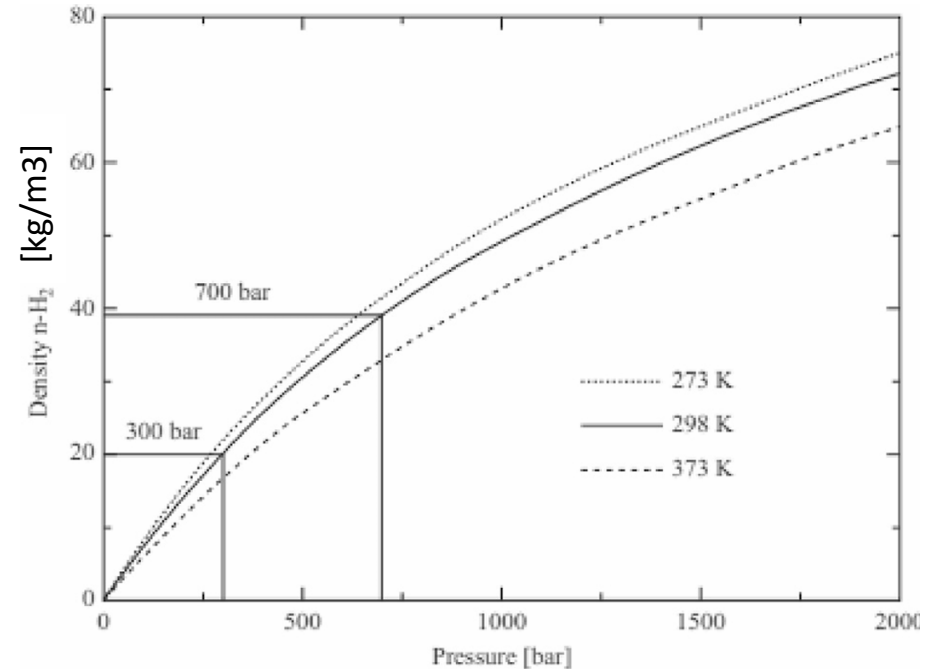


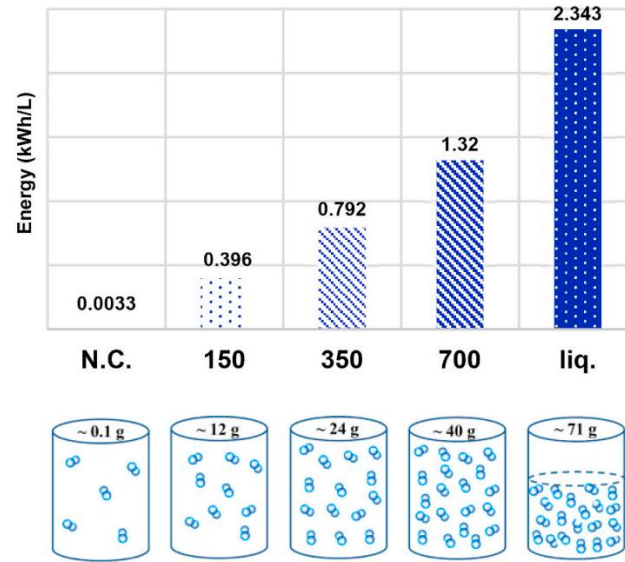
# HYDROGEN COMPRESSION TECHNOLOGIES

# Pressure Effects on Hydrogen Density

- At normal temperature and pressure:  
hydrogen density =  $0.09 \text{ kg/m}^3$
- Compressed to 35 MPa:  
density increases to  $23 \text{ kg/m}^3$
- Compressed to 70 MPa:  
density increases to  $38 \text{ kg/m}^3$
- Higher pressure storage dramatically improves volumetric energy density
- This enables practical onboard storage for vehicle applications



# Hydrogen Density vs. Pressure

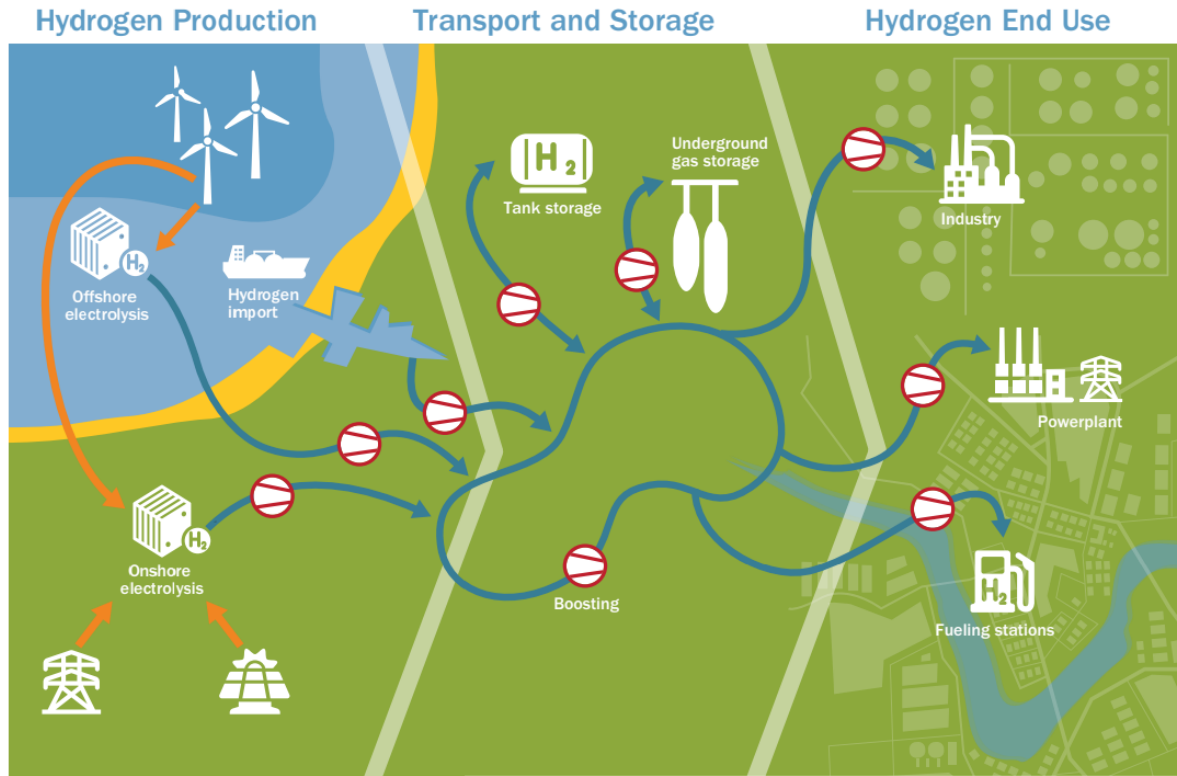


(a)

As pressure increases from 1 bar to 700 bar, the hydrogen density increases from 0.1 g/L to 40 g/L, and consequently energy volumetric density increases from 0.0033 kWh/L to 1.32 kWh/L respectively. As rule of thumb to pressurize hydrogen to 700 bar, the consumed power is about 10% of the gas energy content.

# The Hydrogen Compression Challenge

- Hydrogen is the lightest element with the lowest density, requiring substantial energy input for compression
- Key challenges: hydrogen embrittlement when metals are exposed to high pressures and temperatures. Hydrogen embrittlement (HE) is a process resulting in a decrease in the fracture toughness or ductility of a metal due to the presence of atomic hydrogen.
- Hydrogen's small molecular size enables diffusion through seals and joints, causing leakage and safety concerns



**Figure 2.** Typical applications where compression will be required in our future hydrogen economy, based on a hydrogen distribution grid.

## 1. Injection into the transport grid.

This application comprehends **the boosting of hydrogen to approximately 30 to 80 bar**, as is currently foreseen for the European Hydrogen Backbone (van Rossum, et al., 2022). In the early stage of (inter) national hydrogen distribution networks, the pressure is expected to be built up by the injection **compressors** only, which are **located near electrolyzers**. In case of increased exploitation of these networks, **larger pressure losses** can be foreseen that may bring the **need for booster** compressors. Compressors will also be required at injection stations that blend hydrogen into existing natural gas streams.

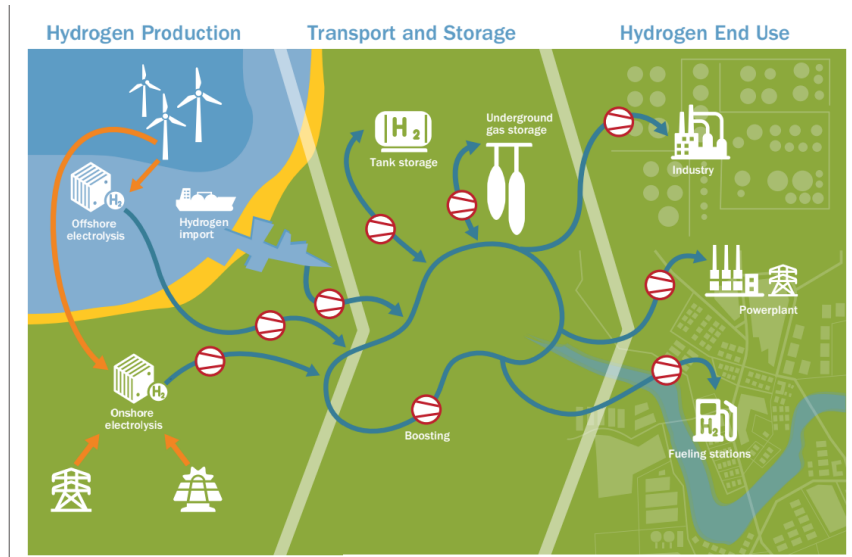
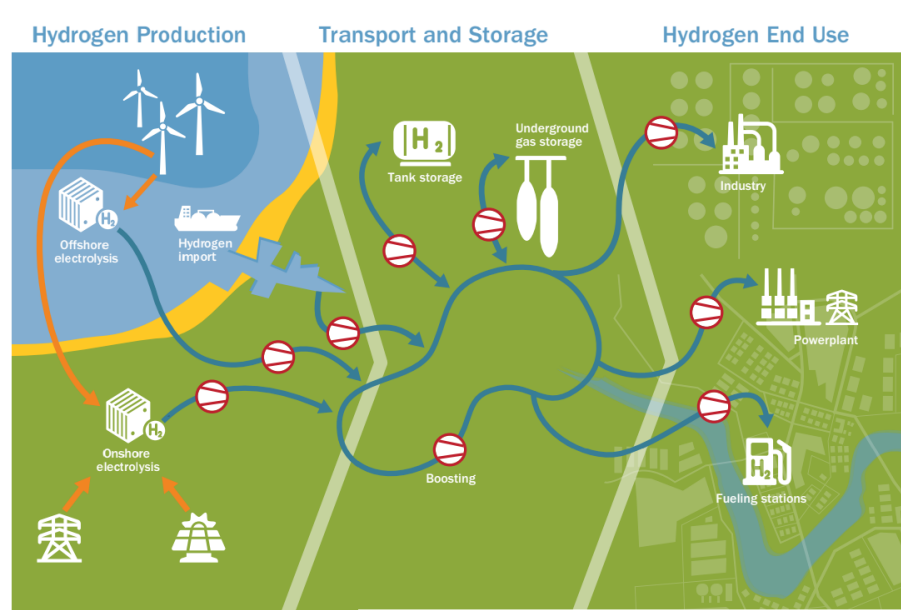


Figure 2. Typical applications where compression will be required in our future hydrogen economy, based on a hydrogen distribution grid.

## 2. Storage Hydrogen.

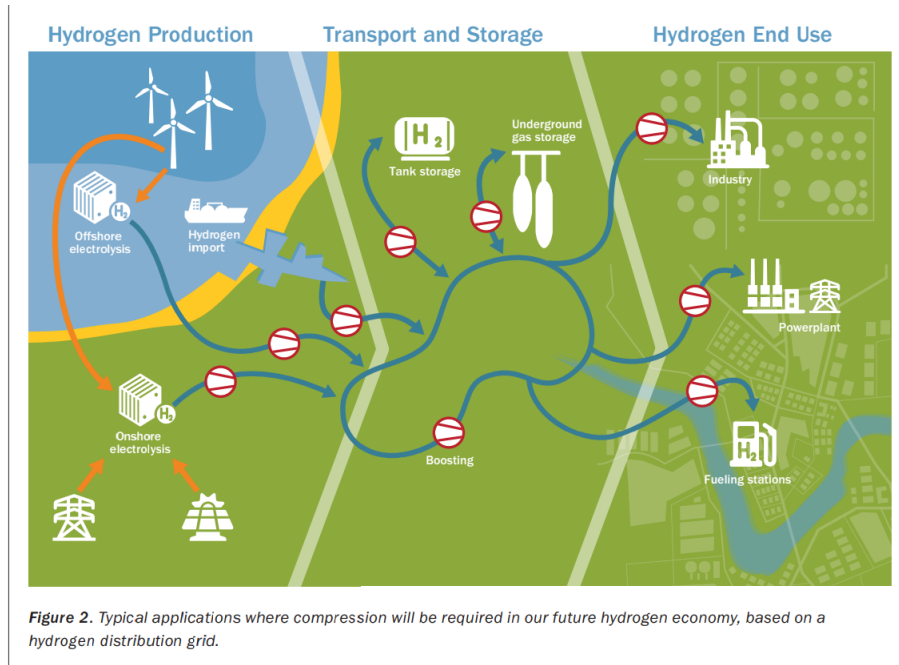
Storage is an important component in the hydrogen value chain to assure security of supply at periods of lower production. Tank storage sites are expected at different scales at several locations, both onshore and offshore. On a larger scale, salt caverns are currently prepared as underground hydrogen storage sites, to allow for larger capacities on longer time scales. **Typical discharge pressures are foreseen to go up to 200 bar for salt caverns**, whereas storage tanks may even be operated at higher discharge pressures.



*Figure 2. Typical applications where compression will be required in our future hydrogen economy, based on a hydrogen distribution grid.*

### 3. Fueling stations.

The discharge pressures, flow rates and number of starts and stops make the **compressors used at fueling stations a topic on its own**. Discharge (fueling) pressures are currently **350 bar** for large vehicles, to **700 bar** for personal vehicles. Although the use of hydrogen in passenger cars may be limited, the long distance heavy duty transport is a likely candidate to use hydrogen or a synthetic fuel as an energy source. Fueling station compressors are different from other storage compressors because of the requirements on high hydrogen purity, relatively small capacity and intermittent operation.



