, some form of passive throttle or active pressure regulation valve on the foot of the recharge main will be advisable in order to maintain an overpressure in the recharge line and to prevent degassing. On the other hand, if the plant room is at ground level or in a cellar, and if the re-injection well head is under pressure, it may be possible to operate the re-injection main simply as an open pipe, without a throttle or check valve (Figure 8.7; Bakema, 2001).

Note that if you are planning the operate the re-injection well with a sealed well head (Figures 8.6 and 8.7), you should verify that the aquifer is confined by a low-permeability covering layer. If the aquifer is unconfined and you attempt to re-inject under excess pressure, you risk the water table rising above the ground surface and causing groundwater flooding of the plant room and any neighbouring areas. If the aquifer is confined, you should also ensure that the well casing is securely grouted, so that excess pressure of groundwater cannot leak out behind the well casing. You should also ensure that the re-injection pressure does not exceed the 'weight' of the confining layer of strata – otherwise, there is a risk that you will simply lift the 'lid' off the aquifer with important geotechnical consequences. These considerations will usually lead you to impose a maximum allowable recharge pressure on your well.

8.10 Hydraulic feedback in a well doublet

8.10.1 Hydraulic feedback

Let us, for the purposes of Sections 8.10–8.12, assume that we are designing a ground source cooling scheme, where we abstract cold water at a rate Z from an aquifer at temperature θ_{gout}, and re-inject it to the same aquifer at the same rate at a higher temperature θ_{ginj} via an injection well at a distance L down the hydraulic gradient of the aquifer. Let us call such an arrangement an *open-loop well doublet*. The heat rejected to the groundwater (G) is given by

$$G = (\theta_{ginj} - \theta_{gout}) \cdot S_{VCwat} \cdot Z \tag{8.26}$$

where S_{VCwat} is the specific volumetric heat capacity of water. The average cooling load (C) delivered to the building is then estimated by

$$C = (\theta_{ginj} - \theta_{gout}) \cdot S_{VCwat} \cdot \frac{Z}{(1+(1/\mathrm{SPF}_C))} \tag{8.27}$$

where SPF_C is the seasonal performance factor for the cooling system. This should be very high ($1/\mathrm{SPF}_C \approx 0$) for free cooling systems, while SPF_C may be between 4 and 6 for heat-pump-based systems.

In a fit of optimism, we might hope that, if we site the abstraction well up the hydraulic gradient from the injection well, all the waste heat will be carried away (to some unspecified destination) by groundwater flow (Figure 8.8) and our cooling scheme will function sustainably. Of course, the plume of waste heat from the re-injection well may impact groundwater users or environmental features further down-gradient and may prove unacceptable to environmental authorities. However, if our injection well is situated too close to the abstraction well (Figure 8.9), we may start to induce a component of warm groundwater flow back towards the abstraction well. Clyde and Madabhushi (1983) have in fact demonstrated that this will occur if

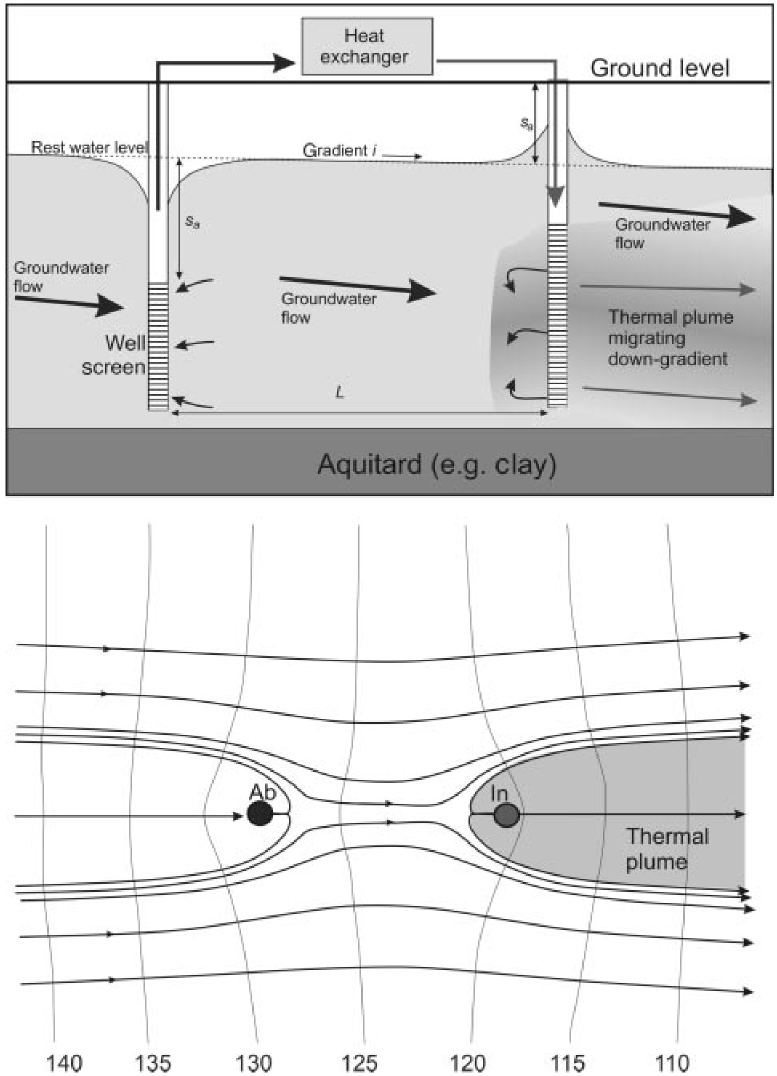


Figure 8.8 A well-doublet system, comprising an abstraction well (Ab), situated immediately up the hydraulic gradient from a re-injection well (In). The upper diagram shows a schematic cross section, and the lower diagram shows a plan view. The distance L between the wells is greater than $2Z/T\pi i$ (see Equation 8.28), and there is no hydraulic (or thermal) feedback between the wells. In the lower diagram, black arrows show groundwater flow lines; thinner numbered lines are groundwater contours, with arbitrary head values (groundwater levels decline from left to right). s_a shows the available drawdown/upconing for each well.

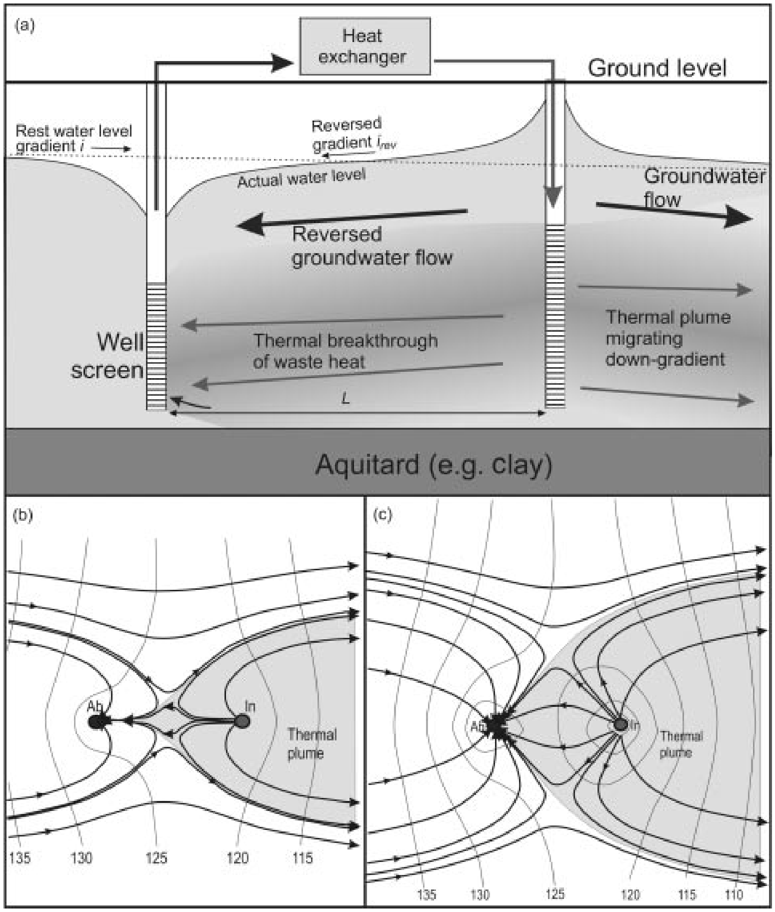


Figure 8.9 A well-doublet system, comprising an abstraction well, situated immediately up the hydraulic gradient from a re-injection well. (a) This shows a schematic cross section. The distance L between the wells is less than $2Z/T\pi i$, and there is hydraulic (and thermal) feedback between the wells. (b) This shows a plan view of the situation where L is slightly less than $2Z/T\pi i$ and there is a minimal amount of hydraulic feedback. (c) This shows the situation where L is significantly less than $2Z/T\pi i$. Symbols as for Figure 8.8.

$$L < \frac{2Z}{T\pi i} \tag{8.28a}$$

or, put another way,

$$\beta > 1 \text{ where } \beta = 2Z/(T\pi iL) \tag{8.28b}$$

where i is the natural hydraulic gradient (metres per metre; dimensionless) and T is the aquifer transmissivity. If the spacing of the well doublet is shorter than this critical value, there is a possibility that a proportion of the warm, re-injected wastewater will find its way back into the abstraction well. This will increase the temperature of the abstracted water and render the ground source cooling scheme less efficient or even, in the worst case, unsustainable. Let us plug some typical values into Equation 8.28: assuming $T = 150\,\mathrm{m^2\,day^{-1}}$, $Z = 10\,\mathrm{L\,s^{-1}} = 864\,\mathrm{m^3\,day^{-1}}$ and $i = 0.01$, then we find that the spacing L must be at least 367 m for there to be no risk of hydraulic or thermal feedback. In many cases, it is impossible to achieve such a large spacing between abstraction and recharge wells, and we thus often have to accept a risk of hydraulic and thermal feedback. Just because the risk is present, however, does not mean that the system is doomed to failure. In fact, such a scheme can be sustainable or can, at least, have a very long lifetime, because

1. breakthrough of heat in the abstraction well does not happen immediately. In fact, it may take many years (although it may only take weeks or months in some fissured or karstified aquifers);
2. only a small portion of the water abstracted from the abstraction well may consist of recirculated water from the injection well;
3. if we have a heating demand in winter and a cooling demand in summer, we may be able to operate the scheme reversibly, effectively recovering the summer's warm wastewater during the winter heating season without any thermal breakthrough occurring.

To assess the risk of thermal breakthrough, we thus need to consider the speed with which groundwater travels between the injection and abstraction wells. Before we do this, however, we should note that, if we are crazy enough to locate our injection well directly up-gradient of the abstraction well, it is likely that we will develop a closed thermal cell (Figure 8.10), with all our re-injected warm water feeding back to our abstraction well.

8.10.2 Rate and proportion of hydraulic breakthrough

One way of assessing hydraulic breakthrough time (i.e. the time it takes for groundwater to travel from the injection well to the abstraction well) is to use a computer-based numerical model, which has a so-called particle tracking capability. This will allow us to simulate the motion of large numbers of water molecules along different flow paths (and often to consider dispersion effects) and to calculate an average hydraulic breakthrough time in what is rapidly becoming a complex hydrogeological problem. There are two analytical methods we can use to estimate this breakthrough time, however, (1) by simple application of Darcy's law and (2) by the Doublet Breakthrough method.

Darcy's law

We can calculate, from Equation 8.25, the expected steady-state drawdown and upconing of groundwater levels in the aquifer adjacent to the abstraction and injection wells.

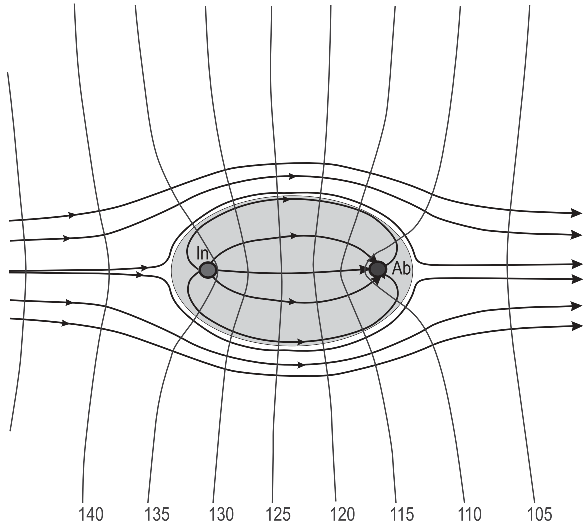


Figure 8.10 A well-doublet system, comprising an abstraction well, situated immediately down the hydraulic gradient from a re-injection well. This plan view shows that the thermal 'plume' forms a closed cell, with all the re-injected water being relatively rapidly returned to the abstraction well. Symbols as for Figure 8.8.

In our example here ($T = 150\,\text{m}^2\ \text{day}^{-1}$, $Z = 10\,\text{L}\,\text{s}^{-1} = 864\,\text{m}^3\ \text{day}^{-1}$), if we assume that $r_w = 100\,\text{mm}$ and the well separation $L = 100\,\text{m}$, we calculate that the drawdown adjacent to the abstraction well is 6.3 m and the upconing adjacent to the injection well is 6.3 m. Thus, we have a head difference (Δh) between the injection and abstraction well of 12.6 m. We can, if we are feeling sufficiently daring, make three dubious assumptions: (1) we can forget that the flow field around the wells will not be linear, (2) we can assume that the hydraulic gradient between the two wells is constant, and (3) we can assume that the natural hydraulic gradient (i) is negligible. We can thus say that the reverse hydraulic gradient (i_{rev}) between the two wells is 12.6 m/100 m or 0.126. Darcy's law states that the flow rate per unit cross-sectional area of aquifer (the so-called Darcy velocity v_D) is given by

$$v_D = Ki_{rev} = \frac{K\Delta h}{L} \tag{8.29}$$

But we must remember that this is not the actual linear velocity of groundwater flow. As flow only takes place through pore space and not through solid mineral grains, we also have to divide by the effective porosity (i.e. the interconnected porosity through which flow takes place, n_e) to obtain the actual linear flow velocity (v):

$$v = \frac{Ki_{rev}}{n_e} = \frac{K\Delta h}{Ln_e} \tag{8.30}$$

Thus, the hydraulic breakthrough time (t_{hyd}) can be estimated as follows:

$$t_{hyd} = \frac{L}{v} = \frac{L^2 n_e}{K\Delta h} \tag{8.31}$$

This probably represents a worst-case estimate of minimum breakthrough time along the most direct flow path between the injection and abstraction wells.

Double Breakthrough time for zero hydraulic gradient

If we are troubled by the dubious assumptions involved in using Darcy's law (which we should be!), we may choose to apply a somewhat more sophisticated analytical method derived from a geometric consideration of the flow paths in a well doublet. In fact (Raudkivi and Callander, 1976; Luo and Kitanidis, 2004; Banks, 2011a), a whole system of equations exists to mathematically describe our well-doublet system, using the parameter β (see Equation 8.28b). All of these equations assume that the groundwater (and heat) transport in the well doublet is horizontal – that is two-dimensional. They assume that no groundwater (or heat) migrates upwards or downwards. This is often quite a good assumption for groundwater (but it is less valid for heat). We will examine the implications of this assumption later. If we assume that the natural hydraulic gradient (i) is zero, the flow paths will all be arcs of circles (Figure 8.11).

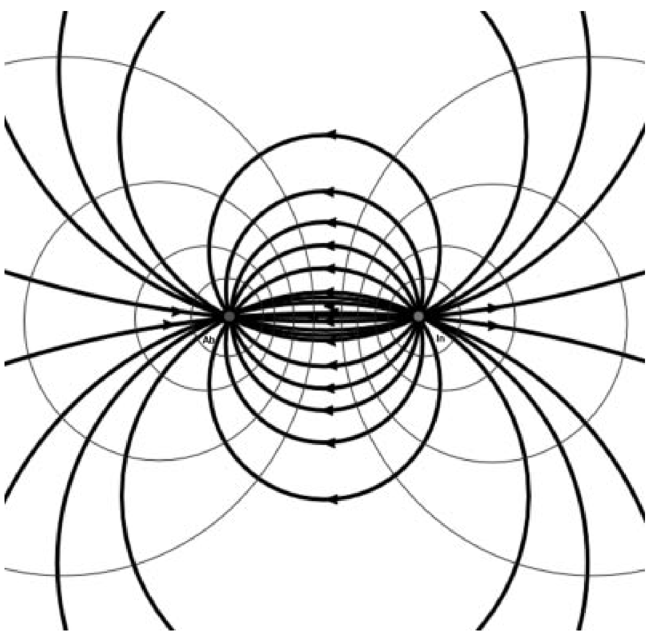


Figure 8.11 A well-doublet system, comprising an abstraction well (left) and a re-injection well (right); in this case there is no natural hydraulic gradient. Both the flow lines between the wells (thicker arrowed lines) and groundwater head contours (thinner lines) all form arcs of circles, making the scenario amenable to geometric analysis.

The doublet formula for the travel time (t_{hyd}) along the shortest flow path (a straight line) between the injection and abstraction well is found to be (Grove, 1971; Güven *et al.*, 1986; Himmelsbach *et al.*, 1993; Banks, 2009b)

$$t_{\text{hyd}} = \pi n_e D \frac{L^2}{3Z} \tag{8.32}$$

where D is the effective aquifer thickness (m). This equation is probably most appropriate for simple dipole systems of one abstraction and one injection well. The Darcy approach (above) may be better justified where we have a row of several abstraction wells separated from a row of injection wells, and where flow conditions may approach linear flow rather than radial flow.

Doublet Breakthrough modification for finite hydraulic gradient

Equation 8.32 presupposes a negligible initial natural hydraulic gradient (i). If this gradient is not negligible, several authors (Lippmann and Tsang, 1980; Clyde and Madabhushi, 1983; Luo and Kitanidis, 2004; Banks, 2011a; Barker, 2012) note that a more general form of the equation exists for the case where the injection well is situated down the regional (natural) hydraulic gradient from the abstraction well (as in Figure 8.9). The hydraulic breakthrough time is given as

$$t_{\text{hyd}} = \frac{Ln_e}{Ki}\left[\frac{\beta}{\sqrt{\beta-1}}\tan^{-1}\left(\frac{1}{\sqrt{\beta-1}}\right)-1\right] \tag{8.33}$$

When using these equations, note the following:

1. Hydraulic breakthrough times are proportional to effective porosity. In fissured and fractured limestones or crystalline rocks, this effective porosity may be <0.01 and we can expect potentially rapid breakthrough.
2. The equation does not account for hydrodynamic dispersion. In real aquifers, some fractures or pore channels are wider than others; some water molecules flow down the middle of channels while others drag along the side of pore spaces. In other words, dispersion means that some water molecules arrive before the theoretical breakthrough time, while others arrive after. The equations give the average breakthrough time along the most rapid flow path. In karstified or fractured aquifers, we may get a surprisingly rapid breakthrough if we have encountered a transmissive flow pathway (large-aperture fracture or fissure) in our wells.

Extent of breakthough in a doublet

The set of equations documented by Luo and Kitanidis (2004) and Banks (2011a) also allow us to predict the proportion (f_{recirc}) of abstracted water that will have been recirculated from the re-injection well:

$$f_{\text{recirc}} = 1 - \frac{2}{\pi}\left(\tan^{-1}\left(\frac{1}{\sqrt{\beta-1}}\right)+\frac{\sqrt{\beta-1}}{\beta}\right) \tag{8.34}$$

A worked example

Let us return to our example (above) of a doublet with a well separation of 100 m, $T = 150\,\mathrm{m}^2\,\mathrm{day}^{-1}$, $Z = 10\,\mathrm{L\,s}^{-1} = 864\,\mathrm{m}^3\,\mathrm{day}^{-1}$ and $r_w = 100\,\mathrm{mm}$. Let us now additionally assume that the aquifer is 75 m thick, implying a hydraulic conductivity of $150\,\mathrm{m}^2\,\mathrm{day}^{-1}/75\,\mathrm{m} = 2\,\mathrm{m\,day}^{-1}$ and has an effective porosity $n_e = 10\%$, or 0.1.

Let us first assume that the natural, regional hydraulic gradient is flat ($i = 0$). Using Equation 8.31, we can roughly estimate a hydraulic breakthrough time of

$$t_{\mathrm{hyd}} = (100\ \mathrm{m})^2 \times 0.1 / (2\ \mathrm{m\ day}^{-1} \times 12.6\ \mathrm{m}) = \mathrm{c.}\ 40\ \mathrm{days}$$

We can then try the more valid Equation 8.32 to refine our answer:

$$t_{\mathrm{hyd}} = \pi \times 0.1 \times 75\ \mathrm{m} \times (100\ \mathrm{m})^2 / (3 \times 864\ \mathrm{m}^3\ \mathrm{day}^{-1}) = \mathrm{c.}\ 91\ \mathrm{days}$$

This second method provides a more valid answer in this context.

Now, we can assume a natural regional hydraulic gradient of $i = 0.01$, with the re-injection well immediately down-gradient of the abstraction well.

Firstly, we calculate that $\beta = 2 \times 864\,\mathrm{m}^3\,\mathrm{day}^{-1}/(150\,\mathrm{m}^2\,\mathrm{day}^{-1} \times \pi \times 0.01 \times 100\,\mathrm{m}) = 3.67$. We can use Equation 8.33 to estimate that

$$t_{\mathrm{hyd}} = 500 \times (2.25 \times 0.549 - 1)\ \mathrm{days} = 117\ \mathrm{days}$$

In other words, the regional hydraulic gradient is opposing the hydraulic breakthrough and t_{hyd} increases. Equation 8.34 indicates that the fraction of recirculated water is

$$f_{\mathrm{recirc}} = 1 - 0.637 \times (0.549 + 0.445) = 0.37,\ \text{or } 37\%$$

8.11 Heat migration in the groundwater environment

The astute reader will have noted that, hitherto, we have only talked about the breakthrough of groundwater and water molecules from the injection to the abstraction well. Can we assume that heat travels at the same speed as groundwater and is merely carried along passively? In fact, we cannot. In a groundwater aquifer, heat travels by three mechanisms:

1. conduction through mineral grains and water-filled pores;
2. advection with bulk groundwater flow;
3. exchange between moving groundwater and the matrix of the aquifer (mineral grains and any immobile pore water).

Many hydrogeologists assume that heat exchange is very rapid and that the aquifer matrix equilibrates with groundwater temperature speedily. De Marsily (1986) estimated that thermal equilibration between a sand grain (1-mm diameter) and surrounding groundwater takes less than 1 min, while equilibration between a 10-cm cobble and groundwater may take 2 h. This means that a single point in an aquifer system can be characterised by a single temperature (θ) at any given time t (which applies both to the matrix and to the mobile groundwater). This is called the *assumption of instantaneous thermal equilibration*. The mathematically inclined can write an equation describing heat transport in such an aquifer (De Marsily, 1986; Shook, 2001a) – in one dimension only it is

$$\lambda^* \frac{d^2\theta}{dx^2} - S_{VCwat} \frac{d(v_D\theta)}{dx} = S_{VCaq} \frac{d\theta}{dt} \tag{8.35}$$

where λ^* is the effective thermal conductivity of the saturated aquifer (see Box 8.3), S_{VCwat} and S_{VCaq} are the specific volumetric heat capacities of the groundwater and the saturated aquifer (i.e. mineral matrix plus pore water), respectively. v_D is the Darcy velocity of groundwater flux through the aquifer (= vn_e). The first term in Equation 8.35 describes heat transport by conduction only, the second term describes heat transport by convection and the third term describes the change in heat stored in a unit volume of aquifer with time.

Let us imagine a re-injection well that fully penetrates an aquifer of thickness D. It steadily pumps out warm water into the surrounding aquifer, of effective porosity n_e, at a rate Z. Let us assume that there is no regional hydraulic gradient and the flow field is radial. Let us calculate the time t_{hyd} it would take for a water molecule to travel a radial distance r from the well. This time is given by the time it would take to fill a cylinder of aquifer of radius r. The volume of water contained in the cylinder is given by multiplying the cylinder's volume by the effective porosity

$$2 \cdot \pi \cdot r \cdot D \cdot n_e = Z \cdot t_{hyd} \tag{8.36}$$

If the re-injected water is at a temperature $\Delta\theta$ higher than the ambient aquifer temperature, the amount of heat able to be contained by this cylinder of aquifer is given by $2\pi rDS_{VCaq}\Delta\theta$. Thus, the time t_{the} it would take for a heat signal to reach radius r is given by

$$2 \cdot \pi \cdot r \cdot D \cdot S_{VCaq} \cdot \Delta\theta = Z \cdot t_{the} \cdot S_{VCwat} \cdot \Delta\theta \tag{8.37}$$

Thus, because heat is absorbed from the groundwater (which only fills the aquifer pore space) into the mineral grains that comprise the aquifer matrix, the heat signal is effectively slowed down or retarded. The ratio between the thermal and hydraulic travel time is given termed the thermal retardation factor R_{the}:

$$R_{the} = \frac{t_{the}}{t_{hyd}} = \frac{S_{VCaq}}{n_e S_{VCwat}} \tag{8.38}$$

BOX 8.3 The Analogy between Heat Transport and Solute Transport in Groundwater

In its full three-dimensional glory, Equation 8.35 can be written (De Marsily, 1986) as follows:

$$\mathrm{div}\left(\frac{\lambda^*}{\rho_{\mathrm{wat}} \cdot S_{\mathrm{Cwat}}} \mathrm{grad}\theta\right) - \mathrm{div}\,(v_{\mathrm{D}} \cdot \theta) = \frac{\rho_{\mathrm{aq}} \cdot S_{\mathrm{Caq}}}{\rho_{\mathrm{wat}} \cdot S_{\mathrm{Cwat}}} \cdot \frac{\mathrm{d}\theta}{\mathrm{d}t}$$

where the following (in addition to gradθ) are tensors:

$\lambda^\star$ = effective thermal conductivity of the saturated aquifer material, modified to take into account effects of hydrodynamic dispersion;

v_{D} = Darcy velocity (rate of flow of groundwater per metre cross section).

The following are scalars:

θ = temperature at a given point in the aquifer–groundwater system;

S_{C} = specific heat capacity ($\mathrm{J\,kg^{-1}\,K^{-1}}$);

ρ = density.

The subscripts 'wat' and 'aq' refer to the groundwater and the bulk saturated aquifer (pore water plus matrix), respectively. We can compare this to the equation for the migration of a non-adsorbed solute in groundwater, where C is a dimensionless concentration of that solute (relative to natural background):

$$\mathrm{div}\,(D^\star \cdot \mathrm{grad}C) - \mathrm{div}\,(v_{\mathrm{D}} \cdot C) = n_{\mathrm{e}} \cdot \frac{\mathrm{d}C}{\mathrm{d}t}$$

where $D^\star$ is a dispersion term reflecting both molecular diffusion and hydrodynamic dispersion. By analogy then, we can now say that

- $\lambda^\star/(\rho_{\mathrm{wat}} S_{\mathrm{Cwat}})$ is a term for 'thermal dispersion', reflecting both thermal conduction and hydrodynamic dispersion. It *may* be comparable but will not necessarily be the same as $D^\star$.
- The velocity v of the non-sorbing chemical solute (which travels with bulk groundwater flow) is related to v_{D} by a factor n_{e}.
- The velocity of the heat front v_{the} is related to v_{D} by a factor $(\rho_{\mathrm{aq}} S_{\mathrm{Caq}})/(\rho_{\mathrm{wat}} S_{\mathrm{Cwat}})$.

Thus, $v_{\mathrm{D}} = v n_{\mathrm{e}} = v_{\mathrm{the}} \rho_{\mathrm{aq}} S_{\mathrm{Caq}}/(\rho_{\mathrm{wat}} S_{\mathrm{Cwat}})$ and $v_{\mathrm{the}} = v n_{\mathrm{e}} S_{\mathrm{VCwat}}/S_{\mathrm{VCaq}}$.

BOX 8.4 The Péclet Number

This is a dimensionless number that characterises the relative importance of advective and conductive heat transfer processes in fluid systems. It was invented by the French physicist Jean Claude Eugène Péclet. For a given characteristic length (L), the Péclet Number (Pe_L) is given by

$$Pe_L = {}^{Lv}\!/_{\alpha} = {}^{LvS_{VC}}\!/_{\lambda}$$

where v is the fluid velocity, S_{VC} is the volumetric heat capacity, α = thermal diffusivity and λ = thermal conductivity. A Péclet Number greater than 1 implies that advective heat transfer is dominant, while a Péclet Number less than 1 suggests that conductive heat transfer may be more important.

Heat transport in a groundwater flow field in an aquifer is usually slower than the transport of the water molecules (or a conservative chemical tracer travelling with the water flow).

Note that we have, in Equations 8.36–8.38, considered advection of heat with groundwater flow and heat exchange with the mineral matrix (assumed to be instantaneous). We have *not* explicitly considered heat conduction through the aquifer. This is because, in many aquifers, groundwater flow is faster than heat conduction. In some geological strata, conductive heat transport may become significant, as compared with advection, either because (1) the strata are of very low permeability or (2) because there is no groundwater head gradient to drive groundwater flow.

The relative importance of heat conduction versus heat advection in any hydraulic system is described by the Péclet Number (Pe) – Box 8.4.

8.11.1 Thermal breakthrough in well doublet

The result of this consideration is that, if we wish to estimate the time for thermal breakthrough in an open-loop well doublet, we can simply multiply the hydraulic breakthrough time by a factor R_{the}. Thus, if the natural hydraulic gradient is zero (Figure 8.11), the Equation 8.32 becomes (Gringarten, 1978; Clyde and Madabhushi, 1983; Banks, 2009b, 2011a)

$$t_{\text{the}} = \pi D \frac{S_{\text{VCaq}} L^2}{3 S_{\text{VCwat}} Z} \tag{8.39}$$

Güven *et al.* (1986) described how the temperature in the abstraction well increases with time, as hydraulic breakthrough occurs. They assume no three-dimensionality (i.e. no heat leakage into over- or underlying strata). The shape of the temperature

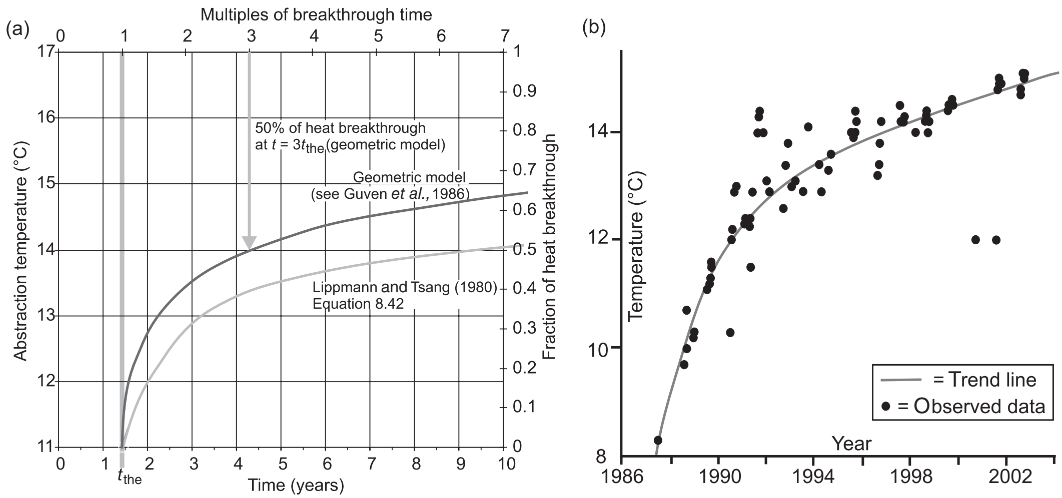


Figure 8.12 (a) The calculated evolution of temperature of water from an abstraction well in a well doublet (natural hydraulic gradient $i = 0$). The re-injection well is operating at a constant temperature of 17°C and the ambient groundwater temperature is 11°C. Thermal breakthrough is assumed to occur after 1.43 years of operation; (b) the real evolution in temperature in an abstraction well of an open-loop doublet system in Winnipeg, Canada, described in Section 8.12.1. *Diagram (b) is redrawn after Ferguson and Woodbury (2005) and reproduced by kind permission of the National Research Council of Canada.*

breakthrough curve is given in Figure 8.12a. It is seen that 50% breakthrough has occurred after a time $3t_{the}$. The type curve assumes no regional natural hydraulic gradient.

If the initial regional natural hydraulic gradient is finite, Equation 8.33 becomes (Clyde and Madabhushi, 1983; Banks, 2011a)

$$t_{the} = \frac{LS_{VCaq}}{S_{VCwat}Ki}\left[\frac{\beta}{\sqrt{\beta-1}}\tan^{-1}\left(\frac{1}{\sqrt{\beta-1}}\right)-1\right] \tag{8.40}$$

It is important to realise that these systems of equations build upon a number of simplifying assumptions:

- That groundwater flow obeys Darcy's law and that the aquifer can adequately be simulated as a homogeneous, saturated porous medium (although even heterogeneity can be tackled – Shook, 2001a).
- The equations do not account for dispersion effects. Some thermal breakthrough will inevitably occur ahead of the calculated mean travel time. If there are major open fractures present or if the aquifer is heterogeneous with rapid flow pathways, this may result in macrodispersive effects and the very rapid breakthrough of a significant portion of the thermal signature.

- That thermal equilibration is instantaneous between groundwater and aquifer matrix. If flow is through a limited number of widely spaced, discrete fractures, rather than through an extensive network of pores and small fractures, thermal equilibration between water and matrix (i.e. the blocks of rock between fractures) will not be instantaneous. This will lead to overestimation of thermal travel times (see Barker, 2010; Banks, 2011a).
- That there is no vertical conductive transfer of heat to strata overlying or underlying the aquifer (although, if this occurs, it will typically improve reservoir lifetime and can be tackled mathematically – Gringarten, 1978; see Section 8.12).

In short, the equations are likely to be adequate in sedimentary aquifers such as sands, gravels and many sandstones. They are even likely to be significantly conservative, as they do not take into account three-dimensionality (i.e. transfer of heat with overlying and underlying strata). In fractured or fissured aquifers, such as limestones, Chalk or crystalline rocks, there is a danger that they may overestimate thermal breakthrough times, especially over short distances. Some progress towards developing thermal breakthrough models for fractured rock aquifers has been made by Shook (2001b), Law *et al.* (2007) and Barker (2010).

8.11.2 A return to our worked example

Let us return to our example of a doublet scheme of separation $L = 100\,\mathrm{m}$, pumping at a rate $Z = 10\,\mathrm{L\,s^{-1}} = 864\,\mathrm{m^3\,day^{-1}}$ in a sand aquifer of transmissivity $T = 150\,\mathrm{m^2\,day^{-1}}$, thickness 75 m, hydraulic conductivity $K = 2\,\mathrm{m\,day^{-1}}$ and effective porosity $n_e = 0.10$. Let us further assume that we abstract groundwater at 11°C and re-inject it at 17°C.

Our doublet Equation 8.32 yielded an estimated hydraulic breakthrough time, in the absence of any regional groundwater gradient, of 91 days. We know that the specific heat capacity of water is around $4180\,\mathrm{J\,L^{-1}\,K^{-1}}$. Let us assume that our aquifer is composed of quartz-dominated sand grains and has a saturated volumetric heat capacity $S_{\mathrm{VCaq}} = 2.4\,\mathrm{MJ\,m^{-3}\,K^{-1}}$ (Box 8.5). The retardation factor for the aquifer can then be estimated from Equation 8.38 as follows:

$$R = \frac{2.4}{0.1 \times 4.18} = 5.74 \tag{8.41}$$

This allows us to predict a thermal breakthrough time (Equation 8.39) of around 5.74 × 91 days = 522 days (1.43 years), and a temperature evolution in the abstraction well as shown in Figure 8.12a. Note that the predicted temperature in the abstraction well only exceeds 14°C after 4–9 years' continuous operation. If we assume a natural hydraulic gradient $i = 0.01$ and that the injection well is down-gradient of the abstraction well, the hydraulic breakthrough time was calculated as 117 days. The application of the retardation factor (Equation 8.40) gives a thermal breakthrough time of 671 days (1.84 years).

BOX 8.5 Volumetric Heat Capacity of Aquifers

Let us consider an aquifer of saturated quartz sand, with a total porosity of 23% (n = 0.23). If quartz has a specific heat capacity of 740 J kg^{-1} K^{-1} (Ward, 1992) and a density of 2620 kg m^{-3}, we can estimate the volumetric heat capacity of quartz as 1.9 MJ m^{-3} K^{-1} (De Marsily, 1986). The volumetric heat capacity of the quartz sand aquifer is then given by

$$\begin{aligned} S_{\mathrm{VCaq}} &= nS_{\mathrm{VCwat}} + (1-n)S_{\mathrm{VCmat}} \\ &= (0.77 \times 1.9\ \mathrm{MJ\ m^{-3}\ K^{-1}}) + (0.23 \times 4.18\ \mathrm{MJ\ m^{-3}\ K^{-1}}) = 2.4\ \mathrm{MJ\ m^{-3}\ K^{-1}} \end{aligned}$$

where S_{VCmat} is the volumetric heat capacity of the aquifer mineral matrix. If the sandstone is dry, we can substitute the volumetric heat capacity of air (0.0012 MJ m^{-3} K^{-1} – Eskilson *et al.*, 2000) for that of water. This yields a volumetric heat capacity of the dry sand of 0.77 × 1.9 MJ m^{-3} K^{-1} = 1.5 MJ m^{-3} K^{-1}.

8.12 The importance of three-dimensionality

Our consideration of well-doublet schemes in Sections 8.10 and 8.11 has assumed two-dimensionality in terms of both groundwater flow and heat transport. As regards groundwater, this assumption is often (but not always) acceptable – sedimentary aquifers are often approximately horizontal and groundwater flow often parallels the ground surface. Regarding heat transport, the assumption is more dubious, because heat can migrate, by conduction, into the overlying and underlying strata. We have already seen (Section 8.11) that heat is absorbed from groundwater into the mineral matrix of the aquifer, and this has the effect of retarding the heat signal. If heat is absorbed into the overlying and underlying strata, this will have the effect of retarding and attenuating the heat signal still further.

Thus, if we use two-dimensional analytical models, of the type described by Equations 8.34–8.40, we risk being overly conservative. Indeed, transport of heat back to the abstraction well (thermal breakthrough) and transport of a heat plume down the regional hydraulic gradient may well be much slower than these equations appear to predict. Carbon Zero Consulting (2010) performed some generic simulations of well-doublets using two-dimensional analytical and numerical (finite element) models and found very little discrepancy. Their three-dimensional numerical models predicted significantly longer thermal breakthrough times within the doublet and significantly shorter down-gradient thermal plumes, purely due to conductive heat transfer into overlying and underlying strata (Figure 8.13). Indeed, typical thermal plume lengths were up to three times longer in two-dimensional models than in three-dimensional ones. They found, moreover, that where the aquifers were very shallow, heat could migrate away from the aquifer, through the ground surface into the atmosphere – restricting the migration of heat still further.

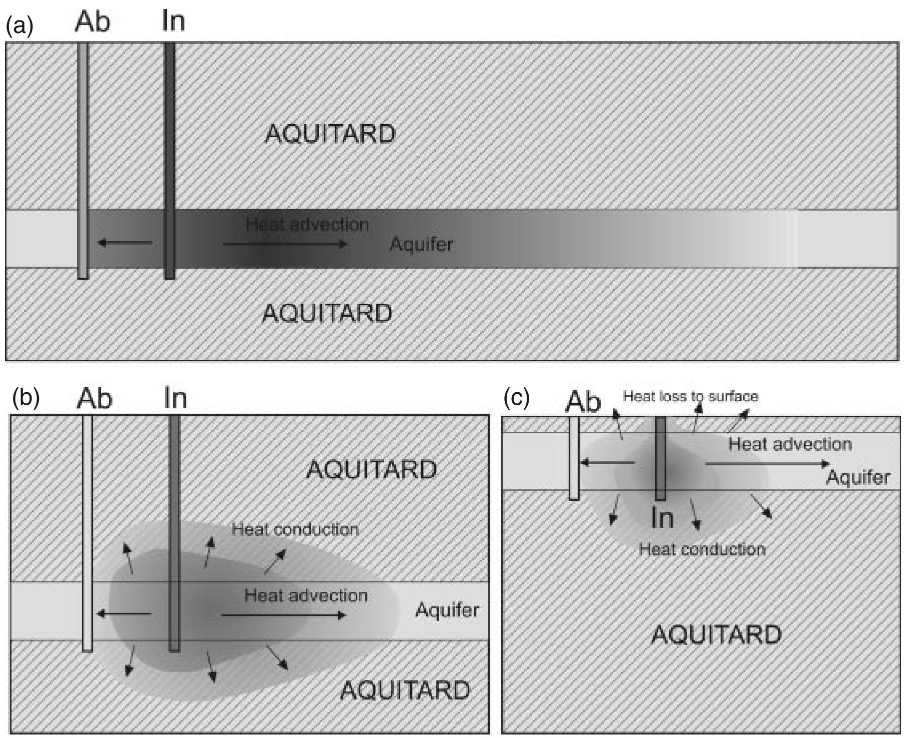


Figure 8.13 (a) The migration of heat with groundwater flow from a well doublet in a two-dimensional aquifer. The two-dimensionality means that heat cannot be conducted into over- and underlying strata. Heat is constrained to the aquifer and thus migrates relatively far and fast. (b) This figure illustrates the effect of three-dimensionality in restricting heat migration, while (c) shows that, with shallow aquifers, the surface can act as a heat sink (or heat source in ground source heating schemes). These diagrams illustrate the conclusions of work by Carbon Zero Consulting (2010). Ab is the abstraction well and In is the re-injection well. The natural groundwater head gradient is from left to right.

It is worth noting, in passing, that Lippmann and Tsang (1980) and Clyde and Madabhushi (1983) provided a formula for predicting how the temperature of the abstracted water (θ_{gout}) from a well will evolve, following thermal breakthrough (for times t greater than the breakthrough time t_{the}):

$$\frac{\theta_{\text{gout}} - \theta_{\text{ginj}}}{\theta_0 - \theta_{\text{ginj}}} = 0.34 \exp\left(-0.0023\frac{t}{t_{\text{the}}}\right) + 0.34 \exp\left(-0.109\frac{t}{t_{\text{the}}}\right) + 1.37 \exp\left(-1.33\frac{t}{t_{\text{the}}}\right) \tag{8.42}$$

where θ_{ginj} is the temperature of the injected water (which is assumed to be constant, although in a real operation, it may increase as the temperature of the abstracted water increases), θ_0 is the initial ambient groundwater temperature, t is time following commencement of abstraction/injection and t_{the} is the thermal breakthrough time.

This equation predicts a substantially slower temperature evolution than does the two-dimensional geometric algorithm (Figure 8.12a) of Güven *et al*. (1986), for the reason that it does take some account of three-dimensional heat migration. It does make some fairly specific assumptions regarding three-dimensionality, however, and its generic applicability is likely to be limited.

8.12.1 A real example from Manitoba, Canada

Ferguson and Woodbury (2005) provided one of very few documented case studies of the breakthrough of hot water from an open-loop doublet system into an abstraction well, at Winnipeg, Canada. The system is a pseudo-'doublet' comprising two closely spaced abstraction wells in a limestone aquifer and a single injection well some 90 m away. The transmissivity of the aquifer is very high and estimated as around 620 m^2 day^{-1}, while the depths of the wells are in the range of 68–90 m. The system was operated at a maximum rate of 13 L s^{-1} for 8 h every working day, giving a long-term average of 3.9 L s^{-1}. Such a fissured (probably karstified), high transmissivity aquifer, with a low effective porosity (n_e) and short doublet spacing is clearly a recipe for rapid feedback of water and heat breakthrough. Indeed, plugging these values into Equation 8.32 (setting n_e arbitrarily at 0.01) yields a hydraulic breakthrough time of around 3 weeks. A thermal breakthrough of around 3 years is calculated, but for reasons discussed in Section 8.11.1, this is likely to be significantly overestimated in fissured limestone aquifers. In reality, thermal breakthrough occurred with very little delay (less than a few months), and over the course of the subsequent 8 years, the temperature of the abstracted water rose by almost 6°C (Figure 8.12b), presumably compromising the scheme's long-term viability.

8.12.2 Multiple well doublets

We have considered the situation where our heating and cooling requirement comprises a single well doublet. If our requirement is very large, we may require more than one well doublet – but we may have only limited space. It can be demonstrated that there is an optimal method for placing abstraction and re-injection wells to make maximum use of the thermal storage that the aquifer represents (and thus to maximise the lifetime of the operation). The solution proposed by Gringarten (1978) is the chequerboard arrangement shown in Figure 8.14.

8.13 Mathematical reversibility

The examples considered in Sections 8.10–8.12 have concerned a well-doublet cooling scheme, where waste heat is re-injected back to the aquifer. Fortunately, the mathematics is essentially reversible, so that the same equations can be used for the migration of a thermal signal from a ground source heating scheme, where cold water is

Figure 8.14 Gringarten (1978) argued that, where multiple abstraction and injection wells are necessary, the most efficient pattern is an alternating grid of abstraction (Ab) and injection (In) wells – a so-called five-spot pattern.

injected to the aquifer. The migration of the plume of cool water will be retarded in the same way as a plume of warm water, as heat is released from the aquifer matrix, and from the over- and underlying strata.

8.14 Sustainability: thermally balanced systems and seasonal reversal

In our theoretical example of Section 8.11.2, we calculated a potential thermal breakthrough of warm water in the abstraction well after less than 2 years. The predicted temperature in the abstraction well may exceed 12°C after 2 years but may take 9 years of continuous operation to reach 14°C (Figure 8.12a). Thus, the potential for thermal breakthrough does *not* mean that the system fails immediately (and remember that our example failed to take any account of three-dimensional heat losses), although its lifetime may ultimately be limited. In such assessments, we should remember the following:

- We can significantly decrease the risk, speed and magnitude of thermal breakthrough by increasing the separation (L) of the wells. The two-dimensional breakthrough time increases with L^2.
- We should also remember that, in thick aquifers, we can also separate the abstraction and re-injection operations vertically. We could place the abstraction well screen (for example) in the interval 30–70 m and the re-injection well screen at

100–140 m depth. If more than one aquifer is present beneath a site, we could abstract from one aquifer and re-inject to the second, effectively removing the risk of thermal breakthrough. The environmental regulator may object to this on water resources grounds, however, and there may be issues with compatibility of groundwater chemistry.

- We may not be pumping the doublet continuously at a given rate Z. There may be seasons of the year where the scheme is dormant.
- We may have a so-called *thermally balanced* scheme, where the amount of heat rejected during the summer is approximately equal to that abstracted during winter. In such a case, we are injecting successive warm and cold pulses of water into the aquifer at (approximately) six monthly intervals. If the separation of the abstraction and injection wells is large enough and the thermal breakthrough time long enough (i.e. greater than several months), it is likely that the thermal storage properties of the aquifer will 'smear' out these temperature pulses during their passage through the aquifer, resulting in little net change in temperature at the abstraction well.

Even better, in a thermally balanced scheme, we may be able to reverse the polarity of our well doublet. Thus, we may abstract cold water from our first well and reject warm water to a second injection well in the summer. In winter, we can convert the second well to a pumping well to re-abstract the warm water. After passage through a heat pump system, the chilled water is then rejected to the first well. The following summer, the polarity is reversed again. It will be obvious that such balanced schemes should be very energetically efficient (we are extracting warmth from preheated groundwater and coolth from pre-cooled water) and indefinitely sustainable. Such schemes are a form of underground thermal energy storage (UTES) called *aquifer thermal energy storage* (ATES). We have encountered such a scheme in Box 3.8 and will meet more examples in Chapter 14.

8.15 Groundwater modelling

Naturally, with larger schemes, things soon get complicated. The magnitude of our scheme may require us to drill multiple injection and abstraction wells (Figure 8.14). Our aquifer may not be homogeneous or isotropic. Our heating and cooling loads may be variable and complex, as may our re-injection rates and temperatures. There may be other groundwater abstractors nearby that perturb the flow dynamics of our scheme. Furthermore, it may not be possible to neatly align our abstraction well directly up-gradient of our injection well. To tackle problems of greater complexity, it may be necessary to invoke a numerical computer model that can simulate coupled groundwater flow and heat transport, using finite element or finite difference algorithms based on the equations of Box 8.3. As heat transport in groundwater is directly analogous to contaminant transport (Box 8.3), it may be possible to utilise (with careful thought) models that are designed to simulate groundwater pollution. There are, however, several models that are developed with the explicit capability to simulate heat transport in aquifers. These currently include (but are not limited to) the following:

- HST3D (Heat and Solute Transport in 3-Dimensional Groundwater Flow Systems). This is a finite difference code in the public domain for simulation not only of groundwater flow and heat transport, but also of contaminant transport, produced by the United States Geological Survey (Kipp, 1997).
- SHEMAT (Simulator for HEat and MAss Transport) is a German programme with similar functionality to HST3D (Clauser, 2003).
- FEFLOW® (Finite Element subsurface FLOW system) is a commercial (and expensive) finite element programme, with a powerful capability to simulate variable-density groundwater flow problems, coupled with heat or solute transport.

8.16 Examples of open-loop heating/cooling schemes

We have already encountered Sweden's Arlanda Airport ATES scheme in Box 3.8, and will meet the Norwegian equivalent, at Oslo Gardermoen Airport, in Box 14.2. A simple open-loop scheme, at the Hebburn Eco-Centre, has briefly been described in Box 4.4. For those who are interested, the BGS (Abesser, 2010) has recently published a review of the performance of open-loop ground source heating and cooling schemes in the United Kingdom. Here follow a few further selected examples of British open-loop schemes, from published literature.

8.16.1 Groundwater-based industrial cooling in Selby, Yorkshire

Todd and Banks (2009) described a well doublet that provides groundwater cooling to a food processing factory in Selby, Yorkshire. The two wells are separated by around 170 m and are about 70 m deep into the Sherwood Sandstone aquifer, which is confined by a sequence of Quaternary silts and clays. The ambient groundwater temperature is around 11°C and groundwater is abstracted at a rate of up to $12\,L\,s^{-1}$. The water is permitted to be re-injected at up to 20°C, although temperatures of 17°C are more typical. No heat pump is used; heat is merely rejected from the industrial process to the groundwater via passive heat exchange. The cooling capacity is thus up to around 450 kW. The photos in Figure 8.6 show the abstraction well and the sealed well head where the water is re-injected under pressure. The scheme has operated since around 1998, apparently without problems, despite relatively high iron concentrations in the confined Sherwood Sandstone of Yorkshire (note the sealed well head and lack of access to oxygen). The migration of any thermal plume of warm groundwater down-gradient from the re-injection well is monitored by the Environment Agency. Todd and Banks (2009) applied the two-dimensional well-doublet equations described in Sections 8.10 and 8.11. They also used the two-dimensional computer model SHEMAT (see Section 8.15) and predicted that thermal breakthrough should have occurred within 3 years at the abstraction well and that the down-gradient thermal plume should have arrived at the nearest observation well, around 150 m down-gradient, within c. 10 years.

In 2008, the abstracted water temperature was found to be around 12.1°C, suggesting that a very limited degree of thermal breakthrough may have occurred (though in no way jeopardising the sustainability of the scheme), while no increase in temperature was found at the observation well. The fact that the measured thermal impacts were somewhat less significant than those predicted by two-dimensional analytical equations and numerical models seems to confirm that three-dimensional heat migration is important and can have a substantial mitigating effect.

8.16.2 Open-loop schemes based on mine water in Scotland

Banks *et al*. (2009b) described two modestly sized (c. 65 kW) ground source heat pump schemes in Scotland, supplying space heating and domestic hot water to apartment blocks. In both cases, boreholes were drilled to abandoned, flooded coal mines. The mine water is pumped directly through the evaporators of the heat pumps and re-injected at a higher stratigraphic level via boreholes located a short distance away. In one scheme, at Shettleston, Glasgow, the scheme has operated without apparent problem, for over 10 years. In some ways, this is surprising, as such mine waters are typically hydrochemically awkward, containing high levels of iron and manganese. However, the fact that the pumped mine water is not allowed to come into contact with oxygen and is re-injected below the groundwater level within the re-injection well appears to have prevented the oxidation and precipitation of iron and manganese minerals. The elements seem to remain dissolved in the mine water. At the second scheme, at Lumphinnans in Fife, a similar system seemed to have operated acceptably until a vandalism incident occurred in which the re-injection main was damaged. Following this incident, the thermally spent water was simply allowed to cascade down into the re-injection borehole. It was thus exposed to oxygen; iron was able to oxidise and precipitate, clogging the recharge well with an ochreous mass. Banks *et al*. (2009b) demonstrated the use of hydrochemical modelling to illustrate the chemical processes occurring.

8.17 Further reading

In a single chapter, you have been subjected to a crash course in some aspects of quantitative hydrogeology. If you desire more information, there are several, less quantitative introductory books on hydrogeology that I can recommend:

- Mike Price's (1996) concise introductory volume *Introducing Groundwater*, focussing to some extent on British geological conditions.

- Paul Younger's (2006b) recent *Groundwater and the Environment* – blessedly equation-free.
- C.W. Fetter's (2001) established *Applied Hydrogeology* – gives you the equations, but in a painless way!
- Rick Brassington's (2006) *Field Hydrogeology* for groundwater practitioners or Kevin Hiscock's (2004) university-level textbook.

For groundwater engineering, try the following:

- *Water Wells and Boreholes* by Bruce Misstear, Dave Banks and the late Lewis Clark (2006). This is an updated and expanded version of Clark's (1988) *Field Guide*.
- *Groundwater and Wells* (1986) by Fletcher D. Driscoll – groundwater engineering from an American perspective.

STUDY QUESTIONS

8.1 A well of diameter 180 mm (radius = 0.09 m) penetrates an aquifer of transmissivity 900 m^2 day^{-1}. It is pumped continuously for 3 weeks at a rate of 25 L s^{-1}. The storage coefficient of the confined aquifer is around 0.002. What drawdown do you predict at the end of the pumping period?

8.2 Carrying on from question 8.1, the water is re-injected back to the aquifer in a similar injection well, located 88 m away. What eventual steady-state drawdown do you predict in the abstraction well? Repeat the calculation for a separation of 150 m.

8.3 The aquifer, in questions 8.1 and 8.2 is around 80 m thick, and the regional groundwater head gradient is 0.01 (1 m in 100 m). An inert chemical tracer is added to the re-injected water, and the re-injected water is heated up by 10°C. How long would it take (1) the tracer and (2) the heat from the re-injection well to reach a monitoring well located 150 m immediately down-gradient? The volumetric heat capacity of the aquifer is 2.3 MJ m^{-3} K^{-1} and the effective porosity is 8%.

9

Pipes, Pumps and the Hydraulics of Closed-Loop Systems

It is by teaching that we teach ourselves, by relating that we observe, by affirming that we examine, by showing that we look, by writing that we think, by pumping that we draw water into the well.

Henri Frédéric Amiel (1853)

The society which scorns excellence in plumbing as a humble activity and tolerates shoddiness in philosophy because it is an exalted activity will have neither good plumbing nor good philosophy: neither its pipes nor its theories will hold water.

John W. Gardner (1961)

We have already encountered *indirect circulation closed-loop heat pump systems* in Chapter 7, and we will consider them in more detail in Chapters 10 and 11. In such systems, we extract heat from the ground to a heat pump using an underground heat exchanger (typically a closed loop of polyethene pipe) buried in a borehole or trench. We circulate a heat exchange fluid – a *carrier fluid* – through this pipe. The ground loop of a closed-loop system will typically comprise one or more hydraulic circuits connected, in parallel, to a manifold. From the manifold, header pipes lead back to the heat pump or heat exchanger. Each parallel circuit represents one (or, sometimes, more) closed-loop borehole, horizontal pipe or slinky trench (see Figure 9.1).

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

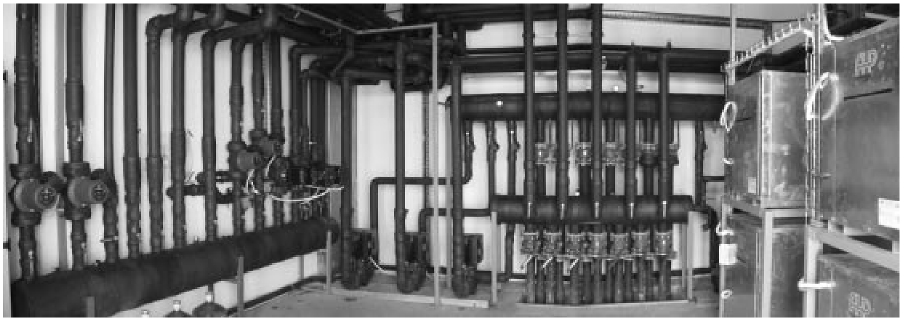


Figure 9.1 A ground source heating and cooling plant room at a car showroom in Bucharest, Romania, showing manifolds, fluid circulation pumps (left) and an array of heat pumps (right). *Reproduced by kind permission of ASA Geoexchange SRL, Bucharest.*

The operator of the closed-loop system needs to pump a *carrier fluid* around the ground loop. The operator will want to minimise the expenditure of energy on pumping, as this 'parasitic power' usage will detract from the energy efficiency of the system. As a very general rule, we should be concerned if the electrical (or other) energy consumed by circulating carrier fluids around the ground loop and any building loop exceeds 10% of the electrical power consumed by the heat pump compressor. To understand why, let's do a brief calculation. Let us assume that our 20-kW heat pump is operating with a COP_H of 3.5. The electrical energy consumed by the heat pump will be 5.71 kW. If our circulation pumps consume 10% of this (571 W), the effective system performance will now be

$$\frac{20\text{ kW}}{5.71\text{ kW} + 0.57\text{ kW}} = 3.18 \qquad (9.1)$$

instead of 3.5.

One can, of course, argue that a proportion of this pumping energy will be converted to usable heat in the ground loop, improving the situation somewhat, but we are still potentially looking at up to 9% decline in efficiency.

We need to expend energy pumping the carrier fluid around the closed loop in order to overcome hydraulic resistances in the hydraulic system. These resistances arise from the following:

- internal friction in the fluid, which will be related to the fluid's viscosity and factors such as turbulence (Box 9.1);
- friction and drag against the walls of pipe sections;
- the resistance of the heat exchanger of the heat pump;
- the resistance caused by bends (angles) in the ground loop;
- the resistance caused by constrictions in the ground loop (valves, flow meters, manifolds, other fittings);
- the resistance caused by pockets of air trapped in the ground loop.

BOX 9.1 Dynamic and Kinematic Viscosity

Viscosity is a measure of how 'thick' a fluid is. Strictly speaking, it describes the fluid's resistance to deformation under stress. Crude oil is viscous, alcohol less so and water has relatively low viscosity. Viscosity tends to decrease with increasing temperature. There are two common measures of viscosity – *kinematic viscosity* and *dynamic viscosity*.

Dynamic viscosity (μ) is also known as absolute viscosity and is often measured in centiPoise (cP). It has the dimensions $[M][L]^{-1}[T]^{-1}$:

$$1\ \text{Poise} = 1\ \text{P} = 0.1\ \text{Pa s} = 0.1\ \text{N m}^{-2}\ \text{s} = 0.1\ \text{kg m}^{-1}\ \text{s}^{-1}$$

$$1\ \text{cP} = 0.001\ \text{Pa s} = 0.001\ \text{kg m}^{-1}\ \text{s}^{-1}$$

Kinematic viscosity (ν) is simply dynamic viscosity divided by fluid density. It has the dimensionality $[L]^2[T]^{-1}$ and is often measured in Stokes (St):

$$\nu = \frac{\mu}{\rho}$$

$$1\ \text{St} = 10^{-4}\ \text{m}^2\ \text{s}^{-1}$$

Each of these hydraulic resistances represents a pressure drop, which must be overcome by the pump (Box 9.2). The bigger the pressure loss, the larger the pump that will be required to circulate fluid and the greater the parasitic pumping power. For pure water, we often talk of pressure losses in terms of metres of water 'head' (Δh, see Box 1.2). However, as the fluids we are considering in closed-loop systems are often solutions of antifreeze, we should talk about pressure losses (ΔP) in pascals (Pa) or kilopascals:

$$\Delta h = \frac{\Delta P}{g\rho_{cf}} \tag{9.2}$$

where g is the acceleration due to gravity = 9.81 m s^{-2} and ρ_{cf} = the density of the carrier fluid in question (kg m^{-3}). 10 kPa is equivalent to 1.02-m head loss when considering pure water. For other fluids, the equivalence depends on the fluid density. For an antifreeze solution of density 1052 kg m^{-3}, 10 kPa is equivalent to 0.97 m of fluid 'head' or a 0.97-m column of the fluid in question.

Remember: 1 bar = 100 kPa = 100 000 N m^{-2} = approximately 10 m of water head.

We can calculate the theoretical energy (E_{pump} in watts) required by a 100% efficient pump to overcome a given pressure ΔP (Pa), when moving a quantity of water F (m^3 s^{-1}):

BOX 9.2 The Darcy–Weisbach Equation

The Darcy–Weisbach equation is a general equation for the pressure drop (ΔP in pascals) along a length L of pipe, resulting from the steady-state flow (F) of an incompressible liquid in a hydraulically smooth circular pipe of radius r:

$$\Delta P = \frac{\rho F^2 L}{4\pi^2 r^5} C_{\mathrm{DW}}$$

where C_{DW} is the (dimensionless) Darcy–Weisbach constant, which depends on fluid turbulence (and thus on viscosity) and pipe roughness. ρ is the fluid density.

For laminar flow (Equation 9.5), $C_{\mathrm{DW}} = 64/Re$, where Re = Reynolds Number.

For turbulent flow, one can use approximate solutions such as

$$\text{the Blasius equation for turbulent flow in smooth pipes } C_{\mathrm{DW}} = \frac{0.3164}{\sqrt[4]{Re}}$$

or the Petukhov equation for (transient-) turbulent flow in smooth pipes:

$$C_{\mathrm{DW}} = \frac{1}{(0.79 \ln Re - 1.64)^2} \text{ (Acuña and Palm, 2008)}$$

$$E_{\mathrm{pump}} = F\Delta P \tag{9.3}$$

Thus, if our ground loop has a pressure loss of 70 kPa at a design carrier fluid flow rate (F) of 0.7 L s^{-1}, then the minimum pumping energy required is

$$E_{\mathrm{pump}} = 0.0007 \text{ m}^3 \text{ s}^{-1} \times 70\,000 \text{ N m}^{-2} = 49 \text{ N m s}^{-1} = 49 \text{ J s}^{-1} = 49 \text{ W}$$

Of course, no pump is 100% efficient at converting electrical power into mechanical (pumping) power, so we may find that we actually need a 100-W electrically rated pump to satisfy our requirements.

9.1 Our overall objective

Our challenge now becomes to design a ground loop which

- satisfies any geological or other engineering criteria, which constrain its depth or length.

BOX 9.3 Turbulent Flow and Reynolds Number

The Reynolds Number is a dimensionless parameter that describes how turbulent the flow of a fluid is. It was invented by the physicist George Stokes (1819–1903), but later 'popularised' by the fluid dynamicist Osborne Reynolds (1842–1912).

For a circular pipe of hydraulic radius r, the Reynolds Number is calculated as

$$Re = \frac{2r\rho F}{\mu A} = \frac{2\rho F}{\pi r \mu} = \frac{2rF}{\nu A} = \frac{2F}{\pi r \nu}$$

(the last term is true provided that the hydraulic radius is similar to the real radius), where F is the fluid flow rate, A is the cross-sectional area, μ and ν are the dynamic and kinematic viscosities (Box 9.1) and ρ is the fluid density.

In a circular pipe, the fluid flow is laminar below a Reynolds Number of around 2300. Somewhere between $Re = 2300$ and 2500, the fluid flow begins to acquire a tendency to be turbulent. Between $Re = 2500$ and 4000, the flow regime is described as 'transitional turbulent'. Above $Re = 4000$, the flow becomes fully turbulent.

In order to obtain reasonably efficient heat transfer in a circular subsurface closed-loop heat exchanger, we usually aim for a Reynolds Number in excess of 2500 and ideally around 3000. If the Reynolds Number becomes too big, hydraulic pressure losses may become excessive.

- has sufficient length to provide an adequate heat exchange capacity with the ground for our peak load and base load scenarios. We will see how to do this in Chapters 10 and 11.
- has a high heat transfer efficiency [a low thermal resistance (R_b) – see Chapter 10] in its underground portions, especially at times of peak load. It turns out that turbulent fluid flow in the pipe is highly beneficial for heat transfer, as it ensures that the fluid comes into intimate contact with the pipe walls. Thus, we usually try to attain transient-turbulent conditions in the heat exchange pipe at times of peak load (in practice, this will usually mean a Reynolds Number of at least 2500 or 3000 – see Box 9.3).
- has an acceptably low hydraulic resistance to minimise parasitic pumping power.

Note that these last two criteria tend to work against each other. Transient-turbulent flow implies good heat transfer efficiency, but poor hydraulic performance.

9.2 Hydraulic resistance of the heat exchanger

The heat exchanger of the heat pump (i.e. the evaporator, in heating mode) will have a significant hydraulic resistance, which will depend on flow rate of carrier fluid through it. The heat pump manual will also usually specify a minimum carrier fluid flow rate, which is necessary to prevent the evaporator from freezing and/or a nominal design fluid flow rate. The manual should also specify the pressure loss that occurs across the evaporator at that fluid flow rate. It may also provide a curve showing how the pressure loss increases as the fluid flow rate through the evaporator increases (it is usually a non-linear relationship, such that pressure loss increases disproportionately to fluid flow). Thus, while we can specify a fluid flow through most heat pumps that exceeds the nominal design flow rate, we should be careful about doing so as we risk incurring excessive pumping energy costs. Heat pump manuals will typically suggest an optimal carrier fluid flow rate through the heat exchanger, which is fast enough to protect against risk of freezing and low enough not to incur large hydraulic penalties. It turns out that this optimum rate is often around 3–3.5 $\mathrm{L\,min^{-1}}$ per kilowatt of heat pump output (in heating mode).

The temperature drop ($\Delta\theta$) of the carrier fluid across the heat pump evaporator is given by

$$\Delta\theta = H[1-(1/\mathrm{COP}_H)]/(S_{\mathrm{VCcar}}F) \tag{9.4}$$

where F is the carrier fluid flow rate, S_{VCcar} is its volumetric heat capacity and H is the heat pump output. If we assume a COP_H of 3.5, then 1-kW heat output implies 0.714-kW heat extraction from the carrier fluid. If a typical carrier fluid has a volumetric heat capacity of around 3.9 $\mathrm{kJ\,L^{-1}\,K^{-1}}$ at c. 0°C, then a flow rate of 3.3 $\mathrm{L\,min^{-1}}$ per kilowatt (0.055 $\mathrm{L\,s^{-1}}$) implies a typical temperature drop across the evaporator of 3.3°C.

The hydraulic losses across the heat pump heat exchanger are, to a large extent, heat pump specific and can be looked up in the operator's manual. We will thus forget about them for the time being and focus on the remainder of the ground loop.

9.3 The hydraulic resistance of pipes

The pressure drop that will occur during fluid flow within a straight circular pipe will depend on several factors:

- the roughness of the inside walls of the pipe (which will depend on the pipe material and construction quality);
- the fluid's viscosity and density, both of which will vary with the type of antifreeze, the concentration of antifreeze and the temperature;
- the nature of fluid flow (i.e. whether it is turbulent or laminar), as characterised by the Reynolds Number (see Box 9.3);
- the pipe diameter.

In general, the following factors improve (decrease) hydraulic resistance:

- smooth pipe;
- low fluid viscosity (i.e. high temperature and low concentrations of antifreeze);

- low Reynolds Number (but, remember, we will normally require a Reynolds Number of at least 2500 during peak load, to ensure transient-turbulent flow, efficient heat transfer and a low borehole/pipe thermal resistance);
- large pipe diameter.

We will immediately see that there is a potential problem with ground source heating systems that operate below 0°C. We need to add an antifreeze solution, which in itself increases fluid viscosity. On top of this, the low temperatures also increase viscosity. Higher-temperature (cooling-dominated) systems, especially in warmer climates, do not face the same challenges with hydraulic efficiency, as viscosities are usually lower, and antifreeze may not be needed if the systems are designed so that average fluid temperatures do not drop below, say +5°C.

The pressure drop (ΔP in pascals) during the flow of a liquid in a hydraulically smooth circular pipe is typically given by the following formulae, which are variants of the Darcy–Weisbach equation (Box 9.2):

$$\Delta P = \frac{16\rho F^2 L}{\mathrm{Re}\,\pi^2 r^5} \text{ for laminar flow (i.e. } Re < 2300\text{)} \left(\text{i.e. } C_{\mathrm{DW}} = \frac{64}{Re}\right) \tag{9.5}$$

$$\Delta P = \frac{0.0791\rho F^2 L}{\sqrt[4]{\mathrm{Re}}\,\pi^2 r^5} \text{ Blasius approximation for turbulent flow in smooth pipe}$$

$$\left(\text{i.e. } C_{\mathrm{DW}} = \frac{0.3164}{\sqrt[4]{Re}}\right) \tag{9.6}$$

where

- r is the pipe's hydraulic radius (m);
- F is the fluid flow rate ($m^3\,s^{-1}$);
- ρ is the fluid density ($kg\,m^{-3}$);
- L is the pipe length (m);
- Re is the Reynolds Number (which depends on density and viscosity – see Box 9.3).

In most ground source heat systems, we can probably assume that factory-new polyethene pipe is close to being hydraulically smooth: hence, pipe roughness does not feature in the above equations.

Note that the Blasius equation is only an approximation, but it is a reasonably simple explicit equation for turbulent flow in a hydraulically smooth pipe. Alternative, more complicated explicit solutions and better implicit solutions do exist, however, for turbulent flow in pipes of varying roughness: the reader is encouraged to explore the Prandtl/Colebrook equation for turbulent flow or the graphical method of the Moody diagram. In many ground source heating schemes, we will probably be dealing with transient-turbulent flow ($2500 < Re < 3500$), and the pressure loss will be intermediate between the solutions given by Equations 9.5 and 9.6.

The reader should note that Equations 9.5 and 9.6 only give the hydraulic losses in straight pipe sections, such as the shanks of an underground U-tube heat exchanger, or the header pipes connecting the ground heat exchangers to the heat pump. Every bend, manifold and valve will cause an additional hydraulic resistance in the circuit, which must be considered. This textbook will not consider these in detail, but you can find out more by looking at standard fluid engineering texts, by looking at manuals associated with hydraulic fittings or by using pipe network simulation models, which will allow you to calculate the total hydraulic resistance of all the components of a ground heat exchange scheme.

9.4 Acceptable hydraulic losses

This book has been at pains to caution against excessive reliance on rules of thumb, and we would encourage all designers to perform site-specific calculations of pressure losses in ground-loop systems. However, it is handy to have some idea of what excessive and acceptable pressure losses look like. Thus, if we ignore hydraulic losses in the heat pump, we can say that, usually, pressure losses greater than 40–60 kPa (c. 4–6 m of hydraulic head) in the headers and ground loop would often be regarded as excessive. Of these 40–60 kPa, we could maybe ascribe half of the losses to various fittings, manifolds and header pipes and half (say, 20–30 kPa) to the ground heat exchanger itself.

9.5 Hydraulic resistances in series and parallel

Hydraulic circuits perform analogously to electrical circuits in the way that resistances combine with each other:

Remember: Hydraulic resistances in series are additive.

Thus, the total hydraulic resistance of a circuit is equal to the sum of the hydraulic resistances of the heat pump, the fittings, the header pipes and the ground heat exchanger.

Remember: The total hydraulic resistance of several resistances in parallel is given by the reciprocal of the sum of the reciprocals.

$$\frac{1}{R_{\text{total}}} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \qquad (9.7)$$

If all the parallel resistances are equal, then the carrier fluid flow is divided equally among each parallel circuit. The pressure loss in an array comprising several such parallel elements is *simply equal to the pressure loss in one parallel element*.

9.6 An example

We will now try and use these equations in a simple example. Let us imagine that we require a ground heat exchanger comprising five boreholes, connected in parallel, each containing a 90-m-long U-tube of 40-mm outer diameter (OD) SDR 11 polyethene pipe [internal diameter (ID) 32.7 mm, internal radius 16.4 mm]. The hydraulic resistance of each U-tube will be identical and the flow of carrier fluid in each should also be identical. Let us assume that the five boreholes support a heat pump of 30-kW peak heating capacity, with a nominal carrier fluid flow rate (from the heat pump manual) of 100 $\mathrm{L\,min^{-1}}$ or 1.67 $\mathrm{L\,s^{-1}}$. This means that each borehole should carry an equal flow of 20 $\mathrm{L\,min^{-1}}$ or 0.33 $\mathrm{L\,s^{-1}}$. Let us also assume that the carrier fluid has the following properties at c. 0°C (note that the temperature we choose should be representative of the actual operating temperature of the system):

- μ = dynamic viscosity of fluid = 0.004 23 $\mathrm{kg\,s^{-1}\,m^{-1}}$;
- ρ = fluid density = 1046 $\mathrm{kg\,m^{-3}}$.

9.6.1 Hydraulic losses in header pipes

In the above example, our total peak fluid flow rate is 1.67 $\mathrm{L\,s^{-1}}$. Let us assume that this is transferred from the heat pump to the ground loops in polyethene flow and return header pipes each 25 m long (total length = 50 m). Let us assume that we preliminarily select 63-mm OD SDR 11 pipe. This implies an internal radius of 25.8 mm (see Box 9.4).

These parameters yield a rather high Reynolds Number of 10 200, indicating turbulent flow. The Blasius equation then gives a pressure loss in the header pipes of 10.2 kPa. This *may* be acceptable (see Section 9.4) provided that the other losses (fixtures and fittings) and the ground-loop losses are modest.

We could increase the pipe diameter to the next sizes up: 75-mm OD or 90-mm OD. These sizes would yield lower Reynolds Numbers of around 8570 and 7140, respectively, and would reduce the hydraulic losses to 4.5 and 1.9 kPa, respectively, which would almost certainly be acceptable. However, increasing the pipe diameter can have significant cost implications, both in terms of pipe cost and in terms of the additional quantity of antifreeze required to fill the pipe. Again, we return to the quandary of capital expenditure to improve system efficiency and reduce operating cost.

If we were daft enough to select 40-mm OD pipe for our headers, we would end up with a Reynolds Number of 16 100 and a whopping pressure loss of 88 kPa – almost certainly unacceptable.

BOX 9.4 Polyethene Pipe Specification

When specifying the dimensions of polyethene pipe, we commonly come across several key terms. The following terms specify the dimension of the pipe. Standard polyethene pipe dimensions are 20-, 25-, 32-, 40-, 50-, 63-, 75- and 90-mm OD:

OD = outer pipe diameter (usually in millimetres);
ID = interior pipe diameter (usually in millimetres);
SDR = standard dimension ratio. This is the ratio of the OD to the wall thickness. Thus, 32-mm OD SDR 11 pipe will have a wall thickness of 2.9 mm. It will thus have an inner diameter of $(32 - 2 \times 2.9)$ mm = c. 26 mm. Similarly, 40-mm OD SDR 11 pipe has an internal diameter of c. 32.7 mm.

Pipe is also classified according to its nominal pressure rating, using a PN designation. PN10 pipe withstands a nominal 10-bar pressure (excess internal pressure) at around 20°C. Other common ratings include PN6, PN8, PN12.5 and PN16 (6, 8, 12.5 and 16 bars, respectively).

The pressure rating obviously depends on the plastic grade and the wall thickness. It so happens that for the common dimensions used for HDPE (PE100) ground-loop pipe, the SDR 11 designation coincides with PN16 for water (may be cited as PN10 for gas).

Finally, pipes can be classified according to polyethene quality:

HDPE is *high-density polyethene* (density > 941 kg m^{-3}).
HPPE is *high-performance polyethene* and broadly corresponds to PE100. This may also be HDPE.
MDPE is *medium-density polyethene* (density 926–941 kg m^{-3}), typically equivalent to PE80.
PEX is *cross-linked polyethene*, where the polymer molecules are cross-bonded, giving the plastic more thermally resistant properties, tolerating higher temperatures. Usually regarded as a form of HDPE.

The designation PE100 may also be encountered, and it essentially signifies high-density, high-performance polyethene. It can withstand a minimum circumferential tension [or minimum required strength (MRS)] of 10 MPa for 50 years at 20°C. SDR 11 PE80, which can be regarded as MDPE, can, on the other hand, withstand a circumferential stress of 8 MPa for 50 years at 20°C.

According to Brömstrup (2004), the relationship between the minimum circumferential tension (MRS, in megapascals) and the maximum operational pressure (MOP in bars) is given by

(*Continued*)

BOX 9.4 (*Continued*)

$$\text{MOP} = 20\ \text{MRS}/S_F\ (\text{SDR} - 1)$$

where the safety factor coefficient (S_F) should be no less than 1.25 for water.

SDR 11 PE100 pipe is often used as a default for the subsurface portions of ground heat exchangers, with a view to offering a high degree of protection against leakage to groundwater. In some countries, MDPE or PN8 pipe has been used, especially for shallow and horizontal ground loops.

Remember that the pressure ratings of pipes refer to a standard temperature of 20°C (and the ratings incorporate a significant safety margin, although this should not be construed as an invitation to under-design). When designing high-temperature ground heat exchange systems, remember that pressure tolerances will drop and alternative materials may need to be used (PEX or steel).

Remember also that the pressure ratings refer to excess interior pressures. SDR 11 HDPE pipe may be able to tolerate 16 bars excess interior pressure, but it may not necessarily tolerate the same amount of excess external pressure. Be aware of this when backfilling boreholes with high-density grouts – this may generate very high bottom-hole external pressures acting on the ground loop (Eugster, 2011).

9.6.2 *Hydraulic losses in ground loops*

Let us continue with the above example and assume that we have selected single U-tubes of 40-mm OD (ID 32.7 mm, internal radius 16.4 mm). We can calculate the Reynolds Number of the fluid in the U-tube from Box 9.3. We obtain a figure of around $Re = 3200$. This is fine for our purposes – falling squarely within the 'transient-turbulent' flow regime required for efficient heat transfer and thus minimising R_b, the borehole thermal resistance.

Let us, for the moment, ignore the hydraulic resistance of the U-piece (which will *not* be negligible) and focus on the resistance of the two straight 90-m sections of pipe (total $L = 180$ m). We can now use the Blasius Equation 9.6 to estimate the pressure loss in the straight-line portions of the U-tube by setting $F = 0.33\ \text{L s}^{-1}$. We calculate a pressure drop of around 19 kPa (equivalent to around 1.86 m of fluid head). This is likely to be acceptable (see Section 9.4).

Thus, our first attempt at pipe selection seems reasonable, and we should end up with a thermally efficient and hydraulically efficient system (19-kPa loss in ground loop, plus 10 kPa in the headers). If the pressure losses in bends, manifolds and valves do not exceed 20 kPa, then we should end up with a total pressure loss of <50 kPa.

Let us see what *would* have happened if we had selected a (1) 32-mm OD single U-tube, (2) 32-mm OD double U-tube and (3) 40-mm OD double U-tube (Table 9.1).

Table 9.1 The calculated Reynolds Number (*Re*), pressure loss and minimum hydraulic power requirement to circulate a flow of $100\,L\,min^{-1}$ (= $1.67\,L\,s^{-1}$) carrier fluid through an array of 5 × 90 m boreholes equipped with different types of single and double U-tube. Only pressure losses and power requirements for the straight shanks of the U-tube are considered. Fluid dynamic viscosity = $0.00423\,kg\,s^{-1}\,m^{-1}$, density = $1046\,kg\,m^{-3}$.

	32-mm OD single U	40-mm OD single U	32-mm OD double U	40-mm OD double U
Flow per parallel circuit ($L\,s^{-1}$)	0.33	0.33	0.17	0.17
Re	4010	3210	2000	1600
Pressure loss (kPa)	55	19	11	4.5
Minimum pumping power required (W)	91	32	18	7.5

Thus, we can see that using single 32-mm OD pipe in this system generates unacceptably high hydraulic losses (55 kPa) and a much higher overall pumping energy cost. Using double pipe, on the other hand, produces very low hydraulic losses (laminar flow), but does not produce the transient-turbulent flow required for efficient heat transfer. It is possible, however, that the increased surface area of double U-tubes may compensate for the lack of turbulence, from the point of view of heat transfer efficiency. In other words, laminar flow in double U-tubes *may* yield similar overall borehole thermal resistances to transient-turbulent flow in single U-tubes. Thus, double U-tubes *may* (subject to further assessment) be a viable and hydraulically efficient option in this case, albeit at increased capital expense (more pipe and carrier fluid).

We have, however, demonstrated that, for many ground source heat schemes, with boreholes of depth c. 100 m, 40-mm OD single U-tubes can represent a good compromise between hydraulic efficiency, thermal efficiency and capital cost. Each scheme will be specific, however, and hydraulic performance will be strongly dependent on the type of antifreeze selected and the minimum operating temperatures (which occur at times of peak load, when it is most important to have efficient heat transfer and a transient-turbulent Reynolds Number).

Let us now consider the scenarios where single 32-mm U-tube and double U-tube do represent good solutions for heat exchange pipe.

9.6.3 Use of 32-mm OD pipe

Let us continue with the example considered above (30-kW scheme), but let us now assume that there are mine workings at 80 m depth, which we wish to avoid. Instead of five boreholes to 90 m, we now drill seven boreholes to 65 m (455 m total). The fluid flow per borehole is a little less than $0.24\,L\,s^{-1}$. If we use a 40-mm pipe, we now find that we have a Reynolds Number of <2300 – too low to ensure turbulent flow and efficient heat transfer. If we use a 32-mm OD pipe (26-mm ID), we do get a transient-turbulent Reynolds Number of around 2860, and a pressure loss of 22 kPa in the straight-line portions of the U-tube, which may well be acceptable.

Thus, narrower (e.g. 32-mm OD) U-tubes are most appropriately used in relatively shallow boreholes with a modest heat extraction rate per borehole (i.e. where the total fluid flow is divided between a large number of boreholes, yielding a low flow rate per bore).

