# AERAULIC PLANT
# (DUCTWORK SYSTEM)

# PRESSURE LOSSES AND SIZING

- **NEED TO SET PHYSICAL DIMENSION OF THE PIPES AND THE MACHINE THAT WILL PROVIDE THE MOVEMENT (FAN)**

- **To size any pipe, the air flow (G) will be needed as input**

$[m^2]$ **S**

$[m^3/s]$ **$G = S \circ v$**

$[m/s]$ **v**

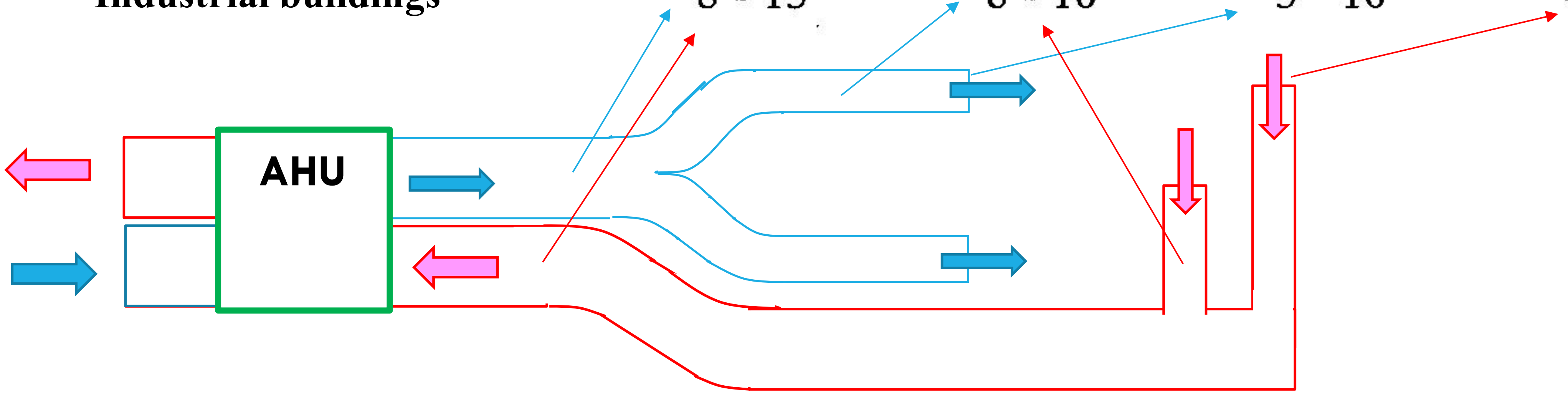
**SMALL pipe size**

| PROS | CONS |
|---|---|
| Small dimension means small overall cost | High fluid velocity, must take into account vibrations and noise |
| Reduced necessary thechnical space and easier positioning | High fluid-duct wall frictions, higher costs to keep the flow moving |

# Recommended air velocities in ducts

## Air velocity [m/s]

| | Main ducts | Secondary ducts | Air outlet | Return vents |
|---|---|---|---|---|
| Residential | 4 - 5 | 3 - 4 | 2 - 3 | 1,5 - 2 |
| Public buildings, schools | 5 - 8 | 4 - 6 | 3 - 5 | 2 - 3 |
| Offices | 8 - 11 | 6 - 8 | 5 - 8 | 3 - 4 |
| Industrial buildings | 8 - 15 | 6 - 10 | 5 - 10 | 4 - 10 |



To set a fluid in motion a pressure difference between inlet and outlet sections will be needed.

$\Delta P = \Delta P_k + \Delta P_p + \Delta P_f$

$\Delta P_k$: kinetic energy variation

$\Delta P_p$: potential energy variation

$\Delta P_f$: friction pressure loss

Friction pressure loss (or pressure loss) is due to two components:

Localised pressure drop: occurs whenever flow meets discontinuity in its path

Continuous pressure drop: occurs because of fluid-wall interactions

# CONTINUOUS PRESSURE DROP

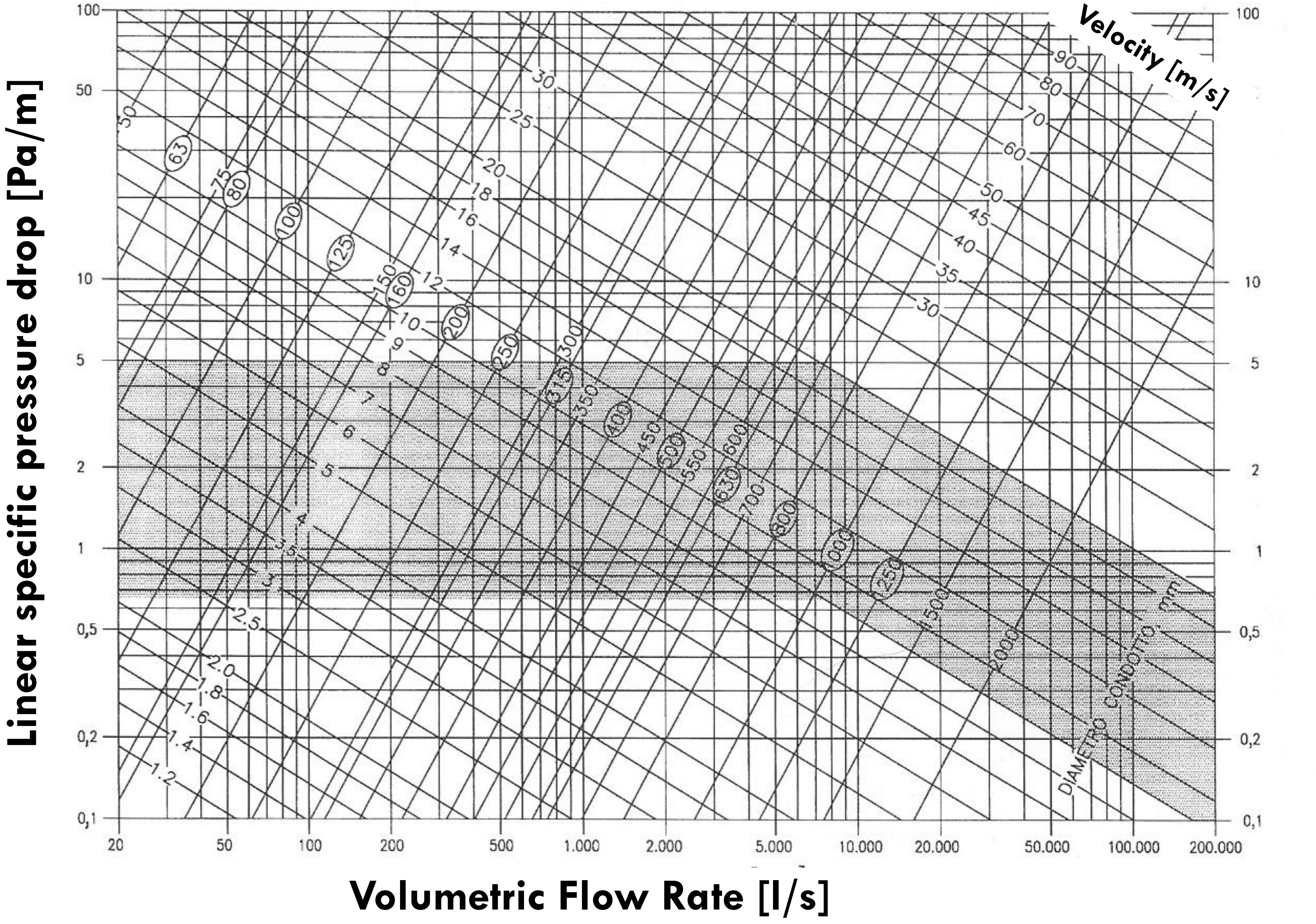
$$\frac{\Delta P_c}{L} = \frac{\rho\, v^2}{2}\frac{f}{D} \quad [\frac{Pa}{m}]$$

- $\rho$ fluid density $\left[\frac{\mathrm{kg}}{\mathrm{m}^3}\right]$
- v mean fluid velocity $\left[\frac{\mathrm{m}}{s}\right]$
- f friction factor [/]
- L pipe length [m]
- D pipe diameter [m]

The friction factor **f** is determined either analytically of via appropriate diagrams.
It depends on:

- Fluid density, viscosity and velocity
- Duct diameter and roughness

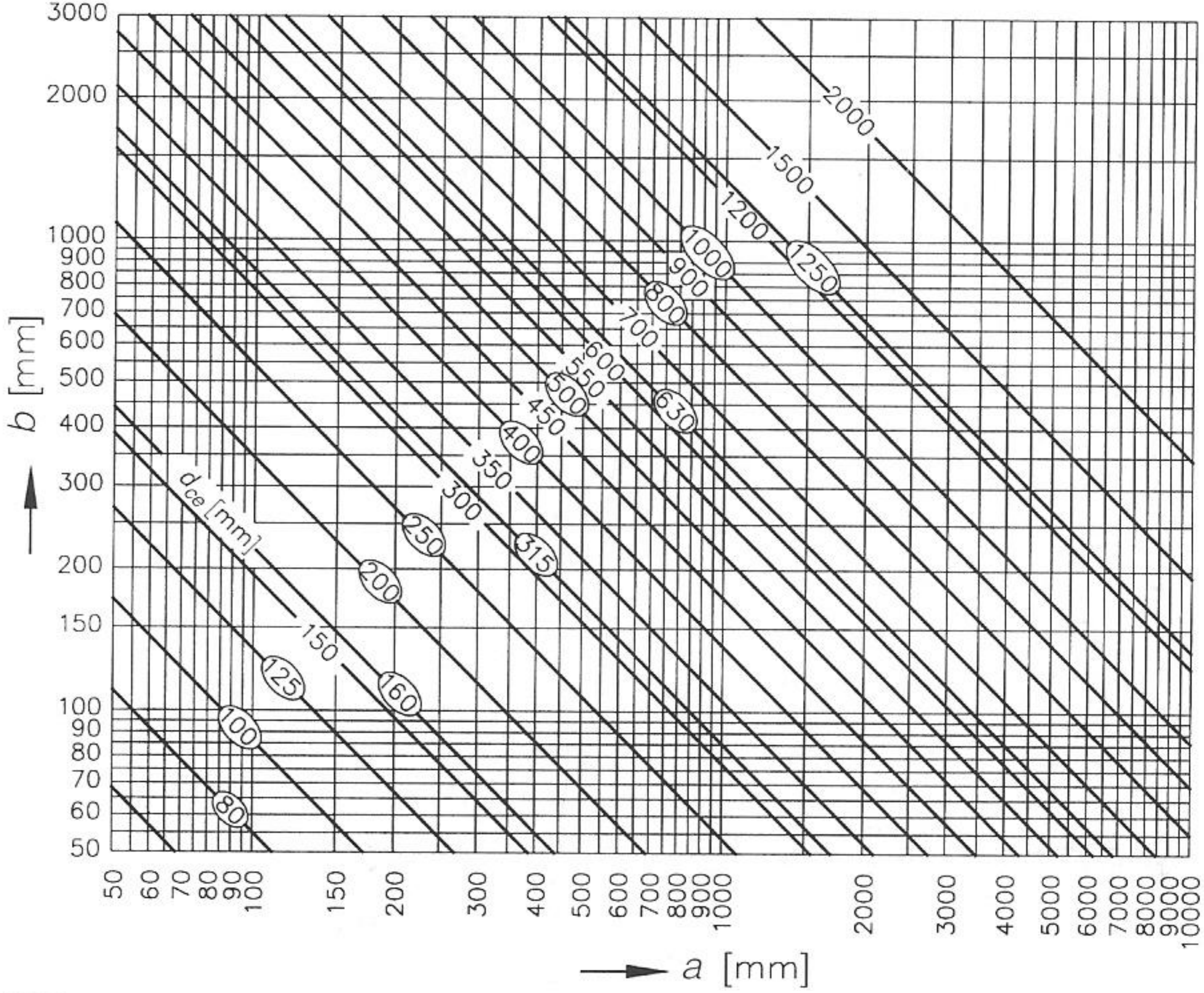
# Diagram for distributed losses for quick choice for ε = 0.09 mm and $\rho$ = 1.2 kg/m$^3$




Standard diameters of the circular ducts according to UNI 10381-2

**Dimensions of rectangular ducts equivalent to circular ducts standardized diameters:**

$$d_{ce} = 1{,}30\ \frac{(a \times b)^{0{,}625}}{(a + b)^{0{,}250}}$$



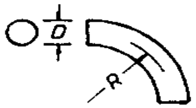
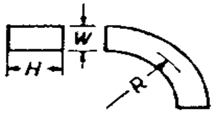

Standard diameters of the circular ducts according to UNI 10381-2

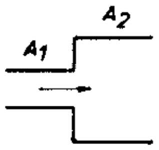
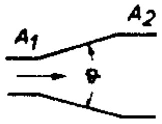
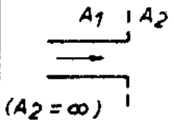
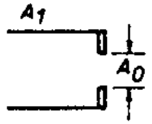
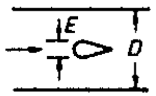
# LOCALISED PRESSURE DROP

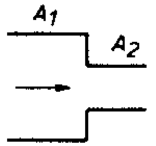
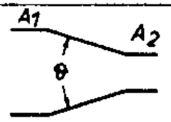
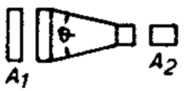
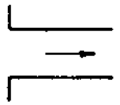
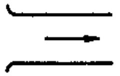
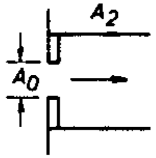
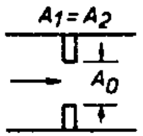
$$\Delta P_l = \frac{\rho\, v^2}{2}\xi \quad [\text{Pa}]$$

- $\rho$ fluid dencity $\left[\frac{\text{kg}}{\text{m}^3}\right]$
- v mean fluid velocity $\left[\frac{\text{m}}{s}\right]$
- $\xi$ localised p.d. coefficient [/]

$\xi$ coefficient is established on the basis of the particular type of loss, usually tabulated for the most common circuit elements (curves, branches etc.)