*Figure 2. Typical applications where compression will be required in our future hydrogen economy, based on a hydrogen distribution grid.*

#### 4. End user demand.

Whether it is hydrogen use for high temperature heating in industrial processes, for feedstock in chemical processes or the use in power generation, each end user will have its specific requirements to the capacity, pressures, temperature, purity etc. of the hydrogen it will process. Tailor made compression solutions will therefore be required at the end users. End users that are located close to a hydrogen distribution grid may not require compression, in case the grid pressure is always higher than the required pressure.

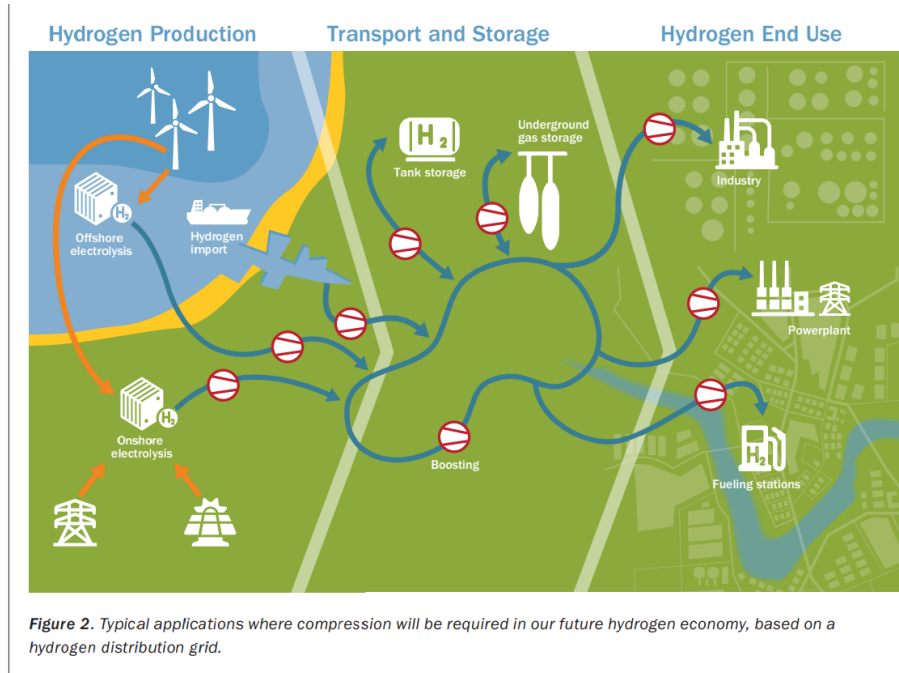
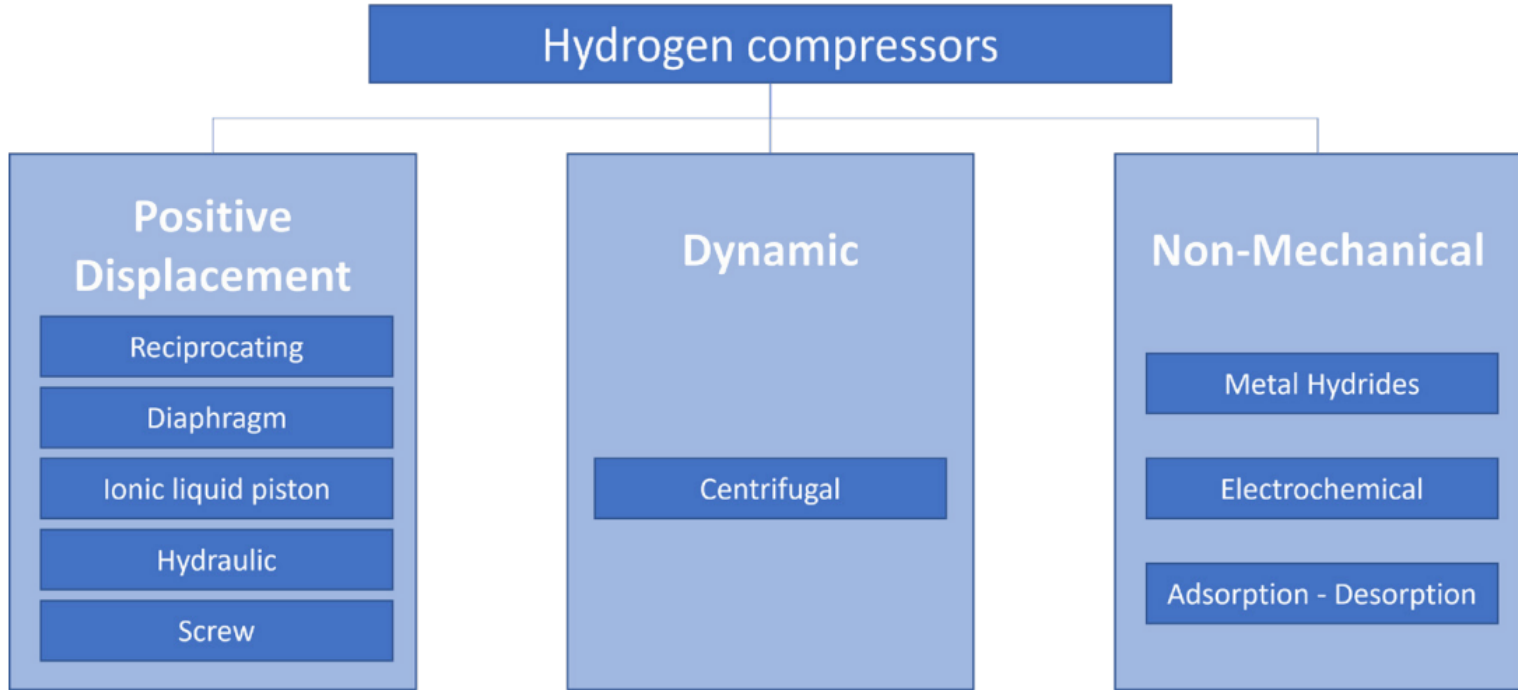


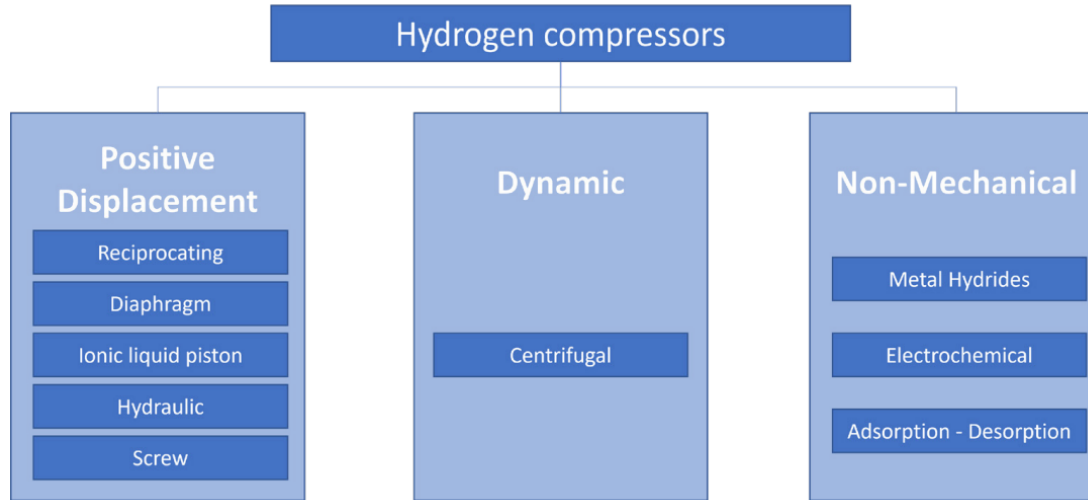
Figure 2. Typical applications where compression will be required in our future hydrogen economy, based on a hydrogen distribution grid.

# Compressors for Hydrogen applications



*Figure 3. Types of compressors.*

# Compressors for Hydrogen applications



– Positive displacement, a certain volume of gas is captured, compressed by a gradual reduction within a control volume and discharged at elevated pressure.

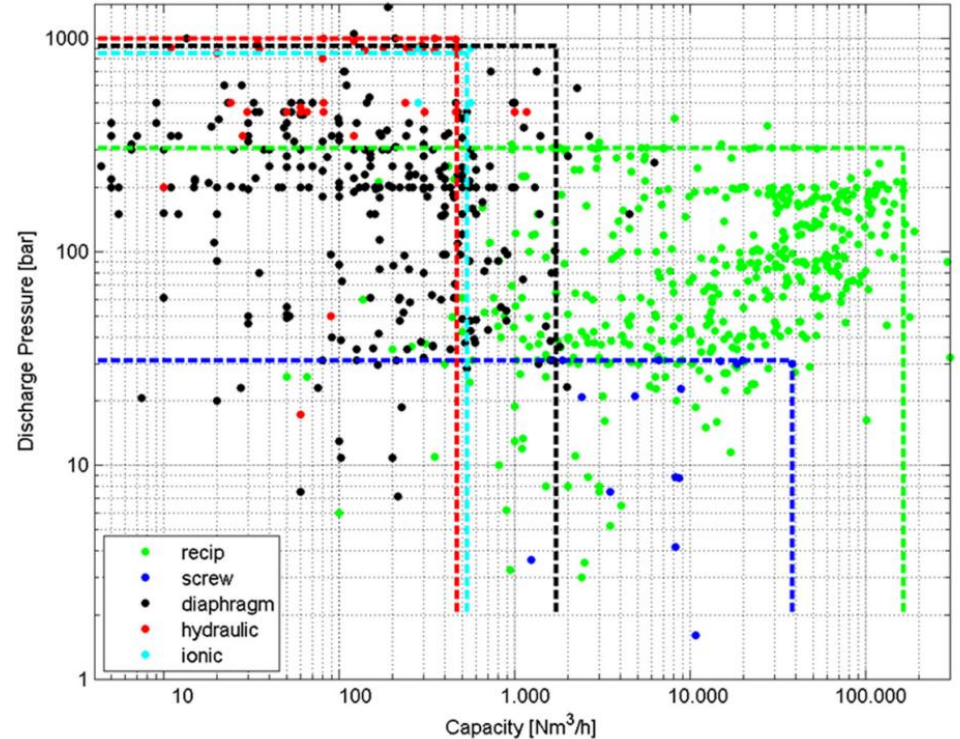
– Centrifugal, where kinetic energy (impeller velocity) is added to the incoming gas stream and, subsequently, converted to into static energy (pressure).

– Non-mechanical compression, using different principles such as electrochemistry or via adsorption (or absorption) of hydrogen into porous materials

# Compressors type

The dots in the map are actual industrial reference cases for compression of pure hydrogen

The major compression technology used for pure hydrogen applications is **positive displacement**



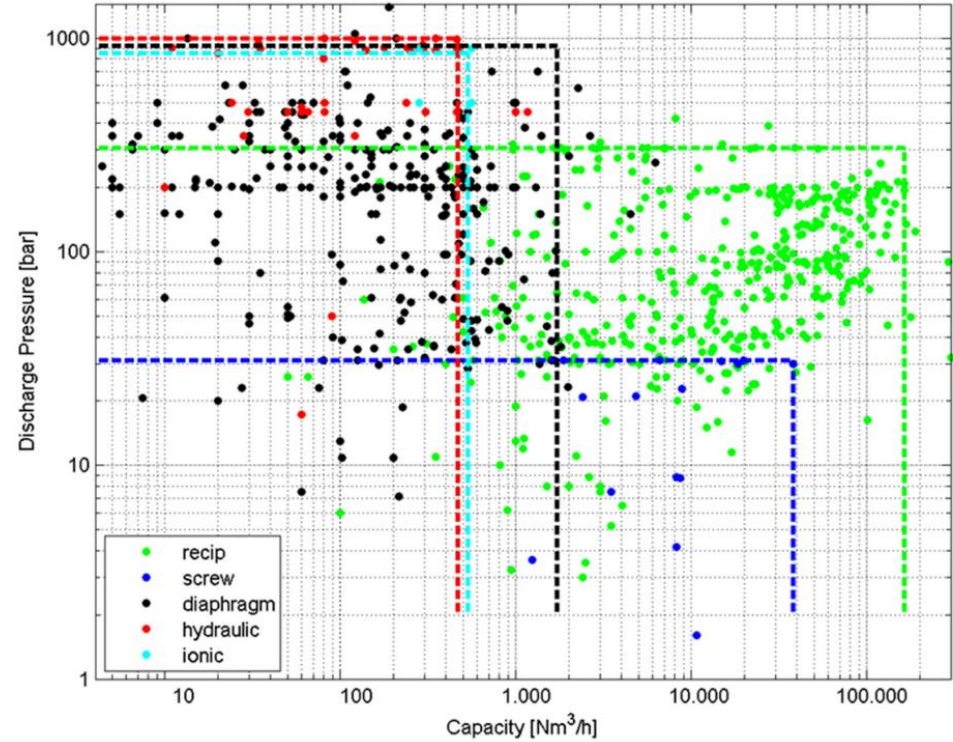
**Fig. 1.** Mechanical compressor map with industrial references (dots) and performance ranges (dotted lines) for hydrogen applications [6]. The capacity along the horizontal axis can be converted to mass flow rates according to  $1000 \text{ Nm}^3/\text{h} \cong 90 \text{ kg/h}$ .

The **reciprocating compressors (green dots)** span a large range of flows, up to a pressure of around 400 bar and include both lubricated and non-lubricated type.

The **diaphragm (black), hydraulic (red) and ionic (cyan)** compressors extend toward very high pressure (1000 bar), but have a limited capacity.

The **screw compressors (blue)** cover an intermediate range of capacity, but are limited to a discharge pressure of typically 30 bar.

Centrifugal compressors are absent in this plot, as no industrial references were found on pure hydrogen (active field of development in industry)



**Fig. 1.** Mechanical compressor map with industrial references (dots) and performance ranges (dotted lines) for hydrogen applications [6]. The capacity along the horizontal axis can be converted to mass flow rates according to  $1000 \text{ Nm}^3/\text{h} \cong 90 \text{ kg/h}$ .