9.6.4 Use of double 40-mm OD pipe

It is quite possible to purchase a double (or even triple) U-tube, where the two 'U' elements of the tube are connected in parallel. Such heat exchangers have a high heat exchange area (good), but are tricky to install, slightly more expensive than single U-tubes and require large volumes and flow rates of carrier fluid.

Let us assume that our 30-kW heat pump scheme (requiring 1.67 L s^{-1} of fluid flow) is to be installed in a city centre, at a very constrained plot of land. Let us assume that we only have room for three boreholes, to 150 m (450 m total). Table 9.2 shows the hydraulic calculations for four different types of pipe.

In this case, if we try to squeeze all the flow down a small number of single U-tubes, we end up with very high flow rates per borehole (0.56 L s^{-1} per borehole), very high Reynolds Numbers and extremely high pressure losses and pumping costs. If we use double 32- or 40-mm U-tubes, we still get a transient-turbulent Reynolds Number and significantly reduced pressure losses and overall pumping costs.

Thus, we conclude that double and triple U-tubes are most appropriate where we are dividing our carrier fluid flow among only a small number of boreholes. This will be the case where we have either a (1) very conductive bedrock, necessitating only limited numbers of boreholes, or (2) very deep boreholes (for other reasons, such as constrained land area).

9.6.5 Coupling boreholes in series

It is normal practice for each borehole in an array of indirect circulation closed-loop boreholes to represent a single circuit, connected in parallel with other boreholes via a manifold. There is, however, in principle, no reason why *more than one* borehole

Table 9.2 The calculated Reynolds Number (*Re*), pressure loss and minimum hydraulic power requirement to circulate a flow of 100 L min^{-1} (= 1.67 L s^{-1}) carrier fluid through an array of 3 × 150 m boreholes equipped with different types of single and double U-tube. Only pressure losses and power requirements for the straight shanks of the U-tube are considered. Fluid dynamic viscosity = 0.004 23 kg s^{-1} m^{-1}, density = 1046 kg m^{-3}.

	32-mm OD single U	40-mm OD single U	32-mm OD double U	40-mm OD double U
Flow per parallel circuit (L s^{-1})	0.56	0.56	0.28	0.28
Re	6680	5350	3340	2670
Pressure loss (kPa)	223	77	66	23
Minimum pumping power required (W)	372	129	111	38

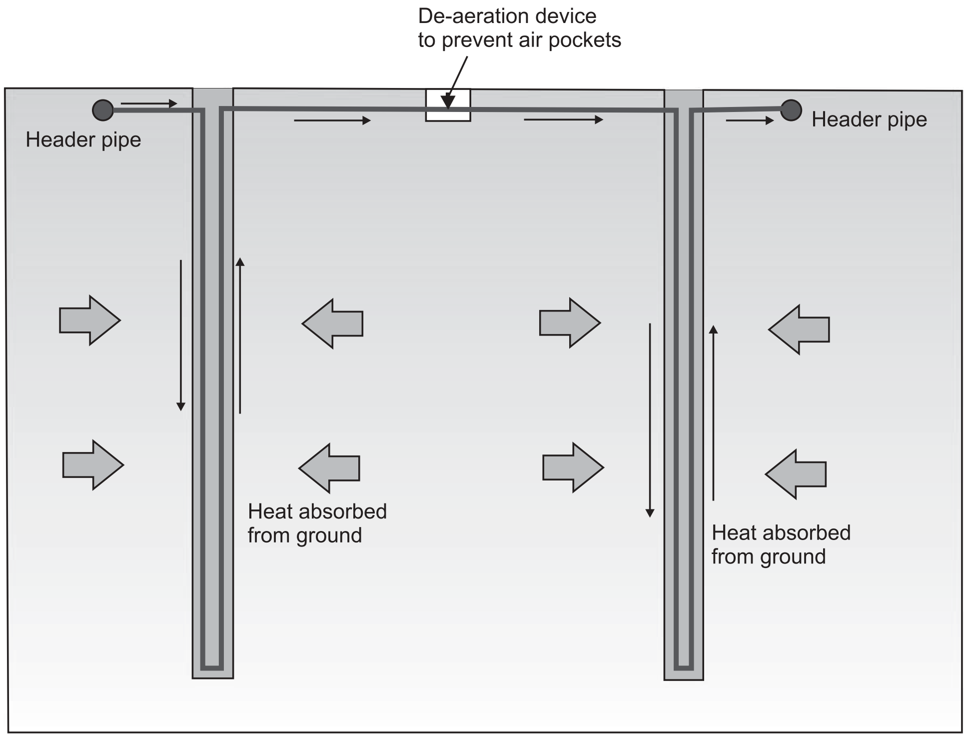


Figure 9.2 Two 50-m-deep boreholes connected in series. A de-aeration valve is fitted at the 'high' point of the carrier fluid circuit to prevent trapped pockets of air obstructing fluid flow.

cannot be connected in series on a single parallel circuit. Figure 9.2 shows two 50-m-deep boreholes connected in series: the total pipe length is very similar to that in a single 100-m borehole and the thermal performance should be very similar. While the fluid circulating through the first borehole will be colder than that in the second borehole (and hence more heat may be extracted from the ground), the average fluid temperature will be similar to the average fluid temperature in a single 100-m bore and the overall heat extraction efficiency will be similar.

One should preferably avoid connecting boreholes in series for several reasons:

- If a leak develops, two boreholes will be 'lost' instead of just one.
- There is a 'high point' in the system, where pockets of air can accumulate and obstruct the fluid flow. A de-aeration/air bleed valve is thus advisable between the boreholes.
- If the boreholes are deep, the length of the fluid circuit, and the hydraulic resistance, can become high.

Connecting boreholes in series may make sense where our array comprises an unusually large number of boreholes, for example, because the boreholes are short (due to geological constraints) or because the ground has an unfavourable thermal conductivity. In this case, there is a danger that, when the flow rate required by the heat pumps is divided among a large number of parallel circuits, the flow per circuit will not be adequate to produce transient-turbulent flow and efficient heat transfer. In that case, we could install two boreholes in series on each parallel circuit, thereby halving the number of parallel circuits required and doubling the amount of flow per circuit. Thus, although the hydraulic resistance per circuit may increase, the number of circuits decreases and the overall hydraulic power required may remain within acceptable limits (although it is especially important to thoroughly evaluate the hydraulic performance of such arrangements).

9.6.6 Horizontal loops

The hydraulic considerations and calculations outlined above are, in principle, exactly the same for parallel circuits comprising horizontally installed pipes within trenches. In particular, an installer should remember that, when working with coiled 'slinky' installations, the lengths of pipe (and thus the hydraulic resistance) can accumulate rapidly. For example, a 30-m trench can incorporate 250–300 m of compact slinky pipe (see Chapter 11).

9.6.7 'Improved' heat exchange pipe

We have seen that the hydraulic design of ground loops involves 'balancing' a number of factors: we want to attain transient-turbulent flow and efficient heat transfer, but we don't want excessively high flow rates or excessive turbulence, as this increases hydraulic resistance and pumping costs. For smooth circular pipes, the Reynolds Number is rather clearly constrained (Box 9.3). However, some pipe manufacturers have experimented with the interior surface of the pipe to try to induce a degree of turbulence or 'swirl' at a *lower* flow rate. One promising design involves a modest degree of 'rifling' on the interior surface of otherwise normal high-density polyethene (HDPE) pipe – this 'rifled' heat exchange pipe is currently on the market and becoming increasingly used in ground heat exchange applications (Acuña and Palm, 2008).

9.7 Selecting pumps

In Table 9.2, we saw that our optimal solution required the use of double 40-mm U-tubes, and we calculated a minimum pumping energy cost (assuming 100% efficient pumping) of 38 W for the straight-line portions of the U-tube. If we assume that the entire hydraulic circuit (including heat exchanger, manifolds, bends and valves) required around 100 W and if the pump unit was 60% efficient, this would translate to an electrical power requirement of up to 170 W. If the 30-kW heat pump had a

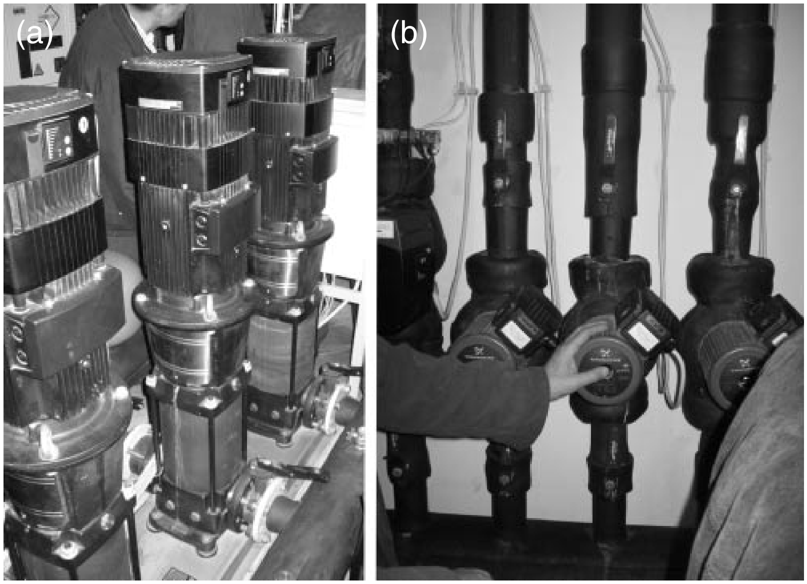


Figure 9.3 (a) Large-scale and (b) smaller carrier fluid circulation pumps. *Photos by David Banks.*

COP_H of 3.6, the electricity used by the compressor would be 8.3 kW. In other words, our pump costs would be low (around 2% of the compressor requirement).

The pumps that are usually (not always) used to circulate carrier fluids are centrifugal pumps, based on a pressure differential created by a rotating impeller (Figure 9.3). They belong to the category of variable displacement pumps, that is, pumps whose discharge (pumping rate) decreases as the operating pressure differential increases.

When we select a pump for a closed-loop circuit, the pump manufacturer will usually provide us with a *pump curve*. This is a set of diagrams (or tabulated data) that relates the pumping head (or achievable pressure differential) to the pumping rate and the power consumption. Such pump curves are directly analogous to the heat pump curves we have encountered in Figures 4.6 and 6.9. Many pump curves provided by manufacturers automatically assume that we will be pumping water. Be careful; pumps will usually perform slightly worse with a denser and more viscous antifreeze solution. We should also remember to convert metres head of water into the equivalent pressure differential.

Figure 9.4 shows a typical pump curve for an electrical centrifugal pump. Let us assume that we wish to pump a carrier fluid of 25% ethylene glycol at a rate of $0.6\,L\,s^{-1}$ ($36\,L\,min^{-1} = 2.16\,m^3\,h^{-1}$) against a predicted pressure drop of 50 kPa. We can see that the pump in Figure 9.4 should be up to the job – the pump curve shows that against 50 kPa, the pump is capable of delivering $2.6\,m^3\,h^{-1}$, consuming a power of 190 W. Note that the theoretical power requirement to pump $2.6\,m^3\,h^{-1}$ ($0.000\,72\,m^3\,s^{-1}$) against 50 kPa is given in Equation 9.3 as 36 W. The overall efficiency of this pump is thus

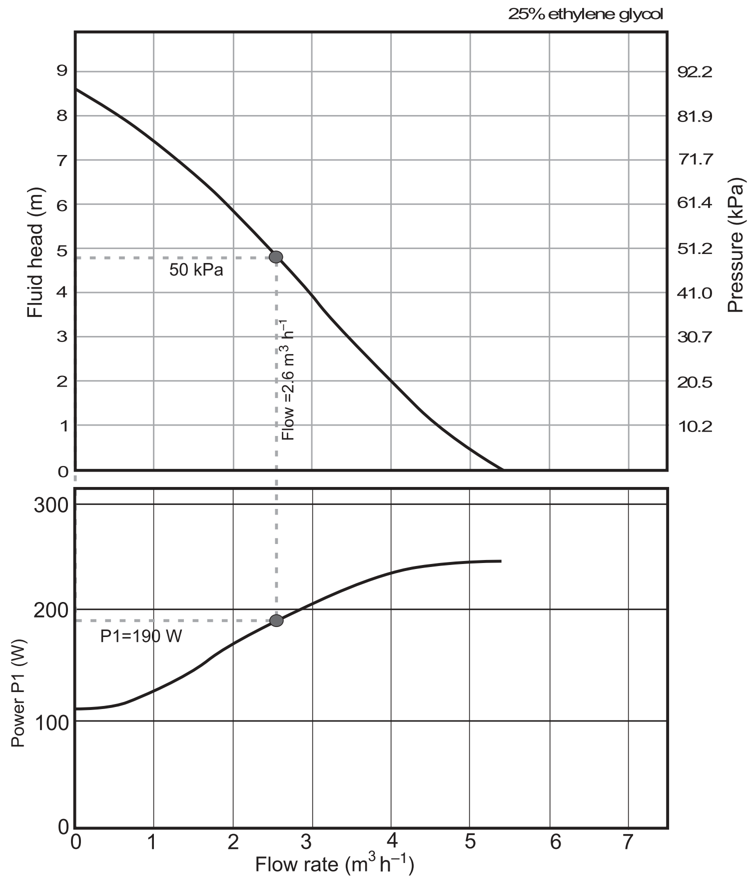


Figure 9.4 An example of a simplified pump curve for a circulation pump using ethylene glycol. Many pumps are variable speed and will have more complicated curves than this.

very low at around 19% and we should maybe look around for an alternative pump. The low efficiency may be due to the fact that the pump is of poor quality or that it is not optimised for pumping antifreeze solutions. It may also be that the proposed operational regime does not coincide with the point of best efficiency of that particular pump.

Remember that, if our heat pump array uses several compressors (or a variable compressor) that cut in and out according to heating/cooling demand, we may wish to also vary the fluid flow rate according to the compressor operation. For this purpose, we may use a number of smaller pumps, or we may use variable speed pumps. The pump curves for such variable speed pumps will, needless to say, be much more complicated than the simplified example shown in Figure 9.4.

Note that heat pump units may be supplied with an integrated circulation pump (maybe two pumps: one for the ground loop and one for the building circuits). These integrated pumps will likely have been selected to provide optimal performance for a particular carrier fluid and a particular ground-loop hydraulic resistance and flow rate – check what these are in the heat pump manual. Other heat pumps may be supplied without an integrated circulation pump.

9.7.1 Pump efficiency

Pumps can lose efficiency in two discrete steps. Firstly, there is the efficiency with which the electrical motor converts electrical energy to mechanical rotational energy (shaft power). Secondly, there is the efficiency with which a rotating shaft moves the fluid (e.g. via a rotating screw or impeller). Pump brochures thus typically cite two measures of pump efficiency:

$$\text{Pump efficiency} = \frac{\text{Hydraulic power (as given in Equation 9.3)} - \text{kW}}{\text{Mechanical power imparted by shaft} - \text{kW}} \tag{9.8}$$

$$\text{Pump} + \text{motor efficiency} = \frac{\text{Hydraulic power (as given in Equation 9.3)} - \text{kW}}{\text{Electrical power input} - \text{kW}} \tag{9.9}$$

The electrical input is often shown on pump curves as P1, the shaft (mechanical) power as P2. One would certainly hope to attain an overall (pump plus motor) efficiency of over 50%, and maybe in the region of 60–70%.

The efficiency of a given centrifugal pump is not constant, but varies with flow rate. There will be a given flow rate at which efficiency is optimal (lower and higher flows result in a decrease in efficiency), and this point is termed the *best efficiency point* (BEP). It is important to size a pump such that it will operate close to optimum efficiency. Also, be aware that efficiency may decrease with time, due to wear, especially of the pump impeller.

9.8 Carrier fluids

We have already seen that it is important to minimise the viscosity of our carrier fluid in order to minimise hydraulic resistance and maximise the likelihood of achieving a degree of turbulence in the ground heat exchange pipe. We can see from Figure 9.5 and Table 9.3 that

- carrier fluid viscosity increases with decreasing temperature (this is yet another reason to avoid operating the ground loop at excessively low temperatures);
- carrier fluid viscosity increases with increasing concentration (i.e. decreasing freezing point). The higher the designed temperature of our ground loop, the lower the concentration of antifreeze we need (while still ensuring an appropriate degree of frost protection).

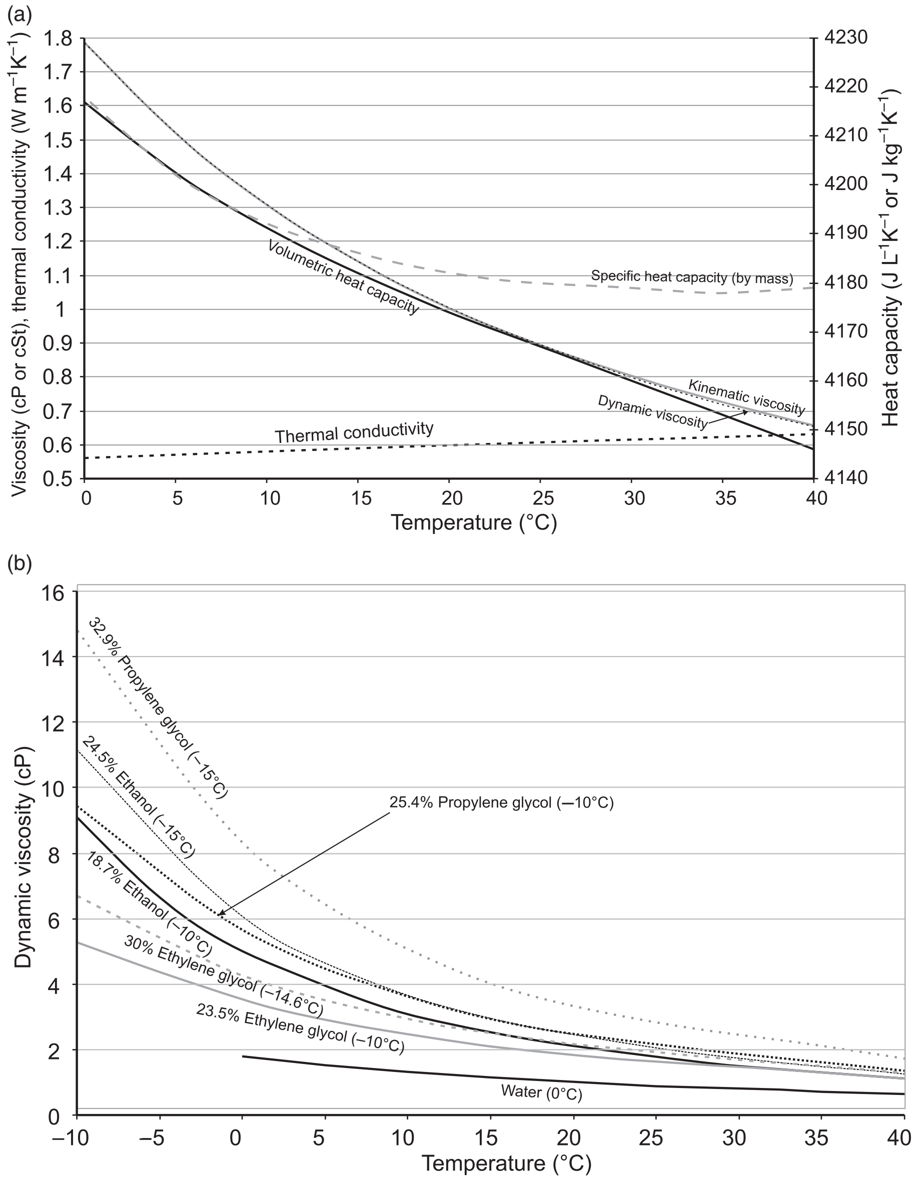


Figure 9.5 How (a) the thermal properties of water and (b) the dynamic viscosity of differing carrier fluids vary with temperature. For each fluid, the antifreeze concentration is in wt% and the freezing point is given in parentheses. After data provided by Melinder (2010).

Table 9.3 Properties of a selection of antifreezes (at 0°C, water at +5°C), including thermal conductivity (λ), density (ρ), dynamic viscosity (μ) and volumetric heat capacity (S_{VC}). The comments on toxicity are indicative only, and you should refer to a formal material safety data sheet (MSDS) if planning to use these products. Biodegradability refers to a typical near-surface, oxidising environment. Compiled from the data provided by Melinder (2010) and Kemira (2006). Freezium™ is a proprietary product based on a solution of potassium formate.

Concentration weight (%)	Fluid	Freezing point(°C)	λ(W m^{-1} K^{-1})	ρ (kg m^{-3})	μ (cP)	S_{VC} (J L^{-1}K^{-1})	Toxic?	Biodegrades?	Salt?
	Water at +5°C	0	0.57	1000	1.519	4202	No		
	Antifreezes at 0°C								
23.5	Ethylene glycol	–10	0.471	1035	3.53	3953	Yes	Y	N
30.5	Ethylene glycol	–15	0.444	1046	4.38	3841	Yes	Y	N
25.4	Propylene glycol	–10	0.448	1026	5.64	3962	Low	Y	N
32.9	Propylene glycol	–15	0.417	1035	8.32	3884	Low	Y	N
18.7	Ethanol	–10	0.455	977	5.00	4288	Low	Y	N
24.5	Ethanol	–15	0.425	972	6.05	4175	Low	Y	N
19.95	Methanol	–15	0.462	973	3.23	3962	Yes	Y	N
18.82	Sodium chloride	–15	0.548	1147	2.59	3916	Low	N	Y
23.9	Potassium acetate	–15	0.492	1129	3.34	3800	Relatively low	Y	Y
24.0	Freezium™	–15	0.51	1149	2.21	3838	Low[1]	Y	Y

[1] According to information provided by the manufacturer.

Table 9.4 The flow rate required (F_{turb}) to yield transient-turbulent flow ($Re > 2500$) in a single, hydraulically smooth 40-mm OD and 32-mm OD SDR 11 pipe, using the dynamic viscosity and density data from Table 9.3.

Concentration weight (%)	Fluid	Freezing point (°C)	F_{turb} for 40-mm OD pipe (SDR 11) (L min^{-1})	F_{turb} for 32-mm OD pipe (SDR 11) (L min^{-1})
	Water at +5°C	0	5.9	4.7
	Antifreezes at 0°C			
23.5	Ethylene glycol	–10	13.1	10.5
30.5	Ethylene glycol	–15	16.1	12.9
25.4	Propylene glycol	–10	21.2	17
32.9	Propylene glycol	–15	31	24.8
18.7	Ethanol	–10	19.7	15.8
24.5	Ethanol	–15	24	19.2
24	Freezium™	–15	7.4	5.9

Table 9.4 shows the flow rates required in single polyethene pipes to ensure a Reynolds Number in excess of 2500 (i.e. the onset of transient-turbulent flow, desirable for efficient subsurface heat exchange). We will also see that, of the common antifreezes, monoethylene glycol is the least viscous and the most hydraulically efficient. Unfortunately, it is also toxic, because it is metabolised to glycolic and oxalic acid in the human body (see Box 9.5). Thus, the use of ethylene glycol in ground loops is strongly discouraged, or even illegal, in several countries (e.g. Scandinavia and a number of US states). In the United Kingdom, the ground source heat pump industry is itself taking steps to block its use (GSHPA, 2011). Even if there is no legal hindrance to ethylene glycol in your country, you should avoid using it in sensitive situations such as

- in heat exchangers installed directly in natural lakes or watercourses (Chapter 12), where leakages could harm the native ecosystem;
- in aquifers within source protection zones of potable water wells, or within the immediate vicinity of significant springs, wetlands or other groundwater-supported habitats or environmentally sensitive features. Remember that the environmental half-life of ethylene glycol is typically days or weeks when evaluating what is a safe distance in terms of groundwater travel time. In chemically reducing aquifers, glycol may degrade more slowly.

9.8.1 *Alternative carrier fluids*

Because of the toxicity of ethylene glycol, a number of other carrier fluids have been evaluated (Table 9.3). Inorganic salt solutions (sodium, calcium or magnesium chloride) are generally not regarded as toxic, are not excessively viscous and can have freezing points lower than zero. Indeed, they have historically commonly been used as heat transfer fluids (hence the term 'brine' for carrier fluid, which is still used).

BOX 9.5 About Ethylene Glycol

Ethylene glycol (also known as glycol alcohol or 1,2-ethanediol) is an attractive heat transfer fluid, due to its relatively low viscosity and lack of corrosivity. In its pure form, it is a clear, colourless, viscous, hydrophilic liquid. Ethylene glycol is a rather simple organic compound, comprising two connected carbon atoms, each with an alcohol group attached, that is, $OH–CH_2–CH_2–OH$.

It is toxic to humans and many other animals, although it is not believed to bioaccumulate. A toxic dose in humans would be considered around $0.1\,g\,kg^{-1}$ of body weight, with reported oral lethal doses varying from around 0.8 to $1.5\,g\,kg^{-1}$ of body weight in humans (Oxford University, 2009; Antizol, 2011). To put this into context, the toxicity of propylene glycol is around 10 times less (i.e. the toxic doses are around 10 times greater). Melinder (2010) regarded an LD_{50} (the dose that produces a 50% fatality rate in laboratory tests) of greater than $2\,g\,kg^{-1}$ as being necessary for a substance to be not classified as harmful.

Ethylene glycol does not sorb strongly to soils, but it does readily biodegrade in soils and in oxygenated surface waters and groundwater environments. The exact rate of biodegradation is dependent on many factors (including microbiota, oxygen availability and pH), but typical environmental half-lives are cited in days to weeks. When it decomposes, ethylene glycol consumes oxygen; thus, excessive glycol leakages to water bodies can deplete them of oxygen. An excellent summary of the ecotoxicity and behaviour of ethylene glycol in the environment is given by WHO (2000).

Degradation of ethylene glycol has also been reported in anaerobic environments. One might expect, however, that if it enters such reducing environments (e.g. deep, oxygen-free aquifers), it is likely to be more persistent and its biodegradation rate somewhat slower. Countries such as Denmark, which rely heavily on such deep groundwaters for their drinking water supply, discourage the use of ethylene glycol in closed-loop boreholes.

The US Environmental Protection Agency has suggested that a lifetime exposure to $14\,mg\,L^{-1}$ in drinking water is unlikely to cause harm in humans (ATSDR, 2010).

Their main drawback is, however, that they are potentially rather corrosive, being ionic substances. Organic salts (such as potassium formate or potassium acetate) are less corrosive (but are still ionic salts) and can represent attractive choices. The commercial product Freezium™ claims to be based on potassium formate and reports an impressively low viscosity.

Propylene glycol is chemically very similar to ethylene glycol, but with three carbon atoms instead of two. Fortunately, it behaves completely differently in the human body and is regarded as having a low toxicity (around 10 times less than ethylene glycol). Unfortunately, it is extremely viscous, and its use can result in high hydraulic head losses and difficulties in achieving turbulent flow (Table 9.4).

We are familiar with the toxicity of ethanol (common alcohol) – we willingly drink it in concentrations of 40–60% as vodka or absinthe. It is also biodegradable and has a high volumetric heat capacity. Again, however, it is quite viscous and its viscosity increases steeply below 0°C (see Figure 9.5). Moreover, above certain concentrations, it becomes potentially flammable: the concentrations at which you can use ethanol may, therefore, be restricted by law or industry best practice in your country.

In the Scandinavian nations, which are nervous about the use of ethylene glycol in ground loops, propylene glycol and ethanol (sometimes denatured with isopropanol to form so-called IPA spirit) are the most commonly used carrier fluids. The biodegradation properties of organic antifreezes are relatively well studied, due to their use in airport de-icing operations. They are generally relatively degradable, although cold, wet ground conditions can severely impede biodegradation – see French *et al.* (2001) and Jaesche *et al.* (2006).

You will also encounter a number of proprietary heat transfer fluids that claim to be formed from 'vegetable-based' chemicals or 'non-toxic' ethylene glycol and that claim attractive hydraulic properties. Such innovations are undoubtedly interesting, but it is unclear whether environmental authorities have sufficient information to evaluate their potential environmental or human health impact in the case of spillage or leakage to aquifers from ground loops. The toxicity of ethylene glycol may well be able to be inhibited by use of an additive – but if that additive has a different geochemical behaviour to glycol, it may become separated from the glycol in an aquifer environment. Greater transparency from manufacturers is undoubtedly needed if the benefits of innovative transfer fluids are to become accepted.

9.8.2 Maintenance of carrier fluids

When a ground loop has been filled with a carrier fluid and sealed, it is tempting to forget the fluid and assume it will circulate efficiently indefinitely. We have already seen that most of the organic fluids degrade, either chemically or biologically. As they degrade, their hydraulic performance and frost protection may deteriorate, sludges may form and viscosity may increase. In the worst instances, the bacteria that regard glycol as food may colonise the ground loop, forming biofilms or bacterial sludge that can clog up the loop.

Ground loops thus require some maintenance. The carrier fluid should be inspected at regular intervals and tested for signs of degradation. Oxidative degradation is often accompanied by an increase in acidity. Some carrier fluids change colour when they degrade.

Biodegradation is encouraged by the presence of bacteria, oxygen and bacterial nutrients (e.g. nitrogen, phosphorus). Thus, when installing the loop, be as clean as possible. Seal the ends of the loop so that soil fragments cannot enter it. Thoroughly flush the loop with clean water after installation. Use clean, sterile water to mix the carrier fluid solution. Pressure test the loop to ensure joints are properly tight.

9.8.3 Additives to carrier fluids

Be aware that carrier fluids may contain additives. These may include

- corrosion inhibitors (especially in the case of organic salts);
- biocides or inhibitors to prevent (bio)degradation;
- scale inhibitors.

Make sure that you are comfortable with these additives and that their toxicity will not negatively impact on the overall toxicity of the carrier fluid.

9.9 Manifolds

Most ground source heating and cooling schemes involve arrays of buried heat exchangers that are typically plumbed in parallel to a manifold, which is in turn connected to a heat pump or building heat exchanger in a plant room. The buried heat exchangers may be closed-loop boreholes (Chapter 10), buried heat exchange pipes or 'slinkies' in trenches (Chapter 11), or heat exchangers submerged in lakes or canals (Chapter 12).

We will typically want to ensure that the circulating carrier fluid is distributed approximately equally between each of the parallel circuits. In other words, if the heat pump array has a peak carrier fluid flow rate of $1.67\,L\,s^{-1}$ and is based on five closed-loop boreholes, we will want to ensure that each borehole experiences a peak flow of $0.33\,L\,s^{-1}$.

Let us imagine an even simpler situation: a system based on three boreholes arranged as in Figure 9.6. An ideal hydraulic situation would be to connect each of those three boreholes with near-surface header pipes back to a manifold. Indeed, it is possible to buy an almost prefabricated chamber with inbuilt manifolds for this purpose (Figure 9.7). The header pipes will all have an additional hydraulic resistance (which we should take account of in the equations of Section 9.3). However, if they are all of approximately the same length and design, their hydraulic resistances should be equal and thus the flows should be evenly distributed. If, however, one of the boreholes is 5 m closer to the manifold than the other two, its circuit will have a lower hydraulic resistance, and there is a danger that it will receive a disproportionately high flow. We could try to rectify that situation by incorporating a spare 10-m length of pipe somewhere in the circuit to try to restore the hydraulic balance.

With larger arrays, things may become more complicated. Let us take an extreme example of a line of boreholes (Figure 9.8a). If we connect each borehole back to a manifold at one end of the array, we can clearly see that the circuit to borehole 1 is

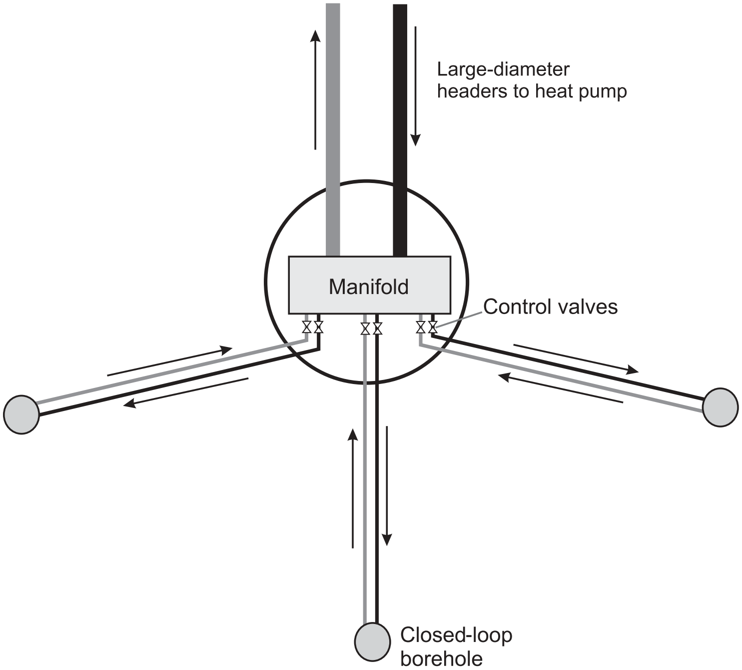


Figure 9.6 A plan of an array of three closed-loop boreholes connected to a manifold chamber in parallel. The control valves provide the possibility to balance flows (if required) and close down the flow to one circuit if a leakage develops.

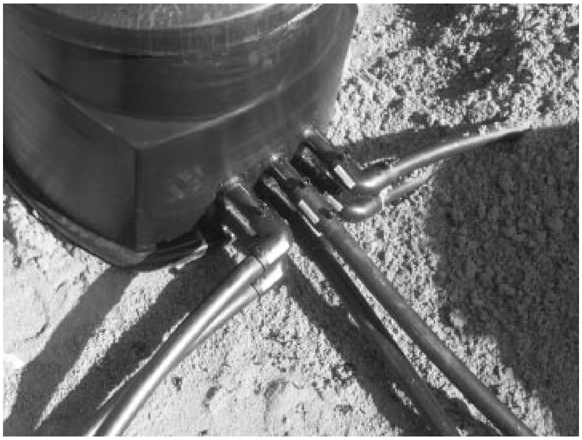


Figure 9.7 A prefabricated manifold chamber, showing three ground-loop circuits connected in parallel, using electrofusion couplings. *Photograph reproduced by kind permission of Water and Energy Field Services Ltd.*

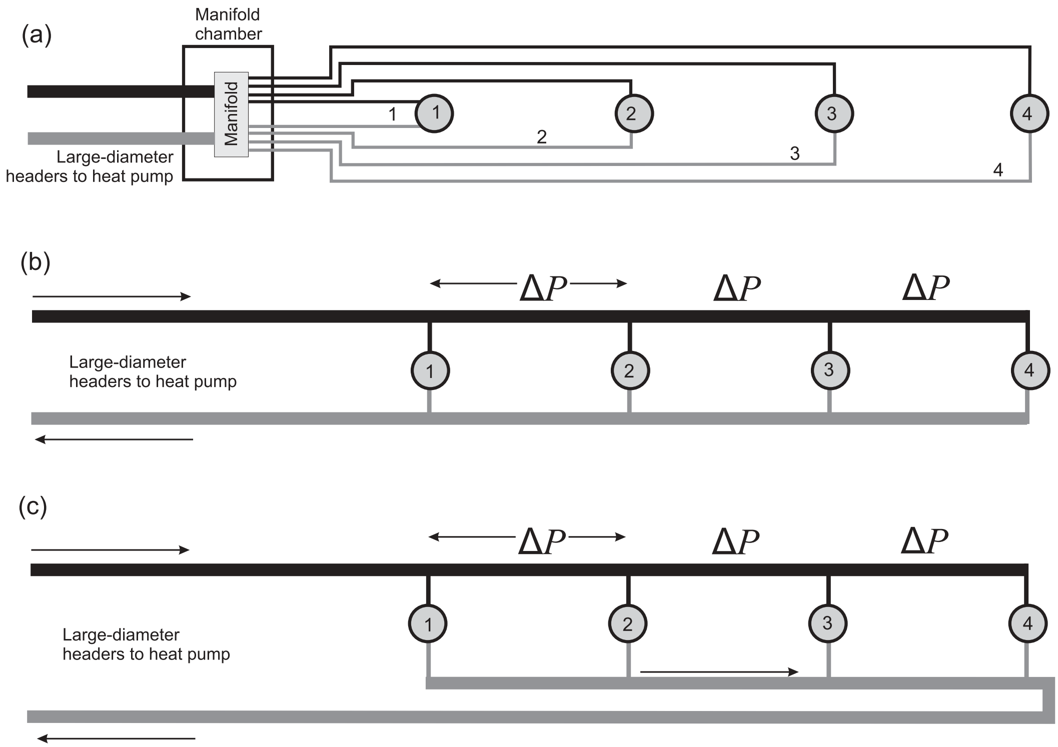


Figure 9.8 Plan illustrating three plausible ways of connecting four indirect circulation closed-loop boreholes in a linear array back to a heat pump.

much shorter and will have a lower hydraulic resistance than that to borehole 4. The following are possible solutions to this:

1. Use manual or automatic valves on the manifold (Figure 9.9) to adjust the resistance of each circuit until the flows are balanced. Unfortunately, although we end up with balanced flows, we increase the overall hydraulic resistance as we are essentially increasing the resistance of each borehole circuit to match that of borehole 4.
2. If the header pipes from the manifold to the borehole are larger diameter than those installed in the borehole, the overall hydraulic resistance of each circuit will be dominated by the borehole U-tube and the header pipes will only have a minor influence. Far less balancing will be necessary at the manifold. Unfortunately, installing large-diameter pipes is expensive both in cost of pipe and increased volume of antifreeze.

It is very common for each parallel circuit on a manifold to be equipped with some form of valve, both to balance the flow, if necessary, and to be able to shut off that

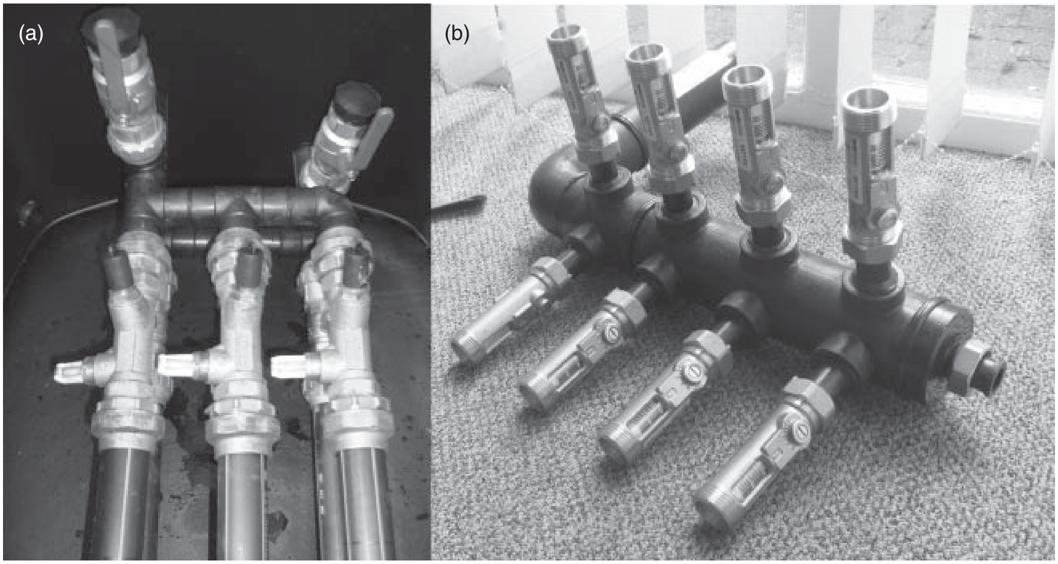


Figure 9.9 Photographs showing two types of carrier fluid manifold: (a) *reproduced by kind permission of Water and Energy Field Services Ltd.*; (b) fitted with automatic flow balancing valves (*photo by David Banks*).

circuit in the event of a carrier fluid leakage or other problem, without having to shut down the entire borehole array (Figure 9.10). The pipes connecting the manifold (Figure 9.8a) to the heat pump will typically be of larger diameter than the ground loops, as they must convey the total flow of carrier fluid with minimal additional hydraulic resistance.

Figure 9.8b shows an alternative solution to the problem of the line of boreholes, with large-diameter flow and return headers extending along the line. There is one obvious problem with this arrangement – there is no central manifold chamber or rack where the flow to individual boreholes can be shut off in the event of a problem (e.g. a leakage). In this arrangement, a leakage may mean that we lose the entire borehole line, or we must accept the need for individual shut-off valve chambers adjacent to each borehole. If the header pipes are narrow, there will be a pressure loss (ΔP) along the header from borehole to borehole. Thus, the hydraulic pathway through borehole 1 will have a lower hydraulic resistance than that through borehole 4 and may represent a preferential flow pathway. To avoid this, we must ensure that the headers are of large enough diameter that ΔP is negligible.

Alternatively, we can employ what is termed a *reflex* or *reverse return* (Figure 9.8c). With this arrangement, all borehole pathways experience the same theoretical hydraulic resistance as they all 'see' the same length of header. This arrangement still suffers, however, from the lack of separate control and shut-off of individual boreholes.

In very large borehole arrays, a combination of both concepts may be employed, with clusters of boreholes arranged around manifolds mounted on large-diameter

Figure 9.10 A rather complex arrangement of carrier fluid manifolds in a subsurface chamber; *photograph reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

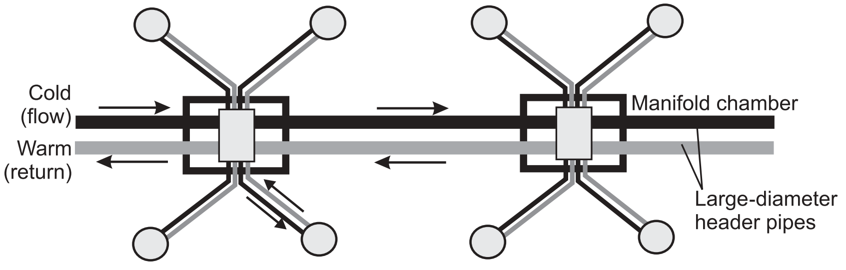


Figure 9.11 An 'octopus' arrangement of closed-loop boreholes around a manifold.

headers (Figure 9.11). Skarphagen (2006) has argued that such an 'octopus' configuration is a particularly effective means of minimising pipe length and ensuring relatively balanced flows.

9.10 Hydraulic testing of closed loops

It is of great importance to ensure that each ground heat exchange element, installed in a trench or borehole, is hydraulically tight and leak-proof. Leakage of antifreeze from a closed-loop system poses operational problems for the owner and may incur liability if it results in pollution of an aquifer or a surface water body.

When a prefabricated closed-loop element (e.g. a borehole U-tube) arrives on site, it should be in pristine condition. Some installers will verify this by carrying out a brief pressure test by applying compressed air, sealing the tube and verifying that it can maintain a pressure of 4–6 bars for at least 30 min (EN 15450, 2007).

The ground loop will then be installed in a trench or borehole. In the latter case, it will then typically be backfilled by a grout slurry. In order to withstand (1) the pressure of water or drilling mud in the borehole remaining after drilling and (2) the pressure of the grout slurry, the U-tube may be water filled and pressurised during installation. To ascertain that the tube can withstand the external grout pressure, a pressure calculation should be made of the weight (pressure) of grout at the bottom of the borehole, based on the grout slurry density and depth.

Immediately following installation, but before the grout has begun to set or cure, the loop is flushed and subjected to a regime of flow and pressure testing. Water is usually flushed around the ground loop to remove any debris (soil fragments, plastic shavings, etc.) and air. During flushing, the direction of flow should be able to be reversed. Flushing is usually adequate if a flow velocity of at least $0.61\,\mathrm{m\,s^{-1}}$ can be maintained everywhere in the circuit, without visible air bubbles or debris emerging, for at least 15 min (Eugster, 2011; GSHPA, 2011).

Following purging, pressure testing of the water-filled pipe can take place. Several methods are available, but the method selected should

1. involve a specific calculation of the test pressure required, such that the test pressure within the loop exceeds the external pressure of the grout slurry at the base of the borehole by a minimum of 0.5 bar (Eugster, 2011);
2. consider the fact that plastic pipe deforms under pressure. It is important to be able to identify whether an apparent decrease in pressure is due to expansion of the pipe (acceptable) or to leakage (unacceptable).

To illustrate the former consideration, let us consider a 110-m-deep ground loop filled with water. At the top of the loop, an excess pressure of P bars is applied. At the bottom of the loop, the water pressure will be around $P + (110 \times 1000 \times 9.81)/100\,000$ bars = $10.8 + P$ bar. If the borehole is backfilled with a grout slurry of density $1550\,\mathrm{kg\,m^{-3}}$, then the grout pressure at the base of the bore will be $(110 \times 1550 \times 9.81)/100\,000$ bars = c. 16.7 bars. Thus, the minimum excess test pressure applied at the surface needs to be at least 6.4 bars to ensure that there is an outward pressure differential of at least 0.5 bar at the base of the loop. It is common to use a test pressure of at least 6 bars (VDI, 2001a).

As regards the second consideration, quite complex methods are available that involve pressurising a plastic pipe and then analysing the shape of a pressure decay curve: for example, WRC (1999). Such methods become increasingly important as the size of the tested system increases. Alternatively, VDI (2001a) and Eugster (2011) suggested using a test (Figure 9.12) involving a pre-loading phase to stress the polyethene pipe, followed by a pressure drop back down to the test pressure (as calculated above). VDI (2001a) suggested a pre-loading phase of 30 min and a test period of at least 60 min,

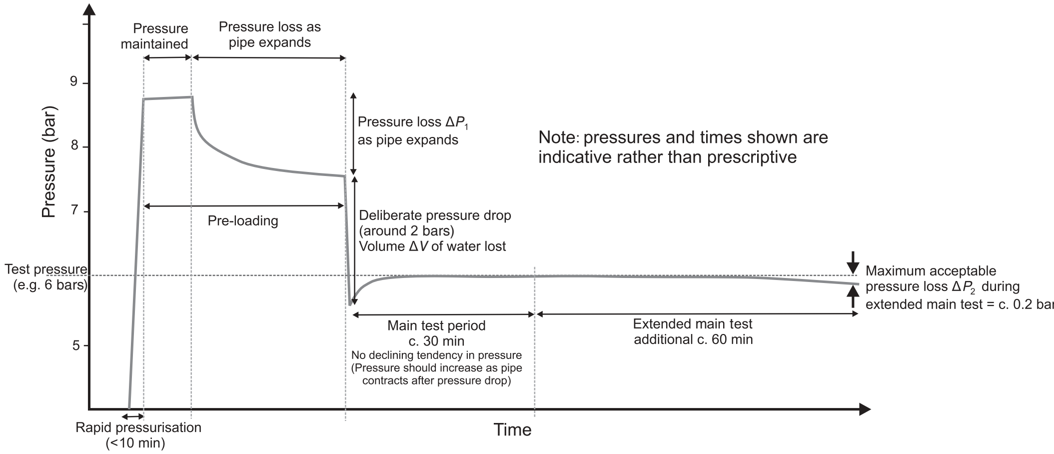


Figure 9.12 A pre-load pressure test of the type recommended by EN 805 (2000) and VDI (2001a) for polyethene/polypropene pipes. There should be no declining trend for pressure in the main test period. One should also evaluate the pre-loading pressure loss ΔP_1, the volume of water removed during the pressure drop phase and the pressure loss ΔP_2 in the optional extended main test.

during which any pressure loss should not exceed 0.2 bar (with a test pressure of at least 6 bars).

Finally, a flow test can be performed, ideally in both directions. Here, water is pumped around the ground loop; the pressure differential across the loop and the flow rate are measured by manometers and a reliable flow meter, respectively. Several measurements are made at flow rates approximating the design operational rate, to verify that the expected pressure drop and flow rate correspond to theoretical calculations of the type shown in Sections 9.3–9.5.

It is important that pressure testing should be carried out immediately (within a few hours) after loop installation and before any grout has begun to cure or set. If pressure testing is carried out while grout is setting, the expanding polyethene pipe may push the grout out. At the end of the test, the pipe will contract, but the grout may not squeeze back into place, possibly leaving a gap between the pipe wall and the grout. This will lead to thermal inefficiency and a possible pollution pathway into the deep subsurface.

For more on testing, the reader is referred primarily to WRC (1999), VDI (2001a), IGSHPA (2007) and Eugster (2011), although HVCA (2006), EN 805 (2000) and EN 15450 (2007) will also be of interest.

9.11 Equipping a ground loop

There is clearly no standard design or specification for a ground loop. However, one should ensure that a ground loop is installed with the following features:

- circulation pumps;
- shut-off valves for each parallel circuit. How much of the array might you 'lose' if a serious leak develops?
- a sensor system to detect drops in pressure (leakage?) or flow (obstruction?), which will alert the user and shut down the heat pump, hopefully minimising carrier fluid loss and freeze damage to the evaporator;
- a low-temperature cut-out, set at an appropriate level for the antifreeze being used (frost protection, may be included in heat pump);
- fittings to allow refilling and flushing of loop;
- pressure maintenance and monitoring system (expansion vessel, pressure relief valve and manometer);
- a flow meter may be appropriate for larger systems, although many types will add additional hydraulic resistance;
- temperature sensors on flow and return pipes (may be included in heat pump);
- de-aeration/air bleed valve(s) at high point(s) on circuit;
- on larger systems, an electronic data logger to collect operational data.

STUDY QUESTIONS

9.1 Connect the correct last half of each statement with the correct first half:

1. a 32-mm OD single U-tube
2. a 40-mm OD double U-tube
3. a 40-mm OD single U-tube

is often a good choice of heat exchanger for

1. typical closed-loop boreholes of depth c. 100 m
2. arrays comprising many shallower boreholes
3. arrays comprising relatively few, but deep boreholes.

9.2 A carrier fluid comprising 25.4 wt% propylene glycol in water is flowing at 0°C through a 40-mm OD (SDR 11) polyethene pipe of length 150 m. What is the pressure loss in kilopascal if the flow rate is (1) 15 L min^{-1} and (2) 25 L min^{-1}?

10

Subsurface Heat Conduction and the Design of Borehole-Based Closed-Loop Systems

Heat is in proportion to the want of true knowledge.
Laurence Sterne, *The Life and Opinions of Tristram Shandy, Gentleman* (1761)

10.1 Rules of thumb?

Chapter 7 introduced us to vertical, borehole-based, indirect closed-loop systems. These employ a carrier fluid circulating within a heat exchange pipe [typically, a high-density polyethene (HDPE) U-tube] installed down a borehole. In heating mode, the carrier fluid is chilled by the heat pump: as it circulates down the borehole, it absorbs heat from the rocks/sediments in the borehole wall and conveys this heat back to the heat pump. This heat is extracted by the heat pump and the chilled carrier fluid returns in its closed circuit down the borehole.

In cooling mode, the heat pump (or a passive heat exchanger) rejects heat to the carrier fluid. This warm carrier fluid then transfers heat into the rocks or sediments in the wall of the borehole.

Chapter 9 introduced us to the necessity of considering the hydraulic design of such closed-loop heat exchangers, so that we avoid using excessive quantities of electrical energy on pumping the carrier fluid around the closed loop. In this chapter, we will

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

focus on the heat exchange aspects of the borehole, and we will learn how to estimate the numbers and depths of boreholes required to support a given ground source heating/cooling scheme.

We saw, in Chapter 7, that for the ground source heat pump (GSHP) systems installed in the British Isles, each metre of drilled borehole typically supports 37–104 W (average 67 W) of installed peak heat pump capacity (Figure 7.11a). This is referred to as the *specific installed thermal output*. If we assume that these peak outputs relate to heating mode and that a heat pump scheme has a typical coefficient of performance (COP) of 3.4, then these values equate to *specific peak heat extraction rates* (i.e. heat extracted or absorbed from the ground) of 26–73 $W\,m^{-1}$ (average 47 $W\,m^{-1}$). This reflects the relatively low range of thermal conductivities in the geological environment (Tables 3.1 and 3.4). The thermal yield of ground source heat schemes contrasts with the groundwater yield of water wells, which can range from a few tens of $L\,h^{-1}$ (or even less) in hard crystalline bedrock to 200 $L\,s^{-1}$ or more in fissured limestones (Misstear *et al.*, 2006). This reflects the huge range of hydraulic conductivities in different lithologies, spanning nine or more orders of magnitude (Table 8.1).

Many installers of small domestic heat pump systems will simply use this 'rule of thumb' (i.e. around 50 $W\,m^{-1}$ peak heat extraction rate) and add a small factor of safety for design purposes (drilling a few extra metres to provide a safety margin may be cheaper than indulging in a site-specific design assessment). They may modify this rule of thumb depending on their understanding of the thermal conductivity of the geology. This rule of thumb appears to be in line with international experience, which is summarised below on the basis of Rosén *et al.* (2001):

- In the United States, typical *specific installed thermal outputs* of 68–82 $W\,m^{-1}$ are reported for boreholes installed with single U-tubes.
- In Switzerland, *specific heat extraction rates* above 75 $W\,m^{-1}$ are not recommended.
- In Austria, recommended *peak-specific heat extraction rates* range from 30 $W\,m^{-1}$ for dry sediments to 70 $W\,m^{-1}$ for granite, for a temperature difference of 10°C between carrier fluid and undisturbed ground.
- In Germany, peak *specific heat extraction rates* of 20–25 $W\,m^{-1}$ are recommended for low conductivity (<1.5 $W\,m^{-1}\,K^{-1}$) strata, 50–60 $W\,m^{-1}$ for medium-conductivity strata and 70–84 $W\,m^{-1}$ for high conductivity (>3 $W\,m^{-1}\,K^{-1}$) strata. In each cited range, the lower value applies to systems with long operational usage (2400 h $year^{-1}$) and the higher value to short operational usage (1800 h $year^{-1}$). More detailed tables and nomograms are given for specific rock types, thermal conductivities and so on, by VDI (2001a).
- Across Europe in general, the average peak *specific heat extraction rate* is estimated at 62 $W\,m^{-1}$ (and taken over the entire heating season, 159 $kWh\,m^{-1}\,year^{-1}$) for systems with operating times of 1600–2400 h $year^{-1}$ (Rosén *et al.*, 2001).
- *However*, see Section 10.11 for details of somewhat more conservative specific heat extraction rates proposed in the United Kingdom.

Such rules of thumb may be a good starting point for design but, *please note*, excessive reliance on them is likely to be dangerous. Even from the international experiences outlined above, we can begin to sense that a simple rule of thumb (a certain number of watts per drilled metre) is likely to be *too* simplistic. The performance of a closed-loop system will depend on the following:

1. The thermal conductivity, specific heat capacity and temperature of the ground. We have already seen that the conductivity does not vary too much from rock type to rock type (i.e. typically within the range of 1.5–6 $W m^{-1} K^{-1}$) and heat capacity varies even less. Ground temperature is also relatively constant within a given region (although it may be significantly higher beneath urban areas – see Chapter 16).
2. The operational pattern of the scheme: for example, the number of equivalent full-load operational hours per heating season and the duration of peak operation on a daily basis. The number of equivalent full-load operational hours per heating season for domestic properties can range from 3000 to 4000 in Scandinavia to as few as 1000 in middle Europe.
3. The operating temperature of the ground loop. In heating mode, this will typically be around 0°C (although it can fall lower during times of peak demand). In cooling systems, however, there is no 'natural' upper limit, and loop temperatures of up to 40°C can sometimes be encountered. Kavanaugh and Rafferty (1997) noted that, in one example, raising the loop's operating temperature from a value of 29.4°C to a value of 32.2°C could result in a 14% shorter ground loop – although this would be at the expense of the efficiency of the heat pump.
4. The efficiency of the heat pump.

The 'rules of thumb' discussed thus far in Section 10.1 tend to be derived from consideration of relatively small, heating-only (or, at least, heating-dominated) schemes in temperate and northern Europe, because the factors above tend to remain fairly constant. However, ground temperature will vary somewhat from country to country and we should remember that 'rules of thumb' from chilly Sweden or Canada may not necessarily be directly applicable to the United Kingdom or Italy: the ground temperature will be lower in Sweden and the achievable differential between carrier fluid and ground may also be less. This means that one might expect generally lower rates of heat absorption as well. Indeed, SVEP (1998) provided a set of nomograms for the dimensioning of Swedish closed-loop systems where the climatic zone (ground temperature) is explicitly taken into consideration. The maximum temperature difference between carrier fluid and undisturbed ground is assumed to be 7°C in the far north of Sweden and 12°C in the south.

With larger projects, cooling requirements become more dominant. Furthermore, in sizable schemes, the number of boreholes begins to be large compared with available land area. In these circumstances, the following additional factors also begin to come into play:

5. Thermal interference between closely spaced boreholes.
6. Complex heating and cooling loads. Within a single day, one may have a heating demand in the early morning and a cooling demand in the afternoon. Carrier fluid temperature differentials and COP values will be different in cooling mode to those in heating mode. Seasonally reversible underground thermal energy storage (UTES) schemes may require shorter borehole arrays than comparable 'heating-only' schemes because, in winter, heat is being abstracted from an aestifer that has been preheated by waste heat 'dumped' the previous summer.

Thus, as schemes become larger and larger, the 'rules of thumb' that are founded on a number of assumptions become less and less reliable. It is absolutely insufficient to 'guestimate' a specific heat absorption rate and add a margin of safety: the 25% difference between a rock thermal conductivity of 2.5 and 2.0 $\mathrm{W\,m^{-1}\,K^{-1}}$ becomes highly significant in terms of drilling costs for a borefield comprising, say, 20 boreholes. Thus, especially for large ground source heating and cooling projects, we require a sophisticated understanding of subsurface heat storage and transfer, site-specific input data and nuanced design tools.

10.2 Common design flaws

The most common design flaws in closed-loop heating or cooling systems can be summarised as follows:

- Using simplistic rules of thumb (e.g. 7-kW output per 100-m borehole), without appreciating that many of the assumptions underlying these 'rules' may be violated (e.g. see Section 10.1). Whether schemes are unidirectional or reversible and, in the latter case, whether the heating and cooling loads are balanced or not (see Glossary) may have a significant impact on the necessary drilled metres of borehole.
- Placing boreholes too close together such that thermal interference becomes significant. Some installers use borehole spacings as low as 4–5 m. While such short spacings may be necessary if land area is very limited, this author would suggest using a minimum spacing of 10 m as a starting point for design of schemes with a highly dominant heating or cooling load. For balanced, reversible schemes, smaller spacings will be acceptable (see Chapter 14).
- Lack of appreciation that GSHP cooling systems *may* require greater borehole lengths to deliver a given cooling load, than heating systems do to provide the same heating load (depending on the loop's operating temperatures). This is because, in active cooling schemes, we are often disposing of waste heat from the compressor, in addition to heat from the building (see Chapter 6).
- Using too short a design life for simulation of the performance of a closed-loop borehole array. Some designers use a simulation period as low as 10 years. We will see later that the system may not begin to approach a steady state until after some

three decades or more have elapsed. We would certainly hope that most buildings and heating schemes would have a design life of more than 10 years.
- Lack of consideration of the hydraulics of the ground loop (see Chapter 9).

10.3 Subsurface heat conduction

The most important mechanism for subsurface heat transfer from the aestifer to the heat exchanger of a typical vertical closed-loop heating system is conduction (forced convection becomes dominant once the heat is in the circulating carrier fluid). We have already met Fourier's law (which has its hydrogeological analogue in Darcy's law):

$$Q = -\lambda \cdot A \cdot \frac{d\theta}{dL} \tag{10.1}$$

where
Q = flow of heat in joules per second, or watts ($\mathrm{J\,s^{-1}}$ or W);
λ = thermal conductivity of the material ($\mathrm{W\,m^{-1}\,K^{-1}}$);
A = cross-sectional area under consideration ($\mathrm{m^2}$), perpendicular to direction of heat flow;
θ = temperature (K);
L = distance coordinate in the direction of decreasing temperature (note that heat flow is in the direction of decreasing temperature: hence the negative sign in the equation); $d\theta/dL$ = temperature gradient ($\mathrm{K\,m^{-1}}$).

Now, we are in a position to consider how three-dimensional heat conduction in the subsurface causes temperature to change with time. Let us consider a very small volume of aestifer material (V_{aest}) of dimension $\delta x \times \delta y \times \delta z$, isotropic thermal conductivity λ and volumetric heat capacity S_{VC} (Figure 10.1). The heat flow in through the front face of the element (from Fourier's law) is

$$Q_{xin} = -\delta y \cdot \delta z \cdot \lambda \cdot \left(\frac{\partial\theta}{\partial x}\right)_x \tag{10.2}$$

and the heat flow out through the rear face is

$$Q_{xout} = -\delta y \cdot \delta z \cdot \lambda \cdot \left(\frac{\partial\theta}{\partial x}\right)_{x+\delta x} \tag{10.3}$$

Thus, the change in heat flow in the x direction is given by

$$Q_{xout} - Q_{xin} = -\delta x \cdot \delta y \cdot \delta z \cdot \lambda \cdot \left(\frac{\partial^2\theta}{\partial x^2}\right) = -V_{aest} \cdot \lambda \cdot \left(\frac{\partial^2\theta}{\partial x^2}\right) \tag{10.4}$$

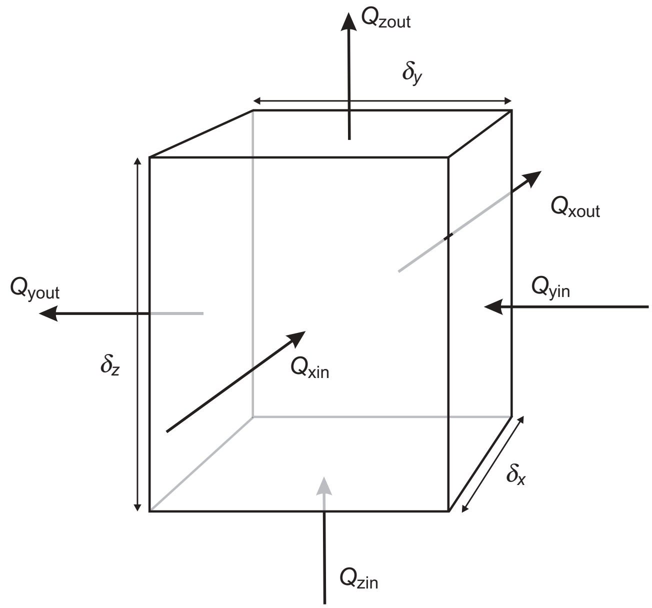


Figure 10.1 Schematic diagram of the conductive heat flow through a representative volume of aestifer, of negligible dimension $\delta x \times \delta y \times \delta z$.

We can perform a similar exercise for the y and z directions. As the dimensions of the volume become negligible, we can construct the following differential equation, based on the assumption that, for a given increment of time, the net heat influx to the volume is equal to the volumetric heat capacity multiplied by the change in temperature:

$$V_{\text{aest}} \cdot \lambda \cdot \left(\frac{\partial^2 \theta}{\partial x^2}\right) + V_{\text{aest}} \cdot \lambda \cdot \left(\frac{\partial^2 \theta}{\partial y^2}\right) + V_{\text{aest}} \cdot \lambda \cdot \left(\frac{\partial^2 \theta}{\partial z^2}\right) = V_{\text{aest}} \cdot S_{\text{VC}} \cdot \frac{\partial \theta}{\partial t} \tag{10.5}$$

$$\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial z^2} = \frac{S_{\text{VC}}}{\lambda} \frac{\partial \theta}{\partial t} \tag{10.6}$$

This equation essentially states that, for every tiny volume of aestifer, the heat that enters must equal the heat that leaves.[1] If it does not, a progressive temperature change with time results. If more heat enters than leaves, the temperature increases and vice versa. (This is, in fact, the same as the equation given in Box 8.3, less the

[1] The hydrogeological equivalent of this equation states that the amount of water entering each small block of aquifer should equal what goes out: if it doesn't, it results in a change in groundwater head:

$$\frac{\partial^2 h}{\partial x^2} + \frac{\partial^2 h}{\partial y^2} + \frac{\partial^2 h}{\partial z^2} = \frac{S_s}{K} \frac{\partial h}{\partial t} = \frac{S}{KD} \frac{\partial h}{\partial t} = \frac{S}{T} \frac{\partial h}{\partial t}$$

term for heat advection with groundwater.) This equation is the foundation of computer-based numerical heat transfer models: the aestifer is mathematically broken down into a finite difference (or finite element) grid of small volumes of rock and Equation 10.6 is solved by the computer simultaneously for each small block of aestifer. Note that Equation 10.6 assumes that thermal conductivity is isotropic – it is the same in the x, y and z directions. In reality, it may not be (Box 10.1), making the equation a little more complex but still tractable.

BOX 10.1 Thermogeological Anisotropy

In reality, thermal conductivity is not isotropic: it is a tensor and has a magnitude that varies with direction. In particular, horizontal thermal conductivity can be different from vertical conductivity. This may be due to the primary sedimentary or crystalline structure of the rock, it may be due to fracturing or jointing in a lithified rock or it may be due to fine-scale layering or lamination within a sediment or sedimentary rock. Indeed, as early as 1885, M. Jannettaz reported that the thermal conductivity along the planes of foliation in European schists, slates and gneisses was 1.5–1.98 times greater than the conductivity perpendicular to them (Prestwich, 1885). Bloomer (1981) found ratios of between 1.19 and 2.17 for British Jurassic mudrocks. Midttømme *et al.* (1998) found ratios of between 1.04 and 1.74 for English Tertiary and Jurassic mudrocks.

Consider a volume of sedimentary rock, 10 m thick, comprising 10 000 very thin layers, each of thickness D = c. 1 mm, of alternating sand ($\lambda = 2.4\,\mathrm{W\,m^{-1}\,K^{-1}}$) and clay ($\lambda = 1.6\,\mathrm{W\,m^{-1}\,K^{-1}}$). The (horizontal) thermal transmissivity (T_{the}) of this 10-m-thick aestifer can be found by summing the product of conductivity and thickness for each layer:

$$T_{\mathrm{the}} = \sum_{n=1}^{10\,000} \lambda \cdot D = \left(5\,000 \times 0.001\ \mathrm{m} \times 2.4\ \mathrm{W\ m^{-1}\ K^{-1}}\right) + \left(5\,000 \times 0.001\ \mathrm{m} \times 1.6\ \mathrm{W\ m^{-1}\ K^{-1}}\right) = 20\ \mathrm{W\ K^{-1}}$$

The bulk average horizontal thermal conductivity λ_{bh} is obtained by dividing this by the total thickness of the aestifer (it is, in fact, the thickness-weighted arithmetic mean of all the layers). It is this average thermal conductivity that we would typically use for the design of closed-loop boreholes in a layered sequence of sediments or rocks:

$$\lambda_{\mathrm{bh}} = \frac{T_{\mathrm{the}}}{10\ \mathrm{m}} = 2.0\ \mathrm{W\ m^{-1}\ K^{-1}}$$

(*Continued*)

BOX 10.1 (*Continued*)

If we consider the vertical conduction of heat, we can consider the aestifer as a collection of thermal resistances connected in series. The thermal resistance (R) of each layer is given by

$$R = \frac{D}{\lambda} = 4.2\times10^{-4}\ \text{m}^2\ \text{K W}^{-1}\ \text{ for sand and } 6.3\times10^{-4}\ \text{m}^2\ \text{K W}^{-1}\ \text{for clay}$$

The total vertical thermal resistance (R_{tot}) of the 10-m-thick sequence is the sum of the thermal resistances of 5000 sand and 5000 clay layers, or 5.21 $\text{m}^2\,\text{K}\,\text{W}^{-1}$. Thus, the bulk vertical thermal conductivity λ_{bv} is given by

$$\lambda_{bv} = \frac{10\ \text{m}}{R_{tot}} = 1.9\ \text{W m}^{-1}\text{K}^{-1}$$

This is, in fact, the thickness-weighted, harmonic mean of the conductivities of the various layers.

10.4 Analogy between heat flow and groundwater flow

Surely, if Darcy's law is analogous to Fourier's law and if the heat balance Equation 10.6 has a direct parallel in groundwater flow theory, we should be able to apply analogues of the Theis (Equation 8.8), Cooper–Jacob (Equation 8.11) and Logan (Equations 8.16 and 8.19) equations to the radial flow of heat towards a closed-loop borehole. And we can . . . almost! In fact, the 'rules of thumb' of Section 10.1, which relate heat yield to borehole depth, are the thermogeological version of the Logan approximation.

There is one important difference between hydrogeology and thermogeology, however: this difference lies in the boundary conditions for an aquifer and an aestifer. Let us first consider a typical unconfined aquifer (Figure 10.2). The base of the aquifer is usually some low-permeability aquitard: it can be approximated to a no-flow boundary. The top of the aquifer is *either* (a) another aquitard, in the case of a confined aquifer, *or* (b) a water surface (the water table), in the case of an unconfined aquifer, with a potentially fluctuating head, but which receives (in the long term) an approximately constant annual supply of rainfall recharge. Conceptually, therefore, the upper boundary is either (a) a no-flow boundary (perfectly confined aquifer) or a constant-flow boundary (unconfined aquifer). Theis's and Cooper–Jacob's equations predict that, if we pump a borehole in the aquifer, a cone of drawdown of head develops around the borehole. The cone of drawdown continues to develop spatially

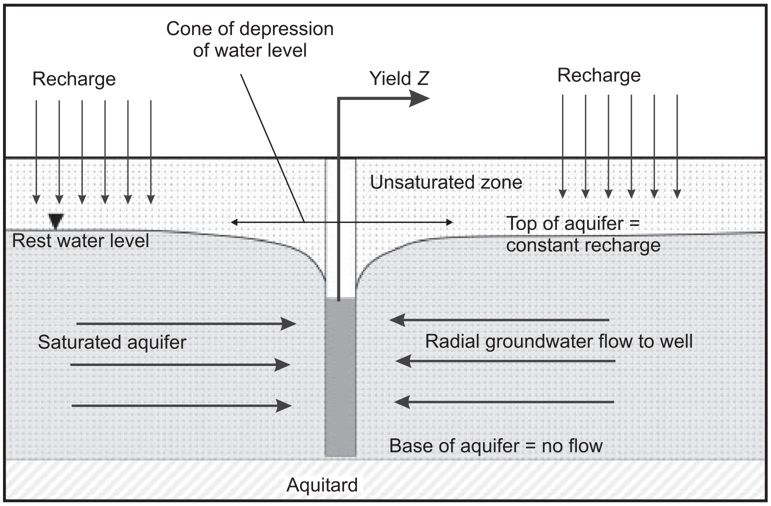


Figure 10.2 Schematic conceptual model of a pumped well in an unconfined aquifer, showing typical boundary conditions.

(in theory) *ad infinitum* and indefinitely in proportion to the logarithm of time (Figure 10.3).[2]

Let us now consider a closed-loop borehole in an aestifer (Figure 10.4). The aestifer has no physical base, but we can conceptually consider the bottom of our model as a constant flow boundary supplying a flux of geothermal energy of several tens of $mW\,m^{-2}$. The top of the aestifer is not a constant-flow boundary. In fact, the top of the aestifer can be regarded as a constant-temperature boundary in the long term (several years timescale) at the annual average surface temperature. If the ground is hotter than the average surface temperature, heat will be lost via the surface. If we abstract heat from our borehole and cool down the ground, we will induce heat flow from the surface into the ground. The colder the ground, the greater the heat flow induced from the surface (ultimately, from the atmosphere). Thus, the boundary conditions for our aestifer are different to our aquifer. The aestifer has a constant temperature boundary, whereas the aquifer (at least in our drawing) has no constant head boundary. If we pump heat from our closed-loop borehole, a zone of depressed temperature develops in the rock around the borehole. Initially, heat stored in the surrounding rock will be conducted radially in towards the borehole (the early phase of Figures 10.5 and 10.6). Later, the ground will cool down to such an extent that heat

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[2] In reality, the cone of drawdown will usually stabilise in a steady-state condition, but to do this, it must either (1) induce recharge from a constant head boundary such as a river, lake or the sea, or (2) suppress natural discharge such as baseflow to rivers, springs or wetlands. In other words, only head-dependent sources of recharge can stabilise a cone of depression, and not constant rate sources of recharge (rainfall) – see Theis (1940) and Bredehoeft *et al*. (1982).

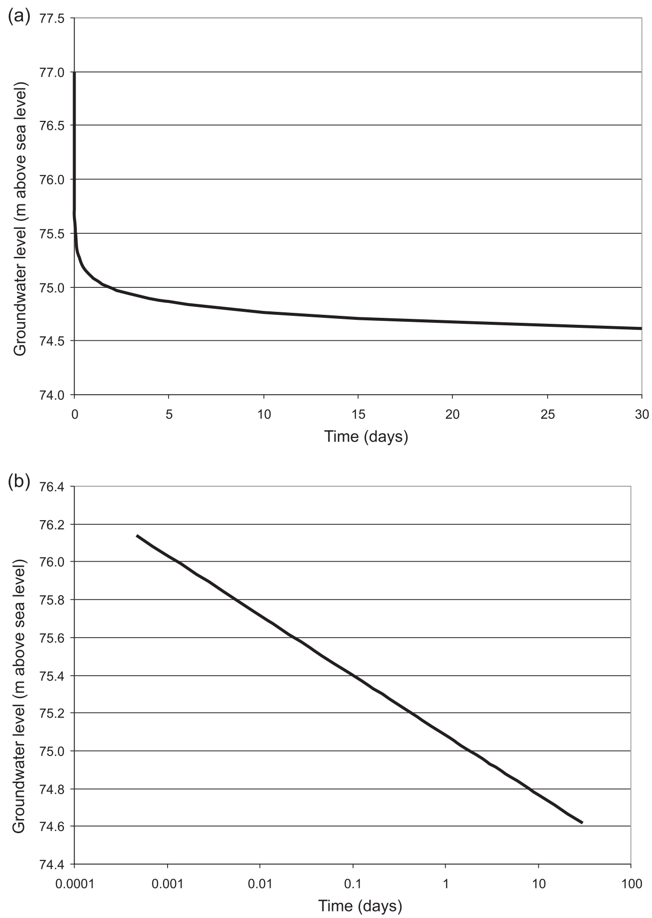

Figure 10.3 The development of groundwater level (a) with time and (b) with $\log_{10}$ (time) in a 200-mm-diameter groundwater well, being pumped at a constant rate of $10\,L\,s^{-1}$. Here, it is assumed that the well is hydraulically efficient and that drawdown is small relative to aquifer thickness. The initial rest water level is 77 m above sea level, the transmissivity is $500\,m^2\,day^{-1}$ and storage is 10%.

flow will be increasingly induced from the surface and the underlying rocks. Eventually, the heat flow induced from the surface will balance the heat abstracted and a steady-state condition will develop (the late phase of Figures 10.5 and 10.6).

The other major difference between the aquifer and the aestifer is that, in our aquifer, there is usually little or no vertical head gradient prior to pumping (i.e. head does not change greatly with depth in many high-permeability aquifers). In our aestifer, we initially have a vertical temperature gradient corresponding to the geothermal gradient. Our borehole has an approximately constant average temperature along its length, however, as we are circulating a carrier fluid through it. It is the temperature of this carrier fluid that we are interested in simulating for the practical design of our ground source system.

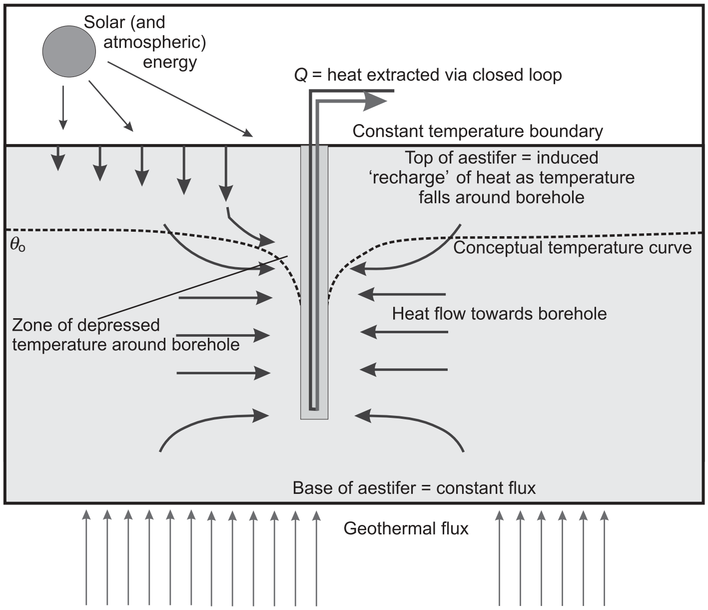

Figure 10.4 Schematic conceptual model of a closed-loop ground source heat scheme, abstracting heat from an aestifer, showing typical boundary conditions. θ_o = initial 'far-field' temperature.

10.5 Carslaw, Ingersoll, Zobel, Claesson and Eskilson's solutions

10.5.1 Early phase of heat abstraction

The fundamental equations governing the radial conduction of heat towards a line 'sink' are based on the solutions of Horatio Carslaw (1921). He envisaged (Figure 10.7) a line 'source' of heat (e.g. an electrical resistance filament) embedded in a conducting sheet of thickness D, sandwiched between two insulators. If a constant input of heat (H) is generated by the 'line' source, the rate of heat transfer into the conductor is $q = H/D$ in $\mathrm{W\,m^{-1}}$. Carslaw wanted to find a solution to the temperature (θ) at any point in the conductor, at a distance r from the source, at a time t after the heat was switched on. He was essentially trying to solve our Equation 10.6 in radial coordinates and in two dimensions:

$$\frac{\partial^2\theta}{\partial x^2} + \frac{\partial^2\theta}{\partial y^2} + \frac{\partial^2\theta}{\partial z^2} = \frac{S_{VC}}{\lambda}\frac{\partial\theta}{\partial t} \qquad (10.6)$$

with the following conditions:

- at $t = 0$, $\theta = \theta_0$ for all values of r and z;
- as $r \to \infty$, $\theta = \theta_0$ for all values of t (where θ_0 is termed the *far-field temperature*).

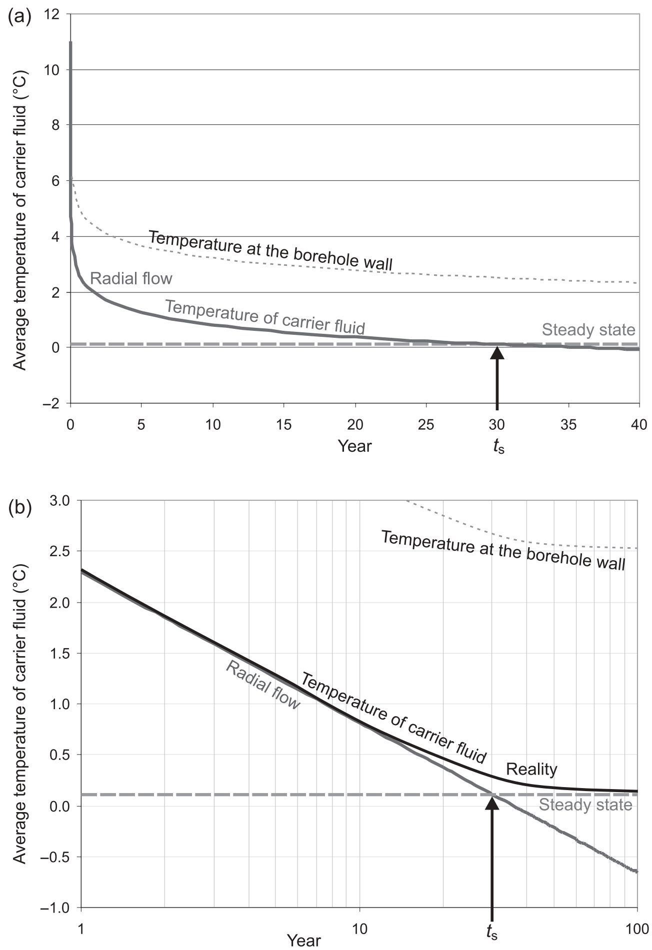

Figure 10.5 The development of average temperature (a) with time and (b) with $\log_{10}$ (time) of the carrier fluid in a closed-loop borehole. Here, the borehole diameter is 126 mm, the constant heat extraction rate is 2 kW, $R_b = 0.12\,\mathrm{K\,m\,W^{-1}}$, ground thermal conductivity $2.48\,\mathrm{W\,m\,K^{-1}}$, $S_{VC} = 2.4\,\mathrm{MJ\,m^{-3}\,K^{-1}}$ and the undisturbed ground temperature 11°C over the borehole's 100 m length. The diagram also shows the calculated temperature of the borehole wall (2.4°C hotter than the carrier fluid).

Carslaw found that the solution was far from trivial:

$$\theta - \theta_0 = \frac{q}{4\pi\lambda} E(u) = \frac{q}{4\pi\lambda}\left[-\gamma - \ln(u) - \sum_{n=1}^{\infty}(-1)^n \frac{u^n}{n.n!}\right] \tag{10.7a}$$

where $E(u)$ is the so-called exponential integral function (the Theis well function – see Chapter 8) and $u = r^2 S_{VC}/(4\lambda t)$.