| TIPO | FIGURA | CARATTERISTICHE | PERDITA DI PRESSIONE C | L/D | L/W |
|---|---|---|---|---|---|
| CURVA A N° |  | RETTANGOLARE O ROTONDA; CON O SENZA ALETTE | N°/90 x PERDITA DI UNA CURVA UGUALE A 90° | | |
| CURVA A 90° SEZIONE ROTONDA |  | GOMITO | 1,30 | 65 | |
| | | R/D = 0,5 | 0,90 | | |
| | | 0,75 | 0,45 | 23 | |
| | | 1,0 | 0,33 | 17 | |
| | | 1,5 | 0,24 | 12 | |
| | | 2,0 | 0,19 | 10 | |
| CURVA A 90° SEZIONE RETTANGOLARE |  | H/W R/W | | | |
| | | 0,25 GOMITO | 1,25 | | 25 |
| | | 0,5 | 1,25 | | 25 |
| | | 0,75 | 0,60 | | 12 |
| | | 1,0 | 0,37 | | 7 |
| | | 1,5 | 0,19 | | 4 |
| | | 0,5 GOMITO | 1,47 | | 49 |
| | | 0,5 | 1,10 | | 40 |
| | | 0,75 | 0,50 | | 16 |
| | | 1,0 | 0,28 | | 9 |
| | | 1,5 | 0,13 | | 4 |
| | | 1,0 GOMITO | 1,50 | | 75 |
| | | 0,5 | 1,00 | | 50 |
| | | 0,75 | 0,41 | | 21 |
| | | 1,0 | 0,22 | | 11 |
| | | 1,5 | 0,09 | | 4,5 |
| | | 4,0 GOMITO | 1,38 | | 110 |
| | | 0,5 | 0,96 | | 65 |
| | | 0,75 | 0,37 | | 43 |
| | | 1,0 | 0,19 | | 17 |
| | | 1,5 | 0,07 | | 6 |
| CURVA A 90° SEZIONE RETTANGOLARE CON DEFLETTORI |  | R/W $R_1/W$ $R_2/W$ | | | |
| | | GOMITO 0,5 | | | 28 |
| | | 0,5 0,4 | 0,70 | | 19 |
| | | 0,7 0,6 | | | 12 |
| | | 1,0 1,0 | 0,13 | | 7,2 |
| | | 1,5 | 0,12 | | |
| | | GOMITO 0,3 0,5 | | | 22 |
| | | 0,5 0,2 0,4 | 0,45 | | 16 |
| | | 0,75 0,4 0,7 | 0,12 | | |
| | | 1,0 0,7 1,0 | 0,10 | | |
| | | 1,5 1,3 1,6 | 0,15 | | |
| GOMITO CON ALETTE |   | C = 0,10 ÷ 0,35 SECONDO LA COSTRUZIONE | | | |
| GOMITO A T CON ALETTE |  | CONSIDERARLO UGUALE AD UNA CURVA ANALOGA. PERDITA BASATA SULLA VELOCITA' IN ENTRATA | | | |
| T CURVILINEO |  | | | | |

| TIPO | FIGURA | CARATTERISTICHE | COEFFICIENTE | |
|---|---|---|---|---|
| ESPANSIONE BRUSCA |  | $A_1/A_2$ | $C_1$ | $C_2$ |
| | | 0,1 | 0,81 | 81 |
| | | 0,2 | 0,64 | 16 |
| | | 0,3 | 0,49 | 5 |
| | | 0,4 | 0,36 | 2,25 |
| | | 0,5 | 0,25 | 1,00 |
| | | 0,6 | 0,16 | 0,45 |
| | | 0,7 | 0,09 | 0,18 |
| | | 0,8 | 0,04 | 0,06 |
| | | 0,9 | 0,01 | 0,01 |
| ESPANSIONE GRADUALE |  | θ | $C_r$ | |
| | | 5° | 0,17 | |
| | | 7° | 0,22 | |
| | | 10° | 0,28 | |
| | | 20° | 0,45 | |
| | | 30° | 0,59 | |
| | | 40° | 0,73 | |
| EFFLUSSO BRUSCO |  | $A_1/A_2 = 0,0$ | 1,00 | |
| ORIFIZIO DI EFFLUSSO A SPIGOLI VIVI |  | $A_0/A_1$ | $C_0$ | |
| | | 0,0 | 2,50 | |
| | | 0,2 | 2,44 | |
| | | 0,4 | 2,26 | |
| | | 0,6 | 1,96 | |
| | | 0,8 | 1,54 | |
| | | 1,0 | 1,00 | |
| SBARRA ATTRAVERSO IL CANALE |  | E/D | C | |
| | | 0,10 | 0,7 | |
| | | 0,25 | 1,4 | |
| | | 0,50 | 4,0 | |
| TUBO ATTRAVERSO IL CANALE |  | E/D | C | |
| | | 0,10 | 0,20 | |
| | | 0,25 | 0,55 | |
| | | 0,50 | 2,0 | |
| SBARRA A PROFILO AERODINAMICO ATTRAVERSO IL CANALE |  | E/D | C | |
| | | 0,10 | 0,07 | |
| | | 0,25 | 0,23 | |
| | | 0,50 | 0,90 | |

| TIPO | FIGURA | CARATTERISTICHE | COEFFICIENTE |
|---|---|---|---|
| CONTRAZIONE BRUSCA SPIGOLI VIVI |  | $A_2/A_1$ | $C_2$ |
| | | 0,0 | 0,34 |
| | | 0,2 | 0,32 |
| | | 0,4 | 0,25 |
| | | 0,6 | 0,16 |
| | | 0,8 | 0,06 |
| CONTRAZIONE GRADUALE |  | θ | |
| | | 30° | 0,02 |
| | | 45° | 0,04 |
| | | 60° | 0,07 |
| TRASFORMAZIONE AD AREA COSTANTE |  | $A_1 = A_2$ | C |
| | | $\theta \leq 14°$ | 0,15 |
| INGRESSO A FLANGIA |  | $A = \infty$ | C |
| | | | 0,34 |
| INGRESSO A CANALE |  | $A = \infty$ | C |
| | | | 0,85 |
| INGRESSO GRADUALE |  | $A = \infty$ | C |
| | | | 0,03 |
| ORIFIZIO DI INGRESSO A SPIGOLI VIVI |  | $A_0/A_2$ | $C_0$ |
| | | 0,0 | 2,50 |
| | | 0,2 | 1,96 |
| | | 0,4 | 1,39 |
| | | 0,6 | 0,96 |
| | | 0,8 | 0,61 |
| | | 1,0 | 0,64 |
| ORIFIZIO A SPIGOLI VIVI NEL CANALE |  | $A_0/A$ | $C_0$ |
| | | 0,0 | 2,50 |
| | | 0,2 | 1,86 |
| | | 0,4 | 1,21 |
| | | 0,6 | 0,64 |
| | | 0,8 | 0,20 |
| | | 1,0 | 0,0 |