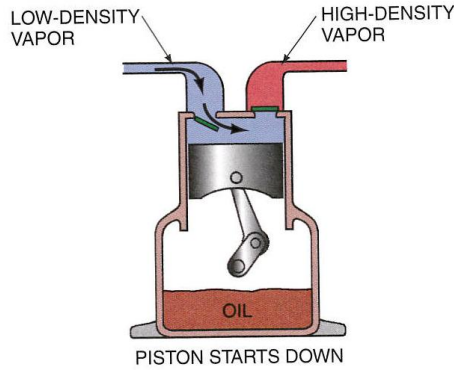
# Hydrogen purity

As regards the purity of the hydrogen to be processed, where the molar mass of the gas is considered to be less than 2.5 g/mol and, especially, the hydrogen is considered to be free of lubricating oil

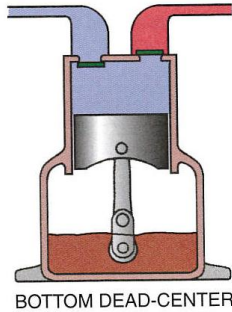
Depending on the application, the purity of the hydrogen may be a key aspect. Many reciprocating compressors use lubricated pistons, thereby increasing the maintenance intervals. The lubrication, however, introduces oil into the hydrogen stream. For applications where high-purity hydrogen is required (e.g. fuel cells and hydrogen liquefaction), non-lubricated reciprocating piston compressors must be used. The absence of lubrication may introduce extra wear and may increase running surface temperatures due to friction. Non-lube piston compressors have been used successfully up to 350 bar.

Higher outlet pressures require lubricated piston compressors which may then require additional equipment in the form of oil separation and filtering

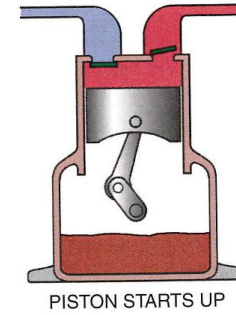
# Reciprocating Compressors Fundamental



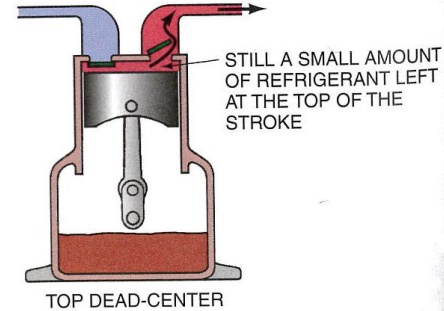
**Figure 23-42** An illustration of what happens inside the reciprocating compressor while it is pumping. When the piston starts down, a low pressure is formed under the suction reed valve. When this pressure becomes less than the suction pressure and the valve spring tension, the cylinder will begin to fill. Gas will rush into the cylinder through the suction reed valve.



**Figure 23-43** When the piston gets near the bottom of the stroke, the cylinder is nearly as full as it is going to get. There is a short time lag as the crankshaft circles through bottom dead-center, during which a small amount of gas can still flow into the cylinder.



**Figure 23-44** When the piston starts back up and gets just off the bottom of the cylinder, the suction valve will have closed, and pressure will begin to build in the cylinder. When the piston gets close to the top of the cylinder, the pressure will start to approach the pressure in the discharge line. When the pressure inside the cylinder is greater than the pressure on the top side of the discharge reed valve, the valve will open, and the discharge gas will empty out into the high side of the system.



**Figure 23-45** A reciprocating compressor cylinder cannot completely empty because of the clearance volume at the top of the cylinder. The manufacturers try to keep this clearance volume to a minimum but cannot completely do away with it.

# Reciprocating compressors: effect of dead space

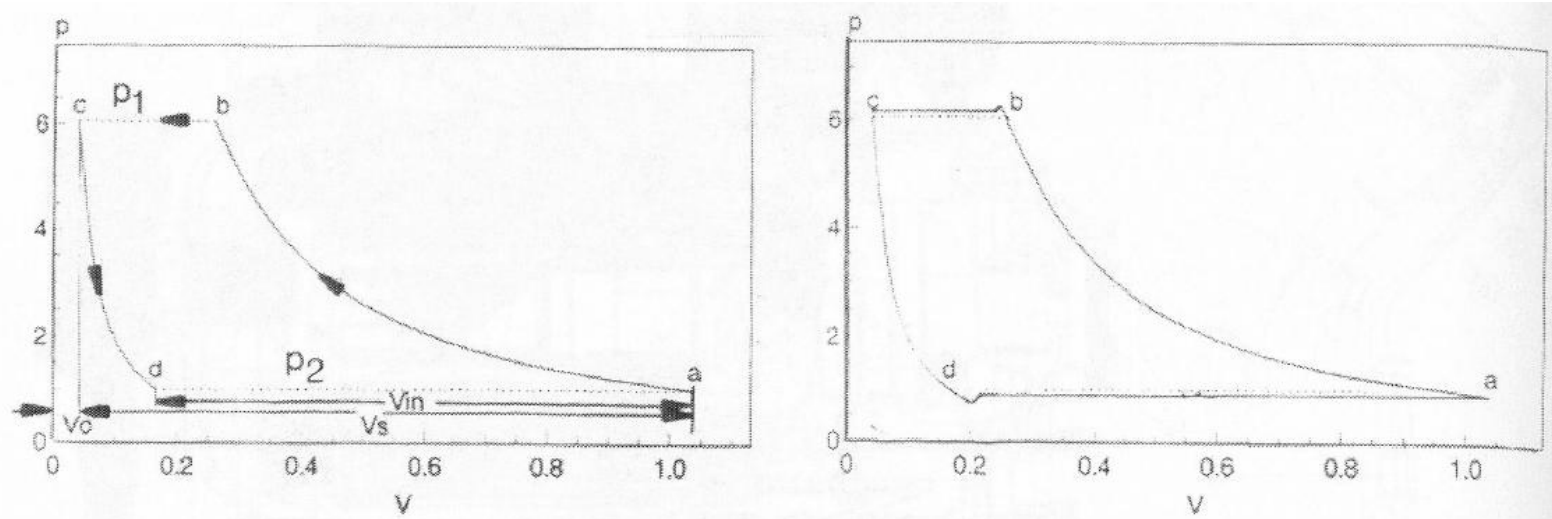


Figure 7.17. Pressures in the cylinder of a reciprocating compressor. The left diagram indicates an idealised cycle, while the influence of pressure losses in the suction and discharge valves are indicated in the diagram to the right.

$$\text{swept cylinder volume } V_s = V_a - V_o$$

## Reciprocating compressors: effect of dead space

$$\eta_{s,th} = \frac{V_{in}}{V_s} = \frac{(V_a - V_d)}{V_s}$$

where  $V_a = V_s + V_o$

For the idealized case the process c  $\rightarrow$  d is isentropic ( $p \cdot V^\kappa = \text{const}$ )

$$V_d = V_o \cdot \left( \frac{p_1}{p_2} \right)^{1/\kappa}$$

$$\eta_{s,th} = 1 - \frac{V_o}{V_s} \cdot \left( \left( \frac{p_1}{p_2} \right)^{1/\kappa} - 1 \right)$$

# Reciprocating compressors: effect of dead space

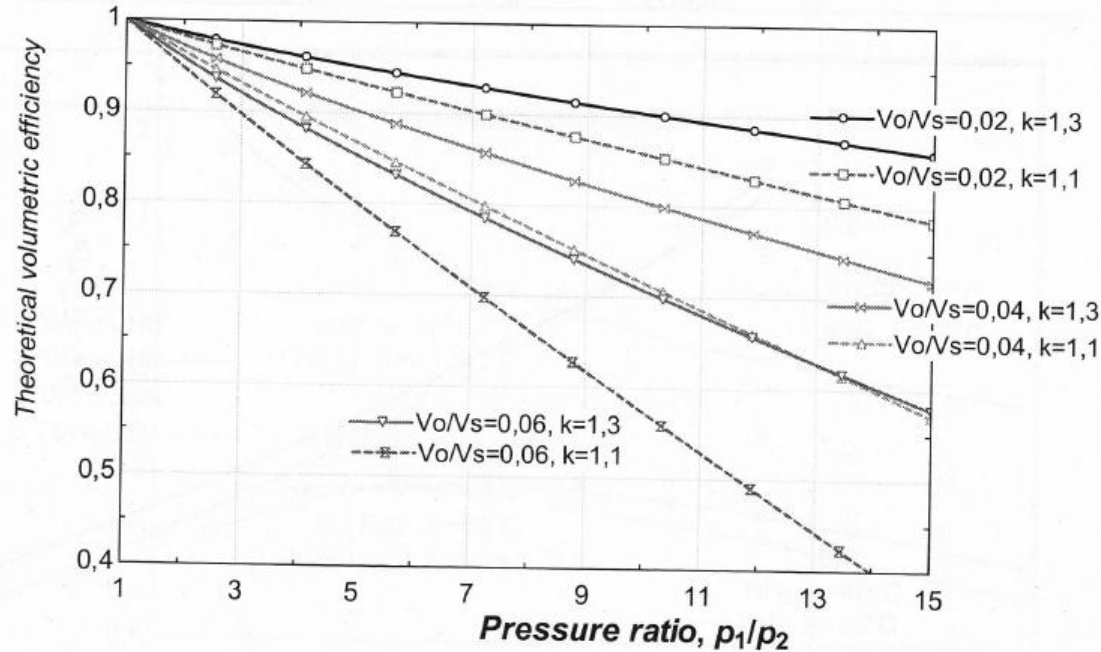
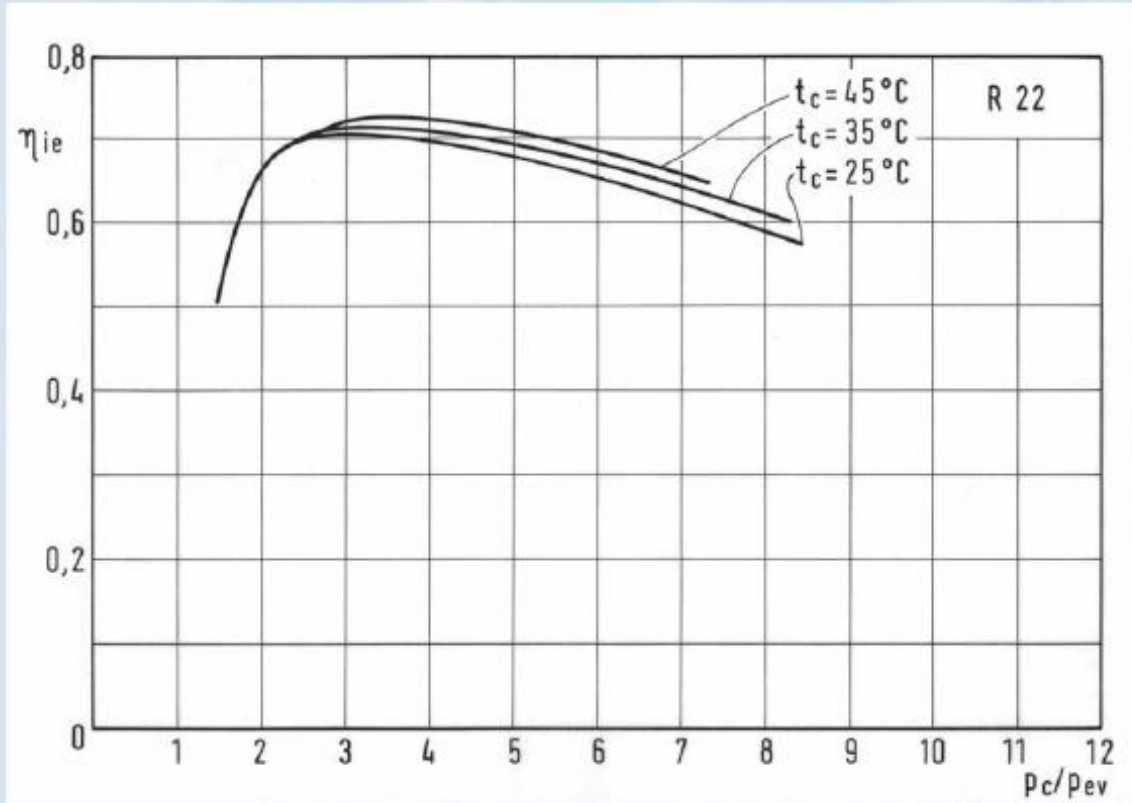


Figure 7.18 Influence of the dead space on the theoretical volumetric efficiency for reciprocating compressors.

# ISENTROPIC EFFICIENCY FOR OPEN COMPRESSORS



# Reciprocating compressors: The effect of irreversibilities on the compressor efficiency

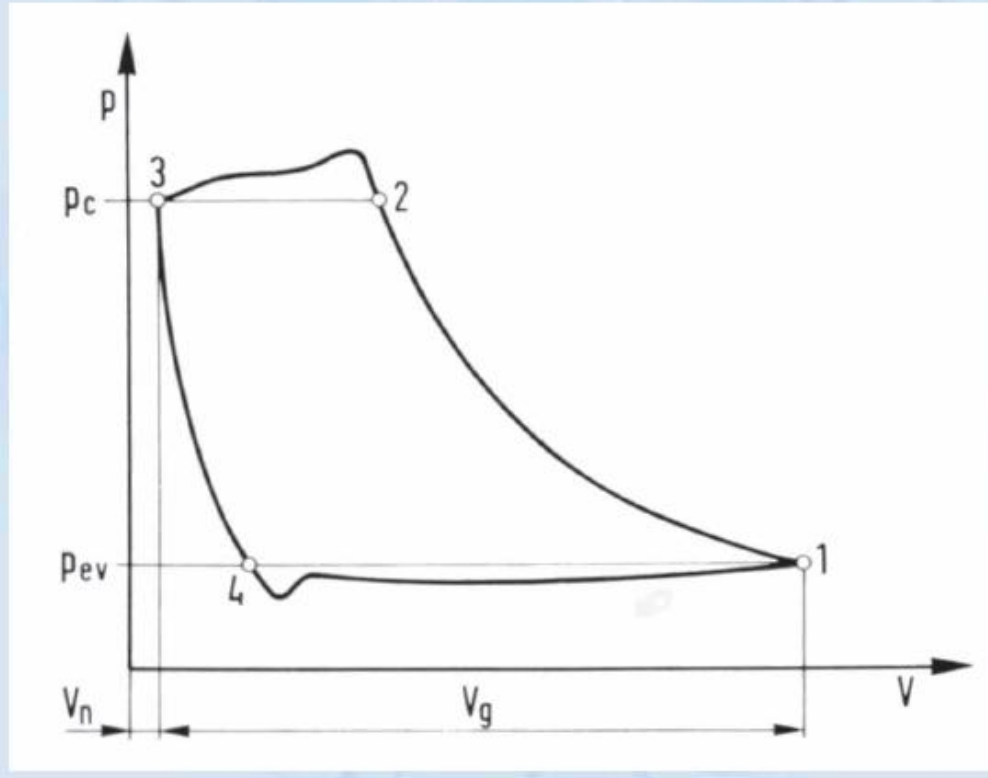
There are four types of losses that can penalize the compression efficiency:

1. Fluid friction
2. Mechanical friction
3. Heat transfer
4. Sealing defects

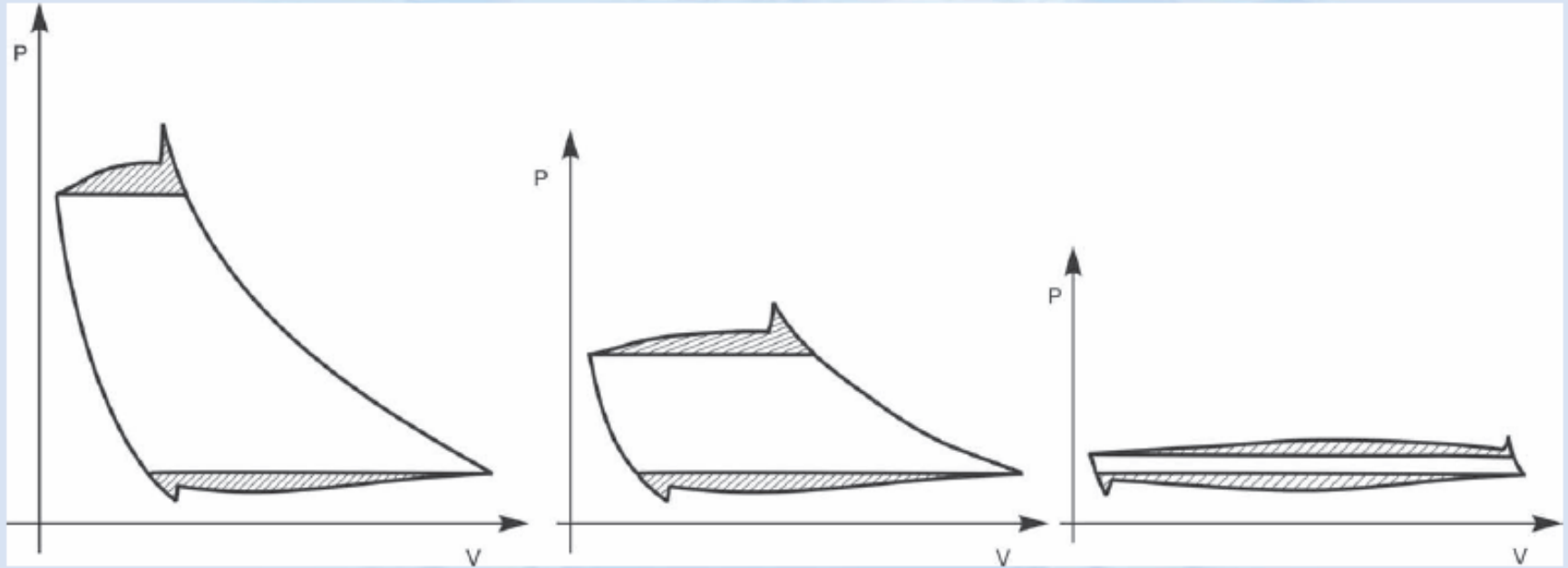
*For the case of reciprocating compressors, the effect of sealing defects is not significant, both for the valves and the couple cilinder-piston.*

*Therefore, the losses are due to the other three reasons*

# INDICATOR DIAGRAM OF A COMPRESSION CYCLE



# INDICATOR DIAGRAMS WHEN VARYING PRESSURE RATIO



## Indicator diagram and work losses

The isentropic efficiency is the ratio of ideal compression work to actual real work.

The real work can be seen as the sum of the ideal work (white area) plus the losses due to the three causes.

The indicator diagram shows the losses when the fluid goes through the valves (not the losses due to heat transfer or due....). There is a local pressure drop, proportional to the square of fluid velocity. This fluid velocity depends on the linear velocity of the piston (zero at top and bottom positions, maximum in the middle).

## Reciprocating Compressors: gas velocity through valves

The *velocity of the gas when flowing through the valves* is influenced by the free area of the valve ports (valve bore)  $A_b$ , the cylinder area ( $A_C = \pi D^2/4$ ) and the speed by which the piston moves in the cylinder. For design purposes a characteristic superficial average gas velocity,  $w_{g,ave}$ , can be defined by using the average piston velocity ( $w_{p,ave}$ ) as

$$w_{g,ave} = w_{p,ave} \cdot A_C/A_b \quad w_{p,ave} = 2 \cdot S \cdot n/60$$

The pressure drop is directly proportional to the density of the gas (for qualitative comparisons use  $\Delta p = \zeta \cdot \rho_g \cdot w_g^2 / 2$ ).

## Indicator diagram and work losses

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Therefore the pressure loss at the valves has a minimum effect on the volumetric efficiency because in the bottom position the gas in the cylinder is almost at the equilibrium with the evaporating pressure. On the contrary, such pressure loss at the valve can be important for the energy efficiency, above all when the pressure ratio gets close to one because the ideal work is very small and thus losses are relatively important.

The work lost in the valves due to fluid friction per unit of mass does not depend on the fluid density but only on its velocity.

# Reciprocating Compressors: gas velocity through valves

The *velocity of the gas when flowing through the valves* is influenced by the free area of the valve ports (valve bore)  $A_b$ , the cylinder area ( $A_C = \pi D^2/4$ ) and the speed by which the piston moves in the cylinder. For design purposes a characteristic superficial average gas velocity,  $w_{g,ave}$ , can be defined by using the average piston velocity ( $w_{p,ave}$ ) as

$$w_{g,ave} = w_{p,ave} \cdot A_C/A_b \qquad w_{p,ave} = 2 \cdot S \cdot n/60$$

Recommendations for this superficial design gas velocity are given with different values in different sources in the literature. In general speed up to  $w_{g,ave} \cong 60$  m/s seems to have been used successfully in designs for ammonia. For heavier gases (like R22) lower speeds are preferred, up to 45 m/s.

Pierre (1982) refers to an investigation giving the following results in conclusion *for ammonia*:

*Suction valve:*

Increase of  $w_{g,ave}$  from 18 to 66 m/s: decreased:  $\eta_S$  by 0,3 % and  $\eta_k$  by 3,3 %

*Discharge valve:*

Increase of  $w_{g,ave}$  from 25 to 171 (!) m/s: decreased:  $\eta_S$  by 1 % and  $\eta_k$  by 2,4 %

## Reciprocating Compressors: gas velocity through valves

The pressure drop is directly proportional to the density of the gas (for qualitative comparisons use  $\Delta p = \zeta \cdot \rho_g \cdot w_g^2 / 2$ ). This means that the valves will be more difficult to design with low pressure drops for gases with high density  $\rho_g$  (or large molecular weight, m). For a comparison at *the same pressure* the permissible speed for equal pressure drop will be  $w_g = \text{constant} / \sqrt{\rho_g}$  and hence also proportional to  $1 / \sqrt{M}$ .

Consider the molecular weights for a few refrigerants:

Ammonia:	$M_{R717} = 17$	kg/kmol	R404A	$M_{R404A} = 97,6$	kg/kmol
propane	$M_{R290} = 44,1$	kg/kmol	R407C	$M_{R407C} = 86,2$	kg/kmol
R22:	$M_{R22} = 86,5$	kg/kmol	R410A	$M_{R410A} = 72,6$	kg/kmol
R134a:	$M_{R134a} = 102$	kg/kmol			

For example it could be expected that the a permissible gas velocity using R134a compared to ammonia would correspond to the ratio  $\sqrt{17 / 102} \cong 0,41$  or  $w_{gR134a} \cong 0,41 \cdot w_{gR717}$ . With less than half the velocity this means that the valve areas for R134a should be *more than twice* the area necessary in an ammonia compressor for equal pressure drop. Similar relations apply for other HFC and HCFC-refrigerants.

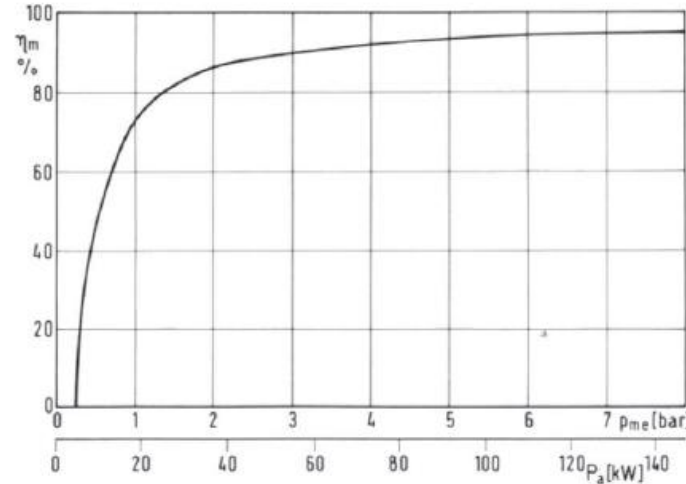
Also the size of the compressor has an influence. In small capacity compressors for the heavier refrigerants the superficial average gas velocity,  $w_{g,ave}$ , often is in the order of 15 m/s for suction, as well as discharge, valves.

# Mechanical efficiency of reciprocating compressors

Losses due to mechanical friction are additional work that must be spent, in addition to the work of the indicator diagram.

The mechanical efficiency can be defined as the ratio of indicator work to actual spent work.

For most operating conditions the mechanical efficiency can be considered as constant.



**Mechanical efficiency versus compressor power**

## Indicator diagram and heat transfer losses

There is heat transfer between the suction fluid and the walls in the cylinder: this is the most important for the heat transfer losses.

This heat transfer does not change significantly the indicator diagram, and therefore the indicator work remains the same. Instead, it affects the **isentropic efficiency and volumetric efficiency** because the gas is heated up when it enters the compressor and thus its density decreases (specific volume increases) which means that the compressed mass is reduced but the work remains unchanged.

This loss is difficult to calculate but it increases with the pressure ratio.

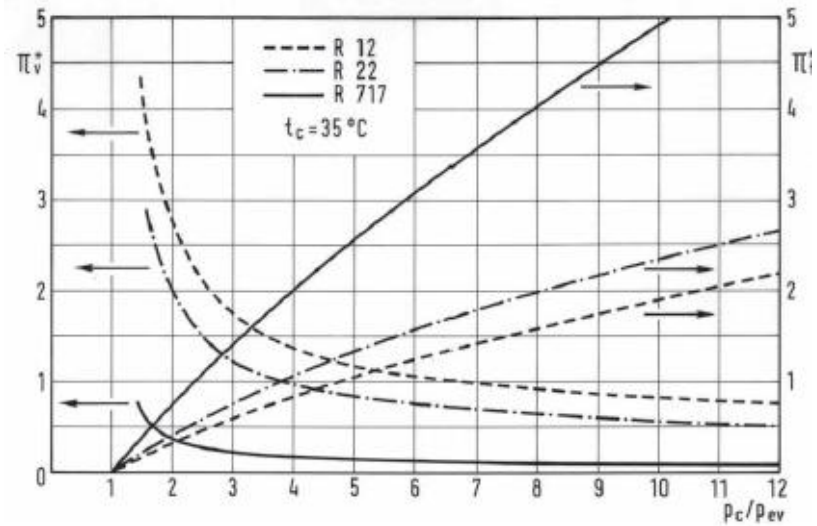
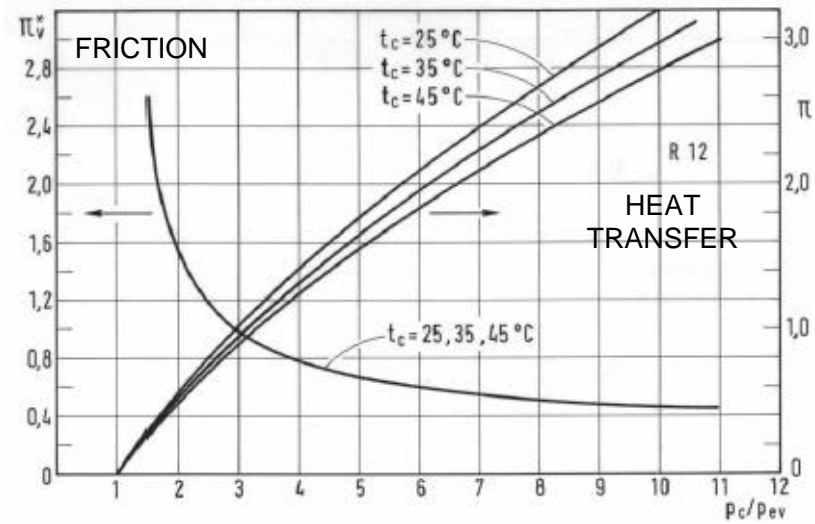
This heat transfer strongly affects the volumetric efficiency making it lower than the theoretical volumetric efficiency.

The importance of these losses depend on the fluid: for example, with ammonia low fluid friction losses in the valves but high heat transfer losses.