A geologist would realise that Carslaw's scenario (Figure 10.7) looks very much like a closed-loop borehole (a 'line source' of heat in cooling mode or 'line sink' in heating mode) penetrating a sequence of strata. Indeed, during the 1940s, a number of workers,

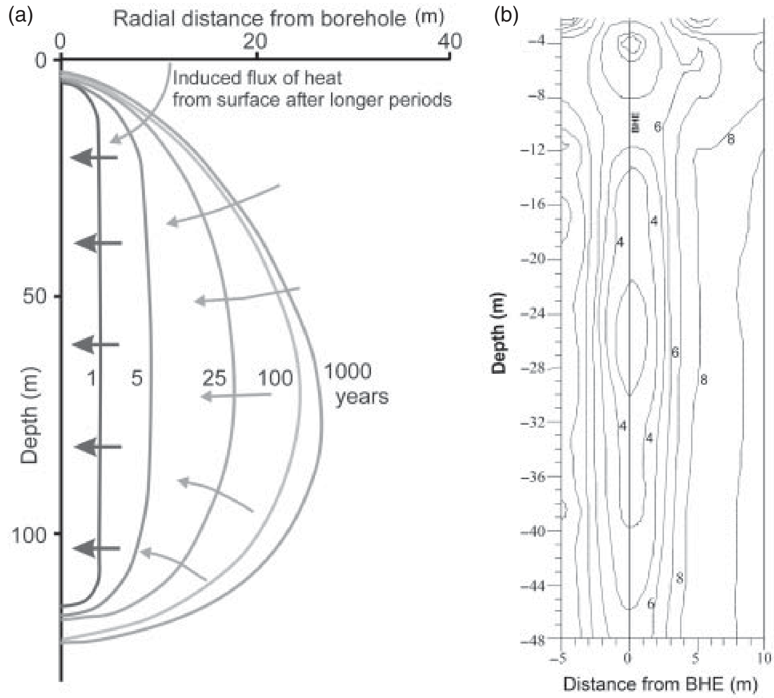

Figure 10.6 The development of the 'thermal front' in the rock surrounding a closed-loop borehole. (a) The contours representing the loci of a 1°C drop in temperature around a closed-loop borehole, extracting heat at a constant rate of $22\,W\,m^{-1}$ from the aestifer (thermal conductivity $3.5\,W\,m^{-1}\,K^{-1}$, diffusivity $1.62 \times 10^{-6}\,m^2\,s^{-1}$), after different periods of time. The black arrows show lines of heat flux after 1 year's operation (radial flux towards borehole), while the grey arrows show lines of flux after 25 years (borehole is beginning to induce heat flow from the surface and to approach a steady state). *Redrawn from a diagram by Claesson and Eskilson (1987b) and published in Energy, Vol. 13(6), 509–52, © Elsevier (1988).* (b) Real monitored ground temperature (contours in °C) around a closed-loop borehole heat exchanger (BHE) in Germany, after one heating season (7 months) of operation. *After Sanner (2002) and reproduced by kind permission of Dr Burkhard Sanner.*

including Leonard and Alfred Ingersoll, Otto Zobel (Ingersoll *et al*., 1948) and Guernsey *et al*. (1949), applied Carslaw's and similar equations to various geological heat exchange geometries.

It turns out that the geological scenario differs from Carslaw's (Figure 10.7) in only two significant respects:

1. the initial temperature of the conductor (the ground) is not constant, but increases slowly with depth, according to the geothermal gradient;
2. the natural ground is devoid of perfect thermal insulators! In fact, there will be a three-dimensionality to the real situation – heat can be derived from above (overlying rocks and atmosphere) and from below (underlying rocks and geothermal heat flux).

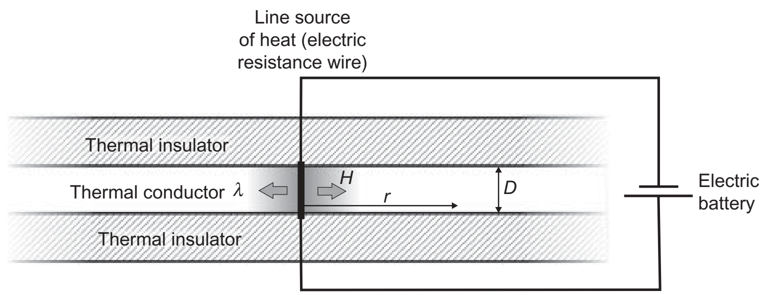

Figure 10.7 Carslaw's original conceptualisation of an infinitely broad thermally conductive plate, of thickness *D*, sandwiched between insulators, with a line source of heat.

The Swedes Johan Claesson and Per Eskilson (1987a,b) have provided a particularly coherent investigation of numerical and analytical solutions to the extraction of heat from a closed-loop borehole in an aestifer, and it is their work that forms the basis for the subsequent discussion. Via numerical modelling, they were able to derive a curve similar to Figure 10.5b, but a simple analytical solution proved elusive. They were, however, able to gain traction on the problem by the use of two simplifying assumptions:

Assumption 1: that the geothermal gradient can be neglected and that we can consider the aestifer as initially having a uniform temperature equal to the average temperature over the borehole length. Thus, if a 100-m borehole (prior to any heat extraction) has a temperature near the surface of 9°C and a temperature at its base of 11°C, we would say that the aestifer has an initial or 'far-field' average temperature θ_o of 10°C. If we imagine that we circulate a carrier fluid around the ground loop, prior to switching on our heat pump, it will tend to equilibrate with this average temperature θ_o.

Assumption 2: that, in the early phase of Figure 10.5, we can neglect any vertical component of heat flow (i.e. induced heat flow from overlying rocks, from the underlying rocks and from the surface) and simply simulate horizontal radial heat flow from the aestifer to the borehole. In other words, in the early phase of its operation, the closed-loop borehole is simply removing heat from storage in the earth's subsurface reservoir immediately adjacent to the borehole.

We can thus rewrite Equation 10.7a as 10.7b, specifically applying to the *extraction* of q W m^{-1} heat from a closed-loop borehole:

$$\theta_o - \theta_b = \frac{q}{4\pi\lambda} E(u) = \frac{q}{4\pi\lambda}\left[-\gamma - \ln(u) - \sum_{n=1}^{\infty} (-1)^n \frac{u^n}{n.n!}\right] \qquad (10.7b)$$

where $u = r_b^2 S_{VC}/4\lambda t$.

θ_b is the average temperature (K) at radius r_b from the centre of the line heat sink – that is, at the wall of the borehole – at time t. If we assume that the borehole is *perfectly thermally efficient* (i.e. that there is no temperature loss or thermal resistance between the wall of the borehole and the carrier fluid), then θ_b is also the average temperature of the carrier fluid:

$\theta_0 - \theta_b$ = temperature 'drawdown' or displacement (K);
q = heat extraction rate per metre of borehole ($W\,m^{-1}$); that is, heat extraction rate divided by effective depth, taking account of any upper portion of a borehole that may be thermally insulated from the rock;
λ = thermal conductivity of the rock ($W\,m^{-1}\,K^{-1}$);
r_b = borehole radius (m), and γ = Euler's constant = 0.5772.

This is our thermal analogue of Theis's equation (Equation 8.8). Actually, it is more correct to say that Theis's equation was derived from this radial heat flow equation (Box 8.1). As in the case of the Cooper–Jacob approximation (Equation 8.11), we can simplify this expression for low values of u (high values of t):

$$\theta_o - \theta_b \approx \frac{q}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{r_b^2 S_{VC}}\right) - 0.5772\right] \tag{10.8}$$

This equation implies that, in the early ('radial flow') phase of Figure 10.5, the temperature of the carrier fluid declines with the logarithm of time. This approximation (Equation 10.8) is valid for values of t within the following range (Equations 10.9a,b):

$$t > \frac{5 r_b^2 S_{VC}}{\lambda} \tag{10.9a}$$

If the value of t is lower than this constraint, the mathematics of the approximation breaks down. For both Equations 10.7 and 10.8, if t is too high (i.e. greater than the following constraint – see Equation 10.10):

$$t < \frac{t_s}{10} \tag{10.9b}$$

the log-linear relationship between time and θ_b begins to diverge, albeit slightly, from the real curve (Figure 10.5) as the system begins to induce heat flow from the overlying and underlying rocks and from the ground surface. The three-dimensionality of heat conduction begins to become important and the system (hopefully) begins to slowly approach steady state.

10.5.2 Late (steady state) phase of heat extraction

The time t_s, after which 'steady state' (Equation 10.11) begins to be a better description of the temperature evolution than radial flow (Equations 10.7b and 10.8), is given by Claesson and Eskilson (1987a) as

$$t_s = \frac{e^{\gamma} D^2 S_{VC}}{18\lambda} \approx \frac{D^2 S_{VC}}{9\lambda} \tag{10.10}$$

where
γ = Euler's constant = 0.5772;
D = borehole depth (m) over which heat extraction takes place.

The steady-state temperature of the carrier fluid, towards which our real temperature curve converges, as it begins to draw on heat from the surface and from the overlying and underlying rocks, is given by

$$\theta_o - \theta_{s,b} = \frac{q}{2\pi\lambda} \ln\left(\frac{D}{r_b\sqrt{4.5}}\right) \approx \frac{q}{2\pi\lambda} \ln\left(\frac{D}{2r_b}\right) \text{ if } D \gg r_b \tag{10.11}$$

where $\theta_{s,b}$ = steady-state temperature of collector fluid in the ideal borehole (K) = steady-state temperature of borehole walls, assuming that the thermal resistance of the borehole structure itself is negligible.

10.5.3 Heat rejection to a closed-loop borehole

For a ground source cooling system, the heat load rejected to the ground (G) is related to the cooling effect (C) and the COP_C (or, in the long term, the seasonal performance factor, SPF_C) of the heat pump by

$$G \approx C\left(1 + \frac{1}{COP_C}\right) \text{ or } G \approx C\left(1 + \frac{1}{SPF_C}\right) \tag{10.12}$$

Exactly the same Equations 10.7–10.11 can be used for heat rejection to the ground, except that the heat extraction rate will be negative and the borehole temperature θ_b will progressively rise above θ_o.

10.6 Real closed-loop boreholes

10.6.1 Borehole thermal resistance

The equations described in Section 10.5 assume that the closed loop is in ideal thermal contact with the aestifer, with no additional thermal resistance within the borehole itself. In reality, of course, there will be thermal resistances related to (among other factors)

- the conductivity of the filling of the borehole annulus (typically grout);
- thermal short-circuiting (heat leakage between the upflow and downflow shanks of the U-tube), which will depend on the temperature difference between the shanks and their spacing;
- thermal resistance associated with transfer of heat from the grout, through the U-tube wall, to the carrier fluid.

We must thus take into account this borehole thermal resistance (R_b): it imparts an additional temperature loss between the aestifer and the carrier fluid, over and above that predicted by Equations 10.7b, 10.8 and 10.11.

The borehole thermal resistance is due to the interplay of a number of factors. In reality, it is quite complex to understand, especially for short-term pulses of heat. However, in the longer term, beyond several hours, it turns out that we can lump all of these factors into a single mathematical black box (R_b), which behaves as a linear term with respect to heat extraction.[3] Note that, for short times (less than a few hours), we cannot necessarily regard borehole thermal resistance as a simple linear term, and for a rigorous understanding, we may need to indulge in more complex mathematical modelling (see Sutton *et al.*, 2002).

Many readers will recall Ohm's law from school, which describes the flow of electric current (I) through an electrical resistance (R) under the influence of a voltage (potential difference – V). It is, in fact, the electrical analogue of Darcy's law and Fourier's law. It states that

$$I = V/R \text{ or } V = IR \tag{10.13}$$

Let us consider a simple electric circuit, comprising a battery of voltage V and two resistances in series (R_1 and R_2). If we consider the current I that will flow around the circuit and if we remember that resistances in series are additive, we can say that

$$V = I(R_1 + R_2) = IR_1 + IR_2 \tag{10.14}$$

Similarly, we can consider our heat flow q (analogous to electric current I) as having to pass through two thermal resistances in series to reach the carrier fluid in a closed-loop borehole (Figure 10.8).

R_1 is the natural thermal resistance of the ground, which is given by Equation 10.7b, 10.8 or 10.11:

[3] This is the analogue of well loss in groundwater engineering (see Section 8.7.1). However, whereas hydraulic well loss is non-linear relative to abstraction rate, borehole thermal resistance is (at least in our simple approximation) linear.

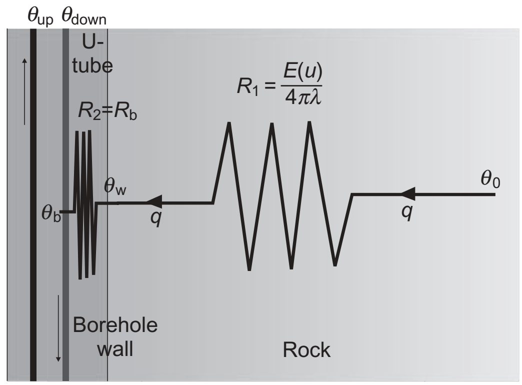

Figure 10.8 Heat conduction from the ground to a closed-loop borehole conceptualised as two thermal resistances in series: R_1 is the thermal resistance of the ground, R_2 is the thermal resistance of the borehole. θ_0, θ_w and θ_b are the temperature of the natural, undisturbed aestifer, the wall of the borehole and the carrier fluid (average), respectively. q is the heat flow per metre of borehole.

$$R_1 = \frac{E(u)}{4\pi\lambda} \approx \frac{1}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{r_b^2 S_{VC}}\right) - 0.5772\right] \tag{10.15}$$

R_1 is a little bit strange as it is not a constant, but evolves with time. R_2 is the borehole thermal resistance R_b. The flow of heat is driven by a thermal potential difference (analogous to the voltage V); that is the temperature difference between the carrier fluid and the natural ground temperature. We can thus write a modified version of Equation 10.7b for real boreholes:

$$\theta_o - \theta_b = q(R_1 + R_2) = qR_b + \frac{q}{4\pi\lambda}E(u) \tag{10.16}$$

In this equation, one half of the equation gives the temperature difference between the carrier fluid (θ_b) and the average temperature of the borehole wall (θ_w):

$$\theta_w - \theta_b = qR_b \tag{10.17}$$

while the other half gives the temperature difference between the undisturbed aestifer (θ_0) and the average temperature of the borehole wall (θ_w):

$$\theta_o - \theta_w = \frac{q}{4\pi\lambda}E(u) \approx \frac{q}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{r_b^2 S_{VC}}\right) - 0.5772\right] \tag{10.18}$$

We can also rewrite Equations 10.8 and 10.11 for real boreholes:

$$\theta_o - \theta_b \approx qR_b + \frac{q}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{r_b^2 S_{VC}}\right) - 0.5772\right] \text{for } t > \frac{5r_b^2 S_{VC}}{\lambda} \tag{10.19}$$

$$\theta_o - \theta_{s,b} = qR_b + \frac{q}{2\pi\lambda}\ln\left(\frac{D}{r_b\sqrt{4.5}}\right), \text{ if } D \gg r_b \tag{10.20}$$

Note that, when we talk about the temperature of the carrier fluid in the closed-loop borehole (θ_b), we mean the average fluid temperature: the mean of the fluid temperature emerging from the upflow shank and that entering the downflow shank. The difference between these temperatures can be found from Equation 9.4:

$$\theta_b = (\theta_{up} + \theta_{down})/2 \tag{10.21}$$

The thermal resistance of a borehole can be regarded as an empirical quantity that may be measured for every borehole by a thermal response test (Chapter 15). Its units are K m W^{-1}. However, various authors have suggested formulae for estimating its value: for example, Shonder and Beck's (2000) *gpm* software provides the following very simple formula for a U-tube heat exchanger:

$$R_b = \frac{\ln(r_b/r_U)}{2 \cdot \pi \cdot \lambda_g} \tag{10.22}$$

where r_U and r_b are the radii of the U-tube pipe and borehole, respectively, and λ_g is the thermal conductivity of the grout (or other backfill). Kavanaugh and Rafferty (1997) implied that this is probably too simplistic, contending that

$$R_b = R_g + R_U \tag{10.23}$$

where R_g is the thermal resistance of the grout or other backfill and R_U is the resistance related to the U-tube. They then argue that, for a situation where the shanks of the U-tube are neither touching each other or the borehole wall:

$$R_g = \frac{(r_b/r_U)^{0.6052}}{17.44\lambda_g} \tag{10.24}$$

and R_U = 0.043 K m W^{-1} (for SDR 11 pipe – see Box 9.4) and 0.054 K m W^{-1} (for SDR 9) for turbulent flow in HDPE U-tube. For transitional flow (*Re* = 2300–4000), an additional 0.005 K m W^{-1} can be added to the R_U for turbulent flow; for laminar flow (*Re* = 1000–2300), a figure of 0.014 K m W^{-1} is added.

Alternatively, if all this seems far too complicated, analytical software programs such as Earth Energy Designer (EED; Blomberg *et al.*, 2010) can be used to calculate the borehole thermal resistance from given input parameters, including the dimension, configuration and material of the collector loop, its shank spacing, the type and flow rate of the carrier fluid, the borehole diameter and the grout/backfill's thermal properties. As an example, let us imagine a 125-mm-diameter borehole installed with 32-mm outer diameter (OD) HDPE U-tube of wall thickness 3 mm (SDR 11). The borehole is backfilled with grout of thermal conductivity 1.5 W m^{-1} K^{-1} and the carrier fluid is flowing turbulently. Equation 10.22 yields a value of R_b = 0.145 K m W^{-1}. The Equations 10.23 and 10.24 yield values of R_g = 0.087 K m W^{-1} and R_b = 0.130 K m W^{-1}. To place these values into context, Mands and Sanner (2001) and Sanner *et al.* (2000) cited values of borehole thermal resistance between 0.06 and 0.50 K m W^{-1}, derived from empirical thermal response tests in Germany. All except two of the German tests yield values below 0.12 K m W^{-1}, however, while boreholes filled with thermally enhanced grout yield values of 0.06–0.08 K m W^{-1}. A 'good' closed-loop borehole thermal resistance would thus be <0.11 K m W^{-1}. A poor R_b would generally be considered >0.14 K m W^{-1}.

Banks *et al.* (2009c) reviewed the results of thermal response tests from the United Kingdom (Figure 10.9 and 15.9). The bulk of the derived borehole thermal resistances fell in the range of 0.08–0.14 K m W^{-1}. It was found, however, that double U-tubes generally yielded lower thermal resistances than single 40-mm OD U-tubes, which in turn had slightly lower thermal resistances than single 32-mm OD U-tubes. This is in line with what one would expect from the relative surface areas of the different U-tube geometries and dimensions (Figure 10.10).

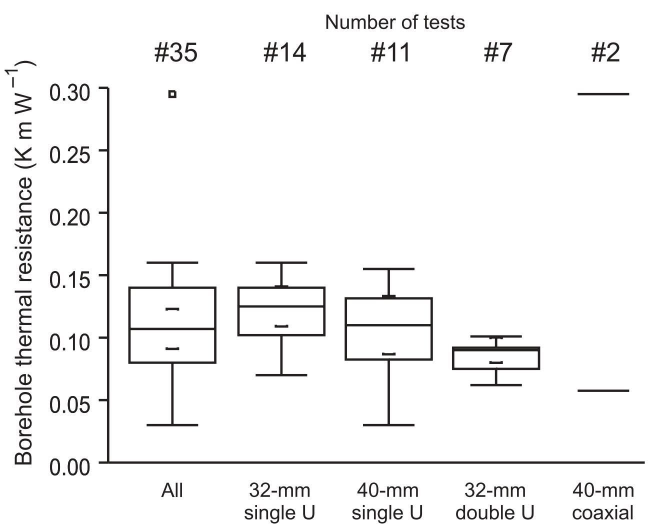

Figure 10.9 Borehole thermal resistances measured in 35 thermal response tests on closed-loop boreholes in the United Kingdom, using water as the heat carrier fluid. This is based on an expanded version of the data set reported by Banks *et al.* (2009c). Operational closed-loop boreholes, with antifreeze-based carrier fluids, may exhibit slightly inferior values of R_b, as the flow is likely to be less turbulent. The central box of the boxplot shows the interquartile range, with a horizontal line at the median and square parentheses indicating a 95% confidence level on the median. The whiskers show the extraquartile data, with the square symbol indicating an outlier.

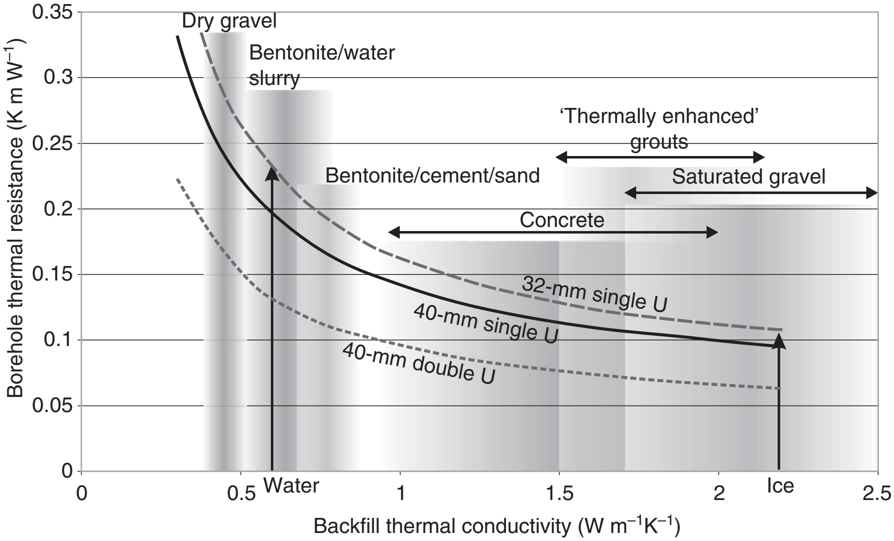

Figure 10.10 The dependence of borehole thermal resistance (R_b) on backfill material, calculated by the program EED on the following assumptions: 127-mm-diameter borehole installed with a U-tube comprising SDR 11 HDPE and shank spacing 60 mm, carrier fluid 25% ethylene glycol circulating at a rate capable of delivering a Reynolds Number of c. 2630 ($0.265\,L\,s^{-1}$ for 32-mm single U, $0.333\,L\,s^{-1}$ for 40-mm single U and $0.666\,L\,s^{-1}$ for 40-mm double U). Excellent contact between the backfill and the borehole wall is assumed. Typical ranges of thermal conductivities from Blomberg *et al*. (2010) and other sources.

10.6.2 Minimising borehole thermal resistance

Obviously, in designing our closed-loop borehole, it is in our interests to minimise borehole thermal resistance. For example, if a 100-m-deep closed-loop borehole, extracting 4.5 kW heat (i.e. $45\,W\,m^{-1}$), has a borehole thermal resistance of $0.14\,K\,m\,W^{-1}$, then there will potentially be an additional temperature drop of $45\,W\,m^{-1} \times 0.14\,K\,m\,W^{-1} = 6.3°C$ between the borehole wall and the carrier fluid. If the borehole was more thermally efficient and if R_b was only $0.1\,K\,m\,W^{-1}$, the temperature drop would be 4.5°C and the heat pump would be operating on a carrier fluid almost 2°C warmer than in the first case. This would translate into an increased COP_H of the heat pump (Figure 4.6).

We can achieve a low value of R_b by ensuring a number of factors are optimised. Figures 10.10 and 10.11 illustrate a number of these. Initially, one might suppose that the use of poorly conductive HDPE for the U-tube, rather than, say, copper, would result in an unnecessarily high value of R_b. There is some truth in this (Figure 10.11), although HDPE tends to be used in modern systems due to its robustness, cheapness and resistance to corrosion.

Secondly, we should specify a carrier fluid flow rate adequate to produce transient-turbulent flow in the carrier fluid. Figure 10.11 demonstrates that R_b declines dramatically with the onset of turbulent flow.

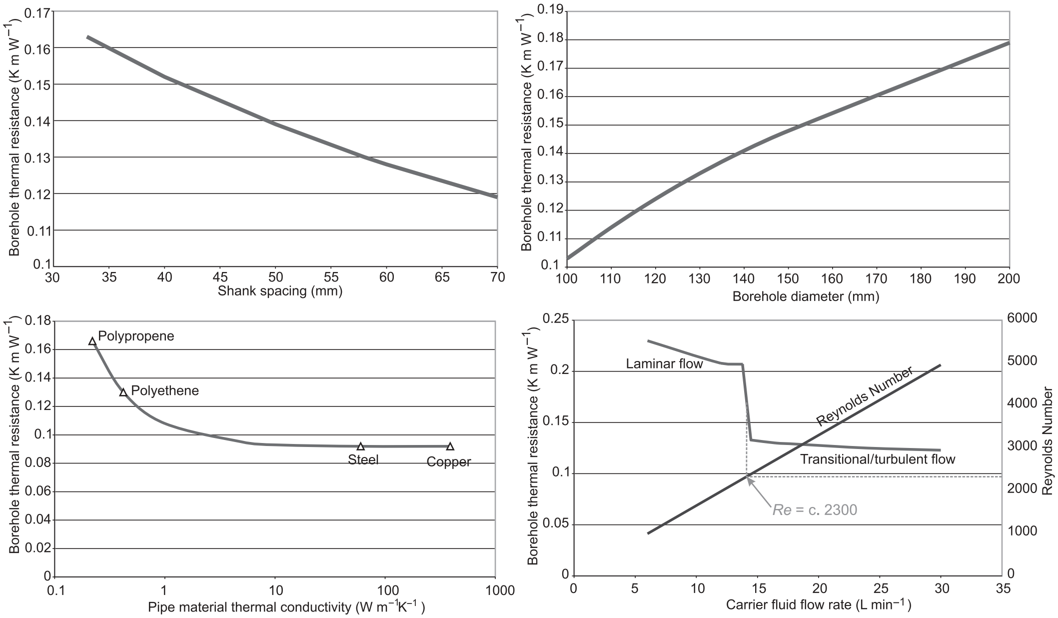

Figure 10.11 The dependence of borehole thermal resistance (R_b) on shank spacing, borehole diameter, U-tube pipe material and fluid flow rate. Note especially the difference that laminar/turbulent flow transition makes to R_b. Calculations were made using the program EED (Blomberg *et al*., 2010), which assumes a Reynolds Number of around 2300 for the laminar/turbulent transition. The baseline assumptions for each case were: 127-mm-diameter borehole installed with a single U-tube comprising 32-mm HDPE, of wall thickness 3 mm and shank spacing 58 mm, backfilled with a thermal grout of conductivity 1.5 W m^{-1} K^{-1}, carrier fluid 25% ethylene glycol circulating at 16.5 L min^{-1}.

Thirdly, we should ensure a large (and constant) shank spacing between upflow and downflow tubes to minimise thermal short-circuiting. The effect of shank spacing is shown in Figure 10.11, while Figure 10.12a and b illustrates the difference between a correctly and an incorrectly spaced U-tube. In practice, small plastic or sprung 'spacers' can be utilised to keep the shanks of a U-tube at a constant separation during installation. Shank spacing is especially important if the thermal conductivity of the backfill or grout is low.

Fourthly, the thermal conductivity of any borehole backfill around the heat exchanger is important for ensuring a low borehole thermal resistance, and the following sections consider this in more detail.

10.6.3 Borehole backfill – grouts

We should ensure that any backfill or grout emplaced in the borehole around the U-tube has a high heat transfer coefficient. Pure bentonite and cement-based grouts

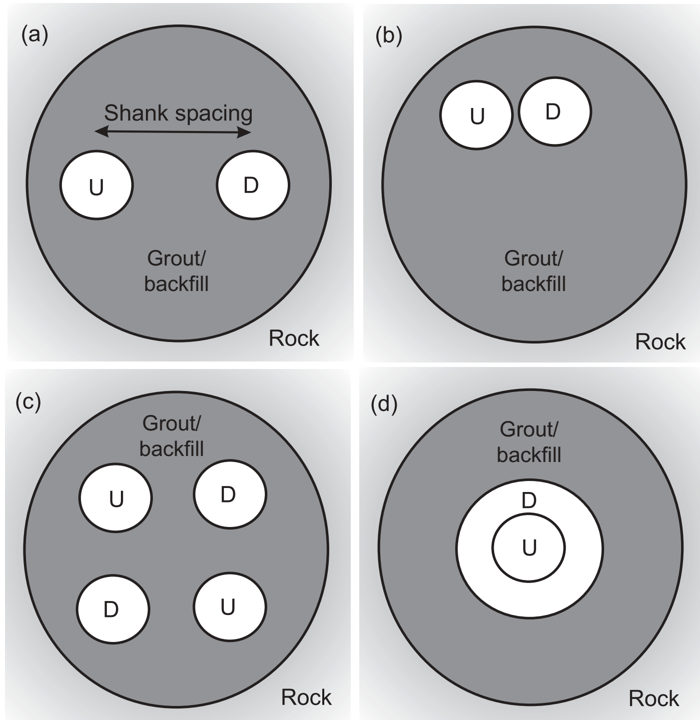

Figure 10.12 Borehole cross section illustrating different types of ground-loop installations: (a) a correctly installed single U-tube, with large shank spacing; (b) a poorly installed single U-tube, with small shank spacing; (c) a double U-tube; (d) a closed coaxial collector. 'U' and 'D' show upflow and downflow portions.

have rather low thermal conductivities. Moreover, Portland-cement-based grouts can have a tendency to shrink away from the U-tube on setting (unless additives are used to prevent this), both reducing the thermal contact and providing a pathway for contaminated surface water. Furthermore, Portland cement releases considerable heat of hydration on setting, which can damage plastic U-tubes (McCray, 1999). We tend therefore to favour grouts that have a high quartz content (and thus a high thermal conductivity), but which also have a high enough cement or bentonite content to ensure a low hydraulic conductivity and a good hydraulic seal. The so-called thermally enhanced grouts are typically of two types:

1. A thick slurry of bentonite, fine quartz sand/silt and water, such as 'Thermal Grout 85' (see Glossary). Such thermally enhanced bentonite/sand grouts are reported to have conductivities in excess of $1.5\,\mathrm{W\,m^{-1}\,K^{-1}}$, while some manufacturers claim conductivities as high as $2\,\mathrm{W\,m^{-1}\,K^{-1}}$. However, Rosén *et al*. (2001) and VDI (2001a) expressed concern that high-bentonite concoctions may be susceptible to damage

upon freezing (bentonite has a high water content, which can freeze out as ice wedges). VDI (2001a) suggested that mixtures of bentonite, quartz silt or sand, furnace fly ash ± cement will give good mechanical properties and frost resistance down to −15°C, although thermal conductivity is much more modest ('over $0.8\,\mathrm{W\,m^{-1}\,K^{-1}}$').

2. Cementitious thermally enhanced grout. In New Jersey, this is defined as (approximately) a 2 to 1 mix by mass of fine silica sand to cement, with small quantities of sodium bentonite and sulphonated naphthalene superplasticiser (GeoExchange, 2003). Allan and Philippacopoulos (2000) found that their 'Mix 111', comprising 2.13 parts sand to 1 part cement, with added superplasticiser, provided a field thermal conductivity as high as $2.19\,\mathrm{W\,m^{-1}\,K^{-1}}$. They observed, however, some problems with shrinkage of the grout away from the U-tube, but still claimed a relatively low overall hydraulic conductivity (a little more than $10^{-9}\,\mathrm{m\,s^{-1}}$). They also experimented with 'Mix 114' and 'Mix 115', where some of the cement was exchanged for granulated blast furnace slag and fly ash, respectively, to reduce the heat of hydration.

Any grout should, of course, be emplaced slowly in the borehole from the bottom up, using a tremie pipe, to avoid air pockets and to ensure a good thermal contact between grout and U-tube. The impact of grout thermal conductivity on R_b is shown in Figure 10.10.

10.6.4 Alternatives to the use of grout

Rather than using a grout, the borehole can be backfilled around the ground loop with a quartz sand or gravel, allowing natural groundwater to fill the pore spaces – resulting in a high thermal conductivity (Figures 7.20 and 10.10). This approach only works, of course, below the groundwater level – the groundwater level should thus ideally be relatively close to the surface. A low-permeability seal is emplaced in the uppermost section of the borehole to prevent ingress of surface water.

Alternatively, the U-tube can be 'dangled' in a borehole filled with natural groundwater (a concept that has been favoured in Scandinavia – Rosén *et al.*, 2001; Skarphagen, 2006; Figure 7.20). This may not seem a promising solution, given water's low thermal conductivity, but a number of factors can conspire to render overall heat transfer rather effective, including (1) formation of high-conductivity ice around the loop,[4] (2) forced convection of heat by groundwater flux, (3) the establishment of free convection cells within the borehole's column of water and (4) a density-driven 'thermosiphon' effect in the aquifer around the borehole (Gehlin *et al.*, 2003). These 'grout-free' solutions presuppose a shallow groundwater level and lack of objection from environmental regulators.

[4] In Norway, this accumulation of ice around a U-tube is called an 'ispølse' or ice sausage!

10.6.5 Ground-loop configuration

Hitherto, we have considered the use of simple HDPE 'U-tubes' as the means by which we extract heat from the deep subsurface by conduction and convey it to a heat pump (Figures 7.14 and 10.13). There are, of course, other configurations of ground loop that can be used, which will increase the amount of heat exchange surface area within the borehole and thus minimise the borehole's thermal resistance. We can, for example, utilise a 'double' (or even a triple) U-tube, with two upflow and two downflow shanks, usually connected in parallel (Figure 10.12). This does (theoretically) have the capability to significantly decrease the borehole's thermal resistance (Figures 10.9 and 10.10). However, its disadvantages are that (1) it is more difficult to install and grout than a single U-tube and (2) it may necessitate a higher carrier fluid flow rate to achieve transient-turbulent flow conditions (see Chapter 9).

Other types of ground loop are available. Canadian researchers (Rosén *et al*., 2001) have even utilized a spiral ground loop – a form of vertical slinky – in large-diameter boreholes.

Coaxial heat exchangers (Figure 10.12) have the advantage of a regular, round cross section: they can thus be installed in soft non-cohesive sediments by using 'push' (e.g. penetrometer) techniques or 'sonic drilling' techniques. In such techniques, the use of grout can be avoided (subject to local environmental regulations): the natural

Figure 10.13 Heat is conveyed from a closed ground loop (in a borehole or trench) to the heat pump by a carrier fluid. Where there are two or more carrier fluid circuits in parallel, they will be connected by means of a manifold (with shut-off valves). *Reproduced by kind permission of Kensa Engineering Ltd.*

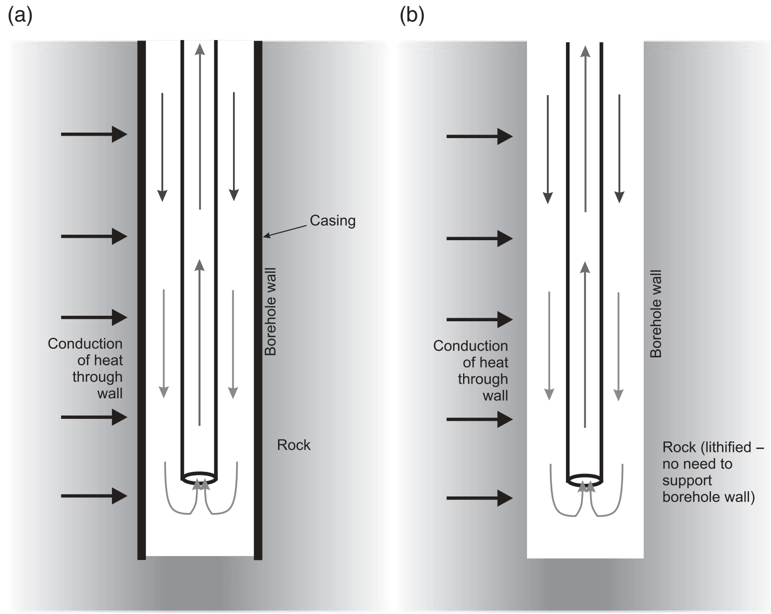

Figure 10.14 An 'open' coaxial collector, where the drilled borehole itself forms the outer flow tube. In low-permeability, stable, self-supporting formations, the borehole wall may not be lined with casing (if this is the case, antifreeze or other potentially polluting substances will not be able to be used): (a) with casing and (b) without casing.

sediments are simply allowed to collapse onto the outer wall of the heat exchange pipe after drilling, following the removal of temporary casing.

A final possibility, in some circumstances, would be to employ an 'open' coaxial configuration where the borehole wall forms the outer flow conduit. The borehole wall may consist of bare rock, or of steel/plastic casing. The circulating carrier fluid (which may be natural groundwater) thus absorbs heat directly from the borehole wall (Figure 10.14). Clearly, environmental regulators are likely to raise objections to the use of antifreeze or other additives in such systems if the borehole wall is unlined, due to the possibility of their release to the wider groundwater environment.

10.7 Application of theory – an example

10.7.1 Design constraints

Equations 10.16–10.20 can be used as the basis for the design of a closed-loop heat extraction system: specifically, the numbers and depths of closed-loop boreholes necessary. But what are our fundamental design constraints? What design criteria are we aiming to achieve? These criteria will normally be related to either (1) the minimum acceptable efficiency (seasonal performance factor) of our heat pump system, or (2) the minimum acceptable carrier fluid temperature. In fact, of course, these two criteria are inextricably linked. The carrier fluid temperature should not drop too low during the operation of a GSHP system, for several reasons:

1. Economic: the heat pump COP_H may decrease to a point where its efficiency is unacceptably low or where any efficiency advantage over, say, an air-sourced heat pump, is lost.
2. Geotechnical: although there is no fundamental operational reason why some degree of ground freezing cannot be accepted (and can, indeed, be advantageous, given the high thermal conductivity of ice), extensive ground freezing is usually regarded as undesirable for geotechnical reasons, especially if buildings are located above or in close proximity to the boreholes. Furthermore, the mechanical properties of some clays may be damaged by repeated freezing and thawing.
3. The bottom line: temperatures should not drop so low that the carrier fluid freezes (25% ethylene glycol has freezing point of around –11°C).

The following design criteria are therefore tentatively suggested, in the context of temperate mid-latitude European climates (although they should not be regarded as absolute or definitive – temperature constraints will ultimately depend on geotechnical, energy efficiency, operational and climatic considerations). Bear in mind that, by 'base load', we mean the longer-term (weekly or monthly) average heat delivery or extraction rate. By 'peak load' we refer to the short-term maximum heat delivery or extraction rate, as experienced on the coldest day of the period in question.

1. The mean temperature of the carrier fluid under average 'base-load' conditions in the coldest month should remain above (and preferably significantly above) 0°C over the design life of the system. This should ensure that any extensive ground freezing is avoided and should also ensure reasonably high seasonal performance factors for the heat pump system (see Note 1 below).
2. The minimum temperature of the carrier fluid should not approach its freezing point during 'peak-load' conditions (see Note 2).
3. Having said this, we would typically not want the average temperature of the carrier fluid during 'peak-load' conditions (i.e. coldest day) to drop below, say, –3°C (implying an entry temperature to the heat pump of maybe –1.5°C). The efficiency (COP_H) of many heat pumps reaches unacceptably low values below such temperatures (see Note 3). Indeed, recent British best practice (GSHPA, 2011; MIS, 2011a,b) suggests that the entry temperature to the heat pump should always remain above 0°C on the coldest days, over a 20-year design life. Assuming a temperature drop of 3°C across the evaporator, this suggests a minimum average carrier fluid temperature of around –1.5°C under peak loading (see Section 10.11).

VDI (2001a) corroborated these suggestions by stating that, under German conditions, the temperature of the carrier fluid entering the ground loop should not be more than 11°C lower than undisturbed ground temperature under base-load conditions (weekly average) and not more than 17°C lower than undisturbed ground temperature under peak-load conditions. If one assumes an undisturbed ground temperature of 11–12°C, this suggests a minimum 'base-load' ground loop entry temperature of around +0 to +1°C and a minimum 'peak-load' ground-loop entry temperature of –5 to –6°C. In turn, this suggests a minimum average 'base-load' fluid temperature of around +1–3°C and a minimum average 'peak-load' fluid temperature of –3 to –4°C (see Note 2).

In designing a closed-loop system, our task is therefore to (1) define minimum acceptable fluid temperatures and heat pump efficiency criteria for our system during its design life, and (2) to design a ground-loop system that, during its design life and under prevalent operational conditions, maintains average carrier fluid temperatures (θ_b) at acceptable levels:

Note 1: one could argue that, in order to avoid ground freezing, the average carrier fluid temperature under 'base-load' conditions should not drop below $-qR_b$ °C over the design life of the system, where q is the 'base-load' specific heat extraction rate. This means that, for an average base-load heat extraction rate of $20\,\mathrm{W\,m^{-1}}$ and a borehole thermal resistance of ~$0.1\,\mathrm{K\,m\,W^{-1}}$, the average temperature difference between the borehole wall and carrier fluid will be 2°C. Thus, we *could* operate with an average (base load) carrier fluid temperature as low as –2°C without the risk of significantly freezing the surrounding rock or sediment. The temperature of the grout or borehole backfill would be expected to drop below 0°C however.

Note 2: remember that the temperature of the fluid entering the ground loop from the heat pump may be 3–4°C cooler than the fluid temperature entering the heat pump from the ground. It may thus be around 1.5–2°C lower than the average carrier fluid temperature θ_b (defined as the average between the ground-loop entry and exit temperatures). Bear in mind the possibility that crystal formation may start to occur on the evaporator well before the bulk carrier fluid temperature approaches it freezing point.

Note 3: the extent to which a low 'peak-load' carrier fluid temperature will be of concern will depend on the proportion of time the system will be operating at peak load. If the system only operates at peak load for a few hours or days each year, the 'peak-load' minimum carrier fluid temperature will be of less concern than for a system where peak loading occurs on a regular basis.

Note 4: The minimum acceptable carrier fluid temperatures will depend on the climate and energy economy of the country in question. In Norway and Sweden, where the natural undisturbed ground temperatures are cold and where electricity is not especially carbon intensive, it may be defensible to run GSHPs at relatively low carrier fluid temperatures (the Swedish program 'Earth Energy Designer', e.g., suggests a minimum average carrier fluid temperature of no lower than –5°C). In the United Kingdom, where ground temperatures are relatively warm and where carbon-intensive electricity is competing against carbon-efficient natural gas, one should probably be designing GSHP systems with higher carrier fluid temperatures and higher COPs. In Central and Southern Europe (where a summer cooling load may balance a winter heating load), one will encounter systems where the average carrier fluid temperature does not drop significantly below +5°C, and where water can be used instead of antifreeze (see Chapter 14, Figure 14.5).

10.7.2 An example

Equation 10.16,[5] 10.19 or 10.20 can readily be programmed into a computer-based spreadsheet, or the calculations may be performed manually. Let us consider a domestic heat pump system, with a COP_H of 3.5 and a peak heat demand on the coldest winter days of 6 kW. This implies that the 'peak-load' heat extraction rate from the ground is 4.286 kW (the remaining 1.714 kW comes from the heat pump compressor). Let us assume that the heat pump is running 2100 h $year^{-1}$, giving a total heat extraction of 9000 kWh $year^{-1}$. The long-term average heat extraction ('base load') over the course of a year is thus around 1.027 kW (= 9000 kWh $year^{-1}$/8766 h $year^{-1}$).

Let us further assume that the ground source system utilises a 126-mm-diameter, 100-m borehole drilled into granite with a thermal conductivity (λ) of 2.48 W m^{-1} K^{-1} and a volumetric heat capacity of 2.4 MJ m^{-3} K^{-1}. Finally, let us assume a borehole thermal resistance (R_b) of 0.12 K m W^{-1} and an initial average mid-European ground temperature (θ_o) of 11°C. We can thus use Equation 10.16 with the following input parameters:

$$\theta_o = 11°\text{C}$$

$$S_{VC} = 2\ 400\ 000\ \text{J m}^{-3}\ \text{K}^{-1}$$

$$\lambda = 2.48\ \text{W m}^{-1}\ \text{K}^{-1}$$

$$R_b = 0.12\ \text{K m W}^{-1}$$

$$r_b = 0.063\ \text{m}$$

Firstly, let us simulate the annual average long-term base load (1.027 kW) over a 30-year design life of the heating system (q = 10.27 W m^{-1}, t = 30 years = 947 × 10^6 s). Equation 10.16 predicts that the average carrier fluid temperature will evolve as shown in Figure 10.15), reaching a value of +5.41°C after 30 years. Remember that we can calculate the temperature drop within the borehole itself as qR_b = 0.12 K m W^{-1} × 10.27 W m^{-1} = 1.23°C. This implies a temperature at the borehole–rock interface of +6.64°C. So far, so good.

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[5] The exponential integral function used in Equation 10.16 can be found in an add-in for Microsoft Excel, termed *xnumbers.xla*, which is downloadable from the Internet.

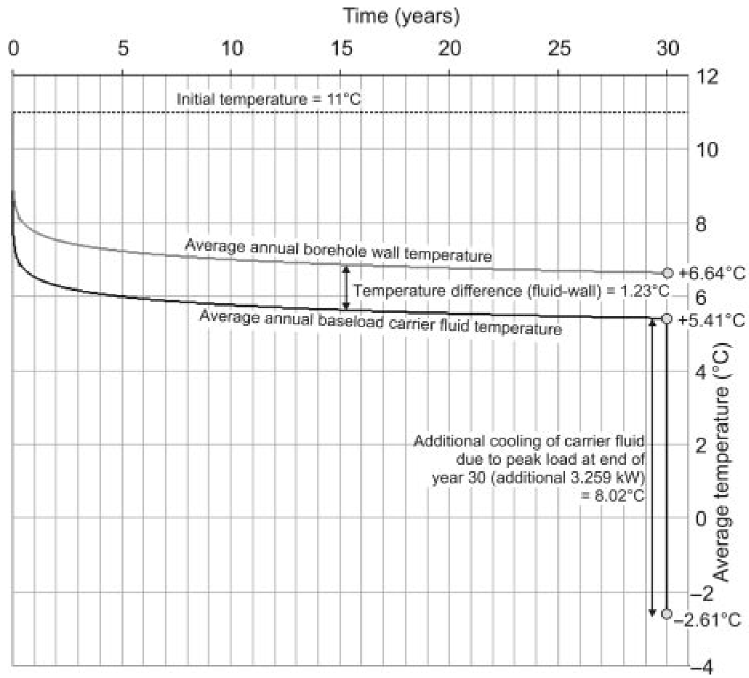

Figure 10.15 The evolution of average carrier fluid temperature over 30 years for a 100-m-deep, closed-loop ground source heat borehole, delivering a peak heat load of 6 kW with a COP_H of 3.5. The solid curve shows the annual average *trend* of carrier fluid temperature (corresponding to a continuous heat extraction of 1.027 kW), while the final downward 'spike' shows the approximate minimum fluid temperature under short-term (24 h) peak heating load, superimposed on the base-load trend after 30 years.

We now need to consider the peak heating condition, namely, on the coldest day of winter, heat is being extracted at the maximum 'peak-load' rate of $q = 42.86\,\mathrm{W\,m^{-1}}$ for a maximum duration of maybe 24 h ($t = 86\,400$ s). Here Equation 10.16 predicts a temperature displacement of 10.55°C, 5.41°C of which takes place in the aestifer and 5.14°C of which takes place across the borehole structure (qR_b). Thus, after 24 h of peak load, our average carrier fluid temperature may have dropped to +0.45°C. This still seems quite acceptable.

However, we must assume the worst: that this peak load takes place on top of the base-load condition at the end of our 30-year design life. In other words, we are imparting an additional 3.259 kW ($32.59\,\mathrm{W\,m^{-1}}$) of heat extraction on the coldest day of the thirtieth year, in addition to our 1.027-kW base load. The additional $32.59\,\mathrm{W\,m^{-1}}$ of peak load for 24 h gives a temperature displacement of 8.02°C. Thus, the minimum peak-load temperature of the carrier fluid after 30 years is predicted to be 5.41°C − 8.02°C = −2.61°C.

In summary, we can see that our borehole is adequately designed; the base-load temperature does not drop below 0°C during its design life and the peak-load minimum temperature remains well above the freezing point of most common antifreeze solutions. Indeed, if we say that our 'minimum peak-load average fluid temperature criterion' is around –3°C, then the borehole design satisfied this criterion (but see Section 10.11).

Note that, for the short-term (24 h) peak-load heat extraction pulse, the temperature drop in the borehole structure is almost as large as the temperature drop in the rock mass (the aestifer). For the long-term base load, the temperature drop in the aestifer is much greater than the temperature drop in the borehole structure. We have thus learned an important lesson: if our heat load is extremely 'peaky' with high heat extraction for short durations, it is extremely important to minimise the borehole thermal resistance. For less 'peaky' heat loads (i.e. a steady continuous extraction of a modest quantity of heat), the impact of borehole thermal resistance is much less important.

10.7.3 Mathematical checks

Let us be rigorous and check our calculation against the constraints of Equations 10.9a and 10.9b. We can calculate the quantity $5r_b^2 S_{VC}/\lambda$ as 5 h. This means that, for times in excess of 5 h, we could have used our logarithmic approximation (Equation 10.19), rather than the exponential integral (Equation 10.16) to solve the problem. Below 5 h, the mathematical assumption underlying the logarithmic approximation breaks down.

We can calculate the quantity $t_s/10$ as 3 years. After 3 years, Equations 10.16 and 10.19 begin to very slightly overestimate the real temperature drop (i.e. underestimate the carrier fluid temperature) as the system begins to induce heat flow from the surface. Thus, a degree of conservatism is built into our approach.

10.7.4 Steady state

At time t_s, the steady-state Equation 10.20 becomes a better approximation of reality than the radial flow Equation 10.16. In our example above, Equation 10.10 predicts that t_s is around 30.3 years. After this time, the temperature evolution flattens out and begins to tend towards the predicted steady-state average fluid temperature $\theta_{s,b}$ corresponding to the 1.027-kW 'base' heat extraction rate. In our example, from Equation 10.20, we calculate that this is also +5.41°C (as the simulation time is approximately equal to t_s). Thus, we can see that our selection of a 30-year simulation time was sensible: not only can it be argued that it corresponds approximately to the foreseeable life of a heating system, but we would anticipate that the system would have approached steady-state behaviour by this time.

We should remember, however, even after time t_s, that our aestifer may still be some way from true mathematical equilibrium! Claesson and Eskilson (1987a,b) provided a rather complex expression to estimate the proportion of heat derived from the surface (i.e. heat replenishment from the soil and atmosphere) at any given time. They indicate that, for a typical closed-loop borehole after 25 years of heat extraction, only some 32% of the extracted heat is derived via the ground surface. The remaining 68% is still derived from thermal storage in the rocks.

10.7.5 A more sophisticated approach

The calculation in Section 10.7.2 simply considered an annual average 'base load', with a superimposed 24-h 'peak load'. In reality, the base load will vary from month to month, depending on the external temperature, and the average fluid temperature in January will be colder than in July. Let us assume that our annual heat extraction of 9000 kWh is distributed as shown in Table 10.1.

Fortunately, we can simulate the average long-term 'base-load' carrier fluid temperatures by superimposing a set of monthly heat signals ('step functions') on top of each other, For example, in January, we can simulate the average carrier fluid temperature by applying Equation 10.16 or 10.19 to a heat extraction 'step' of 20.08 W m^{-1} for a 100-m borehole. At the end of January, we can superimpose on this a second step down of –2.83 W m^{-1} to simulate the slight reduction in load and warming of the fluid temperature in February. At the end of February, a third step of –1.85 W m^{-1} is superimposed and so on. For example, at the end of March, in the first year of operation, the average 'base-load' temperature would be given by

$$\theta_o - \theta_b = (20.08 - 2.83 - 1.85\ \mathrm{kW})R_b + \frac{20.08}{4\pi\lambda}E\left(\frac{r_b^2 S_{VC}}{4\lambda t_1}\right) - \frac{2.83}{4\pi\lambda}E\left(\frac{r_b^2 S_{VC}}{4\lambda t_2}\right) - \frac{1.85}{4\pi\lambda}E\left(\frac{r_b^2 S_{VC}}{4\lambda t_3}\right) \tag{10.25}$$

Table 10.1 An example of a heat-load profile for a hypothetical house, heated by a ground source heat pump of peak output 6 kW, operating 2100 hours per year peak-load equivalent (12 600 kWh output per year), assuming an average COP$_H$ of 3.5. This forms the basis for the example calculation in Section 10.7.5.

	Heat extraction per month (kWh)	Average monthly heat extraction rate (W)	Change in monthly heat extraction rate (W)	Peak-load extraction rate and duration in coldest months
January	1467	2008	2008	4.286 kW, 24 h day^{-1}
February	1260	1725	–283	4.286 kW, 24 h day^{-1}
March	1125	1540	–185	
April	891	1220	–320	
May	576	789	–431	
June	0	0	–789	
July	0	0	0	
August	0	0	0	
September	549	752	752	
October	783	1072	320	
November	1053	1441	370	4.286 kW, 24 h day^{-1}
December	1296	1774	333	4.286 kW, 24 h day^{-1}
Sum	**9000**			

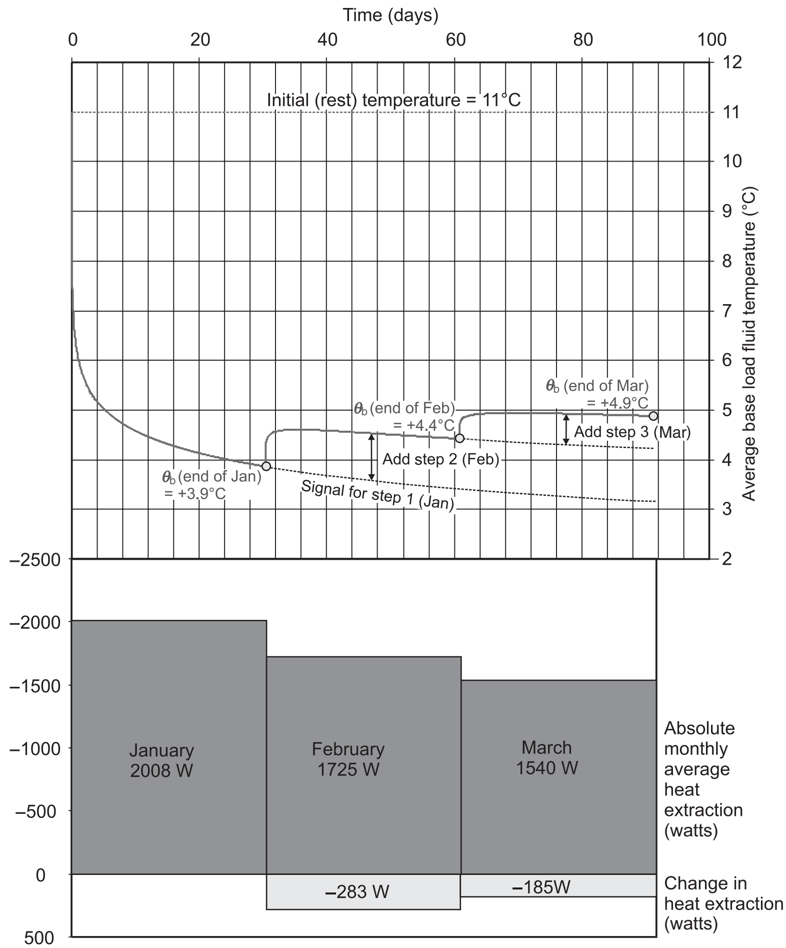

Figure 10.16 The evolution of monthly average ('base load') carrier fluid temperatures in a single, 100-m-deep, closed-loop borehole serving the heat extraction rates shown in Table 10.1. The first 3 months of operation only are shown, simulated by the superimposition of three monthly step functions. See the text in Section 10.7.5 for further information.

where $t_1 = 3$ months, $t_2 = 2$ months and $t_3 = 1$ month. Figure 10.16 shows the simulated evolution of average carrier fluid temperature (as a result of monthly base-load steps) for the first 3 months of the first year of operation of our heat pump scheme. The diagram predicts that the average carrier fluid temperature due to the long-term base load at the end of January is +3.9°C; by the end of February, the temperature is slightly warmer at +4.4°C and, by the end of March, +4.9°C. We can continue and superimpose monthly step functions to simulate the base load throughout the 30-year (360-month) design life of the system. It is by consideration of many such superimposed heat 'steps' that most computerized analytical design models function.

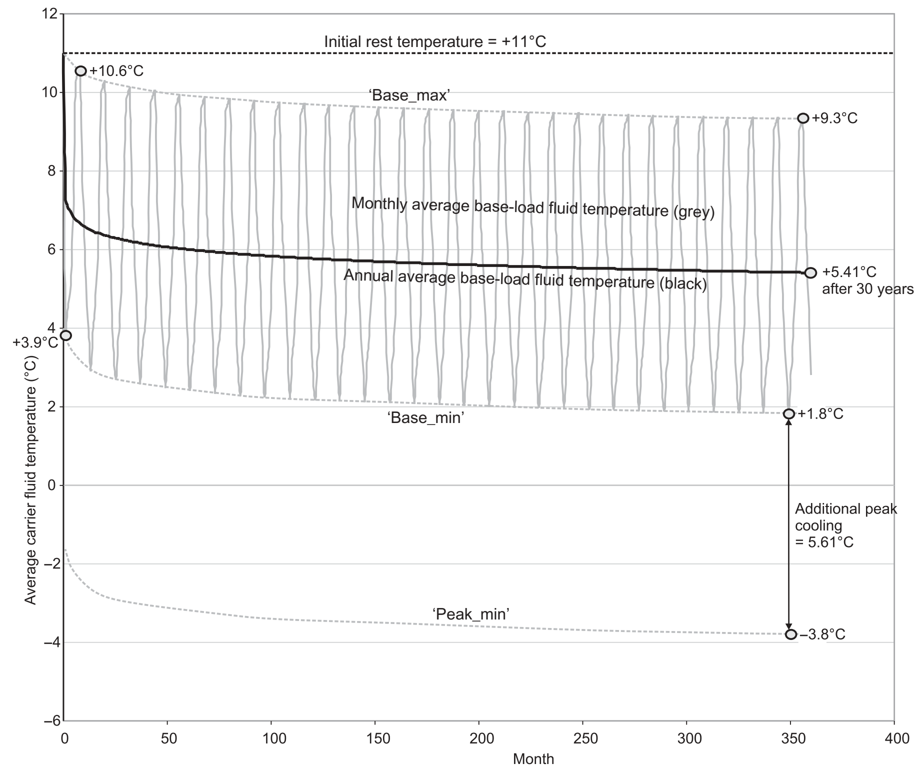

Figure 10.17 The evolution of monthly average ('base load') carrier fluid temperatures in a single, 100-m-deep, closed-loop borehole serving the heat extraction rates shown in Table 10.1, over a 30-year period, simulated by the superimposition of 360 monthly step functions. The worst-case 'peak load' average carrier fluid temperature is simulated by adding a 24-h 'peak load' step at the end of the coldest month (January). The locus of this worst-case 'peak load' average carrier fluid temperature is shown as 'Peak_min'. The bold, black curve shows the average annual fluid temperature (same curve as Figure 10.15). See the text in Section 10.7.5 for further information. Remember that the simulated fluid temperatures are the average of the upflow and downflow temperatures.

Figure 10.17 shows the evolution of the average monthly 'base-load' carrier fluid temperature over a 30-year system design life. We can see that the average monthly 'base-load' carrier fluid temperature is at a minimum in January, varying from +3.9°C in year 1 to +1.8°C in year 30 (we may refer to this as the 'Base_min' temperature, in the parlance of programs such as 'Earth Energy Designer'). The average monthly 'base-load' carrier fluid temperature is at a maximum in August, as the heat pump has not been running for the three summer months; this varies from +10.6°C in year 1 to +9.3°C in year 30 (we may refer to this as the 'Base_max' temperature).

Our critical condition will occur on the coldest day of January in year 30, when the heat pump will be running for maybe 24 h day^{-1} at a rate of 6 kW (extraction rate of 42.86 W m^{-1}). In January, we have already simulated a base-load heat extraction of

20.08 W m^{-1}, so the peak-load condition represents an additional pulse of 22.78 W m^{-1} for 24 h. Such a heat pulse yields an additional temperature displacement of 5.61°C. If we assume that this peak load occurs in January of year 30, we subtract this temperature shift from the average monthly base-load temperature of +1.8°C, yielding a 'worst-case' average fluid temperature of –3.8°C on the coldest day of the coldest month of year 30. The locus of this worst-case condition (January peak load) throughout the system's design life is often referred to as the 'Peak_min' temperature.

This revised estimate of a 'Peak_min' temperature of –3.8°C in year 30 may seem to be rather cold, but we should remember that this is a worst-case condition (at the end of a 24-h peak-load cycle on the coldest day of the coldest month) and for most of the year, carrier fluid temperatures will be significantly higher than this. Indeed, the monthly average temperature in January of year 30 is as high as +1.8°C (this average includes periods when the heat pump is not operating, so the average temperature 'seen' by the heat pump during operation will be somewhere between the 'Base_min' and 'Peak_min' temperatures). Moreover, we should remember that our calculations have considered radial heat flow only; we have not yet considered stabilisation of temperatures due to three-dimensional heat flow (Equation 10.20), which would result in calculated temperatures in year 30 being somewhat higher. Indeed, commercial programs such as EED (Blomberg *et al.*, 2010) take this into account (in fact, if we run this example on EED, we obtain a 'Base_min' temperature of +2.1°C and a 'Peak_min' of –3.5°C in year 30). Having said all this, we can conclude that our 100-m borehole is probably just about adequate to support our 6-kW heat pump and the extraction rates in Table 10.1. We can also begin to see whence our '100-m borehole for a peak heat demand of 4–10 kW' rule of thumb comes from (Section 10.1). But we can also see that a brief design calculation takes a matter of minutes, and we do not really need to rely on 'rules of thumb'. Indeed, we should have become aware of the importance of the assumptions we make about:

- the duration of peak load (12 h, 24 h, 1 week?);
- the percentage of peak load representing average base load (this will depend heavily on the occupancy patterns of the house);
- what we assume to be the minimum acceptable carrier fluid temperatures.

These assumptions will significantly affect the outcome of our design.

10.8 Multiple borehole arrays

10.8.1 Simulations using analytical computer models

Such manual calculations and spreadsheet-based solutions run into difficulties when we are dealing with large, complex ground source heating and cooling schemes (e.g. Box 10.2). In such situations,

BOX 10.2 Case Study: Dunston Innovation Centre, Chesterfield

The Dunston Innovation Centre, Chesterfield. Photo by D. Banks.

The Dunston Innovation Centre is a 3300 m^2 (gross floor area) complex of office and conference space, located near a former colliery site on the northern outskirts of Chesterfield, UK. The site was developed by Chesterfield Borough Council to a high standard of thermal efficiency and is rented to a number of tenants, all of whom will have their own heating or cooling requirements. A distributed ground source scheme was thus selected (Section 7.6.4), using around 97 small reversible console-type water-to-air heat pumps (see Figure 4.9c), producing a flow of warm or cool air. The console units are individually controlled by and metered to the tenants (Climate Master, 2004). The heat pumps are mounted on a loop through which carrier fluid is circulated to the ground array. The central conference facility, in a 'Rotunda', is heated and cooled by a somewhat different philosophy, using a 22-kW reversible water-to-water heat pump unit. Here, heat is drawn from (or rejected to) the ground loop, transferred to a small secondary fluid circuit and distributed to the conference area as warm or cool air by fan coil units.

The ground loop itself comprises 32 boreholes of 60 m depth, drilled through the Carboniferous Coal Measures strata below the site and laid out in a grid beneath a landscaped area adjacent to the building. The scheme supports installed cooling and heating capacities of 242 and 130 kW, respectively, and was commissioned in 2001/02 (Climate Master, 2004; Earth Energy, 2005).

Chesterfield Borough Council anticipated a significant annual saving in costs, compared with traditional air-conditioning and heating systems. Sometimes, it is quite difficult to demonstrate that these cost savings are real, but the Council fortunately has another similarly sized complex, in nearby Tapton, which uses conventional gas-fired heating. The first year's operation revealed that Dunston's

BOX 10.2 (*Continued*)

total energy bill was around £5500 cheaper than Tapton's, despite the fact that Tapton had no air conditioning. If one takes the latter factor into account, together with maintenance costs, one would calculate a realistic payback time on the initial investment. The ultimate test of the scheme is, of course, client satisfaction: both users of the complex and Chesterfield Borough Council appear highly satisfied with the outcome, to the extent that a closed-loop ground sourced approach has also been utilised at the town's new Tourist Information Centre and the new Venture House office complex at Dunston (J. Vaughan, Chesterfield BC, *pers. comm.*).

- there may be a very complicated pattern of heating and cooling throughout the year. In some months, heat may be both abstracted from the ground (in the early morning, for example) *and* dumped to the ground (in the afternoon);
- there will be an array comprising a number of boreholes, which will thermally interfere with each other;
- carrier fluid temperatures may vary significantly throughout the year, implying that we cannot assume a single value of COP (as we did in the example of Section 10.7), as the COP will vary with temperature.

A number of analytical computer programs are available to simulate how ground-loop fluid temperatures evolve in such complex scenarios. Three of the most widely used are

- EED (Earth Energy Designer) from Sweden (Blomberg *et al.*, 2010);
- GLHEPro (Ground Loop Heat Exchanger Professional) from Oklahoma State University, USA (Spitler, 2000). This is mathematically analogous to EED but with slightly more flexible input options;
- GLD (Ground Loop Design) from Gaia Geothermal, USA, which has the advantage of allowing COP values to vary with carrier fluid temperature.

In these programs, multiple borehole arrays are dealt with by factoring the calculations using a so-called 'g-function' – a mathematical function dependent on the geometry and shape of the array. One can simulate lines of boreholes, rectangular blocks of boreholes, 'open rectangles', where boreholes are spaced around the perimeter of an area, and even L-shaped arrays. In fact, the function $E(u)$ in Equation 10.7 is a kind of 'g-function' for a single borehole – it tells us how the thermal resistance of the aestifer surrounding the borehole responds to heat extraction throughout

Table 10.2 An example of a heat-load profile for a hypothetical building that has been used as the basis for examples in Section 10.8. The 256 MWh annual load, represents 1600 h year^{-1} equivalent peak load (160 kW) operational hours. However, the heat pump array is assumed to comprise 4×40 kW units, so the instantaneous peak load in warmer months can be 40, 80 or 120 kW.

Month	Monthly heat load (MWh)	Peak heat load (kW)	Duration of peak load (h)
J	52	160	13
F	37	160	11
M	26	160	8
A	12	120	6
M	10	80	7
J	0	0	0
J	0	0	0
A	0	0	0
S	13	80	8
O	25	120	9
N	33	160	9
D	48	160	12
Total	**256**		

time. Different g-functions will apply to other multiple borehole geometries. Those interested in pursuing the mathematics further can read IGSHPA (1988), Eskilson (1986) or Claesson and Eskilson (1987c).

The complex heating and cooling patterns are simulated using successive 'step functions'. The type of calculation performed in Section 10.7.5 involved superimposing a 24-h 'step' of peak loading on top of a series of superimposed monthly base-load steps. In most analytical simulation models, the base load is typically specified as the heating and cooling load *per month* of a typical year, and is simulated as a combination of sequential monthly steps. One could, of course, also perform such calculations on a daily basis, but for many practical purposes, a monthly base load is regarded as adequate to define the long-term trend. Moreover, for each month, the magnitude and duration of the peak heating and cooling loads is specified for the coldest and hottest days. A load input file for EED or GLHEPro may therefore look something like Table 10.2: in this example, the scheme has a peak output of 160 kW. During the coldest months, however, 52 MWh month^{-1} of heat is supplied (corresponding to 71 kW on average, implying 10- to 11-h peak-load equivalent operation per day on average. A continuous peak load of 13 h day^{-1} is assumed as a worst-case criterion). If the seasonal performance factor of the scheme (SPF_H) is 3.5, then 71% of the heat supplied comes from the ground (i.e. 37.1 MWh of the monthly heat load for the coldest month), and 29% from electrical energy input.

Given a peak heating load of 160 kW, we might make a first guess that $160\,000\,\text{W}/60\,\text{W}\,\text{m}^{-1} \approx 2667$ m of drilled borehole would be required, or 27×100 m boreholes. Let us use EED (Blomberg *et al.*, 2010) to simulate the evolution of carrier fluid temperatures in an array of 27 boreholes, given that

- the aestifer is an arkosic sandstone with thermal conductivity of 2.9 $W\,m^{-1}\,K^{-1}$ and specific heat capacity of 2.0 $MJ\,m^{-3}\,K^{-1}$;
- the boreholes are 133 mm in diameter and 100 m deep;
- the U-tube is of 40-mm OD SDR 11 HDPE and the shank spacing is 70 mm;
- the boreholes will be backfilled with a thermal grout of conductivity 1.8 $W\,m^{-1}\,K^{-1}$ and a contact resistance of 0.01 $K\ m\ W^{-1}$ will be applied in EED;
- the carrier fluid is 25% ethylene glycol, flowing at a rate of 20.3 $L\,min^{-1}$ = 0.34 $L\,s^{-1}$ per borehole;
- the average ground surface temperature is 11°C, with a geothermal heat flux of 65 $mW\,m^{-2}$ (this allows the initial average temperature of the ground θ_o to be calculated).

The program calculates a Reynolds Number of 2686 and a borehole thermal resistance of 0.106 $K\,m\,W^{-1}$. It further calculates the average initial ground temperature θ_o (for the first 100 m depth) to be 12.12°C. If we assume that the 27 boreholes are arranged in a 3 × 9 grid, with a spacing of 5 m, EED predicts the carrier fluid temperature evolution depicted in Figure 10.18. The envelopes 'Base_min' and 'Base_max' bracket the base-load average fluid temperatures; the 'Peak_min' curve represents the limit of minimum temperatures under peak loading in the coldest month. Of course, even these curves are only temperature *trends*; the real fluid temperatures will exhibit a complex pattern of diurnal fluctuations representing the daily switching on and off the heat pumps. The 'base-load' curves show the general trend of fluid temperatures;

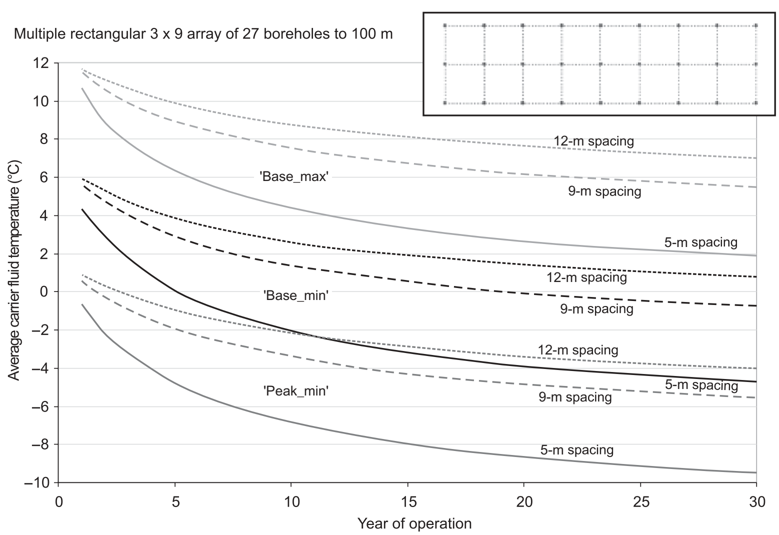

Figure 10.18 The influence of borehole spacing on performance. The evolution of carrier fluid temperatures over a 30-year period in a 27-borehole closed-loop array, drilled to 100 m on a 3 × 9 grid, with a peak heat output of 160 kW. Borehole spacings range from 12 to 5 m. See text for further details. The inset shows the borehole array shape in plan.

the 'peak-load' curve represents the minimum fluid temperatures under 'worst-case' conditions.

We can see that, for a 5-m spacing, at the end of the 30-year period, typical 'base-load' carrier fluid temperatures vary between –4.7°C in winter and +1.9°C in summer. In January, under peak-loading conditions, average carrier fluid temperatures may drop as low as –9.5°C. Remember, this figure is an average of the downflow and upflow temperatures: the downflow temperature may be 1.5–2°C colder than this. It seems clear that our system is way too cold – far lower than the minimum temperatures suggested in 10.7.1 – probably resulting in hugely inefficient heat pump performance and ground freezing. This illustrates the danger of using 'rules of thumb' to estimate numbers of boreholes required in multiple arrays. It also demonstrates the effect of thermal interference, when many closed-loop boreholes are spaced close together.

10.8.2 Effect of borehole array spacing

We could try increasing the spacing of our boreholes to minimise thermal interference. Figure 10.18 shows that, with a spacing of 9 m, 'Base_min' average carrier fluid temperatures in year 30 reach around –0.7°C, and peak load temperatures reach –5.5°C. This is certainly a significant improvement, but still somewhat too cold for comfort.

We have to increase our borehole spacings to around 12 m in this example before we approach a condition which begins to seem acceptable. In this case (Figure 10.18), 'Base_min' average carrier fluid temperatures in year 30 remain above freezing point at +0.8°C, while peak load temperatures do not drop below –4.0°C.

We have thus learned an important lesson: *when designing multiple borehole arrays to serve loads that are 'heating only' or strongly heating dominated, we need to maximise borehole spacing to minimise thermal interference and maintain acceptable carrier fluid temperatures*. The same applies for systems serving loads that are 'cooling only' or strongly cooling dominated. The designer should probably try and achieve borehole spacings of at least 10 m in order to keep thermal interference within reasonable limits. The implication of this is that we require large areas of land for such arrays and we also need substantial lengths of expensive header pipes and trench to connect the widely spaced boreholes. Yet again, we see the inherent conflict between keeping capital costs low and designing efficient systems. Unfortunately, in today's urban areas, land prices are very expensive and we may have a limited area within which to work. We may be forced to live with a spacing of <10 m and we may find that we have to compensate by drilling deeper boreholes to deliver a system capable of maintaining a given output.

We will see, in Chapter 14, that if our heating and cooling loads are well balanced over an annual cycle, we can get away with much smaller borehole spacings, as each borehole only needs to access 1 year's worth of thermal storage in the adjacent rocks, instead of 30 years' worth!

10.8.3 Effect of borehole array geometry

We have seen, in the above example that, unless we can achieve large borehole spacings, an array of 3 × 9 = 27 boreholes to 100 m is probably a little small to support the heating loads in Table 10.2. We could (if we are feeling rich) decide to drill an array of 36 boreholes, on a 4 × 9 grid, at a spacing of 10 m. The program Earth Energy Designer (Blomberg *et al.*, 2010) calculates that, for such an array, we only need to drill the boreholes to 88 m to maintain 'Peak_min' temperatures above –3°C over a 30-year design life (Figure 10.19). In this case, the average January carrier fluid temperature ('Base_min') does not drop below +1.1°C. This array of boreholes measures 80 × 30 m = 2400 m^2 and has a total depth of 3168 m (50.5 W m^{-1} specific peak output).

We could consider other ways of organising our 36 boreholes to 88 m. We could choose a 6 × 6 array. This has dimension of 50 × 50 m (2500 m^2) and results in similar, though slightly lower carrier fluid temperatures – 'Base_min' = +0.8°C and 'Peak_min' = –3.2°C in year 30. Such an array is compact, but the boreholes in the centre of the array do not have access to a large volume of rock to act as a source of heat. The central boreholes can only 'see' the surrounding boreholes and thus will strongly interfere with them.

Alternatively, we could choose a much more 'open' array geometry, such that each borehole is exposed to an unexploited volume of rock to act as a heat source. In an elongated 2 × 18 array, each borehole is exposed, on at least one side, to a huge thermal rock store, unexploited by other boreholes. This array performs much better and thermal interference is much lower (Figure 10.19). 'Base_min' is only +2.5°C and 'Peak_min' = –1.5°C in year 30. Even better is an 'open' 10 × 10 rectangular configuration of 36 bores. Here each borehole 'sees' a largely unexploited rock mass on both sides and thermal interference is minimised. Here 'Base_min' is +3.6°C and 'Peak_min' = –0.4°C in year 30.

Hence, we have learned a second lesson regarding good design: *when designing multiple borehole arrays to serve loads that are 'heating only' or strongly heating dominated, we need to select 'open' array shapes (linear arrays or open rectangles) to minimise thermal interference and maintain acceptable carrier fluid temperatures, rather than compact square arrays*. The same applies for systems serving loads that are 'cooling only' or strongly cooling dominated.

If we really wanted to optimise a linear array, we could drill angled boreholes (Figure 7.16) with alternate boreholes drilled in opposite directions. Here we are minimising the thermal interaction between adjacent boreholes. Despite the small surface footprint of such a borehole array, it accesses a very large subsurface volume of rock (Figure 10.20).

We will see, in Chapter 14, that if our heating and cooling loads are well balanced over an annual cycle, we can use much more compact, closed array shapes. Indeed, if our objective is to store heat or coolth from season to season, such closed, compact shapes may actually be an advantage.

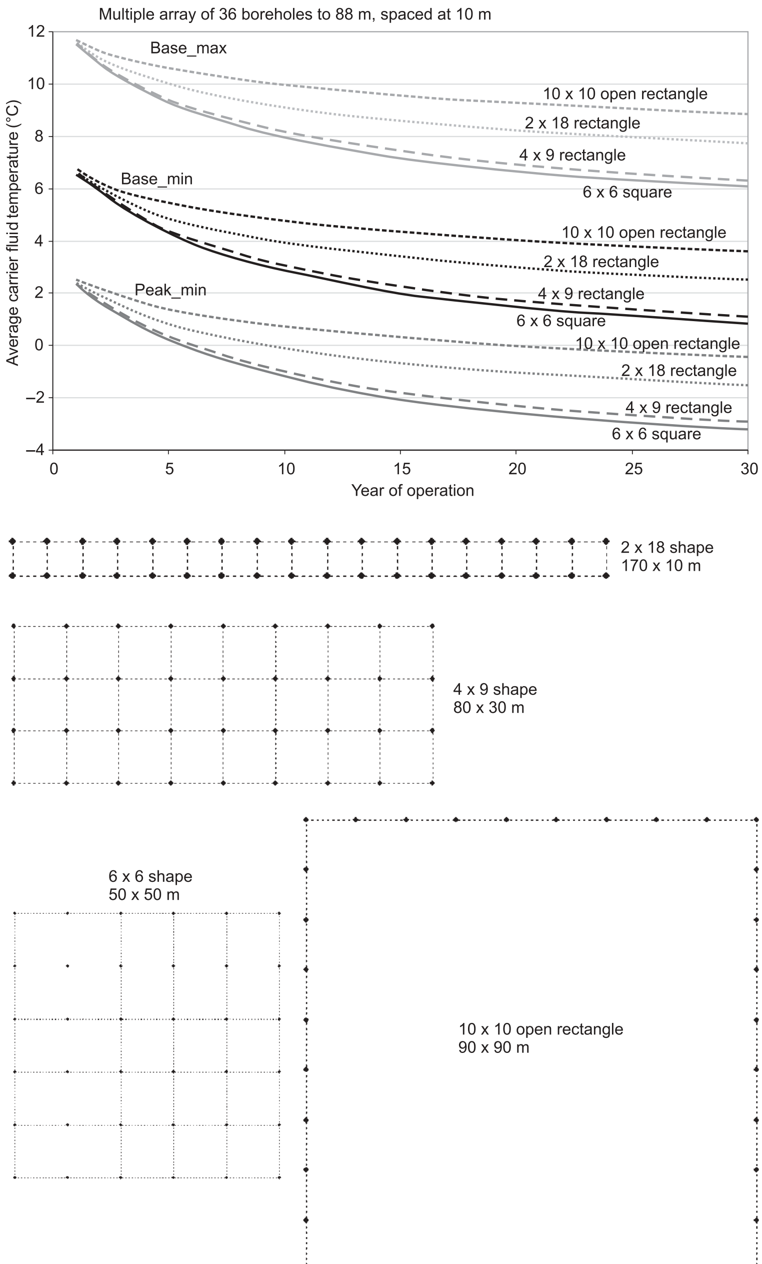

Figure 10.19 The influence of borehole array geometry on performance. The fluid temperature evolution in a 36-borehole array, drilled to 88 m and supporting a heat pump peak output of 160 kW. Rectangular arrays (2 × 18, 4 × 9 and 6 × 6) are shown, as is a 10 × 10 open rectangular array, all with borehole spacings of 10 m. The array shapes are shown below the main diagram.

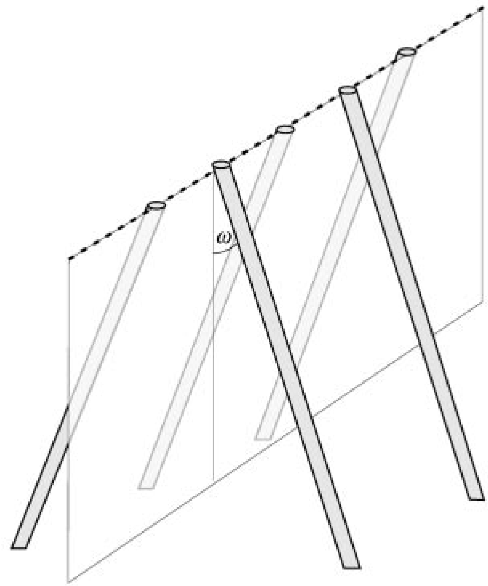

Figure 10.20 An 'open' linear borehole array optimised for continuous heat extraction or rejection. Boreholes are drilled at an angle ω to the vertical, with alternate boreholes in opposing directions. This minimises thermal interference and allows a large volume of rock to be accessed from a narrow surface footprint.

10.9 Simulating cooling loads

Thus far, we have considered situations where heat is only extracted from a closed-loop array. We can also use the techniques described above to simulate the delivery of a cooling effect by 'dumping' waste heat into an aestifer. We can simply substitute a 'negative' value of q into Equation 10.16, 10.19 or 10.20. In this case, of course, the temperature of the ground and carrier fluid rises with time. Eventually, we would hope that a steady state would be reached, where the heat loss via the borehole array is balanced by a heat flux from the ground *to* the surface and the surrounding rocks. Our job is to design a borehole array that allows the heat flux to be dumped without excessive temperature rises in the ground or the carrier fluid. Instead of talking about 'Base_min' and 'Peak_min' temperatures (Section 10.7.5), we will now base our design on 'Base_max' and 'Peak_max' temperatures.

What do we mean by 'excessive'? In the case of heating systems, there are obvious lower temperature limits imposed by the desire not to cause widespread ground freezing or freezing of the carrier fluid. In cooling mode, there are no such obvious upper limits. True, excessive ground heating can cause a small amount of ground movement by thermal expansion (see Chapter 16), but in practice, our upper acceptable temperature limit is more likely to be determined by efficiency factors: the hotter the carrier fluid, the less efficiently our heat pump will operate. Indeed, we may suspect that if we are operating with carrier fluid temperatures consistently in the upper twenties or thirties degrees centigrade, any competitive advantage that a ground 'sink' heat pump has over an air 'sink' heat pump (i.e. conventional air conditioning) begins to be eroded (see Chapter 4). Above 30°C, the installer should also certainly begin to check if the performance warranty of the HDPE pipe is valid, while above 40°C, the installer will probably wish to consider using alternatives to conventional HDPE for the underground heat exchanger, as its performance (pressure resistance) deteriorates as temperature increases.

