# Heating plants

## HVAC System design

Marco Manzan

**University of Trieste**
**Department of Engineering and Architecture**

April 2025



## Air conditioning systems

1. liquid **(water or water with additives)**, are suitable for controlling only the internal temperature and not the humidity; they are sized based on the sensible load, they are usually used for heating and cooling.
   They can be of three different types:
   - radiators (winter use only)
   - fan coils (winter and summer use)
   - radiant panels (winter for heating and summer for cooling).
2. direct expansion **systems** (winter and summer use and for small and medium powers).
3. all-air **systems**, are suitable for controlling both the temperature and the internal humidity; can be divided in turn
   - single duct
   - double duct
   - constant or variable flow rate
4. Mixed systems **air** and **water**
   - water part controls the temperature
   - air part controls the humidity

## Radiators

Classical use for heating, heat exchange by

- radiation about 30 %
- convection about 70 %

The thermal output depends on the difference between the average temperature of the radiator and the ambient air

$$\Phi = c(\Delta\theta_a)^n$$

$c$ is a typical coefficient of the radiator

$n \sim 4/3$ for convection in turbulent regime

$\Delta\theta_a$ average temperature difference between the radiator and ambient air:



## heat output change with temperature

The difference between the average temperature of the radiator and the air can be expressed as:

$$\Delta\theta_a = \left[\frac{(\theta_m + \theta_r)}{2} - \theta_{air}\right]$$

$\theta_m$ inlet temperature

$\theta_r$ outlet temperature

according to UNI EN 442 the heat flux is calculated in nominal conditions with $\Delta\theta_a = \Delta\theta_n$ with $\theta_{aria} = 20^\circ$:

$$\left.\begin{aligned}\theta_m &= 85^\circ C\\ \theta_r &= 75^\circ C\end{aligned}\right\} \implies \Delta\theta_n = 60 \text{ K}$$

$$\left.\begin{aligned}\theta_m &= 75^\circ C\\ \theta_r &= 65^\circ C\end{aligned}\right\} \implies \Delta\theta_n = 50 \text{ K}$$

# heat power change with temperature

## temperature different from the nominal one

$$c = \frac{\Phi_n}{(\Delta\theta_n)^n}$$

$$\Phi(\Delta\theta_a) = c(\Delta\theta_a)^n \; = \; \Phi_n \left( \frac{\Delta\theta_a}{\Delta\theta_n} \right)^n$$

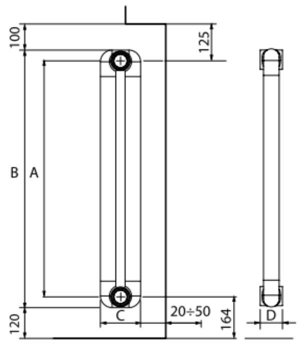


# Example of ta technical information sheet

## exampla taken from a technical sheet by FONDITAL

TRIBECA

| | | |
|---|---|---|
| Standard supply | 235 - 335 - 350 - 435<br>500 - 535 - 600 | from 4 to 20 elements |
| | 685 - 700 - 800 - 835 | from 4 to 16 elements |
| | 900 - 935 - 1000 - 1135<br>1200 - 1400 - 1435<br>1600 - 1735 - 1935 | from 4 to 9 elements |
| | 1800 - 2000 | from 4 to 12 elements |
| **Colours** | see colours table | |
| **Maximum working pressure** | 16 bar | |
| **Test pressure** | 24 bar | |
| **Aleternum treatment** | Supplied as standard | |




MEASURES EXPRESSED IN MILLIMETRES

| Model | Heat output | | | | | |
|---|---|---|---|---|---|---|
| | ΔT 20 | ΔT 30 | ΔT 40 | ΔT 50 | ΔT 60 | ΔT 70 |
| | W/sect. | W/sect. | W/sect. | W/sect. | W/sect. | W/sect. |
| **235** | 9,6 | 16,0 | 23,1 | 30,6 | 38,6 | 46,9 |
| **335** | 12,5 | 21,1 | 30,5 | 40,5 | 51,1 | 62,3 |
| **350** | 13,0 | 21,8 | 31,5 | 41,9 | 52,9 | 64,4 |
| **435** | 15,2 | 25,6 | 37,1 | 49,4 | 62,5 | 76,1 |
| **500** | 16,9 | 28,5 | 41,3 | 55,1 | 69,7 | 85,0 |
| **535** | 17,8 | 30,1 | 43,6 | 58,2 | 73,6 | 89,8 |
| **600** | 19,5 | 32,9 | 47,8 | 63,8 | 80,8 | 98,6 |
| **685** | 21,6 | 36,6 | 53,3 | 71,2 | 90,2 | 110,2 |
| **700** | 22,0 | 37,3 | 54,2 | 72,5 | 91,8 | 112,2 |
| **800** | 24,5 | 41,6 | 60,6 | 81,1 | 102,8 | 125,8 |
| **835** | 25,4 | 43,1 | 62,8 | 84,1 | 106,7 | 130,5 |
| **900** | 27,0 | 45,9 | 67,0 | 89,7 | 113,9 | 139,3 |

| Model | Heat output | | | | | |
|---|---|---|---|---|---|---|
| | ΔT 20 | ΔT 30 | ΔT 40 | ΔT 50 | ΔT 60 | ΔT 70 |
| | W/sect. | W/sect. | W/sect. | W/sect. | W/sect. | W/sect. |
| **935** | 27,9 | 47,5 | 69,2 | 92,7 | 117,7 | 144,0 |
| **1000** | 29,6 | 50,3 | 73,4 | 98,3 | 124,9 | 152,8 |
| **1135** | 33,0 | 56,2 | 82,0 | 110,0 | 139,8 | 171,2 |
| **1200** | 34,6 | 59,9 | 87,5 | 115,7 | 149,3 | 182,8 |
| **1400** | 39,7 | 67,9 | 99,2 | 133,3 | 169,6 | 207,9 |
| **1435** | 40,7 | 69,5 | 101,6 | 136,4 | 173,5 | 212,7 |
| **1600** | 45,1 | 77,1 | 112,6 | 151,2 | 192,3 | 235,6 |
| **1735** | 48,9 | 83,4 | 121,8 | 163,4 | 207,8 | 254,6 |
| **1800** | 50,7 | 86,4 | 126,3 | 169,4 | 215,4 | 263,9 |
| **1935** | 54,5 | 92,9 | 135,7 | 181,9 | 231,3 | 283,3 |
| **2000** | 56,4 | 96,1 | 140,2 | 188,1 | 239,0 | 292,7 |

https://www.fondital.com/downloads/5401/4480/CTC-GAMMA-RADIATORI-EN-EN.pdf

## Mass water flow

After sizing the radiator the required water flow can be computed

$$\Phi(\Delta\theta_a) = \Phi_n \cdot \left(\frac{\Delta\theta}{\Delta\theta_n}\right)^n = \dot{m} \cdot c \cdot \Delta\theta_{mr}$$

con

- $\dot{m}$ mass water flow(water or additivate water).
- $c$ specific heat capacity (4,187 kJ/kgK for water)
- $\Delta\theta$ inlet and outlet temperature difference $\Delta\theta_{mr} = \theta_i - \theta_r$ .