# DISTRIBUTION LINE DESIGN

## Step n 1

**Set up network geometry and its location in the building. Fans and/or the AHU should be placed as centered to the building as possible in order to restrict the network extension.**

**Check the weight of the equipment to check the static loads for the building structure.**

**Analyse the problem of noise and vibrations transmitted to the surroundings.**

## Step n 2

**Decide the flow rate for each section of the plant according to the specifics needs of each room (both extraction and supply).**

## Step n 3

Choice of the pipes size of the whole plant.

Pressure losses determination (as sum of localised and continuous pressure drops)

The result will most certainly differ for the various sections of the plant; need to equalize the distribution network by adding calibration/equalizing dampers in specific points (main branches, terminals etc).

## Step n 4

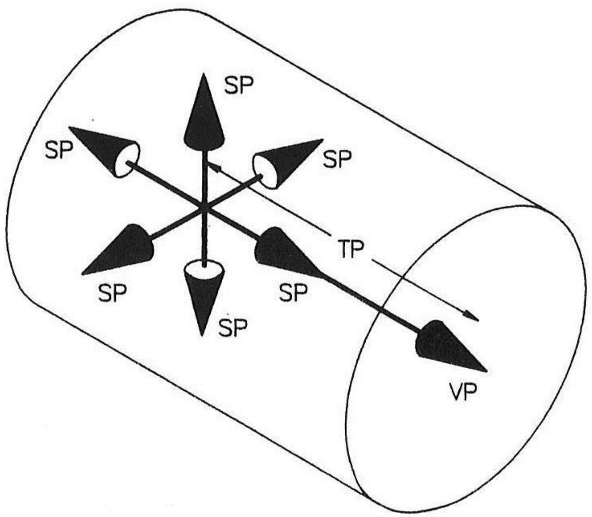
Choosing the appropriate operating machine (fan) that will provide the requested flow and supply the necessary head pressure to overcome the network pressure losses.

# Fan

When dealing with ventilation, the fan has the duty to produce the pressure that will keep the flow moving while facing all the circuit resistances (pressure drops).

The fan produces the TOTAL PRESSURE which is the sum of two components:

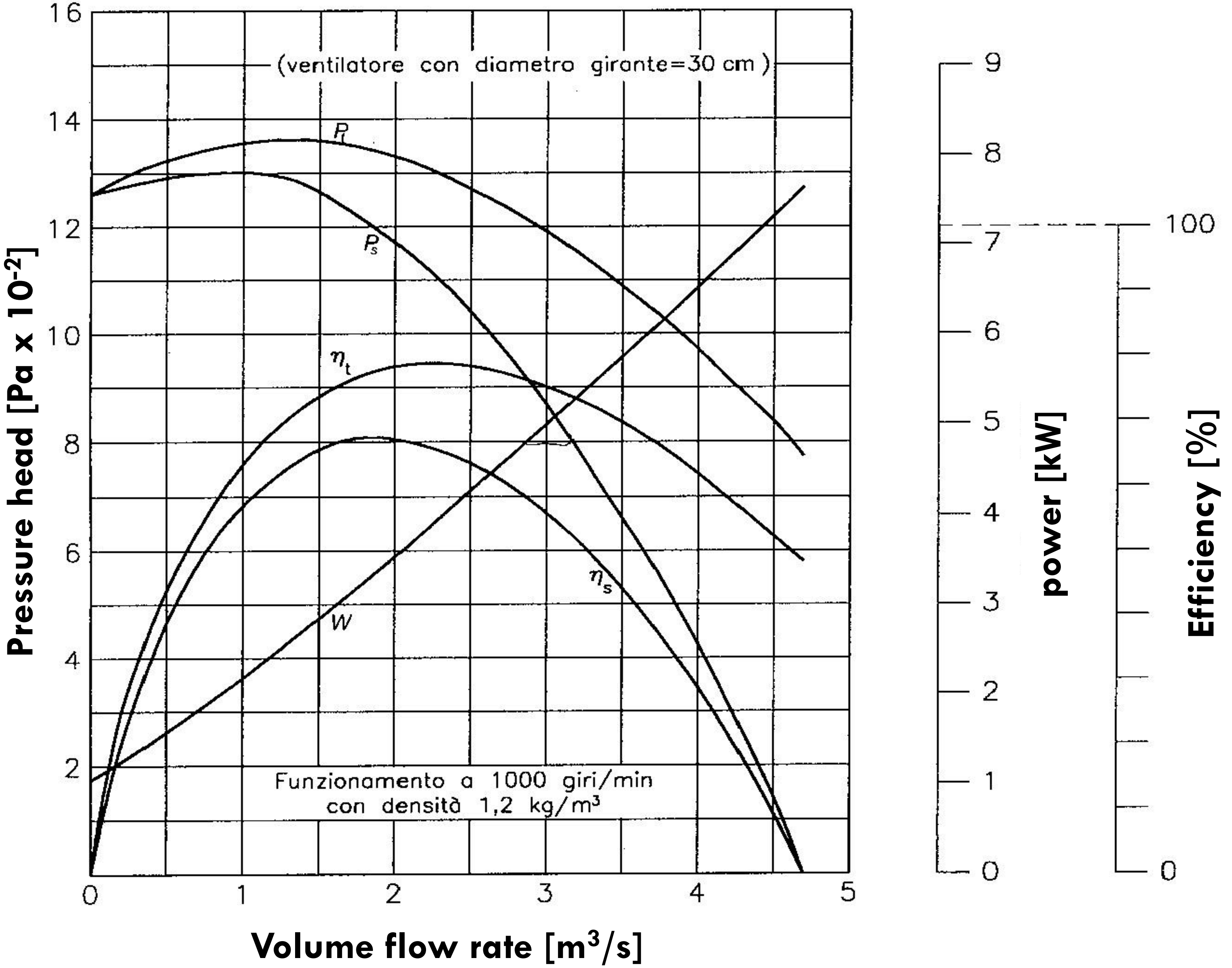
- *Static pressure* (any direction)
- *Dynamic pressure* (always aligned with the flow)