Kavanaugh and Rafferty (1997) suggested that carrier fluid temperatures should typically be between 5 and 11°C below the ground temperature in heating mode and 11–17°C above it in cooling mode. There are no hard and fast rules, but the author would tentatively suggest maximum base load ('Base_max'; i.e. monthly average) carrier fluid temperatures of around 25°C for cooling-dominated ground source systems in temperate climates, with peak-load maximum temperatures ('Peak_max') of up to 30°C.

Note that, in UTES systems, where the ground is being deliberately used to store waste heat from the summer (as opposed to merely dissipating it into the surrounding ground), higher carrier fluid temperatures may be defensible or even attractive (see Chapter 14).

10.10 Simulation time

We can see from the examples of Sections 10.7–10.8 that the majority of evolution in temperature usually occurs in the early years of the closed-loop system's operation. Nevertheless, we have also seen (Section 10.7.4) that it can take more than 30 years before we achieve anything approaching a steady-state condition. Analytical computer codes such as EED offer us the possibility of specifying the time period that we wish to simulate. Architects design our buildings to have a lifetime of at least several decades (and hopefully more), while a heat pump may have a useful life of 20 years or more. It seems reasonable then to run our simulations for a period of at least 25–30 years.

For some large, multiple-borehole systems, which are heavily 'unbalanced' (i.e. dominated by either heating or cooling loads), the system may never approach steady state: the ground will continue to cool down (heating systems) or heat up (cooling systems), as it cannot exchange sufficient heat with the surrounding rocks or with the surface. Take a look, for example, at the 6×6 array in Figure 10.19, which has not really reached any clear 'steady state' after 30 years of simulation. In such a case, we should consider very carefully the simulation time for our models and not simply select an arbitrary 30-year cut-off for the simulation. We may also wish to consider whether we can find alternative sources of heat or 'coolth' by means of which we can bring our ground heat exchange system more into balance, thus enhancing its sustainability. More on this in Chapter 14.

10.11 Stop press

I have several times, in the course of this book, indicated that design standards are being pushed upwards, in part due to bad press caused by under-dimensioned and underperforming GSHP systems. As this book is being completed, the UK Department of Energy and Climate Change has issued a new set of standards (MIS, 2011a,b,c), which are also reflected in the UK Ground Source Heat Pump Association's standards (GSHPA, 2011). These recommend that GSHP systems should be designed such that the carrier fluid entry temperature to the heat pump should never drop below 0°C over a 20-year design life. Assuming a differential of 3°C across the evaporator, this implies that the 'Peak_min' mean carrier fluid temperature should not drop below –1.5°C over a 20-year simulation. This is even more rigorous than the 'Peak_min > –3°C over 30 years' criterion tentatively suggested in this chapter and is, indeed, more rigorous than most other international guidance. The result will be even more drilled metres of heat exchanger (or more efficient borehole constructions) to support a given heat load and, hopefully, higher SPF_H values. In more specific terms, it implies the following *specific peak heat extraction rates* as initial guidance for design of *small* ground source *heating-only* systems (MIS, 2011b) for typical ground temperatures of 10–11°C:

For poorly conductive formations ($\lambda = 1.5\,\mathrm{W\,m^{-1}\,K^{-1}}$), around 30 and 25 $\mathrm{W\,m^{-1}}$ of borehole length for 1800 and 2400 equivalent peak load hours, respectively.
For more conductive formations ($\lambda = 2.5\,\mathrm{W\,m^{-1}\,K^{-1}}$), around 40 and 35 $\mathrm{W\,m^{-1}}$ of borehole length for 1800 and 2400 equivalent peak load hours, respectively.
For highly conductive formations ($\lambda = 3.5\,\mathrm{W\,m^{-1}\,K^{-1}}$), around 49 and 43 $\mathrm{W\,m^{-1}}$ of borehole length for 1800 and 2400 equivalent peak load hours, respectively.

BOX 10.3 The Hellström Efficiency

Some 'dodgy' suppliers of ground heat exchange pipe have a tendency to claim that, if only we can find an efficient enough type of U-tube, we can extract unlimited amounts of heat from the earth and achieve enormous heat pump COPs. While it is definitely advantageous to reduce the borehole thermal resistance R_b, such claims are clearly nonsense. We are inducing the flow of heat to the borehole via two thermal resistances in series: the thermal resistance of the ground and the borehole thermal resistance. Even if we could reduce the latter to zero, we would still have the former to contend with (and which we can do little to alter).

To assist us in understanding this, the Swedish thermophysicist, Göran Hellström, defined a ratio (the Hellström efficiency, η_H), which is essentially the heat extraction rate achievable from a real borehole, divided by the heat extraction

(*Continued*)

BOX 10.3 (*Continued*)

rate from an ideal borehole. Thus, if $\eta_H = 100\%$, then R_b is negligible and the heat extraction rate is solely determined by the thermal resistance of the aestifer. Of course, the heat extraction rate achievable will also depend on the pattern of heat extraction throughout an annual cycle and on the thermal properties of the ground. For typical European geology ($\lambda = 2.5\,W\,m^{-1}\,K^{-1}$) and heat extraction patterns, it is claimed that an η_H of 100% corresponds to a *specific peak heat extraction rate* of around $85\,W\,m^{-1}$ of borehole depth. It is also claimed that the most efficient heat exchangers (triple metal U-tubes with thermally enhanced grout) correspond to an η_H of slightly over 80%; double HDPE thermally grouted U-tubes correspond to η_H around 70%, while single, thermally grouted HDPE U-tubes correspond to an η_H of around 60% (Sanner, 2011).

STUDY QUESTIONS

10.1 A 100-m-deep, closed-loop borehole extracts 1.5 kW of heat from the ground. The thermal conductivity of the ground is $2.2\,W\,m^{-1}\,K^{-1}$, its volumetric heat capacity of $2.3\,MJ\,m^{-3}\,K^{-1}$ and the average undisturbed temperature at 50 m depth is 12°C. What is the predicted ground temperature at 50 m depth, 1 m from the centre of the borehole after 1 h of heat extraction, after 1 day of heat extraction, after 1 year of heat extraction and after 15 years of heat extraction? What is the predicted temperature at a 10-m distance and at a 30-m distance? Note: if you are using an Excel spreadsheet, you can find the exponential integral function as an add-in called 'xnumbers.xla'.

10.2 Download the demonstration version of Earth Energy Designer (EED) from http://www.buildingphysics.com/. In the demo version, the properties of the ground are fixed to specific values and you are unable to save your work. However, you can use the program to simulate some of the problems discussed in this chapter. Convince yourself of the importance of the various factors influencing borehole thermal resistance (or 'effective borehole thermal resistance' as it is termed in the EED output file, which breaks the resistance down into a number of component parts – see Box 10.3). Convince yourself of the importance of borehole spacing and array shape for multiple borehole arrays. You could even try entering some cooling loads – and watch those carrier fluid temperatures rise!

10.3 A heat pump has a peak heat output of 6 kW and a COP_H under the relevant operational conditions of around 3.3. The carrier fluid has a specific heat capacity of $3.795\,kJ\,K^{-1}\,kg^{-1}$, a density of $1.052\,kg\,L^{-1}$ and a flow rate of $0.33\,L\,s^{-1}$. If the average temperature of the carrier fluid during operation is –1°C, as calculated by a program such as *Earth Energy Designer*, for example, at what temperature does it enter the heat pump and at what temperature does it leave?

11

Horizontal Closed-Loop Systems

> A rapid increase in heat pump sales . . . is likely as long as energy costs increase faster than wealth.
>
> R.D. Heap (1979)

From Chapters 4 and 7, it should have become apparent that heat pump technology provides us with many ways of extracting useful heat from our environment: from the air, from cows' milk, from wastes and even from permafrost. In Chapter 8, we have seen that, if we have a reservoir of groundwater in the subsurface, we can extract the heat directly from this water. To do this, however, we need an aquifer, an expensive water well and (usually) a submersible pumping system. There is, fortunately, a far simpler way of extracting heat from the ground, which requires none of these things. All we need is a buried closed loop to act as a heat exchanger between the ground and our heat pump. We have seen, in Chapter 10, that we can install this closed loop vertically, in a borehole. There is, however, a much cheaper means of burying a closed-loop heat exchanger in the ground, provided that we own a large enough land area – a pipe buried in a trench. This is what we mean by a horizontal closed-loop system.

We have already briefly discussed horizontal ground loops in Chapter 7. To recap, these comprise some form of buried pipe, conveying a refrigerant (in direct circulation

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

systems) or a carrier fluid (in an indirect circulation system) that extracts heat from or dumps heat to the shallow subsurface. In the rest of this chapter, we will focus on the more commonly used indirect circulation systems.

In order to design and install an efficient horizontal closed-loop ground heat exchange system, we will need the following:

- adequate quantities of land area. However closely we space our pipes and trenches, we still need a sufficient area of land to act as a 'collector' of atmospheric and solar energy for our ground loop (see Section 11.5);
- readily excavable soil down to the depth of burial (typically 1.2–2 m);
- some assurance that the buried collector will not be disturbed by subsequent construction or excavation works (pipes, cables, foundations, etc.).

11.1 Principles of operation and important parameters

The pipe is usually buried in a trench at a depth of some 1.2–2 m and the carrier fluid is usually a solution of antifreeze, circulating under transient-turbulent flow conditions (Chapter 9), in order to ensure efficient heat transfer.

To some extent, a horizontal buried pipe functions in exactly the same way as a vertical closed-loop borehole, rotated through 90°. When a heat pump is connected to it, heat is initially induced to flow in radially from the surrounding subsoil. We are thus initially drawing on a geological thermal 'store' that is progressively depleted and cooled down. In a longer time perspective, we would hope that heat would be induced to flow in from the surface, to replenish the chilled subsoil, and that some form of quasi-equilibrium would be established.

With a deep vertical borehole-based system (Chapter 10), the heat exchanger is typically surrounded by a huge block of more or less thermally conductive rock. It may take years or decades for the influence of the surface to make itself known and for a steady-state condition to be established (Section 10.5.2). With a horizontal trench-based system, we are not drawing on a deep 'block' of rock in the same way that we are with a vertical borehole-based closed-loop system. However, the ground surface is much closer to the heat exchanger: we would expect its influence to be more immediate and for some form of steady-state condition with the atmosphere to be established more rapidly.

In essence, therefore, with a horizontal heat exchanger, we are extracting heat in the winter from the shallow soil around the ground loop and relying on this depleted heat reservoir to be replenished from the atmosphere during the spring/summer seasons. We are, effectively, using the earth's surface as a solar collector and its subsurface as a temporary thermal storage. The question then becomes: how can we most effectively harvest this absorbed solar energy input? The following factors will need to be considered:

- The ground area overlying the loop will need to be large enough to receive sufficient replenishment of solar and atmospheric heat during the summer season. Advection of heat with rainfall infiltrating the ground surface may also be important.
- A sufficient length of heat exchange pipe should be installed in the trench to support the peak heating (or cooling) loads.
- The depth of burial should be optimised, such that the heat exchange pipe is sufficiently isolated from the extremes of winter temperature (and such that an adequate soil thermal store is available as a heat source), but such that heat can be 'recharged' to the ground during the warmer months of the year.
- The ground should be conductive enough to transmit heat efficiently to the loop.
- The contact between the ground and the pipe should be thermally efficient.
- The pipe should be constructed of a material that is durable, tough and sufficiently thermally conductive.
- The carrier fluid should efficiently exchange heat with the loop wall, should not be too viscous, should be of low toxicity (in case of leaks), should have a freezing point below the minimum operating temperature of the system and should ideally not be flammable.

11.2 Depth of burial

The depth of burial recommended for horizontal loop systems by various practitioners is typically in the range of 1–2 m. VDI (2001a) suggested a depth of 1.2–1.5 m for temperate European (German) climates, to ensure complete replenishment of the heat reservoir during summer. The depth of burial should ensure

- that the pipe is isolated from diurnal fluctuations in temperature (which only penetrate a few tens of centimetres – Figure 3.7) and the worst excesses of winter temperature;
- that the pipe is below the usual depth of frost formation in the subsurface due to prolonged periods of sub-zero wintertime surface temperatures (although ice may form around the collector due to heat extraction);
- that the pipe is close enough to the surface for summertime solar radiation and induced heat flow from the surface to replenish the shallow soil's heat reservoir each year.

Deeper burial is recommended in circumstances where (1) the climate is extreme (i.e. the annual temperature 'swing' is high) (see Section 3.5 and Equation 3.11) and/or (2) the thermal diffusivity of the soil or sediment is high (i.e. high thermal conductivity/low volumetric heat capacity). If the thermal diffusivity is high, a thermal signal (e.g. cold winter temperatures) will be transmitted rapidly down to the heat exchange pipe. Shallower depths of burial are possible where the climate is less extreme and where soil/sediment thermal diffusivities are low.

Health and safety considerations may also be important when considering the depth of burial. In the United Kingdom, for example, depths of burial of around 1.2 m have been common, not for any thermophysical reason, but because industry 'best practice' has tended to invoke more onerous health and safety requirements in excavations deeper than 1.2 m.

VDI (2001a) suggested that a plastic warning tape be placed in the backfilled trench, some distance above the collector pipe, to warn future excavators of its presence. VDI (2001a) further recommended that areas underlain by horizontal collector pipes should not be built on.

11.3 Loop materials and carrier fluids

The ground loop can be made up of a number of materials. Copper has, of course, a very high thermal conductivity, but it is rather expensive and is not regarded as being particularly resilient: it may be subject to corrosion and also to damage by, for example, later excavation works. It is therefore normal for the ground loop to be constructed of a plastic. Of the common varieties, polyethene has one of the highest thermal conductivities, while being tough and durable (Table 11.1). Both medium-density polyethene (MDPE) and high-density polyethene (HDPE) can be used in shallow ground loops. As horizontal ground loops are not subject to the same geological pressures as deep vertical closed loops, one can argue that their pressure tolerance can be lower. As ground loops are commonly pressurised under operation to somewhere between 1½3 bars, pipe materials with a pressure rating of at least PN6 (6 bars) should be used. However, due to increased concern over leakage of antifreeze, it is becoming common in many countries for installers to utilise the same SDR 11 (i.e. PN10 to PN16) HDPE pipe as is used for vertical borehole heat exchangers (see Box 9.4).

With horizontal ground loop, we enjoy much more flexibility as regards the geometry of installation and the type of pipe that can be installed. Pipe diameters can range from 40-mm outer diameter (OD), down to diameters 20-mm OD or less. Of course,

Table 11.1 Thermal conductivities of common pipe materials (after the databases of [1]Eskilson *et al.*, 2000; [2]Rosén *et al.*, 2001; [3]VDI, 2001b; and [4]Engineering Tool Box, 2007).

Material	Thermal conductivity ($W\,m^{-1}\,K^{-1}$)
High-density polyethene (HDPE)[2]	0.45
Polyethene (generic)[1]	0.42
Medium-density polyethene (MDPE)[2]	0.4
Polypropene (PP)[1]	0.22
Polybutene[3]	0.22
Polyvinyl chloride (PVC)[1]	0.23
Steel	20[3]; 60[1]; 16–54[4]
Copper[1,2,4]	390–401

different pipe diameters will have different surface areas, heat exchange capacities and hydraulic resistances – all factors that must be considered when designing the system.

As regards carrier fluids, the same range is available as for vertical, borehole-based systems (Chapter 9), and the choice of fluid will be governed by

1. expected temperature of operation (if fluid temperatures are expected to remain significantly above 0°C, water may be considered as a carrier fluid);
2. hydraulic properties;
3. toxicity considerations and local/national legislation.

When installed, the horizontal loop will be subject to a regime of flow and pressure testing (see Chapter 9) and will typically be pressurised under operation to a pressure of 1½3 bars. Horizontal loops, being buried at relatively shallow depth, are more vulnerable to accidental damage than borehole-based vertical loops. The heat pump circuit should be supplied with a pressure-sensitive trip switch. Thus, if a leak does develop in the ground loop such that pressure falls and/or circulation ceases, the heat pump should cut out, preventing freeze damage to the evaporator (and hopefully minimising leakage, too).

As with borehole-based vertical systems, a carrier fluid flow rate should ideally be selected to ensure transient-turbulent fluid flow (Reynolds Number > 2500), at least at times of peak loading. Fluid turbulence optimises heat transfer and minimises the thermal resistance associated with the pipe itself. The pipe dimensions and carrier fluid should be selected to ensure an appropriate Reynolds Number during periods of peak loading, but also to minimise hydraulic pressure losses and excessive pumping costs (Chapter 9).

11.4 Ground conditions

As with vertical borehole-based closed-loop systems, the main factors influencing the performance of horizontal closed loops will include the initial temperature, thermal conductivity and volumetric heat capacity of the ground.

The initial temperature of the ground will be governed by the average surface temperature. However, as the depths being considered are shallow (1–2 m), the average temperature of the ground will vary seasonally. We must thus take into account the annual temperature swing, the mean surface soil temperature and the thermal diffusivity of the soil/sediment when designing our ground loop (see Equation 3.11 and Section 3.5).

11.4.1 Variability of ground properties

As horizontal loops are buried at shallow depth, we are commonly not excavating into solid rock, but into some kind of subsoil, weathered material or recent (Quaternary)

Table 11.2 Thermal conductivities and volumetric heat capacities of common subsoil materials and loose sediments. Note that thermal conductivity and volumetric heat capacity typically increase with moisture content. Thermal conductivity also typically increases with increasing density (compactness). See Figure 11.1 and/or VDI (2008).

	Thermal conductivity ($W m^{-1} K^{-1}$)	Volumetric heat capacity ($MJ m^{-3} K^{-1}$)
Sediments		
Dry sand/gravel	0.3–1.2	1.2–1.7
Unsaturated sand	0.7–2.0 (*1.0–1.4*)	1.6–2.2 (*1.8*)
Saturated sand/gravel	1.7–3.8 (*2.4*)	2.2–2.9 (*2.5*)
Drier clay	0.8–1.2	1.5–1.7
Wet clay	0.9–2.2 (*1.6–1.8*)	1.6–2.8 (*2.4*)
Peat	0.2–0.7	0.5–3.8

Data from *inter alia* Farouki (1982), Sundberg (1991), Clauser and Huenges (1995), Eskilson *et al.* (2000), Banks and Robins (2002), Waples and Waples (2004a), Lienhard and Lienhard (2006) and VDI (2008). Italics show recommended values cited by Eskilson *et al.* (2000) and/or VDI (2008).

sediment. The excavated material will thus typically consist of a sand, gravel, silt or clay (or mixture of these in various proportions). The thermal properties of these are summarised from various sources in Table 11.2.

The proportions of sand, silt and clay in the subsoil can vary dramatically across a single site. Deciding on an appropriate thermal conductivity to use when designing a horizontal ground loop can thus be problematic. We will usually end up making some form of conservative estimate. Fortunately, there are some techniques for determining subsoil thermal conductivities *in situ*; these are described in more detail in Section 15.11. Where thermal conductivity can be measured at various locations in a site, one should probably use the geometric mean of at least 12- to 16-point measurements for the purposes of design (King *et al.*, *in press*).

11.4.2 Moisture content

Excavations of trenches into subsoil materials will usually (but not always) be above the water table. The subsoil or sediments will typically be only partially saturated with water and, moreover, the moisture content may vary from season to season (in temperate climates – high during winter, low during summer). The thermal conductivity and volumetric heat capacity of sediments varies strongly with moisture content (Figure 11.1). Design of horizontal ground-loop systems is thus complicated by the fact that the thermal properties of the ground will vary throughout the year.

Purely theoretically, we could consider a quartz sand, with 25% porosity. If it is perfectly dry, we could calculate the volumetric heat capacity as the volume-weighted arithmetic mean of its constituents: quartz ($1.9 MJ m^{-3} K^{-1}$) and air (negligible). This gives us a value of $1.4 MJ m^{-3} K^{-1}$. If we allow that its thermal conductivity can be roughly estimated as the geometric mean of the constituents (using values from Table

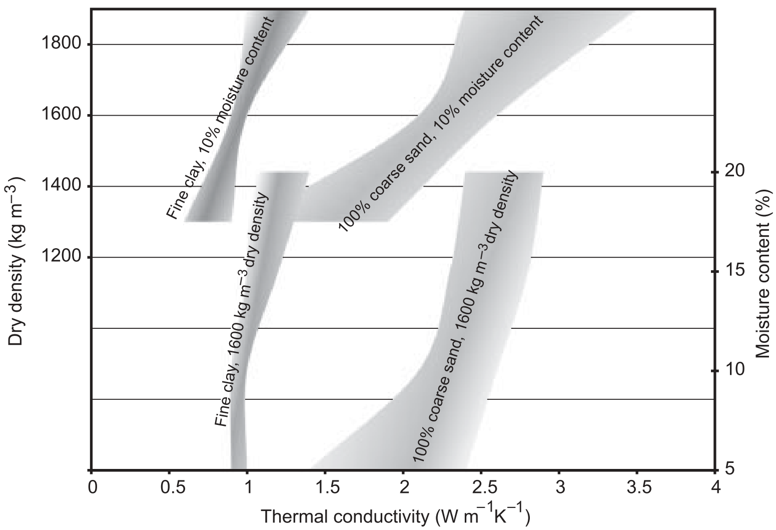

Figure 11.1 How the properties of a sand and a clay vary with dry density (in the range of 1300–1900 kg m^{-3}, upper part of diagram) and moisture content (in the range of 5–20%, lower part of diagram), based on data provided by Farouki (1982).

3.1), we obtain 1.8 W m^{-1} K^{-1}. If the sand becomes saturated with water, these values increase to 2.5 MJ m^{-3} K^{-1} and 4.1 W m^{-1} K^{-1}, respectively. If the saturated sand then freezes such that pores become filled with ice, the values are recalculated as 1.9 MJ m^{-3} K^{-1} and 5.3 W m^{-1} K^{-1}. These are high values, but remember that we have selected an unrealistically pure quartz sand for this example. Furthermore, although the geometric mean is a reasonably simple means of estimating the thermal conductivity of a composite material, it is far from ideal. A preferred method (the Russell equation) is presented by Russell (1935) and discussed by Spencer *et al.* (2000) – and even they admit that this rather complex formula has its limitations. Further discussion of methods of calculating bulk conductivities and of the thermal properties of porous materials is provided by Clauser and Huenges (1995), Renard and de Marsily (1997), Renard *et al.* (2000) and Waples and Waples (2004b).

11.4.3 Vapour migration and ice formation

The complexity does not stop with seasonal variation in moisture content. The moisture content (and thus the thermal conductivity and volumetric heat capacity) of the ground can change due to the operation of the ground-loop system itself. For example, if heat is extracted from the ground, moisture may condense on the chilled ground loop, increasing the moisture content of the ground and its thermal conductivity. Condensation also lowers the water vapour pressure in the soil gas and induces the diffusion of additional water vapour through the soil towards the loop, which in turn condenses. This is termed *vapour migration*. If the loop operates at sub-zero temperatures, some of the moisture condensing on the loop may freeze, to form a halo of ice around the pipe. This again changes (favourably) the thermal conductivity of the ground, as ice is more thermally conductive than water. Ice formation also releases heat in the form of latent heat of freezing (fusion) to the tune of 0.34 $MJ\,kg^{-1}$ of pore water (see Box 3.3). Thus, we should maybe not necessarily worry too much about modest amounts of ice formation, especially in climates where the ground is cold to start with. Indeed, Scandinavian thermogeologists are rather proud of the 'ispølser' (ice sausages) that may form around ground loops, as was John Sumner (1976a) in his day. We should worry, however, if our ground loop is beneath a paved or asphalted area, or is adjacent to a built area, as ice formation can cause expansion and heave of the ground – so-called *frost heave*. We should also worry if we experience large accumulations of ice on the ground loop, as this is a sign that the loop is operating for long periods at sub-zero temperatures and is thus likely to be performing very inefficiently.

In designing our system, one of our criteria should be that any ice lens around the ground loop should not grow to such an extent that it joins up with that around a parallel loop or that it merges with any front of natural frozen ground extending down from the surface during winter frost conditions. If this does occur, we will effectively be trying to extract heat from ice. Moreover, we might doubt whether such an extensive ice lens could be melted by solar warmth penetrating down into the soil during the summer (Figure 3.4). We should also be aware of recent tendencies (see Section 10.11) to encourage relatively high carrier fluid temperatures, in order to improve system performance and carbon efficiency. Recent British guidelines (MIS, 2011a) require that design carrier fluid entry temperatures to the heat pump should always remain above 0°C.

In cooling mode, if we are 'dumping' heat to the ground, the subsoil around the ground loop will warm up. This will have the effect of vaporising moisture, which will then tend to be driven away from the ground loop. The ground surrounding the loop will dry out, its thermal conductivity and volumetric heat capacity will decrease, and the ground loop will perform progressively less efficiently. This is one reason why, although horizontal loops can be designed to accept some component of cooling load, one should be very careful about designing horizontal ground loops for load profiles that are dominated by intense cooling.

11.4.4 Drainage and ground surface

Horizontal ground loop can be installed beneath paved, asphalted or grassed areas. We have seen, however, that moisture content is very important when determining the thermal performance of the ground. In general, a high moisture content is beneficial and we should prefer to install our horizontal ground loops beneath permeable surfaces, in areas where the soil is naturally moist or even in areas that are receiving artificial drainage water. Advection of heat with infiltrating rainfall may also be significant in enhancing heat transfer around the ground loop.

Other aspects that may be important for the thermal performance of the ground loop are the slope of the land and its aspect (i.e. north-facing or south-facing).

11.5 Areal constraints

We will see, in Section 11.6, that there are various means by which we can install heat exchange pipe into the ground. There are some methods that allow us to cram large amounts of pipe into a trench – coiled 'slinkies' for example. We should remember, however, that the objective of a horizontal ground heat exchange system is to ensure that, over an annual cycle, sufficient heat from the sun and the atmosphere enters the overlying ground to replenish the heat that has been extracted. There will thus be an ultimate limit to the areal density of heat extraction. In other words, to extract a given amount of heat over an annual cycle, we will typically require a minimum land area to be underlain by heat exchange pipe (irrespective of the heat exchanger geometry).

This minimum area will depend on climate, terrain, drainage, aspect and other factors. However, in Germany, VDI (2001a) suggested that total annual extraction of heat from the soil should not exceed 50–70 kWh m^{-2} each year (180–250 MJ m^{-2} $year^{-1}$) to ensure long-term sustainability. For a heat pump with a seasonal performance factor (SPF) of 3.5, this equates to the delivery of 70–98 kWh m^{-2} by the heat pump annually. These figures should be seen in the context of typical net solar/atmospheric radiation rates of several tens of W m^{-2}, for temperate European climates (Section 3.4).

Rosén *et al.* (2006) concurred, citing limiting heat extraction rates of 60, 70 and 85 kWh m^{-2} per year for northern, central and southern Sweden, respectively, in order to avoid permanent ground freezing.

If waste heat or surplus heat is re-injected to the ground loop during a cooling season, the total annual availability of heat may be increased.

11.6 Geometry of installation

There is clearly an almost unlimited number of ways of burying a pipe in the ground; so, in this section, we will consider just a few (Figure 11.2). Those interested in some of the theoretical background on how different geometries of ground-loop function should refer to Section 4 of IGSHPA (1988). Those interested in practical, empirical experiences of how horizontal closed loops function, with an emphasis on Scandinavian experiences, should read the volumes by Rosén *et al.* (2001, 2006).

In all cases (at least in temperate European and North American climes), it would appear that if the pipe is buried much more than 2 m deep, the time for 'recharge' of the heat extracted during the heating season may take more than one summer. If the pipe is too shallow however (<0.8 m according to data cited by Rosén *et al.*, 2001), it is suggested that root systems of vegetation may be damaged by the installation.

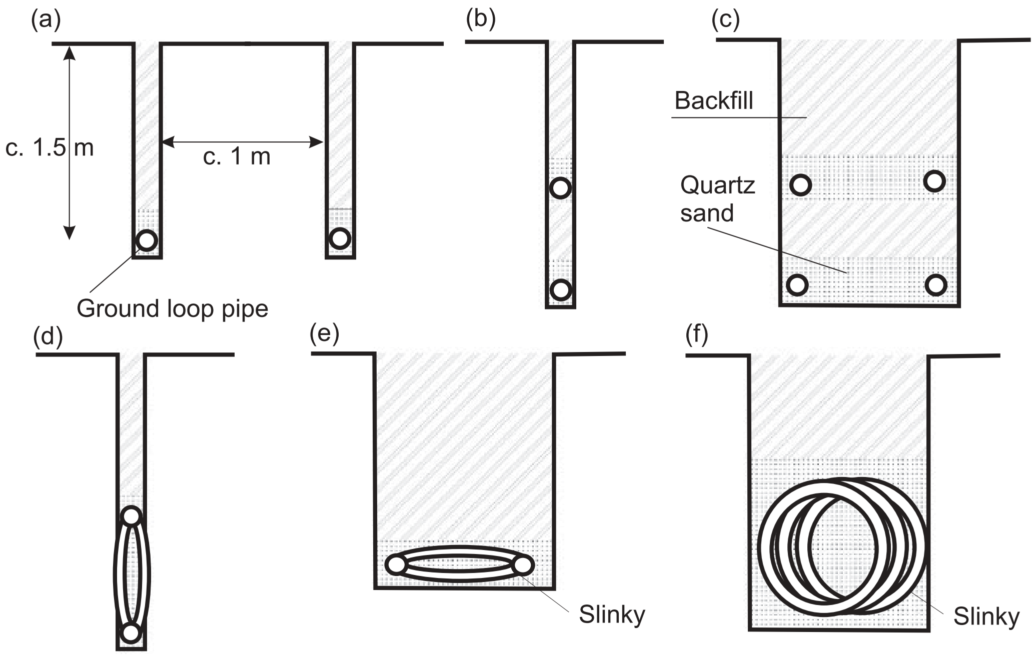

Figure 11.2 Possible configurations of horizontal ground loops in trenches: (a) single pipes in parallel trenches; (b) vertically installed double-pipe system (these may be flow and return pipes, or two separate parallel circuits); (c) 'square' four-pipe system (these may be two flow, two return or four separate circuits); (d) vertically installed slinky; (e) horizontally installed slinky; and (f) three-dimensional spiral slinky. Note that pipes *may* be bedded in conductive quartz sand; although this is not always necessary nor even recommended (see text).

11.6.1 Backfill around pipes

We have seen in Chapter 10 that we can regard the transfer of heat between the ground and the carrier fluid as being impeded by two thermal resistances: (1) a time-dependent thermal resistance associated with the ground itself, but also dependent on the geometry of the closed-loop installation, and (2) an approximately constant thermal resistance associated with the heat exchanger itself. In the case of a horizontal, trenched, heat exchanger, there will be some kind of 'pipe' thermal resistance R_p, which is analogous to the borehole thermal resistance R_b of Chapter 10. This will depend on the pipe material, the carrier fluid, the flow conditions and the degree of thermal contact between the pipe and the ground.

In many cases, it is common practice to simply backfill the trench around and overlying a buried horizontal pipe with the excavated material. This is often acceptable but, in some cases, large clods of soil will leave air gaps or spaces around the pipe, which will be to the detriment of a good thermal contact. This problem of 'bridging' by large clumps of soil can be overcome by using a water jet or water lance to break up clods and wash them down around the pipe during backfilling (Jones, 1995). In some cases, it may be appropriate to backfill around the pipe with a granular, quartz-rich, thermally conductive medium, such as quartz sand (Figure 11.2). Such a sand should ideally be dense, poorly sorted (i.e. a variety of grain sizes) and the grains should not be too well rounded – these criteria ensure a good thermal contact between sand grains and a low porosity. Rosén *et al.* (2006) described a system where a coiled horizontal 'slinky' is underlain by 5 cm of sand, overlain by 10 cm of sand and thereafter the trench is backfilled with the natural excavated material.

Sand backfill may be expensive, however, and may sometimes be detrimental. For example, in dry soils, especially where the loop is being used to reject heat, the thermal conductivity of a dry sand may be extremely poor (Table 11.2). Sand backfills may be considered where we know that the subsoil will remain relatively moist.

11.6.2 Dimensioning 'rules of thumb'

Some international guidelines for dimensioning horizontal ground-loop systems are provided in this section, although the reader should realise that these are essentially 'rules of thumb', whose dangers we have already recognised (Section 10.1) and which may be climate specific (e.g. to the cold Swedish environment).

We will see that we can support a given heat load by burying a single pipe in a trench, two pipes in a trench at different levels, four pipes in a trench and so on (Figure 11.2). We can also cram ever more pipe (and thus increasing areas of heat exchange surface) into a trench by using overlapping coils of pipe – so-called 'slinkies'. It may appear, at first sight, that we obtain significantly more heat per metre of slinky trench (e.g. 50–100 $W m^{-1}$ installed heating capacity) than we do per metre of single-pipe trench (15–30 $W m^{-1}$), because the slinky trench contains far more heat exchange area. However, the pipe coils within a slinky trench, or the individual pipes in a four-pipe trench, all thermally interfere with each other, so we obtain less heat *per metre of pipe* in a slinky trench (or a four-pipe trench) than we do from a single-pipe trench. Furthermore, while we can lay single pipes c. 1 m apart without excessive thermal interference, we must space four-pipe trenches or slinky trenches much further apart to avoid thermal interference, as the heat extraction density is so much higher.

Thus, to summarise, if we wish to support a given heat load using a slinky trench, or a four-pipe trench, as compared with a single-pipe trench, we will require

- significantly less trench length;
- significantly greater quantities of pipe (and hence far greater carrier fluid volume). We may also run into issues of excessive hydraulic head loss when attempting to circulate carrier fluids;
- possibly, a greater number of parallel circuits and thus a higher fluid flow rate;
- significantly larger spacings between trenches.

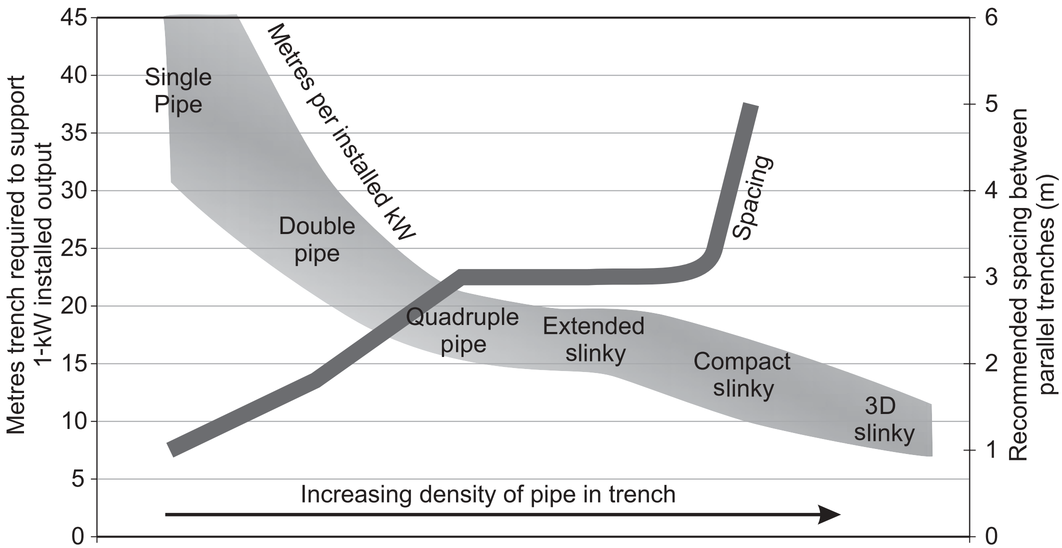

Figure 11.3 The relationship between specific installed thermal output (number of metres of trench required to support 1-kW installed heat pump capacity) and necessary separation between parallel trenches, for different configurations of horizontal ground loop. This figure is intended only to illustrate the inverse relationship between heat extraction density and spacing and should not be used for design purposes.

It turns out that the increased spacing between trenches often approximately balances the reduced trench length required (Figure 11.3 and study questions 11.1–11.3), resulting in similar land areas utilised. Comparative figures for different layout geometries can be seen from the Swedish case studies in Box 11.1. This observation should relate back to the overall areal constraints observed in Section 11.5. For small systems, however, requiring only one or two slinky trenches, the reduced trench length may genuinely translate into a reduced land area requirement.

Thus, the selection of geometry of installation will depend on

- the cost of trenching;
- the cost of pipe and carrier fluid;
- hydraulic factors;
- especially, for small systems, available land area may be a factor.

All things being equal, many installers are adopting a tendency to keep things as simple as possible by installing single-pipe systems where the available land area permits and provided trenching is not prohibitively expensive (Figures 11.4 and 11.5). Single-pipe systems have the advantages of being slightly easier to model, requiring less pipe and carrier fluid and minimising hydraulic problems (i.e. problems of head loss in excessively long pipe runs).

BOX 11.1 Two Swedish Case Studies

Rosén *et al.* (2006) documented two interesting studies of domestic properties served by horizontal ground loops. The first example, at Flistad, comprises an 11-kW heat pump, supported by a 'slinky' installation at 1 m depth in a moist silty–sandy, block-rich moraine subsoil. Six 'slinkies' were installed in six 1-m-wide trenches, spaced 4.5 m apart (centre-to-centre). The total trench length was 215 m (19.5 m kW^{-1} output) occupying 745 m^2 area. The total length of 25-mm OD pipe comprising the slinkies was 1800 m. It is noted that the spacing was conservative and that a lesser land area could have been used. An overall SPF_H of 3.5 was reportedly achieved, with collector fluid temperatures *leaving* the heat pump of between +1 and –3°C in winter, and load-side temperatures of 35–50°C. It was calculated that around 600 m of single 40-mm OD pipe trenches, spaced at 1.2 m, 350 m of double 40-mm pipe trench, spaced at 1.2 m, or 250 m of quadruple 40-mm pipe trench, spaced at 1.5 m would have been required to give a similar heat exchange capacity.

At Motala, a 7-kW heat pump was served by a 'double-pipe' system in a gravelly, silty sand/sandy silt moraine soil. The two pipes, of 40-mm OD, were buried at around 0.55 and 1–1.2 m deep, respectively. The ground loop comprised two parallel circuits (at the two different depths) of 250 m (250 m of trench, 500 m of pipe), arranged in eight trenches, spaced at 1.5–1.8 m. The metres of trench required equates to 36 m per installed kilowatt capacity. In terms of pipe, this represented a figure of 14 W m^{-1} installed thermal output. The installation occupied 357 m^2 (but was again spaced conservatively, and an area of 294 m^2 was calculated to have been adequate). An SPF of 3.7 was reportedly achieved, with collector fluid temperatures leaving the heat pump of between 0 and –4°C in winter, and load-side temperatures of 35–50°C.

It was recognised, however, that the system design in both cases may not have been optimal. The table below shows the optimised designs for single and double pipe and slinky (8-m pipe for metre trench, assumed) installations at Flistad, respectively.

Flistad	40-mm OD single pipe	Double pipe	Slinky
Trench length (m)	600	370	191
Trench length per installed kilowatt (m kW^{-1})	55	34	17
Installed heat output per metre pipe (W m^{-1})	18	15	7
Peak heat extraction per metre of pipe (W m^{-1}). Assumes coefficient of performance (COP) of 3.5.	13	11	5
Trench spacing (m)	1.2	1.6	2.8
Area (m^2)	700	580	520

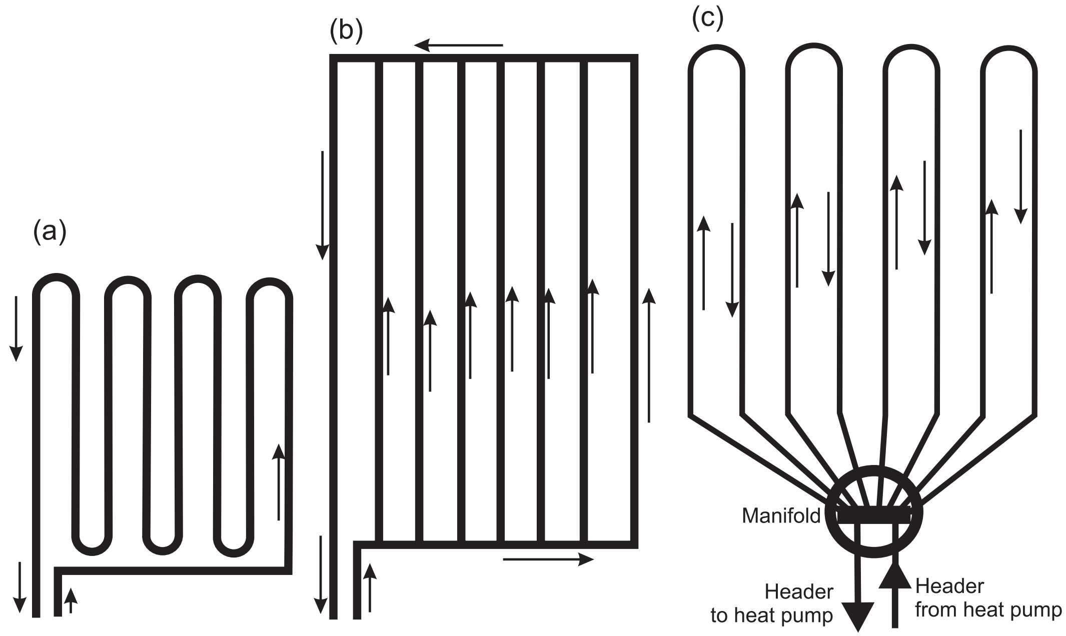

Figure 11.4 Three possible ways of installing parallel trenches of single pipe: (a) in series; (b) and (c) in parallel. In the case of (a), the total pipe run must not be excessively long (to avoid hydraulic head loss) and the trenches will be relatively short.

Figure 11.5 An array of horizontal pipework, comprising individual loops of single polyethene (PE) pipe returning to a manifold chamber (of the type shown in Figure 11.4c), being installed below a British field. The array is designed to support a heat pump installation of 15 kW. *Photo reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

11.6.3 Single horizontal pipe

Rosén *et al.* (2001) cited data from the United States suggesting that each metre of buried single horizontal pipe (Figure 11.3) will support 15 W of installed GSHP capacity in dry soils and up to 30 W in 'normal' soils. They also cited a typical value for 'heating-only' European conditions of 19 $W\,m^{-1}$ pipe. Rosén *et al.* (2001) also noted the generally higher values cited from American practice and suggested that this may reflect a typically shorter GSHP heating season in the United States, and their frequent use for cooling as well as heating (in other words, there will be artificial recharge of 'waste' heat during the summer), which will hasten the regeneration of heat in the soil. It should also be noted that these figures will also depend to a modest degree on the diameter of pipe used. Note that the figures referred to above are '*specific installed thermal outputs*': the installed peak capacity of the system divided by the metres of pipe (or trench).

Other authors cite such 'rules of thumb' as '*specific heat absorption*' or '*specific heat extraction*' rates, that is, the peak rate of heat transfer from the ground to the ground loop. In heating schemes, this value is less than the 'specific installed thermal output' by a factor of $[1 - (1/COP_H)]$. For, example, for 2400 equivalent operational hours per year in south-central Sweden, Rosén *et al.* (2001, 2006) recommended a maximum specific heat extraction rate of 18 $W\,m^{-1}$ for single pipes in damp till, without the carrier fluid entry temperature to the heat pump dropping below −5°C. For dry soils, the equivalent figure is 14 $W\,m^{-1}$, and for wet/saturated soils, the rate is 25 $W\,m^{-1}$.

Often, an installation will consist of parallel rows of buried horizontal pipe at spacings of around 1 m (Figure 11.4), although the details vary from country to country. In Sweden, for example, it is common to use single 40-mm OD polyethene pipe, spaced at 1.2 m to prevent ground freezing (but remember that, in Sweden, the ground is cold to start with). Rosén *et al.* (2001) further recommended that the annual energy extraction from single-pipe systems should not exceed 50 $kWh\,m^{-1}$.

Ochsner (2008a) recommended a typical installation depth of 0.8–1.5 m for horizontal pipes in Europe. He recommended that spacing be calculated on the basis of required land area (Section 11.5), divided by pipe length, but suggests a minimum pipe spacing of 0.5 m in damp well-packed soil and 0.8 m in drier sandier soil. Furthermore, he recommended a collector area in damp, packed soil of 24–36 m^2 per kilowatt installed heat pump capacity ($SPF_H = 3.5$).

At the other extreme, one German company has promoted a system based on parallel circuits of 20-mm OD polyethene pipe, buried at 1.0–1.4 m depth, with a separation of as little as 0.25 m between adjacent pipes. Each circuit should be a maximum of 75 m long and might support a typical peak extraction rate of 750 W (i.e. 10 $W\,m^{-1}$ pipe). It is further claimed that a typical peak extraction rate of up to 50 $W\,m^{-2}$ would be achieved for areas underlain by such a heat exchanger, although these rates will obviously depend on the soil type and the details of the heating load (Waterkotte, 2009).

One of the most widely cited guidance notes is that of VDI (2001a), developed for German conditions. VDI recommends burial depths of 1.2–1.5 m and minimum separations (depending on intensity of energy extraction and soil type) of 30–80 cm. The recommendations for the quantity of installed heat exchanger are based on the *area* underlain by parallel single pipes:

- for normal damp soils, with heating seasons equivalent to 1800 h full operation, the maximum specific heat extraction rate should not exceed 20–30 W m^{-2}. For dry, friable soils, a figure of 10 W m^{-2} is used and for water-saturated sands and gravels, as high as 40 W m^{-2}.
- for a heating season of 2400 h, these figures are reduced to 8 W m^{-2} for dry soils, 16–24 W m^{-2} for normal soils and 32 W m^{-2} for saturated sands and gravels.

These figures form the basis for Figure 11.6a, which allows us to estimate the area required to support a given peak installed output, for given soil conditions and heating season durations. Swiss 'best practice' also supports these figures, with specific peak heat extraction rates ranging from as low as 10 W m^{-2} for dry soils to 40 W m^{-2} for high thermal conductivity substrates (Rosén *et al.*, 2001).

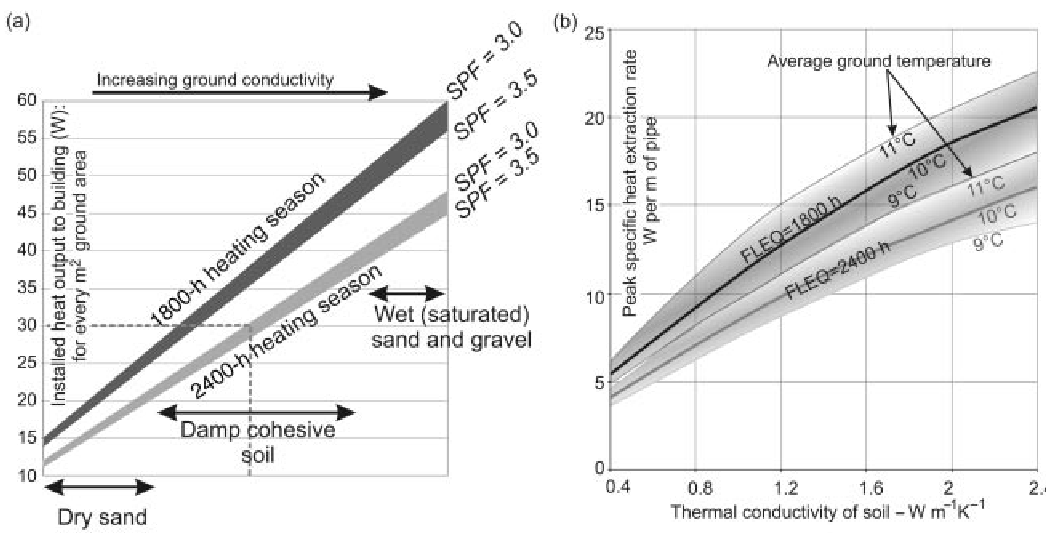

Figure 11.6 (a) Recommended specific installed thermal outputs per square metre for differing ground conditions and different lengths of heating season (FLEQ). *Based on data cited for parallel single horizontal pipes by VDI (2001a) and Rosén et al. (2001).* The dashed line shows an example where, in a damp silty soil, around 30 W of heat pump capacity can be installed per square metre of ground underlain by parallel single pipes, for 2400-h equivalent running hours per heating season and a seasonal performance factor of 3.0. VDI (2001a) appears to assume a typical pipe spacing of 30–80 cm and a depth of 1.2–1.5 m. (b) MIS (2011b) recommendations for peak specific heat extraction rate in the United Kingdom for single parallel 25-mm OD pipes buried at 0.8–1.2 m and spaced at >75 cm, for different annual full-load equivalent hours (FLEQ), a range of typical British soil temperatures and varying soil thermal conductivity. The design criterion is that carrier fluid entry temperature to the heat pump should not fall below 0°C.

The recent MIS (2011a,b) British guidelines recommend specific peak heat extraction rates (i.e. heat absorbed from the ground) of around 7–16 $W\,m^{-1}$ for typical British soils and 1800–2400 h $year^{-1}$ full equivalent load running hours (Figure 11.6b): these rather modest figures reflect the desire for highly efficient operation and with the carrier fluid entry temperature to the heat pump not falling below 0°C.

11.6.4 Multiple pipes per trench

Instead of burying a single pipe per trench, we can bury two pipes per trench (Figure 11.2b), with the pipes at different levels (the lower one being located at, maybe, 1.2–2 m depth). The two pipes can be the flow and return pipes of a single circuit or can represent two separate parallel circuits. American studies tend to be unanimous in concluding that around 50–60% (and even as much as 80%) more energy per metre of trench can be extracted compared with a single pipe (Rosén *et al.*, 2001). Thus, if a single-pipe system has a specific peak heat extraction rate of 18 $W\,m^{-1}$ in a given soil type, a two-pipe trench might have a specific peak heat extraction rate of 27–30 $W\,m^{-1}$ of trench. Rosén *et al.* (2006) suggested that, for two-pipe systems, the trench length can be reduced to 65% of that required for an equivalent single-pipe system.

The reason for this appears to be that, early in a heating 'episode' or season, there is no thermal interference between the two pipes and they effectively extract heat at double the rate of a single pipe. Later, interference occurs as the temperature fields of the upper and lower pipes begin to overlap – this ultimately reduces the heat available per metre of trench. To maintain a long-term equilibrium with atmospheric/solar heat input, Rosén *et al.* (2006) suggested that such double-pipe trenches should be spaced at correspondingly larger intervals in parallel systems (around 1.8 m). The advantage of a double-pipe system over a single-pipe system is that it reduces the total length of ditch required. It increases the length of pipe required, and the volume of collector fluid, however. Double-pipe systems would be particularly effective in situations with a short heating season, such that interference between the two pipes is minimal. An example of a double-pipe system is given in Box 11.1.

Clearly, many other geometric variants are possible, including four pipes per trench either stacked vertically or in a 'square' configuration (Figure 11.2c). Rosén *et al.* (2001) suggested that four-pipe systems yield even greater specific peak heat extraction rates (per metre trench), some 120% to >150% higher than with single-pipe systems. This implies that trench lengths can be reduced to 40–45% of those required single-pipe systems. The spacing between the trenches must also increase, however, with values of 2.5–3.6 m being cited by Rosén *et al.* (2001).

11.6.5 Coiled collectors – 'slinkies'

Clearly, stuffing more and more pipe into a single trench appears to yield dividends in terms of specific heat extraction rate per metre of trench, at least in the short term. The downside of this is

- that there is an inevitability of the 'law of diminishing returns' kicking in;
- the necessity for increased spacing of trenches, as intensity of heat extraction increases, in order to avoid thermal interference (and as the heat available will ultimately be constrained by available land area and insolation);
- that our gains in terms of reduced trench length may be offset by the disproportionately large amounts of pipe and carrier fluid that we require (although, admittedly, polyethene pipe is relatively cheap).

The apparent culmination of this tendency is the 'slinky', which we have already encountered in Chapter 7. The 'slinky' concept was first pioneered by Otto Svec in Canada in the late 1980s. Such configurations of overlapping coils are thus also referred to as Svec coils or 'curtate cycloids'.

Here, large lengths of polyethene pipe are installed in trenches in the form of overlapping or adjacent coils, typically at installed diameters of around, or slightly less than, 1 m (Figure 7.15). There is no wholly standardised design for slinky geometry: slinky lengths and diameters can vary. In order to obtain relatively low diameters for such coils, it may be necessary to use relatively small pipe diameters. For example, while Swedes tend to use 40-mm OD pipe for single-pipe installations, they may need to use 25- or 32-mm OD pipe for slinkies, to avoid overstressing the pipe in the relatively tight coils (Box 11.1). Successive coils may overlap to varying degrees (a so-called compact slinky) or may not overlap at all (an extended slinky), if space is available. An extended slinky extracts less heat per metre of trench, but *more* heat per metre of pipe, as there is less thermal interference between adjacent coils. If we consider Figure 11.7, we will see that

- if the original slinky coil diameter is D^o;
- if the slinky is stretched out to fit in a trench, such that each successive coil is displaced by a pitch (P);

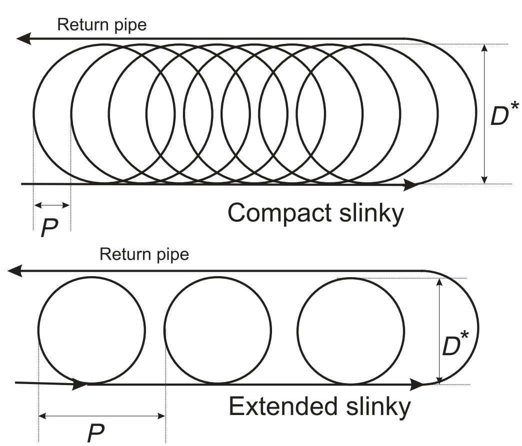

Figure 11.7 Two methods of installing 'slinkies': in compact and extended modes. P is the pitch between successive coils, while D^* is the *in situ* loop diameter.

then the original coil diameter will reduce slightly such that the *in situ* (stretched) diameter D^* is approximately given by

$$\pi D^* = \pi D^o - P \tag{11.1}$$

Furthermore, including the length of the return pipe, but neglecting the length of any headers, the length of trench (L_t) and pitch (P) are related by the formula (Jones, 1995; Rosén *et al.*, 2006)

$$L_t = nP + D^* \tag{11.2}$$

and the length of pipe (L_p) is given by

$$L_p = n\pi D^* + 2nP + \tfrac{1}{2}\pi D^* + D^* \tag{11.3}$$

where n is the number of coils in the trench. For example, let us consider a compact slinky of installed diameter $D^* = 0.8$ m and with a pitch of only 250 mm. If it is installed in a 20.8-m-long trench, there will be around 80 loops of pipe, containing a length of 243-m pipe. In other words, every metre of trench contains over 11 m of pipe (and we are including the return pipe here).

Slinkies can either be installed vertically in a narrow-slot trench (Figure 11.2d) or horizontally in a broader (typically c. 1 m wide) trench (Figure 11.2e). Either way, the 'rules of thumb' for dimensioning do not appear to vary greatly. The depth of installation is typically between 1.2 and 2 m. The spacing between parallel slinky trenches depends on the density of heat transfer: for extended slinkies, a spacing of 3 m between adjacent trenches is likely to be sufficient, while a spacing of up to 3–5 m may be required for compact (overlapping) slinkies (Alliant Energy, 2007) to minimise problems of thermal interference. Rosén *et al.* (2006) were a little more optimistic, suggesting that a minimum spacing of 2.5 m is appropriate for compact slinkies.

Typically, individual slinkies contain around 150–240 m pipe (depending on pipe diameter and carrier fluid type: lengths greater than this are not recommended, due to head losses in very long pipes and difficulties in achieving the flow rates necessary for turbulent flow). Trench lengths are often in the range of 20–30 m for compact slinkies and installed diameters are typically in the range of 0.6–1 m.

According to Alliant Energy (2007), 230–240 m of ¾-inch (19 mm, presumed internal diameter) HDPE slinky pipe, in compact horizontal configuration, with a pitch of 43 cm, installed in a 30-m-long, 1.8-m-deep, 90-cm-wide trench is adequate to support 1 ton of heat pump capacity (3.5 kW). With this American recommendation, we should be aware that the postulated system may support a significant component of cooling demand. For the (usually heating-dominated) systems designed for the British Isles, an average of 10.5 m of slinky trench is typically installed per kilowatt of installed peak output (the values range from 10 to 14 m – Figure 7.11b). Note that these lengths are lower than those calculated for the Swedish examples of Box 11.1 – but remember that Swedish ground is rather cold to start with, and that the Swedish heat pumps performed very efficiently, suggesting generally high fluid temperatures.

The recent MIS (2011a,b) British guidelines are significantly more conservative (see Section 10.11), recommending specific peak heat extraction rates (i.e. heat absorbed from the ground) of around 20–50 W per metre of trench for typical British soils ($\lambda = 0.8$–$1.6\,\mathrm{W\,m^{-1}\,K^{-1}}$); 1800–2400 h year^{-1} full-load equivalent (FLEQ) running hours; 32-mm OD SDR 11 extended slinkies buried at 0.8–1.2 m depth, with ground temperatures of 9–10°C and trench spacing >3 m.

Comparison with single pipe

Rosén *et al.* (2006) considered compact slinkies containing 8-m pipe per metre of trench as typical for Swedish conditions. They reckon that the trench length can be reduced to 38% of what is required for single pipes in dry soils to support a given load. In moist soils, the trench length can be reduced to 34–38% and in wet soils to 30–36%. In other words, the use of compact slinkies can result in trench lengths being reduced to one-third of the length required for single-pipe systems, albeit at the expense of eight times as much pipe.

As regards extended slinkies, Rosén *et al.* (2001) suggested that an extended slinky requires around 45% of the trench length calculated for a single-pipe system.

3D spiral slinkies

It should be noted that we can also install a slinky as a three-dimensional spiral, if we are prepared to construct a large enough trench (Figure 11.2f). According to information cited by Rosén *et al.* (2001) for a climate similar to Ontario, 150 m of slinky pipe can be packed into a 20- to 25-m-long, 60-cm-wide, 1.8-m-deep trench in this manner and can be used to support an installed heat pump capacity of 3.5 kW. This corresponds to only 7-m trench per installed kilowatt.

11.7 Modelling horizontal ground exchange systems

Horizontal systems are inherently less suited to mathematical modelling than vertical borehole-based systems, for several reasons:

- the shallow subsoil is thermally less homogeneous, with moisture content and thermal properties varying rapidly laterally;
- moisture content and thermal properties of shallow subsoils vary seasonally and as a result of heat pump operation;
- the ground loop is thermally affected by the ground surface over a scale of weeks to months (rather than years to decades, as is the case with deep borehole-based systems). Thus, the ground surface cannot be strictly represented as a constant temperature boundary at the annual average soil temperature;

- the geometry of all but the most simple (i.e. single pipe) ground heat exchangers rapidly becomes mathematically complex to describe.

Despite this, mathematical models *are* available to assist in the design of horizontal ground-loop and slinky installations. These can be applied manually, as in the case of the IGSHPA (1988) manual, which gives thermal response functions for many different ground-loop configurations. Computer-aided design tools are also available, although they are not as sophisticated as those used for vertical systems (Spitler, 2005; Gaia Geothermal, 2009).

We can begin to understand how such mathematical models function by simply considering a single heat extraction pipe. In principle, this is identical to our line-source/sink model of Sections 10.5–10.7, merely rotated through 90° (Figure 11.8). As with the vertical borehole model, after the heat pump is initially switched on, the pipe will begin to suck heat radially inward from the surrounding soil, a process that can mathematically be described by Equation 10.7 or 10.16. Later on, the ground heat exchanger begins to experience the thermal impact of the ground surface – that is, it begins to induce a flow of heat from the atmosphere via the soil and to approach a steady-state condition. With a deep borehole heat exchanger, it may take many years or decades before this surface effect is observed (Section 10.5.2). With a shallow trenched pipe, installed only a short distance below the surface, the surface effect may make itself known within a few weeks or months.

The ground surface can, as we have seen, be regarded as a kind of 'fixed temperature' boundary condition (albeit, on the scale of weeks or months, a fixed temperature that varies throughout the year, according to say, a monthly average). If the subsoil is cooler than the soil, heat flows from the soil towards the ground loop; if the subsoil around the loop is warmer than the soil (as it may be in cooling mode), heat is lost to the soil surface.

We can employ a neat mathematical trick to simulate this. It turns out that

- a line heat sink (a heat extraction pipe), extracting heat at a rate q, located at a depth z below a fixed temperature boundary (the ground surface)

is mathematically identical to

- a line heat sink (a heat extraction pipe), extracting heat at a rate q, located at a depth $2z$ below an imaginary 'mirror image' of itself (i.e. a 'virtual' heat rejection pipe, rejecting heat at a rate q, at a height z above the ground surface).

This mathematical sleight of hand is shown in Figure 11.8b. If we apply Equation 10.7 or 10.16 to a situation where a constant rate of heat extraction occurs from the subsurface heat exchange pipe,

$$\theta_0 - \theta_p = qR_p + \frac{q}{4\pi\lambda}E(u) - \frac{q}{4\pi\lambda}E(u') \tag{11.4}$$

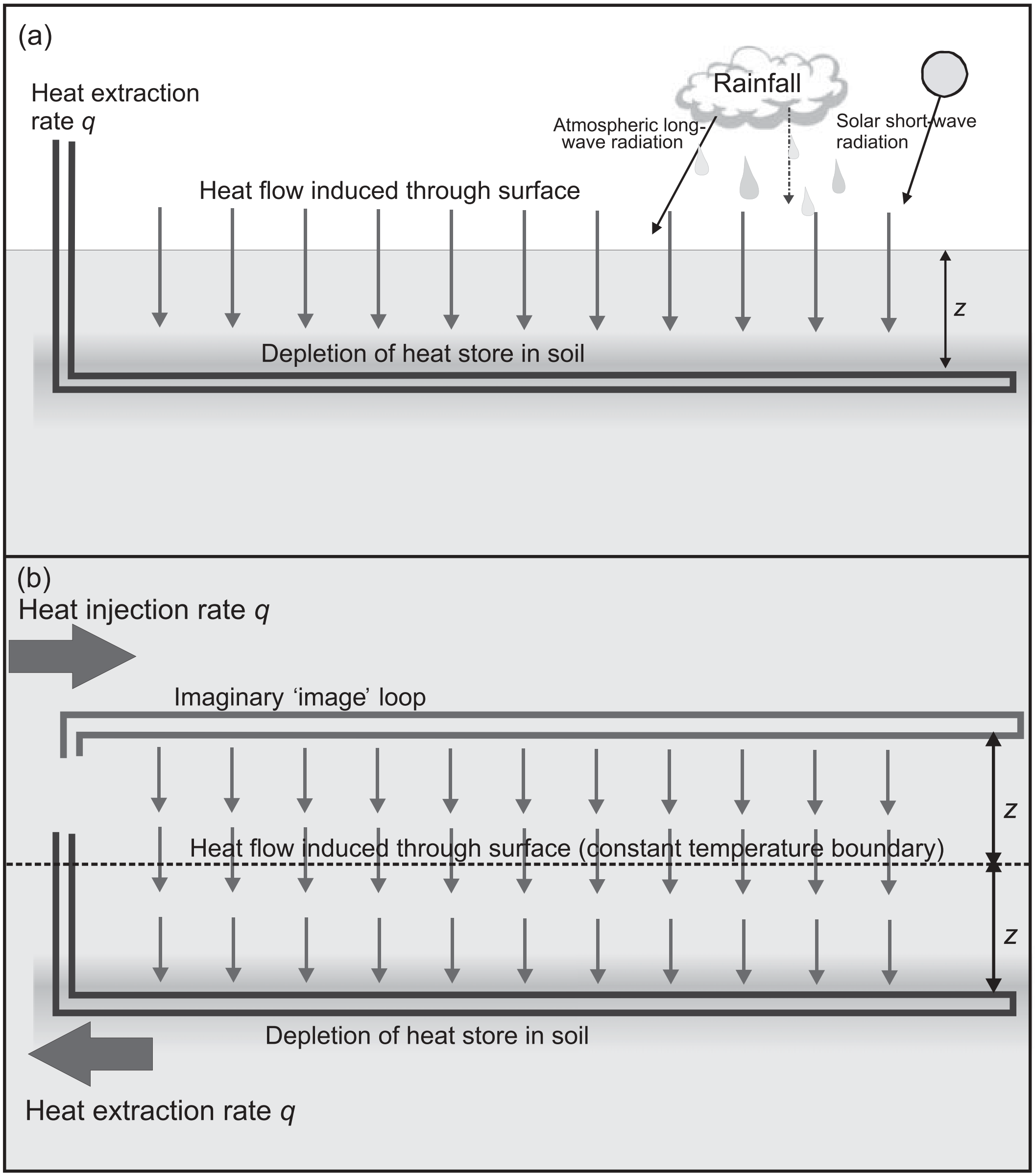

Figure 11.8 (a) The physics of constant heat extraction (q) from a buried heat exchange pipe at depth z. Eventually, the pipe induces heat flow from the fixed temperature boundary, which is the ground surface. This is mathematically identical to (b) the heat flow from an imaginary 'image' pipe, located a distance z above the ground surface, dumping heat to the ground at a rate q.

where

$$u = r_p^2 S_{VC}/4\lambda t \text{ (i.e. the value of } u \text{ for the real pipe)} \tag{11.5}$$

and

$$\text{(for the virtual pipe) } u' = (2z)^2 S_{VC}/4\lambda t \tag{11.6}$$

and where
- θ_p is the average temperature of the carrier fluid in the heat extraction pipe;
- θ_0 is the undisturbed temperature of the subsoil *and* the fixed temperature of the surface (constant temperature boundary);
- q is the constant heat extraction rate from the heat exchanger ($W\,m^{-1}$);
- r_p is the effective radius of the heat extraction pipe (m);
- R_p is some kind of pipe resistance term ($K\,m\,W^{-1}$);
- t is the time after heat extraction commenced (s);
- λ and S_{VC} are the thermal conductivity ($W\,m^{-1}\,K^{-1}$) and volumetric heat capacity ($J\,m^{-3}\,K^{-1}$) of the subsoil.

If we now apply a logarithmic approximation of the type shown in Equation 10.19, for high values of t

$$\begin{aligned}\theta_o - \theta_p &\approx qR_p + \frac{q}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{r_p^2 S_{VC}}\right) - 0.5772\right] - \frac{q}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{(2z)^2 S_{VC}}\right) - 0.5772\right] \\ &= qR_p + \frac{q}{2\pi\lambda}\ln\left(\frac{2z}{r_p}\right)\end{aligned} \tag{11.7}$$

In other words, at high values of t, some kind of equilibrium is reached, where the temperature of the pipe is stabilised by induced heat flow from the surface. Moreover, the greater the depth, the lower that temperature will be.

Figure 11.9 shows the results of Equation 11.4 applied to a specific set of parameters, which may be regarded as relatively typical: $q = 10\,W\,m^{-1}$, $\lambda = 1.6\,W\,m^{-1}\,K^{-1}$, $S_{VC} = 2.0\,MJ\,m^{-3}\,K^{-1}$, $r_p = 20\,mm$ and $R_p = 0.05\,K\,m\,W^{-1}$. The undisturbed temperature $\theta_0 = 11°C$. Figure 11.9 shows the average carrier fluid temperature evolution for different depths of burial z, ranging from 0.6 to 2.0 m. We can see that, for very shallow depths of burial, the loop starts to equilibrate with the surface heat flux after a relatively short time of around 1 week. This is probably too short, if we wish to protect the loop from extreme winter surface temperatures when maximum heat extraction is occurring. At a depth of 1.4 m, the loop starts to 'see' the surface after around 1–2 months, which is regarded as a much more acceptable 'thermal buffer'. At a depth of 2.0 m, the temperature of the carrier fluid continues to decline for 3 or 4 months before the impact of the surface makes itself felt. Thus, we can begin to see whence our recommended depths of burial are derived. Bury the loop too shallowly and, in a severe

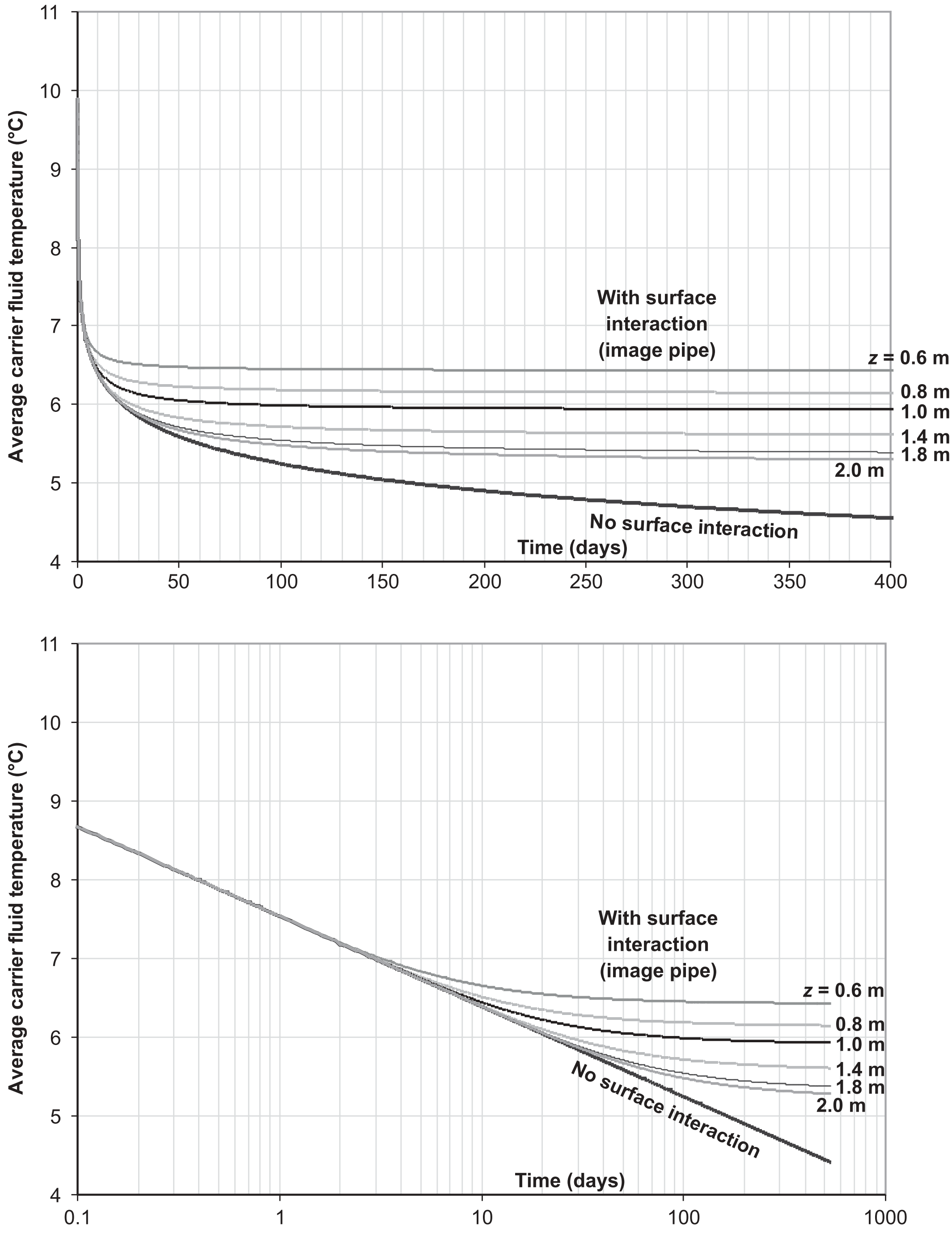

Figure 11.9 The average carrier fluid temperature, simulated using Equation 11.4, in a single heat exchange pipe, situated at varying depths z (ranging from 0.6 to 2.0 m) below the surface. The diagram also shows the temperature evolution curve in a conventional 'line sink' heat exchanger (such as a deep borehole) that does not experience the influence of the surface. The parameterisation of the equation is as follows: $q = 10\,\mathrm{W\,m^{-1}}$, $\lambda = 1.6\,\mathrm{W\,m^{-1}\,K^{-1}}$, $S_{VC} = 2.0\,\mathrm{MJ\,m^{-3}\,K^{-1}}$, $r_p = 20\,\mathrm{mm}$, $R_p = 0.05\,\mathrm{K\,m\,W^{-1}}$. The undisturbed temperature $\theta_0 = 11°C$.

winter climate, the loop is inadequately buffered from the cold winter surface temperatures at the times of maximum heat demand. Bury the loop too deeply, however, and it may take too long for the loop to 'feel' the beneficial effects of thermal replenishment from the surface and temperatures may drop to rather low levels. We can also see that Equation 11.4 gives us a means of mathematically modelling the performance of a horizontal ground loop. The real situation is, of course, much more complex than Equation 11.4 suggests, for several reasons:

- The surface temperature and undisturbed ground temperature are not fixed at a single value θ_0, but will vary (albeit in a known manner) according to the seasons and the depth.
- Ground loops do not experience a constant rate of heat extraction q. However, we have already learned, in Section 10.7.5, to simulate variable heating loads by means of superimposed base-load and peak-load step functions.
- Most horizontal ground loops have much more complex geometries than a single pipe, and may interfere with adjacent heat exchange elements. We have also learned, however, that physicists can derive appropriate 'g-functions' for many different ground-loop geometries (Section 10.8).

It is on the basis of such mathematical treatments that manual (IGSHPA, 1988) and computer-aided (Gaia Geothermal, 2009) design tools are able to simulate horizontal ground loops.

From Box 11.2, we will note that extensive horizontal closed-loop systems can be used for both heating and cooling large building spaces. We should be aware, however,

BOX 11.2 The Green House, Annfield Plain

The Green House, Annfield Plain. Photo by D. Banks.

The Green House is a major 3000-m^2 modern office complex in a former coal-mining area of County Durham, UK. At an early stage, a desire was expressed to

(*Continued*)

BOX 11.2 (*Continued*)

provide space heating and cooling by ground source heat pumps, to be delivered by a combination of underfloor heating and fan coil units.

Initially, the intention was to employ an open-loop system, using the abundant water that often occurs in the flooded, abandoned coal mine workings that are so prevalent in this part of the world (Younger, 2004). Such mine-water-sourced heat pump schemes are not without problems, due to the water's high iron content leading to the potential deposition of iron hydroxide ('ochre') in heat exchangers. Nevertheless, British and international experience demonstrates that mine-sourced schemes are viable (Banks *et al.*, 2004a, 2009b). Unfortunately, a feasibility study suggested (and two boreholes confirmed) that the Coal Measures strata beneath the Annfield Plain site were largely dry to at least 100 m depth. This is most likely due to the presence of an old underground regional mine drainage network that dewaters the abandoned coal mine workings and adjacent strata. The mine water in this drainage system flows northwards under gravity and is eventually discharged to the River Derwent, some 7 km away, at Hamsterley John.

The next possibility was to employ closed-loop vertical boreholes as ground heat exchangers. Trial boreholes were constructed and tested (Fernández Alonso, 2005), although it proved inordinately difficult to grout the ground loop into a borehole that passed through open, abandoned mine workings – the grout seemed to slump into the workings, distorting the loop. The problem could have been overcome by 'casing out' the mine workings, but this would have added significantly to the cost of drilling.

Eventually, a plan was hatched to use most of the available land area around the building to install a huge network of horizontal slinky trenches. It proved difficult to integrate this work with the rest of the construction process, and to protect loops and manifolds from wandering excavators and bulldozers. Eventually, however, the system was commissioned, using fifteen 55-m-long slinky trenches, containing a total of 6500-m pipe, to deliver peak loads of 80-kW heating and 30-kW cooling (2 × 40 kW heat pumps in dedicated heating mode and 1 × 30 kW in dedicated cooling mode). This works out at just over 10-m trench per 1-kW installed capacity. The ground source scheme is integrated with photovoltaic cells and a wind turbine to generate electricity.

that we are relying on 'leakage' of heat to and from the surface in order to achieve this. Horizontal closed-loop systems are not especially good at 'storing' surplus heat from the summer for subsequent use in the winter. In order to use the subsurface as a thermal 'store', we probably need to construct carefully designed, deep, vertical borehole systems (see Chapter 14).

The temperature of the carrier fluid in horizontal ground loops during operation in heating mode is typically in the interval +5°C to –5°C. In Germany, VDI (2001a) made the more specific recommendation that the temperature of the fluid returning from the heat pump to the ground loop should not deviate by more than ±12°C from the undisturbed ground temperature under base-load (i.e. weekly average) conditions, or by more than ±18°C under peak conditions. If we assume that the undisturbed ground temperature is 11°C in temperate Europe (including the United Kingdom), this implies typical acceptable base-load temperatures (from the heat pump) of –1°C in heating mode and +23°C in cooling mode, with extremes under peak loading not exceeding –7 and +29°C, respectively. The temperatures of the fluid returning *to* the heat pump will typically be around 3°C less extreme than these last figures.

If we wish to avoid the risk of substantial ground freezing, we should ensure that our base-load average temperatures remain above 0°C.

We should also note the political pressure in some countries for more efficient heat pump systems. As is the case with vertical, borehole-based systems (Chapter 10), if we invest extra money in designing more efficient (and maybe longer) ground loops, we typically reap a return in the form of improved heat exchange, less extreme carrier fluid temperatures and more efficient heat pump performance.

11.8 Earth tubes: air as a carrier fluid

As a parting thought, we should be aware that we have limited ourselves in Chapters 7 and 9–11 to considering only water-based solutions as carrier fluids. There is, of course, good reason for this: water has a huge specific heat capacity. We *can*, however, consider air as a carrier fluid. Indeed, we could conceive of a system of subsurface pipes (or even boreholes) through which air is circulated, gradually equilibrating towards the subsurface temperature. Such systems are probably most useful for summer air conditioning: clearly, a throughflow of air, which has been cooled by flowing through the ground (at, say, 10°C in Britain) could potentially be very useful for cooling a house or small office. Alternatively, in winter, the circulated and warmed air could be used for preheating a more conventional system, or the warmed air could be utilised in an air-sourced heat pump.

Such air-based ground circulation systems have been christened 'earth tubes'. Clearly, we need to power a fan to drive the air around the earth-tube system, which consumes power. VDI (2004) believed that such earth tubes are not energy-effective unless they can achieve a temperature differential of at least ±2°C relative to the outside air temperature. Pipes are typically made of polyethene, polyvinyl chloride (PVC) or concrete and installed in trenches at depths of between 1½and 3 m. Consideration must be given to problems of condensation and how such condensate will be collected.

VDI (2004) estimated that residential houses of floor area 100–250 m^2 typically require air throughflows of 100–500 $m^3 h^{-1}$. Furthermore, flow velocities in pipes of

2–4 $m\,s^{-1}$ are typically sufficient to permit adequate heat exchange with the ground. The pipe diameter selected will depend on the flow rate: for small houses requiring 100 $m^3\,h^{-1}$ of airflow, some 25–28 m length (depending on the thermal properties of the ground) of 100-mm-diameter pipe will be required to provide a minimum standard of thermal performance (VDI, 2004). The corresponding flow velocity here is 3.5 $m\,s^{-1}$ and the pipe heat exchange area is ≈8–9 m^2. For a larger house, with an air throughflow rate of 500 $m^3\,h^{-1}$, some 40–50 m of 300-mm-diameter pipe will be required. This corresponds to 2 $m\,s^{-1}$ flow velocity and 38–47 m^2 of exchange area.

Earth tubes can be utilised to provide air conditioning and preheating to larger building spaces, although some form of computer-aided design is likely to be required.

STUDY QUESTIONS

11.1 You have been asked to design a horizontal ground heat exchanger for a small house in south-central Sweden, to support an 8-kW heat pump. The heat pump will operate for 2400 equivalent peak load hours per year and will have a target SPF_H of 3.5. The soil is a very wet sandy clay. The garden of the house is 70 m long and 70 m wide. If you use a conventional Swedish approach of single straight 40-mm OD polyethene pipes, spaced at 1.2 m, how much pipe do you believe you will need? See the guidance in Section 11.6.3 based on the assumption that the carrier fluid entry temperature does not drop below –5°C (which is, in fact, pretty jolly cold).

11.2 If, instead of 40-mm OD straight pipe, you were to use a narrower-diameter compact slinky, with around 8-m pipe per metre of trench, how many metres of trench and pipe would you estimate that you require? See Section 11.6.5.

11.3 In both cases, calculate the area of ground underlain by heat exchanger. How does this compare with the overall areal limits discussed in Section 11.5?

Note that these questions involve estimates by semi-quantified 'rules of thumb', which are not necessarily a substitute for a full quantitative design process.

12

Pond- and Lake-Based Ground Source Heat Systems

> A thick, acrid, black smoke hung on the ground, envloping the trees. . . . 'I still don't understand', he said, 'how the smoke saves them from the frost'. 'The smoke replaces the clouds, when the sky is clear', replied Tanya. 'And why are clouds so important?' 'You don't get frost when the weather is overcast and cloudy'.
>
> Anton Pavlovich Chekhov, *The Black Monk* ('Чёрний Монах') 1894

In Chapter 11, we have seen how we can extract heat from shallow soils using horizontal 'closed loops' or coils of heat exchanger. We have also seen that the wetter the soil, the better it is as a source for a ground source heating scheme. We can also use heat exchange coils to extract warmth from rivers, ponds and lakes, where the surrounding water provides an excellent heat source (remember the very high specific heat capacity of water – Table 3.1) and a heat transfer medium.

Alternatively, we can base an 'open-loop' (Chapter 8) heat extraction or rejection system on a surface water body, where water is physically extracted from the source, passed through a heat exchanger or heat pump and then returned to the surface water.

Such systems can be highly efficient, large-scale and economically attractive, although their success presupposes a good understanding of the physics and the heat budget of the surface water body.

Of course, we are not limited to using natural lakes and rivers. We can construct our own surface water bodies (which can double up as drainage balancing ponds or

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

ornamental features). We can also use old mineral workings or mines. Wet restoration, coupled with the installation of a heat exchange system, can be an effective means of converting a potential environmental liability (e.g. a spent quarry or sand/gravel pit) into a source of renewable energy (Dodds and Banks, 2010).