## water mass flow

Once computed the mass flow the piping can be sized using specified velocities which depend on:

- pressur losses
- noise
- corrosion
- air

## recommended water velocity

| Recommended velocity (m/s) for hot and chilled water networks | | | |
|---|---|---|---|
| | pipes main | pipes secondary | branches to heating bodies |
| steel pipes | 1.5 - 2.5 | 0.5 - 1.5 | 0.2 - 0.7 |
| copper pipes | 0.9 - 1.2 | 0.5 - 0.9 | 0.2 - 0.5 |
| plastic pipes | 1.5 - 2.5 | 0.5 - 1.5 | 0.2 - 0.7 |



## Types of fluid flow

### laminar flow

- regular flow
- low velocities
- low pressure drops or head loss
- reduced heat exchange

### turbulent flow

- high velocities
- chaotic motion
- high pressure drops and strong heat exchange

## dimensionless groups

### Reynolds number

- heat exchanges and pressure drops are computed using correlations
- *Re* fundamental parameter for calculating flow type
- ratio between inertial forces and viscous forces
- for each geometry determines whether the motion is *laminar* or *turbulent*

$$Re = \frac{\rho \cdot u \cdot \cancel{L^2} \cdot \cancel{u}}{\mu \cdot \cancel{u}/L \cdot \cancel{L^2}} = \frac{\rho \cdot u \cdot L}{\mu}$$

$u$ speed

$\rho$ density

$\mu$ dynamic viscosity $\mathrm{kg/(m\ s)}$

- laminar flow $Re < 2000$ in round ducts and pipes.
- transition $2000 \leq Re < 4000$
- turbulent $4000 \leq Re$



## Steady Flow Energy Equation

### relationship between pressure and velocity in a duct

$$(p_2 - p_1) + \frac{1}{2}\rho \cdot (u_2^2 - u_1^2) + g \cdot \rho \cdot (z_2 - z_1) + \Delta p_l = 0$$

$u$ velocity

$p$ pressure

$z$ elevation

$\Delta p_l$ pressure loss

# Steady Flow Energy Equation

## total pressure

$$P_t = p + \frac{1}{2} \cdot \rho \cdot u^2$$
$$P_{t,1} - P_{t,2} = \rho \cdot g \cdot (z_2 - z_1) + \Delta p_l$$

- the pressure difference between inlet and outlet depends on head losses and height difference
- the formula is valid for closed-circuit and open-circuit systems
- for closed-circuit systems the elevation head term disappears
- $\Delta p_l$ takes into account the losses along the pipe and fittings discontinuities



# pressure loss

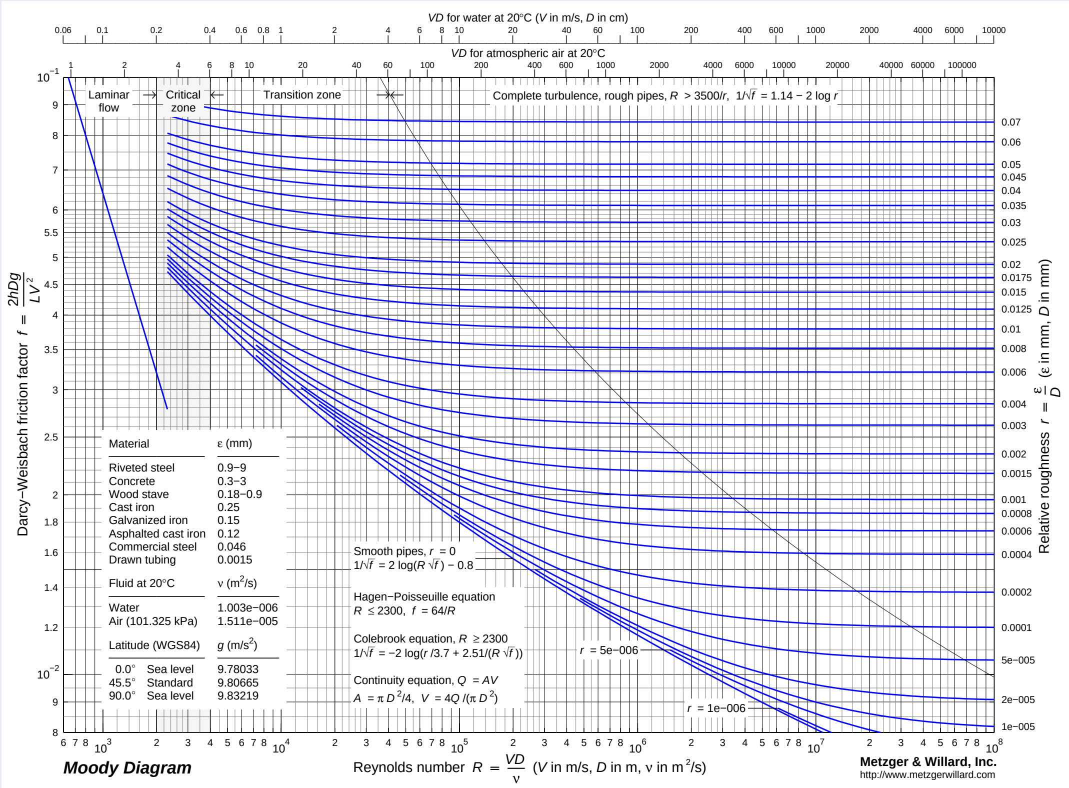
## Friction Factor

$$\frac{\Delta p}{L} = r = F_a \frac{1}{D} \rho \frac{v^2}{2}$$

- $r$ [Pa/m] pressure drop per unit length $\frac{\Delta p}{L}$
- $L$ length of the duct
- $D$ diameter of the duct
- $\rho$ density of the fluid
- $v$ velocity of the fluid
- $F_a$ friction factor

- pressure drops are proportional to the square of the velocity of the fluid
- depend on the flow regime, laminar or turbulent
- can be calculated with diagrams or formulas

# Moody diagram






## $f$ and $Re$

- laminar flow $f$ is affected mainly by the viscous force of the fluid flow is a function of $Re$ only.

$$f = \frac{64}{Re}$$

- smooth tube $Re > 4000$ surface roughness submerged in laminar sublayer, $f$ decreases vith $Re$

$$f = \frac{0.316}{Re^{0.25}}$$

- with an increase of $Re$ laminar becomes thinner than roughness. $f$ increases
- if $Re >$ *Rouse limit* $f$ depends on relative roughness $\epsilon/D$ only

## friction factor

can be obtained with Colebrook equation:

$$\frac{1}{\sqrt{F_a}} = -2 \cdot log\left(\frac{k}{3,7 \cdot D} + \frac{2,51}{Re\sqrt{F_a}}\right)$$

where

$k$ absolute roughness

$Re$ Reynolds number

- implicit formulation
- difficult to be used for computing head losses
- other formulas are available in explicit form