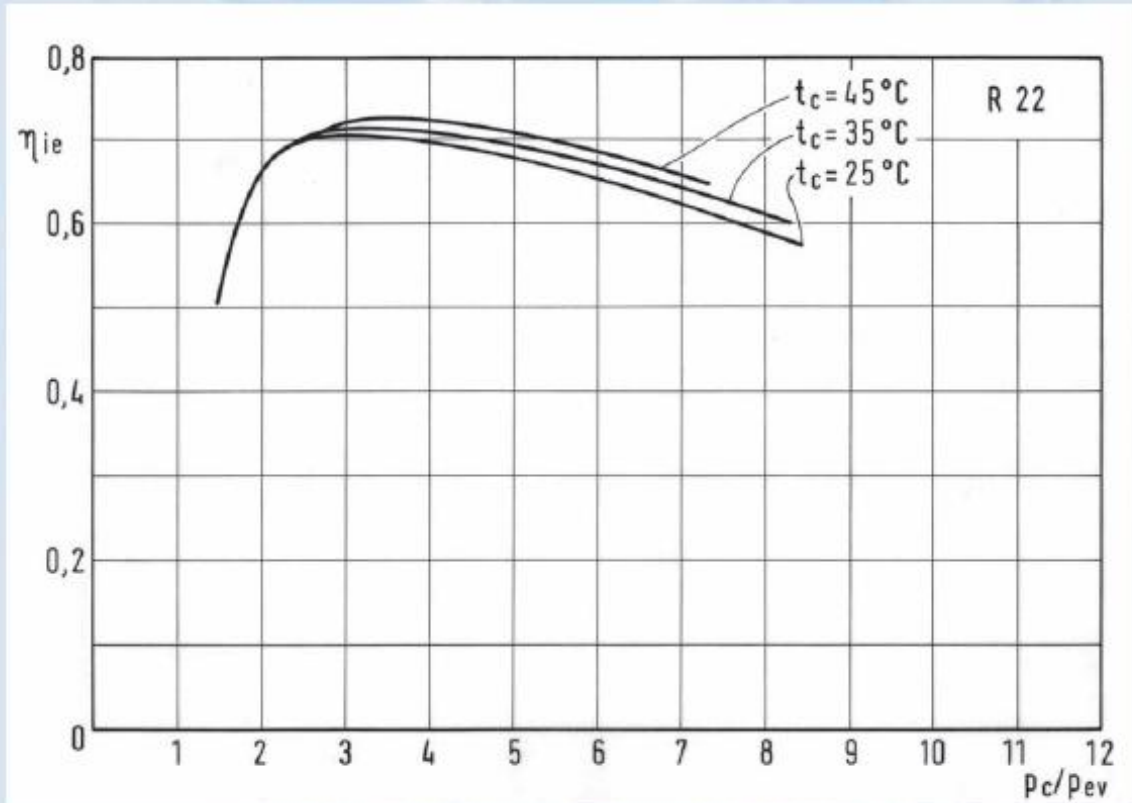
For the design:

- reducing the rotation speed gives an advantage for the fluid friction losses (more in synthetic fluids)
- high ratio stroke/bore increase friction losses and decrease heat transfer losses

# LOSSES FOR FLUID FRICTION AND HEAT TRANSFER



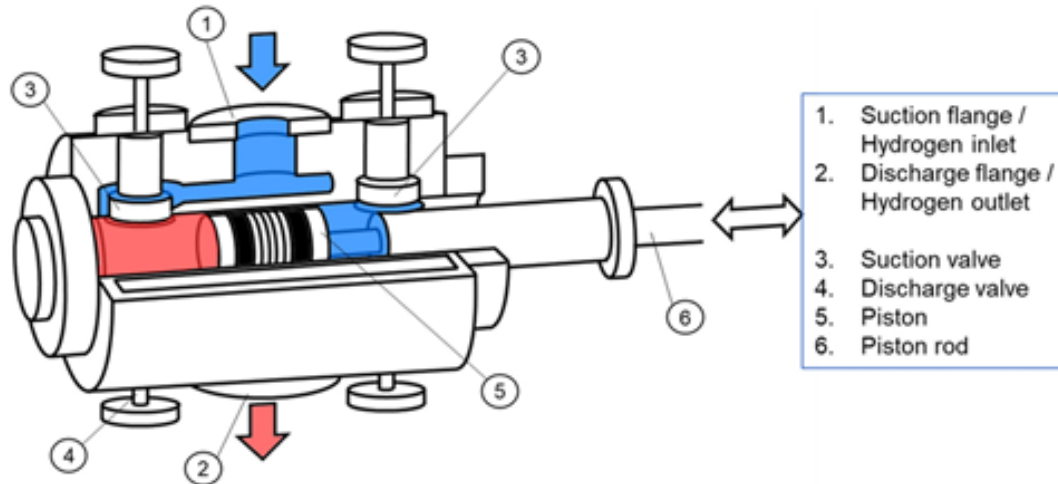
# ISENTROPIC EFFICIENCY FOR OPEN COMPRESSORS



# Reciprocating compressors for hydrogen applications

It has a large range in capacity (up to 200.000 Nm<sup>3</sup>/h) and discharge pressure (in excess of 400 bar for lubricated compressors, up to 225 bar for non-lubricated compressors).

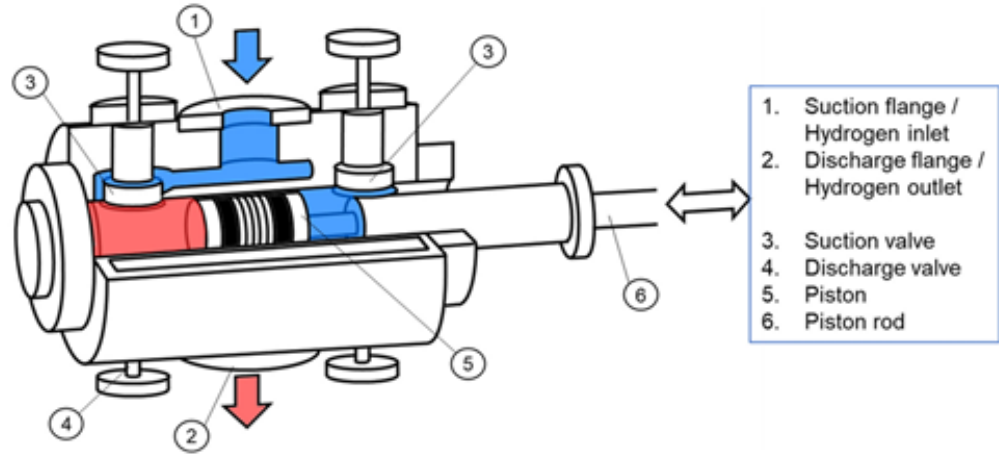
The typical pressure ratio per stage is 1.6-2.5. In case a large total compression ratio is required, this is achieved by introducing multiple stages, typically up to 4 depending on the process. A schematic overview of the elements in a reciprocating compressor cylinder is presented in the Figure.



As a consequence of the operating principle, reciprocating compressors have a large ability to adapt to varying process conditions such as flowrate or gas composition changes.

In terms of capacity control, various proven techniques exist for reciprocating compressors, such as speed control, bypass lines, suction valve throttling, suction valve unloading, clearance pockets and stepless reverse flow capacity control.

The most common (proven) techniques for hydrogen compression are **suction valve unloading** and **stepless reverse flow control**.

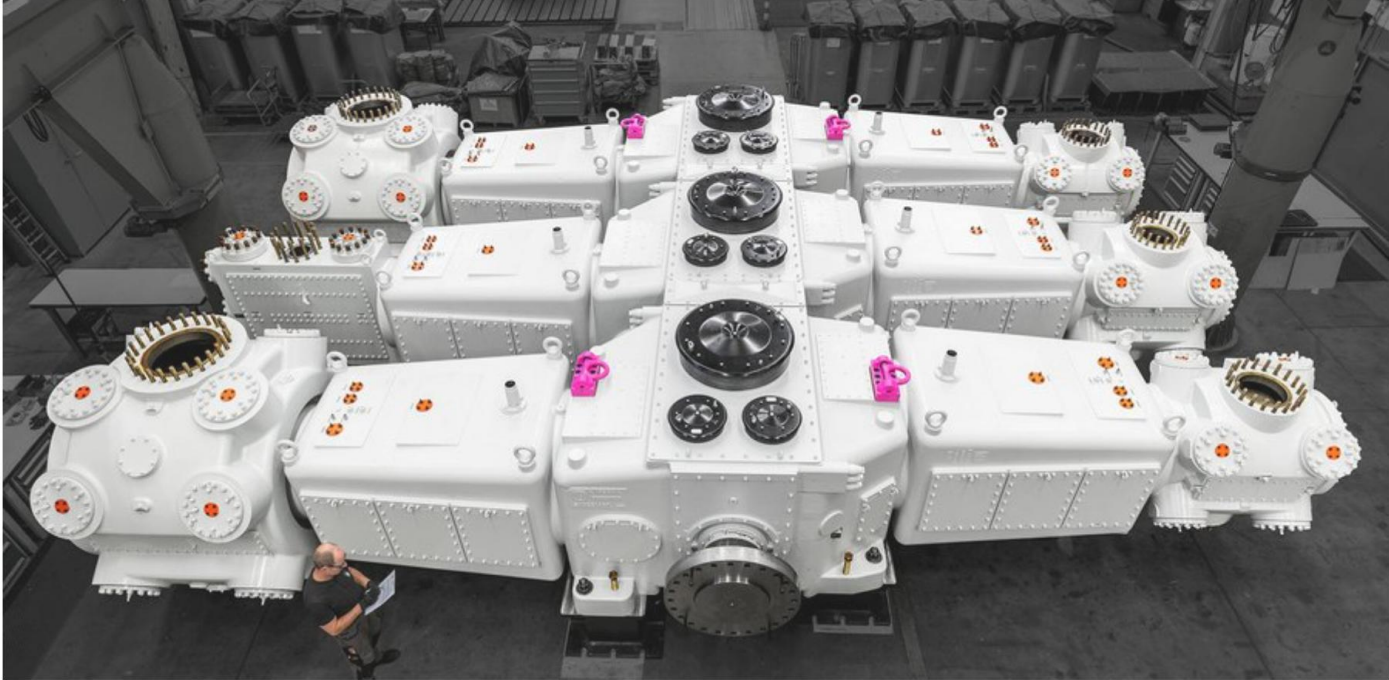


As a consequence of the operating principle, reciprocating compressors have a large ability to adapt to varying process conditions such as flowrate or gas composition changes.

The gas composition and corresponding molecular weight of hydrogen gas is very sensitive to even small percentages of contamination.

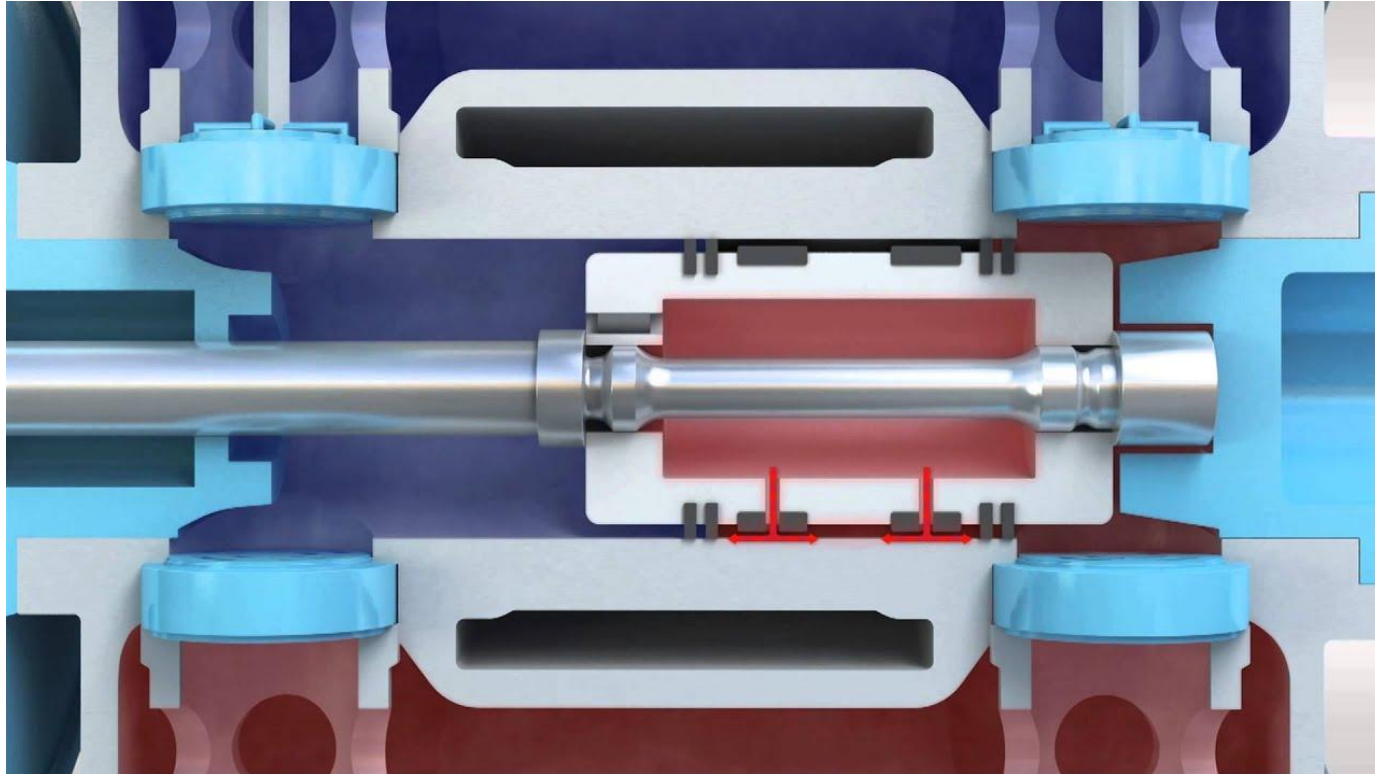
By the nature of its design, the reciprocating compressor has wearing parts. Most obvious are the compressor valves, but also pressure packings (piston rod), piston rings and rider rings (wear bands) are subject to wear. With adequate design and operation, replacement of wearing parts can be scheduled within the overall maintenance strategy. Intermittent operation may introduce extra challenges on wearing parts.