12.1 The physics of lakes

Some lakes can be regarded as chemically and thermally homogeneous reservoirs of water and heat, but many are not; nor will they be static. They will usually be in some form of dynamic hydrological and thermal equilibrium with their environment. Our attempts to extract either water or heat (or both) will disturb that equilibrium in a manner that may or may not be acceptable to the ecological systems (or the other human interests) that depend on the lake.

Shallow lakes (less than, say, 4 m; Kavanaugh and Rafferty, 1997), where solar radiation can reach and warm the lake bed and where wind-induced turbulence reaches throughout the depth of the lake, are more likely to be homogeneous in their composition than deeper lakes. Convection and turbulence processes in shallow lakes are likely to ensure that water is mixed throughout the lake depth.

In deeper lakes (say, >6 m), however, significant stratification can develop, as described in detail by Henderson-Sellers (1984). In the following, remember that water is at its most dense at around +4°C. Let us consider a lake in a temperate climate: as the air temperature increases during spring and summer, the upper layer of the lake will warm up and become less dense. Due to the density difference and the fact that wind-induced turbulence does not penetrate to great depths, a stratification will develop, with an increasingly warm *epilimnion* layer floating over a cool, denser *hypolimnion*. The two are separated by a *thermocline* – a steep temperature gradient. The depth at which the thermocline develops in a stratified lake varies from about 3 m (in turbid water, where solar radiation cannot penetrate deeply) to over 15 m in clear water lakes (University of Alabama, 1999). Throughout summer, relatively little heat penetrates deep into the lake: atmospheric and solar heat largely serves to raise the temperature of the epilimnion still further. In autumn, the air temperature cools, the lake's surface layer loses heat and begins to cool down. The top waters become denser than the layers immediately underneath and they start to sink: in other words, the stratification disintegrates in an *autumn overturn*. This process continues down to around +4°C, with the coolest, densest waters accumulating at the base of the lake. As the air and surface temperatures drop below +4°C, the water density starts to decrease again and a new stratification develops, with the coldest waters floating on top of slightly warmer waters (c. +4°C). Eventually, the top layer may freeze. Thus, in climates where temperatures fall near or below freezing point, the water at the base of a deep lake will remain at around +4°C throughout the winter, unless there is a significant groundwater or surface water throughflow to disturb the stratification.

In spring, the coldest top layer heats up towards +4°C, becomes denser and starts to sink – the so-called *spring overturn*. Thereafter, a stratigraphy re-establishes itself, with dense, cold water below the now rapidly warming surface waters. Such processes are characteristic of temperate climates where the winter temperatures fall below +4°C. Because of the two overturns each year, such lakes are termed *dimictic*. We now see why such deeper lakes in temperate and higher latitude climates are so attractive for ground source heating and cooling schemes – they have relatively warm (around +4°C) water below the thermocline in winter and relatively cold water (around +4°C) in summer!

In warmer climates, where winter temperatures stay above +4°C, only an autumn overturn will occur and the lake is said to be *monomictic*. In tropical climates, where there is little seasonal variation in temperature, such overturns seldom occur and the lakes are said to be *oligomictic*.

In lakes with pronounced thermoclines, we may often see a chemocline as well. The dense, cool bottom layer of water may be relatively immobile (at least for several months of the year): it may become depleted in oxygen and other oxidised species, and become rich in reducing gases such as hydrogen sulphide (H_2S), produced by sulphate-reducing microbes in bottom sediments.

Of course, local factors can modify these generalised conceptual models significantly. Large throughflows of surface water can mix the lake contents and prevent water stratification from developing. Influxes of groundwater from the banks or bed of a lake can cool portions of the lake in summer and warm them in winter. Indeed, infrared imaging of lakes can be very useful in identifying groundwater influxes (Banks and Robins, 2002).

Figure 12.1 shows a summer thermochemical profile of a deep Siberian lake (Banks *et al.*, 2001, 2004b) with a thermocline between around 7 and 12 m depth and a redox gradient (a type of chemocline). Above 9 m depth, the water is close to saturation with dissolved oxygen. Below 13 m, dissolved oxygen is effectively absent and concentrations of hydrogen sulphide increase steadily with depth. In other respects, the lake is far from typical: the lake is saline, due to high evaporation rates in the semi-arid southern Siberian climate, coupled with low surface water inflow and the somewhat saline nature of groundwater inflows. The salinity is a little over half that of seawater, as reflected in the recorded dry residue concentrations. Thus, it is not merely temperature that controls the density and the dynamics of the lake; one might expect saline water to form by evaporation in the upper layers of the lake during summer and, possibly more importantly, by the formation of a thick cap of (fresher) ice during winter. It is possible that the saline water thus formed sinks, due to density differential, and accumulates at the base of the lake. The sinking of saline waters, formed during winter freezing, may explain the unusually low temperatures (<2°C) at the base of the lake. An added complicating feature is the possibility that the lake is partly fed by submerged groundwater inflow and, in Siberia, this may be colder than 4°C. These complicating factors are speculative but should serve to remind the reader that lake dynamics are complex.

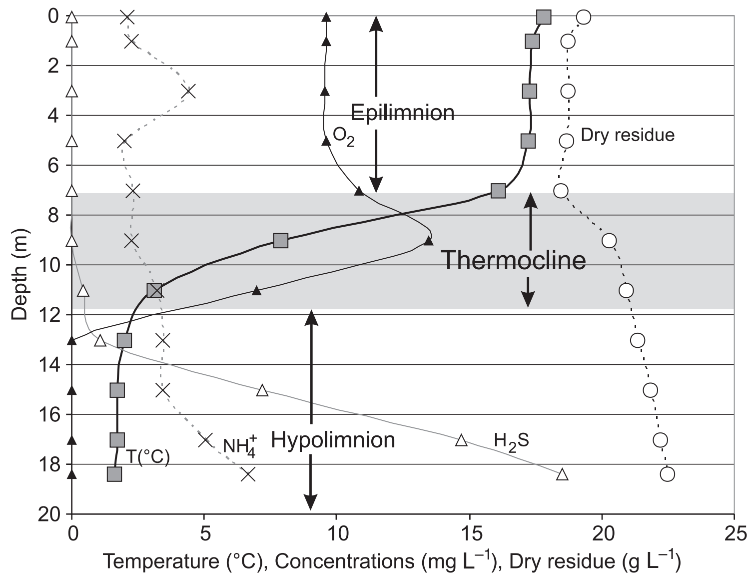

Figure 12.1 A typical summer thermochemical profile for a deep lake in a cold climate exhibits many of the features of this profile from Lake Shira, southern Siberia, measured in September 2000. Note the presence of a thermocline separating deep cold water from shallow warm water, and the redox gradient. Lake Shira is, however, saline (dry residue is a measure of salinity) and has extremely cold basal temperatures: in these respects, it is not typical of lakes in less extreme, non-continental climates. *Based on data from Banks et al. (2001)*.

12.2 Some rules of thumb

Our objective, when supplying heat or cooling from a pond or lake, is to ensure that we do not cause unacceptable change to the lake's temperature, its water level or throughflow (note that, if we increase a lake's temperature, we will also increase losses through evaporation), its chemistry or to its stratification. By 'unacceptable change', we usually mean change that will significantly damage the lake's ecology, amenity value or value as a resource. Clearly, if we are basing a large heating or cooling operation on a pond or lake, we need to thoroughly evaluate both the thermal and water budgets of the lake. However, with smaller schemes, Kavanaugh and Rafferty (1997) suggested that if

- the lake is deeper than 3–4 m,

  AND

- the peak cooling load (resulting in waste heat) is <174 kW ha^{-1} (17.4 W m^{-2}),
- the peak heating load (extraction of heat from lake) is <87 kW ha^{-1} (8.7 W m^{-2}),

OR

- there is a substantial replenishment or throughflow of water (and thus heat) to the lake,

then a lake-sourced heating or cooling scheme is likely to be acceptable. If these criteria are violated, the scheme could easily still be plausible but will require a more detailed risk assessment.

12.3 The heat balance of a lake

If we need to perform a more detailed risk assessment of the impact of a heating/cooling scheme on a pond or lake, we need to construct not only a water balance, but also a heat balance of the lake (Henderson-Sellers, 1984; Hostetler, 1995; Rouse *et al.*, 2005). Figure 12.2 illustrates the main heat transfer mechanisms for a natural lake; these include

- evaporative heat loss (i.e. *latent heat flux*) q_{evap}: this may be modified by plant cover;
- *sensible heat flux*: this comprises conductive/convective heat loss or gain to/from air q_{sens};
- conductive heat loss or gain to/from ground q_g;

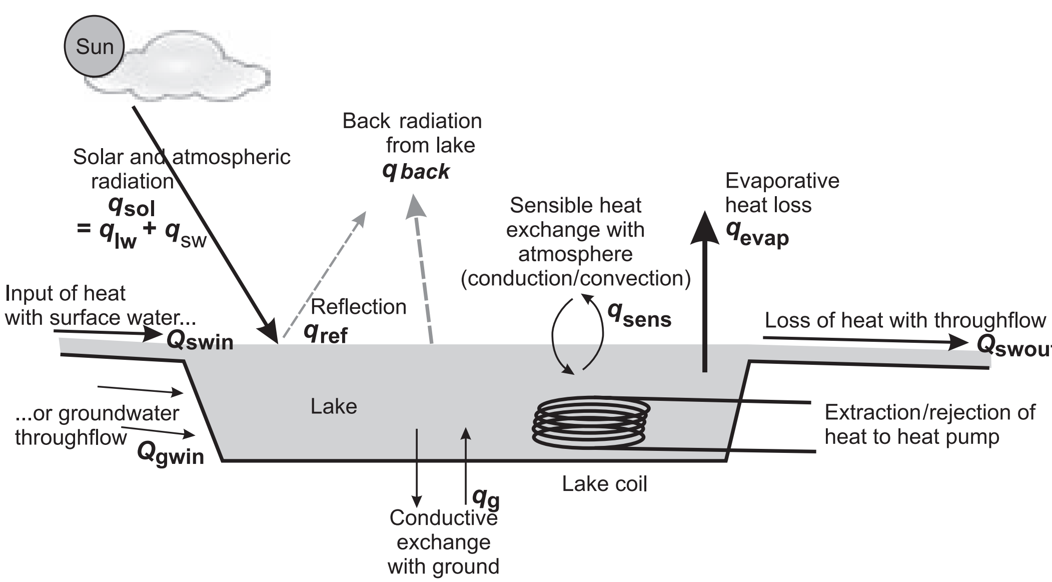

Figure 12.2 The main components of a lake's heat balance. q denotes a heat flux which is proportional to lake area ($W\,m^{-2}$), while Q denotes an absolute heat flow.

- gains from solar and atmospheric radiation q_{sol}. This term can be split into (Section 3.4)
  a. insolation of short-wave solar radiation from the sun (q_{sw}) and
  b. incoming long-wave radiation from the clouds and the warm atmosphere (q_{lw});
- reflective losses of solar and atmospheric radiation q_{ref}. This will be governed by the albedo of the lake with respect to the long- and short-wave components of incoming radiation (α_{lw} and α_{sw}), respectively;
- back radiation (long wave) of heat from lake q_{back};
- advective heat fluxes, comprising
  a. input of heat with surface water inflow Q_{swin};
  b. loss of heat with surface water outflow Q_{swout};
  c. input of heat with groundwater inflow Q_{gwin};
  d. loss of heat with groundwater outflow Q_{gwout}.

Note that the symbol q denotes a heat flux per square metre of lake area, while Q denotes a total heat flow.

These components will vary according to many factors, such as time of day, latitude, season, water temperature, wind speed, cloud cover and humidity. In theory, all these factors are knowable and the relevant equations can be programmed into a spreadsheet to construct a heat balance. Unfortunately, the physics governing a number of these processes cannot be precisely pinned down and a number of different equations can be found in the scientific literature. For example, Linacre (1992), Linacre and Geerts (1997), Kavanaugh and Rafferty (1997) and the University of Alabama (1999) all provide algorithms of relevance to these calculations – the results show a significant variation, however, even for a single set of climatic data. For the interested reader, Marta Faraldo Sánchez (2007) provides an excellent example of thermal balance calculations, for a series of wetlands treating waste water from an abandoned mine.

As an example, let us consider a ground source cooling scheme, where, during summer, we expect to dump Q_{gsc} MJ day^{-1} of heat energy to a lake. The University of Alabama (1999) argued that, for many relatively shallow, non-stratified lakes, the main natural modes of heat loss are (1) evaporative loss and (2) back-radiation losses (which become particularly important in the heat budget during cool nights), while the main mode of heat gain is from solar/atmospheric radiation. In many cases, sensible heat exchange will also be significant.

The dumping of heat raises the lake's temperature: back radiation and evaporation increase. Eventually, these heat losses become so great that (hopefully) a new thermal balance is achieved with additional heat loss via evaporation and back radiation balancing the new heat input from the cooling scheme. Remember that increased evaporation implies not just additional heat loss, but additional water loss, which may be important for the water balance of the lake. In fact, 1 kg of evaporated water loss corresponds to 2.26–2.5 MJ of heat loss (depending on temperature – Box 3.3). In other words, an evaporation rate of 0.01 L s^{-1} corresponds to 22.5–25 kW.

Let us, however, work through the various steps of the heat balance calculation.

12.3.1 Solar and atmospheric energy input

We have already seen, in Section 3.4, that we need to consider two types of solar/atmospheric energy input. Firstly, there is the direct and diffuse (scattered) input of short-wave radiation from the sun (q_{sw}). The total solar irradiance at the edge of the atmosphere on the earth's cross section ($\pi \cdot r^2_{earth}$) is around 1366 W m^{-2}. The average irradiation on the earth's actual surface area, at the top of the atmosphere ($4 \cdot \pi \cdot r^2_{earth}$) is thus 342 W m^{-2}, or averaged over the daylight hours, 683 W m^{-2}. The amount that actually arrives at the earth's surface will depend on latitude, season and cloudiness of the day. We can look this figure up directly in literature, or can estimate it from Prescott's formula (Linacre, 1992):

$$q_{sw} = R_{ex}\left(a + b\frac{n}{D}\right) \tag{12.1}$$

where n is the actual hours of direct sun out of a potential maximum of D daytime hours. R_{ex} is the solar radiation incident at the outer edge of the atmosphere (extraterrestrial irradiance), which will vary according to season and latitude (tables provided by Linacre, 1992, cite monthly means of between 89 W m^{-2} in January to 475 W m^{-2} in June at 50°N). Be careful to select a value of R_{ex} that is appropriate for the time period you are considering in your model. The a value is a constant relating to the 'diffuse' component of short-wave radiation that is scattered during its passage through the atmosphere and typically has a value ranging from 0.18 in England to 0.28 in the tropics. The b constant relates to direct irradiance and has a value ranging from 0.55 in England to 0.47 in the tropics. Penman (1948) suggested values for southern England of $a = 0.19$ and $b = 0.55$.

In addition, we must remember that some of the incoming radiation from the sun (and back radiation from the earth) is absorbed by the atmosphere and clouds. These heat up and re-emit the energy in the form of long-wave radiation. Several equations for estimating this are provided by Henderson-Sellers (1984). Linacre (1992) cited Monteith's (1973) equation as a relatively simple means of estimating downwelling long-wave radiation (q_{lw}):

$$q_{lw} = 208 + 6\theta_{sc} \text{ W m}^{-2} \tag{3.7 and 12.2}$$

or, alternatively

$$q_{lw} = 258 + 3\theta_{sur} \text{ W m}^{-2} \tag{12.3}$$

where θ_{sc} is the mean daily screen temperature in degrees Celsius or θ_{sur} is the mean daily surface temperature in degrees Celsius. Atmospheric long-wave radiation increases with cloud cover, and Linacre (1992) suggested that this calculated flux should be multiplied by a factor ($1 + 0.0034Cl^2$) where Cl is cloud coverage in oktas (eights of the sky).We now begin to understand why cloud cover provides a source of heat at night and we can provide an answer to Chekhov's hero (see introductory quote) as to why we seldom experience frosts on cloudy nights.

12.3.2 *Reflection from the water surface*

Remember that a proportion of the incoming radiation will immediately be reflected away from the water surface. According to Laval (2006), short-wave albedos (α_{sw}) of between 6% (summer) and 10% (winter) are typical for open water, with long-wave albedos (α_{lw}) being even less. However, we should remember that albedo may be modified if the lake is very turbid or if the lake bottom is shallow (i.e. reflection from the lake base as well as the water surface).

12.3.3 *Back radiation of heat from the lake*

The lake itself has a finite temperature and thus radiates electromagnetic energy back into the sky (see Section 3.2.3). The back radiation from the lake is largely a function of lake water temperature. According to Hostetler (1995), the relevant equation is related to the Stefan–Boltzmann law (Equation 3.5):

$$q_{\text{back}} = \varepsilon\sigma\left(\theta^{\text{o}}_{\text{watsur}}\right)^4 \text{ in W m}^{-2} \tag{12.4}$$

where ε = emissivity ($\approx$ 0.97), σ = Stefan–Boltzmann constant = $5.67 \times 10^{-8}\,\text{W m}^{-2}\,\text{K}^{-4}$ and $\theta^{\text{o}}_{\text{watsur}}$ = the temperature of the water surface in K.

12.3.4 *Advective heat gains and losses*

Heat gains and losses with water flow can simply be evaluated from Equation 12.5:

$$Q_{\text{advection}} = ZS_{\text{VCwat}}\left(\Delta\theta\right) \tag{12.5}$$

where Z = the water throughflow rate ($\text{m}^3\ \text{s}^{-1}$), S_{VCwat} is the volumetric heat capacity of water ($\text{J m}^{-3}\,\text{K}^{-1}$) and $\Delta\theta$ is the temperature change in the water. Thus, the advective heat input to the lake can be calculated from

$$Q_{\text{advection}} = Q_{\text{swin}} + Q_{\text{gwin}} - Q_{\text{swout}} - Q_{\text{gwout}} \tag{12.6}$$

where

$$Q_{\text{swin}} = Z_{\text{swin}}S_{\text{VCwat}}\left(\theta_{\text{swin}} - \theta_{\text{lake}}\right) \tag{12.7}$$

and

$$Q_{\text{gwout}} = Z_{\text{gwout}}S_{\text{VCwat}}\left(\theta_{\text{gwout}} - \theta_{\text{lake}}\right) \tag{12.8}$$

and so on, where Z_{swin} and θ_{swin} denote the flow rate and temperature of the incoming surface water.

12.3.5 Evaporative losses (latent heat flux)

The rate of evaporation is typically calculated from water vapour pressure differential (which depends on temperature) and wind speed. This is done by using

1. empirical equations from region-specific studies;
2. simple theoretical relationships, based on Dalton's law of evaporation, which relates evaporation rate E to a constant (c), some function $f(\ )$ of wind speed (v_w) and the difference between the saturated vapour pressure of water (from Figure 12.3) at the water surface temperature (e_0) and the actual vapour pressure of the surrounding air (e_a). e_a is also equal to the vapour pressure of saturated air at the dew point temperature (Box 12.1), which can in turn be found from the wet bulb and dry bulb temperature measurements of the air. Such equations take the form

$$E = c \cdot f(v_w) \cdot (e_0 - e_a) \tag{12.9}$$

3. the more theoretically based approaches such as that of Penman (1948), which is discussed fully by Linacre (1992).

Various versions of Dalton's law of evaporation can be found in the literature and are summarised by Henderson-Sellers (1984) and Faraldo Sánchez (2007). These authors recommended (for UK use) a version derived by the US Geological Survey (Richards and Irbe, 1969):

$$E = 1.12 \times 10^{-11} \cdot v_{w8} \cdot (e_0 - e_a) \text{ in m s}^{-1} \text{ (or m}^3 \text{ s}^{-1} \text{ m}^{-2}) \tag{12.10}$$

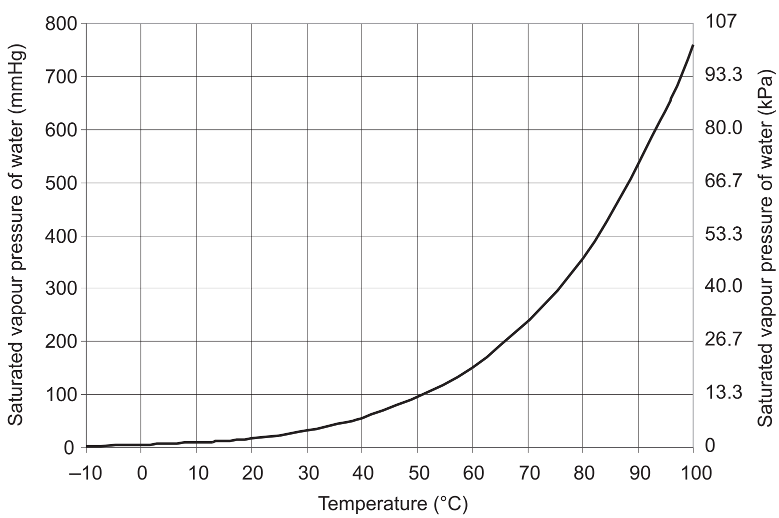

Figure 12.3 The dependence of the saturated vapour pressure of water on temperature. Formulae for calculating saturated vapour pressure are published by Tetens (1930) – see Box 12.1.

BOX 12.1 Dew Point

The dew point is the temperature at which air, when cooled, first becomes saturated with water vapour. At that point, the vapour pressure of water in the air is equal to the saturated vapour pressure of water. The dew point temperature θ_d (°C) is given by manipulation of the Magnus–Tetens formula (Barenbrug, 1974):

$$\theta_d = \frac{b\Omega}{a - \Omega}$$

where $\Omega = (a\theta/(b + \theta)) + \ln(\mathrm{RH})$, $a = 17.27$, $b = 237.7$°C, θ = actual temperature (°C) and RH = relative humidity at temperature θ on a scale of 0–1.

Note also that the saturation vapour pressure over a water surface (e_0 in Section 12.3) can be found by

$$e_0 = 610.8 \times \exp\left[\frac{a\theta}{b + \theta}\right] \text{ in Pa}$$

where θ = temperature at the water surface (°C).

or

$$E = 9.68 \times 10^{-4} \cdot v_{w8} \cdot (e_0 - e_a) \text{ in mm day}^{-1} \left(\text{or L day}^{-1}\text{ m}^{-2}\right) \quad (12.11)$$

In this formula, e_0 and e_a are in pascals, while v_{w8} is the wind speed at an elevation of 8 m in m s^{-1}.

Linacre (1992) himself offered:

$$E = 2 \times 10^{-3} \cdot v_w \cdot (e_0 - e_a) \text{ in mm day}^{-1} \quad (12.12)$$

Kavanaugh and Rafferty (1997) cited a version in somewhat mixed units:

$$E = a(e_0 - e_a) \times (b + cv_w) \text{ in mm day}^{-1} \left(\text{or L day}^{-1}\text{ m}^{-2}\right) \quad (12.13)$$

where e_0 and e_a are in mmHg, v_w is in miles day^{-1}, and a, b and c are constants, which may vary between empirical studies from different regions. Kavanaugh and Rafferty (1997) cited an equation where $a = 0.72$, $b = 0.417$ and $c = 0.004$ (for the cited set of mixed units).

Having found E as mm day^{-1} (= L day^{-1} m^{-2}), we can then calculate the heat loss from the lake through evaporation by

$$q_{evap} = L_V \cdot \rho_w \cdot E \text{ in MJ day}^{-1}\text{ m}^{-2} \quad (12.14)$$

$$q_{evap} = L_V \cdot \rho_w \cdot E/86.4 \text{ in kW m}^{-2} \quad (12.15)$$

where L_V is the latent heat of vaporization of water (see Box 3.3) in $MJ\,kg^{-1}$ and ρ_w is the density of water in $kg\,L^{-1}$.

12.3.6 Sensible heat flux

The magnitude and direction of sensible heat transfer (q_{sens}) between the air and a body of water will depend on the temperature difference between them. The heat flux is typically calculated on the basis of the so-called Bowen ratio:

$$\text{Bowen ratio} = \frac{q_{sens}}{q_{evap}} \approx \frac{\psi_c\,(\theta_{wat} - \theta_{air})}{(e_0 - e_a)} \text{ (see Lewis, 1995)} \tag{12.16}$$

where ψ_c is the so-called psychrometric constant in $Pa\,K^{-1}$, θ_{wat} and θ_{air} are the water and air temperatures in °C or K, and e_0 and e_a are as defined above and in pascals. The psychrometric constant is defined as

$$\psi_c = \frac{S_{Cair} P_{atm}}{0.622 L_V} \tag{12.17}$$

where S_{Cair} = specific heat capacity of air ($J\,kg^{-1}\,K^{-1}$), P_{atm} is the atmospheric pressure in pascals (typically around 100 000 Pa) and L_V = latent heat of vaporization of water ($J\,kg^{-1}$). The value typically varies between 66 $Pa\,K^{-1}$ at sea level and 59 $Pa\,K^{-1}$ at 1000-m altitude. The use of this approach may break down where strong winds are a factor.

12.3.7 Heat exchange with ground

In heat balance calculation for lakes and ponds, it is normally assumed that heat exchange with the ground (other than with groundwater flow) is rather low and can be ignored (compared with the other heat fluxes described above). This is because the thermal conductivity of geological materials is modest and the temperature gradients between the lake and the ground are also typically low.

12.3.8 Finding solutions

Our simplified heat balance (assuming negligible groundwater and surface water throughput) for a lake of area A_{lake} and thermally active volume V_{lake}, where we are planning to dump a quantity of waste heat Q_{gsc}, is thus

$$Q_{gsc} + A_{lake}\,(q_{sol} - q_{ref} - q_{back} - q_{evap} \pm q_{sens}) = \Delta\theta \cdot S_{VCwat} \cdot V_{lake} / \Delta t \tag{12.18}$$

Here we can see that the excess heat inputs cause an average change in temperature $\Delta\theta$ in the lake water over a time interval Δt. If our overall objective is to achieve a long-term equilibrium, we need to identify the final temperature where $\Delta\theta$ becomes zero and where

$$A_{\text{lake}}\,(q_{\text{evap}} + q_{\text{back}} \pm q_{\text{sens}}) = Q_{\text{gsc}} + A_{\text{lake}}\,(q_{\text{sw}}\,(1-\alpha_{\text{sw}}) + q_{\text{lw}}\,(1-\alpha_{\text{lw}})) \tag{12.19}$$

We need to be careful to consider which volume (V_{lake}) or layer of the lake our balance is valid for. In temperate and cold climates, as we have seen, it may only be the surface layer of the lake (above a thermocline) that actively absorbs solar radiation and exchanges heat with the atmosphere. We can thus consider Equation 12.19 and find the water temperature for which an equality (i.e. thermal equilibrium) is obtained. This may take several iterations, as we have already seen that q_{back}, q_{evap} and q_{sens} all depend on water temperature. The final equilibrium temperature represents the predicted maximum average temperature of the lake during the operation of our ground source cooling scheme.

If we are extracting heat from the lake, the same approach can be applied. In this case, however, the temperature of the lake will drop until evaporation, back radiation and sensible heat exchange have fallen to new values where a new thermal equilibrium will be attained.

Note that, in such discussions, we are considering some long-term 'average' thermal equilibrium: the lake will continue to experience diurnal and seasonal fluctuations in temperature due both to natural factors and to fluctuations in the heating and cooling loads coupled to the lake.

Figure 12.4 considers this type of calculation for a typical British lake, of dimension 100 m × 100 m. A mean monthly extraterrestrial solar irradiance of 200 W m^{-2} is assumed, with a moderate cloud cover, a relative humidity of 60%, an albedo of 8%,

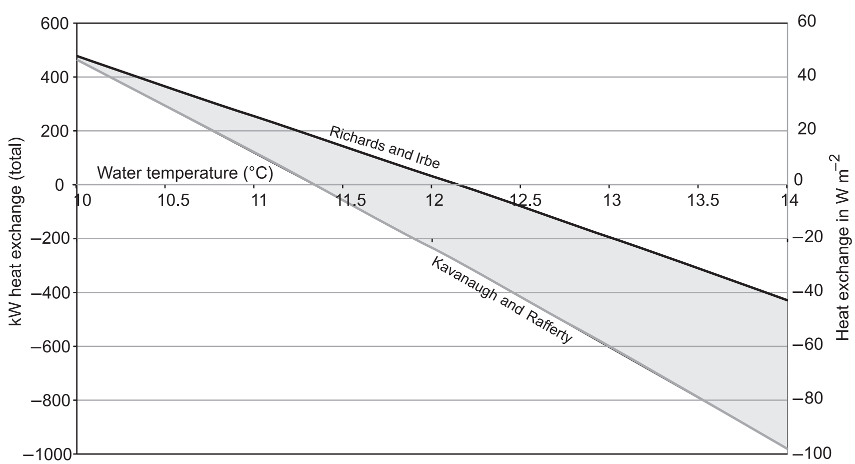

Figure 12.4 An example of a heat balance calculation for a 1-Ha lake (10 000 m^2), with an average extraterrestrial irradiance of 200 W m^{-2}, a short-wave albedo of 8%, a moderately cloudy sky, an air temperature and relative humidity of 14°C and 60% and a wind speed of 4 m s^{-1}. The graph shows the approximate effect of lake water temperature on heat fluxes. The two lines indicate that the evaporative component of the heat budget is calculated by the algorithms cited by Richards and Irbe (1969) and Kavanaugh and Rafferty (1997), respectively.

a wind speed of 4 m s^{-1} and an air temperature of 14°C. It turns out that incoming and back-radiated long-wave radiation are the dominant components in the heat budget, although it is evaporation and sensible heat exchange that are decisive for the heat balance. We find that, at a lake water temperature of 11.3–12.2°C [depending on whether we use Kavanaugh and Rafferty's (1997) or Richards and Irbe's (1969) algorithm for evaporation], an approximate thermal equilibrium exists. If the lake temperature drops from this equilibrium temperature by 1°C, we find that the lake has a net surplus of 22–35 W m^{-2} heat, implying that *if we remove 220- to 350-kW heat via a ground source heat scheme, we would drop the average lake temperature by around 1°C.* Conversely, if the lake temperature increases by 2°C, we find that the lake has a net deficit of 46–73 W m^{-2} heat, implying that *if we dump 460- to 730-kW heat via a ground source heat scheme, we would raise the lake's equilibrium temperature by around 2°C.* The question that we would now have to address would be, are such changes acceptable in terms of the ecology and other users of the lake?

This type of simplified calculation ignores some elements of the heat balance shown in Figure 12.2. In other circumstances, the calculation may need to be modified to take account of these. For example, if the lake freezes, heat conduction through a layer of ice will need to be considered. Furthermore, we should remember that, if the lake is large, it is likely that our heat exchange system will *not* result in a uniform rise in the entire lake temperature; rather, it will result in a substantially greater local change in temperature around the submerged heat exchanger, which will gradually dissipate with distance.

Figure 12.4 is only an example for a specific climatic scenario. For goodness sake, don't use it for design purposes! It is pleasing, however, that the figures derived bear some resemblance to the 'rules of thumb' in Section 12.2. Note also that Box 12.2 describes a 5.4 MW_{th} British scheme, based on a lake of 165 000 m^2.

12.4 Open-loop lake systems

In principle, lake-water-based open-loop systems are similar to groundwater-based open-loop systems (Chapter 8). A prophylactic heat exchanger will often be used to prevent lake water directly entering the heat pump unit, thus restricting any problems of biofouling to the lake water pipe circuit and the heat exchanger. A filter will be placed on the lake inlet pipe to prevent entry of debris to the system and the intake area will need to be periodically checked to ensure that the intake is unobstructed.

The submersible pumps typically used for groundwater-based open-loop systems may also be used in lakes or ponds. However, if the head difference between the lake surface and the surrounding ground surface is less than around 5–6 m, surface-mounted suction pumps may alternatively be used: indeed, they may be preferable due to ease of access and maintenance.

Surface-water-based open-loop systems are suited to cooling schemes and to heating schemes in mild climates. In cold winters, however, water from natural water bodies

BOX 12.2 Case Study: Kings Mill Hospital, Mansfield, UK

The ground source heating and cooling system at Kings Mill Hospital, in the delightful town of Mansfield, UK, is reported to be one of Europe's largest lake-based schemes, with an installed capacity of around 5 MW_{th} heating and 5.4 MW_{th} cooling. Forty-two heat pumps provide the hospital with hot water at 45°C for space heating, and chilled water at 6°C for space cooling via fan coils. The heat pumps transfer heat to and from the nearby Kings Mill reservoir, a lake with a surface area of some 165 000 m^2 and depth of up to 5 m. The lake started life as a mediaeval mill pond, but expanded in the 1830s to support a large number of mills in the valley of the River Maun. Natural winter water temperatures are between 4 and 8°C at the base and 3 and 9°C near the surface of the lake. In summer, temperatures reach 18–21°C at the surface, while remaining cooler at 10–16°C at the base. One hundred and forty flat stainless steel lake plate heat exchangers were fixed onto frames and submerged in the deepest part of the lake, to provide around 1560 m^2 of effective heat exchange area. Carrier fluid is circulated from the heat pumps to the stainless steel lake plates via polyethene header pipes. The design carrier fluid temperatures in the lake plates vary between –2°C in winter and +40°C in summer. No more than light icing of plates was anticipated on the coldest winter days. The scheme is reported to reduce the National Health Service's energy bill by £120 000 per year. Seasonal performance factor (SPF) values of between 4 and 7 have reportedly been achieved. The reported capital cost of the scheme was around £4 million. Information on this installation is sourced from Geothermal International and Skanska (2007), Aston University (2011) and Hermitage Mill Partnership (2011).

may need to be abstracted at temperatures as low as 4°C or below (as was the case with John Sumner's, 1948, pioneering heat pump system, based on the River Wensum in Norwich, UK). As no antifreeze can be added to the water flow, this leaves little margin for heat extraction and such schemes will run the risk of either low efficiency or ice formation in the heat exchanger or discharge pipe. Sumner (1948) noted the importance of ensuring that entry of debris to the heat pump is precluded, as this might constrict flow, increasing the likelihood of ice formation.

The locations of intake and outlet for the open loop will need to be carefully selected to prevent rapid 'short-circuiting' of water from the one to the other. Numerical models are available to simulate the spread of a 'thermal plume' in open surface waters – it may be rapidly attenuated by heat exchange with the atmosphere. It may be possible to use the thermal stratification of deep lakes to optimise the system performance of a cooling scheme, with cold water being abstracted below the thermocline in summer and the waste warm water being rejected above the thermocline. Tempting though such an arrangement may sound, one must be aware of the chemical characteristics of the water below the thermocline. In the case of Figure 12.1, the deep water is oxygen deficient and H_2S rich. Rejection of such water above the thermocline could have drastic consequences (i.e. death!) for the aerobic organisms dwelling in the upper part of the lake.

Of course, we can also use rivers, fjords/sea lochs or the open ocean to support such open-loop schemes. In European cities, disused canals represent large volumes of water that penetrate deep into urban heartlands that may be ripe for redevelopment, with associated heating and cooling demands.

12.5 Closed-loop surface water systems

In Chapters 10 and 11, we have seen that we can use closed-loop heat exchangers, buried in rocks and soils, to extract (or reject) heat. We have seen that the amount of heat we can extract or reject is constrained:

1. in the short term, by the amount of heat exchange surface buried in the ground, and
2. in the longer term, by the amount of heat flux that is induced (or lost) through the ground surface, from the surrounding rock mass or from other heat sources.

Moreover, the heat flow that is induced by the temperature difference between the closed-loop carrier fluid and the natural ground has to overcome two thermal resistances:

3. the thermal resistance of the natural ground, which will depend on the thermal conductivity of the rocks or soil and will vary with time;
4. the thermal resistance of the heat exchanger (closed-loop pipe) itself.

We should be able to envisage that, if we are lucky enough to have access to a pond, or lake (or even a river or stream), we can sink a closed-loop heat exchanger into it and extract heat. Constraints 1, 2 and 4 will still apply, of course: the lake has to have a large enough surface area (or a large enough water throughflow) for long-term heat replenishment, and we must be sure that we install adequate heat exchange area of good efficiency. As regards criterion 3, we might think that the low thermal conductivity of water (Table 3.1) would be highly detrimental to such a system. However, in most lake systems, we fortunately do not have to rely on conduction as the dominant heat transfer process. We can rely on some degree of convection to convey heat to the heat exchanger. This may be forced convection (i.e. natural water currents within the lake) or simply free convection cells set up by temperature gradients around the heat exchanger. In other words, the thermal resistance of a lake-based closed-loop system will largely be governed by the efficiency of the heat exchanger itself and not by the properties of the lake water. Thus, we are largely concerned with

(1) installing enough heat exchange surface area with (2) adequate heat transfer efficiency, in allowing (3) convection or advection to occur freely around the heat exchanger and (4) ensuring that the long-term heat budget of the lake is adequate to meet our needs (Section 12.3). We are less concerned with the details of heat transfer through the lake water, as this will seldom be the limiting factor.

In closed-loop systems, an antifreeze-based carrier fluid is normally circulated through a heat exchange element submerged in the surface water body. This heat exchanger often simply comprises a network of high-density polyethene (HDPE) pipes. The polyethene (PE) should be tolerant of exposure to light (including UV). Closed-loop systems enjoy the advantage, over open-loop systems, of not being as susceptible to biofouling (at least, if the antifreeze is properly checked and maintained – see Section 9.8.2). Moreover, they are also able to extract heat from relatively cold lakes (as antifreeze can be added to the closed-loop carrier fluid). The carrier fluid temperature in heating mode will typically be some 3–5°C lower than the lake temperature and will thus absorb heat by conduction through the walls of the PE pipe. There are several means of installing closed-loop circuits of HDPE pipes in lakes:

- Parallel 'slinkies' (see Chapter 11): extended coils of pipe on the bed of the lake, which is a thermally efficient arrangement but relatively tricky to install.
- Loose coil bundles (i.e. unextended slinkies) of HDPE pipes may be placed on the bottom of the lake. These will usually be anchored to some form of frame or fixed within a wire cage. In the case of artificial lakes (Figures 12.5 and 12.6), these can be installed during construction, before the lake is filled.
- A number of loose coils can be mounted on a 'raft' framework. The raft is floated into position and then sunk to rest on the base of the lake or pond (Figure 12.7).

Figure 12.5 Five pond mats (loose coils of polyethene pipe, anchored to a steel frame) in the base of a newly constructed artificial lake in Yorkshire, UK, designed to support a heat pump installation of c. 22-kW peak output. *Photo reproduced by permission of Geowarmth Heat Pumps Ltd.*

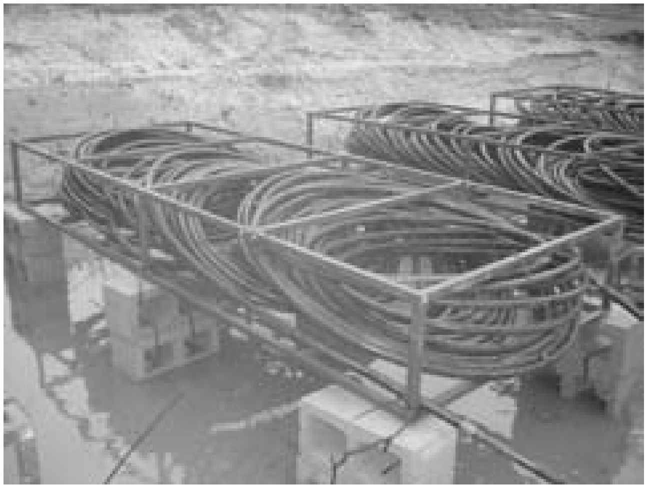

Figure 12.6 Close-up of one of the 'pond mats' in Figure 12.5, each comprising around 200 m of coiled HDPE pipes fastened to a steel frame and anchored with concrete blocks. Note the loose bundling of the pipe to reduce thermal interference. *Photo reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

Figure 12.7 Coils for installation in a natural lake can be mounted in a steel frame, rowed out into the lake, filled and submerged. *Photo reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

The bundles of pipe emplaced on the pond bed, or attached to a raft or other framework, are sometimes referred to as 'pond mats'. The coil bundles should not be 'tight' (Figure 12.6), otherwise thermal 'short-circuiting' between adjacent loops of pipe will reduce the efficiency of heat absorption and throughflow of lake water will be hindered. Rather, each bundle should comprise loose coils of pipe. Rough rules of thumb (Kavanaugh and Rafferty, 1997; University of Alabama, 1999) are as follows:

- Between 20 and 43 m of HDPE pipe length are typically required per peak kilowatt of heating or cooling load, depending on the coil configuration, mode (heating or cooling) and the temperature differential between the carrier fluid and the lake water. Kavanaugh and Rafferty (1997) provided nomograms to calculate this.
- Coil bundles typically contain upwards of 150-m pipe and can be installed as rafts or as discrete bundles on a lake bed, with centres ideally spaced at least 6 m apart (but often less, especially if there is good water circulation within or through the lake).

One UK firm (C. Aitken, GeoWarmth, *pers. comm.*) documented two case studies of lake-sourced systems that have employed (1) 2 × 200 m bundles of PE coil to support 6-kW heating capacity and (2) 7 × 300 m bundles to support 60-kW heating and cooling capacity. These equate to 66- and 35-m pipe length per installed kilowatt, respectively. In the first case, however, the system was consciously over-dimensioned and the second bundle is effectively a 'back-up' – the true figure for specific installed capacity was thus nearer to 33 m per installed kilowatt.

When dimensioning lake coils, we must remember that, if the building to be heated or cooled does not lie immediately on the shore of the lake or the bank of the river, we should take into account heat losses and gains to the carrier fluid during its passage (usually in a trench in the ground) in the header pipes between the building and the lake.

Typically, each bundle of coil is connected in parallel. As with borehole-installed and trenched closed-loop systems, the total carrier fluid flow rate through the closed-loop system ($F_{carrier}$) should ideally be high enough to achieve transient-turbulent flow conditions within each lake heat exchange element (e.g. the bundled coils or slinkies), and thus to ensure efficient heat transfer (Chapter 9).

Although we can circulate the carrier fluid at a temperature below 0°C in an antifreeze-filled closed-loop system, we should be aware that significant build-up of ice on the lake coils could lead to two types of problem. Firstly, a thick layer of ice, while improving heat conduction, will hinder convection of water around the coils and impede convective heat transfer, decreasing the overall efficiency of heat exchange. Secondly, large ice accumulations may lead to the coils becoming buoyant and floating to the surface, if not adequately anchored.

12.5.1 Alternatives to polyethene pipe

HDPE is clearly a cheap and robust means of installing a heat exchanger in a lake. However, it is not especially thermally conductive and for large schemes, we soon start running into kilometres of pipe to satisfy our heating or cooling load. Many clients are thus beginning to consider prefabricated or custom-built metal heat exchange elements, which require a far lower exchange surface area with the water per kilowatt of heat transfer.

One alternative is to install flat heat exchangers ('lake plates'), prefabricated in stainless steel or titanium (depending on the salinity and, thus, corrosivity, of the water). These look something like banks of large metal household radiators mounted on a submerged frame. One popular range, trading under the name 'Slim Jim™', includes lake plates from a 2-ton refrigeration capacity (nominal 7 kW, dimension 61 cm × 183 cm × 1 cm deep) to 8 tons (nominal 28 kW, dimension 122 cm × 366 cm × 1 cm). These exchangers weigh around 3.6 kg per kilowatt capacity and require a carrier fluid flow of around 3.2 L min^{-1} kW^{-1}.

12.6 Closed-loop systems – environmental considerations

With surface-water-based closed-loop systems, particular consideration should be given to the type of any antifreeze employed in the closed loop. Any leakage might come into immediate contact with a sensitive freshwater ecosystem or threaten fisheries or potable water interests. Closed HDPE loops in lakes, seashores and rivers, although fairly robust, will be far more vulnerable to damage (from boats, dredging, etc.) than borehole or trenched systems. Thus, the potential eco-toxicity of any antifreeze is especially important. While ethylene glycol may be an attractive choice in underground loops due to its low viscosity, it is relatively toxic compared with, for example, propylene glycol or ethanol (which have a significantly higher viscosity – see Chapter 9). Thus, the latter are often preferable in lake-based systems, although other alternatives are available on the market today, claiming very low toxicity.

STUDY QUESTIONS

12.1 At a location in temperate Europe, the average monthly extraterrestrial solar irradiance is 170 W m^{-2}. Assuming that there is direct sun for 50% of the total daylight hours, what is the average short-wave insolation on a water surface? If the daily average screen temperature is 15°C, and the sky is 50% covered by cloud, what is the estimated downwelling long-wave radiation?

12.2 A lake has a temperature of 13°C: what would be the estimated long-wave back radiation?

12.3 A swimming pool is 33.3 m long and 9 m wide. On average, it is 2 m deep. If it is to be heated up by a heat pump from a starting temperature of 9°C to a final temperature of 18°C over a 48-h period, what capacity does the heat pump need?

13

Standing Column Wells

> The older view of the nature of heat was that it is a substance, very fine and imponderable indeed, but indestructible, and unchangeable in quantity, which is an essential fundamental property of all matter. . . . We must rather conclude from this that heat itself is a motion, an internal invisible motion of the smallest elementary particles of bodies.
>
> Hermann von Helmholtz

13.1 'Standing column' systems

We have, by now, become familiar with 'open-loop' ground source heat systems, where heat is transported from the subsurface to a heat exchanger by forced convection (or advection) with groundwater flow. We have also dealt with closed-loop systems, where the dominant heat transport mechanism in the rock or sediment is usually conduction.[1] This section deals with systems that are neither open nor closed

[1] Although, in closed-loop schemes, the dominant heat transport process in the rock or sediment surrounding the closed loop will usually be conduction, convective processes in vapour or groundwater may also be important in some circumstances. Heat will pass from the aestifer through any borehole grout and through the walls of a polyethene closed-loop pipe by conduction. However, the process by which heat is transferred from the pipe walls to the internal carrier fluid will involve convection. Furthermore, the carrier fluid conveys heat to the heat pump by forced convection. We can thus see that heat transfer from the aestifer to the heat pump in a closed-loop system is by a complex combination of processes.

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loop but halfway between the two. These are called standing column wells (SCWs). But before we examine these in earnest, let us briefly recap a type of closed-loop configuration we discussed in Chapter 10.

13.1.1 'Open coaxial' closed-loop boreholes

The title of this section suggests that we are reaching the limits of available terminology! Section 10.6.5 briefly dealt with coaxial closed-loop heat exchangers. Figure 10.14 showed us that an 'open coaxial' closed-loop system could be established by circulating water down the bore of a cased borehole (or, in stable low-permeability formations, an unlined borehole) and then pumping it back up a rising main along the centre of the borehole. Or, alternatively, fluid can be pumped down the central pipe and can return up along the borehole wall – the direction of flow makes relatively little difference in terms of overall heat transfer. In a heating system, we would try to design our system so that the re-injected, chilled fluid from the heat pump picks up enough heat from the walls of the borehole, so that it is warm enough to return to the heat pump on re-abstraction. In a cooling system, we would anticipate that the warm fluid from the heat pump/heat exchanger would have lost sufficient heat to the rock in the borehole walls by the time it is re-abstracted.

Although such an arrangement may sound attractive because it can minimise the thermal resistance of the borehole itself, this arrangement is seldom used because of the following:

1. Heat transfer within the aestifer to the borehole wall is dominantly by conduction (especially if the borehole is lined).[1]
2. In permeable formations, unless the entire borehole is lined with plain casing, we are unable to use antifreeze in the carrier fluid (it would be a potential pollutant). This constrains the minimum temperature permissible in the fluid (water) to a few degrees above 0°C.
3. If we propose to use antifreeze solution as the carrier fluid, the entire borehole length should be lined with plain casing, which must prevent any leakage of antifreeze. This results in a rather expensive borehole.

In other words, the gains in heat transfer efficiency over a grouted U-tube arrangement are modest, but the cost of the borehole (due to the need for casing) may be significantly greater. This 'open coaxial' closed-loop configuration would thus be most applicable in stable, self-supporting, low-permeability formations, where open holes can be constructed without large lengths of casing.

13.1.2 Standing column wells (SCWs)

However, if we could induce heat transfer from the geological formation to the borehole by advection in groundwater flow as well as by conduction, the arrangement

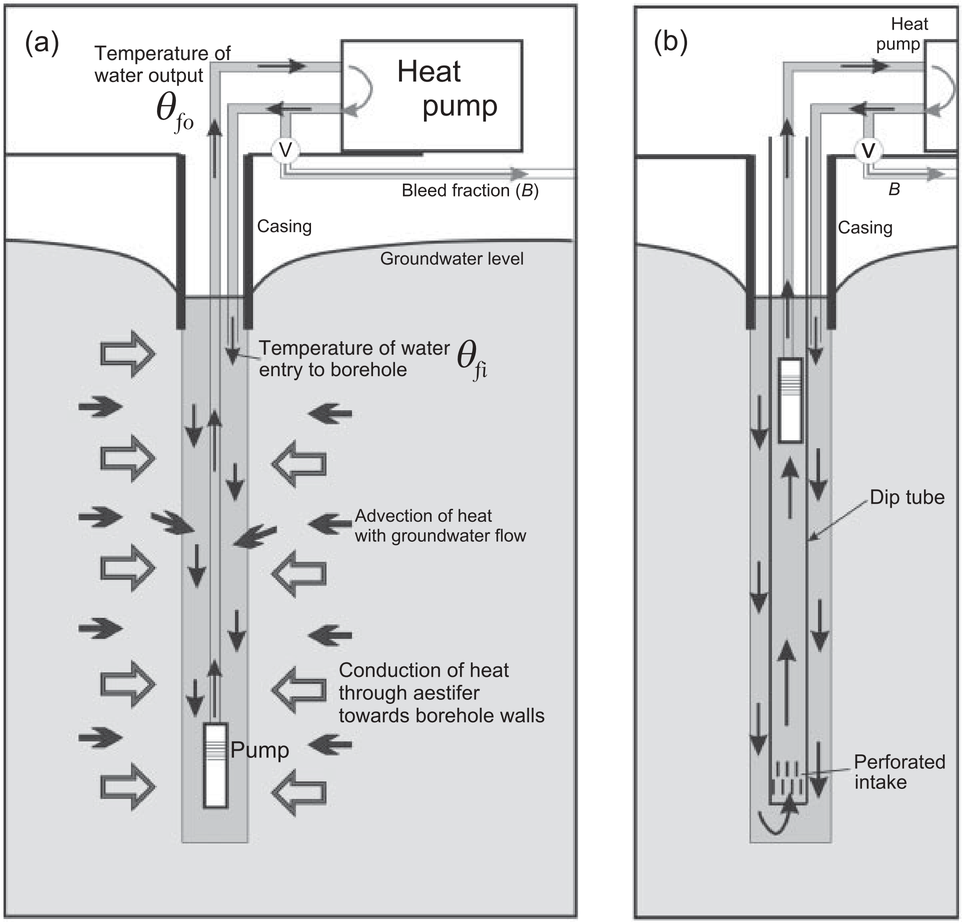

Figure 13.1 (a) Schematic diagram showing a standing column well and (b) a standing column well suited to larger capacity installations where the length of rising main is minimised by using a pump 'sheath' or 'dip tube'. In both cases, the wells are operating in 'bleed' mode. In 'non-bleed' mode, there would be little or no drawdown, but some circulation of groundwater through the formation, due to temperature and pressure gradients along the borehole.

outlined above may start to look more attractive. In this case, groundwater needs to be able to enter the well, so we require either an 'open' (unlined) borehole or a water well screen (see Section 7.3.1). Let us therefore consider the situation in Figure 13.1a, where we recirculate a carrier fluid (groundwater) down the well bore and up a rising main but where we bleed a certain proportion (B) of that flow to waste (following passage through the heat pump). This will have the effect of lowering heads within the borehole and inducing the influx of a corresponding amount of new groundwater to the borehole. This groundwater will also transfer a new 'load' of heat to the system by advection. We might consistently bleed a certain percentage of the total borehole circulation during operation (although this would not be the normal mode of operation). More likely, bleed would be automatically invoked at times of peak heating (or cooling) demand: if the temperature falls below a certain level in the circulating fluid (in heating mode), the bleed valve is opened to induce the flow of fresh (warmer) groundwater into the borehole. In the cooling mode, bleed will be invoked when the temperature rises above a certain level, inducing inflow of cooler groundwater to the bore. This ensures that fluid temperatures entering the heat pump remain within an acceptable range and thus avoids poor values of coefficient of performance (COP).

If the bleed percentage is 0, we essentially have a 'coaxial' closed-loop system of the type described in Section 13.1.1, where heat transfer from the aestifer to the borehole wall is dominated by conduction. If the bleed percentage is 100, none of the pumped water is recirculated to the borehole and we have an open-loop system, where heat transfer from the aestifer is by advection with groundwater. The intermediate arrangement outlined above (where bleed percentage may typically be in the region of 5–30%) is termed a 'hybrid' or SCW system. Around 1000 SCW systems are believed to be operational in the United States (O'Neill *et al.*, 2006). Such wells are typically 6–8 in. (150–200 mm) in diameter and in the United States are usually >100 m deep. The shallower (typically smaller, residential) schemes will have an electrical submersible pump placed at the base of the well (Figure 13.1a), while re-injected water is returned just below the surface of the standing water column (under all operating conditions). In deeper wells, it is common for the submersible pump to be placed within a ≈100-mm-diameter pipe – termed a dip tube. The dip tube extends to the base of the well, where it has a perforated intake section. The pump itself is placed a little way beneath the water level of the standing column of water, allowing lengths of rising main and cable to be minimised (Figure 13.1b). The flow and return mains are sized for flow velocities of up to around $1.5\,m\,s^{-1}$ (Orio *et al.*, 2005). Depending on water quality, the pumped groundwater can flow through the evaporator of the heat pump or through a prophylactic heat exchanger (Section 7.3.2).

As an example of an SCW system, Deng *et al.* (2005) documented a system at a public library in Massachusetts, drilled in Cambro-Ordovician metasediments, where a 10% bleed rate, of 30 min duration, automatically kicks in when the well temperature drops below 4.4°C.

The SCW system has not been widely researched, but the papers by Mikler (1993), Deng (2004), Deng *et al.* (2005) and O'Neill *et al.* (2006) provide a good introduction to the performance of such systems. These authors discovered that the most important design variables for such systems are

- borehole depth;
- rock thermal conductivity;
- rock hydraulic conductivity;
- bleed rate.

The mechanisms of heat transfer within the SCW system include

- heat conduction through the surrounding rock to the borehole wall (and thence by convection into the circulating borehole fluid);
- convective transfer at the borehole walls and surfaces of pipework;
- forced convection/advection with groundwater flow during bleed periods;
- a small component of forced and buoyancy-driven convection within the formation itself, which will be related to the head gradients and temperature gradients along the borehole.

With these many variables and heat transfer mechanisms, we should not be surprised that there are no simple 'rules of the thumb' for designing these systems. Deng *et al.* (2005) proposed a 'simplified model' for simulating the performance of the standing column. Even this simplified model requires considerable mathematical insight and computing skills, but it is currently being incorporated into some design software packages.

13.2 The maths

The simplified model of Deng (2004) and Deng *et al.* (2005) is a one-dimensional (radial) model, with the assumptions of homogeneous and isotropic aquifer conditions and a negligible natural geothermal temperature gradient. It solves heat transport within the aquifer by the one-dimensional advection/conduction model (which is a version of the mathematics encountered in Box 8.3 in radial coordinates):

$$S_{\mathrm{VCaq}} \frac{\partial \theta}{\partial t} + S_{\mathrm{VCwat}} v_{\mathrm{D}} \frac{\partial \theta}{\partial r} = \lambda_{\mathrm{eff}} \left(\frac{\partial^2 \theta}{\partial r^2} + \frac{1}{r} \frac{\partial \theta}{\partial r} \right) \tag{13.1}$$

$$v_{\mathrm{D}} = \frac{FB}{2\pi r D} \tag{13.2}$$

where S_{VCaq} and S_{VCwat} are the volumetric heat capacities of the saturated aquifer and of water, respectively; λ_{eff} is the effective (enhanced) thermal conductivity; B is the bleed rate as a fraction of the total pumping rate F; θ is the temperature; t is the time; and r is the radial distance from the borehole. v_{D} is the Darcy velocity (flow rate per unit aquifer area – see Glossary) and D is the borehole depth within the aquifer or the effective thickness of the portion of the aquifer under consideration.

Here, a so-called enhanced thermal conductivity that includes both genuine thermal conduction and the modifying effects of convective processes related to the pumping borehole is used. The fluid temperature within the borehole is calculated by invoking the heat transport calculated from Equation 13.1 and a separate and explicit term representing the inflow from 'bleed'. It should be obvious that the average temperature of the fluid in the SCW (θ_{f}) is given by

$$\theta_{\mathrm{f}} = \frac{(1-B)\theta_{\mathrm{fi}} + B\theta_{\mathrm{b}} + \theta_{\mathrm{fo}}}{2} \tag{13.3}$$

This allows us to construct a heat balance for the borehole including both the 'bleed' term and 'quasi-conductive' transfer of heat from the borehole wall. Without reconstructing the whole of Deng O'Neill's argument, this takes the form

$$V_{SCW} S_{VCwat} \frac{d\theta_f}{dt} = 2FS_{VCwat}\left[(1-B)\theta_{fi} + B\theta_b - \theta_f\right] + \frac{(\theta_b - \theta_f)}{R_b} D \qquad (13.4)$$

where V_{SCW} is the volume of fluid in the standing column; θ_b is the fluid temperature at the borehole wall (which is assumed to be the temperature of the groundwater bled into the well) derived from Equation 13.1; R_b is a borehole resistance term; and θ_{fi} and θ_{fo} are the temperatures of the recirculating water's entry to and exit from the well.

This simplified model makes a number of very 'dodgy' assumptions: it implicitly assumes that the aquifer approximates to a homogeneous porous medium and also that there is near-instantaneous thermal equilibrium between the rock–sediment matrix and groundwater (see Section 8.11), to the extent that a single temperature can describe both phases. These assumptions will, unfortunately, be least valid in exactly those aquifers where SCWs are most commonly used – fractured, well-lithified aquifers. Nevertheless, the model has been validated against a rather complex two-dimensional, numerical model and found to be broadly satisfactory. However, as one might expect, discrepancies between the simplified model and the complex model do arise in fractured rock aquifer systems, and Deng *et al.* (2005) suggested that these might be of the order of 2°C over 1-year simulation. Discrepancies also arise in shallow boreholes because of end effects, and in situations with high natural geothermal temperature gradients.

13.3 The cost of SCWs

O'Neill *et al.* (2006) compared the requirements for drilled borehole length for three types of ground array, based on the heating and cooling demands of a real building in Boston, Massachusetts. The site was underlain by fractured igneous and metamorphic bedrock of assumed thermal conductivity $3.0\,W\,m^{-1}\,K^{-1}$ and specific heat capacity $2.6\,MJ\,m^{-3}\,K^{-1}$. The undisturbed earth temperature was considered to be 12.2°C. The three ground arrays considered were as follows:

- A 'conventional' grouted U-tube, closed-loop array comprising a line of eight boreholes, drilled to 82 m (total drilled depth 653 m). The borehole thermal resistance was assumed to be $0.14\,K\,m\,W^{-1}$.
- A 152-mm-diameter SCW, with no bleed.
- A 152-mm-diameter SCW with 10% bleed. For the SCW systems, an 'enhanced' thermal conductivity of $3.5\,W\,m^{-1}\,K^{-1}$ was applied.

The study found that, to satisfy the same heating and cooling demand, the SCW with no bleed needed to be 391 m deep, while the SCW with 10% bleed needed to be 263 m deep, compared with a total drilled depth of 653 m for the conventional closed-loop boreholes. O'Neill *et al.* (2006) estimated, furthermore, that the borehole thermal resistance of the SCWs was only 0.0011 $\mathrm{K\,m\,W^{-1}}$. It was found that the operating costs of the three types of scheme over a 20-year life cycle were very similar. The 'cheapest' system was thus largely determined by capital cost. In the selected geological terrain, where the SCW wells could be constructed simply as open holes, the capital costs were strongly related to drilled depth and the SCW well with bleed had the cheapest capital (and thus overall) cost.

It should be noted that, if the geology had required specialist construction of water wells, with a lot of casing, well-screen and/or gravel pack, the costs of the SCW would probably have been dramatically higher than the conventional closed-loop holes. Given that the SCW only gives just over double the heat yield per drilled metre, the analysis would likely have shown that the conventional closed-loop approach would have been cheaper under those circumstances. One could argue (S. Rees, *pers. comm.*), however, that polyethene U-tubes become increasingly difficult or impossible to install in very deep boreholes (Orio *et al.*, 2005, considered depths of up to 450 m). Thus, if available land area is very constrained and if very deep boreholes are required, SCWs may be the favoured option from a constructional viewpoint in certain lithologies.

Moreover, O'Neill *et al.* (2006) assumed a shallow water table in their analysis. They went on to demonstrate that, if the water table was deep (around 30 m in their example), the additional pumping costs of the SCW system with high rates of bleed began to make it uncompetitive with closed-loop or low-bleed options. We can thus summarise that SCW wells (with bleed) are best applied in a rather limited range of geological and logistical circumstances:

- well-lithified or crystalline rock aquifers that are self-supporting and do not require long casing lengths or well screens;
- situations where available land area is tightly constrained;
- aquifers with relatively modest available yields and buildings with modest heat demands (if yields are potentially large, open-loop systems may well be preferable);
- aquifers with relatively shallow water tables (otherwise, pumping costs of lifting the water may render the economics unattractive);
- situations where the well can be used to provide a domestic water supply as well as heating/cooling.

It is thus not coincidental that most of the SCWs found in the United States are in the crystalline rock terrains of the north-east of the country or the Pacific north-west (Orio *et al.*, 2005).

13.4 SCW systems in practice

Orio *et al.* (2005) conducted a study of around 20 commercial and residential SCW schemes in north-eastern United States, ranging in size (installed output) from 17 to 700 kW, typically dominated by heating loads. Depths of boreholes ranged from 73 to 457 m, and some of the larger schemes comprised several SCW wells (up to six wells in one instance). In such multiple-well schemes, automated variable-speed pumps were deemed to be advantageous.

SCWs were typically drilled at a diameter of 6 in. (152 mm), while the static groundwater level was within 37 m of the surface in all cases and typically <20 m from the surface. Orio *et al.* (2005) regarded bleed rates of between 2 and 25% as typical. The number of metres of standing column to support 1-kW load ranged from 2.6 to 7.4 m kW^{-1}, with upper and lower quartiles of 3.4 and 5.2 m kW^{-1} and a median of 4.8 m kW^{-1}. The submersible pumps' power rating was typically in the range of 1–3% (i.e. 10–30 W per kilowatt load) of the total output of the heat pump (although it reached as high as 7% in one instance).

Collins *et al.* (2002) stated that a 1500-ft (457-m) SCW should be adequate to supply a heating or cooling demand of 420 000–480 000 Btu h^{-1} (123–141 kW) – equivalent to 3.2–3.7 m kW^{-1}. They also recommend a spacing of 15–23 m between individual SCW boreholes.

In over 90% of the residential SCW systems surveyed by Orio *et al.* (2005), the SCW systems bled water to domestic water supply systems. This highlights the final major advantage of SCW systems: their ability to produce domestic heat and domestic potable water supply simultaneously.

13.5 A brief case study: Grindon Camping Barn

At Grindon, on a popular walkers' route along Hadrian's Wall in Northumberland, UK, a former telephone repeater station was converted to a hostel to provide accommodation and food for ramblers. The owner of this remote building expressed interest in a borehole to provide a domestic water supply and a base load of ground source heat to the building (to be supplemented by wood fires in the coldest weather). A groundwater-sourced system was thus constructed, supplied by a 104-m-deep borehole (drilled using down-the-hole hammer methods – Figure 13.2) through overlying Carboniferous mudstones and sandstones, the dolerite Whin Sill (along whose outcrop Hadrian's Wall runs) and into the underlying saturated limestones and sandstones. It was initially intended that the system would run as an open loop, with chilled water being discharged to two shallow soakaway boreholes. It soon became clear, however, that the soakaways were unable to accept the entire pumped volume, so the system was reconfigured as an SCW. Groundwater was pumped from the well at a design rate of around 34 L min^{-1}. Part of this flow was used for domestic water supply, while the remainder was circulated directly through the evaporator of a 17-kW heat pump.

Figure 13.2 A down-the-hole (DTH) hammer rig in operation, drilling the standing column well at Grindon Camping Barn. *Photo by Jonathan Steven and reproduced by kind permission of Geowarmth Heat Pumps Ltd.*

Around 80% of the water was recirculated to the abstraction well: it was not returned directly to the well's interior, however, but to the gravel pack installed between the polyvinyl chloride (PVC) well screen and the borehole wall. The waste bleed water (c. 20%) was disposed to the two shallow soakaway boreholes.

Unfortunately, after around 2 years of operation, some problems emerged that appeared to be related to the groundwater's chemistry, which seemed to be somewhat reducing in nature (containing H_2S gas and around $0.5\,mg\,L^{-1}$ dissolved iron). The coaxial heat exchange element of the heat pump evaporator began to develop symptoms of iron incrustation and had to be cleaned of ochreous precipitates by pressure rinsing. Furthermore, the well's gravel pack has also become increasingly clogged, quite possibly with iron oxyhydroxide deposits, and has become progressively less able to accept the recirculated water. The system has thus reverted to open-loop mode, although the site owner has had to establish supplementary disposal routes for 'waste' water.

This experience underlines the extent to which an understanding of groundwater chemistry is important for open-loop and SCW systems. It also emphasises the fact that, while such systems utilising natural groundwaters often require fewer drilled metres, they generally have far greater maintenance requirements than closed-loop systems, whose fluid composition can be controlled.

13.6 A final twist – the Jacob doublet well

A final means of exploiting hydrogeological stratification to increase the sustainability of an abstraction/recharge operation in a single well grows out of the idea of the 'doublet well', proposed by the hydrogeologist C.E. Jacob in the 1960s. In Jacob's original concept, the doublet well consists of an upper portion of the well bore (used for abstraction) and a lower portion (used for circulation and recharge), separated from each other by a system of packers. Jacob proposed the use of this well in situations where a less dense fluid was 'floating' over a denser fluid in an aquifer. For example,

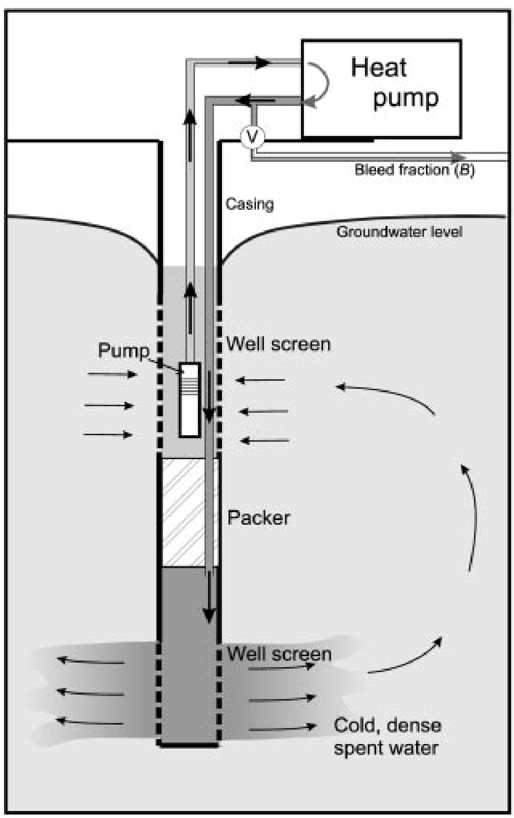

Figure 13.3 A simple vertical doublet well. In fact, C.E. Jacob's original 'doublet well' concept was more complicated than this, utilising three separate chambers for abstraction, recirculation and re-injection – see Wickersham (1977).

in coastal aquifers, fresh groundwater could be abstracted from the upper chamber of the well, while the lower chamber is used for the recirculation and re-injection of the underlying, denser saline groundwater. Jacob also noted that the doublet well could be used to maximise the abstraction of an oil phase, floating on top of denser formation water in an oil reservoir.

Wickersham (1977) noted that the doublet well could also be used in the context of heat pumps. Groundwater could be abstracted from the upper chamber of the borehole and passed through a heat pump to provide space heating (Figure 13.3). All or some of the cold, thermally spent water could then be re-injected into the well's lower chamber. This arrangement is not, strictly speaking, an SCW: we are forcing the water to travel through the surrounding aquifer formation, increasing its contact time with geological materials and the available heat exchange area. The extent to which the spent water migrates laterally into the aquifer would be related to the aquifer's anisotropy: a high ratio of horizontal to vertical hydraulic conductivity would increase the length of the aquifer flow path from the lower chamber back to the upper chamber. In a sense, we could argue that the doublet well is a type of vertically oriented open-loop doublet system, where the abstraction and recharge wells are combined into a single structure, making use of the aquifer's hydraulic stratification.

14

Thinking Big: Large-Scale Heat Storage and Transfer

> The heat pump would appear to offer great possibilities in situations, such as the centre of London, where, for example, the large block of Government buildings now being erected in Whitehall could be warmed by heat extracted from the Thames, with a minimum of smoke and sulphur dioxide production in the centre of the city. Thus one might even recover some of the heat wasted in the cooling water of the Fulham and Battersea power houses.
>
> Mr T.H. Turner, in the discussion following the presentation of John Sumner's (1948) paper to the Institution of Mechanical Engineers in March 1947

Our previous discussions of closed-loop, vertical ground heat exchange systems have tended to focus on quantifying the ground (the *aestifer*) as a source or a sink of heat. We finished Chapter 10 by pointing out that the bigger such a ground source heating or cooling scheme becomes and the more closed-loop boreholes it requires, the more thermal interference there will be between boreholes. Boreholes inside the ground heat exchange array will find it difficult to access heat from 'virgin' unexploited rocks or sediments and will end up competing with adjacent boreholes for access to rock conductivity and heat capacity. In such systems, it is possible that, as heat is extracted, no new equilibrium with the surface or surrounding rocks will be achieved; temperatures of carrier fluid within the array may continue to fall to unacceptable levels.

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14.1 The thermal capacity of a building footprint

Let us imagine that we are constructing a new building on a plot of land of dimension 100 m × 100 m (1 ha = 10 000 m^2). Let us further assume that the volumetric heat capacity of the underlying rock is 2.1 MJ m^{-3} K^{-1} and that we are proposing a borehole heat exchange system of depth 100 m. The total heat capacity of the block of ground that we will be using is

$$10\,000\ \text{m}^2 \times 100\ \text{m} \times 2.1\ \text{MJ m}^{-3}\ \text{K}^{-1} = 2.1\ \text{TJ K}^{-1} = 2.1 \times 10^{12}\ \text{J K}^{-1} = 583\ \text{MWh K}^{-1} \quad (14.1)$$

Let us further assume that the building has a cooling demand of 550 MWh each year. If we assume that the heat pump operates with a seasonal performance factor SPF_C of 5, this means that 660 MWh of heat will be dumped to the ground each year. If there is no heat loss from the block of rock beneath the building footprint to the surrounding rocks, the aestifer will heat up by 660 MWh/583 MWh K^{-1} = 1.1°C per year on average. It will thus not take many years before the aestifer has heated up to such a degree that the cooling operation becomes unsustainable. Of course, this is a highly simplified 'back-of-the-envelope' assessment – we have ignored the fact that the block of rock beneath the building *will* lose heat to the surface and surrounding and underlying rocks. However, if we do this sort of calculation and estimate a predicted temperature change of more than 1°C (upwards for a cooling demand, downwards for a heating demand), it should send us a warning signal that our operation is unlikely to be sustainable.

Let us now assume that, in addition to our cooling load of 550 MWh (with an SPF_C of 5, yielding waste heat of 660 MWh), our building also has a winter heating demand of 400 MWh (with an SPF_H of 4). The net yearly heat flux to the ground will depend on the degree to which cooling and heating are supplied simultaneously (heat recovery) by the heat pump or independently. If, however, we assume the latter, the heat demand represents a heat extraction of 400 MWh × 0.75 = 300 MWh from the ground each year. If we now consider a whole yearly cycle, the net heat flux to the ground is

$$660\ \text{MWh} - 300\ \text{MWh} = 360\ \text{MWh year}^{-1}$$

This represents an average temperature change in our block of rock of 360 MWh/583 MWh K^{-1} = 0.6°C per year. This is still quite a high value, but we should at least now start checking, using a good simulation program, such as *Earth Energy Designer* (Blomberg *et al.*, 2010), how temperatures will evolve in the array, taking account of transfers with the surrounding rocks.

What can we do if we perform such a simple calculation and find that our heat extraction or heat disposal operation is likely to be unsustainable? We can

- drill deeper. Drilling to 200 m, instead of 100 m, effectively doubles the volume and heat capacity of our ground heat exchange array and would be expected to approximately halve the anticipated annual temperature change;
- try to obtain a more balanced (and, therefore, sustainable) heating and cooling profile.

If, for example, we could get rid of 360 MWh of our waste heat, either via conventional cooling (e.g. evaporative coolers) or by selling it to adjacent properties or to a district heating system, we would end up with a perfectly balanced system, with an annual heat extraction rate from the ground of 300 MWh (presumably mostly in winter) and an annual heat rejection load to the ground of (660 MWh – 360 MWh) 300 MWh (presumably mostly in summer). Thus, although the temperature of the carrier fluid in the borehole array, and the temperature of the ground itself, would vary both diurnally and seasonally, there should be no long-term year-on-year temperature change in the ground, and our operation should be perfectly sustainable.

Note that these considerations of sustainability of the thermal capacity of a plot of land typically only pertain to large projects, using large, compact borehole arrays (either vertical closed-loop borehole heat exchangers or multiple open-loop well doublets). With much smaller projects, using only single boreholes or relatively modest arrays, it should be possible to design a solution that yields a long-term sustainable equilibrium with the surrounding rocks and ground surface, even for heavily unbalanced loads (strongly heating or cooling dominated: see Chapter 10, especially Section 10.5.2).

14.2 Simulating closed-loop arrays with balanced loads

We saw, in Chapter 10, that we could use programs, such as *Earth Energy Designer* (Blomberg *et al.*, 2010) to simulate multiple borehole arrays, as well as both heating and cooling loads. Let us take the example from Table 10.2, and now add a cooling load (Table 14.1) of 155 MWh year^{-1} and a peak cooling load of 120 kW.

Let's keep our original SPF_H of 3.5 for heating from Section 10.8.1 (although, we may be tempted to increase it very slightly, as we know that putting a bit of heat back into the ground will tend to lift the ground temperature and thus improve the heat pump efficiency – but let's see what happens). Thus, the annual heating load of 256 MWh implies an annual heat extraction of 182.9 MWh.

Let's use an SPF_C of 4.5 in cooling mode. Thus, the load of 155 MWh implies an annual heat rejection of 189.4 MWh. Our system appears to be almost perfectly balanced, with only a small net rejection of 6 MWh heat per year to the ground.

Note that we have again assumed that our heating and cooling operations are separate (i.e. that the heating and cooling are not performed simultaneously). If the heating and cooling loads were being supplied simultaneously (i.e. with a cooling circuit coupled to the evaporator and a heating circuit coupled to the condenser of the same

Table 14.1 An example of a heating and cooling load profile for a hypothetical building that has been used as the basis for examples in Section 14.2. The 256 MWh annual heating load represents 1600 h year^{-1} equivalent peak load (160 kW) operational hours. The 155 MWh annual cooling load represents 1292 h year^{-1} equivalent peak load (120 kW) operational hours.

	Heating			**Cooling**		
Month	**Monthly load (MWh)**	**Peak load (kW)**	**Duration of peak load (h)**	**Monthly load (MWh)**	**Peak load (kW)**	**Duration of peak load (h)**
J	52	160	13	0	0	0
F	37	160	11	0	0	0
M	26	160	8	0	0	0
A	12	120	6	0	0	0
M	10	80	7	12	60	10
J	0	0	0	40	120	15
J	0	0	0	45	120	16
A	0	0	0	45	120	16
S	13	80	8	13	60	11
O	25	120	9	0	0	0
N	33	160	9	0	0	0
D	48	160	12	0	0	0
Total	**256**			**155**		

heat pump), we could still simulate the ground loop – we would just need to give a little more thought as to how we formulate the load profiles.

Let us again use *Earth Energy Designer* to simulate 30 years of evolution of average carrier fluid temperatures in a 27-borehole closed-loop array, drilled to 100 m on a 3 × 9 grid, and a borehole spacing of 9 m. All other details are the same as in Sections 10.8.1 and 10.8.2. If you remember, with the heating loads only, we found that the 27-borehole array (2700 drilled metres) was barely adequate, even at a spacing of 9 m, with the average carrier fluid temperatures dropping below –5°C (Figure 14.1). With the approximately balanced heating and cooling loads, we find that array is more than adequate – the average wintertime carrier fluid temperature is around +6°C in the coldest month, while even under peak load conditions on the coldest day, the temperature does not drop below +1°C. In summertime, when we are disposing heat *to* the ground, the average carrier fluid temperature in the warmest month reaches 22°C, and does not exceed 28°C even under peak load conditions on the warmest day. Moreover, Figure 14.1 shows that the temperature evolution is almost flat: after the first year of operation, there is almost no long-term year-on-year drift in the temperatures (in fact, there is a barely perceptible upward drift, as we are disposing around 6 MWh net to the ground).

If we reduce the borehole spacing to 5 m, we will recall that this had a large negative effect on the 'heating-only' scheme in Section 10.8.2. With balanced heating and cooling loads, however, reducing the spacing makes very little difference – during the summer it merely results in very slightly higher temperatures. This is because, with balanced heating and cooling loads, we are merely considering the propagation of a heat 'front' around each borehole over an annual cycle, rather than over a 30-year

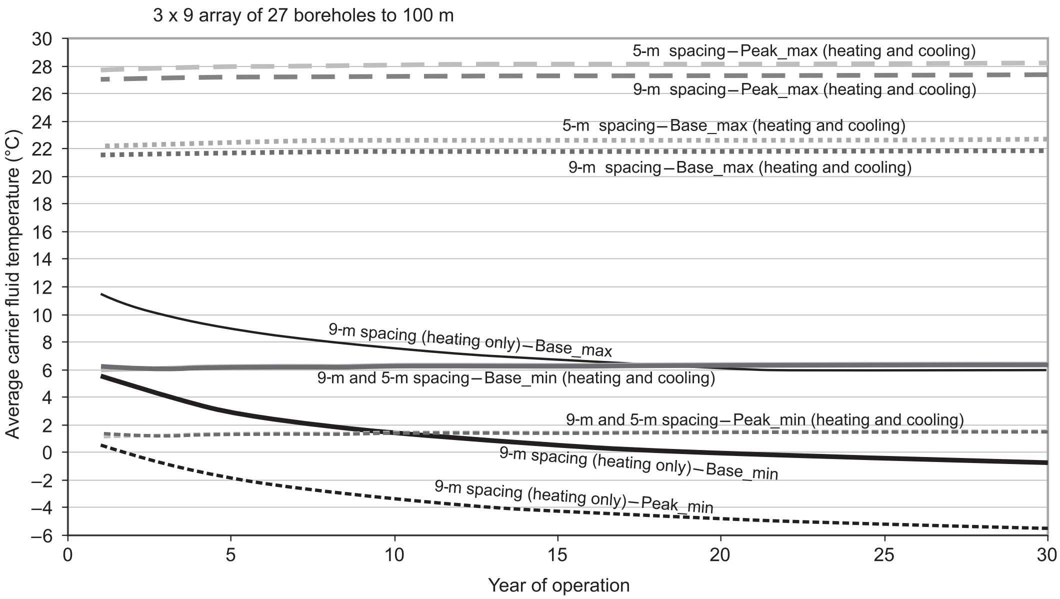

Figure 14.1 Evolution of carrier fluid temperatures in an array of 3 × 9 = 27 closed-loop boreholes, on a rectangular grid, serving the heating and cooling loads documented in Table 14.1. Boreholes are drilled to 100 m, and other details are as described in the text (Section 14.2). Three scenarios are shown: (1) heating loads only, spacing between boreholes = 9 m; (2) heating and cooling loads, spacing 9 m; (3) heating and cooling loads, spacing 5 m. For each scenario, four curves are shown – *Base_min* and *Base_max*, which represent average fluid temperatures throughout the coldest and warmest months, and *Peak_min* and *Peak_max*, which represent worst-case fluid temperatures on the coldest and warmest days of the year.

operational period of constant heat extraction. Over a single year, the 'heat front' from a single borehole barely reaches 5 m from the borehole – to convince yourself of this, return to study question 10.1.

Thus, we have learned an important lesson: *with balanced annual heating and cooling loads, we can use much smaller borehole spacings. We can, in fact, also use much more compact array shapes (i.e. square or cylindrical arrays of the type shown in Figure 14.2, rather than the linear or open shapes shown in Figure 10.20)*.

Figure 14.1 shows us that balanced ground-coupled systems tend to be more efficient as they result in less extreme carrier fluid temperatures and thus in improved heat pump seasonal performance factors.

Alternatively, we could forgo some of these efficiency and marginal cost savings and reap the benefits of reduced capital costs. Figure 14.3 demonstrates that the length of the boreholes in our 3 × 9 = 27 borehole array, spaced at 5 m, could be reduced to around 74 m, without the average carrier fluid temperatures under peak load conditions falling below –3°C (in winter) or rising above +34°C (in summer). Alternatively, a compact square array of 5 × 5 = 25 boreholes, spaced at *only 3 m*, would have much the same effect, if drilled to around 91.4 m.