## Absolute roughness

**low roughness**

$$0.002 < k < 0.007 \quad \mathrm{mm}$$

- copper
- plastic water pipe

**medium roughness**

$$0.02 < k < 0.09 \quad \mathrm{mm}$$

- steel
- galvanized steel

**high roughness**

$$0.2 < k < 1.0 \quad \mathrm{mm}$$

- scaled steel
- corroded steel
- concrete

## Alternative formulas

### Swamee-Jain

$$F_a = 0.25 \cdot \left[ log \left( \frac{k/D}{3.7} + \frac{5.74}{Re^{0.9}} \right) \right]^{-2}$$

### Haaland

$$\frac{1}{F_a} = -1.8 \cdot log \left[ \left( \frac{k/D}{3.7} \right)^{1.11} + \frac{6.9}{Re} \right]$$

### Atsui-saal

$$f^* = 0.11 \cdot \left( \frac{k}{D} + \frac{68}{Re} \right)^{0.25}$$

$f^* > 0.018$ $F_a = f^*$

$f^* < 0,018$ $F_a = 0,85 \cdot f^* + 0,0028$



## simplified formulas
Quaderni Caleffi

practical formulas for $F_a$ with different tube material

low roughness $2\mu m < k < 7\mu m$ (Cu, PE)

$$F_a = 0,316 \ Re^{-0,25}$$

medium roughness $20\mu m < k < 90\mu m$ (acciaio)