Because hydrogen causes embrittlement when in contact with metals over time at high pressures and temperatures, material selection is critical. Suitable materials include special steel, alloy steel, cast iron, and forged steel. Many reciprocating compressors employ inner cylinder linings or coatings that provide multiple advantages: they reduce maintenance costs by allowing coating replacement rather than component replacement, enable cylinder diameter adjustment for operational flexibility, and minimize hydrogen embrittlement effects

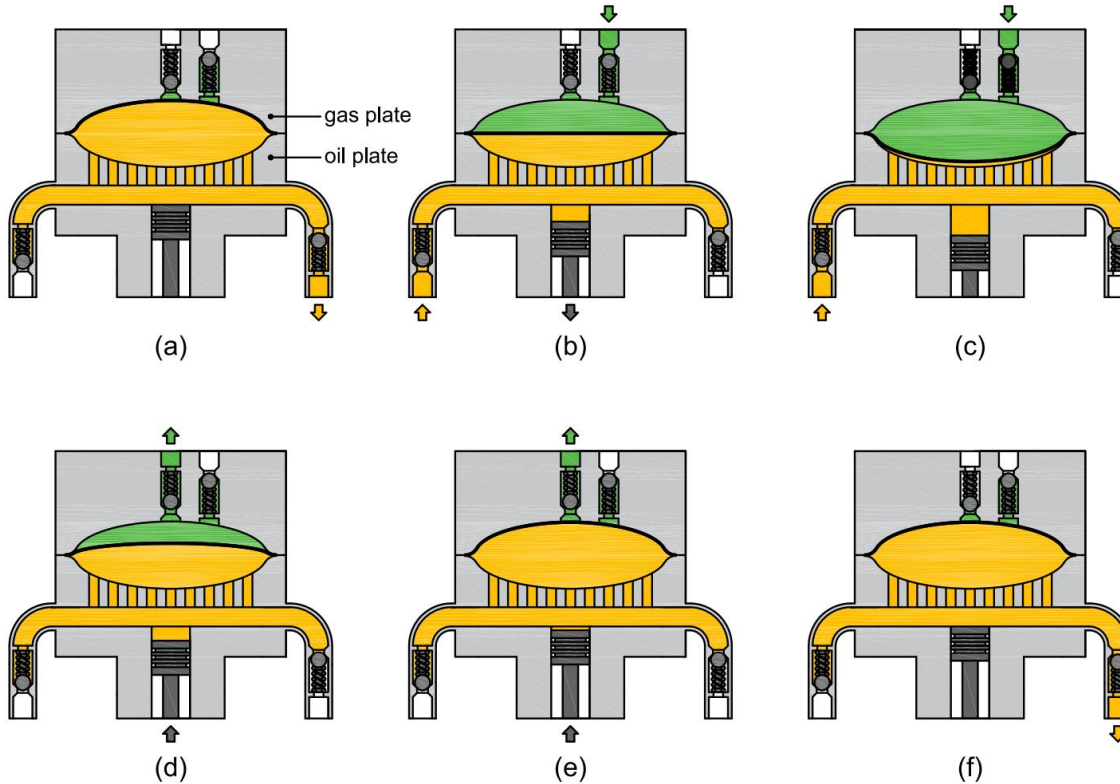


<https://www.neuman-esser.com/en/news-media/magazine/hydrogen-transport-with-pipelines/>

# Free floating piston

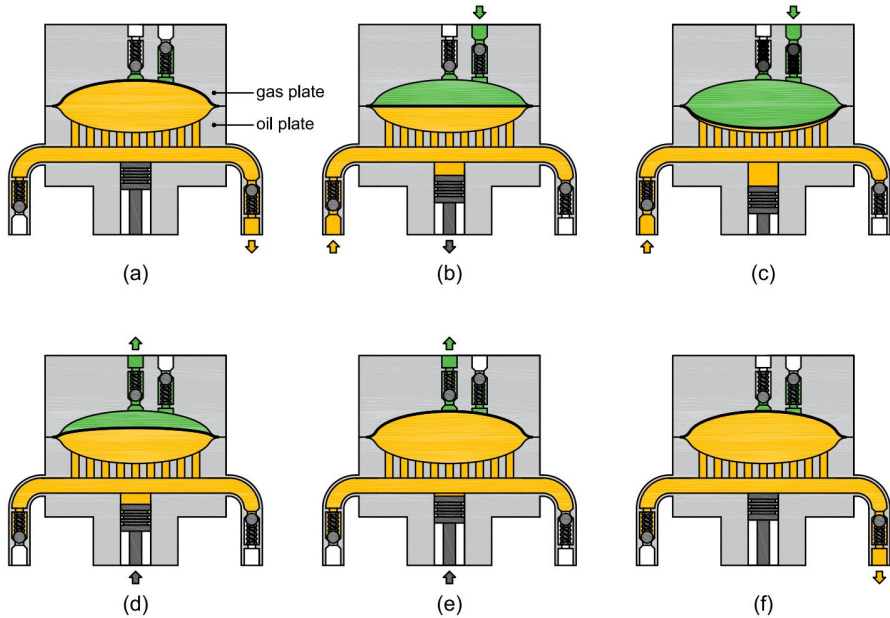


# Diaphragm compressors



Schematic of a diaphragm compressor with representation of the working cycle: orange is used for hydraulic fluid and green for hydrogen (or any other gas).

The diaphragm (in black) separates the gas side from the hydraulic side.



- The piston is at its top dead center (TDC) and the diaphragm is in contact with the gas plate;
- both the piston and the diaphragm are moving downward, with gas entering the system;
- the piston is at its bottom dead center (BDC) but the diaphragm must not reach the oil plate;
- both the piston and the diaphragm are moving upward, with pressurized gas exiting the system;
- the piston has not yet reach its TDC but the diaphragm is in contact with the gas plate;
- the piston has finally reached its TDC and excess oil is throttled through the hydraulic pressure limiter.

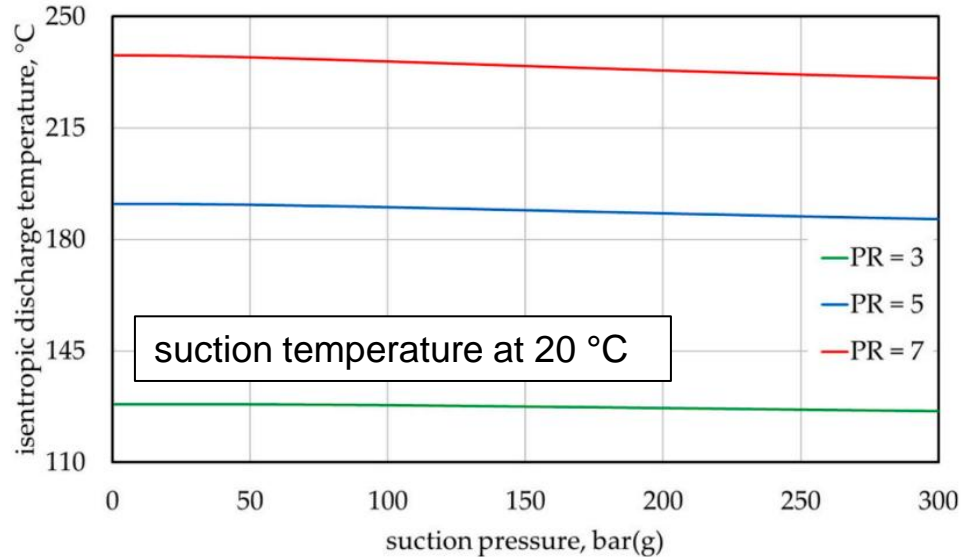


The diaphragm is composed of a metallic sandwich-type set of sheets.

Actually, the most common failure mechanism of the compressor is identified as diaphragm fracture due to low tensile strength and poor surface properties

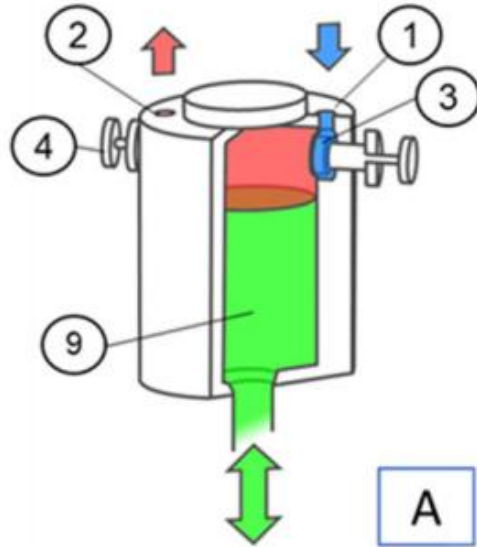
# Thermal problem: poor heat dissipation of the gas head cover

The gas head cover of the compressor is realized with a hydrogen embrittlement-resistant material, which has low thermal conductivity, with limited release of compression heat. Thus, the head cover can get seriously hot with significant thermal stress.



As an example, the temperature of the outer surface of the gas head cover can reach 160 °C when the discharge temperature is as high as around 250 °C, as the result of compressing hydrogen from 5 to 45 MPa in a single stage

# Ionic liquid compressor



- Compressor types:
- A. Ionic Liquid Piston
  - B. Diaphragm
  - C. Hydraulic
- 
- 1. Suction flange / Hydrogen inlet
  - 2. Discharge flange / Hydrogen outlet
  - 3. Suction valve
  - 4. Discharge valve
  - 5. Piston
  - 6. Piston rod
  - 7. Hydraulic Oil
  - 8. Membrane
  - 9. Ionic liquid

In case of the ionic liquid compressor, the hydrogen is compressed and expanded by pumping the ionic liquid in and out of the compression chamber volume. The ionic liquid has a low solubility which eliminates the necessity of separation material between the hydrogen and the liquid (membrane or piston), which also increases cooling capacity and efficiency

# Comparison and comments

The capacity of these (diaphragm, ionic, hydraulic) compressors is generally lower than the capacity of reciprocating compressors.

This is caused by the limited swept volumes (diaphragm, ionic) and the low running speed (hydraulic).

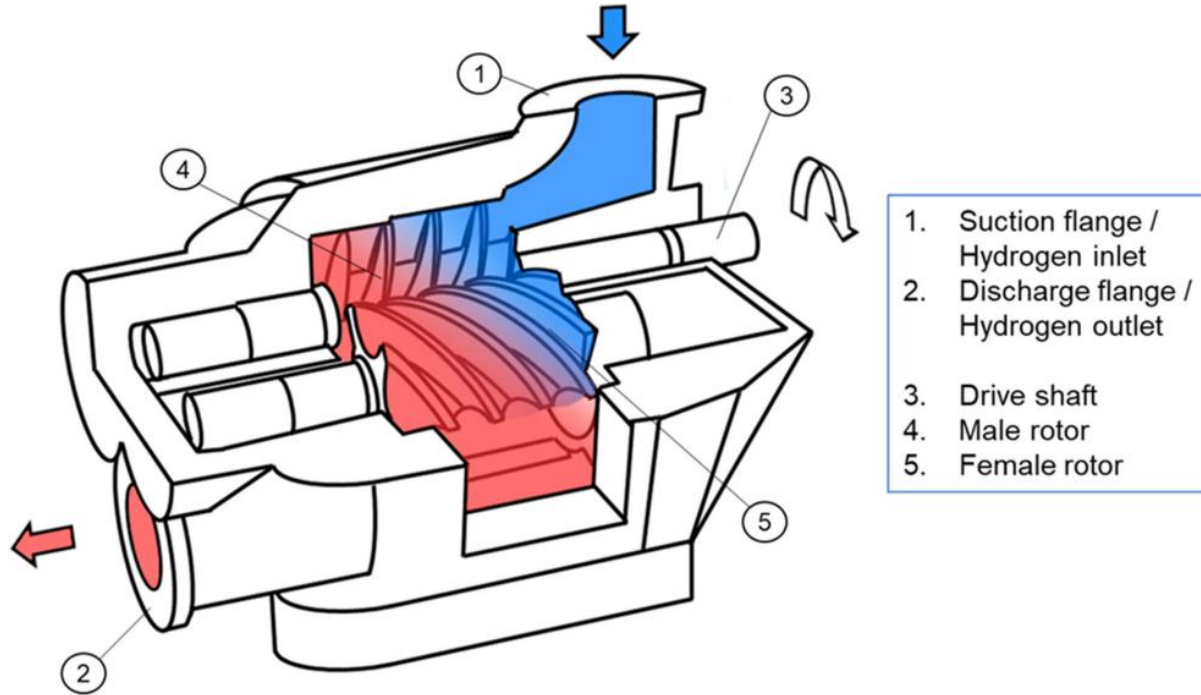
In (pure) hydrogen service, reported capacities range up to 1000 Nm<sup>3</sup>/h, with discharge pressures ranging up to 1000 bar.

Due to the minimal clearances, the favorable cooling capacity and higher allowed discharge temperatures for diaphragm compressors, the pressure ratio per stage in diaphragm compressors is generally higher (2.5-5.5) than for reciprocating compressors. This enables a design with less stages.

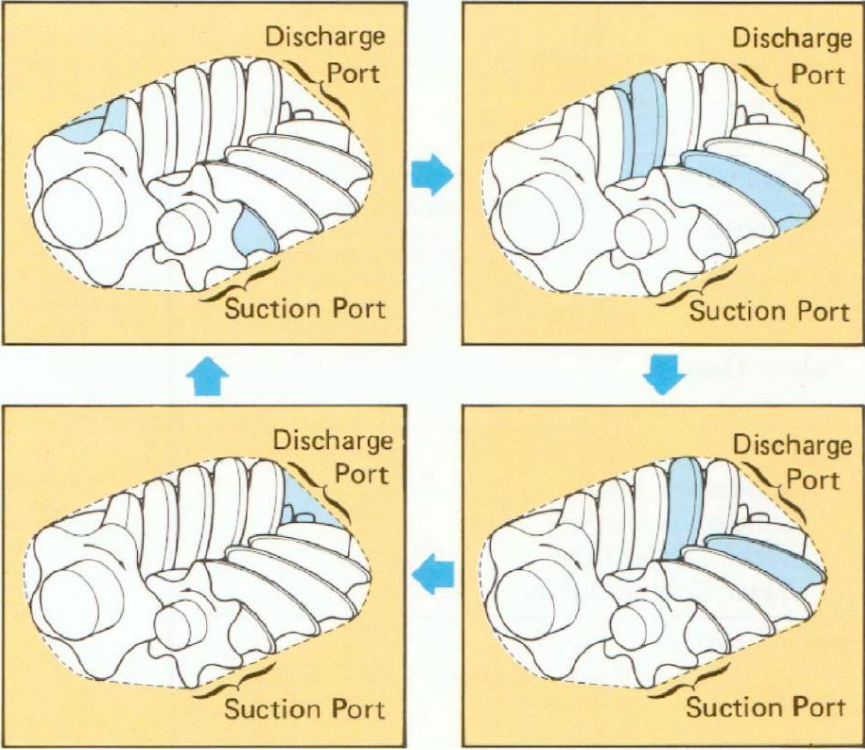
Hydraulic compressors are generally run at low running speeds, to maintain integrity of piston seals and hydraulic seals. Thus, the capacity and efficiency is generally less favorable than for the diaphragm compressor.

The maturity of these compressors is considered high, with many examples running in hydrogen compression systems (in particular for the mobility sector).