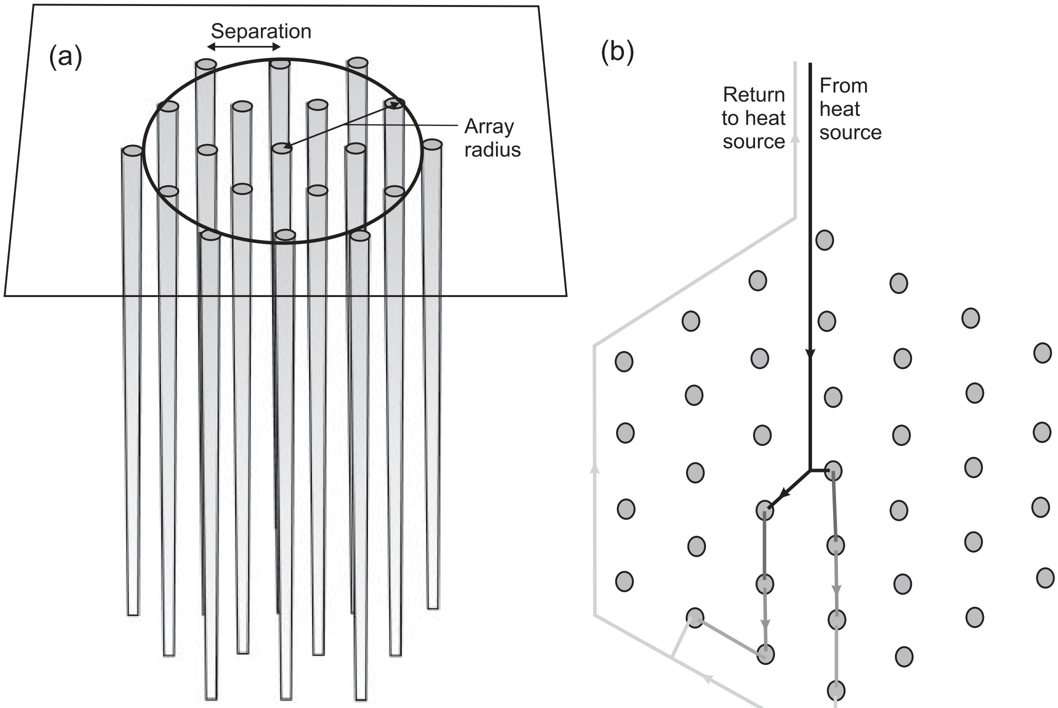

Figure 14.2 (a) A compact hexagonal borehole array designed for UTES: subsurface storage of energy rejected in summer, such that it can be extracted in winter. (b) Plan view showing radial coupling of boreholes in series on several parallel circuits; surplus heat charges the store from the centre outwards. At times of heat demand, heat is discharged by carrier fluid flowing from the outside, inward. Remember that coupling boreholes in series may run the risk of excessive hydraulic resistance (Chapter 9) and of air entrapment between boreholes.

As a reality check, we calculate the volume of this last array as 15 m × 15 m × 91.4 m = 20 565 m^3. The ground volumetric heat capacity is 2.0 MJ m^{-3} K^{-1}, giving a total array thermal capacity of 41.13 GJ K^{-1} or 11 MWh K^{-1}. Thus, we would predict that the extraction or injection of c. 180 MWh of heat would result in an average temperature change throughout the year of up to c. 15 or 16°C. Given that we estimated the average undisturbed ground temperature as c. 12°C (Section 10.8), we can see that this is a slight overestimate – our *Base_min* and *Base_max* temperatures are around +3 and +28°C – due to the fact that there is some heat exchange with the surrounding and underlying rocks.

14.2.1 If the heating and cooling loads are not perfectly balanced

If the heating and cooling loads coupled to our scheme are not perfectly balanced, we will merely see a long-term drift in temperatures upwards (in the case of net cooling

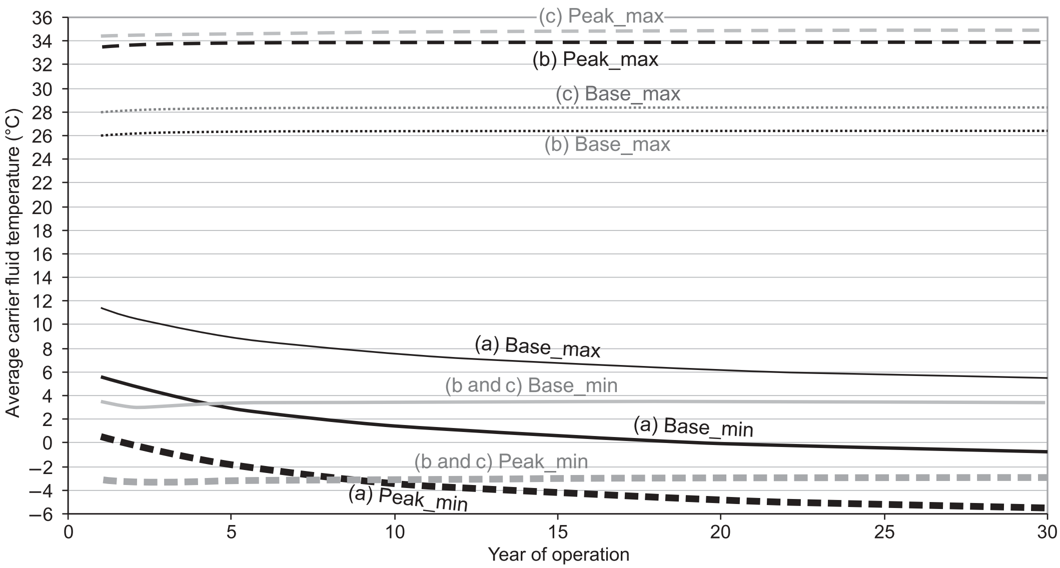

Figure 14.3 Evolution of carrier fluid temperatures in three hypothetical arrays of closed-loop boreholes, on a rectangular grid, serving the heating and cooling loads documented in Table 14.1 – other details are as described in the text (Section 14.2). Three scenarios are shown: (a) heating loads only, 3 × 9 array of 27 boreholes to 100 m, spacing between boreholes = 9 m; (b) heating and cooling loads, 3 × 9 array of 27 boreholes to 73.4 m, spacing = 5 m; (c) heating and cooling loads, 5 × 5 array of 25 boreholes to 91.4 m, spacing = 3 m. See caption to Figure 14.1 for further explanation.

schemes) or downwards (in the case of net heating[1] schemes). At some point in the future, a steady-state situation may or may not be established at an acceptable temperature (depending on the degree of imbalance and the thermal properties of the aestifer). If there are both heating and cooling loads coupled to a ground heat exchange system (open or closed loop), it will always be beneficial, in terms of yielding less extreme temperatures, compared with a heating-only or cooling-only system serving a comparable load.

This is because, in any such system, some of the waste heat from summer remains in the ground and is available for winter extraction. In the case of single boreholes and small arrays, this storage effect will be very modest – between summer and winter, most of the heat will propagate away into the surrounding rocks (this is because a single borehole or small array has a small internal 'volume', but presents a relatively large subsurface area to the surrounding rocks – it has a large subsurface

[1] Remember, when a thermogeologist talks about a 'net heating' or 'net cooling' scheme, he or she refers to the balance between the *ground-coupled loads* and not necessarily to the loads that are delivered to the building.

area-to-volume ratio). In the case of large loads, served by large compact arrays, the borehole array has a large volume but a relatively limited external area exposed to the surrounding rocks. It has a much lower subsurface *area-to-volume ratio* and retains heat much more efficiently.

14.3 A case study of a balanced scheme: car showroom, Bucharest

Any large scheme where there is a component of both heating and cooling coupled to the ground will benefit to some extent from the storage of thermal energy in the rocks or sediments from season to season. This benefit will manifest itself in less extreme summer and winter carrier fluid temperatures (i.e. more efficient heat pump performance) and a reduced tendency to long-term temperature drift.

Polizu and Hanganu-Cucu (2010) described a neat example of an approximately balanced ground-coupled heating and cooling scheme in Bucharest, Romania. Despite lying towards southern Europe, Bucharest can have rather severe winters and the annual average air temperature is around 12°C. A new car showroom was projected to have annual heating and cooling demands of around 572 and 499 MWh year^{-1}, respectively – that is approximately balanced (Figure 14.4). Peak heating loads in winter were estimated as 390 kW and peak summer cooling loads as 421 kW. The company ASA GeoExchange designed a vertical closed-loop borehole array to support these loads. It was designed using modelling software of the type we have encountered

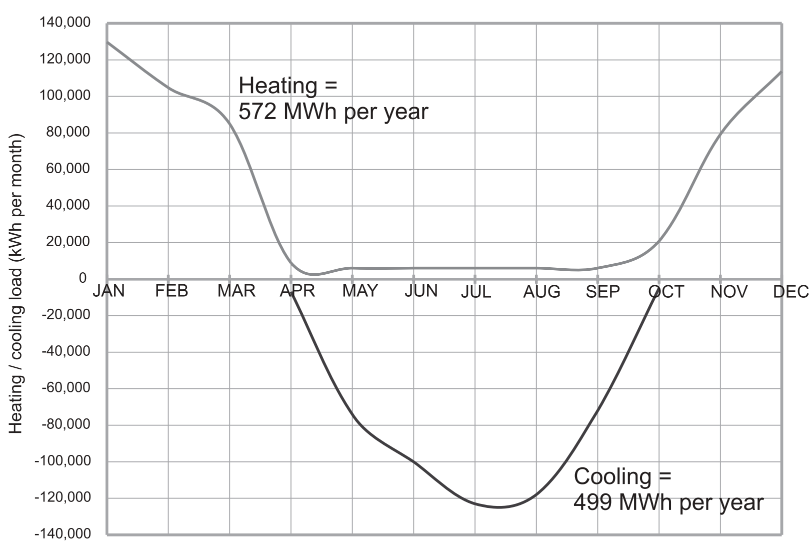

Figure 14.4 The design heating and cooling demands of a car showroom in Bucharest, Romania (in kilowatt-hour per month – see text for discussion) over the course of a year. *Reproduced by kind permission of ASA Geoexchange SRL, Bucharest.*

in Chapter 10. A thermal response test (Chapter 15) was carried out on a pilot borehole, to provide a site-specific estimate of thermal conductivity of around 2.0 W m^{-1} K^{-1}. The average ground temperature was found to be 13°C. It was concluded that a 16 × 7 rectangular array of 112 boreholes to 72 m depth, at a spacing of only 5 m, would be adequate to support the loads. This may sound a lot for a balanced system (52-W installed cooling and 48-W installed heating capacity per metre borehole), but the design criterion was that carrier fluid temperatures should remain above the relatively high figure of around +5°C. Boreholes were drilled at 140-mm diameter, installed with single 33-mm outer diameter (OD) polyethene U tubes and backfilled with thermally enhanced grout. Figure 9.1 shows the plant room and a battery of circulation pumps for the various heat transfer circuits.

The borehole array supports a total of 10 water-to-water heat pumps (which, in turn, provide domestic hot water, warm/chilled water for underfloor heating and fan coil units) and 8 water-to-air heat pumps. Figure 14.5 shows the variation in outside air temperature in 2008–2010 and the corresponding variations in carrier fluid temperature. Note that, when the carrier fluid entry temperature to the heat pump is higher than the exit temperature, the system is dominated by heating; when it is lower, it is dominated by cooling. Note that the carrier fluid entry temperature to the heat pump tends to vary between around +8°C in winter and +20°C in summer. Because carrier fluid temperatures always remain well above 0°C, the system does not employ antifreeze: this has huge benefits in terms of increased hydraulic efficiency (Chapter 9) and in terms of reduced environmental (pollution) risks. The ground temperature in Bucharest is only slightly higher than in London and many temperate European cities. The Romanian experience should make us pause and think: maybe the fact

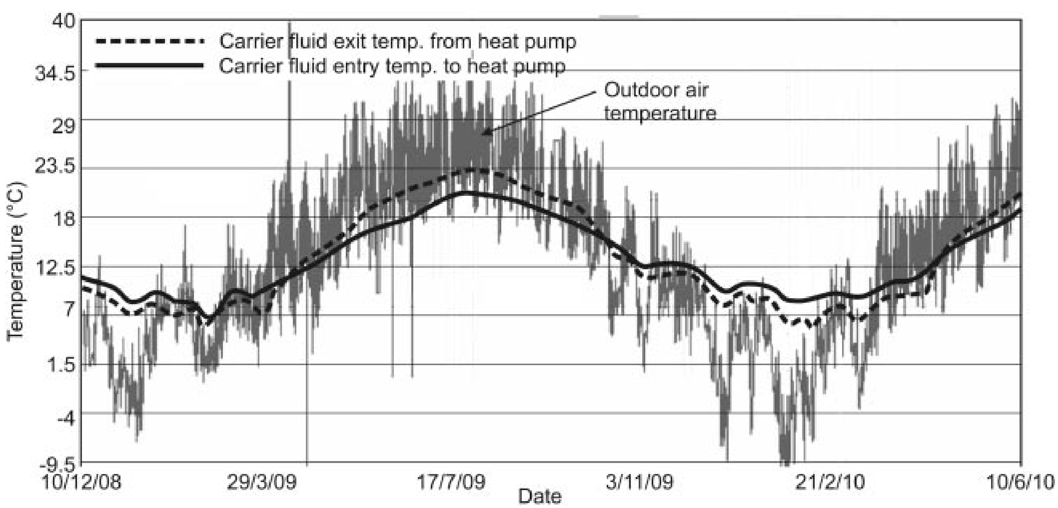

Figure 14.5 The actual variation in carrier fluid temperatures entering and leaving the heat pump array at the car showroom in Bucharest, Romania, compared with external air temperature. *Reproduced by kind permission of ASA Geoexchange SRL, Bucharest.*

that much ground source heating technology has been developed in chilly Sweden has caused us to be over-dependent on antifreeze additives. Maybe temperate Europeans should be following the Romanian example and designing systems with balanced heating and cooling loads, with relatively high ground-loop temperatures and with simply water as a carrier fluid.

14.4 Balancing loads

We have seen that, for large ground heat exchange schemes (both open and closed loop), significant advantages accrue if the ground-coupled heating and cooling loads can be approximately balanced over an annual cycle. The more balanced the loads are, the less extreme the winter and summer temperatures are likely to be, the more efficient the seasonal performance factor and the less any long-term drift in carrier fluid/ground/groundwater temperatures will be. This will typically result in more efficient system performance and a lower risk of unacceptable environmental impacts. For well-balanced systems, we will typically also need a smaller volume of aestifer (i.e. a smaller scheme footprint) to serve as our ground thermal resource.

The example above, from Bucharest, was of a medium sized building in a temperate climate, which naturally had approximately balanced heating and cooling loads. We are seldom so fortunate. In arctic climates, our buildings may be dominated by heating requirements; in the Mediterranean or the tropics, cooling needs may dominate. We have seen in Chapter 6 that, even in temperate or northern climates, cooling needs become dominant as the size of building increases.

So, let us imagine that we are designing a ground-coupled system for a building where the heating or cooling load strongly dominates. If the available footprint for installing the ground heat exchanger is limited, we may find that the only way we can make the system sustainable is to try and bring the ground-coupled loads into balance. There are several ways of doing this.

14.4.1 Supplement the excess loads with conventional solutions

We could, of course, choose the easy solution. If the cooling loads are dominant, we could satisfy a portion of them with conventional air conditioning, maybe coupled to evaporative cooling towers. In this case, we would only couple that fraction of the cooling load to the ground that approximately balances the annual heating load. If the building loads are heavily heating dominated, we could provide part of the heating load with more conventional gas or biomass boilers and we could also examine methods of recovering heat (e.g. heat recovery from exiting ventilation air in winter).

In such a case, we would have to weigh up the additional capital and carbon costs of installing the conventional heating/cooling equipment against the capital and carbon saving of the reduced land footprint required for a thermally balanced ground heat exchanger and its potentially more efficient operation.

14.4.2 Seasonal heat/coolth harvesting

If our ground-coupled loads are characterised by an excess heat requirement, it means that the 'thermal store' around our ground heat exchanger experiences a heat deficit. It is emptied (to heat the building) in the winter but may not be adequately replenished in the summer. We could attempt to 'harvest' excess environmental heat during the summer by circulating the cool carrier fluid from a closed-loop borehole array through:

- simple heat exchangers, such as dry coolers, which will absorb some heat from the air (Figure 6.3);
- arrays of pipes installed beneath asphalt or concrete road decks or car parking lots. The road surface acts as a large solar collector and heats up the carrier fluid;
- solar thermal panels, installed on the roofs of houses (Gabrielsson, 1997; Sanner and Hellström, 1998; Nordell and Hellström, 2000). Box 14.1 describes an example of this at Drake Landing, Canada. In this case, the solar thermal cells replenish the ground array at temperatures of up to 80°C and a large proportion of this heat remains in the ground for re-extraction in winter. In this case, heat is exchanged from the solar thermal cells to the ground-loop carrier fluid via heat exchangers in a central energy station.

In each of the above cases, surplus heat from the environment is harvested and transferred to the ground, replenishing the thermal store.

If our ground-coupled loads are characterised by an excess cooling requirement, it means that the ground experiences a heat surplus. It is filled up with waste heat from the building in summer and not fully discharged in the winter. We could attempt to deliberately discharge this surplus heat to the environment during the winter (or, looked at another way, harvest surplus environmental 'coolth') by circulating the warm carrier fluid from a closed-loop borehole array through

- simple heat exchangers, such as dry coolers, which will reject surplus heat to the cold winter air (Figure 6.3);
- arrays of pipes installed beneath asphalt or concrete road decks or car parking lots and use the surplus heat to melt snow or ice on the road surface. This is done at, for example, Arlanda Airport, Stockholm (Box 3.8);
- natural heat exchangers, such as ornamental fountains (which are quite effective at shedding excess heat; although be aware of potential *Legionella* risks) or heat exchangers installed in cold winter surface water bodies, such as lakes, rivers or ornamental ponds.

14.4.3 Trading heat with the wider community

If a large building has an excessive cooling requirement, even outside of the summer months, one possibility is to supply, or even sell, that waste heat to an adjacent building that has a heat requirement. If a district heating network exists, the surplus heat

BOX 14.1 Case Study: Drake Landing Borehole Thermal Energy Store

Drake Landing is a small community in Alberta, Canada, with an extreme continental climate (which is ideal for seasonal thermal energy storage). Summer temperatures reach the high +20s°C, while in wintertime, temperatures plummet below –30°C. The community comprises around 52 houses, supplied by a district heating system, drawing heat from a closed-loop borehole array. Clearly, such year-on-year intensive heat extraction might soon become unsustainable so, every summer, the borehole array is recharged with heat from an array of solar collectors mounted on the community's roof area. Circulating fluid in these collectors 'harvests' up to 1.5 MW of heat energy and transfers it, via a heat exchanger, to large hot water tanks (short-term energy stores) in a central 'heat station'. Heat can be transferred, via another heat exchanger, to the district heating network supplying the houses, if there is any demand. Any summer excess is circulated, as a hot, water-based carrier fluid, from the short-term store through the closed-loop borehole thermal energy storage (BTES). On hot summer days, the BTES recharge temperature can reach 80°C. The BTES is an array of 144 boreholes to 37 m depth in a 35-m-diameter octagonal shape (in plan), similar to the type shown in Figure 14.3. The boreholes within the BTES array are spaced at an average of 2.25 m; the heat exchange pipe is 25-mm cross-bonded polyethene (PEX), and the 144 boreholes are connected in 24 parallel circuits of six boreholes in series. Each circuit of six boreholes is arranged radially in the array, such that heat is recharged from the solar cells to the centre of the array first. At the end of summer, temperatures in the centre of the array approach 80°C. This heat is efficiently stored in the BTES array until the following autumn/winter, when demand for heat begins to exceed supply. At this juncture, the circulating carrier fluid starts to extract the heat from the BTES to the short-term store and thence, via heat exchangers, to the water-borne district heating system. This runs at around 40–50°C and conveys heat to the individual houses, where it is transferred to each house's airborne heating system (DLSC, 2011). The system is backed up by conventional gas burners. Note the use of short-term thermal storage (hot water cylinders in the central heat station) to provide a diurnal thermal buffer and the BTES array to provide long-term storage – a seasonal thermal buffer. The system operates without the use of any heat pumps (Sibbitt *et al.*, 2007; DLSC, 2011).

could be sold to that network (having been boosted by a heat pump to an appropriate temperature, if necessary). Of course, our biggest heat surpluses from a cooling system are likely to occur in summer – exactly the time when other buildings in the neighbourhood are least likely to require heat.

If a building has an excessive heating requirement, it could purchase 'waste' heat from an adjacent building with a net cooling load, from a combined heat and power

(CHP) plant, from an industrial process or from a district energy system. Again, the biggest need for heat, in the middle of winter, will coincide with the time when surplus heat energy is least likely to be available and when heat costs are at a premium.

Of course, if we could use a ground-coupled heat exchange system (open loop or closed loop) to store our surplus energy from the summer to the winter, either in the form of warm rocks or sediments in a closed-loop borehole array, or in the form of warm groundwater in an aquifer (Box 3.8), we could re-extract it in winter and sell any excess heat to neighbouring buildings or to a district energy system at a premium. Alternatively, if our building has a net heating demand, we effectively accumulate 'coolth' in the ground over the winter months; this surplus 'coolth' could be stored until summer and then sold to neighbouring buildings or to a district cooling system.

We should now be able to understand that ground-coupled heat exchange systems should not merely be viewed as sources or sinks of heat for individual buildings. They can also be used as 'thermal accumulators' or 'thermal batteries', providing seasonal storage in large, possibly complex, district heating and cooling systems.

When small-scale ground source heat systems were discussed (e.g. Chapter 10), the importance of thermal conductivity was emphasized as a determining parameter. In these larger-scale underground thermal energy storage (UTES) systems, volumetric heat capacity is at least as important a factor.

14.5 Deliberate thermal energy storage – closed-loop borehole thermal energy storage (BTES)

In Chapter 10, we have seen that, for schemes where heat is being continuously extracted from (in heating schemes) *or* dumped to (in cooling schemes) an aestifer,

- temperatures in closed-loop borehole arrays evolve over a period of decades until a quasi-steady-state (hopefully) begins to be established, due to heat exchange via the ground surface and the surrounding/underlying rocks;
- arrays with a large surface area will encourage heat exchange via the surface;
- thermal interference between boreholes becomes important and spacing of boreholes is critical (at least 10 m is recommended, if possible);
- a linear or 'open' array, such as that shown in Figure 10.20, performs better than a tightly packed, closed array of boreholes, as it encourages exchange of heat with the broader aestifer external to the array;
- any groundwater flow through the closed-loop borehole array will usually be beneficial, tending to replenish heat to a heat extraction array of boreholes and removing heat from a heat rejection array, by advection.

In this chapter, we have learned that many schemes serving large buildings will have components of heating *and* cooling, both of which can be coupled to the ground.

We thus end up 'dumping' waste heat to the ground in summer and abstracting heat in winter. The loads may not be completely balanced. If we are depositing more heat in summer than we re-abstract in winter, the ground temperatures will steadily creep up with time, and this can be simulated with the analytical tools discussed in Sections 10.7 and 10.8 or with numerical models. If, however, we consider 'balanced' annual heating and cooling loads, we would expect some kind of 'steady state' to be achieved. In these cases, surplus heat is being stored from summer to winter (or surplus 'coolth' from winter to summer).

In some large schemes, our objective may not merely be to achieve 'balanced' heating and cooling loads to enable the design of a neat ground source heating/cooling system that does not experience extreme carrier fluid temperatures (Section 14.3). Our objective may be very deliberate UTES (Sanner and Nordell, 1998). UTES systems using arrays of closed-loop heat exchange boreholes are normally refereed to as borehole thermal energy storage (BTES) systems. VDI (2001b) pointed out that, in thermally balanced BTES systems:

- Temperatures in the system reach a dynamic steady state relatively quickly, although they may vary diurnally and over an annual cycle.
- Thermal interference between boreholes is of lesser importance and closer borehole spacings are possible.
- To avoid leakage of heat outside the borehole array, a more 'closed' array shape will be preferable to an open or linear array. Indeed, to minimise the array's overall surface area-to-volume ratio, an equidimensional cylindrical or hexagonal array may be preferable (Figure 14.2).
- If we wish to minimise heat loss from our subsurface heat store, large throughflows of groundwater will be disadvantageous. Thus, either the hydraulic conductivity should be low to modest *or* there should be a low groundwater hydraulic gradient in the area.
- More extreme peak temperatures may be possible or even desirable.

Figure 14.2 shows a compact hexagonal array suited to BTES. The total thermal storage (H) of an approximately cylindrical array of boreholes of depth D and effective array radius r_{array} is approximated by

$$H = S_{\mathrm{VC}} \cdot \pi \cdot r_{\mathrm{array}}^2 \cdot D \cdot \Delta\theta \tag{14.2}$$

where S_{VC} = volumetric heat capacity of the aestifer material and $\Delta\theta$ is the average change in the temperature of the ground enclosed by the borehole array. This thermal storage is the amount of heat rejected to the ground in summer and abstracted in winter. Such hexagonal or cylindrical arrays are usually charged with heat from the inside outwards, and discharged from the outside inwards (see VDI, 2001b, and Section 14.5.2). Although such a recharge-discharge arrangement will assist in minimising temperature changes in the carrier fluid supplied by the BTES array during use, the carrier fluid temperature will inevitably vary as the array is charged and discharged. It is thus important to identify the maximum and minimum acceptable temperatures to and from the array, as these will effectively define the $\Delta\theta$ and the storage capacity that the array represents.

Clearly, the amount of heat that can be stored is maximised if $\Delta\theta$ is maximised. In northern countries, where storage of surplus heat in the summer is encouraged for subsequent abstraction in the winter, carrier fluid temperatures in excess of 40°C are commonly considered viable. Usage of heat pumps to reject heat at such temperatures may not be especially energetically efficient. If, however, the surplus heat is already available at a high temperature (e.g. waste heat from industrial processes, or heat harvested from efficient solar thermal cells) and can be stored underground, it will permit highly efficient extraction of heat in winter.

VDI (2001b) and Gabrielsson *et al.* (2000) even discussed temperatures as high as 90°C for BTES schemes. Temperatures as high as this will require careful consideration of the materials employed in borehole construction: high-density polyethene becomes unviable as a ground-loop material above 60–70°C (VDI, 2001b) – and any manufacturer's guarantees may become void at lower temperatures. Cross-bonded polyethene (PEX) may remain viable at slightly higher temperatures of up to 80°C or, alternatively, other materials such as steel, polybutene or polypropene may need to be considered. Simulation tools and models should also be carefully assessed for applicability. Bear in mind the temperature dependence of thermal properties of materials and also that vapour movement and fluid convection may become significant at highly elevated temperatures, in addition to conduction.

BTES systems can be characterised by a Utilization Ratio, calculated as the amount of energy recovered after one or more storage cycles, divided by the amount of heat energy supplied to the store over the same cycles. A good BTES scheme should yield a Utilization Ratio of at least 50%, while larger schemes may achieve values in excess of 80% (VDI, 2001b).

14.5.1 Lulevärme BTES, Sweden

One of the earliest BTES schemes, in Luleå, Sweden, used an array of 120 × 65 m deep boreholes in granitic gneiss bedrock to store waste industrial heat. The boreholes were spaced at c. 4 m and encompassed a rock volume of some 120 000 m^3 in a compact 10 × 12 rectangular array. The system operated from 1983 to 1989. Temperatures of 60–70°C (Sanner and Knoblich, 1998) were typical at the centre of the store following charging, falling to 40–50°C during the discharge phase. The waste heat was derived from a nearby steel factory and, over a typical annual cycle, some 2000 MWh of heat was recharged to the BTES array at temperatures of up to 82°C (Hellström, 2005). Each season, around 1000–1200 MWh of heat was recovered to heat a university building, implying a Utilization Ratio of 50–60%. The heating was performed both with and without heat pumps, depending on the temperature of the store during the discharge cycle.

The borehole construction was of particular interest, being similar to the 'open coaxial' type shown in Figure 10.14b, without casing. In other words, the boreholes were drilled 'open hole' at 150-mm diameter into the gneiss. Only the upper few metres (through the overlying Quaternary superficial deposits and unstable bedrock) were cased with 200 mm steel casing. Circulating water (the carrier fluid) was led down to the base of the hole in an internal pipe; it then flowed back up the borehole, coming into direct contact with the borehole walls, discharging via another pipe near the borehole top (Hellström, 2011b).

14.5.2 Other BTES examples

Box 14.1 provides an example of a scheme at Drake Landing in Canada, where a cylindrical BTES array is used to store heat, harvested from solar thermal panels in summer at 80°C. The heat from the solar panels is transferred, by a central exchanger, to the BTES carrier fluid at times when the solar thermal circuit is warmer than the BTES circuit and when the solar heat is not required by the residents' houses themselves. As in many such BTES systems, the inner core of the array is 'charged' with heat first. In this case, this is achieved by connecting several boreholes in series (in each parallel circuit) along a radial direction. In summer, hot fluid passes through the central boreholes and then into the outer bores, thus heating up the core to a high temperature first, and then the margins. In winter, the flow direction is reversed, cool carrier fluid flows from the outside of the array inwards, removing heat from the outside of the array first, and conveying it to the houses. The core of the array is 'saved' and discharged last of all. Note that the Drake Landing scheme does not need to use heat pumps; the stored heat is at a temperature high enough for direct usage.

Another example of a major BTES scheme, based on closed-loop boreholes, is that at Richard Stockton College in New Jersey, where a volume of 1.2 million cubic metres of sediments is accessed by an array of 400 boreholes of 135 m depth. The cooling load (over 5000 kW) is reportedly larger than the heating load, so that the ground will likely have a tendency to warm up over time, although this may be mitigated by heat loss due to groundwater advection (Stiles, 1998).

14.6 Aquifer thermal energy storage (ATES)

We have already seen, in Sections 3.10, 3.11 and 8.14, that the consideration of balanced loads, thermal storage and reversibility also applies to open-loop, groundwater-based systems.

Let us simply consider a non-reversible well doublet, extracting water from an up-gradient abstraction well and recharging it to the aquifer via a down-gradient re-injection well. If the doublet is performing continuous cooling (e.g. the Selby scheme, described in Section 8.16.1) or continuous heating, there are a number of thermal risks associated with its operation:

- that the re-injected warm or cold water will eventually feed back into the abstraction well, if it is not sufficiently distant;
- that the 'thermal plume' of warm or cold groundwater migrating down-gradient from the re-injection well may impinge on other aquifer users or environmental features, causing thermal pollution;
- that the aquifer may tend to heat up over time, if too many large ground source cooling schemes are installed in it;
- that long-term ground temperature changes may cause ground movement, or geochemical/microbiological complications.

It should be relatively easy to see that, if we can do both heating *and* cooling with the well doublet, we will tend to minimise these potentially negative impacts and long-term temperature trends. If the loads are approximately balanced over an annual cycle, there should be *no* long-term changes in temperature and no significant thermal plume. We will also greatly reduce the risk of thermal feedback and should be able to place our well doublet closer together than would otherwise have been the case.

In Box 3.8 and Section 8.14, we have also seen that a well doublet can be reversible and that it can be deliberately used for ATES from season to season (Andersson, 1998; Bakema and Snijders, 1998; Vos, 2007). Figure 14.6 shows how this might work: in the summer, the loads will be dominated by cooling and there will be a net discharge of heat to the groundwater passing through our building's system of heat exchangers. The warmed groundwater will be re-injected into the right-hand well (the 'warm' well), creating a 'bubble' of warm water in the aquifer around that well. In the winter, heating loads might dominate; the system polarity would reverse, such that the 'warm' well now becomes the pumping well. The heat exchangers (which will typically be coupled to a heat pump) extract heat rather efficiently from the warm groundwater flow and the resulting cold water is re-injected into the left-hand 'cold' well, creating a 'bubble' of cold water. The following summer, the polarity is reversed again and the stored groundwater from the 'cold' well is used to perform passive and, if necessary, active (with a heat pump compressor) cooling.

Box 14.2 describes a large-scale application of this concept, at Gardermoen International Airport, Norway. Note that the use of 'colder-than-normal' groundwater for cooling and 'hotter-than-normal' groundwater for heating enables us to obtain larger thermal effects per $L\,s^{-1}$ of groundwater yield than would be the case if we were simply using natural groundwater temperatures (Section 7.3). Furthermore, we do not have to think about long-term, year-on-year migration of heat when designing our 'thermally balanced' doublet separation, we merely need to ensure that the warm and cold 'bubbles' of water remain separate and to consider thermal migration over an annual cycle. Kazmann and Whitehead (1980) have developed tables of minimum well-doublet separations, based on exactly this assumption. We can thus locate our doublet at a smaller spacing than would have been the case in a heating- or cooling-dominated non-reversible system. Both of these factors serve to reduce capital and land costs.

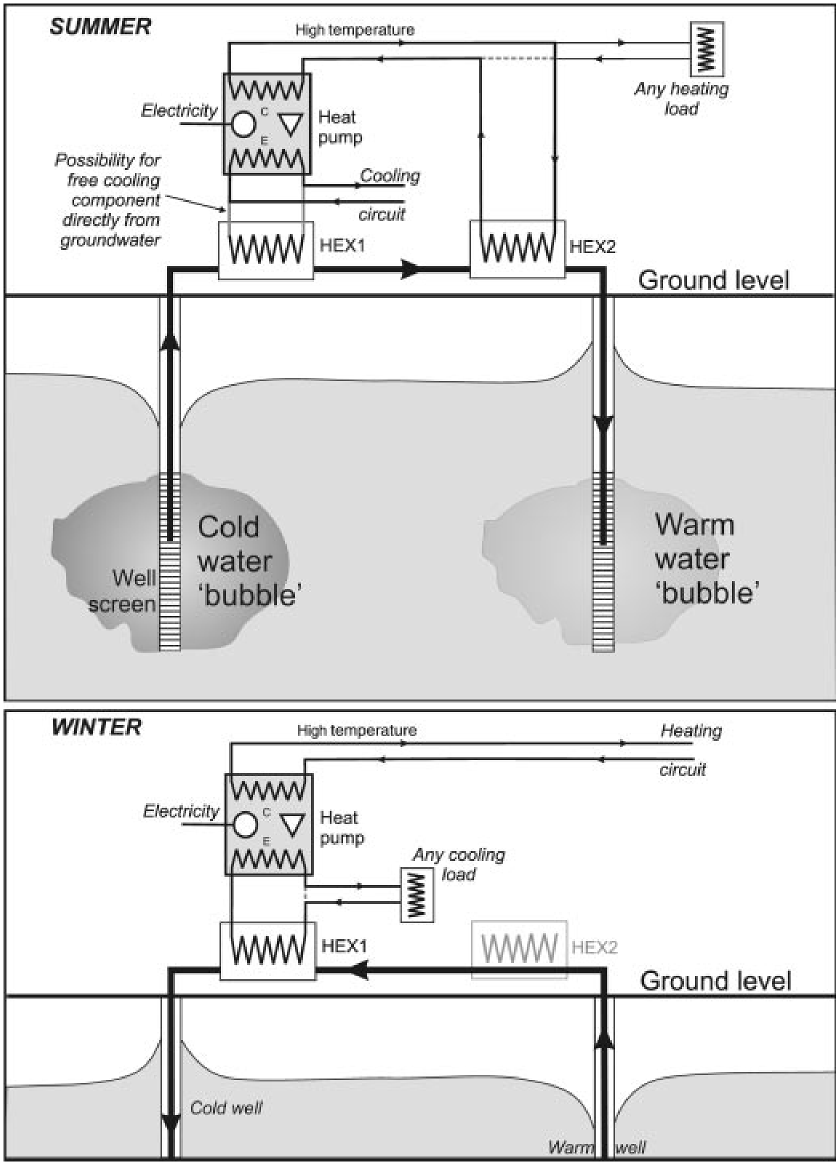

Figure 14.6 Cross sections showing how a hypothetical well-doublet aquifer thermal energy storage scheme might work during summer (top) and winter (bottom). This particular scheme incorporates two heat exchangers (HEX) and a heat pump capable of performing heating and active cooling. HEX1 permits some component of free, passive cooling in summer, and extraction of heat from groundwater by the heat pump evaporator in winter. HEX2 allows rejection of heat from active cooling. Other variations are possible – for example, schemes without heat pumps, performing passive heating and cooling.

BOX 14.2 Case Study: Gardermoen International Airport

Norway's main international airport, at Gardermoen, is also located on the nation's largest aquifer: the Øvre Romerike glaciofluvial sand and gravel aquifer (Odling *et al.*, 1994), containing huge reserves of groundwater at around 5–6°C (Sæther *et al.*, 1992). Before Oslo's airport was shifted from Fornebu to Gardermoen in 1998, it was decided that the massive new airport deserved a ground source heating and cooling scheme, based on pumped groundwater. The scheme is seasonally reversible and stores energy from season to season. During most of the year, water is pumped from nine 45- to 50-m-deep 'warm' abstraction wells (spaced at 50 m and with a combined yield of up to 75 L s^{-1}) and heat is extracted from the groundwater flow (via a heat exchanger) by a heat pump array. The resulting chilled water is used to satisfy any cooling needs in other parts of the airport complex and is then returned to nine similarly spaced 'cold' re-injection wells, some 150 m away from the abstraction wells. In summer, the polarity of the wells is reversed and water is abstracted from the 'cold' wells to perform passive and, if necessary, active cooling, before the water (with its cargo of waste heat) is re-injected to the warm wells. The maximum installed capacity of the system is around 8-MW heating and 6-MW cooling (SINTEF, 2007). The payback period for the additional capital cost of the ground source heating and cooling installation, compared with conventional technology, was reckoned to be around 2 years (Eggen and Vangsnes, 2005). The ground-coupled system seasonal performance factor is estimated at around 5.5. The project is thus regarded as a success and expansion of the system is currently planned – some problems have been noted with well-clogging, however.

Of course, other factors may assist in keeping our warm and cold aquifer 'bubbles' separate:

- geological or topographical factors, such as the bedrock ridge creating two aquifer 'basins' in the Arlanda case study (Box 3.8);
- having our cold and warm wells tapping different vertical portions of a thick aquifer sequence (vertical separation as well as horizontal separation);
- locating our cold and warm wells in separate aquifer strata, if a multi-level aquifer system is available.

For the ATES concept to work well, one would prefer an aquifer system which is sandwiched between two strata of low thermal and low hydraulic conductivity, to prevent vertical dissipation of heat. One would also prefer a situation with little natural groundwater flow, so that advection does not remove heat from storage. Low groundwater flow can be a result of low aquifer transmissivity (although this is detrimental to well yields, see Section 8.6) or low hydraulic gradients in the aquifer (typically associated with flat topography).

For reversible well doublets to function efficiently, each well head needs to be fitted with equipment to allow both pumping and recharge, while responsibly managing pressures in the recharge main to prevent degassing of dissolved gases (see Sections 8.9.3 and 8.9.4). One can install separate pumping mains (fitted with a submersible pump) and recharge mains (which may or may not be fitted with pressure management valves, depending on pressure differentials). Figure 14.7 shows a well head in the Arlanda Airport scheme (Box 3.8), fitted with a large-diameter central pumping main and two smaller diameter, open recharge mains (one or both of which can operate, depending on flow rates, in order to manage recharge pressures). Alternatively, specialist devices exist that allow the pump and recharge valve to be fitted on a single rising main pipe (e.g. Cla-val, 2007).

Furthermore, wells and well screens (if used) need to be constructed with their reversible polarity in mind, ensuring that water will flow efficiently out from the well to the aquifer and *vice versa*. The risk of clogging of the well by particulates or chemical precipitates must be considered and, if necessary, mitigated.

In 2002, Andersson *et al*. (2003) reported some 40 large ATES systems in Sweden, especially in the Skåne region, partially serving commercial buildings and partially

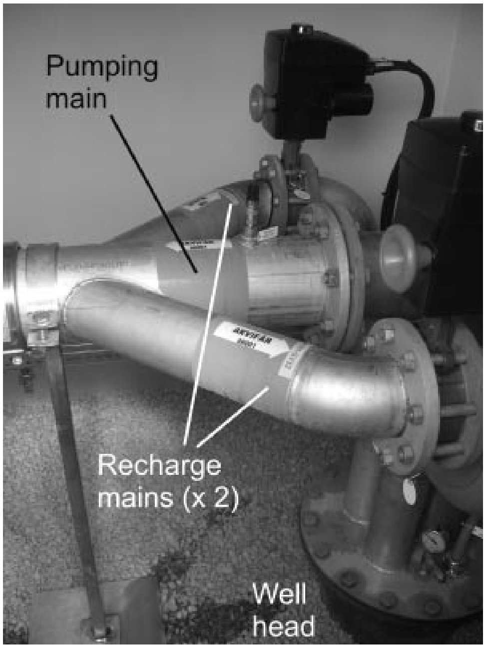

Figure 14.7 A bidirectional well at the Arlanda Airport ATES scheme (Box 3.8), allowing both abstraction of groundwater (thicker central main) and re-injection (flanking, narrower, pipes). *Photo by D. Banks.*

serving district heating and cooling schemes. ATES systems have proved especially popular in the Netherlands (lots of groundwater, multi-level aquifers, low hydraulic gradients): Godschalk and Bakema (2009) reported that some 400 ATES systems had been installed by 2004, rising to around 1000 by 2009. Vos (2007) noted that some Dutch provinces require an energy balance, over a period of 5 years, to be demonstrated in order for such ATES schemes to be permitted. She advocates continuous monitoring of groundwater flows and temperatures into and out of the aquifer in order to verify this thermal balance. Furthermore, in the Netherlands, ATES schemes are typically not permitted to operate at temperatures <5°C or >25°C or 30°C and they should not be operated in the vicinity of drinking water wells.

In reality, of course, it is seldom that a building will have exactly balanced annual heating and cooling loads: in order to balance the heat rejected to or abstracted from the ground, we may need to consider some of the strategies discussed in Section 14.4.

14.7 UTES and heat pumps

The astute reader will have noticed that we have now encountered several examples of UTES schemes where heating and cooling are performed *without any need for heat pumps*:

- Arlanda Airport (Box 3.8), where snow melting and preheating of ventilation air are provided, as well as air conditioning;
- Lulevärme in Sweden, where the temperature of the heat from the BTES was high enough to heat a university building for significant parts of the operational cycle;
- Drake Landing, Canada, where stored solar energy in a BTES is hot enough to provide for the heating needs of the residential community.

Without UTES, we are restricted to the natural temperature of the ground/groundwater: in temperate Europe, this is around 10–13°C, which is too cold to perform direct space heating. With UTES, we can store heat from summer cooling, from solar cells, from industrial processes or from CHP plants at a temperature that is high enough to be used directly.

14.8 Regional transfer and storage of heat

14.8.1 The psychology of rural water supply in Kosovo

In the year 2000, I spent a very happy and interesting time, travelling the countryside south of Mitrovica, Kosovo, visiting villages and trying to advise on potential improvements to village water supplies, many of which were based on natural springs. We proposed all sorts of nifty schemes based on the construction of spring chambers, the laying of supply lines and the development of village distribution networks. Often, our proposals were met with considerable scepticism because some of the village households did *not* want a communal water distribution net. They wanted their own water supply line direct from a spring to their house. This led to the slightly bizarre concept of a single trench from a spring carrying, not a single, large-diameter supply pipe, serving a village distribution system, but a bundle of small pipes, each conveying water to a single household. The latter approach makes us chuckle a little, because it is inefficient and reveals a lack of trust in communal solutions. However, it is also entirely understandable, especially in a community that had barely started to recover from a civil war.

In the United Kingdom, we should not chuckle too loud, however, because exactly the same mentality exists in respect of residential space heating. Every household likes to have its own combi-gas burner, oil stove, fireplace or (heaven forbid) electric bar heater. The concept of not being in control of your own heat source – that is, joining a district heating scheme – is a fairly alien one (Figure 14.8).

14.8.2 District heating

District heating has never caught on in many countries, despite its many attractions. In Norway (Figure 14.8), it is perhaps understandable, given the low density of population in mountainous terrain. In the United Kingdom, one reason is probably that coal-fired power stations and waste incinerators (excellent sources of surplus heat) have typically been constructed miles away from the main population centres, out in the countryside. In the former communist eastern Europe, district heating is the norm,

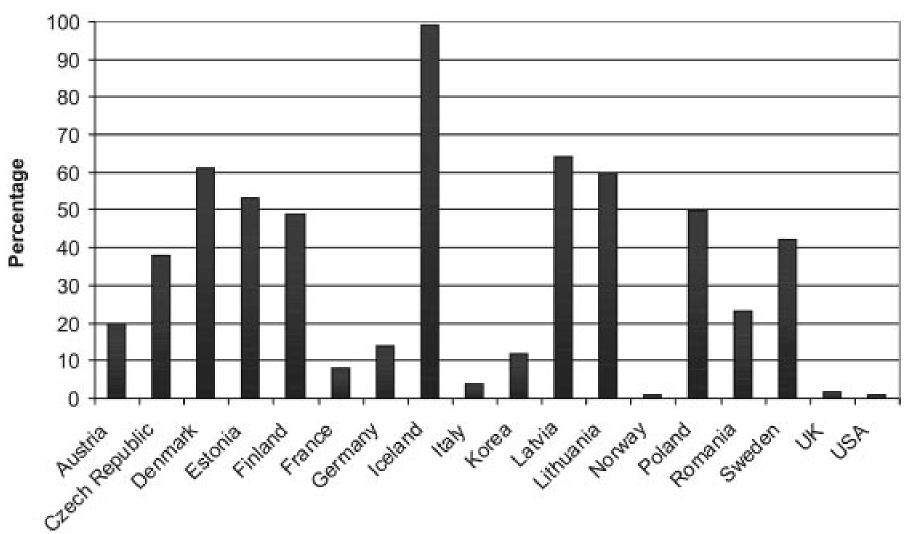

Figure 14.8 The percentage of the population, by country, served by district heating schemes in 2009. Data from Euroheat (2009), UK data from Pöyry (2009).

with power stations being located close to cities and their surplus heat being piped as hot water alongside or under the streets. Several west European countries have also invested heavily in district heating – especially Denmark – and in these schemes, more innovative 'renewable' energy sources, such as biomass or waste incineration, often form a significant proportion of the heat source.

District heating involves the following:

- one or more sources of heat energy (a power station, a CHP plant, a waste incinerator, a heavy industry), supplying a hot fluid (usually hot water) to
- a network of distribution pipes, through which the hot fluid is circulated by pumping, and which in turn feed
- radiators (or other heat exchangers) and/or domestic hot water in residential or commercial properties around the community.

One of the attractions of district heating systems is that they can use a wide variety of heat sources, which may be located on the industrial outskirts of a town. District heating systems allow heat to be transferred from loci of surplus (or waste) heat to loci of heat demand. In reality, no country truly suffers from heat poverty: there is plenty of surplus heat out there – it is often simply in the wrong place at the wrong time. District heating allows heat to be transferred over distances of kilometres, largely solving the problem of heat being in the *wrong place*.

14.8.3 District heating with ground source heat pumps (GSHPs)

There is no reason why conventional geothermal resources (as is the case in Iceland or Southampton – Chapter 3) cannot form the basis or, at least, a component of district heating systems.

Similarly, we have seen that well-designed GSHP schemes can sustainably extract plenty of usable heat from ground or groundwater at 'normal' temperatures. They can also form the basis for district heating schemes. We must remember, however, that the heating distribution network that they feed must operate at a modest temperature (or the temperature must be boosted by a supplementary heat source) to obtain an attractive operational efficiency. Thus, one could envisage a small community of houses being supplied by warm space heating fluid from a central community heat pump plant room, extracting heat from an array of closed-loop boreholes, or from a water well. One would need to overcome several problems, of course:

- the distribution network would have to be well insulated;
- an acceptable ownership and maintenance model for the borehole array and heat pumps must be identified;
- an acceptable means of metering and charging for heat consumed must be found.

However, the Scottish mine water-based schemes in Section 8.16.2 provide examples of clusters of apartments supplied from a centralised heat pump array/thermal store, owned and operated by community housing associations and charged at an attractive rate to tenants.

Alternatively, one could choose a 'distributed' model, as described in Section 7.6.4. A centralised closed-loop borehole array could simply circulate carrier fluid around the members of the district energy scheme. Each house or apartment could own a heat pump mounted on the carrier fluid circuit and would control when and how it was used (heating or cooling mode).

14.8.4 District cooling

Increasingly, larger commercial, hotel and retail properties in large cities have significant cooling requirements in addition to heating requirements. Several cities are now beginning to realise the potential of installing district cooling systems, where chilled water/antifreeze solution is circulated in trenched insulated pipes to serve various customers located throughout a district. Sources of environmental 'coolth' may include cold lakes, fjords, the sea or the ground.

The Stockholm district cooling scheme (Figure 14.9) originally had a capacity of some 60 MW (it has since grown – Capital Cooling, 2005). It uses cold water, at a temperature of +4–6°C, from 35 m depth from one of the city's sea inlets, Värtan, as one of its main sources of 'coolth' (or ultimate heat sink, if you prefer). Heat pumps can be called on to further chill the cooling circuit, if required. An ATES component was added to the scheme to increase its peak capacity. Surplus night-time coolth is

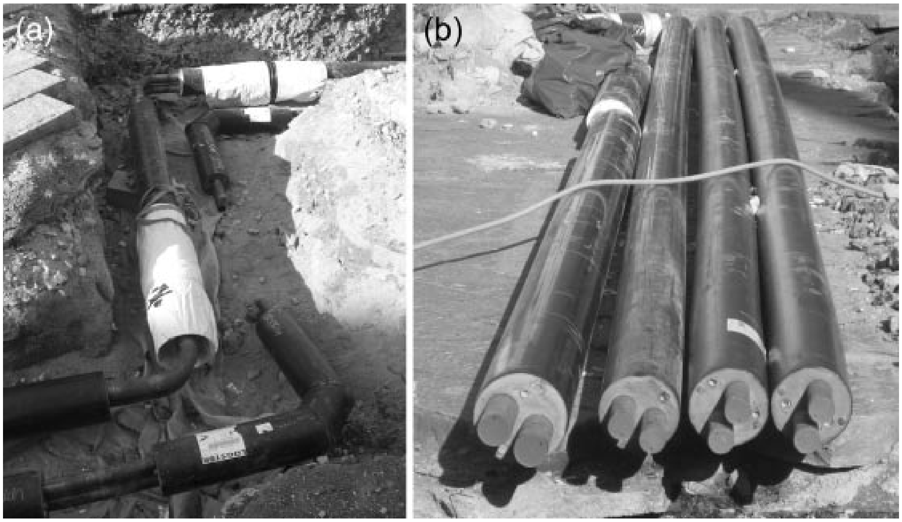

Figure 14.9 (a) Exposed! – part of Stockholm's district cooling scheme, showing (b) the insulated, two channel pipes used in the distribution net. *Photos by D. Banks.*

stored in the ATES scheme, comprising 12 wells with an initial peak yield of $600\,L\,s^{-1}$. The cold and warm wells operate at temperatures between around +4 and +14°C, respectively. At times of cooling demand (generally, daytime), the water from the cold wells is used to chill the district cooling circuit via a heat exchanger – the heat absorbed from the district cooling system results in the temperature of the groundwater being raised to maybe +14°C, which is then recharged to the warm wells. At times of low demand (generally, night-time), groundwater is pumped from the warm wells, its heat is exchanged (discharged) to the near-freezing lake and its temperature drops (to maybe +4°C), and it is recharged to the cold wells (Andersson, 2007b). The wells have experienced problems with sand pumping (with risk of settlement) and iron oxyhydroxide precipitation such that, by 2007, the peak yield was reduced to $360\,L\,s^{-1}$, representing an additional c. 15 MW to the system's cooling capacity.

14.8.5 Multi-nodal district energy systems

A particular advantage of district heating/cooling systems is that they can grow in size and scope. They can incorporate new sources and new customers. Naturally, heat pumps (including GSHPs) fit very well into this picture, as they can supply heat or cold to district heating or cooling systems – or both simultaneously, as all heat pumps have a hot and a cold side.

Box 2.3 describes the Southampton district energy scheme (IEADHC, 2004). It started off as a deep geothermal borehole supplying a district heating scheme – but has evolved into a *multi-nodal district energy scheme*, where a variety of heat sources provide district heating (and electricity through CHP). Moreover, surplus heat is also used to run absorption heat pumps, which support a district cooling scheme. Indeed, such complicated schemes begin to look like complex electrical circuits, where generators and batteries of different types (heat sources and heat pumps) create a potential difference (high or low temperatures), which is transmitted through electrical wires (pipes) to run different electrical components (serve various heating/cooling consumers).

In Section 14.8.1, I asserted that *heat poverty* is a myth. We have *plenty* of surplus and waste heat – but not necessarily in the right place at the right time. We saw that connecting loci of heat surplus with loci of heat deficit via district heating and cooling systems largely overcomes the problem of *wrong place*. I will now argue that heat pumps and thermogeology allow us to overcome the problem of *wrong time*:

- Absorption heat pumps allow us to convert surplus heat in summer (e.g. from a CHP plant or high-temperature geothermal well, or a waste incinerator) into a cooling effect (i.e. chilled water), for which there is a high demand in summer.
- UTES systems (closed-loop BTES and groundwater-based ATES) allow us to store large quantities of heat or 'coolth' from season to season.

In other words, UTES provides as with a thermal 'capacitor' or 'accumulator' (to use the analogy of an electrical circuit). This allows us to incorporate sources of surplus summer heat (waste incinerators, power plants, CHP units, solar panels, large buildings with strong net cooling requirements) into a multi-nodal district energy system, secure in the knowledge that we can use them to charge up a UTES store, which can be 'downloaded' again in winter. Similarly, we can also incorporate heat exchangers installed in pavements for snow melting and de-icing, or heat exchangers in cold winter rivers or lakes, into our district energy system: they will satisfy any winter cooling needs we may have, but any surplus 'coolth' can be stored in a UTES system, to be made available again when we really have a demand for it – in the summer.

UTES is ideal for storage of large quantities of heat: it is probably the only form[2] of thermal storage that can realistically be used for large-scale seasonal storage. Other forms of thermal storage can certainly be incorporated into district energy systems to provide shorter-term storage of smaller quantities of heat – these include tanks of water, production of ice or slurry and other phase change materials.

All of this sounds idealistic – and it clearly takes a huge amount of communal and political will to realise such schemes. However, enough large-scale schemes of this nature are up and running to demonstrate their viability. Those irritating Scandinavian, as ever, seem to lead the way, with complex district energy schemes at several locations. The so-called Bo01 ATES scheme in Malmö, for example, incorporates a seawater-based heat pump and an ATES thermal store into a district heating and cooling network (Andersson, 2007a).

14.8.6 A final thought – regional heat transport with groundwater?

Loci of surplus (waste) heat and heat demand often do not coincide. For example, city centres may be filled with large offices, commercial properties and hotels, all with net cooling demands for much of the year. Residential areas, with net heating demands, are often situated in the outskirts, several hundreds metres or kilometres away. Transport of heat between the two areas could be solved by district heating schemes, although the laying of new pipeline networks in heavily urbanised areas is often impractical or very expensive. We could actually use existing infrastructure, such as transport tunnels, water mains or sewage mains to transport heat – recall (Section 5.4) that Haldane's early heat pump extracted heat from a water main in the 1920s – although such thoughts tend to make utility companies nervous.

Let us suppose that a city centre or industrial area, with a net cooling demand, lies above a major aquifer, directly up the hydraulic gradient from a residential area (in fact, Gandy *et al.*, 2010, described exactly such a situation). The city centre businesses or industries could use groundwater from a well doublet to perform cooling. The warm groundwater with the waste heat would be re-injected to the aquifer and would migrate down-gradient as a thermal plume (Sections 8.11–8.12). Given a long enough

[2] With the possible exception of snow/ice storage.

time perspective – it may take decades and we should remember that most thermal plumes do not extend more than a few hundred metres – this plume may eventually reach the residential area, where the warmed groundwater could be utilised by residential GSHP systems. In such a situation, we would have created an artificial geothermal resource. Most environmental regulators are nervous about thermal plumes and rises in aquifer temperatures, but we should remember that such changes can actually be beneficial for heat consumers.

STUDY QUESTION

14.1 A large building has a peak heating demand of 80 kW. Its annual heating demand is equal to 1800 peak load equivalent operating hours (i.e. 144 MWh). On the coldest winter day, this peak load may be required for a duration of 12 h to maintain an acceptable building temperature. The rock below the site has a volumetric heat capacity of $2.2\,\mathrm{MJ\,m^{-3}\,K^{-1}}$. Assuming a temperature change of 20°C, what volume of (1) water storage tank and (2) BTES system, based on boreholes 35 m deep, would be required to store (i) 1 day's worth of heat demand and (ii) 1 year's worth of heat demand? What do you conclude about the applicability of the two different types of store to buffer diurnal and seasonal variations in heat demand?

15

Thermal Response Testing

Logarithmic plots are a device of the devil.

Attributed to Charles Francis Richter

15.1 Sources of thermogeological data

We have seen in Chapter 10 that, when we are planning a borehole-based closed-loop heating or cooling system, we require values of the ground's thermal conductivity and its volumetric heat capacity as input parameters to the design process. We can acquire these by three routes:

- Generic tables of values (such as Tables 3.1, 3.4 or 11.2). These can be found in many publications; alternatively, databases may be bundled with design software such as Earth Energy Designer (Blomberg *et al*., 2010). In some countries, geological surveys may be able to provide a tailored report of the likely thermogeological parameters at a specified site.
- Laboratory testing. A core or sample of geological material can be returned to a laboratory and the thermal conductivity determined by measuring the temperature change when the material is subject to a constant heat flux.
- *In situ* field tests.

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

15.2 Laboratory determination of thermal conductivity

Several different laboratory methods are available for determining the thermal conductivity of solid samples (see Bloomer, 1981; Midttømme *et al.*, 1998). They may employ steady-state solutions of Fourier's law in several variants, either by (1) applying a constant heat flow to a sample of known cross-sectional area, and measuring the temperature at various points along its length, or (2) applying a known temperature differential across a sample of known cross-sectional area and determining the heat flow through the sample. The 'divided bar' method is an example of such a steady-state method (Benfield, 1939).

One may also use a transient method. One such method involves solving the line-source equation for a needle probe, inserted into a sample of material – as described in Section 15.11. Alternatively, another relatively simple transient method involves applying a constant source of heat to the top of an insulated core or cylinder of sample. The temperature evolution is measured at the base of the sample and the thermal conductivity is derived from the rate of temperature increase. This method is used by several laboratories, including that of the Geological Survey of Norway, as described by Ramstad *et al.* (2009). The method was developed by Middleton (1993), on the basis of equations published by Carslaw and Jaeger (1959). The latter authors considered a semi-infinite slab of material of vertical thickness L at an initial temperature θ_0. They considered the base of the slab to be insulated at a position $z = 0$ and the top of the slab to be subject to a constant heat influx q (W m^{-2}) at $z = L$, commencing at time $t = 0$. The temperature evolution at the base of the slab is then given by

$$\text{Temperature increase} = \left(\theta_{(z=0,t)} - \theta_0\right) = \frac{q\alpha t}{L\lambda} - \frac{qL}{6\lambda} + F(t) \qquad (15.1)$$

where λ is thermal conductivity, α is thermal diffusivity and $F(t)$ is a rather complex function of time. Fortunately, for all but very short times, $F(t)$ tends to zero. This means that the temperature at the base of the slab increases linearly with time t. By plotting temperature change versus time, the intercept of the straight-line portion of the graph on the time axis will yield a value of thermal diffusivity. In other words, when $(\theta_{(z=0,t)} - \theta_0) = 0$, then

$$\frac{q\alpha t_i}{L\lambda} = \frac{qL}{6\lambda} \text{ and } t_\text{i} = \frac{L^2}{6\alpha} \qquad (15.2)$$

where t_i is the intercept on the time axis. This is an elegant method as we don't actually need to know the value of the constant heat flux.

In order to apply this method in practice, one needs approximately one-dimensional heat flow through the core, which is achieved by insulating the side of the core (e.g. with polystyrene; Figure 15.1). The base of the core is likewise insulated and an electronic temperature sensor is attached to the base of the core. In the Norwegian

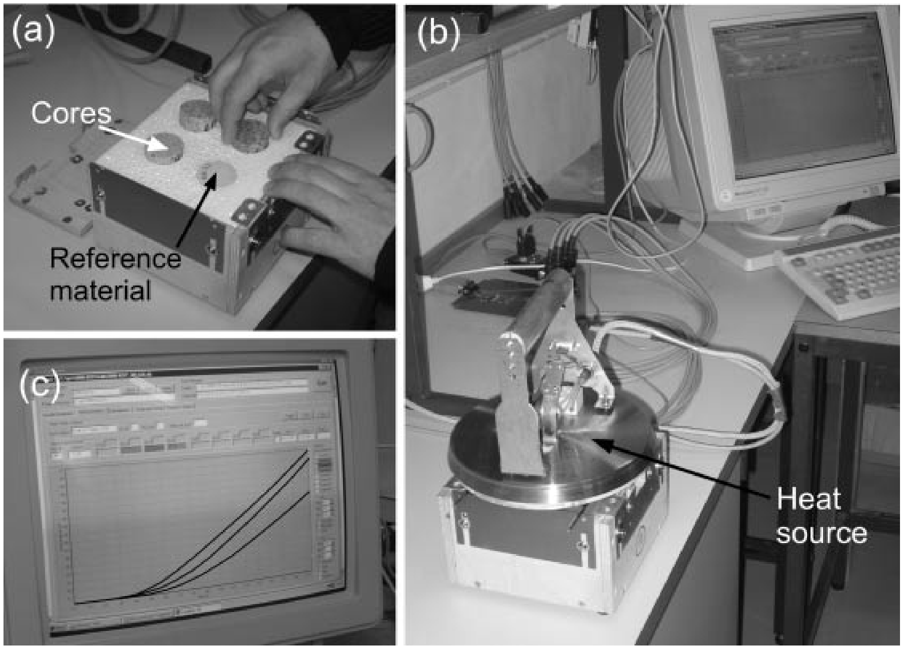

Figure 15.1 A rapid means of measuring the thermal conductivity of cylindrical cores of hard rock in a laboratory (see Section 15.2). (a) Three test cores of rock and one of a standard reference material are placed in a snug insulated holder. (b) A source of constant heat flux is applied (a waffle iron) and (c) the temperature at the base of the cores is measured and the temperature evolution displayed on a computer screen. The thermal diffusivity is calculated from the slope of the response curve. See Ramstad *et al*. (2009). *Photos taken at the Geological Survey of Norway by David Banks.*

apparatus, three 'unknown' cores and one core of known reference material are measured simultaneously in a single polystyrene sample container. The sample(s) are covered by the constant heat source (in Figure 15.1, a Norwegian electric waffle griddle) and the temperatures at the base of the cores are monitored after the heater has been switched on. This method is maybe not the most accurate (but has been tested successfully against other methods), although it is simple and rapid. It also allows validation against the thermal conductivity of the reference material (typically, a substance called Pyroceram). Note that, from this method, we obtain a value of thermal diffusivity. To convert this to thermal conductivity, we need to assume a value of volumetric heat capacity, although this does not vary too much between differing rock types (Table 3.1).

15.3 The thermal response test (TRT)

Clearly, a laboratory test, of the type described above, is of some value. Indeed, many of the data reported in literature (e.g. Table 3.4) are largely based on laboratory determinations. However, such determinations are subject to considerable limitations. One can query whether they are representative: they determine the properties of a sample of only a few centimetres in dimension. In an operational ground source heat scheme, we try to induce heat flow through many hundreds or thousands of cubic metres of rock. If major fractures or discontinuities are present in the aestifer, they are likely to reduce the bulk thermal conductivity of the rock. Such features are unlikely to appear in a core recovered from a borehole (fracturing makes core recovery difficult). Furthermore, ambient groundwater flow in an aestifer may enhance heat transport by advection: laboratory tests reveal nothing of this phenomenon, but it can be identified in field tests (see Section 15.9 and Sanner *et al.*, 2000).

The typical field test used to estimate thermogeological parameters is the *thermal response test* (TRT). We have already seen (Equation 10.19) that the evolution of average carrier fluid temperature (θ_b) with time (t) in a heat-extracting closed-loop borehole is governed by the equation

$$\theta_o - \theta_b \approx qR_b + \frac{q}{4\pi\lambda}\left[\ln\left(\frac{4\lambda t}{r_b^2 S_{VC}}\right) - 0.5772\right] \text{ for } \frac{5r_b^2 S_{VC}}{\lambda} < t < \frac{t_s}{10} \qquad (10.19)$$

where θ_o = average 'far-field' (initial) temperature of the ground over the length of the borehole (K); ($\theta_o - \theta_b$) = temperature 'drawdown' or displacement (K); q = heat extraction rate per metre of thermally active borehole (W m^{-1}); λ = depth-averaged thermal conductivity of the aestifer (subsoil or rock) in W m^{-1} K^{-1}; r_b = borehole radius (m); S_{VC} = average volumetric heat capacity of aestifer (J m^{-3} K^{-1}); t_s = time at which steady state begins to be a better representation of fluid temperature than radial flow (s); and R_b = borehole thermal resistance (K m W^{-1}).

Note that the logarithmic term in Equation 10.14 is written as a natural logarithm (ln or log$_e$). If we now consider heat *injection* to the borehole at rate q, and if we convert the natural logarithm to log$_{10}$ by multiplying by 2.303, this equation can be rearranged as follows:

$$\theta_b - \theta_o \approx qR_b + \frac{q}{4\pi\lambda}\left[2.303\ \log_{10}\left(\frac{4\lambda}{r_b^2 S_{VC}}\right) - 0.5772\right] + \frac{2.303q}{4\pi\lambda}[\log_{10} t] \qquad (15.3)$$

Thus, if we inject heat to a closed-loop borehole at a constant rate q, the temperature will evolve in proportion to $\log_{10} t$. From the rate at which the temperature evolves, we should be able to deduce the ground's thermal conductivity λ. Figure 15.2 shows just such a TRT output plot: heat is being injected to the ground via the closed loop at a rate of around 8.5 kW. The carrier fluid enters the ground loop at a temperature some 3.5°C higher than the return temperature. The mean fluid temperature (θ_b) is calculated simply as the average of the uphole and downhole temperatures.

We can also see that Equation 15.3 should correspond to a straight line if we plot temperature displacement ($\theta_b - \theta_o$) against the logarithm (log$_{10}$) of time (t). The gradient of the straight line is

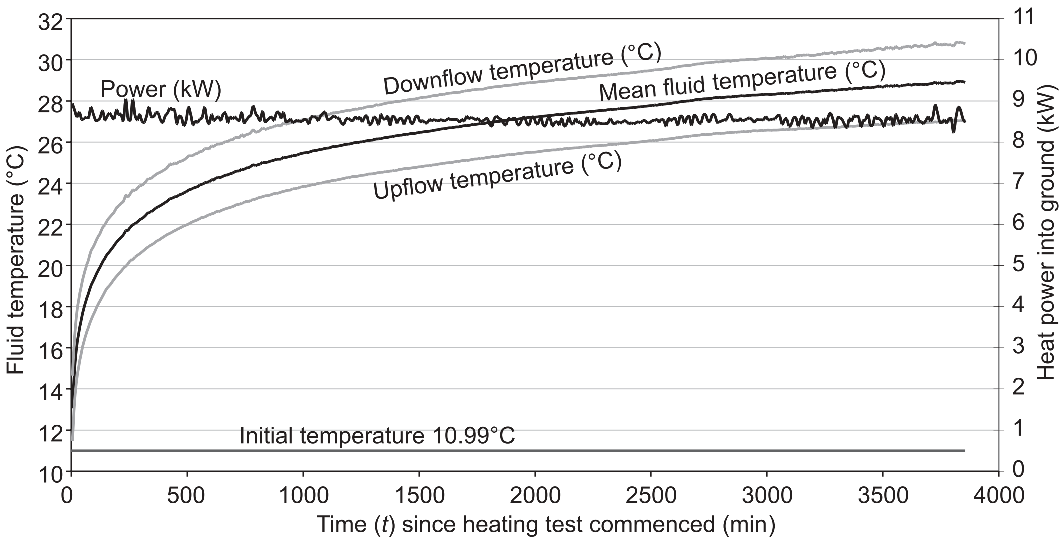

Figure 15.2 Example of output from a TRT, plotted on a linear timescale.

$$\text{Gradient (°C per } \log_{10} \text{ cycle)} = 2.303q/4\pi\lambda \tag{15.4}$$

Because we know q, this enables us to find the thermal conductivity λ. Furthermore, the intercept of the straight line on the temperature displacement axis at $\log_{10} t = 0$ is given by the expression

$$\text{Intercept} = qR_b + \frac{q}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda}{r_b^2 S_{VC}}\right) - 0.5772\right] \tag{15.5}$$

With this equation, we can deduce the value of the borehole thermal resistance (R_b), provided we have a good estimate of S_{VC} (or vice versa).

We should note that Equation 10.19 (and thus Equations 15.3–15.5) is only valid for times greater than a given minimum value. Below this minimum value, the logarithmic approximation is not mathematically valid and one would not expect a straight line on a plot of temperature displacement versus $\log_{10} t$. This minimum required time for a log-linear temperature evolution is usually cited as

$$t > \frac{5r_b^2 S_{VC}}{\lambda} \tag{15.6}$$

although some authors, such as Ingersoll *et al.* (1954) and Gehlin (2002), are more conservative and suggest

$$t > \frac{20r_b^2 S_{VC}}{\lambda} \tag{15.7}$$

For a typical test, with a 140-mm-diameter borehole, $S_{VC} = 2.2\,MJ\,m^{-3}\,K^{-1}$ and $\lambda = 2.0\,W\,m^{-1}\,K^{-1}$, the most commonly accepted criterion (Equation 15.6) yields a minimum time of around 450 min or 7.5 h. Thus, a practical TRT needs to be significantly longer than this minimum time, in order to acquire adequate usable data. It is common to run TRTs for at least 36 h and often 50 h. Note that, the narrower the borehole diameter, the sooner the logarithmic approximation is likely to be valid. For this reason, boreholes greater than 150-mm diameter are seldom recommended for TRTs. While the above equations (Equations 15.6 and 15.7) refer solely to the mathematics of the logarithmic approximation, we should also bear in mind that real closed-loop boreholes have a finite diameter and are backfilled with some material (usually a thermally enhanced grout) whose thermal properties may be different to those of the aestifer. We should therefore add an additional margin to the minimum time of Equation 15.6, in order to ensure that the heat front from the heat exchanger has expanded out into the aestifer, that some form of quasi-steady-state radial flow field has been established through the grout and thus *that we are genuinely measuring the properties of the aestifer material and not of the grout!* Only at a later time will the temperature differential through the borehole backfill approach an approximately stable value: the term qR_b in Equation 15.3. It is quite common for a thermogeologist to discard the first 10–12 h (600–720 min) of any TRT data, even if Equation 15.6 predicts a lower minimum time.

15.3.1 An example of line-source TRT analysis

So much for the theory – let us try it out. Suppose our borehole is drilled in a slatey shale lithology and we have measured the initial undisturbed average ground temperature over the length of the closed loop (θ_o) as 10.99°C. If we plot the curve in Figure 15.2 on a logarithmic ($\log_{10}$) scale, we can see that the data form a pretty good straight line. We will discard the first 800 min of data and draw a straight line through the later part of the data, as shown in Figure 15.3. This line has a gradient of around 5.983 K per $\log_{10}$ cycle. If the heat input during this later part of the test is, on average, 8530 W and the borehole's loop is 115 m deep, then the heat input rate (q) is $74.2\,W\,m^{-1}$. We can use Equation 15.4 to calculate the thermal conductivity as $2.27\,W\,m^{-1}\,K^{-1}$ (or, c. $2.3\,W\,m^{-1}\,K^{-1}$, bearing in mind the uncertainties inherent in the method).

Furthermore, we can estimate that the intercept at $\log_{10}(t) = 0$ (or $t = 1$ s) is –14.14 K. By rearranging Equation 15.5,

$$R_b = \frac{\text{Intercept}}{q} - \frac{1}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda}{r_b^2 S_{VC}}\right) - 0.5772\right] \tag{15.8}$$

Given that the borehole radius is 0.07 m, and assuming a volumetric heat capacity for the rock of $2.3\,MJ\,m^{-3}\,K^{-1}$, which is typical for a shale, we can estimate that the borehole thermal resistance is around $0.08\,K\,m\,W^{-1}$ (quite a low value, possibly typical of a double U-tube heat exchanger).

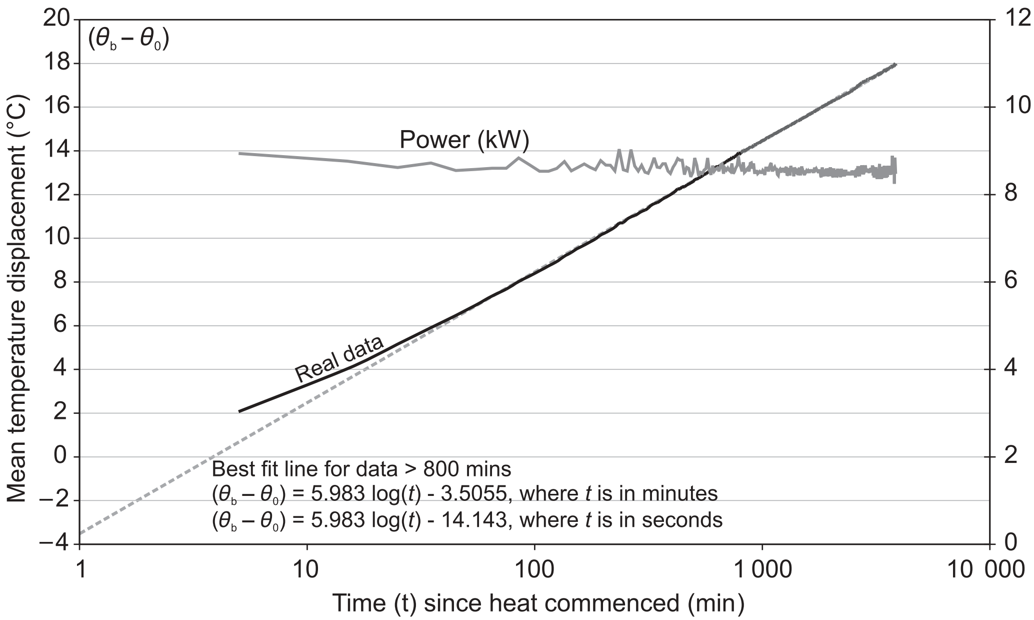

Figure 15.3 Example of output from a TRT, plotted on a log (time) scale, based on the data from Figure 15.2. The temperature displacement is the difference between the average fluid temperature θ_b in the ground loop and the initial average ground rest temperature θ_o.

We can then re-substitute these values of λ and R_b back into Equation 15.3 and simulate the temperature evolution curve to check that it provides a good fit to the observed data. We can also substitute our value of λ into Equation 15.6, yielding a minimum time of 414 min for our logarithmic approximation to become valid. Thus, our decision to neglect the first 800 min of data seems reasonable.

We may choose to calculate a 'running gradient' of the straight-line portion of the response curve, by considering the gradient of progressively later sections of the test data as the test progresses. This allows us to calculate a running estimate of thermal conductivity. If heat transfer from the borehole is solely by radial conductive processes, the gradient should be relatively constant and our estimate of thermal conductivity should not change with time.

15.3.2 *Alternative analytical methods*

The reader should also be aware that other methods are available to 'solve' (or 'inverse model') TRTs than the simple *line source equation* (15.3). These methods include the so-called *cylinder method*, which does not need to make the assumption that the borehole is treated as an infinitesimally thin line-source of heat, and various numerical models. These alternative solutions are discussed by Gehlin (2002).

15.4 The practicalities: the test rig

The TRT, at least in the form described in this chapter, was first carried out in Sweden by Palne Mogensen and two students from Stockholm's *Kungliga Tekniska högskolan* in 1983. Detailed descriptions of testing are provided by Gehlin (2002), Brekke (2002), Signorelli *et al*. (2007) and Banks (2011b). To carry out a TRT, we need, in essence, the following:

- a representative, completed, thermally equilibrated closed-loop borehole;
- a circulation pump and a flow meter for the carrier fluid;
- a source of constant heat input – this can be a (1) large electrical resistance element, powered by a generator; (2) a reliable, calibrated gas burner; or (3) a heat pump, in heating or cooling mode (although care should be taken to ensure that a constant heating/cooling effect is maintained over the expected temperature range of the test).
- temperature sensors mounted at the well head on the upflow and downflow shanks of the ground loop;
- a means of recording data, such as a data logger or computer.

These elements are shown schematically in Figure 15.4. Figures 15.5 and 15.6 show a variety of actual test rigs, ranging from a self-contained trailer-mounted rig using

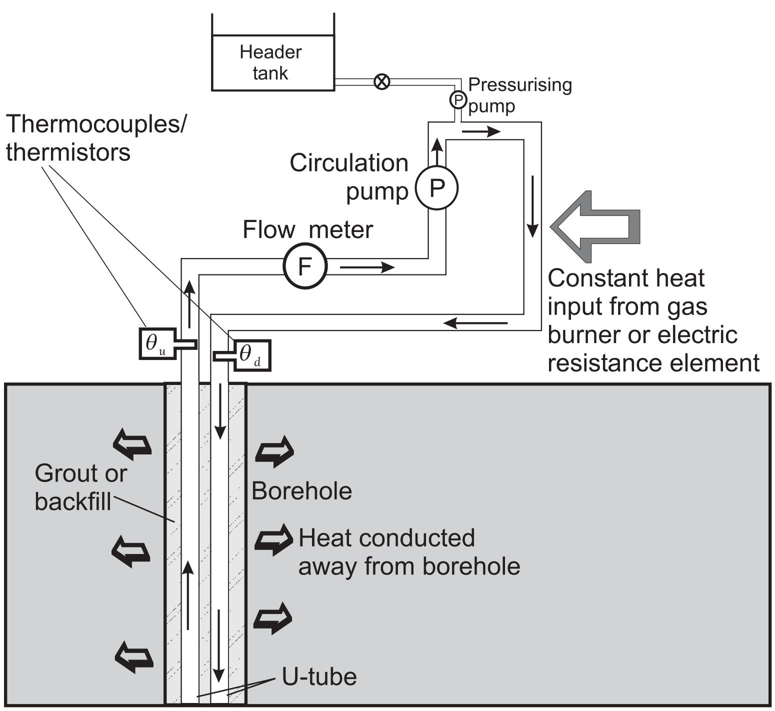

Figure 15.4 Schematic figure of the operational principles behind a TRT rig. A real rig will also include bleed valves and purging circuits to remove air and solid debris from the ground loop.

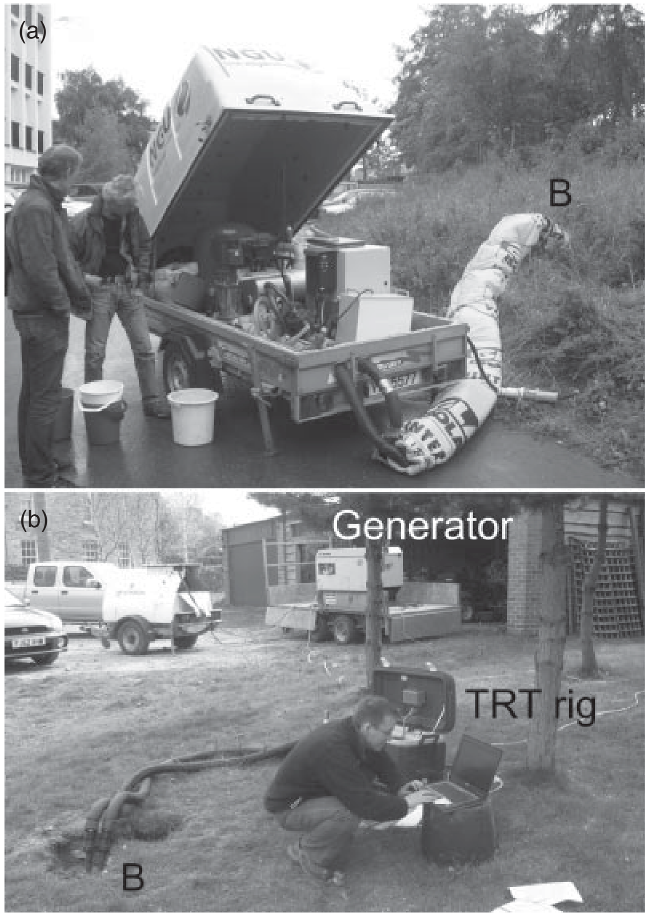

Figure 15.5 Two real TRT rigs: (a) the Geological Survey of Norway's test rig, connected by heavily insulated header pipes to a test borehole in greenstone in Trondheim (*photo by David Banks*). (b) A very small portable rig operated by Carbon Zero Consulting, with its pumps, sensors and electrical resistive heating elements all contained in a small insulated box and powered by an external generator. The rig's data logger is currently being programmed and monitored by an external computer (*photo by David Banks*). In all cases, the borehole head is marked 'B'.

gas burners to a small electric resistance rig contained in a portable box, powered by an external generator. Note, in all cases, that the header pipes between the rig and the borehole need to be efficiently insulated (preferably with reflective insulation) to prevent solar/atmospheric heat gains or losses.

The mean fluid temperature (θ_b) is calculated as the average of the upflow (θ_u) and downflow (θ_d) temperatures:

$$\theta_b = \frac{(\theta_u + \theta_d)}{2} \tag{15.9}$$

Figure 15.6 (a) A self-contained trailer-mounted test rig operated by Geowarmth Heat Pumps Ltd. and powered by liquid petroleum gas (*photo reproduced by kind permission of Geowarmth Heat Pumps Ltd.*). The borehole head is marked by 'B'. (b) The ground loop running from a test borehole at the Annfield Plain site (Box 11.2) to the test rig (*reproduced by kind permission of Pablo Fernández Alonso*); (c) the interior of the test rig (*photo by David Banks*).

The rate of heat rejection to the ground can be calculated by

$$q = q_{\mathrm{heater}} + q_{\mathrm{pump}} = F \cdot S_{\mathrm{VCcar}} \cdot (\theta_{\mathrm{d}} - \theta_{\mathrm{u}}) / D \tag{15.10}$$

where F is the flow rate of the carrier fluid and S_{VCcar} is its volumetric heat capacity. D is the effective depth of the ground loop within the borehole. We can also verify the heat input, in rigs using electric resistive element heating, from the product of the current and voltage applied to the heaters and by adding a small margin to represent the heat derived from the circulation pump. Note that q is the summed heat input from the heaters/burners q_{heater} and the rather small heat input from the circulation pump q_{pump}.