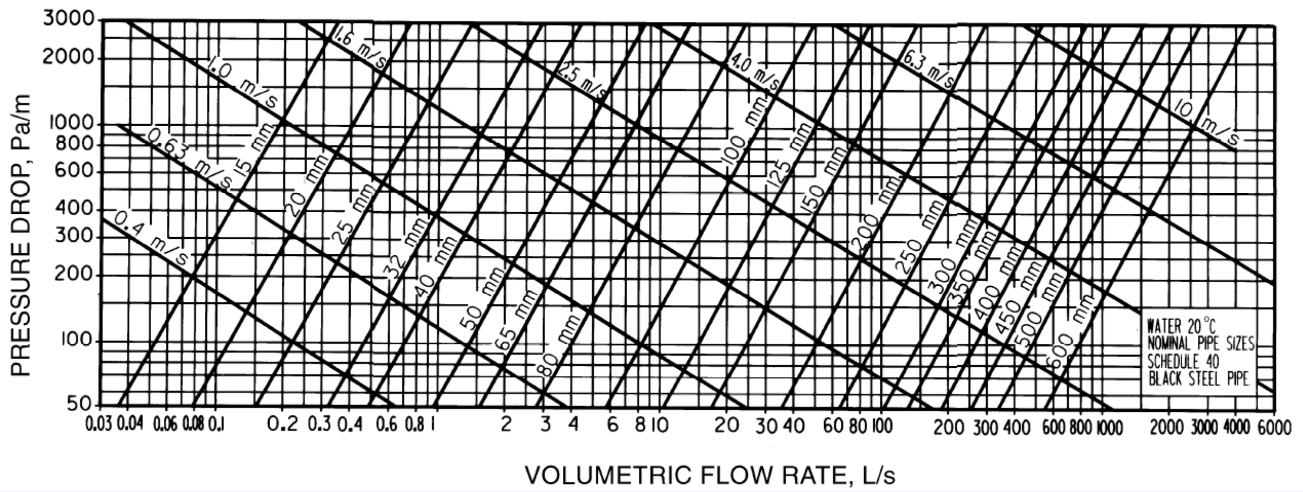
$$F_a = 0,07 \ Re^{-0,13} \ D^{-0,14}$$

high roughness $0,2mm < k < 1mm$ Colebrook equation or alternatives

# Friction chart

from ASHRAE

## steel pipes

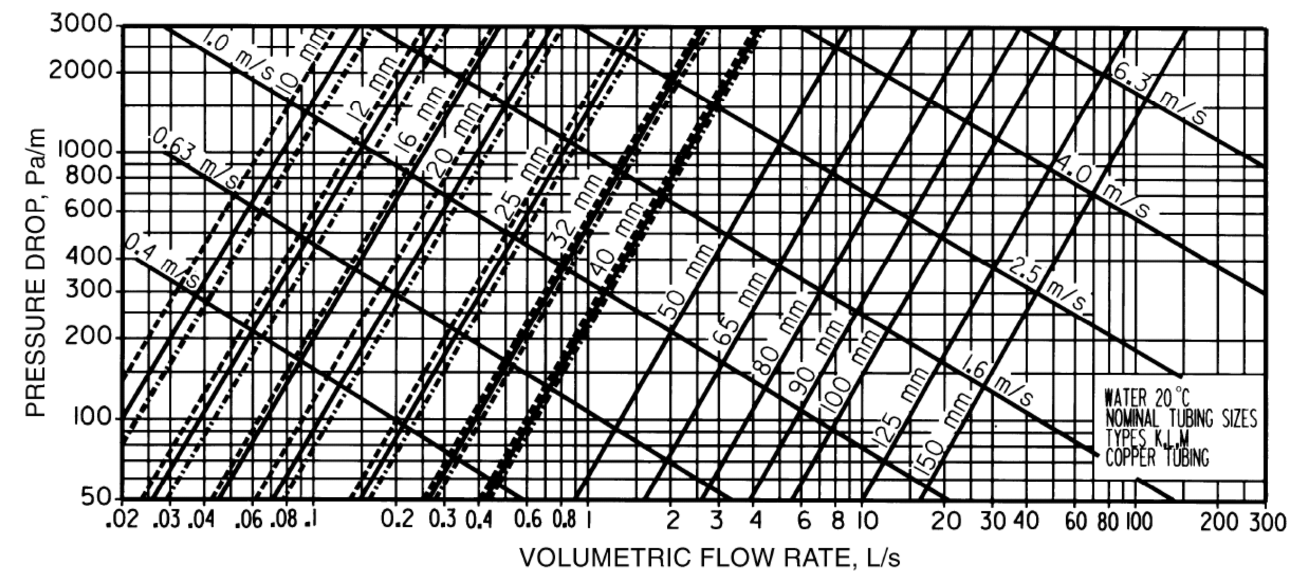





# Friction chart

from ASHRAE

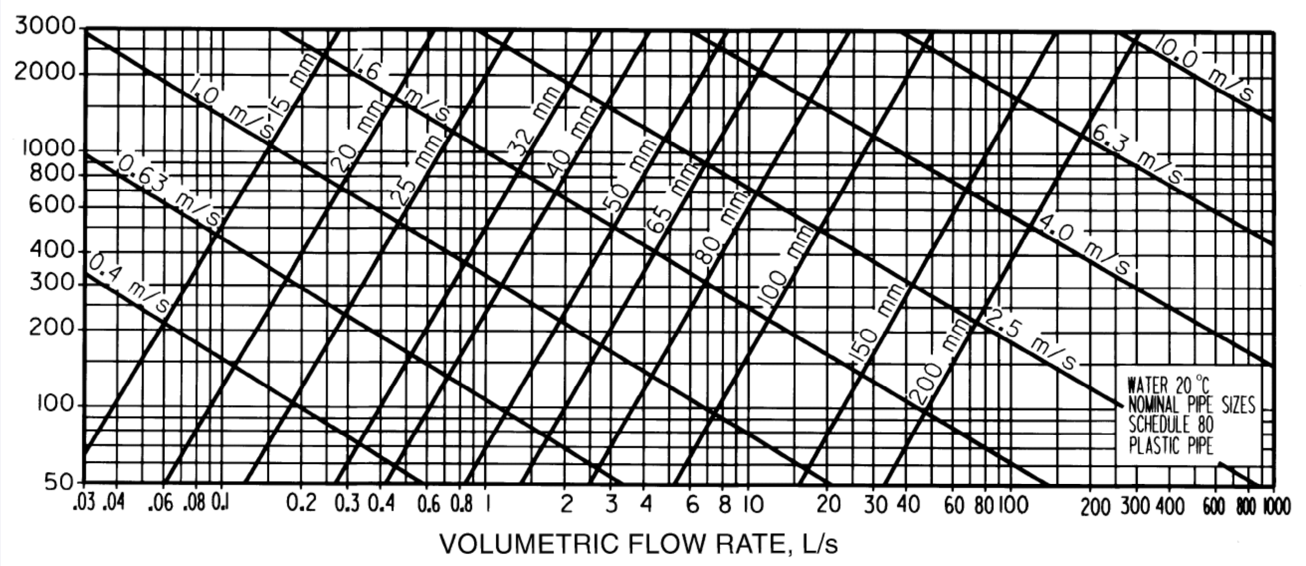
## copper pipes



# Friction chart

from ASHRAE

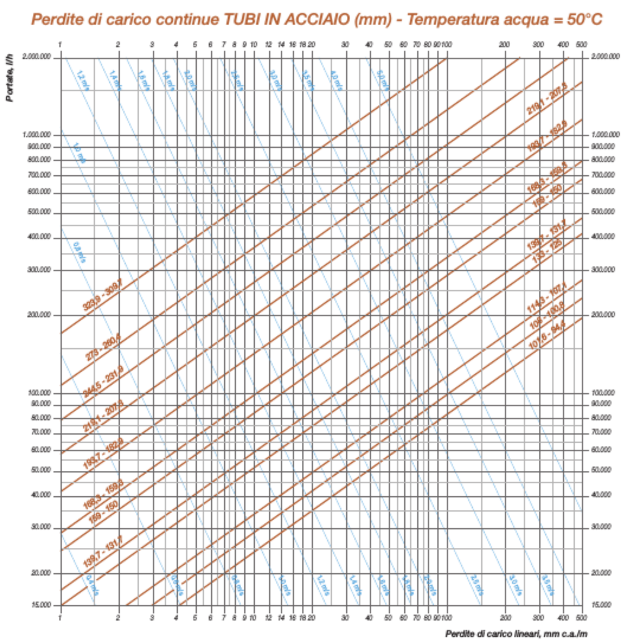
## plastic pipe

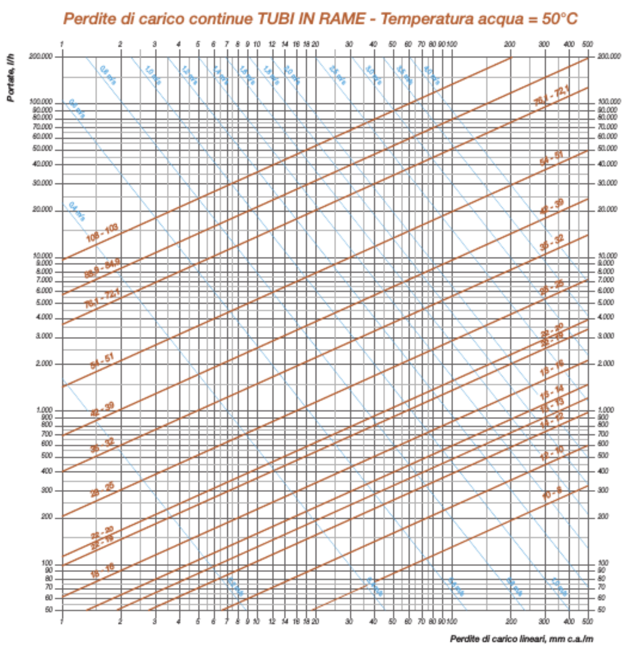





# Friction chart

from Caleffi



## pipe fitting losses

- circuits with bends fittings valves
- resistance coefficients are introduced

### Computing methods

- direct
- equivalent length
- kv factors $kv$ and $kv_{001}$



## direct method

### pressure loss in fittings

$$z = \xi \cdot \rho \cdot \frac{u^2}{2}$$

$\xi$ loss coefficient

### total pressure loss

$$\Delta z = L \cdot r + (\sum \xi) \cdot \rho \cdot \frac{u^2}{2}$$

# typical pressure loss coefficients

## quaderni caleffi

| Diametro interno tubi rame, PEad, PEX | | 8÷16 mm | 18÷28 mm | 30÷54 mm | >54 mm |
|---|---|---|---|---|---|
| Diametro esterno tubi acciaio | | 3/8"÷1/2" | 3/4"÷1" | 1 1/4"÷2" | >2" |
| Tipo di resistenza localizzata | Simbolo | | | | |
| Curva stretta a 90° r/d = 1,5 | | 2,0 | 1,5 | 1,0 | 0,8 |
| Curva normale a 90° r/d = 2,5 | | 1,5 | 1,0 | 0,5 | 0,4 |
| Curva larga a 90° r/d > 3,5 | | 1,0 | 0,5 | 0,3 | 0,3 |
| Curva stretta a U r/d = 1,5 | | 2,5 | 2,0 | 1,5 | 1,0 |
| Curva normale a U r/d = 2,5 | | 2,0 | 1,5 | 0,8 | 0,5 |
| Curva larga a U r/d > 3,5 | | 1,5 | 0,8 | 0,4 | 0,4 |
| Allargamento | | 1,0 | | | |
| Restringimento | | 0,5 | | | |
| Diramazione semplice con T a squadra | | 1,0 | | | |
| Confluenza semplice con T a squadra | | 1,0 | | | |
| Diramazione doppia con T a squadra | | 3,0 | | | |
| Confluenza doppia con T a squadra | | 3,0 | | | |
| Diramazione semplice con angolo inclinato (45°- 60°) | | 0,5 | | | |
| Confluenza semplice con angolo inclinato (45°- 60°) | | 0,5 | | | |
| Diramazione con curve d'invito | | 2,0 | | | |
| Confluenza con curve d'invito | | 2,0 | | | |