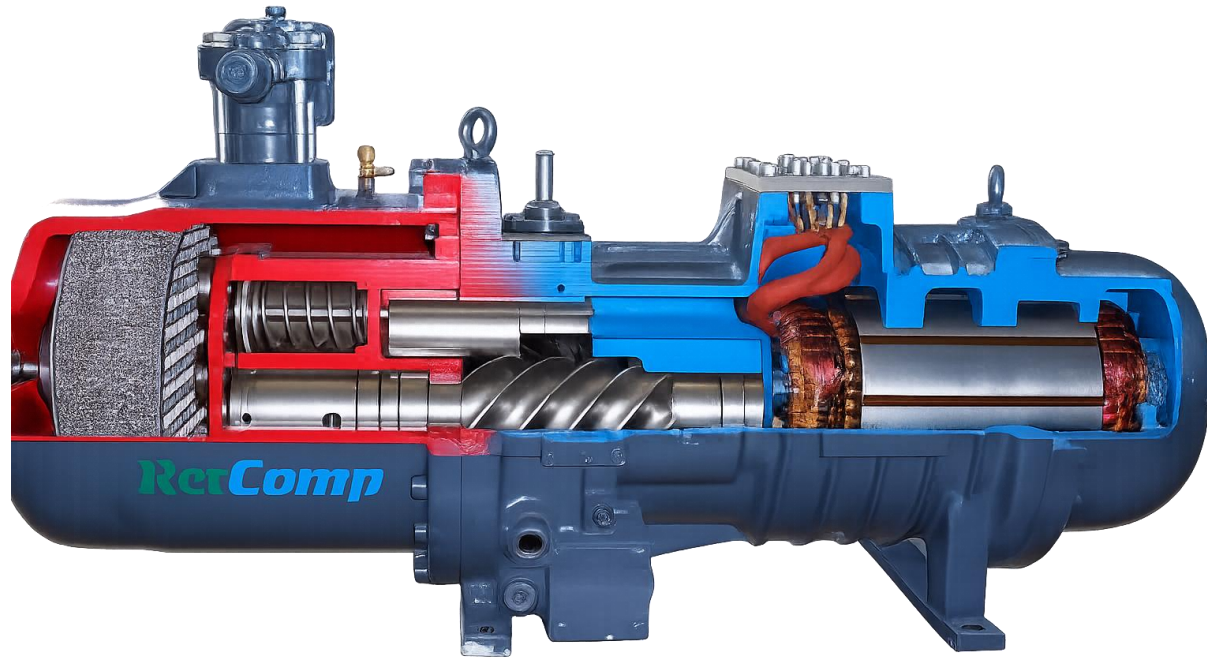
# Screw compressors



# Screw compressors



# Screw compressors



# Screw compressors

Built-in volume ratio

*Rapporto volumetrico intrinseco*

$$v_i = V_S/V_i$$

volume of gas at inlet ( $V_S$ )

volume when the discharge port opens, ( $V_i$ ).

# Screw compressors: Mismatch losses

$$V_i = \frac{V_{in}}{V_{out}}$$

3 values of  
BUILT-IN volume ratio

$$V_{i1} > V_i > V_{i2}$$

$$V_{out,i1} \quad V_{out,i} \quad V_{out,i2}$$

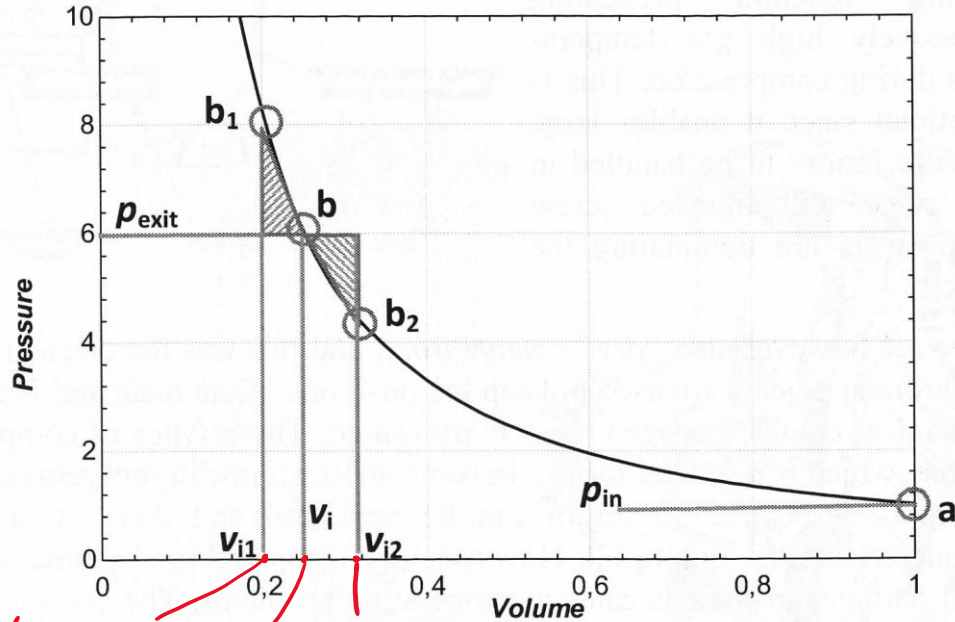


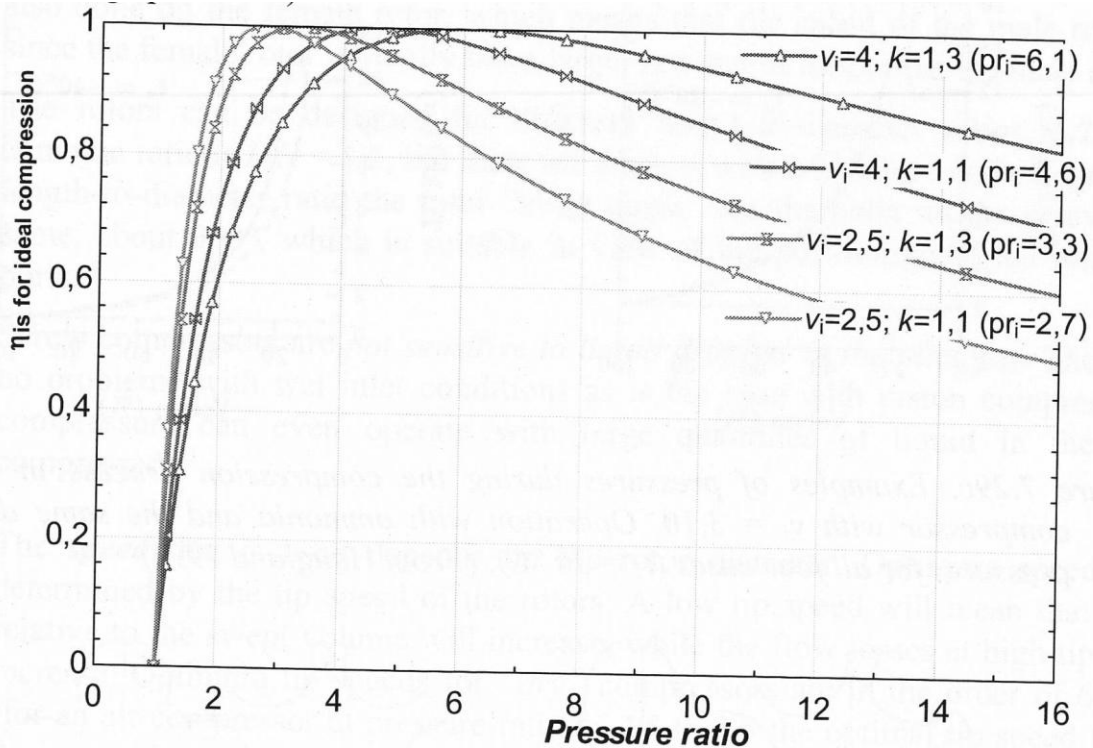
Figure 7.29a.  $p, V$  diagram for compression processes with different built-in volume ratios.

# Screw compressors

$$\eta_{is} = \frac{(p_1/p_2)^{(k-1)/k} - 1}{\pi_i^{(k-1)/k} - \frac{k-1}{k} \cdot \pi_i^{-(1/k)} \cdot (\pi_i - p_1/p_2) - 1}$$

$$\pi_i = v_i^k \quad \text{Built-in pressure ratio}$$

# Screw compressors

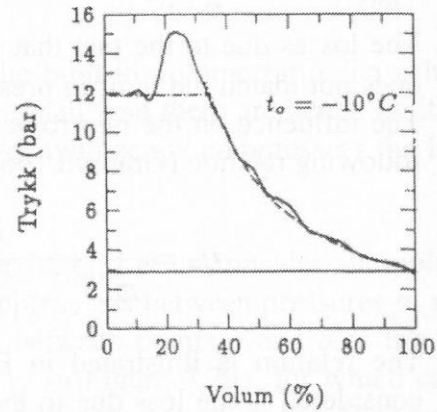
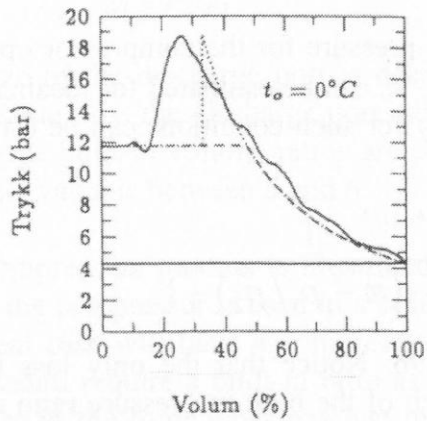


Do you remember the four types of losses that can penalize the compression efficiency?

Friction  
Heat transfer  
Mechanical  
Sealing

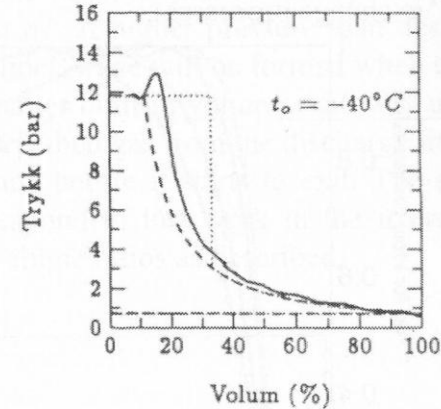
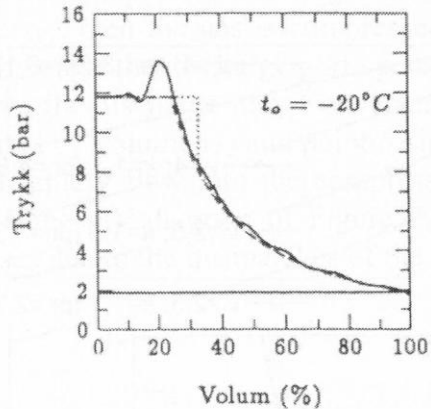
Figure 7.29b Influence of the built-in volume ratio on isentropic efficiency. The only loss considered is the one due to pressure ratio mismatch.

$$C_{pi} > P_e$$



$$\pi_i = P_e$$

$$\pi_i < P_e$$



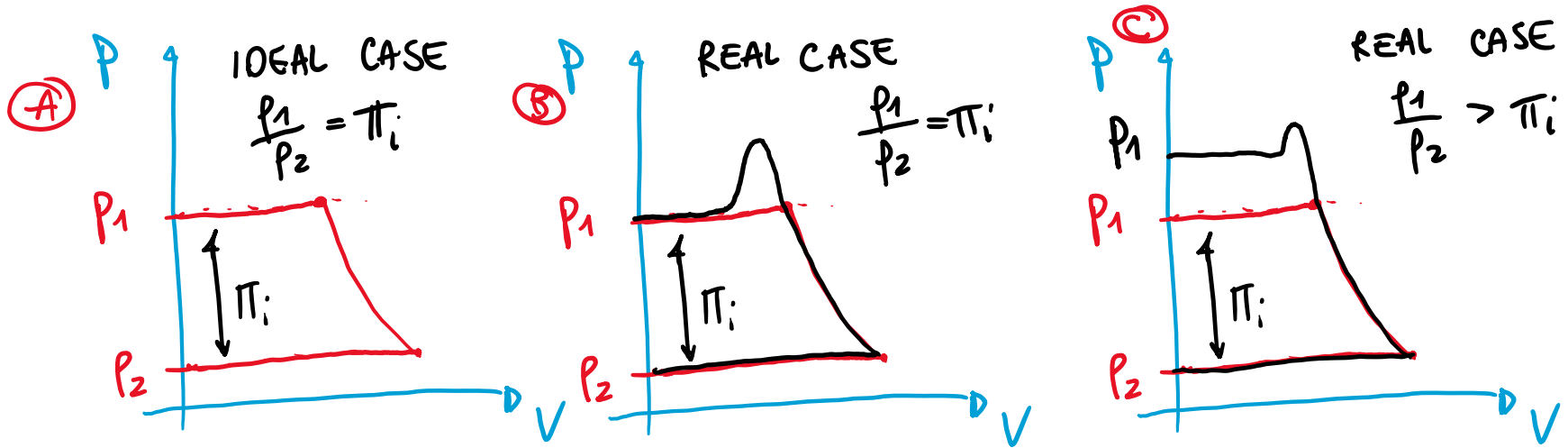
$$\pi_i \ll P_e$$

Figure 7.29c. Examples of pressures during the compression process in a screw compressor with  $v_i = 3,10$ . Operation with ammonia and the same discharge pressures for all four cases ( $t_1 = 30^\circ\text{C}$ ). (From Haugland 1993)

# Screw compressors: efficiency

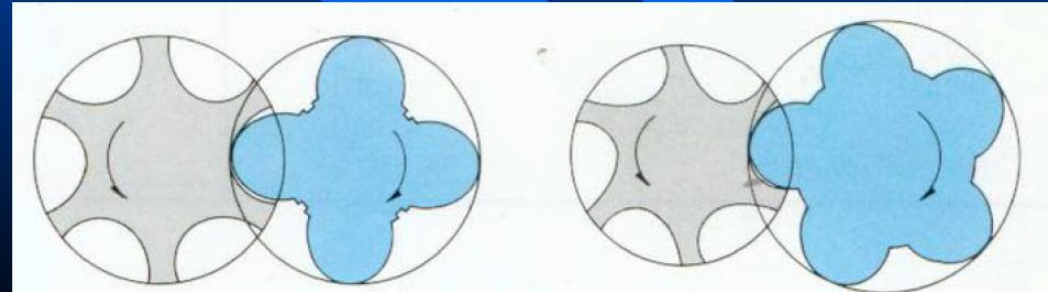
$$\pi_i = v_i^k \quad v_i = V_S/V_i \quad V_i \text{ is the gas volume when the discharge port opens}$$

When the gas volume reaches  $V_i$  the discharge port opens, but not all the gas can exit immediately because of the (initially) limited flow area. Therefore the pressure increases (in fact the rotors continue to rotate). In order for the gas to exit the compressor, the pressure must increase and therefore the maximum pressure reached by the gas is higher than the one calculated from the built-in pressure ratio (see fig. B, real case). For this reason in practice a screw compressor exhibits an optimum efficiency when the pressure ratio is somewhat larger than the built-in.



In the theoretical case, the best situation is when the pressure ratio is equal to the built-in pressure ratio (fig. A). In the real case, the optimum efficiency is reached when the pressure ratio is higher than the built-in pressure ratio (fig. C).