The flow rate should also be selected to simulate the temperature differential $(\theta_{\mathrm{d}} - \theta_{\mathrm{u}})$ that would be typical for a heat pump system (c. 3 to 5°C). The carrier fluid itself can be a solution of the antifreeze that will be utilised under operational conditions. Alternatively, as we are heating (and not cooling) the fluid, it can simply be water. Water has the advantage of a low viscosity, which decreases further with increasing temperature. It is thus easy to achieve turbulent flow conditions. Remember, when analysing the data, that the volumetric heat capacity of the carrier fluid will vary a little with temperature and may need to be corrected to give a reliable value of q. We should check that the test's flow rate F is adequate to yield the transient-turbulent fluid flow conditions ($Re > 2500$) that ensure efficient heat transfer. Even so, the value of R_b derived from tests using water as a carrier fluid *may* tend to underestimate the operational value slightly, as the water flow at high temperatures will typically be far more turbulent than an antifreeze solution at low temperatures.

In practice, using water as a carrier fluid in a 32- to 40-mm outer diameter (OD) ground loop, a flow rate of around 3–3.5 $\text{L min}^{-1}\,\text{kW}^{-1}$ is able to deliver both turbulent flow and a reasonable temperature differential with a heat input of, say, around 5 kW.

We should note, in passing, that we can also perform a TRT by extracting heat from the loop (i.e. chilling the carrier fluid) using a heat pump. The analysis procedures are the same as for heat addition, except that we plot $(\theta_o - \theta_b)$ against $\log_{10} t$, rather than $(\theta_b - \theta_o)$. Signorelli *et al*. (2007) believed that a 'cooling' test produces slightly more representative results, but it is rather more logistically difficult to perform. One advantage of the 'cooling' test might be that it avoids the drying out of the sediment/rock due to vapour migration during heat injection – this could affect thermal conductivity.

15.5 Test procedure

15.5.1 Test criteria

When performing a test, we need to (1) obtain a good estimate of the initial 'rest' temperature of the ground (θ_o, averaged over the borehole length) prior to the test; (2) ensure accurate measurement of fluid temperature and flow rate (turbulent); (3) ensure a stable and uninterrupted power input; and (4) continue the test for a sufficient time so that we obtain a representative temperature evolution curve that reflects conditions in the aestifer and not the borehole backfill.

There are at least two sets of international guidelines on the performance of TRTs:

- those of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE, 2002, 2007), which have in turn influenced the guidelines of the International Ground Source Heat Pump Association (IGSHPA, 2007);
- those published by a working group of the Implementing Agreement on Energy Conservation through Energy Storage of the International Energy Agency (Sanner *et al*., 2005).

In addition, national standards or guidelines may exist (e.g. GSHPA, 2011), which usually tend to reflect the consensus viewpoint of the international guidelines. The salient points of most of these guidelines can be summarised as follows:

- There should be a delay after drilling, loop installation and loop filling, and before a closed-loop borehole is tested, such that the borehole can attain a thermal equilibrium with the ground. ASHRAE (2002) suggested a minimum of 3–5 days (longest delay in low-conductivity formations). Remember also that cement has an exothermic setting reaction (it releases 'heat of hydration' during curing), so that considerably longer delays may be necessary if cementitious grouts have been used.
- Drilling of adjacent boreholes should be avoided during the test period, as this can disturb ground temperatures.
- The borehole diameter should be less than 152 mm (IGSHPA, 2007).
- Any surface portions of the ground-loop circuit should be short and thermally insulated (Sanner *et al.*, 2005). Moreover, the current author suggests that they should be covered in reflective material to minimise absorption of solar radiation. The ambient (air) temperature should also be recorded during the test, so that possible interference from heat 'leakage' at the surface can be identified.
- The system should be purged of air before testing commences (Sanner *et al.*, 2005). It will usually also be pressurised to simulate operational conditions and to prevent pump cavitation.
- Before testing, the initial average ground temperature should be measured either (1) by lowering a temperature probe down the fluid-filled U-tube, measuring the temperature at regular depth intervals (evenly spaced over borehole depth) and averaging these, or (2) by circulating the carrier fluid without heat input for 10–20 min, and averaging the temperature readings as the first volume of carrier fluid (corresponding to the volume of the ground loop) emerges from the upflow shank.
- The test should be at least 36–48 h long (ASHRAE, 2002). Sanner *et al.* (2005) recommended at least 50 h. In practice, the test should be long enough to yield an interpretable straight-line response (Figure 15.3) that is representative of the aestifer's thermal properties.
- The fluid temperature should be measurable to an accuracy of <0.3°C, the power input (heater plus circulation pump) to <2% and the fluid flow rate to <5% accuracy (ASHRAE, 2002).
- A flow rate should be selected that results in a temperature difference of 3–7°C between the upflow and downflow fluid fluxes (ASHRAE, 2002). The flow should also be turbulent (Sanner *et al.*, 2005).
- The selected heat input rate should correspond to around 50–80 $W m^{-1}$ of drilled depth (ASHRAE, 2002) or 30–80 $W m^{-1}$ (Sanner *et al.*, 2005). The lower rates correspond to lower-conductivity formations.
- During testing, the standard deviation in heat power input should be less than 1.5% (and spikes should be less than 10%), according to ASHRAE (2002). For comparison, the standard deviation in power input for $t > 800$ min in Figures 15.2 and 15.3 is around 1%.
- If the borehole needs to be re-tested, at least 10–14 days need to elapse following the cessation of the first test, to allow the borehole to fully thermally 'recover' (ASHRAE, 2002).

15.5.2 Prior to testing

Prior to testing, we have seen that we need to measure the initial 'rest' temperature of the ground, averaged over the borehole length. If the borehole has equilibrated with the ground, this is effectively reflected in the initial temperature of the carrier fluid. For example, in the United Kingdom, the initial temperature of the carrier fluid in a 100-m-deep borehole may increase from 10°C at the surface to 12°C at the base of the U-tube, corresponding to the geothermal gradient (at least, under ideal, undisturbed conditions). θ_o would then be ≈ 11°C.

We can determine θ_o by running a weighted temperature sensor, attached to a measuring tape, down the U-tube (or simply down the groundwater-filled borehole annulus itself, if the borehole has not been backfilled). We would take temperature measurements at regular intervals, having allowed the sensor to equilibrate at each point. A measurement interval of 1–2 m is usually more than adequate. We would thus end up with a fluid temperature log (Figure 15.7a) of the U-tube and could simply take an average of these measurements (e.g. average of 50 measurements, every 2 m, in a 100-m borehole) to obtain a good estimate of θ_o.

A quicker, though possibly less accurate, method is to circulate fluid through the U-tube and the test rig, without any heat input. A 100-m-deep, 40-mm OD [33-mm internal diameter (ID)] U-tube contains ≈ 171 L of carrier fluid. If the carrier fluid is circulating at 20 L min^{-1}, it will take ≈ 8 min to flush the carrier fluid once around the ground loop. Thus, if we switch on the circulation pump, with no heat input, and monitor the upflow fluid temperature for ≈ 8 min, the average reading can be taken as representative of θ_o (Figure 15.7b). However, take care not to continue measurements for too long – the circulation pump may itself contribute a few hundred watts of heat to the carrier fluid, causing its temperature to increase slowly with time, even if the main heaters are not switched on.

Before we start the test, the carrier loop should be purged of any air or solid debris and the circuit pressurised. Many test rigs will be constructed with a side circuit and appropriate valve work to purge and bleed air from the system. Moreover, the rig will usually incorporate some form of filter to ensure that solid debris does not enter the heater/burner assembly.

15.5.3 After the test – thermal recovery

After our test is complete, our instinct will probably be to turn everything off and return home. Before we do this, we should make sure our data are safely saved in a logger or computer (and backed-up). We should check that power input throughout the test has been constant and that we have achieved an adequate temperature response.

If we are in any doubt about the quality of our data, we can perform a *thermal recovery test*. Here, we turn the heat input off, but continue circulating the fluid. The temperature curve slowly returns towards θ_o in a shape that is the inverse of the heating curve (Figure 15.8) and which can be analysed in exactly the same way. In fact, the curve tends to return to a value slightly greater than θ_o, as there will still be a small amount of heat entering the ground from the circulation pump (usually less

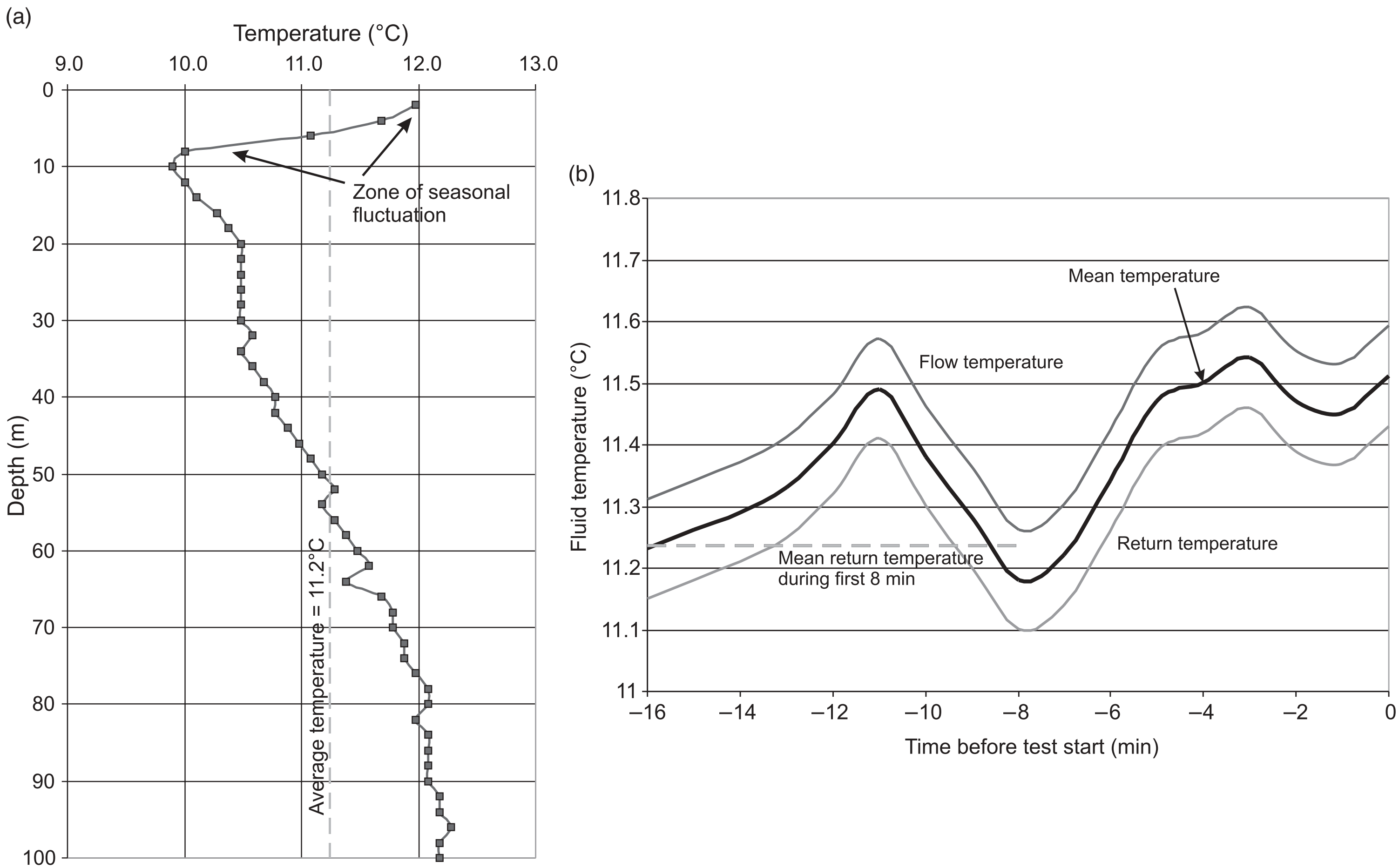

Figure 15.7 (a) An example of the pre-test 'dipping' of fluid temperature in a closed-loop U-tube, to determine undisturbed ground temperature. This figure implies an average undisturbed ground temperature (over the borehole length) of 11.2°C. (b) An example of the pre-test fluid temperatures obtained by circulating the carrier fluid prior to heater switch-on. The temperature rises very slightly with time due to heat from the circulation pump. The average undisturbed ground temperature can be obtained from the average fluid return temperature from the ground over the first fluid cycle(s). This figure implies an average undisturbed ground temperature of 11.2°C. (Both figures were modified after Banks, 2011b, and *reproduced by kind permission of © Holymoor Consultancy Ltd.*)

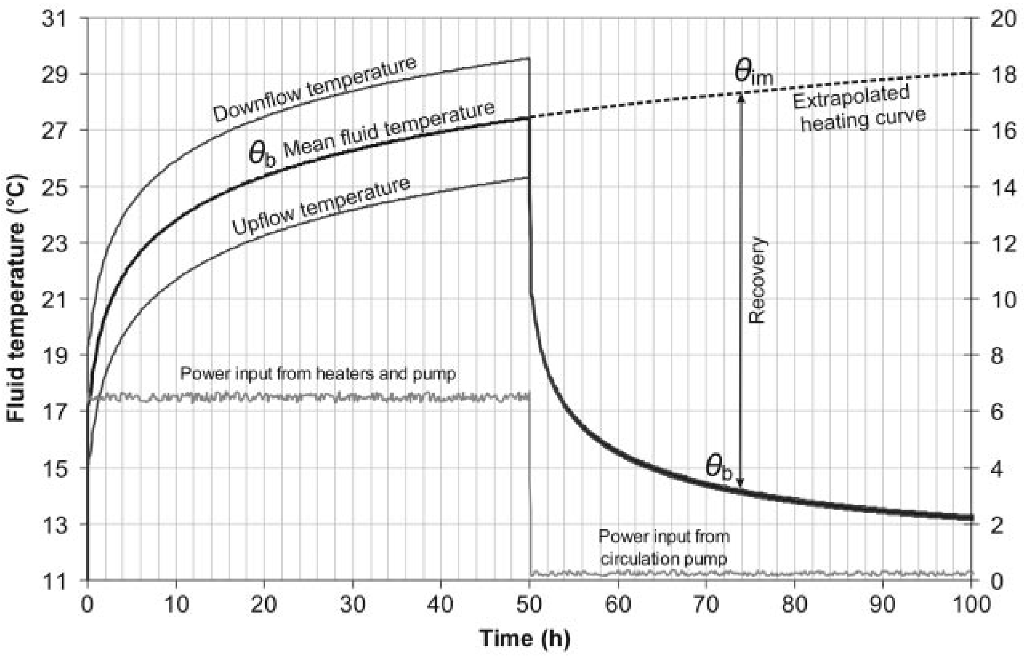

Figure 15.8 An example of the output (fluid temperatures and heat power vs time) from a thermal response test, followed by a recovery test. Refer to the text for further details.

than 300 W). The main drawbacks of the recovery test are, firstly, the additional time required to perform it (and time is money in our society!) and, secondly, the complicating factor that the circulation pump continues to add a small amount of heat to the carrier fluid during the recovery. The latter effect can, however, be factored into the recovery test analysis, provided we can estimate the magnitude of heat input from the pump. For the recovery period (see Figure 15.8), we can say

$$\begin{aligned}\text{Recovery} &= \theta_{\text{im}} - \theta_{\text{b}} \\ &= q_{\text{heater}} R_{\text{b}} + \frac{q_{\text{heater}}}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda}{r_{\text{b}}^2 S_{\text{VC}}}\right) - 0.5772\right] + \frac{2.303 q_{\text{heater}}}{4\pi\lambda}\log_{10} t'\end{aligned} \tag{15.11}$$

$$\begin{aligned}\theta_{\text{b}} &= qR_{\text{b}} + \frac{q}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda t}{r_{\text{b}}^2 S_{\text{VC}}}\right) - 0.5772\right] - q_{\text{heater}} R_{\text{b}} - \\ &\quad \frac{q_{\text{heater}}}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda t'}{r_{\text{b}}^2 S_{\text{VC}}}\right) - 0.5772\right] \\ &= q_{\text{pump}} R_{\text{b}} + \frac{q_{\text{pump}}}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda}{r_{\text{b}}^2 S_{\text{VC}}}\right) - 0.5772\right] + \frac{2.303}{4\pi\lambda}\left(q\log_{10} t - q_{\text{heater}}\log_{10} t'\right)\end{aligned} \tag{15.12}$$

where

- *Recovery* is the difference between the measured average carrier fluid temperature θ_b during recovery and what the average carrier fluid temperature would have been if the heat had not been switched off θ_{im} (i.e. we use an extrapolated heating curve as the 'baseline' against which recovery is measured – see Figure 15.8);
- $q = q_{heater} + q_{pump}$ = total heat input during heating phase of test; q_{heater} = the heat input from the heater/burner that has been switched off (i.e. not including the circulation pump); q_{pump} = heat input from circulation pump;
- t is the time since the heating test started and t' is the time since the recovery started.

Therefore, to derive values of λ and R_b, we need to plot the Recovery against $\log_{10}t'$ (Equation 15.11). Equation 15.12 can be rearranged as follows, if we assume that q_{pump} is negligible compared with q_{heater} (which may not always be the case),

$$\theta_b = \text{Constant} + \frac{2.303q}{4\pi\lambda}\log_{10}\left(\frac{t}{t'}\right) \tag{15.13}$$

In such a case, we can plot the mean carrier fluid temperature during recovery against $\log_{10}(t/t')$ and derive a value of λ from the gradient.

15.6 Sources of uncertainty

It is important not only to derive values of λ and R_b from the TRT but also to be able to give the client some idea of the level of confidence in these values. Gehlin (2002) and Signorelli *et al*. (2007) believed that the accumulated error in the results from a 'line gradient' analysis, of the type detailed in Section 15.3, typically reach 10%. Zervantonakis and Reuss (2006) cited typical levels of confidence of 9% in λ, and 14% in R_b. There are four main sources of uncertainty in a TRT: if we can identify and quantify these, we can begin to place a quantitative value on the uncertainty in our result:

- Uncertainty in measured temperatures: errors in absolute temperatures will affect the derived value of R_b although, provided the relative temperature changes are correct, the slope of the temperature evolution (and thus λ) will not be affected.
- Uncertainty in calculated power input: this is perhaps the main uncertainty, as it will affect the derived values of both R_b and λ.
- Uncertainty in assumed input values, especially of S_{VC}. Note that we have three unknowns (S_{VC}, λ and R_b) in our Equation 15.3, but we can only solve it to yield values of two of them. The confidence that we have in the assumed value of S_{VC} (which does not vary greatly between saturated geological media, but which can

vary by, say, 10–20%) will be reflected in the confidence that we have in our result for R_b (Equation 15.8).

- Uncertainty in the 'straight-line' fit to the log-transformed data: this will affect the derived values of both R_b and of λ, although in the same 'direction' (i.e. the greater the slope of the curve, the lower the derived λ *and* the lower the derived R_b: the uncertainties in the two parameters are thus not independent).

15.7 Non-uniform geology

Our test analysis assumes that the borehole penetrates a single rock type. In reality, the TRT delivers an integrated value of thermal conductivity over the length of the borehole – effectively, a thermal transmissivity. Indeed, we can rewrite Equation 15.4 in terms of the absolute heat input (Q in watts) as

$$\text{Gradient} = 2.303q / 4\pi\lambda = 2.303\text{Q} / 4\pi T_{\text{the}} \tag{15.4}$$

Thermal transmissivity (T_{the}) is defined as the product of thermal conductivity and thickness for each individual horizontal stratum (units W K^{-1}; see Box 10.1). If various strata are present throughout the borehole length, the transmissivity of each unit is summed to yield a total thermal transmissivity.

For example, if our tested borehole (Section 15.3.1) penetrates 21 m of sandy–silty clay ($\lambda = 1.7\,\text{W m}^{-1}\,\text{K}^{-1}$) and then 94 m of a slatey shale ($\lambda = 2.4\,\text{W m}^{-1}\,\text{K}^{-1}$), the thermal transmissivity integrated along the borehole length would be $(2.4 \times 94) + (1.7 \times 21)$ $\text{W K}^{-1} = 261\,\text{W K}^{-1}$. The mean thermal conductivity (which is the result that the test gives us) is then given by the thermal transmissivity divided by the thermally active borehole depth = $261\,\text{W K}^{-1}/115\,\text{m} = 2.27\,\text{W m}^{-1}\,\text{K}^{-1}$ (the result of our analysis in Section 15.3.1).

As the TRT only provides a value of mean conductivity, we can only deduce the thermal conductivity of the slate/shale if we have information (or make an assumption) about the conductivity of the clay.

15.8 Non-constant power input

If we have been unable to maintain a constant power input over the course of the test (e.g. because the efficiency of heat transfer from a gas burner to the carrier fluid decreases slightly as the carrier fluid's temperature increases), do not despair. Provided we know how the power input has varied with time (and we can calculate this from Equation 15.10), we can still solve the maths. We need to do this by somewhat complex inverse modelling techniques, however, and for this we must use a computer. Fortunately, Shonder and Beck (2000) of the United States' Oak Ridge National Laboratory have developed a neat piece of software (named 'gpm' – geothermal properties measurements) to fit parameters to a data set by inverse modelling, producing output in terms of ground thermal conductivity, θ_o and borehole thermal resistance, all with associated estimates of uncertainty.

If the fluctuations in power input form well-defined 'steps' – for example, a power failure of 2 h due to the generator running out of diesel – we can use the concept of superimposed step functions (as discussed in Section 10.7.5) applied to our line source (Equation 15.3) and solve the mathematics in a spreadsheet environment.

15.9 Groundwater flow

External factors can also affect the measured gradient of the curve of temperature evolution (Figure 15.2). Equation 15.3 assumes, for example, that all heat transport from the borehole is by radial conduction. In reality, and especially in aquifer strata, other processes, such as advection with groundwater flow, may also contribute to removing heat from the borehole. This effect is usually relatively minor but, if significant, it may lead to a shallower gradient on the plot of temperature versus log time, and a higher apparent thermal conductivity than would otherwise have been the case (especially for groundwater flow velocities of greater than 0.1 m day^{-1}; Signorelli *et al.*, 2007). Advection with groundwater flow often manifests itself as a progressive shallowing of the gradient of the temperature versus log time curve with time, producing a characteristic upwardly convex shape (Sanner *et al.*, 2000). This is due to the temperature front caused by the test expanding out into the aquifer and intersecting a progressively larger area of groundwater flux. If we calculate a running value of thermal conductivity from progressively later sections of the test curve, we would find that the apparent thermal conductivity would not be constant, but would seem to increase with time.

Indeed, the 'apparent' conductivity in such advection-affected circumstances can reach unrealistically high values. Figure 15.9 shows the results of 35 TRTs from the United Kingdom. Most values are believable: typical average ground temperatures are in the range of 11–14°C (except in Cornwall, where geothermal heat fluxes are unusually high – Figure 3.10). Thermal conductivities are typically around 1.5–2.5 W m^{-1} K^{-1}: low in mudstones, Chalk and basalt; high in sandstones (Sherwood Sandstone, Coal Measures) and greywackes. A few values fall in the range of 8–21 W m^{-1} K^{-1} – clearly too high to represent purely conductive heat transfer and most likely reflecting advection with groundwater flow.

We should *not* use such a 'groundwater-enhanced' value of apparent thermal conductivity for the design of a ground source heating and cooling (GSHC) system, simply because it is not a true thermal conductivity and because heat transfer due to advection with water flow cannot be scaled up to multiple borehole arrays in the way that conductive heat transport can. If we see the effects of advective heat transport in a TRT, we should probably tend to select a lower value of apparent

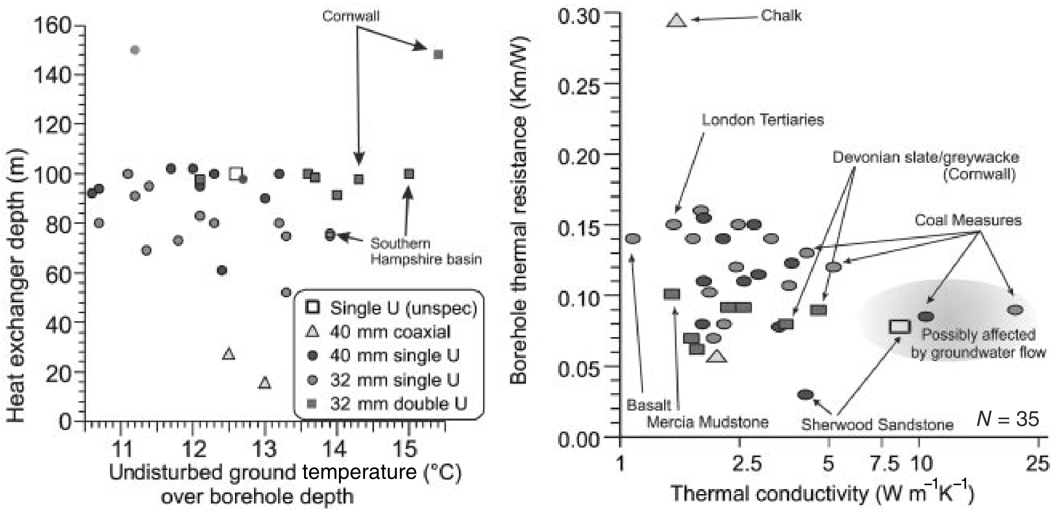

Figure 15.9 The results of around 35 thermal response tests carried out in the United Kingdom: (left) borehole depth versus average ground temperature; (right) borehole thermal resistance versus calculated average thermal conductivity – the shaded ellipse shows the tests that may be affected by advection of heat with groundwater flow. Updated after © Banks *et al.* (2009c). *Copyright retained by and reproduced by permission of the authors.*

thermal conductivity from the slope of the relatively early data (although still greater than our minimum time criterion) as being most likely to be representative of true thermal conductivity.

15.10 Analogies with hydrogeology

The hydrogeologists among you will have spotted that the TRT is the thermogeological analogue of the water well pumping test. In the former, we stress the aestifer by imposing a constant heat input (q), and measure its response in terms of temperature change (θ_b). In the latter, we stress the aquifer by extracting a constant rate of groundwater flow (Z), and measure its response in terms of head change. We analyse aquifer pumping tests by a method exactly analogous to that described in Section 15.3: the Cooper and Jacob (1946) method, which is an approximation of the Theis (1935) equation (see Section 8.6).

Borehole thermal resistance also has a hydrogeological analogue – well loss or well inefficiency (Bierschenk, 1963; Hantush, 1964). Whereas the thermal inefficiency of a borehole is expressed in Equation 15.3 as the term qR_b (i.e. a linear relationship with q), hydrogeological well loss is non-linear and commonly expressed as a power law such as CZ^2 or CZ^n. Here, C and n are simply constants (see Section 8.7.1).

15.11 Thermal response testing for horizontal closed loops

There is no fundamental reason why we cannot carry out a TRT on a horizontal closed loop, using similar equipment and principles to those described above. The interpretation of such a test will not be simple, however. We have already seen that (1) the geometry of horizontal closed loops is seldom straightforward and (2) the loop response may be strongly affected by interaction with shallow soil conditions, moisture migration and the ground surface. Thus, it is tricky to derive a single, simple, reliable equation (such as Equation 15.3 for vertical loops) describing the behaviour of horizontal loops (see Section 11.7). Attempts have been made, however, to interpret horizontal TRTs by inverse modelling (e.g. Fernández Alonso, 2005).

An alternative approach is to make a number of point thermal conductivity tests at various locations within the area in which a horizontal loop is planned to be installed. King *et al*. (*in press*) described the use of a thermal needle probe, inserted into natural subsoils at the base of either (1) a hand-drilled auger hole or (2) a mechanically excavated trial pit, to measure thermal conductivity at a depth of around 1 m. The thermal needle probe is essentially a short (c. 17 cm), narrow (c. 6- to 7-mm diameter) metal rod, inserted into the subsoil and heated at a constant rate by an electrical current (Figure 15.10). The ground and the rod itself heat up in a manner

Figure 15.10 An example of a thermal needle probe being used to measure local *in situ* subsoil thermal conductivity in a 1-m-deep trial pit. *Photo by David Banks.*

described by Equation 15.3. In fact, the narrow rod has the same 'line-source' geometry as a borehole, only much smaller and narrower, so that exactly the same analysis method can be used to derive thermal conductivity. The increase in rod temperature is plotted against the logarithm of time and the thermal conductivity is derived from the slope of the graph. The probe also measures the undisturbed ground temperature prior to testing. Because the probe is far narrower than a borehole, a much shorter test is required before the probe 'sees' the thermal response of the ground: in fact, tests of only 5 min are usually sufficient to return a stable value of thermal conductivity. King *et al.* (*in press*) described the test methodology in far more detail and also evaluated the methodology at two sites in Central England. The methodology is subject to some potential drawbacks, however:

- There will always be some doubt as to whether mechanical excavation or augering has disturbed the soil into which the probe has been inserted (although King *et al.*'s results suggest the outcome to be relatively robust and independent of excavation methodology).

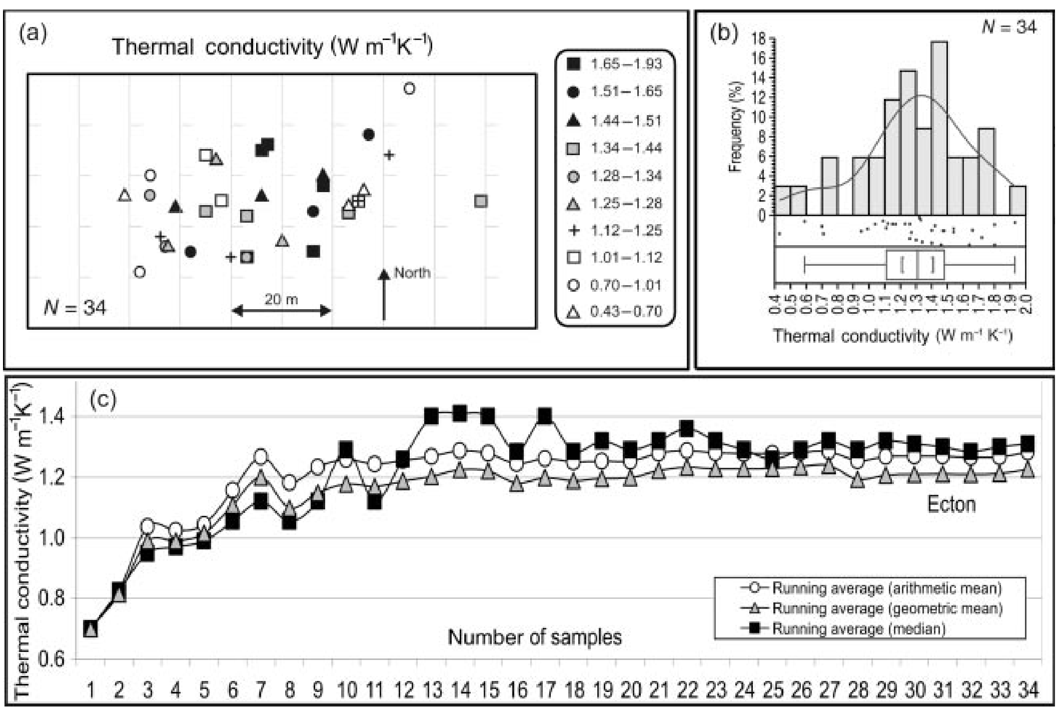

Figure 15.11 The results of subsoil needle probe thermal conductivity testing at around 1 m depth at the Ecton site, Central England. (a) Geographical distribution of determined thermal conductivity across the site ($N = 34$), (b) statistical distribution of thermal conductivities at the same site ($N = 34$) and (c) running estimates of central values (arithmetic mean, geometric mean and median) of thermal conductivity determinations at the site. *All data after Holymoor (2010) and King et al. (in press) and reproduced by kind permission of © Holymoor Consultancy Ltd.*

- The test only yields a thermal conductivity at a given point in time. One must remember that soil temperatures and conductivities may vary seasonally and with changing soil moisture content.
- The test only measures the thermal conductivity of a rather small volume of ground, which is unlikely to be representative of the entire site.

King *et al*. (*in press*) carried out numerous measurements across their test sites. In both cases, they found a wide spread in test results: in one case, the results were distributed rather randomly across the site (Figure 15.11); in the other case, they showed a clear spatial pattern (possibly representing systematic lithological or moisture content changes across the site). King *et al*. concluded that, in order to use this methodology to derive a single representative value of 'bulk' thermal conductivity, which can be used as input to loop design software, at least 12 (and preferably 16) separate determinations were required across the site before the mean and median values stabilised (Figure 15.11c). Based on considerations documented by Bloomer (1981), Renard and de Marsily (1997) and Renard *et al*. (2000), they further recommended that the geometric mean of a large number of separate determinations across a site was most likely to yield a meaningful value of bulk thermal conductivity.

STUDY QUESTIONS

15.1 You carry out a 50-h thermal response test on a 100-m-deep, 140-mm-diameter borehole, with an input power of 6.21 kW. You plot the mean carrier fluid temperature displacement ($\theta_b - \theta_o$) against $\log_{10}$ of time (t) for the data after 10 h and obtain a straight line with the following equation:

$$(\theta_b - \theta_o) = 4.27 \log_{10} t + 1.261$$

where t is in minutes and θ is in K.

Assuming that the rock's volumetric heat capacity is 2.2 MJ m^{-3} K^{-1}, what are your best estimates of thermal conductivity λ and borehole thermal resistance R_b?

15.2 In the test in question 15.1, what is the minimum time (t) at which the logarithmic proportionality assumption is likely to be valid?

A thermal needle test of the type described in Section 15.11 is carried out with a probe of diameter 6.35 mm. If we assume that the soil has a thermal conductivity of 1.4 W m^{-1} K^{-1} and a volumetric heat capacity of 1.5 MJ m^{-3} K^{-1}, what is the minimum time (t) at which the logarithmic proportionality assumption is likely to be valid? How does this compare with the typical test length of 300 s?

16

Environmental Impact, Regulation and Geohazards

The ground remembers major events in its surface temperature history . . .

Lachenbruch and Marshall (1986), also cited by Chapman (2001)

16.1 The regulatory framework

Thus far, this book has argued that the heat stored in the subsurface environment is a resource that can be utilised for space heating and cooling. Moreover, we have seen that it is a resource that can be utilised with a rather low 'carbon footprint' and minimum visual impact. Finally, it is a resource that can be, in many cases, economically competitive with conventional space heating and cooling solutions.

Surely, this technology should be welcomed with open arms and our policy-makers should be removing all barriers to its uptake? If only life were so simple! In practice, the installer of any ground source heating or cooling (GSHC) system will need to overcome certain bureaucratic and regulatory hurdles in most countries.

The first edition of this book attempted to present an overview of regulatory frameworks and legislation for GSHC systems within Europe and further afield. This edition will be far less ambitious – regulations regarding GSHC systems are changing so fast that any attempt to take a snapshot of the situation will become outdated almost immediately.

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

Within Europe, the regulatory framework for GSHC varies strongly (Rybach, 2003b; EGEC, 2006): in the United Kingdom and Ireland, for example, closed-loop GSHC systems are very largely unregulated; in Germany and Demark, regulations regarding the drilling and environmental assessment of such schemes are relatively onerous. In the United States, regulations vary from state to state (Den Braven, 1998; Geoexchange, 2003): guidelines have been developed by the United States Environmental Protection Agency (USEPA) and the National Ground Water Association (NGWA) (USEPA, 1997; McCray, 1998, 1999) and codes of good practice developed by the International Ground Source Heat Pump Association (IGSHPA). Even in Russia, despite the relatively low uptake of ground source heat systems, there is an architectural guideline (Vasil'ev *et al.*, 2001; Vasil'ev, 2003) whose title roughly translates as 'Management of heat pump applications utilising secondary energy resources and non-traditional renewable energy resources'.

The planning, construction and operation of GSHC schemes may fall under one or several different classes of legislation, including

- water resources legislation;
- pollution prevention legislation;
- energy efficiency legislation;
- mining legislation (in some countries, geothermal energy is considered a 'mineral resource');
- specific geothermal energy legislation;
- planning legislation;
- codes of good practice, published by industry or environmental bodies, which may not be legally binding.

In this chapter, we will *not* make any attempt to systematically cover all of these. Be aware of legislation and codes of good practice in *your* country.

In addition to satisfying any national laws and regulations, a developer may need to gain planning permission from a local or regional authority. Planning authorities may impose their own (sometimes apparently rather arbitrary) constraints to ensure that new developments have a minimised CO_2 footprint. For example, at the time of writing, several local authorities in the United Kingdom, including London, operate a policy of demanding a certain percentage of renewable energy (10% in London) to be incorporated into certain new developments (Merton Council, 2011). Again, this plays to the strengths of GSHC solutions and can be seen as a major reason for the recent stimulation of the ground source heat pump (GSHP) market in Britain.

In the following, we will focus to some extent on the European situation, although the general observations and principles apply worldwide.

16.1.1 The Water Framework Directive

European thermogeologists need to be very aware of a few key pieces of European Union (EU) legislation:

- Directive 2000/60/EC – the Water Framework Directive;
  - its Groundwater Daughter Directive 2006/118/EC;
- Directive 2009/28/EC – the Renewable Energy Directive;
- Directive 2002/91/EC on the Energy Performance of Buildings, seeking to enhance the thermal performance of buildings and their energy (and thus, CO_2) efficiency. The directive emphasises the importance of insulation in the winter and thermal performance in the summer, with an emphasis on passive rather than active cooling.

Of particular importance is a formulation in the Water Framework Directive, which defines 'pollution':

> 'Pollution' means the direct or indirect introduction, as a result of human activity, of substances **or heat** into the air, water or land which may be harmful to human health or the quality of aquatic ecosystems or terrestrial ecosystems directly depending on aquatic ecosystems, which result in damage to material property, or which impair or interfere with amenities and other legitimate uses of the environment.

It also defines a 'pollutant' as follows:

> 'Pollutant' means any substance liable to cause pollution.

From the above statements, I would make the following observations (and, please bear in mind that I am not a lawyer):

- The definition of pollution enshrines a risk-based approach, presupposing the presence of a source (substances or heat), a pathway (air, water or ground) and a receptor (ecosystems, infrastructure, amenity value or users).
- Release of heat to the ground can cause pollution . . .
- . . . but *only* if a risk scenario can be envisaged (i.e. a plausible receptor, which might be adversely affected). Under other circumstances, the release of heat to the ground could be construed as creating an anthropogenic resource (i.e. a reservoir of heat that could be re-extracted and used).
- Release of 'coolth' (i.e. extraction of heat) does not cause pollution, according to the Directive, although this activity could potentially damage ecosystems, damage material property (through frost heave) or reduce the utility of the environment.
- Heat is not a *substance* . . . and thus is not a pollutant. As the remainder of the Directive focuses heavily on controlling discharges of pollutants, this creates immediate problems for any regulatory authority wishing to regulate the discharge of heat to the geosphere.

Needless to say, heat is not the only potential form of pollution that we should consider when planning a GSHC scheme. Pollution can also occur due to inadvertent release of refrigerants or antifreezes to groundwater, or due to improperly sealed boreholes providing a conduit for contaminated surface water to enter an aquifer.

European directives need to be translated into national legislation, and this may happen in different ways. In England and Wales, it appears that a very literal interpretation of the Water Framework Directive applies: the Environment Agency is able to regulate discharges of substances (e.g. hot water from an open-loop well-doublet scheme) but not discharges of heat (e.g. from a closed-loop scheme being used for cooling purposes). In other European states, different interpretations prevail and some environmental regulators clearly see closed-loop boreholes as falling within their remit.

16.1.2 Groundwater-based open-loop systems

In general, open-loop, groundwater-based GSHC schemes are more likely to be strictly regulated than closed-loop schemes, because they involve the physical abstraction of groundwater and, usually, its discharge to a natural recipient. One of the results of the Water Framework Directive is that, at least in the EU, most abstractions of groundwater, above a certain *de minimis* quantity, will require a licence or permit. In support of a licence application, a regulator will normally require some kind of survey of potential environmental impacts, a pumping test and a hydrogeological risk assessment to be carried out. The regulatory body will usually determine the licence application on the basis of (1) the lack of any unacceptable hydrogeological impacts on other nearby aquifer users (e.g. other water wells) or groundwater-supported environmental features (e.g. springs, streams, wetlands) and (2) the overall water resources status of the groundwater body (is the aquifer over-exploited or is there plenty of surplus groundwater available). An open-loop scheme that recharges thermally 'spent' water back to the aquifer will normally be more acceptable to a regulator, as the overall groundwater resources of the aquifer are not being depleted and the local hydrogeological impacts will also be minimised.

In addition to an abstraction permit or licence, an open-loop scheme is also likely to require a discharge consent to return the water to a natural recipient (e.g. the aquifer, a river or a lake). In this case, the regulator will consider whether the discharge could result in

- any increased flooding risk in the recipient;
- any change in water quality, remembering that groundwater from a deep aquifer may have a very different chemistry to the water in a recipient river (it may be more saline, more reducing, lower in dissolved oxygen or even contain natural substances, such as hydrogen sulphide, that are toxic);
- any additional contamination risk, due to additives that may have been introduced to the water during its passage through a heat exchange system;
- a thermal risk to ecosystems or water users within, or associated with, the recipient water body.

Typically, warm water discharges are likely to be less acceptable to regulators than cold water discharges, as the solubility of gases, such as oxygen, decreases as temperature increases. Regulators will typically be able to place constraints on the permitted temperature, quantity and water quality of any discharge. The ability to regulate the temperature and the flow rate of a discharge implies the ability to regulate the quantity of heat discharged.

In summary, in the case of open-loop GSHC systems, a regulator will usually be seeking to

- prevent wastage of groundwater reserves by placing limits on the net abstraction rate permissible (or insisting on 100% re-injection of used water back to the original aquifer);
- prevent detrimental changes to the temperature or chemical quality of surface water recipients receiving discharges from open-loop systems;
- prevent or limit widespread (or sometimes even localised) changes in aquifer water levels as a result of abstraction or re-injection;
- prevent large-scale changes in aquifer groundwater temperatures beyond the immediate location of the GSHC scheme. Thus, the regulator may stipulate operational constraints on 'doublet' systems and may limit (1) the net amount of heat that can be abstracted or re-injected over a given period (e.g. one or more annual cycles) or (2) the maximum or minimum acceptable temperature of re-injection of 'spent' water.

If thermally spent water is discharged to a sewer, permission will normally be required from the utilities company operating the sewer and a charge will often be levied.

In some countries, the abstraction and discharge permitting may be covered by a single process, resulting in a unified 'environmental permit' (which may also cover other environmental aspects of the scheme, in addition to the purely hydrogeological).

Installers contemplating a groundwater-based open-loop system should be aware that the entire licensing/permitting process can easily take up to 1 year to complete.

16.1.3 Closed-loop systems

For closed-loop systems, as there is usually no actual abstraction or discharge of groundwater, or any actual discharge of pollutants (only the future possibility of a leakage), the legal tools *may* not exist for environmental regulators to control such systems. This is the case in, for example, England and Wales. This does not mean, however, that the Environment Agency of England and Wales is unconcerned. It has recently issued a comprehensive guidance document on good environmental practice when installing a GSHC scheme, with emphasis placed on groundwater protection (EA, 2011a). It also has the power to serve a notice on irresponsible drillers/installers who they believe to be threatening groundwater resources (e.g. by drilling through a heavily contaminated site or by drilling in a well's source protection zone), stopping an installation from proceeding. The Agency has also worked closely with the UK trade body, the Ground Source Heat Pump Association: several of the Environment Agency's concerns are reflected in the Association's recently published standards (GSHPA, 2011). The English approach has thus been to encourage voluntary adherence to good environmental practice as regards closed-loop systems. If things go awry – if a closed-loop borehole leaks antifreeze into an aquifer or if a driller unwittingly penetrates an artesian aquifer – the Environment Agency has a range of powers to prosecute the offending organisation.

In other countries, even the smallest closed-loop system may be subject to regulation. In Denmark, for example, all closed-loop systems require an environmental permit from the municipality and are governed by specific regulations (*Bekendgørelse* 1203, among others – see Villumsen, 2008). This legislation specifies the minimum distance between adjacent closed-loop boreholes (20 m), between a closed-loop borehole and someone else's water supply borehole (300 m) and the permitted types of antifreeze [ethylene glycol, propylene glycol or so-called 'IPA spirit', comprising ethanol denatured with 10% isopropanol (IPA)]. In Germany, closed-loop systems are also tightly regulated (including the distance that a closed-loop borehole can be drilled relative to a neighbour's property boundary – not less than 6 m, typically). The fact that closed-loop systems affect the temperature of groundwater means that they can be regulated under German water law (Rybach, 2003b).

16.2 Thermal risks

Some environmentally concerned people believe that the temperature of the ground is constant and determined by nature and that any change in the ground temperature is unacceptable. Unfortunately, all GSHC schemes work by changing the temperature of the ground – we cannot have the cheap, low-carbon energy that ground heat represents without accepting at least modest, local changes in ground temperature. In much the same way, we cannot extract pure potable groundwater from an aquifer without accepting local changes in the groundwater levels.

Of course, we can over-abstract groundwater, such that the rate of abstraction exceeds the rate of replenishment from rainfall and from the surrounding aquifer. In such a case, changes in the water table become unacceptably large or widespread. In the same way, we *can* potentially over-abstract heat or 'coolth', if our extraction or rejection of heat exceeds the capacity of the ground surface and surrounding aestifer to replenish or disperse that heat. It is still not wholly clear, however, what magnitude of ground temperature change is deemed 'unacceptable'. Changing ground temperatures can have geotechnical impacts or, conceivably, microbiological or geochemical impacts. These will be considered later in this chapter. In this section, we will simply consider the direct thermal impacts.

16.2.1 Thermogeological 'background' conditions

It is axiomatic that, in order to be able to identify the effects of anthropogenic pollution and environmental impact, we need first to understand the 'background conditions' (Banks, 2004). For example, if we detect high concentrations of arsenic in an aquifer, we might be tempted to jump to the conclusion that it is due to 'pollution' from a local wood impregnating works. However, we must first establish that the groundwater does not contain naturally high background concentrations of arsenic before we can begin to blame anyone. The same applies to thermal pollution. If we spot an area of apparently elevated groundwater temperatures in a city, we should not automatically assume that it is heat pollution from an open-loop cooling scheme. We first need to identify the natural distribution of groundwater temperatures in the aquifer beneath the city (and it is a three-dimensional distribution – Headon *et al.*, 2009).

Furthermore, we need to recognise any temporal trends in ground or groundwater temperature. We have learned in Chapter 3 that ground temperatures are generally fairly stable and that they reflect the annual average air temperature. A geologist, of course, should be aware that local and global temperatures are not static and we would expect ground temperatures to change in parallel with these.

Global temperatures have warmed considerably since the last ice age, around 10 000 years ago. In fact, temperature profiles in boreholes still bear the thermal signatures of the past ice ages and of isostatic crustal rebound (Šafanda and Rajver, 2001). What we now call recent 'global warming' or 'climate change' (and which almost certainly has anthropogenic and natural components) would also be expected to express itself as small increases in ground temperature (Lachenbruch and Marshall, 1986; Beltrami and Harris, 2001; Bodri *et al.*, 2001; Chapman, 2001; Majorowicz *et al.*, 2006; Kharseh, 2011; Kharseh *et al.*, 2011). Ground temperatures might also be affected by changes in the depth and duration of snow cover and the timing and amount of recharge (Box 3.6).

The last few centuries have also seen significant changes in the local temperature environment, especially in cities. The 'urban heat island' effect, where the air temperatures in cities may exceed rural temperatures by several degrees (Henson, 2006; Mayor of London, 2006), due to changes in the albedo, radiative and storage properties of the urban environment, may be reflected in the temperature of the subsurface. Moreover, surprisingly large changes in ground temperature may be ascribable to all the heat lost by downward conductive heat flow from warm house basements and floors, buried services or even by leakage of warm effluent from sewers (see Figure 3.9). For example, de Beer (2008) presented data suggesting that a Norwegian underground car park has elevated ground temperatures in the vicinity.

Indeed, while we have been busy monitoring groundwaters for minute concentrations of trace organic contaminants in city environments, it seems that hydrogeologists may have overlooked one of the most obvious forms of ground 'pollution' to affect urban areas. In a recent, but remarkable, piece of detective work, Ferguson and

Woodbury (2004) have mapped groundwater temperatures beneath the city of Winnipeg, Canada. The regional aquifer temperature outside the city is typically below 6°C. However, below the urban area, temperatures were found to be some 2–3°C higher than this background, and in some cases, 5°C higher. Ferguson and Woodbury considered whether this could be due to global or local atmospheric warming: they found, however, that Winnipeg's annual average air temperature had only risen by around 1°C over the past century (from +1–2°C to +2–3°C). They also considered whether the increase in aquifer temperature could be due to 'thermal plumes' from active ground source cooling schemes (where warm waste water is injected to the aquifer). They found that, while this could have important local effects, it could not account for the regional anomaly beneath the entire city area (Figure 16.1). They eventually concluded that the effect was largely due to downward heat leakage from the warm urban environment. Unlikely though this seems, Ferguson and Woodbury marshalled evidence from borehole temperature logs to demonstrate that, beneath old built-up parts of the city, the geothermal gradient was reversed (i.e. temperatures decrease with depth) down to depths of up to 100 m. This demonstrates that there has been downward conduction of heat into the subsurface from the surface during the past century or so. The downward speed of this temperature 'signal' is governed by the thermal diffusivity of the rock (see Box 3.7 – which demonstrates that such a mechanism is able to account for the observed gradient reversals). Does this constitute 'thermal pollution'? In one sense, it does, as it is a significant anthropogenic disturbance of natural conditions. In another sense, it may be beneficial – the elevated ground temperature allows ground source heating schemes to perform more efficiently. The downside, of course, is that ground source cooling schemes would perform less efficiently.

The Winnipeg story is far from the only example of a 'thermogeological urban heat island': similar results and 'reversed' geothermal gradients are being reported from Sweden (Hellström, 2011a); Osaka, Japan (Taniguchi and Uemura, 2005); Ireland (Allen *et al.*, 2003; Goodman *et al.*, 2004); and England (Figure 3.9 and Banks *et al.*, 2009a).

Thus, I would merely point out to those who worry about GSHC schemes altering the temperature of the ground – *too late!* It's already happened, not because of GSHC, but because city dwellers simply like living in warm houses. The temperature of the ground beneath cities has been altered to considerable depths – *and nobody (not even most geologists) has noticed!*

16.2.2 Local thermal impacts of closed-loop schemes

If you've successfully attempted study question 10.1, you will have found that the zone of ground temperature change around an operational closed-loop borehole is rather limited. The radius of influence typically does not reach much more than 20 m (Figure 10.6; Gabrielsson *et al.*, 2000), and most of the long-term temperature drop occurs within 10 m of the borehole. Thus, if two adjacent closed-loop ground source

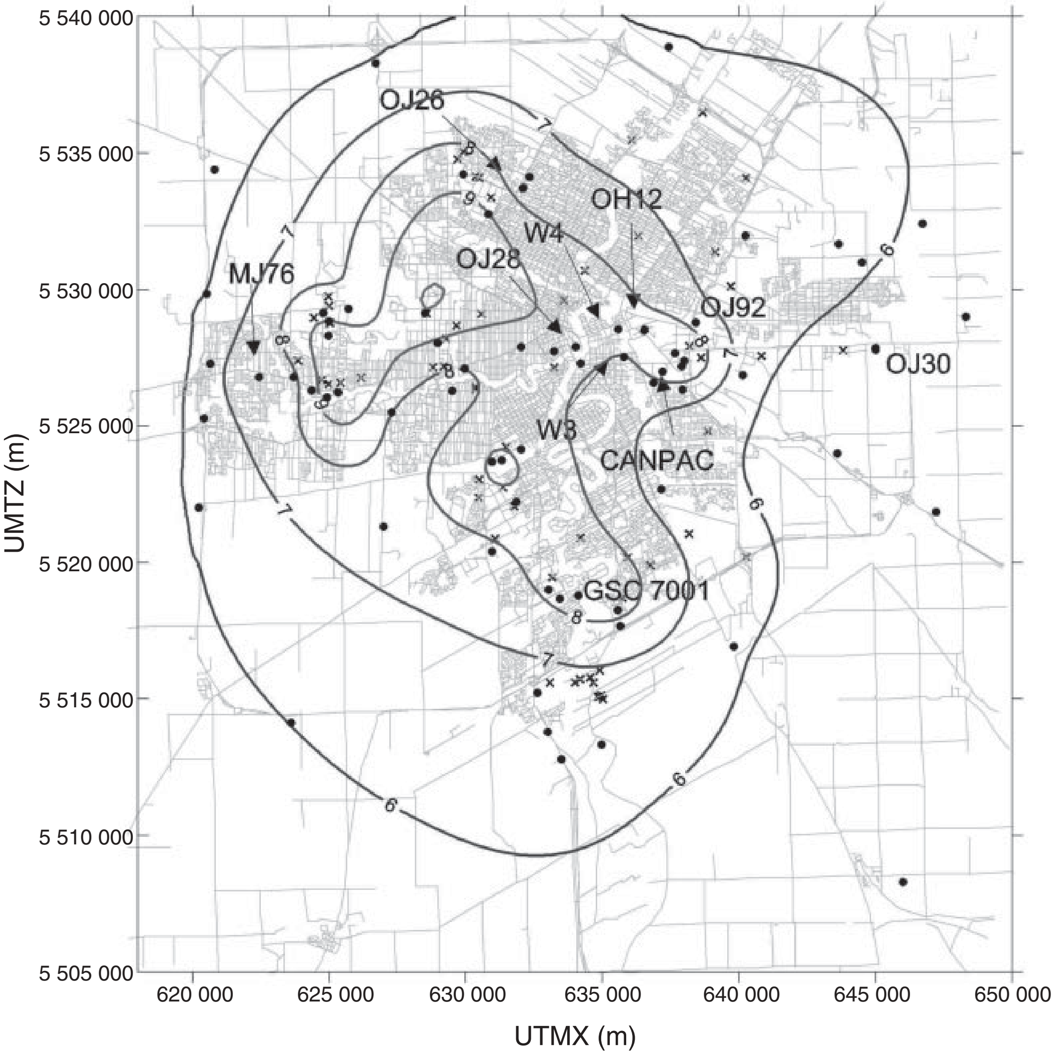

Figure 16.1 Contour map of the Winnipeg area showing temperatures in degrees Celsius (nominally at c. 20 m depth) in the Upper Carbonate aquifer, and locations of temperature measurements (dots). The regional background aquifer temperature is believed to be <6°C. The road network is shown in grey. Locations of injection wells are shown as crosses (after Ferguson and Woodbury, 2004). *Reproduced by kind permission of © American Geophysical Union.*

heat schemes are located >20 m apart, their impact on each other should be rather minor.

Within a distance of 20 m, the two ground source heat schemes could interfere with each other to some extent, to the detriment of their operational efficiency. Indeed, Danish law (*Bekendgørelse* 1203) specifies a minimum distance of 20 m between adjacent closed-loop boreholes. In the German länder, closed-loop boreholes have to be a minimum distance (typically around 6 m – LLUR, 2011) from the boundary of the owner's property (effectively securing a minimum distance of c. 12 m between adjacent schemes). In the United Kingdom, no such regulation exists and litigants would be left to fight out a 'nuisance' claim under Common Law in court.

Of course, the above observations presuppose that the two adjacent schemes are heating dominated. If one was used for heating and one for cooling, the proximity of the two schemes would be to the mutual advantage of both. This applies to both open- and closed-loop schemes (see Section 14.8.6 for more on this).

One could also argue that, if closed-loop ground source heating schemes cool down the ground, they might affect the viability of plants and ecosystems depending on that ground. The greatest impacts might be expected from horizontal closed-loop schemes, where the heat extraction (or rejection) takes place close to the surface. It is a plausible scenario – I merely observe that evidence for significant impacts from well-designed schemes is lacking. The soil is, after all, usually exposed to temperature variations of at least 20°C throughout a year, so it is perhaps unsurprising that it is relatively robust as regards further small man-made changes. Ball *et al*. (1983) and Skarphagen (2006) reckoned that a correctly dimensioned horizontal closed-loop system in a trench should not delay the winter thaw and the growing season of a lawn above it by more than around 2 weeks. However, Skarphagen also recommended avoiding constructing closed-loop systems within the radius of the crown of a tree (which will often approximately correspond to the radius of the root bole). It is unclear if this recommendation is simply to avoid damage to roots during excavation or drilling, or to avoid damage due to temperature changes during operation. For systems rejecting heat at high temperatures, VDI (2001b) suggested that changes in microbiological communities in soils may occur and that a risk assessment should be carried out to identify risks to macroflora. At temperatures in excess of 60°C, soil sterilisation may begin to occur.

Where we use closed (or open) systems to extract or reject heat from/to lakes or ponds, we have already noted the need to consider aquatic ecology in Chapter 12.

16.2.3 Larger-scale open-loop schemes/closed-loop schemes in aquifers

One could argue that larger-scale open-loop well-doublet schemes (i.e. where thermally spent water is re-injected to the aquifer) have a greater potential to cause far-reaching temperature changes than closed-loop schemes, simply because they employ groundwater flow as a carrier for heat. Many large-scale open-loop schemes are cooling dominated, resulting in the rejection of warm groundwater, simply because many large buildings are dominated by a net cooling demand. Thus, large groundwater-based well-doublet cooling schemes result in a 'thermal plume' of warm groundwater that migrates down the hydraulic gradient (in fact, large closed-loop cooling schemes, drilled into a major aquifer, would also emit a similar thermal plume that travels passively down-gradient). Fortunately, the shape and speed of migration of this plume are susceptible to mathematical analysis – see Sections 8.11 and 8.12.

It turns out that heat rejected to groundwater usually travels much more slowly and over shorter distances than most hydrogeologists intuitively expect. This is for two reasons: (1) the ground has such an enormous capacity to absorb heat (volumetric heat capacity) that heat transport is significantly retarded with respect to groundwater flow (Section 8.11); and (2) three-dimensional heat conduction – that is, conduction of heat from the aquifer vertically to the ground surface and to overlying and underlying rocks – efficiently attenuates the thermal plume (Section 8.12). Both modelling (Carbon Zero Consulting, 2010) and empirical studies (e.g. Todd and Banks, 2009 – see Section 8.16.1) suggest that it takes a very large groundwater cooling scheme to have significant thermal impacts beyond, say, 200–250 m.

However, such a thermal plume could have impacts, the risk of which needs to be assessed:

- A plume of warm groundwater could affect other aquifer users, such as well owners. Would the owner of a well really object if the temperature of their water increased by around 2°C? They may do, but they might also find that the cost of domestic water heating is reduced!
- A plume of warm groundwater could affect adjacent GSHC schemes (open or closed loop). The impact would be detrimental if the adjacent scheme was dominated by a cooling demand, but would actually be beneficial for schemes dominated by a need to *extract* heat.
- A plume of warm groundwater could eventually discharge at the surface and result in temperature changes in springs, baseflow-fed rivers or wetlands. Unless the heat source (the cooling scheme) is immediately adjacent to the spring or river, it is very difficult to imagine scenarios where the result would be a really significant change in temperature in the recipient water or wetland. Any temperature change could affect the ecology of the wetland or watercourse, but it seems that sensitive aquatic species such as salmonids and cyprinids are able to tolerate temperature fronts of 2–3°C without significant ill effect (Turnpenny and Liney, 2007).

The density of the proposed open-loop groundwater cooling schemes in some cities has become so large that regulators fear aquifer-wide changes in temperature (Box 16.1). This is a plausible scenario, although we should recall the huge volumetric heat capacity of the ground. Calculations of the type shown in Section 14.1 suggest that heat rejection densities need to get up to values of tens of MWh per ha per annum for a 50-m aquifer before serious temperature change becomes likely. Ferguson and Woodbury (2006) considered four groundwater well-doublet cooling schemes operating in one region of Winnipeg, Canada, and observed that the thermal plumes related to the schemes were overlapping with each other to create a larger coherent region of elevated groundwater temperature. In other words, the four schemes were thermally and hydraulically interfering with each other. They concluded that there is a limit to the number of groundwater-based doublet cooling schemes that a given aquifer unit can support. Clearly, this will depend on the properties of the aquifer and on the water

BOX 16.1 The Chalk Aquifer of London

The Chalk of London had, for some time, been regarded as an under-abstracted aquifer. Indeed, due to the closure, in the mid-to-late twentieth century, of much industry that formerly abstracted Chalk groundwater, there was concern over the rate at which groundwater levels were recovering towards their pre-industrial condition (Banks, 1992). For a period, the regulatory authorities tried to encourage increased abstraction in an attempt to stabilise the rise in groundwater levels (and thus protect underground structures from flooding). Indeed, Ampofo *et al.* (2004b) documented a number of open-loop groundwater cooling schemes in London, all of them with spent water being run to sewers or to the River Thames, rather than being injected back into the Chalk. More recently, however, the rate of abstraction has increased to a level where the Environment Agency is once again beginning to limit abstraction licenses in several areas of the city (EA, 2005, 2006). Those proposing open-loop groundwater heating or cooling schemes are now being encouraged to re-inject spent water back to the Chalk. At the same time, a requirement for most substantial new developments in London to incorporate at least 10% 'renewable' energy (Merton Council, 2011) has led to an explosion in ground source heating and (especially) cooling schemes based on the Chalk aquifer. The density of the proposed open-loop well-doublet groundwater cooling schemes has now reached such a level (EA, 2011b) that the Environment Agency is becoming concerned about the sustainability of the thermal balance of the aquifer and the risk of regional aquifer warming.

The London Chalk story is a re-enactment of the situation that took place in Brooklyn and Long Island in the United States in the 1920s and 1930s (Kazmann and Whitehead, 1980). Here, there was reportedly a rapid uptake of the use of groundwater for providing cooling for buildings. The spent groundwater was simply run to waste, until the local authorities became concerned that groundwater levels in the aquifers were being depleted. Users of groundwater were then encouraged to return spent water to the aquifers. Problem solved? Not wholly! Kazmann and Whitehead reported that concern then steadily grew over the possibility of regionally rising water temperatures within the aquifer system.

and heat fluxes from each scheme, but in the Winnipeg case, Ferguson and Woodbury (2006) recommended that schemes should be spaced no closer than 500 m from each other (i.e. four systems per square kilometre). Gringarten (1978) provided a mathematical analysis of the optimum patterns and spacings for adjacent well doublets in geothermal reservoirs.

When deciding whether to permit a well-doublet scheme in an area where a potential thermal risk or non-sustainability has been recognised, a regulator may opt to limit the permissible temperature differential between natural groundwater and injected water. In Manitoba, Canada, regulators limited groundwater cooling schemes to a 5 K differential (Ferguson and Woodbury, 2005). In England, regulators in London have, over the past few years, tended towards the view that a differential of <10 K is acceptable. In Lombardy, Italy, legislation is also currently unclear, but regulators talk of differentials no higher than 3–5 K being permissible (U. Puppini, *pers. comm. 2006*). In Holland, permitted re-injection temperatures vary from province to province but are typically no cooler than +5°C and no warmer than +25°C (Vos, 2007). The logic behind these differentials is hard to fathom and it may be that limits on net quantities of heat (in joules) being abstracted or rejected are more meaningful than temperatures.

Moreover, given the inherent limits on the sustainability of large unbalanced, non-reversible open-loop doublet schemes, some regulators (EA, 2007; Vos, 2007) are beginning to express a strong preference for an approximate balance between heat re-injected to and heat abstracted from groundwater in major well-doublet schemes. In other words, they are promoting aquifer thermal energy storage (ATES) rather than simple unidirectional heating or cooling schemes.

In addition to concerns over direct thermal impacts and thermal sustainability, one can also envisage geotechnical, geochemical or microbiological effects of temperature changes in the ground. These will be discussed in more detail in subsequent sections.

16.3 Hydraulic risks

Open-loop GSHC schemes often involve the abstraction of water from one well and its re-injection to another. Chapter 8 should have introduced us to the idea that this results in a drawdown in groundwater level around the abstraction well and an upconing in groundwater level around the recharge well (Figure 16.2). Thus, the act of pumping can result in water levels being lowered in other nearby water wells (resulting in increased pumping costs for users), reduced flows in springs/spring-fed rivers or reduced water levels in groundwater-fed ponds or wetlands. In the worst cases, it can result in wells or springs drying up. The risk of this occurring will normally be assessed by the environmental regulator when a licence to abstract water is sought. Lowering of groundwater levels can also (in relatively rare cases) cause subsidence of surrounding land – partly due to the elastic responses of the sediments and rocks to decreases in pore water pressure and partly due to the drying out of clay-rich or organic-rich soils (Box 16.2). The latter form of subsidence is usually the more serious.

Where water is being re-injected to an aquifer, the reverse can happen – groundwater levels can rise and land heave can very occasionally occur. Cellars or basements may become flooded. If the groundwater level is naturally near to the surface, we may find that it rises above ground level when re-injection commences – and may start pouring out of the re-injection well top. The obvious solution is then to seal the well top and re-inject under pressure. However, if the aquifer is unconfined, there is a risk that the

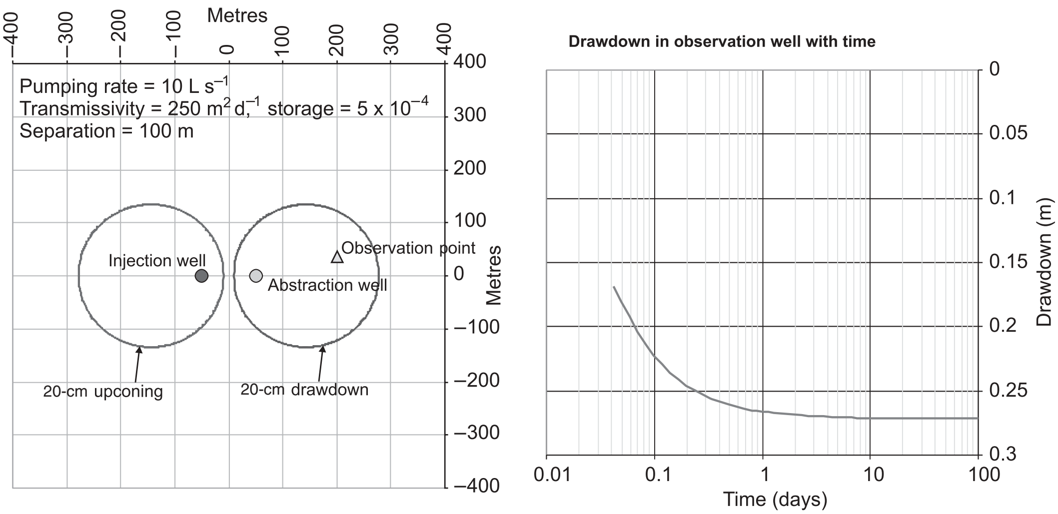

Figure 16.2 The theoretical development of drawdown around a well doublet (of separation 100 m and pumping rate 10 L s^{-1}) in a confined aquifer of transmissivity 250 m^2 day^{-1} and storage 5×10^{-4}. Left: a plan view of the 20-cm equilibrium drawdown and upconing contours around the wells. Significant drawdown does not extend more than c. 200 m from the wells. Right: the evolution of drawdown in the observation well with time (an equilibrium drawdown of c. 27 cm is achieved within a few days).

BOX 16.2 Shanghai, China: Cooling, Subsidence and Artificial Recharge

The Chinese city of Shanghai sits on top of a hydrogeological 'layer cake' of sedimentary aquifers and aquitards, related to various depositional phases of the Yangtze River delta. In fact, it is generally recognised to overlie a 300-m-thick succession of five confined aquifer horizons. As long ago as the 1860s, the city was abstracting water from the two shallowest confined aquifers. As the water was relatively cool (c. 16–20°C), one of its main uses was for cooling workshops and factories. As the groundwater was not re-injected, the continuous abstraction became non-sustainable, the aquifers were over-abstracted, groundwater heads fell and the result was widespread subsidence of the ground surface, first reported in 1921 and later reaching 100 mm year^{-1} in places. Total subsidence reached over 2.5 m in some areas. Indeed, the city's main historical street, 'The Bund' now lies below the level of the adjacent River Huangpu and the Bund's embankment acts as a flood defence.

By the 1960s, legislation was enacted to severely limit the consumptive abstraction of groundwater from the shallower confined aquifers, in order to brake the

(*Continued*)

BOX 16.2 (*Continued*)

The Bund, Shanghai (photo by David Banks). The River Huangpu is on the other side of the embankment on the right of the photo.

rate of subsidence. Thenceforward, it was only permitted to use the deeper aquifers for groundwater abstraction. The factories and workshops that had previously used shallow groundwater for cooling would have been forced to make use of river water or compressor-driven chillers, as the deeper aquifer horizons, from which abstraction was still permitted, were originally too warm (probably 21–26°C) to realistically use for many direct cooling purposes. However, the Shanghai authorities also initiated the artificial recharge of surplus surface water in winter to the confined aquifer horizons, to replenish their water resources and to try and reverse the subsidence trend. The artificial recharge of cool surface water in winter is also claimed to have reduced the groundwater temperatures in the deeper aquifers to levels suitable for summer cooling. This is possibly the first example of 'aquifer thermal energy storage' (Shi and Bao, 1984; Volker and Henry, 1988; Wei, 2006).

groundwater will simply start percolating out of the ground surface around the well (groundwater flooding). Even if the aquifer is confined, groundwater still may find its way up to the surface through the annulus around the borehole casing, if it has been inadequately sealed with grout. Also, if injecting groundwater into a shallow confined aquifer at very high pressure, there is a risk that ground heave may result, due to the excess pressure in the aquifer lifting the 'lid' (the confining layer) off the aquifer.

We have seen in Sections 8.5–8.7 that there are plenty of equations that allow us to predict the drawdown and upconing effects around wells. In the special case where we have a well doublet (an abstraction well and re-injection well in relatively close proximity), any groundwater that we remove is replenished to the aquifer. There is thus no widespread change in groundwater levels, merely local upconing and drawdown around the wells, and a steady-state situation is established relatively rapidly. Equations 8.24 and 8.25 describe this situation, while Figure 16.2 shows a typical pattern of drawdown and upconing within a well doublet.

16.3.1 Artesian aquifers

Some aquifers (typically confined, but also unconfined) can contain groundwater heads that are above ground level. Thus, when a borehole is constructed into them, it overflows at the ground surface. When drilling a water well, a hydrogeologist will usually have been involved, who can identify this potential hazard scenario. When drilling a closed-loop borehole, there may (in some countries) have been no need to obtain a permit or to involve a hydrogeologist. There is thus a significant risk that an unwary driller will strike an artesian aquifer – the risks are greatest in lower lying areas and valley bottoms. The groundwater flow from an artesian borehole will depend both on the artesian head ('overpressure') and the transmissivity of the aquifer. Once started, a large artesian flow can be remarkably difficult to stop, especially if no casing has been installed, onto which a flange plate can be bolted.

Methods of mitigating artesian risk differ according to the drilling method employed (Figure 16.3). In rotary drilling, a heavy drilling mud may be able to counteract the artesian head. In other methods, a string of casing should be securely grouted into any overlying aquitard before the suspected artesian aquifer is penetrated (the casing can then be closed with a flange plate or packer, if necessary – Figure 16.4). In both cases, some hydrogeological homework is required in order to evaluate the likely depth and head in the artesian aquifer.

If planning to install a closed-loop heat exchanger into an artesian borehole, the best advice is – don't even think about it, unless the flows and heads are very small. The likelihood of being able to install adequate low-permeability grout to satisfactorily seal the hole is low.

The consequences of uncontrolled artesian overflow are the following:

- prosecution – you may be breaching all sorts of water resources legislation;
- flooding (Figure 16.4);
- pollution, if the groundwater enters a surface water and has a very different hydrochemistry (e.g. highly reducing or saline);
- acute embarrassment, as was the case when a driller, installing a geothermal scheme for the Hesse state Ministry of Finance in Wiesbaden, Germany, in November 2009, struck an artesian aquifer and flooded the central portion of the city. In this case, the driller had little excuse – Wiesbaden is famed for its wells of spa water. Oops.

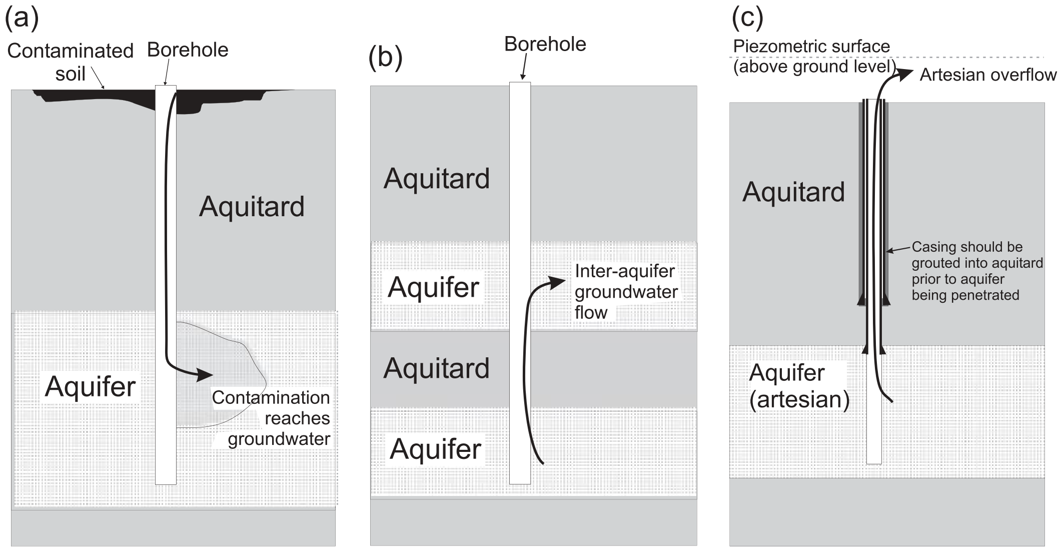

Figure 16.3 Three hydrogeological scenarios for borehole drilling that make environmental regulators nervous: (a) drilling in contaminated land can provide a conduit for contaminants to enter an aquifer; (b) drilling through two aquifers can provide a hydraulic connection and allow leakage of water from one aquifer to another (unless the borehole is either cased through the upper aquifer, or securely grouted/sealed in the section in the intervening aquitard); (c) inadvertent penetration of an artesian aquifer. Artesian pressures may be able to be controlled by a drilling mud of sufficient density. Alternatively, if casing has been grouted into the overlying aquitard prior to penetration, any artesian flow can be controlled.

Figure 16.4 An artesian borehole at High Wycombe, UK. Even when steel casing has been securely installed in an artesian borehole, stemming the flow can be a nightmare! *Photo by David Banks.*

16.3.2 Multiple aquifers

Less obvious, but similar in nature, is the risk posed when a borehole connects two or more different aquifers in a multi-level aquifer system (Figure 16.3). The groundwater heads in the aquifers may be different, with the result that groundwater flows from the higher-head aquifer to the other one. This may not appear to be a major problem – unless one of the aquifers is contaminated, or has a greater value as a water resource or if the two aquifers have different (incompatible) hydrochemistries. Most regulators do not wish to see different discrete aquifers hydraulically connected in any way. To do so could risk being in breach of water resources law. So, if faced with this situation, talk to the environmental regulator. If they permit you to go ahead, they will normally insist that you seal the borehole with a robust, low-permeability grout (or install blank casing through the upper aquifer) to ensure that hydraulic interconnection is precluded.

16.4 Geotechnical risks

Section 16.3 has introduced us to the fact that abstracting or re-injecting groundwater in an open-loop GSHC scheme can cause ground movement, purely due to changes in groundwater level or pressure. There are, however, other mechanisms that can cause movement of ground around operational systems.

16.4.1 Sand pumping

A well-designed water well should abstract water, but leave sediment behind in the aquifer. Sometimes, however, if the well screen (well filter) is incorrectly dimensioned, if the aquifer is a fine-grained uniform sand or if the well is being pumped too hard, we may end up abstracting a small amount of sediment along with the water flow. This is undesirable because it may abrade pump impellers, heat exchangers and valves and may also end up clogging the well screen or aquifer around any re-injection well. In worse cases, we may end up pumping so much sand that we start to physically remove the ground from beneath our very feet. Pumping $10\,L\,s^{-1}$ of groundwater containing $1\,mg\,L^{-1}$ sand results in the removal of almost 1 kg of sediment per day and over 310 kg per year. Houses have been known to disappear into hollowed out ground adjacent to sand-pumping wells.

16.4.2 Frost heave

A closed-loop GSHP system may run with the ground loop at temperatures below 0°C for short periods. A system where the carrier fluid is below 0°C for prolonged periods is unlikely to result in a favourable heat pump efficiency and is unlikely to deliver the cost and carbon savings that we ideally require (there are exceptions to this statement). Furthermore, prolonged running at sub-zero temperatures can result in ice forming in the pore spaces of the rock or sediment. In loose sediments (and especially in horizontal ground loops), the ice expands as it freezes and results in frost heave of the ground surface (Penner, 1962). This may not be a problem beneath pasture land; it may cause unsightly misalignment of paving stones beneath a path or driveway, but may have serious consequences in the vicinity of foundations or load-bearing structures.

It should also be mentioned that repeated freezing and thawing of some loose sediments and some slurry-type grouts can cause 'sorting' of the sediment and migration of fine particles, resulting in a collapse of the sediment or slurry structure.

Finally, while on the subject of freezing, ground loops or header pipes that may run at sub-zero temperatures should be kept well away from any structures susceptible to freezing – or they should be insulated. VDI (2001a) recommended a safe distance of 70 cm from service pipes and drains. IGSHPA (2007) suggested that GSHC pipes passing within 1.5 m of a wall, structure or water pipe should be insulated.

16.4.3 Vapour migration

For closed-loop systems operating in cooling mode (especially shallow horizontal systems), one should be aware of the potential that exists for vapour migration and progressive drying (and even shrinkage) of soils (Section 11.4.3). Significant heating of the ground can also lead to consolidation and creep settlement in clayey, unconsolidated soils (Gabrielsson *et al.*, 2000).

16.4.4 Thermal expansion

Like most materials, rocks expand when heated. Such thermal expansion is only likely to become a significant concern in large cooling-dominated schemes, where heat is being rejected at a high temperature [such as in the borehole thermal energy storage (BTES) systems described in Section 14.5 and Box 14.1]. According to Skarphagen (2006), a terrain 'heave' of 12–17 mm was measured in the early 1980s above a BTES array in Luleå, Sweden (Section 14.5.1), that was operating at a temperature around, or in excess of, 60°C. Linear thermal expansion coefficients of rocks measured in 'normal' temperature ranges include: 3 to $7 \times 10^{-6}\,K^{-1}$ (granite and sandstone; Park *et al.*, 2004) and 5.2 to $7.4 \times 10^{-6}\,K^{-1}$ (mean values for granites; Janio de Castro Lima and Braga Paraguassú, 2004).

16.4.5 Evaporites

Evaporites are rather soluble minerals that were laid down in the geological past in arid environments. They may have been deposited in *sabkhas*, in evaporating saline lakes or hypersaline seas. They include minerals such as halite ($NaCl$), sylvite (KCl), gypsum ($CaSO_4 \cdot 2H_2O$) and anhydrite ($CaSO_4$). They have only been preserved in the rocks beneath our feet today because (typically) they are sandwiched between other low-permeability rocks that keep them from coming into contact with groundwater and dissolving away.

If we drill a well or closed-loop borehole into such rocks, we run the risk of either dissolving the evaporite minerals with our drilling fluid, or creating a conduit along the borehole through which groundwater can migrate and dissolve the evaporite minerals. Result? – Subsidence.

Alternatively, groundwater can migrate along improperly sealed closed-loop boreholes and come into contact with a mineral such as anhydrite. When this occurs, the mineral may hydrate to form gypsum (see above) and expand. The result in this case is ground heave. Exactly this scenario is alleged to have occurred in the German town of Staufen in Baden-Württemberg, causing significant structural damage to several historical buildings and bringing the ground source heat industry into disrepute (Goldscheider and Bechtel, 2009).

Fortunately, substantial evaporite deposits are not so common, at least in temperate and northern Europe, and the formations in which they occur are fairly well known. So, do your geological homework before drilling and avoid these scenarios.

16.5 Contamination risks

16.5.1 Drilling in contaminated land

Another risk of major concern to environmental regulators is that potentially resulting from drilling deep closed-loop boreholes in connection with the redevelopment of 'brownfield'[1] sites. Hopefully, as part of the redevelopment, ground contamination will have been identified and remediated. However, a borehole represents a potential conduit for contaminated surface water or shallow groundwater to migrate from the surface down into the ground, where deep aquifers may be present (Figure 16.3). The main function of a low-permeability grout backfill is to seal the borehole around the heat exchange pipe and prevent this from happening.

Thus, when drilling at a site where contaminated soil or groundwater may be present (and remember that contamination may be found at other locations than just obvious 'brownfield' sites),

- evaluate the risk of contamination;
- understand the hydrogeology;

[1] A 'brownfield' site is a former industrial (or maybe, mining) site, typically in an urban environment, that is being redeveloped. There is a risk of contaminated soil or water at such sites.

- liaise with planning and environmental authorities to develop 'safe' drilling methods and means of disposal of potentially contaminated drilling arisings;
- agree to a grouting programme to ensure that closed-loop boreholes present no risk of contaminant migration into the deep subsurface.

Note that IGSHPA (2007) recommended using a grout with a hydraulic conductivity of less than 10^{-9} m s^{-1}. The grout should also be non-shrinking and frost resistant.

16.5.2 Leakage of refrigerants

Environmental regulators tend not to favour direct circulation heat pump systems, because most of them involve circulating a refrigerant through the subsurface. This refrigerant is often a fluorinated organic compound and *may* be classified as a hazardous substance. Within the EU, such halogenated hydrocarbons are among the substances that should be prevented from entering groundwater. It can be argued that, as long as the closed loop is intact, there is no leakage and no offence being committed, but it certainly has the possibility of making some environmental regulators nervous. In New York State, Collins *et al.*'s (2002) guidelines recommend that, in direct circulation systems, the refrigerant must be non-toxic and non-hazardous, while the loop must be pressure tested at 500 psi (34.5 bars) and equipped with cathodic protection in soils of pH <5.