| Diametro interno tubi rame, PEad, PEX | | 8÷16 mm | 18÷28 mm | 30÷54 mm | >54 mm |
|---|---|---|---|---|---|
| Diametro esterno tubi acciaio | | 3/8"÷1/2" | 3/4"÷1" | 1 1/4"÷2" | >2" |
| Tipo di resistenza localizzata | Simbolo | | | | |
| Valvola di intercettazione diritta | | 10,0 | 8,0 | 7,0 | 6,0 |
| Valvola di intercettazione inclinata | | 5,0 | 4,0 | 3,0 | 3,0 |
| Saracinesca a passaggio ridotto | | 1,2 | 1,0 | 0,8 | 0,6 |
| Saracinesca a passaggio totale | | 0,2 | 0,2 | 0,1 | 0,1 |
| Valvola a sfera a passaggio ridotto | | 1,6 | 1,0 | 0,8 | 0,6 |
| Valvola a sfera a passaggio totale | | 0,2 | 0,2 | 0,1 | 0,1 |
| Valvola a farfalla | | 3,5 | 2,0 | 1,5 | 1,0 |
| Valvola a ritegno | | 3,0 | 2,0 | 1,0 | 1,0 |
| Valvola per corpo scaldante tipo diritto | | 8,5 | 7,0 | 6,0 | — |
| Valvola per corpo scaldante tipo a squadra | | 4,0 | 4,0 | 3,0 | — |
| Detentore diritto | | 1,5 | 1,5 | 1,0 | — |
| Detentore a squadra | | 1,0 | 1,0 | 0,5 | — |
| Valvola a quattro vie | | 6,0 | | 4,0 | |
| Valvola a tre vie | | 10,0 | | 8,0 | |
| Passaggio attraverso un radiatore | | 3,0 | | | |
| Passaggio attraverso una caldaia | | 3,0 | | | |



# equivalent length

### virtual length of pipe

$$L_{tot} = L + \sum L_E$$

$L_{tot}$ virtual length of pipe

$L$ real length of pipe

$L_E$ equivalent length

### total pressure losse

$$\Delta z = L_{tot} \cdot r$$

## equivalence

### direct method

$$\Delta p_c = \xi \cdot \frac{1}{2} \cdot \rho \cdot u^2$$
$$\Delta p_c = r \cdot L_E$$
$$r = \xi \cdot \frac{\rho \cdot u^2}{2 \cdot D}$$
$$L_E = \frac{\xi \cdot D}{F_A}$$



## pressure loss in valves

### Flow coefficient $K_v$

$$G = K_v \sqrt{\Delta p} \qquad G\ [\mathrm{m^3/h}]; \quad \Delta p\ [\mathrm{bar}]$$

### for reduced flow rates and pressures $K_{v0,01}$

$$G = K_{v0,01} \sqrt{\Delta p \cdot 100} \qquad G\ [\mathrm{l/h}]; \quad \Delta p\ [\mathrm{bar}]$$

$K_v$ volumetric flow rate in $\mathrm{m^3/h}$ obtained with $\Delta p = 1$ bar.

$K_{v0,01}$ volumetric flow rate in $\mathrm{l/h}$ with $\Delta p = 0,01$ bar.

## pressure loss in valves

### imperial units

**Flow coefficient $C_v$**

$$G = C_v\sqrt{\Delta p} \qquad G\ [\mathrm{GPM}]; \quad \Delta p\ [\mathrm{psi}]$$

$C_v$ volumetric flow rate in $\mathrm{gpm}$ obtained with $\Delta p = 1\ \mathrm{psi}$.

$\mathrm{gpm}$ gallon per minute

$\mathrm{psi}$ pounds square inch, 1 psi $=$6894.8 $\mathrm{Pa}$

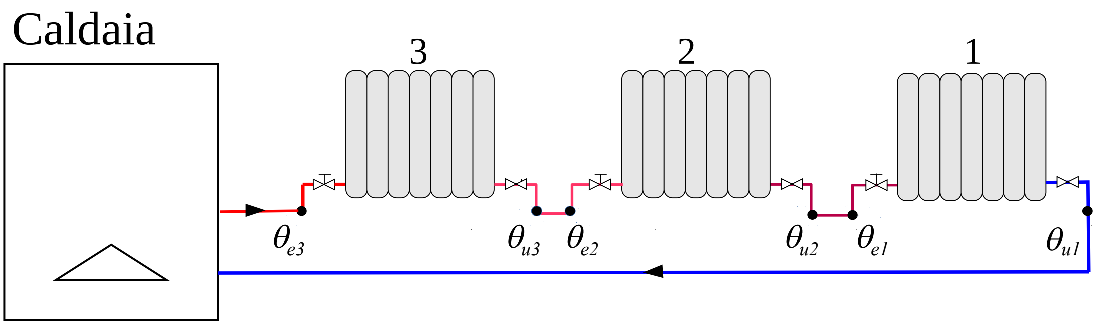


## pipe layout

There are several ways to connect heating terminals to the generator: for domestic plants, 3 layouts are mainly used:

- **one pipe**
- **two pipes**
- **manifold**, dual distribution manifolds (also calle “modul”)

# one pipe distribution




## Temperature

$$
\begin{aligned}
\Delta\theta_{a3} &= (\theta_{e3} + \theta_{u3})/2 - \theta_{aria} \\
\Delta\theta_{a2} &= (\theta_{e2} + \theta_{u2})/2 - \theta_{aria} \\
\Delta\theta_{a1} &= (\theta_{e1} + \theta_{u1})/2 - \theta_{aria} \\
\Delta\theta_{a3} &> \Delta\theta_{a2} > \Delta\theta_{a1}
\end{aligned}
$$



# one pipe distribution

## Characteristics

- low installation cost
- requires special attention in connecting radiators
- four way valves or bypass
- temperature drop computed on the whole ring
- the temperature of the radiator changes along the ring
- requires high flow rates to minimize the temperature differences

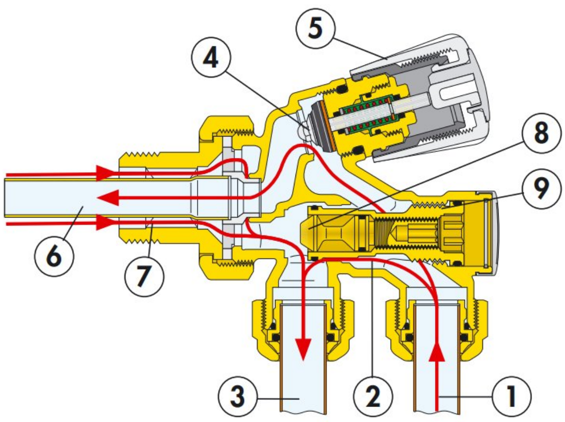
# bypass

- il bypass allows the fluid to pass over each radiator
- two flows, one in the radiator and the other in the bypass



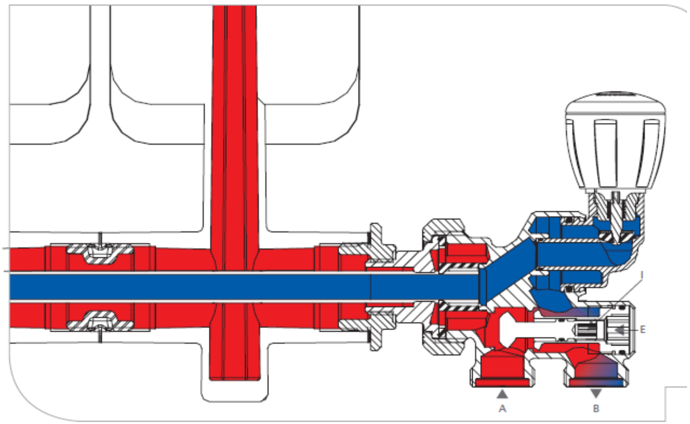
# four way valve

- allows to regulate the flow inside the radiator.
- again twou flows can be identified, one in the radiator and the other in the bypass of the valve.