# Screw compressors



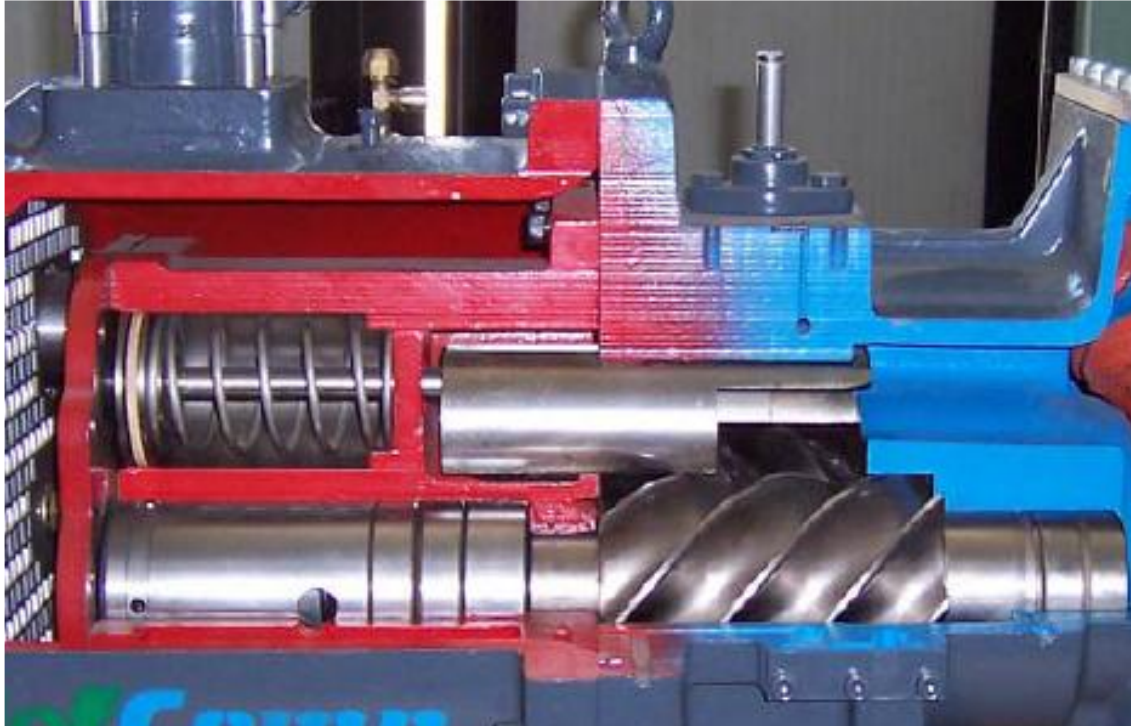
# Screw compressors: Oil

Oil is used to:

- Lubrication
- Sealing (minimize leaking between the rotor and housing and between the two rotors)
- Heat transfer carrier (reduce the temperature at the discharge)

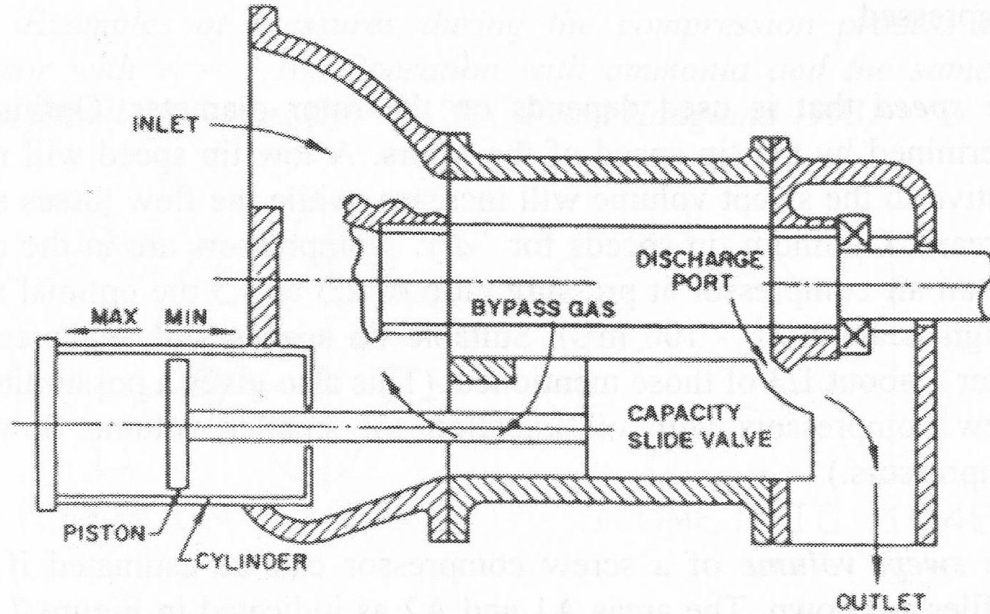
Compressors with oil injection are equipped with oil separators after the compression. Large quantities of oil are injected for sealing, lubrication and cooling of the gas (a compressor of 200 m<sup>3</sup>/h has a normal oil injection of 30 - 50 liters/minute). The oil must hence be removed. Such oil separators can be designed in different ways.

# Screw compressors



# Screw compressors: capacity control

Slide valve reduces the length of the active rotor



*Figure 7.33a. Arrangement for capacity control with a slide valve. In the position shown the valve has opened an area between the house and the flange of the port through which gas can flow back to the inlet (through channels in the house) before compression. (From ASHRAE, 1996).*

# Screw compressors

Screw compressors operate up to **discharge pressures of 30 bar**.

This limitation is caused by leakages inside the compressor (especially for low molecular weight gases like hydrogen), allowable discharge temperatures, and loads on the screw compressor shaft (Wennemar, 2009).

Due to these limitations, dry gas screw compressors are rarely applied in hydrogen applications where significant pressure ratios are needed (Takao Ohama, 2006).

The cooling and sealing benefits, that are inherent in oil or water injected screw compressors, help to mostly solve these challenges.

# Non-mechanical solutions

Hydrogen can be compressed in non-mechanical ways by exploiting different principles, e.g. electrochemically or via adsorption (or absorption) of hydrogen into porous materials.

Compared to conventional compressors, key advantages of non-mechanical compressors are the absence of moving parts (no risk of vibrations, less wear of materials, no use of lubricants) and the ability to maintain and even improve the hydrogen purity during compression (up to 99.99%).

There are still challenges to bring the technology readiness level and scale on par with conventional compressors.

# Non-mechanical compression

Characteristics	Metal hydrides	Electrochemical	Adsorption-desorption
Compression Rate (Nm <sup>3</sup> h <sup>-1</sup> )	<10 Batch process	<10 Continuous process	No data Batch process
Outlet pressure (bar)	200	850	No data
Efficiency (%)	<10	~60	No data
Energy consumption (kWh kg <sup>-1</sup> )	10	4-8	No data
TRL	~6	~7	3
Applications	- Refueling stations	- Refueling stations	- Refueling stations
Advantages	- No moving parts - Thermally driven - Low footprint - High purity H <sub>2</sub> production	- No moving parts - High compression efficiency - Low footprint - High purity H <sub>2</sub> production	- No moving parts - Low costs of adsorbents - Low heat of adsorption
Disadvantages	- High desorption temperature - High heat of desorption - Limited heat transfer - High weight	- Difficulty in manufacturing of high-pressure cells - Difficulty in water management - High electrical cell resistance - Low efficiency at high outlet pressure	- Low thermal conductivity of adsorbents - Difficulty in thermal management - Low-temperature operation (77 K)

Non-mechanical hydrogen compressors can be grouped in the following three categories:

Metal hydrides

Electrochemical

Adsorption-desorption compressors.

Their typical characteristics are summarized in Table

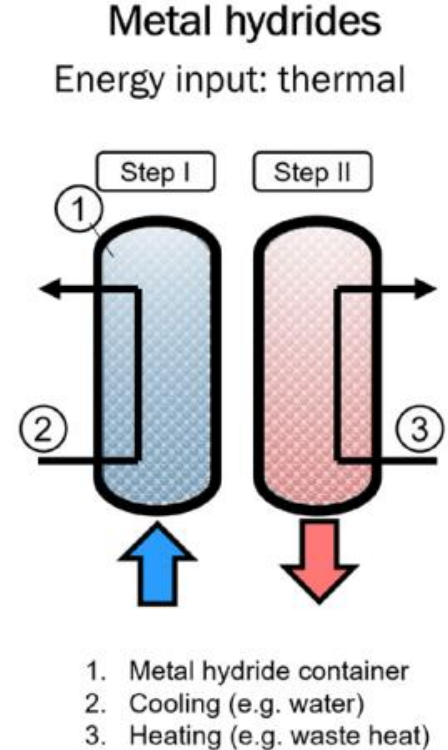
**Table 1.** Characteristics of non-mechanical compressor types, as found in (Sdanghi, et al., 2020), (Hyet, 2020), (Bellosta von Colbe, et al., 2019).

# Metal hydride compressors

Metal hydride compressors exploit the property of some metals, alloys, and intermetallic compounds to **absorb and desorb hydrogen forming hydrides**. The working principle is a **thermally-driven** chemisorption process with reversible adsorption-desorption kinetics: **hydrogen absorption occurs at low temperature**, and lasts until the equilibrium pressure is equal to the feed pressure. When the **metal hydride is heated, hydrogen can be desorbed and released at a higher pressure**. Since the compression requires only thermal energy, this type of compressors can be advantageous in applications where waste heat is available.

Metal hydride compressors are nowadays commercially available for **pressures up to 200 bar**, while compressors at higher outlet pressure are under development (Sdanghi, et al., 2019).

Metal hydride compressors suffer from slow adsorption/desorption kinetics in some cases, and require efficient cooling. Moreover, these compressors have relatively high specific energy consumption due to the low thermal conductivity of the absorbent materials and the high heat of adsorption.



# Electrochemical compressors

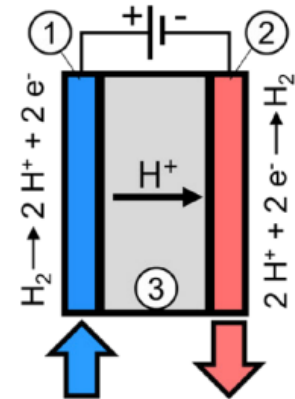
Electrochemical compressors are based on the same principle of proton-exchange membrane (PEM) fuel cells: **an electric current is applied to transport hydrogen through a proton-exchange membrane** from the (low-pressure) anode side to the (high-pressure) cathode side. Unlike PEM fuel cells, the cathode side is blocked (i.e. no air is fed), and water is not a reaction product. Instead, water is fed (as water vapor) along with the hydrogen feed, or on the cathode side to maintain the membrane wet and ensure stable performance. **One of the main advantages of electrochemical compressors is low energy requirements**, since they ensure isothermal compression. In principle, very high discharge pressures can be reached (up to 1000 bar, or up to 875 bar in a single stage (Hyet, 2020))

In practice, the main applications are limited to discharge pressures up to 100 bar.

One of the disadvantages is that the high-pressure hydrogen is wet, and thus a desiccant must be used downstream of the electrochemical compressor to dry the compressed hydrogen (e.g., prior to a subsequent compression stage).

## Electrochemical compressor

Energy input: electrical



1. Low-pressure anode
2. High-pressure cathode
3. Proton-exchange membrane

# Adsorption-desorption compressors

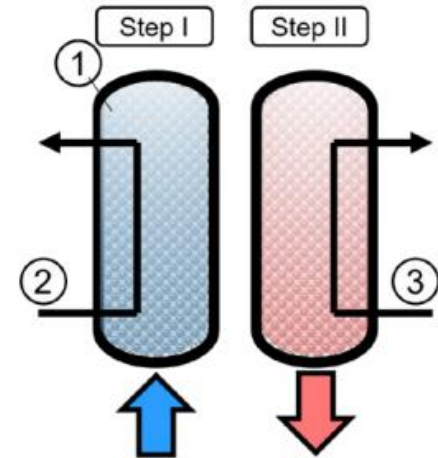
The working principle behind this class of non-conventional compressors is **thermally driven bonding of hydrogen onto microporous materials, similar to metal hydride compressors**. The difference is the operating temperature for adsorption (generally driven at temperatures as low as 77 K, i.e. the temperature of liquid nitrogen) and type of adsorbent used.

The main advantages of this type of non-conventional compressors is **low heat of adsorption**, and relatively high adsorption capacity, with hydrogen density comparable to liquid hydrogen (70 g/L) can be reached at  $T=77$  K.

However, these non-conventional compressors are still in their very early stage of development.

## Adsorption-desorption

Energy input: thermal



1. High-pressure container
2. Cooling (liquid  $N_2$ )
3. Heating (e.g. hot gas)

# Insight on the application: HYDROGEN FUELING

Critical aspects for fueling station compressor types are the **discharge pressure** that can be achieved (350-700 bar), as well as the **purity of the hydrogen** that is discharged by the compressor.

Other important selection criteria are the possibility of **large numbers of starts and stop**.

The critical aspects, particularly discharge pressure, disqualify the screw compressor.

As the hydrogen purity of key importance, a **diaphragm, ionic liquid piston, hydraulic piston or a non-mechanical compressor** type are well fit for the purpose.

Looking at other aspects such as the required capacity or the maturity of the system technology, the diaphragm compressor, the hydraulic piston compressor and the ionic liquid piston compressor type are currently the most logical, and are therefore the most observed types.

# Insight on the application: INJECTION TO TRANSPORT PIPELINES

The necessity of compression downstream of electrolyzers depends on the type of electrolyzer installed. For alkaline stacks, current (conceptual) designs assume that these operate at atmospheric pressure, resulting in the need for compression. For PEM stacks, it is often assumed that they operate at a **pressurised mode up to 30 bar**, which reduces the need for compression.

Production rates can vary significantly for electrolyzers, particularly in case they are off-grid, connected only to wind farms or solar plants.

Compressors that feed the hydrogen into a transportation network will need to compress the hydrogen to a minimum of 30 bar. Temperature and purity are considered important but not critical, and could be treated post compression.

For most systems, **reciprocating compressors are currently selected for compression**, particularly for their maturity as well as flexibly in capacity control.

# Insight on the application: END USE (HIGH TEMPERATURE HEAT)

In the petrochemical industry, fuel gases for combustion are most often a mixture of hydrocarbons and hydrogen (e.g. refinery off gas). The percentage of hydrogen is variable, but can amount up to 80%. Burners, designed to operate also at 100% hydrogen are already available and will be a likely candidate to provide high-temperature heat in process installation, based on pure hydrogen supply.

For the traditional fuel gases, combustion pressures are typically low (~1 barg). On existing refineries, the fuel gas is typically supplied at a pressure of 2-4 barg.

The expected pressure increase will be relatively limited (probably < 10 bar).

Typically, the requirement favors the application of reciprocating compressors or screw compressors. Finally, the fuel gas composition may have an impact on the stability of the combustion. Since existing burners already use a fluctuating mix of components, the gas purity is not considered the most critical requirement (probably lubricated compressors are a feasible option for this application).