New generations of direct circulation systems, using other environmentally benign refrigerants, such as CO_2, are beginning to be marketed, however, which should be far more acceptable.

16.5.3 Leakage of carrier fluids

We have already discussed the different types of carrier fluids used in indirect circulation closed-loop systems in Section 9.8 and Box 9.5. The most commonly used carrier fluids are based on ethylene glycol, propylene glycol or ethanol. The last two of these are relatively non-toxic and biodegrade rapidly in the ground under aerobic conditions. Ethylene glycol is somewhat more toxic, but also biodegrades rapidly (French *et al.*, 2001; Jaesche *et al.*, 2006). Cold and wet soil conditions hamper biodegradation, however, while anaerobic degradation is less well-studied and likely to be slower.

A trend can be observed away from ethylene glycol in the ground source heat industry in some nations, due to its perceived toxicity (despite its attractively low viscosity). The new GSHPA (2011) industry standards in the United Kingdom present toxicity criteria that any carrier fluid must meet. In the United States, Den Braven (1998) noted that, of the 26 states that regulated the type of carrier fluid used in closed-loop systems, the overwhelming preference was for water, potassium acetate solution and propylene glycol solution, with ethylene glycol being permitted in only 1 of the 26 states.

In any case, whether your country regulates carrier fluid type or not, you should probably be cautious about using ethylene glycol (or other potentially toxic) carrier fluids in certain sensitive situations, such as

- in loops installed in natural surface water bodies;
- in ground loops installed in the immediate vicinity of ecologically sensitive receptors;
- in the immediate vicinity, or within the source protection zone, of wells used for potable water supply (note that some countries may restrict drilling activities or prohibit other types of carrier fluid within source protection zones).

It is worth noting that, in the Swiss cantons, closed-loop boreholes are potentially regulated under both water and environmental protection legislation. This is related to the fear that closed-loop boreholes can lead both to groundwater contamination and to hydraulic effects (aquifer interconnection). Of particular importance is the prohibition of borehole-based closed-loop systems within source protection zones around potable groundwater abstraction wells and in certain high-value aquifer units. Indeed, Rybach (2003b) reported that maps of 'exclusion zones' for ground source heat boreholes have been published by several cantons.

One can, of course, argue that the risk of carrier fluid leakage is overemphasised. If a closed-loop geoexchange system is properly constructed, a small leak in a ground loop should only release a very modest quantity of carrier fluid (limited by the over-pressure in the loop and the capacity of the pressure expansion vessel), and a drop in loop pressure should trigger a shutdown of the heat pump/circulation pumps. It would usually take a fairly major loop failure (or the persistent re-filling and re-starting of a leaking ground loop) to result in a large leakage of carrier fluid. The best defence again ground-loop failure is, of course, careful construction (Section 7.4.5) and testing (Section 9.10). A low-permeability grout backfill may provide some extra protection of groundwater in case of a leakage, but it should not be relied upon for this purpose.

16.6 Geochemical risks

We should remember that elevated temperatures reduce the solubility of dissolved gases. Thus, increasing temperatures (e.g. in a groundwater-based open-loop cooling scheme) and poor management of fluid pressures during recharge, can result in exsolution of gas bubbles from groundwater in the re-injection well of an open-loop well doublet, or within the aquifer surrounding it. If these bubbles enter and lodge in aquifer pore spaces, they may reduce the aquifer permeability around the recharge well. This is, however, essentially a local problem and a self-limiting one, as the recharge well will suffer decreased capacity to accept water (see Section 8.9.4).

We should also remember that the subsurface is an environment in which groundwater reacts with minerals, and that these reactions may be temperature-dependent.

For example, it is widely speculated that karst (cave) development in several of Britain's limestone aquifers may have been particularly intense in periglacial conditions towards the end of the last ice age, partly due to large quantities of available meltwater, but also partly because the solubility of carbon dioxide (the 'acid' potentially dissolving the limestone) increases as temperature decreases:

$$CaCO_3 + CO_2 + H_2O = Ca^{++} + 2HCO_3^- \tag{16.1}$$

Younger (2006a) has speculated that ground source heat schemes resulting in the cooling of a limestone aquifer may conceivably lead to increased rates of limestone dissolution. We must also wonder what impacts might be expected from ground source cooling schemes where warm water is rejected back to the aquifer. Use of geochemical models suggests that, within the range of normal temperatures encountered in GSHC systems, such issues are unlikely to be significant (see, e.g., Banks *et al*., 2009b) and that issues involving dissolved gases (e.g. contact with oxygen, which can result in precipitation of iron and manganese minerals, or degassing of CO_2) are far more important for controlling mineral precipitation and dissolution reactions. In GSHC schemes operating at more extreme temperatures (e.g. BTES systems), a more detailed consideration of geochemistry is recommended.

16.7 Microbiological risks

I am often asked the question – will rejection of waste heat into the ground or into an aquifer result in a bloom of bacteria or other micro-organisms? I find it difficult to answer, as I am not a microbiologist. When I ask some microbiologists, the response is often muted – they are not used to being asked such straightforward questions! My musings and queries on this matter have elicited the following observations from myself and others:

- If the questioner is thinking of pathogenic micro-organisms – my response is that such organisms should not be in the groundwater in the first place. Find the source of the contamination, rather than worrying about a ground source cooling scheme.
- If the questioner is thinking about the indigenous, non-pathogenic microbes that are known to inhabit even quite deep groundwater environments, the question becomes more intriguing. Some of these bacteria are known to be biofilm-forming and responsible for clogging tunnel drainage systems and water wells. However, in practical terms, even if a temperature increase does result in increased viability of such bacteria, the consequences are likely to be self-limiting. The first place to be affected by 'biofouling' will be the recharge well of the ground source cooling scheme and the aquifer around it, resulting in clogging and reduced capacity (Cullimore and McCann, 1977).

- There seems to be a general agreement among microbiologists that modest increases in temperature tend to increase microbiological activity – all other factors being equal.
- However, all microbes have optimum ranges of temperature tolerance – thus, excessive temperatures may result in a decrease in viability of some species (after all, we pasteurise milk to sterilise it and can, indeed, pasteurise wells to sterilise them, too).
- However, temperature is unlikely to be the determining factor in limiting microbial biomass – microbes need nutrients such as nitrogen, phosphorus and sulphur; they may need terminal electron acceptors (e.g. O_2, NO_3^-, $SO_4^=$) for respiration and carbon compounds for fermentation or cell growth. If these are limited (and they often are in the groundwater environment), temperature increases alone are unlikely to result in an explosion of biomass.
- Changes in ground and groundwater temperature are likely to result in alterations in the balance of species that compose the microbiota. Species that favour the new temperature conditions will out-compete the old ones.

While it seems that some research on this topic could be beneficial, my (poorly informed) gut feeling is that microbiological consequences are unlikely to represent a serious environmental risk from GSHC schemes.

16.7.1 Macrobiota in aquifers

It is becoming increasingly recognised that there also exist macro-organisms (e.g. crustaceans) that actually dwell within the fractures and fissures of limestone aquifers (Waters and Banks, 1997). They are becoming regarded as an ecosystem that itself may deserve protection. Could such species be impacted by temperature changes due to GSHC? Potentially.

16.8 Excavation and drilling risks

Closed-loop ground source heat schemes are, in many countries, poorly regulated. The larger of such schemes will often form part of an application for planning permission and may thus be subject to some form of control by environmental regulators. Smaller GSHC schemes may, however, go beneath the regulators' 'radar'. Regulators fear that installers of such schemes may not necessarily be fully aware of the various 'searches' that need to be made before commencing drilling a borehole, or even excavating a trench (Box 16.3). Before drilling or excavation commences, a search or risk assessment should be carried out to ascertain if any of the following are likely to be encountered:

- Gas mains, oil pipes or strategic fuel pipelines.
- Electricity cables.

BOX 16.3 Before Drilling a Borehole

There are various precautions that should be undertaken before drilling a borehole. Some of these may be prescribed by national legislation; others are simply good practice and common sense. If you get it wrong and drill through an artesian aquifer, an underground railway or a government nuclear bunker, you can expect a call from an angry environmental regulator/insurance assessor/secret service agent (delete as appropriate).

The following checklist is not exhaustive and the possible risks associated with drilling are described in more detail by Misstear *et al*. (2006):

- Obtain permission from the landowner/site occupier.
- Check that the rig has access to the site (e.g. bearing capacity of the access road).
- Check for overhead power lines or other services that will restrict the operation of a drilling rig or excavator.
- Check for the presence of underground services or other structures (see Section 16.8).
- Assess any geotechnical risks from drilling close to buildings (embankments, slopes, vibration).
- Is the site likely to be contaminated (if yes, develop a specific drilling plan in liaison with local and environmental authorities).
- Is there an issue of noise or vibration? What working hours will be permitted?
- Develop a health and safety plan (see Misstear *et al*., 2006).
- Do you need a permit from the environmental regulator, a water authority or geological survey to commence drilling? Even if you do not need permission to drill, it is often a very good idea to contact the environmental authority informally to let them know your plans and to obtain advice.

- Water pipes, sewers, drains or culverted watercourses; district heating or cooling pipes.
- Telephone, Internet or fibre-optic cables (the last being fiendishly expensive to repair).
- Service tunnels or transport tunnels (e.g. underground railways). The Swedish thermogeologist, Göran Hellström, may be persuaded to tell you the story of the time when a driller drilled straight into the Stockholm Metro (no one was injured, but it could have been a very serious incident).
- Mine workings (beware of contaminated mine water or methane gas). In the United Kingdom, for example, if you are drilling through coal seams or mined strata, you will need permission from the Coal Authority.

- Caves, especially those used for recreational purposes.
- Archaeological remains.
- Underground rooms or bunkers.

Contact the relevant utilities companies to obtain maps of buried services and to obtain permission to excavate/drill if necessary. You should also ask yourself the following questions:

- Are there any overhead power lines at the site? What is the minimum safe distance to erect a drilling mast from such lines? Is the rig earthed?
- Are we drilling or excavating in the vicinity of another water well or ground source heat scheme? Are we planning to drill within a designated source protection zone? Or in the vicinity of any nature reserve or protected habitat?
- Is there any suspicion that we are drilling into contaminated soils or groundwater?
- Do we know what geology or hydrogeology to expect? Is there any risk of artesian or multi-aquifer conditions? Do we have a contingency plan if artesian water is encountered?
- How will we dispose of drilling cuttings (especially if contaminated)?

On completion of drilling, send a drilling log and site location plan to the national repository of geological information (often, the Geological Survey). This may be required by law. In any case, it is good practice: if all drillers report data diligently, an excellent database can be acquired that is of benefit to all.

Even in the case of horizontal 'trenched' closed-loop systems, many of the above points apply, especially those regarding contaminated land and buried services.

16.8.1 Gas risks and confined spaces

A borehole or well provides a conduit through which gases can escape from the ground. If the head of the borehole is within or beneath a building, there is a risk that gases could accumulate within that space. While a low-permeability grout seal in a closed-loop borehole significantly reduces the risk, even a grouted borehole is likely to represent a preferential pathway for gas migration. The risks are greatest at times of low atmospheric pressure. The gases of most concern are as follows:

- Methane (CH_4), which is often associated with coal mines and with landfills, but which can occur in other organic-rich rocks or sediments. In certain proportions in air, it can be explosive.
- Carbon dioxide (CO_2), which can also be associated with mines and landfills. It can also degas from wells and boreholes in many other types of rock, especially limestones and Chalk, and especially at times of low atmospheric pressure. It can accumulate, from the base upwards, in rooms or chambers and can cause asphyxiation. It is probably the most dangerous of all the gas risks discussed here.

- Vapours and gases from contaminated land or landfills (e.g. solvents and other volatile organic compounds).
- Radon, a radioactive gas that can be emitted by certain types of ground that are rich in uranium and radium, including dark organic shales, granites and some limestones. Prolonged exposure can lead to increased risk of lung cancer.

Assess the risks and take remedial action (e.g. ventilation of the space containing the borehole or well heads) if necessary.

16.9 Decommissioning of boreholes

A drilled hole in the ground represents a potential pathway for contaminants from the surface to deep groundwater. Thus, when a borehole is redundant and no longer actively managed, it should be responsibly decommissioned, not just abandoned. If the borehole or well is open, it is probably worth enquiring whether the regional agency responsible for groundwater management can convert it to an 'observation well' for monitoring groundwater quality and levels. If not, we may be forced to backfill the well. A concise guide to well decommissioning is provided by SEPA (2004), which broadly coincides with advice given by NGWA (McCray, 1998, 1999). The overriding objectives of decommissioning are (1) to remove any hazard to the public from an open void in the ground; (2) to remove any carrier fluid from a ground loop; (3) to remove any headworks and pump; (4) to prevent any pollution from entering the subsurface; (5) to prevent the borehole acting as a conduit for groundwater flow, or for gases such as methane or carbon dioxide to the surface; and (6) to return the ground to a condition near to its natural state.

In order to achieve these objectives, we should restrict access to the well site until the well is safely decommissioned, and then remove the pump, rising main and any headworks. Then, having obtained a copy of the original drilling log and construction records, we should commission a specialist contractor to backfill the well. The well can be backfilled along its length by non-polluting, low-permeability, non-shrinking grout or bentonite. Alternatively, if the geology is known, the borehole can be backfilled with clean permeable backfill, sand or gravel adjacent to aquifer horizons, but with low-permeability bentonite or grout adjacent to aquitard horizons. This preserves the integrity of the hydrostratigraphic structure of the aquifer and prevents the well from acting as a conduit for groundwater flow between aquifer horizons. The uppermost section of the well (>2 m) should always be backfilled with a low-permeability concrete, cement or bentonite grout medium. At the surface, a concrete cap should extend >0.5 m beyond the well's perimeter (Figure 16.5).

In the case of a closed-loop borehole, the situation is a little trickier. If the ground loop is simply installed in an open, water-filled borehole, the loop can be removed and the borehole backfilled as described above. If the loop has been grouted into the

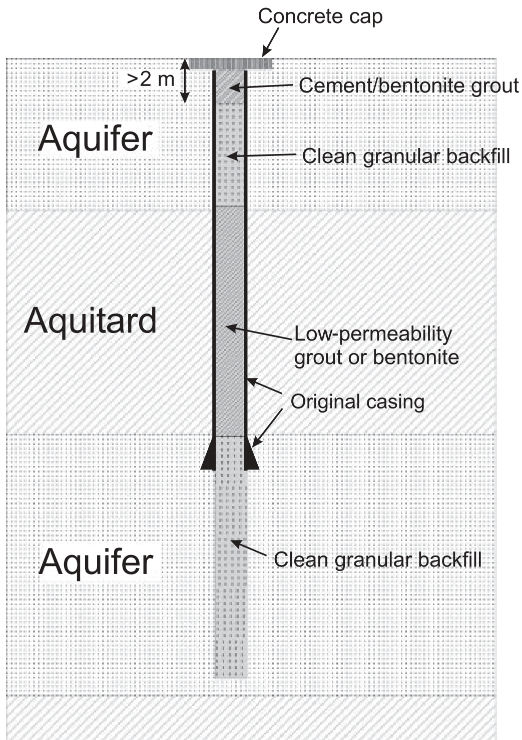

Figure 16.5 One method of backfilling a decommissioned borehole. The backfill type matches the hydrostratigraphy (high-permeability backfill in the aquifer sections, low-permeability grout in the aquitard sections). Alternatively, the entire borehole could have been backfilled with low-permeability, non-shrinking grout.

borehole, however, the only solution is to remove any carrier fluid from the loop and then to grout the U-tube with a low-viscosity grout mix, injected under pressure.

After decommissioning, notification of backfilling should be sent to the national or regional custodian of geological records (often, the Geological Survey).

16.10 Promoting technology: subsidy

In the foregoing sections, we have considered what concerns might arise among regulators over the widespread use of ground-sourced heating and cooling (see also Younger, 2006a). We would hope, of course, that the legislation imposed upon the industry is 'measured' and addresses genuine, rather than imaginary, concerns. The current author would argue that the potential benefits in terms of savings in CO_2 emissions from space heating far outweigh concerns over possible groundwater pollution from a few litres of ethylene glycol solution.

In addition to the regulatory 'stick', many nations offer subsidies as a form of 'carrot' to encourage the uptake of renewable energy technologies, such as GSHPs. While we would naturally hope that a technology such as GSHC can eventually thrive on its own merits, one can argue in favour of subsidy in circumstances where the technology

- has a high initial capital cost and where payback times may be large;
- has not yet fully gained consumer confidence, either due to unfamiliarity with the technology or due to conceptual obstacles.

Many European nations have offered subsidies to GSHP schemes. The subsidies can be of several forms and, in fact, the three main types have been attempted in recent years in the United Kingdom:

- Flat rate initial grants: the United Kingdom's recent Low Carbon Buildings Programme attempted to provide flat rate grants to individuals, communities and businesses to install renewable energy technology. Until recently, house owners could claim up to £1200 or 30% of eligible costs towards a GSHP scheme (LCBP, 2012).
- Tax breaks: the so-called Enhanced Capital Allowances scheme (ECA, 2010) allowed UK businesses to claim 100% of the cost of renewable energy infrastructure (again, including GSHPs) against tax during the first year.
- Payments related to energy production (feed-in tariffs): in the United Kingdom, the two forms of subsidy discussed above are due to be replaced by a Renewable Heat Incentive (DECC, 2011), which pays the user a given sum per kilowatt-hour of renewable heat produced, over a long period. Such a scheme provides the owner with an incentive to use renewable heating sustainably and also attaches some monetary value (revenue) to the installed scheme when the property is sold to a new owner. At the time of writing, the tariff is set at £0.03 (large schemes) and £0.043 per kilowatt-hour (smaller schemes) over a period of 20 years.

Usually, in order to take advantage of such subsidies and tax breaks, it is necessary to utilise either an officially approved heat pump and/or an accredited installer. This should provide authorities with a means of ensuring that heat pumps meet certain standards of reliability and efficiency, and that installers operate within a certain code of professional competence. It may also be necessary to demonstrate that measures have already been taken to achieve a reasonable standard of building thermal performance (insulation, heating control and other energy efficiency measures).

16.11 The final word

Many would argue that subsidy plays an important role in the first few years of growth of a market for a socially and environmentally desirable technology. However, I have tried to argue, during the course of this book, that GSHC not only has significant environmental advantages, but can also compete economically with conventional fossil fuels in many cases. It is a technology that can be attractive to both environmentalists and petrol-heads. We should thus hope that, at some point in the not-too-distant future, the technology will have gained sufficient consumer confidence to stand alone as an unsubsidised product.

Having said this, I have become aware, since the publication of the first edition of this book, that GSHC has significant obstacles to overcome:

- GSHC technology and design is very complex. I have to admit that a technology, whose successful design requires such a wide-ranging understanding of heating, ventilation and air conditioning (HVAC), thermodynamics, geology and hydraulics, is likely to be beyond the scope of many would-be renewable-energy installers and also beyond the understanding of many customers. I can fully understand why many customers and installers regard air source heat pumps and solar thermal panels as far more attractive products.
- The economic and environmental benefits associated with GSHPs are intimately bound up with the cost and carbon content of the electricity that is used to power them. In some west European nations, including the United Kingdom, there is little sign that electricity is being effectively decarbonised (indeed, there is evidence to suggest that the reverse is the case). This erodes away at the environmental arguments for heat pumps.
- As a consequence of the above, in some nations, it is becoming absolutely paramount to design highly efficient systems to reap the environmental and marginal cost benefits that GSHPs can offer. Recent guidelines published in the United Kingdom (GSHPA, 2011; MIS, 2011a,b) are aimed at exactly this, and are steadily forcing up design carrier fluid temperatures and thus the quantities of heat exchanger that need to be installed in the ground. This tends to force up the capital costs of the systems, potentially making them less attractive in terms of the initial investment required.

A phrase often heard in the media is 'fuel poverty' – the fact that large numbers of people (especially the elderly) are unable to heat themselves adequately in the winter. I would argue that fuel poverty ought to be a myth – particularly when we bear in mind that, in most European city centres, there exist offices, businesses, underground railways (Ampofo *et al*., 2004a) and hotels that are willing to pay large sums to 'throw away' waste heat. I would thus argue that we have *plenty* of heat – it is just in the wrong place at the wrong time. Thus, I have become increasingly convinced that the way forward for our cities of the twenty-first century is

- in investing in district heating and cooling systems, which can solve the problem of heat being in the wrong place. Such systems have the potential to connect a number of centralised or decentralised sources of heat and 'coolth' (of which geothermal/GSHC may be but one). In other words, GSHC should be integrated with other technologies and not be in competition with them. Could we use existing conduits (water mains, sewers, transport tunnels) as means of transporting heat from one node to another?

- in developing various forms of thermal energy storage, to efficiently store heat or coolth from times of surplus to times of demand. Several different forms of large-scale thermal storage are available, but underground thermal energy storage (BTES or ATES – see Chapter 14) must rank among the most important. Such thermal stores represent the 'accumulators' or 'capacitors' within district heating and cooling networks.

The possibilities that the science of thermogeology and the technology of GSHC offer are manifold, and there is no shortage of research tasks. The main challenge is to realise that these tasks can only be tackled meaningfully in a multidisciplinary manner, with seamless collaboration between the geologist, the architect, the engineer and the regulator. Those involved in marketing the GSHP should do so with integrity, realising that it represents a powerful tool in humankind's array of energy technologies, but acknowledging that it is not appropriate for all customers, that it poses significant challenges in terms of skills integration and that it requires a significant investment of capital 'up front' in the expectation of long-term environmental and monetary benefits. As Heap (1979) wisely observed, we should protect against 'the twin dangers of excessive enthusiasm for apparent novelty and of undue scepticism for a concept for which worthwhile applications have been slow in developing'.

STUDY QUESTIONS

16.1 A closed-loop ground source cooling scheme, rejecting 15 kW of heat to the ground, is constructed 100 m from a water supply well, pumping clean, potable groundwater at a rate of 25 L s^{-1} and at a constant temperature of 10.6°C. What is the maximum change in the pumped groundwater temperature that one might expect as a result of the new ground source cooling scheme?

16.2 One day, a single borehole in the closed-loop scheme suffers a defect and slowly releases 40 L of 25% v/v monoethylene glycol (equivalent to 10 L of pure glycol) to the aquifer during the course of 24 h. What is the maximum concentration of glycol one could expect in the pumped groundwater (if the well was not shut down)? Assume that the density of neat glycol is 1.113 kg L^{-1}.

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Study Question Answers

1.1 The groundwater head gradient is 8 m/1000 m = 0.008. The cross-sectional area of the aquifer in the direction of flow is 30 m × 1000 m = 30 000 m^2. Thus, by Darcy's law,

$$\text{Flow } Z = 30\,000 \text{ m}^2 \times 0.008 \times 3 \times 10^{-4} \text{ m s}^{-1} = 0.072 \text{ m}^3 \text{ s}^{-1} = 2\,270\,000 \text{ m}^3 \text{ a}^{-1}$$

1.2 The radius of the core is 15 mm = 0.015 m. The cross-sectional area of the core is given by $\pi r^2 = 7.07 \times 10^{-4}\,\text{m}^2$. The temperature gradient is given by (28°C – 22°C)/0.055 m = 109 K m^{-1}. By Fourier's law, the heat flow is

$$109 \text{ K m}^{-1} \times 3.1 \text{ W m}^{-1} \text{ K}^{-1} \times 7.07 \times 10^{-4} \text{ m}^2 = 0.24 \text{ W}$$

1.3 *Heat, flowing from a high-temperature source to a low-temperature exhaust, is able to drive a heat engine, which can perform mechanical work.*

We can use mechanical energy (work) to power a heat pump, which can 'lift' heat from the cool ground to a warmer space-heating environment.

2.1 The conventional units (m^2 day^{-1} of transmissivity and metres head of water) assume that the fluid in question is fresh water, with a 'normal' viscosity and density. When we are considering geothermal fluids, the temperature and density will be anything but normal (125 g L^{-1} salinity and 76°C). We thus use intrinsic units of permeability/transmissivity and we use SI units of pressure (MN m^{-2}) rather than metre head of water.

Given that 1 D is approximately equal to 0.84 m day^{-1}, we can say that 1 D m = 0.84 m^2 day^{-1}, when we are considering fresh water.

Given that force is equal to mass × acceleration, we can say that a cube of fresh water 1 m high and with a base of 1 m^2 exerts a pressure of 1000 kg × 9.81 m s^{-2} per m^2 = 9810 N m^{-2} = 9.81 kN m^{-2} = 9.81 kPa. Thus, 1 MN m^{-2} is equivalent to 102 m head of fresh water.

If the water (as at Southampton) has a salinity of 125 g L^{-1}, the density of the water is now around 1125 kg m^{-3} (not accounting for density changes with temperature). Thus, 1 m of brine exerts a pressure of 11.04 kN m^{-2}. Thus, 1 MN m^{-2} is equivalent to 91 m head of Southampton brine.

2.2 At Soultz, we are considering temperatures in excess of 100°C. Thus, fluid density may change significantly with temperature and, if steam forms, the fluid density

An Introduction to Thermogeology: Ground Source Heating and Cooling, Second Edition. David Banks.

drops dramatically. Thus, we can no longer talk about $L\,s^{-1}$, especially in systems where water and steam may exist, and we must rely on the principle of mass conservation to talk about $kg\,s^{-1}$ of H_2O (irrespective of fluid phase or density).

2.3 Box 2.1: Larderello, Nesjavellir. Box 2.2: Typical open-loop ground-sourced heat pump systems, Arlanda Airport. Box 2.3: Typical closed-loop ground-sourced heat pump systems, Annfield Plain, Dunston Innovation Centre (although at this location, an aquifer was present and could potentially have been used). Box 2.4: Southampton, Eastgate. Box 2.5: Soultz-sous-Forêts, Rosemanowes.

3.1 Using Equation 3.12,

$$\theta_z = \theta_0 + z\frac{d\theta}{dz} = \theta_0 + z\frac{Q}{A\lambda}$$

$z = 150\,m$, $Q/A = 0.064\,W\,m^{-2}$ and $\lambda = 1.8\,W\,m^{-1}\,K^{-1}$. Thus, the geothermal gradient $= 0.064/1.8\,K\,m^{-1} = 0.0356\,K\,m^{-1}$. As $\theta_0 = 9.8°C$, the temperature at 150 m $= 9.8°C + 150\,m \times 0.0356°C\,m^{-1} = 15.1°C$.

3.2 The heat required to drop the temperature of the tank to 0°C = $(10 - 0)°C \times 2.5\,MJ\,m^{-3}\,°C^{-1} \times 20\,m^3 = 500\,MJ$. The quantity of water in the sand is $0.18 \times 20\,m^3 = 3.6\,m^3$. The latent heat of fusion of water is $335\,kJ\,kg^{-1}$ = c. $335\,MJ\,m^{-3}$ (Box 3.3). Thus, the heat required to convert $3.6\,m^3$ of water at 0°C to ice at 0°C $= 3.6\,m^3 \times 335\,MJ\,m^{-3} = 1206\,MJ$. The total refrigeration requirement is thus 500 MJ + 1206 MJ = 1706 MJ. The rate of heat extraction required is $1.706 \times 10^9\,J/(14 \times 24 \times 60 \times 60)\,s = 1410\,W = 1.41\,kW$.

4.1 The electricity used by the heat pump is given by 12 kW/3.3 = 3.64 kW. Thus, the remainder of the heat is extracted from the ground (G):

$$G = H\left(1 - \frac{1}{\mathrm{COP}_H}\right) = 12\ \mathrm{kW}\ (1 - 1/3.3) = 8.36\ \mathrm{kW}$$

4.2 Now, the electricity consumption is lower, at 12 kW/3.6 = 3.33 kW. The heat extracted from the ground is thus greater, at 8.66 kW. Thus, paradoxically, although our heat pump has become more efficient, we may find we need to install additional heat exchange surface into the ground to transfer the increased quantity of heat.

4.3 12.3 MWh of heat = 12 300 kWh $year^{-1}$. If we divide this figure by the heat pump's nominal rating of 6 kW, we arrive at a compressor run time of 2050 h per year. One year has 8766 h – thus, the heat pump operates for 0.23 (23 %) of the year. On average, the heat pump has consumed 12.3 MWh/3.3 = 3.73 MWh of electricity during the year, and only the remaining 'ground source' heat (8.57 MWh) qualifies as being renewable or 'green' (see the EU Renewable Energy Directive 2009/28/EC).

6.1 Equation 6.5 states that the heat rejected to the environment (i.e. to the ground G) is given by

$$G = C + E = C\left(1 + \frac{1}{\mathrm{COP}_C}\right)$$

where C is the cooling effect and E is the electrical energy. Thus, the heat rejected to the ground is 21 kW × (1 + 1/4.2) = 21 kW × 1.238 = 26 kW. The compressor would use 21 kW/4.2 = 5 kW of electricity.

6.2 Rearranging Equation 6.6, we get

$$\Delta\theta = \frac{\text{Heat rejected}}{Z\, S_{\text{VCwat}}}$$

Thus, the temperature change is given by

$$100\ \text{kJ s}^{-1}/4.18\ \text{kJ L}^{-1}\ \text{K}^{-1}/5\ \text{L s}^{-1} = 4.8\ \text{K}$$

Thus, the final groundwater temperature would be 16.8°C.

6.3 The curves in Figure 6.9 do not extend to 40°C, so we should really contact the manufacturer for specific data. However, if we do a wild extrapolation, we might expect a rather low COP_C of around 3.

7.1 The heat transferred from the first fluid is given by the product of the temperature change, the heat capacity and the flow rate:

$$3.3\ \text{K} \times 4.18\ \text{kJ L}^{-1}\ \text{K}^{-1} \times 1.2\ \text{L s}^{-1} = 16.55\ \text{kJ s}^{-1} = 16.55\ \text{kW}$$

Thus, the temperature increase of the cold fluid is given by

$$16.55\ \text{kJ s}^{-1}/\left(1.6\ \text{L s}^{-1} \times 3.65\ \text{kJ L}^{-1}\ \text{K}^{-1}\right) = 2.8\ \text{K}$$

7.2 Firstly, we calculate the thermal conductance (Λ) of each layer, by dividing the conductivity by thickness. Then we calculate the thermal resistance (R) as the inverse:

Layer 1: $\Lambda_1 = 1.3\,\text{W m}^{-1}\,\text{K}^1/10\,\text{m} = 0.13\,\text{W m}^{-2}\,\text{K}^{-1}$; $R_1 = 7.69\,\text{K m}^2\,\text{W}^{-1}$
Layer 2: $\Lambda_2 = 2.3\,\text{W m}^{-1}\,\text{K}^1/20\,\text{m} = 0.115\,\text{W m}^{-2}\,\text{K}^{-1}$; $R_2 = 8.70\,\text{K m}^2\,\text{W}^{-1}$
Layer 3: $\Lambda_3 = 1.6\,\text{W m}^{-1}\,\text{K}^1/20\,\text{m} = 0.08\,\text{W m}^{-2}\,\text{K}^{-1}$; $R_3 = 12.5\,\text{K m}^2\,\text{W}^{-1}$

The total thermal resistance of the sequence is obtained from the sum of these = $28.89\,\text{K m}^2\,\text{W}^{-1}$. The thermal conductance of the sequence is the inverse of this: $0.0346\,\text{W m}^{-2}\,\text{K}^{-1}$. The equivalent composite thermal conductivity is obtained by multiplying the conductance by the sequence thickness:

$$0.0346\ \text{W m}^{-2}\ \text{K}^{-1} \times 50\ \text{m} = 1.73\ \text{W m}^{-1}\ \text{K}^{-1}$$

8.1 Let's use the Cooper–Jacob Equation 8.11 setting r equal to the well radius = 0.09 m, $S = 0.002$, $T = 900\,\mathrm{m^2\,day^{-1}}$, $Z = 25\,\mathrm{L\,s^{-1}} = 2160\,\mathrm{m^3\,day^{-1}}$ and $t = 21$ days:

$$s_\mathrm{w} = \frac{2.30Z}{4\pi T}\log_{10}\left(\frac{2.25Tt}{r_\mathrm{w}^2 S}\right) = \frac{2.30\times 2160}{4\pi\times 900}\log_{10}\left(\frac{2.25\times 900\times 21}{0.09^2\times 0.002}\right) = 4.14\ \mathrm{m}$$

Remember that the Cooper–Jacob equation is only valid for $u < 0.01$, so let's check:

$$u = \frac{r^2 S}{4Tt} = \frac{0.09^2\times 0.002}{4\times 900\times 21} = 2\times 10^{-10}\ \text{(that's OK!)}$$

8.2 We can now use Equation 8.25, firstly with a well separation of 88 m, then with 150 m:

$$s_\mathrm{w} = \frac{Z}{2\pi T}\ln\left(\frac{L}{r_\mathrm{w}}\right) = \frac{2160}{2\pi\times 900}\ln\left(\frac{88}{0.09}\right) = 2.63\ \mathrm{m}$$

$$s_\mathrm{w} = \frac{Z}{2\pi T}\ln\left(\frac{L}{r_\mathrm{w}}\right) = \frac{2160}{2\pi\times 900}\ln\left(\frac{150}{0.09}\right) = 2.83\ \mathrm{m}$$

Thus, the existence of the recharge well dramatically decreases the drawdown in the abstraction well. The further the well doublet is spaced apart, the greater the drawdown (and upconing in the re-injection well) becomes.

8.3 The hydraulic conductivity of the aquifer is the transmissivity divided by the aquifer thickness = $900\,\mathrm{m^2\,day^{-1}}/80\,\mathrm{m} = 11.25\,\mathrm{m\,day^{-1}}$. The Darcy velocity v_D of the groundwater (flow per unit cross-sectional area) is obtained from Darcy's law:

$$v_\mathrm{D} = Ki = 11.25\ \mathrm{m\,day^{-1}}\times 0.01 = 0.1125\ \mathrm{m\,day^{-1}}$$

The actual groundwater flow velocity v is obtained by dividing by effective porosity $n_\mathrm{e} = 0.08$:

$$v = v_\mathrm{D}/n_\mathrm{e} = 0.1125\ \mathrm{m\,day^{-1}}/0.08 = 1.41\ \mathrm{m\,day^{-1}}$$

Thus, we might expect the chemical tracer to appear at the observation well after:

$$t_\mathrm{hyd} = 150\ \mathrm{m}/1.41\ \mathrm{m\,day^{-1}} = 107\ \mathrm{days}$$

The thermal retardation factor is given by Equation 8.38:

$$R_\mathrm{the} = \frac{t_\mathrm{the}}{t_\mathrm{hyd}} = \frac{S_\mathrm{VCaq}}{n_\mathrm{e}S_\mathrm{VCwat}} = \frac{2.3}{0.08\times 4.18} = 6.88$$

Thus, we might expect the heat to appear at the observation well after

$$t_\mathrm{hyd} = 107\ \mathrm{days}\times 6.88 = 736\ \mathrm{days}$$

We should recognise, however, that this is only a very gross estimate because

1. we have assumed that the flow is purely governed by the regional groundwater flow velocity (which in reality will be slightly enhanced by the forced injection at the recharge well – see Barker, 2012, for improved equations);
2. we have neglected any three-dimensional heat transfer, which will tend to further retard the heat signal;
3. we have not considered any dispersive effects.

9.1 1. A 32-mm OD single U-tube is often a good choice of heat exchanger for arrays comprising many shallower boreholes.
2. A 40-mm OD double U-tube is often a good choice of heat exchanger for arrays comprising relatively few, but deep boreholes.
3. A 40-mm OD single U-tube is often a good choice of heat exchanger for typical closed-loop boreholes of depth c. 100 m.

But you should *always do a hydraulic calculation to confirm your selection!*

9.2 A 40-mm OD (SDR 11) pipe has a wall thickness of 40 mm/11 = 3.64 mm. Thus, the pipe's internal diameter is 32.7 mm. Its internal radius is thus 0.0164 m. The dynamic viscosity of 25.4% propylene glycol is (from Table 9.3) 0.00564 kg m^{-1} s^{-1} and its density is 1026 kg m^{-3}. We can use Box 9.3 to calculate the Reynolds Number for 15 L min^{-1} (= 0.00025 m^3 s^{-1}) and 25 L min^{-1} (= 0.000417 m^3 s^{-1}):

$$Re = \frac{2\rho F}{\pi r \mu}$$

We obtain values of

$$\begin{aligned} Re &= 2\times 1026 \text{ kg m}^{-3} \times 0.00025 \text{ m}^3 \ s^{-1} / (3.1415 \times 0.0164 \text{ m} \times 0.00564 \text{ kg m}^{-1} \text{ s}^{-1}) \\ &= 1769 \text{ (i.e. laminar flow) for } 15 \text{ L min}^{-1} \end{aligned}$$

and

$$Re = 2949 \text{ (i.e. transient turbulent flow) for } 25 \text{ L min}^{-1}$$

Be very careful to ensure that all your units are compatible: that is kg, m and s.

For a flow rate of 15 L min^{-1}, we can use Equation 9.5 to calculate the pressure loss, and we obtain a figure of

$$\begin{aligned} \Delta P &= 16 \times 1026 \text{ kg m}^{-3} \times (0.00025 \text{ m}^3 \text{ s}^{-1})^2 \times 150 \text{ m} / \left[1769 \times \pi^2 \times (0.0164 \text{ m})^5\right] \\ &= 7512 \text{ Pa} = 7.5 \text{ kPa} \end{aligned}$$

For a flow rate of 25 L min^{-1}, we can use Equation 9.6 to estimate the pressure loss, and we obtain a figure of

$$\begin{aligned} \Delta P &= 0.0791 \times 1026 \text{ kg m}^{-3} \times (0.000417 \text{ m}^3 \text{ s}^{-1})^2 \times 150 \text{ m} / \left[7.369 \times \pi^2 \times (0.0164 \text{ m})^5\right] \\ &= 24\ 767 \text{ Pa} = 24.8 \text{ kPa} \end{aligned}$$

10.1 We can use Equation 10.7a

$$\theta - \theta_0 = \frac{q}{4\pi\lambda} E(u) \text{ where } u = r^2 S_{VC} / (4\lambda t)$$

and we would set $\lambda = 2.2\,\mathrm{W\,m^{-1}\,K^{-1}}$, $S_{VC} = 2.3\,\mathrm{MJ\,m^{-3}\,K^{-1}}$, $q = -15\,\mathrm{W\,m^{-1}}$, $\theta_0 = 12°\mathrm{C}$, $r = 1\,\mathrm{m}$, 10 m and 30 m, $t = 1\,\mathrm{h}$ (3600 s), 1 day (86 400 s), 1 year (31 557 600 s) and 15 years (473 364 000 s). Solving the equation gives us the following ground temperatures:

	$r = 1\,\mathrm{m}$	$r = 10\,\mathrm{m}$	$r = 30\,\mathrm{m}$
$t = 1\,\mathrm{h}$	12°C	12°C	12°C
$t = 1$ day	11.99°C	12°C	12°C
$t = 1$ year	9.71°C	11.84°C	12°C
$t = 15$ years	8.24°C	10.71°C	11.69°C

These results tell us that a typical closed-loop borehole does not substantially affect the temperature of the ground beyond a radius of around 10 m from the borehole. Note that the above calculations assume radial heat flow only; they do not take account of heat flow from the surface or underlying rocks, which would tend to lead to even lesser temperature changes in the longer term.

10.3 If the heat pump COP_H is 3.3, then the heat extracted from the carrier fluid is $6\,\mathrm{kW} \times (1 - 1/3.3) = 4.18\,\mathrm{kW}$. The volumetric heat capacity of the fluid is given by $3.795\,\mathrm{kJ\,K^{-1}\,kg^{-1}} \times 1.052\,\mathrm{kg\,L^{-1}} = 3.992\,\mathrm{kJ\,L^{-1}\,K^{-1}}$. The temperature change across the evaporator is given by $4.18\,\mathrm{kJ\,s^{-1}}/(0.33\,\mathrm{L\,s^{-1}} \times 3.992\,\mathrm{kJ\,L^{-1}\,K^{-1}}) = 3.17\,\mathrm{K}$. If the average carrier fluid temperature is −1°C, then the fluid enters the heat pump at −1°C + 3.17°C/2 = +0.59°C, and leaves at −1°C − 3.17°C/2 = −2.59°C.

11.1 Section 11.6.3 states that, for 2400 equivalent operational hours per year in south-central Sweden, a maximum specific heat extraction rate of $18\,\mathrm{W\,m^{-1}}$ is recommended for single pipes in damp till, without the carrier fluid entry temperature to the heat pump dropping below −5°C. For wet/saturated soils, the rate is $25\,\mathrm{W\,m^{-1}}$. As our soil is sandy (quartz rich) and saturated, we may be tempted to use a figure of, say, $20\,\mathrm{W\,m^{-1}}$. If the COP_H of the heat pump at peak load is 3.5, then the heat extracted is $8\,\mathrm{kW} \times (1 - 1/3.5) = 5.71\,\mathrm{kW}$ (in fact, the COP_H at temperatures of −5°C will probably be much less than that, resulting in less heat being extracted from the soil, so our assumption is conservative). We therefore require $5714\,\mathrm{W}/20\,\mathrm{W\,m^{-1}} = 286\,\mathrm{m}$ of pipe. We might thus propose installing six runs of 48 m length (four runs of 71.5 m is marginally too long for the garden), which has a total area of $48\,\mathrm{m} \times (5 \times 1.2\,\mathrm{m}) = 288\,\mathrm{m^2}$.

11.2 Section 11.6.5 states that, in moist soils, the use of compact slinkies can reduce the required trench length to 34–38% of that for single pipe and in wet soils to

30–36%. Let us assume a ratio of 34%. This means that we would require $0.34 \times 286\,\mathrm{m}$ of slinky trench, or c. 98 m. We could thus install four slinky trenches of 24.5 m length, each containing 196 m of pipe (which should be hydraulically acceptable, depending on the pipe diameter and the carrier fluid). The total pipe required is thus 784 m and the area underlain by the trenches (if 1 m wide and spaced at 3 m) is $24.5\,\mathrm{m} \times (4\,\mathrm{m} + 3 \times 3\,\mathrm{m}) = 318\,\mathrm{m}^2$.

11.3 Section 11.5 states that heat extraction rates should not exceed $70\,\mathrm{kWh\,m^{-2}\,year^{-1}}$ for central Sweden. If a peak rate of 8 kW is delivered for $2400\,\mathrm{h\,year^{-1}}$, this equates to 19 200 kWh. With an SPF_H of 3.5, this means that 13 714 kWh of heat are extracted per year. For the single pipe system in question 11.1, $288\,\mathrm{m}^2$ is required, resulting in an areal extraction rate of $48\,\mathrm{kWh\,m^{-2}\,year^{-1}}$ – well within the specified limit. The same is also true of the slinky system in question 11.2.

12.1 We can use Equation 12.1, setting R_{ex} to $170\,\mathrm{W\,m^{-2}}$, n/D to 0.5 and using $a = 0.18$ and $b = 0.55$. This gives an answer of $q_{sw} = 0.455 \times 170\,\mathrm{W\,m^{-2}} = 77\,\mathrm{W\,m^{-2}}$. For the long-wave radiation, we can use Equation 12.2, setting θ_{sc} to 15°C, yielding a value of $298\,\mathrm{W\,m^{-2}}$. This figure should be corrected for cloud cover, however, by a factor $(1 + 0.0034Cl^2)$, where Cl is 4 oktas. This yields a correction factor of 1.054 and a final figure for $q_{lw} = 314\,\mathrm{W\,m^{-2}}$.

12.2 The back radiation is calculated from Equation 12.4 as

$$q_{\mathrm{back}} = 0.97 \times 5.67 \times 10^{-8}\ \mathrm{W\ m^{-2}\ K^{-4}} \times (286\ \mathrm{K})^4 = 368\ \mathrm{W\ m^{-2}}$$

12.3 The volume of the pool is $599.4\,\mathrm{m}^3$ and its heat capacity is thus $599.4\,\mathrm{m}^3 \times 4.18\,\mathrm{MJ\,m^{-3}\,K^{-1}}$ (the heat capacity of water) $= 2505\,\mathrm{MJ\,K^{-1}}$. Thus, to raise the water by 9 K requires 22 550 MJ heat. If this is supplied over 48 h (172 800 s), the power required is 22 550 000 kJ/172 800 s = 130 kW. This calculation ignores heat gains and heat losses to the atmosphere and ground during the heating period.

14.1 The daily heat demand is $80\,\mathrm{kW} \times 12\,\mathrm{h} = 960\,\mathrm{kWh}$, or $80\,\mathrm{kJ\,s^{-1}} \times 12 \times 3600\,\mathrm{s} = 3456\,\mathrm{MJ}$. The heat capacity of water is around $4.18\,\mathrm{MJ\,m^{-3}\,K^{-1}}$. Thus, one would require $3456\,\mathrm{MJ}/4.18\,\mathrm{MJ\,m^{-3}\,K^{-1}}/20\,\mathrm{K} = 41.3\,\mathrm{m}^3$ of water to store this heat, with a temperature difference of 20 K. This is equivalent to a tank of water of radius 2.1 m and height 3 m. This is a large tank, but is feasible.

The annual heat demand is $80\,\mathrm{kJ\,s^{-1}} \times 1800 \times 3600\,\mathrm{s} = 518\,400\,\mathrm{MJ}$. Thus, one would require $518\,400\,\mathrm{MJ}/4.18\,\mathrm{MJ\,m^{-3}\,K^{-1}}/20\,\mathrm{K} = 6201\,\mathrm{m}^3$ of water to store this heat, with a temperature difference of 20 K. This is equivalent to a tank of water of radius 22.2 m and height 4 m. This is an unrealistically large volume. To store the heat in a BTES system, one would require $518\,400\,\mathrm{MJ}/2.2\,\mathrm{MJ\,m^{-3}\,K^{-1}}/20\,\mathrm{K} = 11\,782\,\mathrm{m}^3$ of rock. This is equivalent to a borehole array of diameter 21 m and depth 35 m, which is perfectly feasible.

We can thus conclude that tanks of water, or similar thermal storage vessels, are ideal for short-term (e.g. diurnal) thermal storage, but that UTES is ideal for seasonal storage of heat.

15.1 We will use Equation 15.4 to derive the thermal conductivity, implying that

$$\begin{aligned}\lambda &= 2.303q/(4\cdot\pi\cdot\text{gradient})\\ &= 2.303\times 62.1\text{ W m}^{-1}/(4\times\pi\times 4.27\text{ K})\\ &= 2.67\text{ W m}^{-1}\text{ K}^{-1}\,(\text{round up to two significant figures} = 2.7\text{ W m}^{-1}\text{ K}^{-1})\end{aligned}$$

Now, from Equation 15.3,

$$\theta_b - \theta_o \approx qR_b + \frac{q}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda t}{r_b^2 S_{VC}}\right) - 0.5772\right]$$

The intercept, when $\log_{10} t = 0$, is +1.261 K. However, note that time is given in minutes and *not* in seconds – thus, when $t = 1\text{ min} = 60\text{ s}$, the intercept $(\theta_b - \theta_0)$ is +1.261 K. Thus, given that the borehole radius r_b is 0.07 m,

$$R_b = \frac{1.261\text{ K}}{62.1\text{ W m}^{-1}} - \frac{1}{4\pi\lambda}\left[2.303\log_{10}\left(\frac{4\lambda 60\text{ s}}{(0.07\text{ m})^2 S_{VC}}\right) - 0.5772\right]$$

If we substitute $\lambda = 2.67\,\text{W m}^{-1}\text{K}^{-1}$ and $S_{VC} = 2.2\,\text{MJ m}^{-3}\text{K}^{-1}$, we obtain $R_b = 0.020\,\text{K m W}^{-1} - (-0.101\,\text{K m W}^{-1}) = 0.121\,\text{K m W}^{-1}$ (round up to two significant figures = $0.12\,\text{K m W}^{-1}$).

15.2 The minimum time required for the logarithmic assumption to become valid is given by Equation 15.6:

$$t > \frac{5r_b^2 S_{VC}}{\lambda} \tag{15.6}$$

For the test in question 15.1, the minimum time is thus given by $5 \times (0.07\,\text{m})^2 \times 2\,200\,000\,\text{J m}^{-3}\text{K}^{-1}/2.67\,\text{W m}^{-1}\text{K}^{-1} = 20\,187\,\text{s} = 5.6\,\text{h}$. Thus, the practice of using data after 10 h seems reasonable.

For the needle probe test, we set r_b to the probe radius, 0.00317 m, yielding a minimum time of $5 \times (0.00317\,\text{m})^2 \times 1\,500\,000\,\text{J m}^{-3}\text{K}^{-1}/1.4\,\text{W m}^{-1}\text{K}^{-1} = 54\,\text{s}$. Thus, we would expect the data to approximate to a straight line after 1 min into the test, meaning that the typical test duration of 5 min (300 s) is reasonable.

16.1 The rate of heat discharge is $15\,\text{kW} = 15\,\text{kJ s}^{-1}$. Thus, every second, no more than 15 kJ of heat is 'diluted' in 25 L of abstracted groundwater, with a volumetric heat capacity of $4.18\,\text{kJ L}^{-1}\text{K}^{-1}$. Thus, the maximum temperature change that would be predicted is $15\,\text{kJ}/(25\,\text{L} \times 4.18\,\text{kJ L}^{-1}\text{K}^{-1}) = 0.14\,\text{K}$. Thus, the maximum predicted temperature in the abstracted water would be 10.74°C.

16.2 The equivalent amount of glycol released is 10 L, which represents a mass of 10 L × 1.113 kg L^{-1} = 11.13 kg = 11 130 g. If none of this glycol biodegraded or was retarded in the aquifer and if all of it was evenly distributed in the pumped groundwater over the course of the day (=86 400 s), the resulting concentration would be

$$11\,130 \text{ g day}^{-1} / \left(25 \text{ L s}^{-1} \times 86\,400 \text{ s day}^{-1}\right) = 0.0052 \text{ g L}^{-1} = 5.2 \text{ mg L}^{-1}$$

To put this in context, Melinder (2010) reckoned that the minimum lethal dose of ethylene glycol for an adult is around 100 mL (around 111 g). This would represent around 21 000 L of the abstracted water. These calculations were not performed by a qualified toxicologist (but they illustrate that all the information is available)!

Symbols

A note on the dimensionality of energy and power

The joule is the SI unit of energy. It can be regarded as dimensionally equivalent to work. *Work* can be defined as *force* × *distance over which force is applied.*

The dimension of energy and work is thus $[M][L][T]^{-2}[L]$ or $[M][L]^2[T]^{-2}$.

$$1 \text{ joule} = 1 \text{ kg m}^2 \text{ s}^{-2}$$

The watt is the SI unit of power, or rate of delivery of energy. The dimension of power is thus $[M][L]^2[T]^{-3}$.

$$1 \text{ watt} = 1 \text{ kg m}^2 \text{ s}^{-3}$$

Note, however, that when considering electricity, *power* = *current* × *voltage.*

$$1 \text{ watt} = 1 \text{ V C s}^{-1}$$

General symbols

A = cross-sectional area perpendicular to the direction of heat flow or groundwater flow; $[L]^2$; typically in m^2.
e = 2.7182 (the base to which natural logarithms are calculated). Note that $\ln(10) = \log_e(10) = 2.303$.
η = efficiency of electricity production; (dimensionless) – see Section 4.16.
g = acceleration due to gravity = c. $9.81\,m\,s^{-2}$; $[L][T]^{-2}$.
γ = Euler's constant = 0.5772.
L = distance coordinate; [L]; typically in m.
m = mass; [M]; typically in kg.
P = pressure; $[M][L]^{-1}[T]^{-2}$; typically in pascals (Pa); 1 Pa = 1 newton per m^2 ($N\,m^{-2}$).
r = radial coordinate; [L]; typically in m.
t = time; [T]; typically in s.
W = work performed; $[M][L]^2[T]^{-2}$; typically in joules (J).
W = rate of work performed; $[M][L]^2[T]^{-3}$; typically in $J\,s^{-1}$ or W.
x = distance coordinate; [L]; typically in m.

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z = distance coordinate in the vertical direction; [L]; typically in m. Depending on the context, this may be either depth below the earth's surface *or* elevation above some arbitrary datum (i.e. take care!).

Heat flow symbols

$\mathring{A}$ = internal (e.g. radiogenic) heat production per unit volume of the earth's crust; [M][L]$^{-1}$[T]$^{-3}$; typically in W m^{-3}.
α_L = linear coefficient of thermal expansion; typically in K^{-1} (see Glossary).
α_V = volumetric coefficient of thermal expansion; typically in K^{-1} (see Glossary).
α_{sw}, α_{lw} = short-wave and long-wave albedo of a lake or ground surface; typically expressed as a fraction or percentage.
C = cooling load delivered to a building (i.e. rate of heat extraction from a building); [M][L]^2[T]$^{-3}$; typically in J s^{-1} or W.
Cl = cloud cover; typically in oktas (eighths of the sky).
D = effective thickness of aestifer *or* depth of borehole over which heat extraction takes place; [L]; typically in m.
e_0 = saturated vapour pressure of water at the water surface temperature; [M][L]$^{-1}$[T]$^{-2}$; typically in pascals (Pa) or mm of mercury (mmHg).
e_a = actual vapour pressure of water in air; [M][L]$^{-1}$[T]$^{-2}$; typically in pascals (Pa) or mmHg.
E = efficiency (usually expressed as a ratio or percentage).
E = evaporation rate; [L][T]$^{-1}$; often in mm day^{-1}.
E_b = 'black body' energy radiated; typically in W m^{-2}.
ε = emissivity: the ratio of energy radiated by a given real material to the energy radiated by an ideal black body at the same temperature.
η_H = Hellström efficiency (see Box 10.3); usually cited in %.
F = flow rate of carrier fluid in a closed-loop system; [L]^3[T]$^{-1}$.
F_{turb} = flow rate of carrier fluid in a closed-loop system necessary to ensure transitional-turbulent flow; [L]^3[T]$^{-1}$.
G = rate of heat extraction from, or dumping to, a ground-coupled receptor; [M][L]^2[T]$^{-3}$; typically in J s^{-1} or W.
H = heating load delivered to a building; [M][L]^2[T]$^{-3}$; typically in J s^{-1} or W.
H_{env} = heat transferred to or from the environment (e.g. ground, river, lake or air); [M][L]^2[T]$^{-3}$; typically in J s^{-1} or W.
H_{in} = heat input to a heat engine; [M][L]^2[T]$^{-3}$; typically in J s^{-1} or W.
$\bar{h}$ = local coefficient of heat transfer; typically in W m^{-2} K^{-1}.
L_V = latent heat of vaporization; typically in MJ kg^{-1} (or MJ L^{-1} for liquids).
λ = thermal conductivity; typically in W m^{-1} K^{-1}.
λ_g = thermal conductivity of grout in a closed-loop borehole; typically in W m^{-1} K^{-1}.
λ_{bh}, λ_{bv} = bulk horizontal and bulk vertical thermal conductivities of anisotropic rocks (see Box 10.1); typically in W m^{-1} K^{-1}.

Λ = thermal conductance, often in $W\,m^{-2}\,K^{-1}$ (i.e. per m^2 of the material), or in $W\,K^{-1}$ for an entire structure.
Q = rate of heat flow; $[M][L]^2[T]^{-3}$; typically in $J\,s^{-1}$ or W.
q = specific rate of heat flow or transfer; rate of heat transfer per metre length of borehole; $[M][L][T]^{-3}$; typically in $J\,s^{-1}\,m^{-1}$ or $W\,m^{-1}$. See also q^*.
q^* (or, in Chapter 12, q) = specific rate of heat flow; rate of heat transfer per metre of cross-sectional area; $[M][T]^{-3}$; typically in $J\,s^{-1}\,m^{-2}$ or $W\,m^{-2}$.
r = radius or radial distance; [L]; typically in m.
r_{array} = effective radius of a cylindrical or hexagonal array of closed-loop boreholes; [L]; typically in m.
r_b = radius of a closed-loop borehole; [L]; typically in m or mm.
r_p = radius of a closed-loop heat exchange pipe; [L]; typically in m or mm.
r_U = the radius of the U-tube in a closed-loop borehole [L]; typically in m or mm.
R = universal gas constant = $8.314\,J\,K^{-1}\,mol^{-1}$.
R = thermal resistance (or electrical resistance); typically in $m^2\,K\,W^{-1}$ (or ohms).
R_{ex} = extraterrestrial solar irradiance (i.e. solar radiance incident at the outer edge of the atmosphere); $[M][T]^{-3}$; typically in $J\,s^{-1}\,m^{-2}$ or $W\,m^{-2}$.
R_b = thermal resistance of a borehole; typically in $m^2\,K\,W^{-1}$ per m of borehole, or $K\,m\,W^{-1}$.
R_p = thermal resistance of a horizontal closed-loop heat exchange pipe; typically in $m^2\,K\,W^{-1}$ per m of pipe, or $K\,m\,W^{-1}$.
R_n = net incident radiation on the earth's surface; $[M][T]^{-3}$; typically in $J\,s^{-1}\,m^{-2}$ or $W\,m^{-2}$.
S_C = specific heat capacity; typically in $J\,kg^{-1}\,K^{-1}$.
S_{VC} = volumetric heat capacity; typically in $J\,m^{-3}\,K^{-1}$.
S_{VCwat} = volumetric heat capacity of water = c. $4.18\,MJ\,m^{-3}\,K^{-1}$ (but is temperature dependent).
S_{VCaq} = volumetric heat capacity of a saturated aquifer; typically in $J\,m^{-3}\,K^{-1}$.
S_{VCcar} = volumetric heat capacity of a carrier fluid; typically in $J\,m^{-3}\,K^{-1}$.
SDR = the ratio of a pipe's outer diameter to its wall thickness.
σ = Stefan–Boltzmann constant = $5.67 \times 10^{-8}\,W\,m^{-2}\,K^{-4}$.
t_{earth} = hypothetical age of the earth; [T].
t_s = time taken for a closed-loop system to approach steady state (the time at which a steady-state approximation begins to be a better description of the loop's behaviour than a radial flow model); [T].
T_{the} = thermal transmissivity. The product of thermal conductivity and thickness of an aestifer; typically in $W\,K^{-1}$.
θ = temperature; typically in °C.
θ^o = absolute temperature in K.
θ_o = ambient temperature of groundwater *or* initial, far-field temperature of an aestifer.
θ_b = the average temperature of the carrier fluid in a closed-loop borehole (i.e. the average of the upflow and downflow fluid temperatures).

θ_i = initial temperature of the earth's molten interior (according to Kelvin's 'cooling earth' model).
θ_s = temperature of the earth's surface.
$\theta_{s,b}$ = the average steady-state temperature of carrier fluid in a borehole.
θ_{sur} = temperature of the surface of a lake.
θ_{up}, θ_{down} = temperature of a carrier fluid emerging from or entering the shanks of a borehole heat exchanger.
$\Delta\theta$ = a temperature change or differential; typically in K or °C.
U-value = a buildings engineering designation for thermal conductance of a building component per square metre of area, often in $W\,m^{-2}\,K^{-1}$.
U = total thermal conductance of a building in $W\,K^{-1}$.
v_w = wind speed; $[L][T]^{-1}$; typically in $m\,s^{-1}$.
V_{aest} = volume of aestifer under consideration; $[L]^3$; typically in m^3.
V_{lake} = thermally active volume of lake under consideration; $[L]^3$; typically in m^3.
ψ = current geothermal gradient; in $K\,m^{-1}$.
ψ_c = psychrometric constant, typically in the range of 59–66 $Pa\,K^{-1}$.

Groundwater/fluid flow symbols

b_a = aperture of a fracture; [L].
B = bleed rate, as a fraction of total fluid flow in a standing column well ($0 < B < 1$).
B = linear head losses in a pumping well; $[T][L]^{-2}$.
B' = linear well losses in a non-ideal pumping well; $[T][L]^{-2}$.
β = a dimensionless parameter defining a well doublet, where $\beta = 2Z/(T\pi iL)$.
C = non-linear head losses in a non-ideal pumping well; $[T]^n[L]^{(1-3n)}$.
C_{DW} = Darcy–Weisbach constant for fluid flow in pipes; [dimensionless].
D = borehole depth or effective thickness of aquifer; [L]; typically in m.
D^* = dispersion coefficient; $[L]^2[T]^{-1}$.
F = flow rate of carrier fluid in a closed-loop system; $[L]^3[T]^{-1}$.
F_{turb} = flow rate of carrier fluid in a closed-loop system necessary to ensure transitional-turbulent flow; $[L]^3[T]^{-1}$.
h = groundwater head; [L]; typically in m relative to some arbitrary datum.
H = elevation of water table; [L]; relative to the base of an unconfined aquifer.
K = hydraulic conductivity; $[L][T]^{-1}$; typically in $m\,s^{-1}$ or $m\,day^{-1}$.
L = separation; [L]; between abstraction and injection wells in a doublet scheme.
λ^* = effective thermal conductivity of the saturated aquifer material, modified to take into account effects of hydrodynamic dispersion; typically in $W\,m^{-1}\,K^{-1}$.
μ, μ_w = the dynamic viscosity of a fluid or of water; $[M][T]^{-1}[L]^{-1}$; typically in $kg\,s^{-1}\,m^{-1}$.
n = power law coefficient in the Hantush–Bierschenk equation – Equation 8.18.
n = porosity; (dimensionless); typically expressed as a fraction or percentage.
n_e = effective porosity; (dimensionless); typically expressed as a fraction or percentage.

ν = the kinematic viscosity of a fluid; $[L]^2[T]^{-1}$; typically in $m^2\,s^{-1}$.
r_w = radius of a pumped well; [L].
Re = Reynolds number; (dimensionless).
ρ_w = density of water = c. $1000\,kg\,m^{-3} = 1\,kg\,L^{-1}$; $[M][L]^{-3}$.
s = drawdown; [L]; typically in m.
s_a = available drawdown in a pumping well; [L].
s_w = drawdown in a pumping well; [L].
S = groundwater storage coefficient or storativity; (dimensionless).
S_S = specific storage; $[L]^{-1}$.
S_Y = specific yield of an unconfined aquifer; (dimensionless).
T = transmissivity (T_f = transmissivity of a fracture); $[L]^2[T]^{-1}$; typically in $m^2\,s^{-1}$ or $m^2\,day^{-1}$.
u = a dimensionless parameter = $r^2S/4tT$, used in the Theis well function.
v = actual linear velocity of groundwater flow = v_D/n_e; $[L][T]^{-1}$; typically in $m\,day^{-1}$ or $m\,s^{-1}$.
v_D = Darcy velocity (=rate of groundwater flux per unit cross-sectional area of aquifer); $[L][T]^{-1}$; typically $m\,day^{-1}$ or $m\,s^{-1}$.
w = width of aquifer under consideration; [L].
$W(u)$ = the Theis well function; (dimensionless).
Z = rate of groundwater flow; $[L]^3[T]^{-1}$; typically in $L\,s^{-1}$, $m^3\,s^{-1}$ or $m^3\,day^{-1}$.

Electricity flow symbols

F = Faraday's constant, the electric charge on 1 mol of electrons = $96\,485\,C\,mol^{-1}$.
I = current; typically in amps (A); 1 A = 1 coulomb per second ($C\,s^{-1}$).
V = voltage or potential difference; typically in volts (V).
R = electrical resistance; typically in ohms (Ω).

Glossary

Absolute zero: The temperature at which a material has no utilisable thermal energy. Zero on the Kelvin scale or –273.15°C on the Celsius scale.

Aestifer: A body or stratum of rock or sediment that has adequate thermal properties (thermal conductivity, heat capacity) to permit the economic exploitation of ground source heat.

agl: Above ground level.

Air conditioning: The modification and control of the temperature, humidity, quality and circulation of air, usually in an interior space. Normally, the term refers to the cooling of circulating air, achieved by one or a combination of the following mechanisms: direct circulation of, or heat exchange with, a naturally cool fluid or medium; artificial chilling via a refrigeration (heat pump) cycle; evaporative cooling. Chilling of circulating air is often accompanied by dehumidification and removal of condensed water (as the air cools, the amount of water vapour it is able to contain decreases).

Air cycle heat pump: In an open, air cycle heat pump (of the type first proposed by William Thomson in 1852), mechanical compression and expansion are performed on the fluid that actually delivers the heat effect – air. In Thompson's concept, air is allowed to expand into a reservoir, resulting in cooling. The cooled air absorbs heat from the environment through the reservoir walls and then is mechanically compressed into a second cylinder. From here, the warm air is released in a controlled manner to the space to be heated.

Albedo: A measure of the reflectivity of the earth's surface – the ratio of reflected sunlight (electromagnetic radiation) to incident light. The albedo of fresh snow can be as high as 80%, whereas the albedo of water is usually less than 10%.

Aperture: The width of a fracture opening in a rock, or the diameter of a pore space in a sediment.

Aquifer: A body or stratum of rock or sediment that has adequate hydraulic properties (transmissivity, hydraulic conductivity, storage) to permit the economic abstraction of groundwater.

ASHRAE: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

asl: Above sea level.

ASME: American Society of Mechanical Engineers.

Asthenosphere: The plastic, flowing part of the earth's subsurface, below the more rigid *lithosphere*. The asthenosphere lies in the upper mantle.

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ATES: Aquifer thermal energy storage. A form of UTES where heat is stored in the form of warm groundwater (e.g. carrying a cargo of surplus heat from the summer cooling season) that has been injected to an aquifer. The warm groundwater is re-abstracted in winter to provide heating.

Balanced scheme: A balanced ground source heating and cooling scheme will have approximately similar annual loads of heat extracted from and rejected to the ground. An unbalanced scheme will be strongly biased, over the course of a year, towards net heating or net cooling.

bgl: Below ground level.

Bivalent scheme: A ground source heating/cooling scheme that satisfies only a portion of a building's heating or cooling demand, and is supplemented by alternative heating or cooling arrangements.

BTES: Borehole thermal energy storage. The storage of surplus heat in the subsurface for re-extraction at times of high demand, specifically by means of closed-loop arrangements installed in boreholes. BTES is a type of UTES.

bwt: Below well top.

Celsius: Synonymous with centigrade. A unit (°C) and scale of measuring temperature, characterised by 0°C being set to the melting point of ice and 100°C to the boiling point of water.

Cementitious thermally enhanced grout: The State of New Jersey, USA (GeoExchange, 2003), defines this as a mixture of 94 lb (43 kg) of cement with 200 lb (91 kg) of dry fine silica sand, 1.04 lb (470 g) of sodium bentonite and c. 21 oz (600 g) of sulphonated naphthalene superplasticiser (per bag of cement) and 6.19 gal (23.4 L) of water to give a target density of 2.18 kg L^{-1}. See also Allan and Philippacopoulos (1999).

Centigrade: See Celsius.

Closed-loop ground source heating system: A system whereby heat is extracted from the ground by a chilled 'carrier' fluid, circulating within a buried heat exchanger – typically a subsurface loop of pipe installed in a trench or borehole.

Coefficient of performance (COP): The ratio of heating (or cooling) effect delivered by a heat pump to the electricity (or other primary energy) required to power the compressor. It is an instantaneous measure and will depend on the temperature of the heat source and heat sink at that moment in time (see Seasonal performance factor).

Coefficient of thermal expansion: The *linear coefficient of thermal expansion* (α_L) is the fractional increase in the length (L) of a bar of material (or in the dimension of an object) for every degree temperature (θ) rise. It is cited in K^{-1} and is given by the formula $\frac{1}{L}\frac{\partial L}{\partial \theta}$. For rocks, values of 3 to 9×10^{-6} K^{-1} are typical, depending on porosity and mineral composition. The volumetric coefficient of thermal expansion (α_V) is the fractional increase in volume per degree temperature rise. In general, the value of α_V is around three times α_L.

Confined aquifer: A transmissive stratum of sediment or rock that is overlain by a low-permeability *aquitard* and where the groundwater head is higher than the top of the aquifer.

Darcy velocity: The flux rate of groundwater through an aquifer or porous medium per unit cross-sectional area (i.e. $m^3 s^{-1}$ per m^2, or $m s^{-1}$). As groundwater can only flow through pore apertures or fractures, the Darcy velocity (v_D) is related to the actual average linear velocity (v) by the expression $v_D = vn_e$, where n_e is the effective porosity.

DHW: Domestic hot water; that is, the water that comes out of the hot tap in your kitchen or bathroom.

DX: Direct expansion or direct circulation heat pump system – a type of closed-loop system where refrigerant circulates in the ground and where the ground loop functions as the evaporator (or condenser) of the heat pump.

EA: Environment Agency (of England and Wales).

Emissivity: The ratio of energy radiated by a given real material to the energy radiated by an ideal black body at the same temperature.

Enthalpy: a measure of the heat content of a substance per unit mass, depending not only on temperature, but also on pressure and volume (Boyle, 2004).

EU: European Union.

Fahrenheit: An outdated system of measuring temperature, still used in the United States. On the Fahrenheit scale, water freezes at 32°F and boils at 212°F. To convert a temperature in Fahrenheit (θ_F) to a temperature in centigrade (θ_C), use theformula

$$\theta_C = 5(\theta_F - 32)/9$$

Geothermal energy: In this book, we have chosen to restrict the term to relatively high-temperature, high-enthalpy heat occurring in the geosphere. Such high-enthalpy geothermal energy is typically accessed either via very deep boreholes or in locations with a naturally high geothermal gradient (e.g. Iceland), or both. New EU legislation implies that low-enthalpy ground source heat should also be included in the definition of geothermal energy.

Geothermics: The study of geothermal energy.

Ground source heat: Very low-enthalpy heat that exists in the subsurface at 'normal' temperatures – the type of heat that we are considering in the science of thermogeology.

GSHC scheme: Any ground source heating and/or cooling scheme. This may or may not employ a heat pump.

GSHP: Ground source heat pump.

Head: A measure of the potential energy of water due to a combination of elevation (z) and pressure (P). Groundwater always flows from high head to low head. Groundwater head (h) is defined by the formula

$$h = \frac{P}{\rho g} + z$$

where g = acceleration due to gravity and ρ = fluid density.

HDPE: High-density polyethene.

HVAC: Heating, ventilation and air conditioning.

Hydrogeology: The study of the occurrence, movement and exploitation of water in the geosphere. The study of groundwater.

ID: Internal diameter (of a pipe).

IGSHPA: International Ground Source Heat Pump Association (currently based in Oklahoma, USA).

IPA spirit: A carrier fluid typically comprising ethanol, denatured with around 10% isopropyl alcohol (alternatively named isopropanol, abbreviated to IPA).

Kelvin: The SI unit of temperature. A degree Kelvin is equal to a degree centigrade, but the zero points are different: 0 K is absolute zero (–273.15°C), while 0°C is the freezing point of water. To convert a temperature in centigrade (θ_C) to an absolute temperature (θ^o) in Kelvin, use the formula

$$\theta^o = \theta_C + 273.15$$

Latent heat: The heat absorbed or released by a material purely by virtue of a change in phase – for example, from ice to water or from water to steam. The absorption or release of latent heat is not accompanied by any transfer of 'sensible heat' – that is, any change in temperature of the material.

Lithosphere: The upper, relatively rigid part of the earth's subsurface, floating on top of the plastic, ductile *asthenosphere*. The lithosphere comprises the earth's crust and the uppermost part of the mantle.

MDPE: Medium-density polyethene.

mOD: Metres above Ordnance datum. A bizarre British way of saying 'metres above sea level'!

Monovalent scheme: A ground source scheme that fully satisfies a building's heating or cooling demand (see 'bivalent scheme').

Net radiation (R_n): The net radiative flux incident on 1 m^2 of the earth's surface. It comprises the sum of short-wave insolation and long-wave atmospheric radiation, less reflection and less long-wave back radiation from the earth's surface. It varies throughout the day (it may be positive during the morning and negative during the night), but its annual average value is typically positive and several tens of $W m^{-2}$ in temperate regions, exceeding 100 $W m^{-2}$ in the tropics. See Section 3.4.

NGWA: National Ground Water Association (of the United States).

OD: Outer diameter (of a pipe).

Open hole: The practice of drilling into a well-lithified aquifer without any form of well screen or casing. The sides of the well or borehole are strong and stable enough to support themselves. In less stable rocks and sediments, open hole drilling can still be achieved if a dense drilling mud is used to support the borehole walls.

Open-loop ground-coupled heating system: A system whereby water is physically abstracted from a well or spring in an aquifer (or another natural water body: a flooded mine, a lake, a river or the sea). Heat is extracted directly from this flux of water with its advected 'cargo' of heat.

PE: Polyethene (alternatively named polythene or polyethylene).

Permeability: An intrinsic property of a porous medium, describing the ease with which a fluid can pass through it. The *intrinsic permeability* (measured in darcys or m^2) can be applied, irrespective of the fluid in question. If the fluid is water at normal temperatures and pressures, the term *hydraulic conductivity* is used (typically measured in $m\ s^{-1}$).

PEX: Cross-linked polyethene. A type of high-density polyethene that can withstand higher temperatures.

PN: Pressure Number – a pressure rating for polyethene pipe, nominally in bars.

PP: Polypropene (polypropylene).

Prophylactic heat exchanger: A heat exchanger (typically a plate heat exchanger) that transfers heat between a natural fluid (whose composition is not controlled and which may lead to problems of corrosion, biofouling or chemical incrustation) and a carrier fluid of controlled composition (e.g. antifreeze solution). This ensures that the natural water does not enter a building loop or heat pump, thus protecting the evaporator from corrosion or fouling. However, the prophylactic heat exchanger may be subject to these problems and may need to be regularly cleaned, maintained or even (occasionally) replaced.

PVC: Polyvinyl chloride.

Screen temperature: The air temperature as measured in a Stevenson screen: a louvred, shaded, ventilated white box, mounted around 1.5 m above ground level.

SCW: Standing column well (see Chapter 13).

Seasonal performance factor (SPF): The ratio of heating (or cooling) effect, integrated over an entire heating or cooling season, to the amount of electricity (or other primary energy) consumed by the compressor over the same period. A *system* performance factor is the ratio of heat output to total electrical energy consumed by the compressor and any auxiliary devices (circulation pumps, etc.).

Sensible heat: The transfer of heat to or from a material that manifests itself as a change in temperature of the material (contrast with 'latent heat').

SEPA: Scottish Environment Protection Agency.

Sialic: A rock that is rich in *si*lica and *al*uminium minerals. This chemical composition is typical of the upper continental crust.

Specific heat capacity (S_C): This describes how good a material is at storing heat. It is defined as the amount of heat released from a unit mass of material, corresponding to a 1 K temperature change. It is measured in $J\,kg^{-1}\,K^{-1}$.

Specific heat extraction rate: See Specific thermal absorption.

Specific installed thermal output: For a closed-loop scheme, this is the peak installed heating or cooling output of a ground source heat scheme, divided by the number of metres of borehole (vertical systems) or trench (horizontal systems) required to support the system. It is thus typically cited in $W\,m^{-1}$.

Specific thermal absorption: For a closed-loop scheme, this is the peak rate of heat transfer between ground and ground loop (i.e. the amount of heat extracted from or dumped to the ground), divided by the number of metres of borehole (vertical systems) or trench (horizontal systems) required to support the system. It is thus typically cited in $\mathrm{W\,m^{-1}}$. In heating mode, it is *lower* than the *specific installed thermal output* (because the electrical energy input to the heat pump also contributes to the heating system's total output). In active cooling mode, it is *higher* than the *specific installed thermal output*.

Temperature: A measure of the thermal potential energy of a fluid or body. At a molecular level, temperature is related to the kinetic or vibrational energy of motion of the molecules or atoms. Temperature can be measured in degrees Fahrenheit (°F), degrees centigrade/Celsius (°C) or Kelvin (K).

Thermal conductivity: This describes how good a material is at conducting heat. It is measured in $\mathrm{W\,m^{-1}\,K^{-1}}$. It is defined by Fourier's law: a material of thermal conductivity $1\,\mathrm{W\,m^{-1}\,K^{-1}}$ under a temperature gradient of $1\,\mathrm{K\,m^{-1}}$ will conduct 1 W of heat through every square metre of its cross section.

Thermal Grout 85: A specific grout mix of quartz sand, bentonite and water. The State of New Jersey (GeoExchange, 2003) defines it as mix of 54 lb (24 kg) bentonite and 200 lb (91 kg) of fine dry silica sand, mixed with 17.5 gal (66 L) of water to achieve a density of $13.1\,\mathrm{lb\,gal^{-1}}$ ($1.57\,\mathrm{kg\,L^{-1}}$). See also Allan and Philippacopoulos (1999).

Thermogeology: The study of the occurrence, movement and exploitation of low-enthalpy heat in the relatively shallow geosphere. By 'relatively shallow', we are typically talking of depths of down to 300 m or so. By 'low enthalpy', we are usually considering temperatures of less than 40°C.

TRT: Thermal response test. A constant-rate heat injection test, typically carried out on a closed-loop borehole heat exchanger, with the primary objective of deriving the ground's average thermal conductivity.

USEPA: United States Environmental Protection Agency.

UTES: Underground thermal energy storage. Using the subsurface as a heat reservoir, to store excess or waste heat (e.g. from cooling activities or from solar harvesting) during the summer. This surplus heat can then be re-extracted during winter. The term encompasses both ATES and BTES schemes.

VDI: Verein Deutscher Ingenieure (German Association of Engineers).

Volumetric heat capacity (S_{VC}): This describes how good a material is at storing heat. It is defined as the amount of heat released from a unit volume of material, corresponding to a 1 K temperature change. It is measured in $\mathrm{J\,m^{-3}\,K^{-1}}$.

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Units

Mass

1 kg = 2.2046 lb
1 tonne (metric) = 1000 kg = 2205 lb
1 tonne = 1.102 tons
1 lb (pound) = 16 oz (ounces)

Length

1 inch = 25.4 mm
1 foot = 0.3048 m
1 yard = 0.914 m

Area

1 m^2 = 10.76 ft^2
1 ha = 10 000 m^2
1 acre = 4046.9 m^2 = 0.40469 ha
1 acre = 4840 square yards

Volume

1 acre-foot = 43 560 ft^3 = 1233 m^3
1 imperial gallon = 4.5461 L
1 US gallon = 3.7854 L
1 m^3 = 35.31 ft^3
1 barrel (oil) = 42 US gallons
1 barrel (oil) = 158.99 L

Volumetric flow rate

1 imperial gallon per minute = 0.07577 L s^{-1}
1 L s^{-1} = 13.2 imperial gallons per minute

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1 US gallon per minute = 0.06309 $L\,s^{-1}$
1 $L\,s^{-1}$ = 15.85 US gallons per minute
100 $m^3\,d^{-1}$ = 1.157 $L\,s^{-1}$
1 $L\,s^{-1}$ = 86.4 $m^3\,d^{-1}$

Density

1000 $kg\,m^{-3}$ = 1 $kg\,L^{-1}$ = 1 $g\,cm^{-3}$
1 $lb\,ft^{-3}$ = 16.02 $kg\,m^{-3}$

Force

1 newton (N) = 1 $kg\,m\,s^{-2}$

Viscosity

1 centiPoise (cP) = 1 mPa s = 0.001 $N\,s\,m^{-2}$
1 centiPoise (cP) = 0.001 $kg\,m^{-1}\,s^{-1}$

Pressure

1 pascal (Pa) = 1 $N\,m^{-2}$
10 kPa = 1.02 m of H_2O
1 atmosphere = 101 325 Pa
1 atmosphere = 406.8 inches of H_2O = 33.90 ft of H_2O (at 4°C)
1 atmosphere = 760 mmHg = 29.92 inches of mercury (at 32°F)
1 atmosphere = 10.33 m of water (at 4°C)
1 atmosphere = 14.696 pounds per square inch (psi)
1 $kg\,m^{-2}$ = 9.81 Pa
1 bar = 0.9869 atmosphere
1 bar = 100 000 Pa
1 bar = 14.50 psi
1 torr = 1 mmHg = 133.32 Pa

Energy and heat

1 $calorie_{15}$ = specific heat capacity of 1 g water at 15°C = 4.1855 J
1 calorie (thermochemical) = 4.1840 J
1 International Table calorie = 4.1868 J

1 joule = 9.48×10^{-4} British thermal units (Btu)
1 joule = 10 000 000 ergs
1 kWh = 3 600 000 J = 3600 kJ = 3.6 MJ
1 kWh = 3412 Btu
1 Btu (international) = 1055.056 J
1 Btu = 0.000293 KWh
1 therm = 100 000 Btu = 105.5 MJ
1 Thermie = 1 000 000 calories = 4.1855 MJ
1 tonne oil equivalent = 44.76 GJ = 12 433 kWh
1 tonne oil equivalent (alternative definition) = 41.87 GJ and 11 630 kWh

Power

1 Btu h^{-1} (international) = 0.2931 W
1 Btu s^{-1} (international) = 1055.1 W
1 ton (refrigeration) = 12 000 Btu h^{-1}
1 ton (refrigeration) = 3517 W

Specific heat capacity

1 Btu lb^{-1} $°F^{-1}$ = 4186.8 J kg^{-1} K^{-1} = 1 kcal kg^{-1} $°C^{-1}$
1 kJ kg^{-1} K^{-1} = 0.239 Btu lb^{-1} $°F^{-1}$
1 Btu ft^{-3} $°F^{-1}$ = 67.07 kJ m^{-3} K^{-1}
1 MJ m^{-3} K^{-1} = 14.91 Btu ft^{-3} $°F^{-1}$

Specific heat flux

1 Btu h^{-1} ft^{-2} = 3.154 W m^{-2}

Thermal conductivity

1 Btu h^{-1} ft^{-1} $°F^{-1}$ = 1.731 W m^{-1} K^{-1}
1 W m^{-1} K^{-1} = 0.5778 Btu h^{-1} ft^{-1} $°F^{-1}$

Thermal resistance of a borehole

1 h ft °F Btu^{-1} = 0.5778 K m W^{-1}
1 K m W^{-1} = 1.731 h ft °F Btu^{-1}