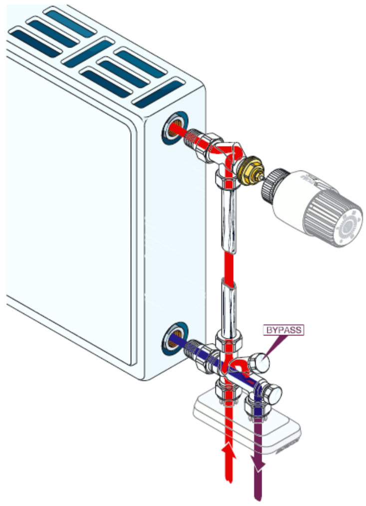



from Caleffi

# four way valve



Giacomini




IVAR



# pipe sizing

each circuit is analyzed at once:

1. heat $\Phi_A$ heat exchanged along the whole ring it is the sum of the heat exchanged by each $\Phi_T$ heat emitter (radiator or fan coil).

$$\Phi_A = \sum_J \Phi_T$$

2. selection of $\Delta\theta_A$, temperature difference, between 10 and 15 K.
3. compute mass flow rate, $G_A$:

$$G_A = \frac{\Phi_A}{c \cdot \Delta\theta_A}$$

4. with the mass or volumetric flow rate select pipe diameters

## pressure loss

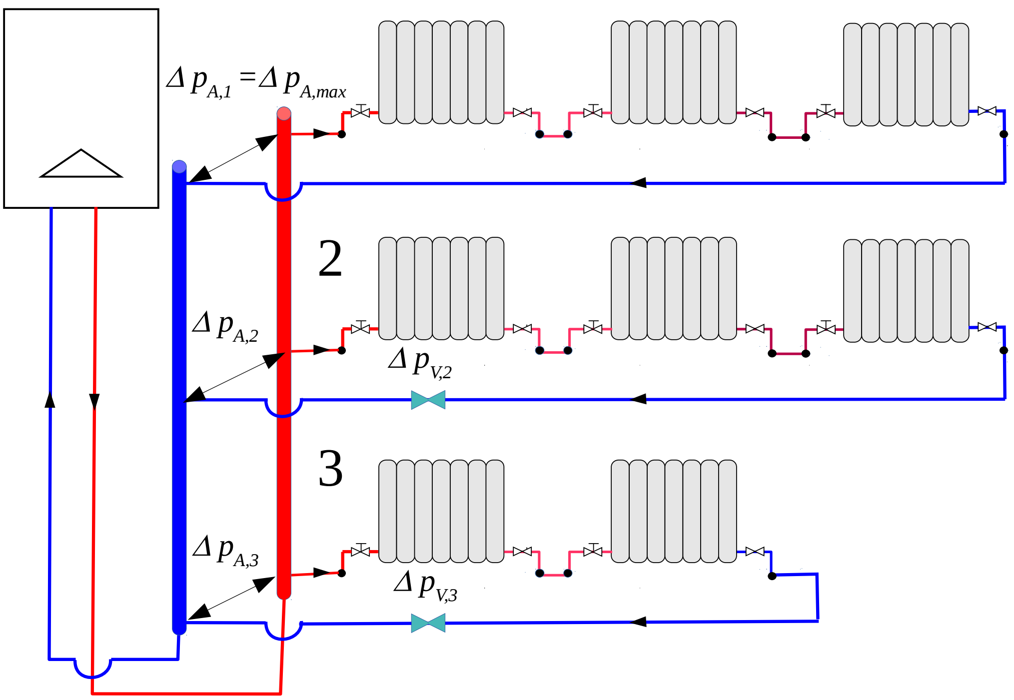
once sized the pipes compute totale loss:

$$\Delta p_A = r_A \cdot L_A + \sum_i \Delta p_i + \sum_j \xi_j \cdot \rho \cdot \frac{v_A^2}{2}$$

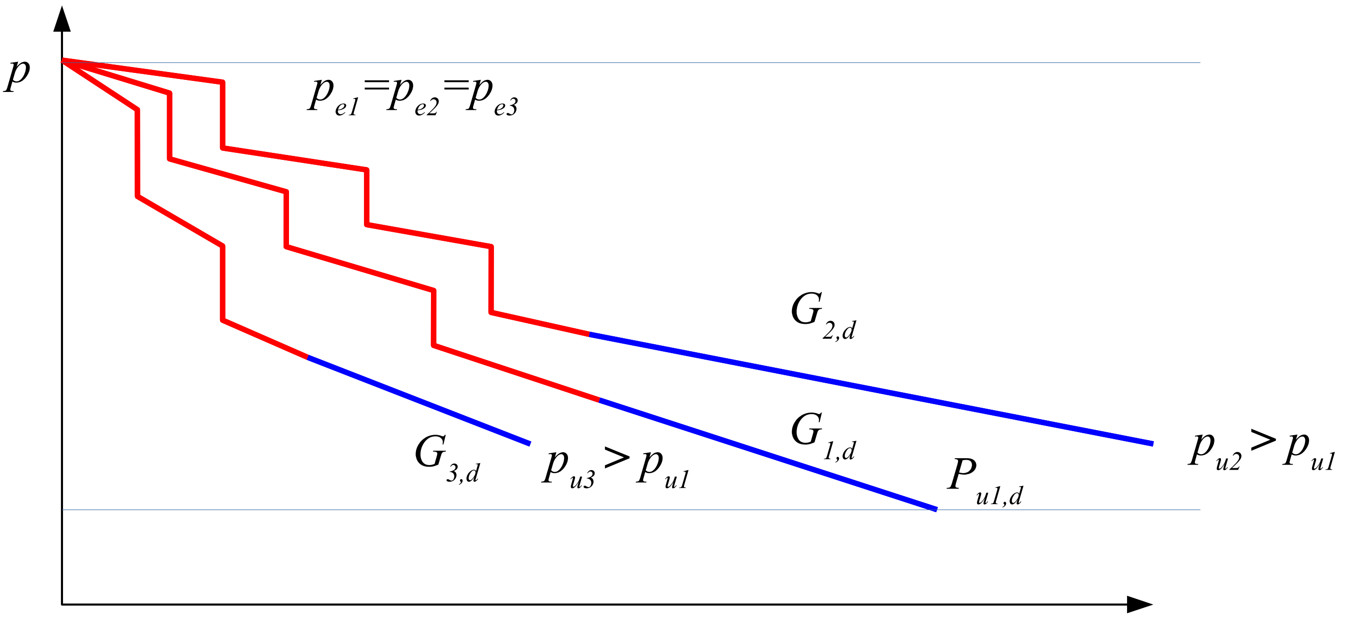
| | |
|---|---|
| $\Delta p_A$ | total loss of the ring |
| $r_A$ | pressure loss for unit length |
| $L_A$ | pipe length of ring |
| $\Delta p_i$ | pressure loss for each emitter |
| $\xi_j$ | pressure loss coefficient |
| $v_A$ | fluid velocity |