TP: Total Pressure

SP: Static Pressure

VP: Dynamic Pressure


(ventilatore con diametro girante=30 cm)
Funzionamento a 1000 giri/min
con densità 1,2 kg/m³
Pressure head [Pa x 10⁻²]
Volume flow rate [m³/s]
power [kW]
Efficiency [%]
$P_t$
$P_s$
$\eta_t$
$\eta_s$
W
0
1
2
3
4
5
2
4
6
8
10
12
14
16
9
8
7
6
5
4
3
2
1
0
100
0

# Static pressure

It is the pressure the flow will apply all over around the duct it is contained in.

It depends on the fan aerodynamic characteristics and acts equally in all directions; it does not depend on the fluid speed.

Taking the ambient pressure as a reference the static pressure will be:

POSITIVE if higher

NEGATIVE if lower

Static pressure provides the necessary energy to speed up the air from its quiet and to keep it moving while winning resistances due to friction and turbulence.

# Dynamic Pressure

It is the fluid pressure due to its kinetic energy.

It is created at the cost of static pressure.

It acts in the same direction as the fluid motion and is always considered positive.

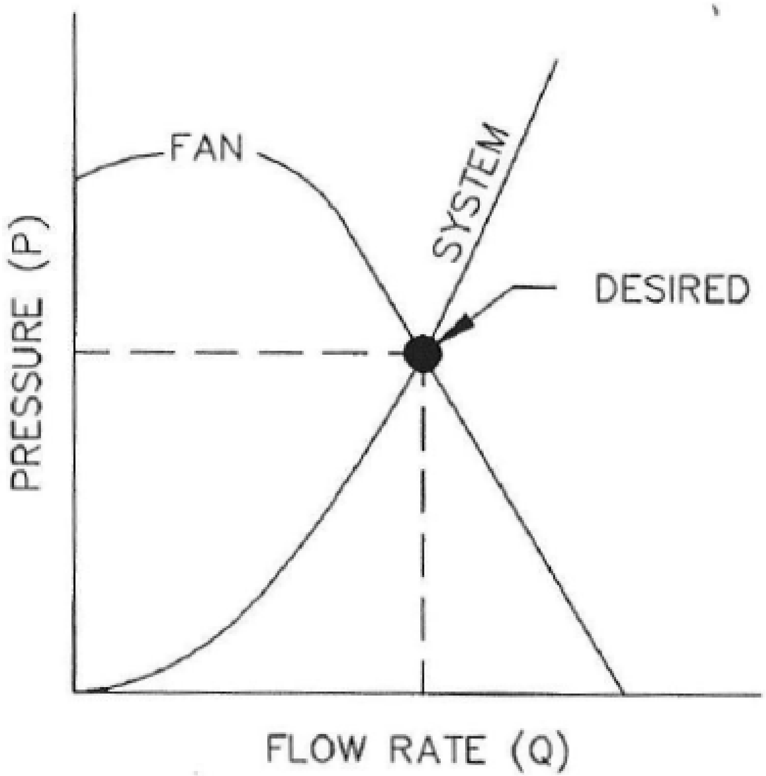
It is function of both speed and density of the fluid.

# Suction system operating point

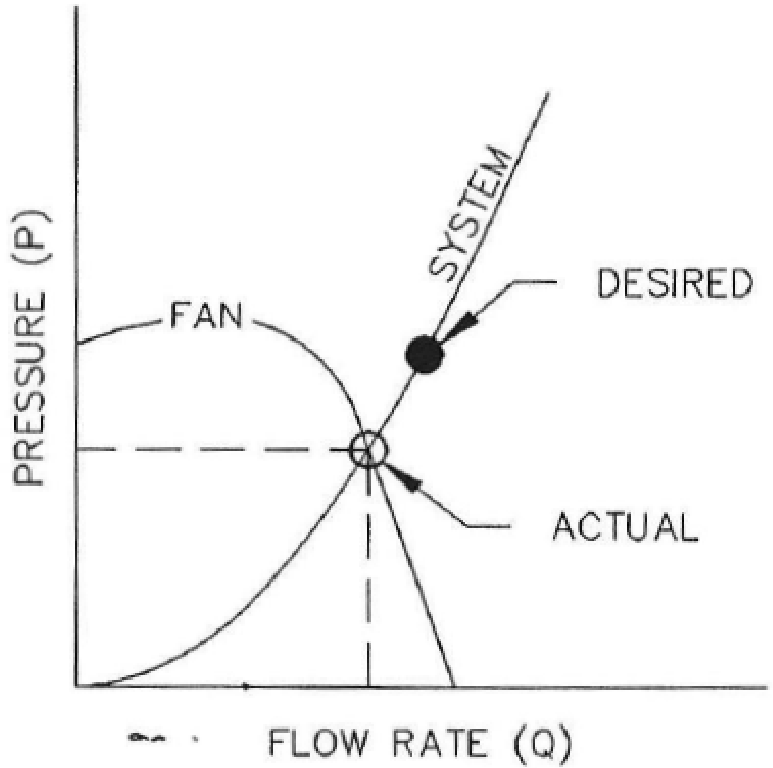
**Fan curve**

**System curve**

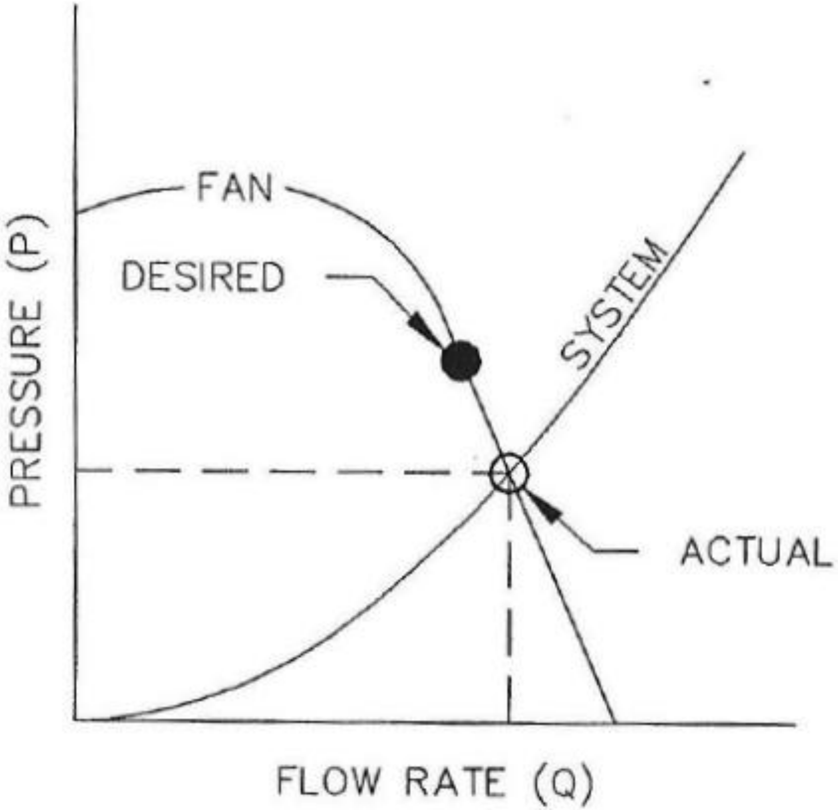
→ **The desired working point comes out of their intersection**