## layout with additional circuits

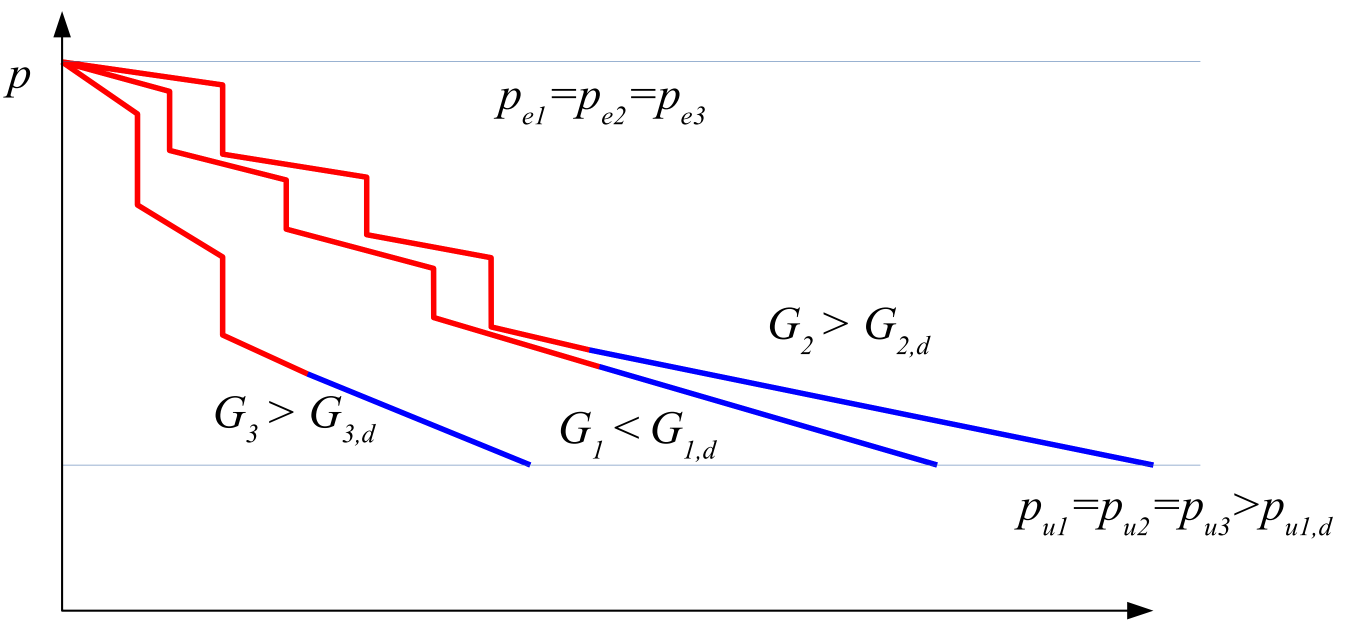


## One pipe circuits in parallel
### Design pressure distribution



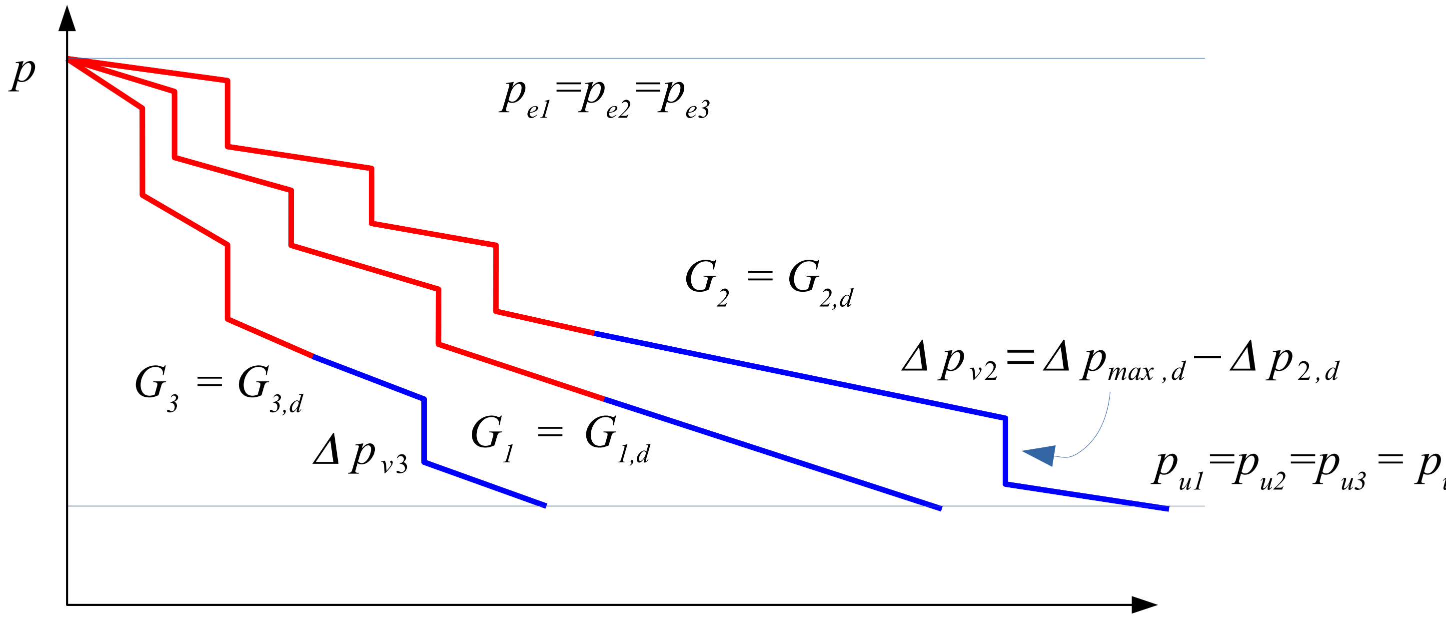



## One pipe circuits in parallel
### Design pressure distribution



## One pipe circuits in parallel

Design pressure distribution, with additional balancing valves






## Sizing different rings in parallel

- pressure losses are different for each ring:
- Additional pressure loss $\Delta P_V$ for the rings with lower pressure loss

$$\Delta p_{V,i} = \Delta p_{A,max} - \Delta p_{A,i}$$

- compute the $k_v$ or the $k_{v001}$ of the balancing valve

$$K_{V,i} = \frac{G_i}{\sqrt{\Delta P_{V,i}}}$$

- Without valves, the fluid flow is large in the rings with lower pressure loss.