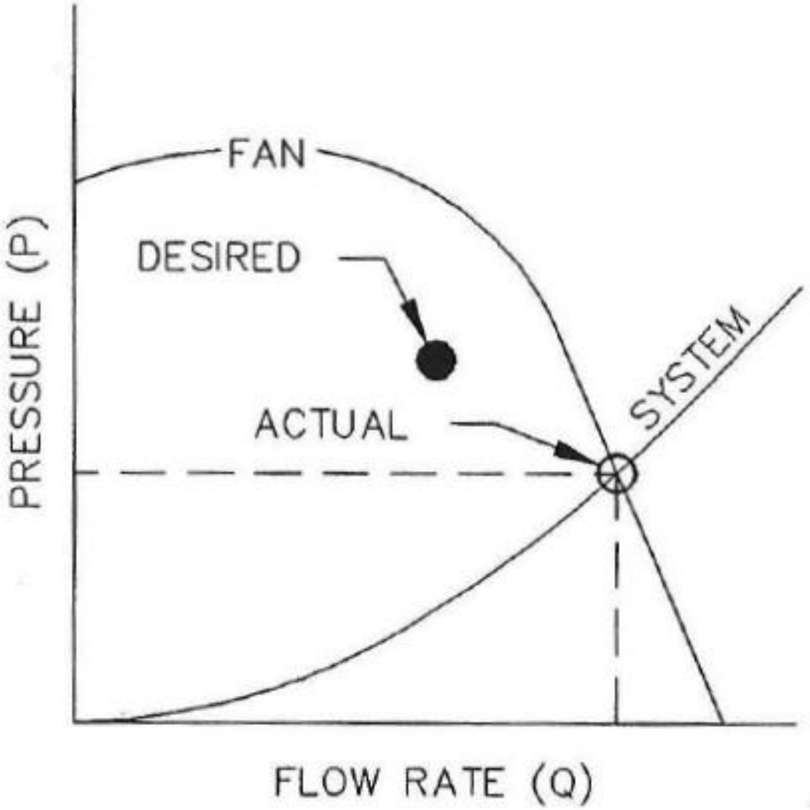

A. FAN AND SYSTEM MATCHED



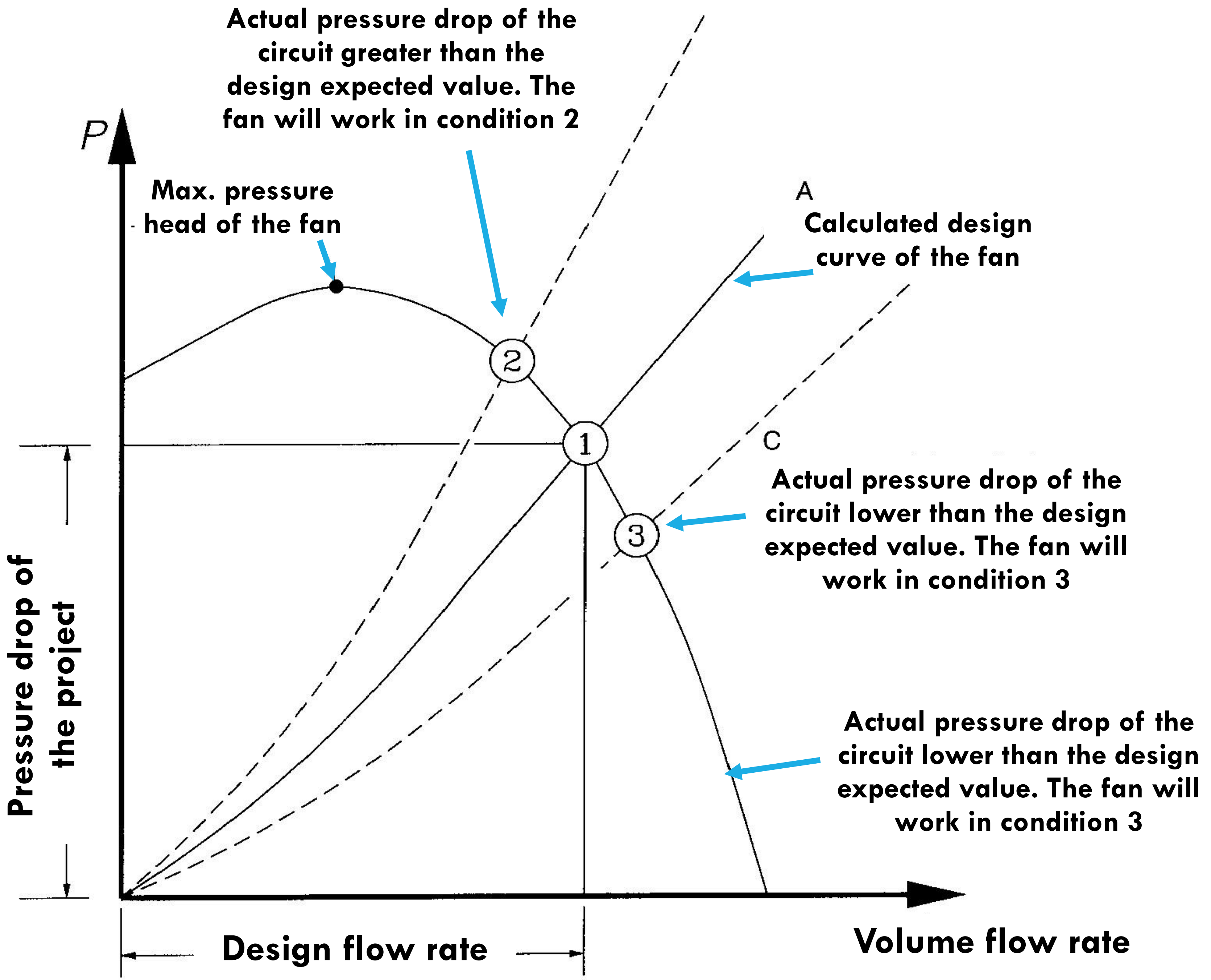

B. WRONG FAN.




C. WRONG SYSTEM.




D. BOTH FAN AND SYSTEM WRONG


Actual pressure drop of the circuit greater than the design expected value. The fan will work in condition 2
P
Max. pressure head of the fan
A
Calculated design curve of the fan
2
1
C
Actual pressure drop of the circuit lower than the design expected value. The fan will work in condition 3
3
Pressure drop of the project
Actual pressure drop of the circuit lower than the design expected value. The fan will work in condition 3
Design flow rate
Volume flow rate