# Change of flow rate with different pressures

Simple formula for computing the flow rate with different pressure losses

$$r = \frac{\Delta p}{L} = F_a \frac{1}{D} \rho \frac{u^2}{2}$$

tubi di media scabrezza

$$F_a = 0,07 \ Re^{-0,13} \ D^{-0,14} \sim u^{-0,13}$$

$$\Delta p \sim u^{1,87}$$

$$G \sim u \sim \Delta p^{\frac{1}{1,87}}$$

$$G' = G \left( \frac{\Delta p'}{\Delta p} \right)^{\left(\frac{1}{1,87}\right)}$$

### considering possible fittings

$$G' = G \left( \frac{\Delta p'}{\Delta p} \right)^{0,525}$$



# Two pipe systems

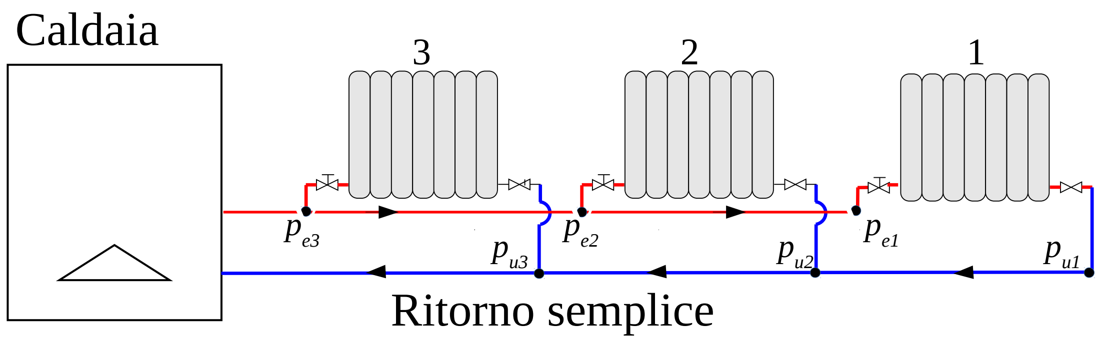
### Direct Return

- classical distribution
- used togheter with other distribution systems
- layout requires balancing of flow rtes
- heat emitters near the generator are subjected to higher pressures diffeneces
- balancing valves are required

### reverse return

- classical distribution
- used togheter with other distribution systems
- In a reverse-return system, the piping lengths for each branch circuit, including the main and branch pipes, are almost equal
- pressure difference is almost constant
- higher pipe length, cost and space problems

## two pipes direct return



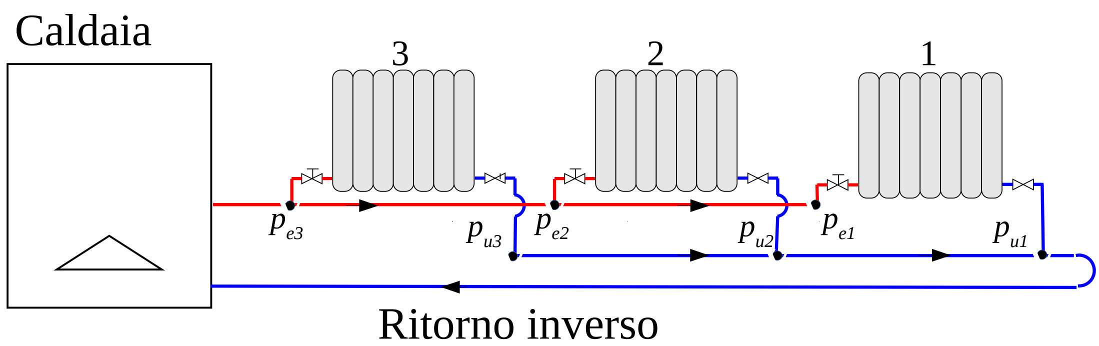

### differential pressure

$$
\begin{aligned}
p_{e3} > \quad & p_{e2} \quad > p_{e1} \\
p_{u3} < \quad & p_{u2} \quad < p_{u1}
\end{aligned}
$$

$$\Delta p_3 = (p_{e3} - p_{u3}) > \Delta p_2 = (p_{e2} - p_{u2}) > \Delta p_1 = (p_{e1} - p_{u1})$$



## two pipes reverse return




### available differential pressure

$$
\begin{aligned}
p_{e3} > \quad & p_{e2} \quad > p_{e1} \\
p_{u3} > \quad & p_{u2} \quad > p_{u1}
\end{aligned}
$$

$$\Delta p_3 = (p_{e3} - p_{u3}) \simeq \Delta p_2 = (p_{e2} - p_{u2}) \simeq \Delta p_1 = (p_{e1} - p_{u1})$$

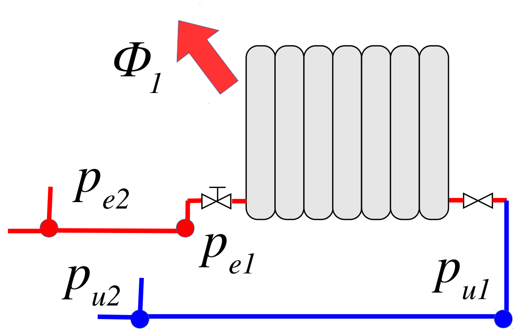
# two pipes direct return
## sizing and balancing

### first terminal

1. compute flow rate and pipe diameter
2. size the terminal computing design pressure lossess $\Delta p_{1,d}$

$$G_1 = \frac{\Phi_1}{c_l \cdot (\theta_{e1} - \theta_{u1})}$$

$$\Delta p_1 = r_1 \cdot L_1 + \sum_j \xi_{1,j} \cdot \frac{1}{2} \cdot \rho \cdot u_1^2$$






# two pipes direct return
## sizing and balancing

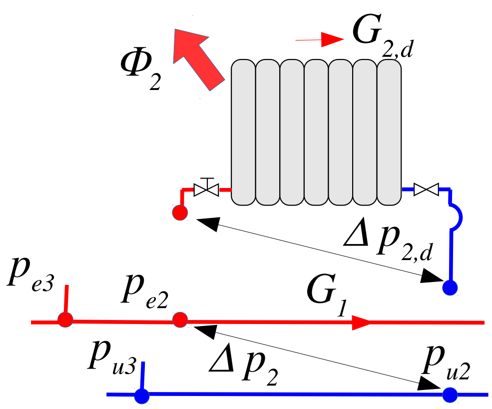
### second terminal

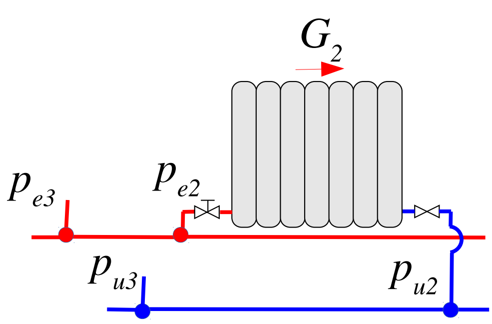
1. compute available pressure difference
2. size the terminal computing the design pressure loss $\Delta p_{2,d}$
3. balnce the system using the available pressure difference $\Delta p$
4. if the new flow rate is too large, add an additional pressure loss $\Delta p_{v,2}$

$$\Delta p_2 = \Delta p_1 + r_{21} \cdot L_{21} + \sum_j \xi_{21,j} \cdot \frac{1}{2} \cdot \rho \cdot u_{21}^2$$

$$G_2 = G_{d,2} \cdot \left(\frac{\Delta p_2}{\Delta p_{d,2}}\right)^{0.525}$$

$$\Delta p_{v,2} = \Delta p_2 - \Delta p_{2,d}$$



# two pipe direct return

## sizing and balancing

### third terminal

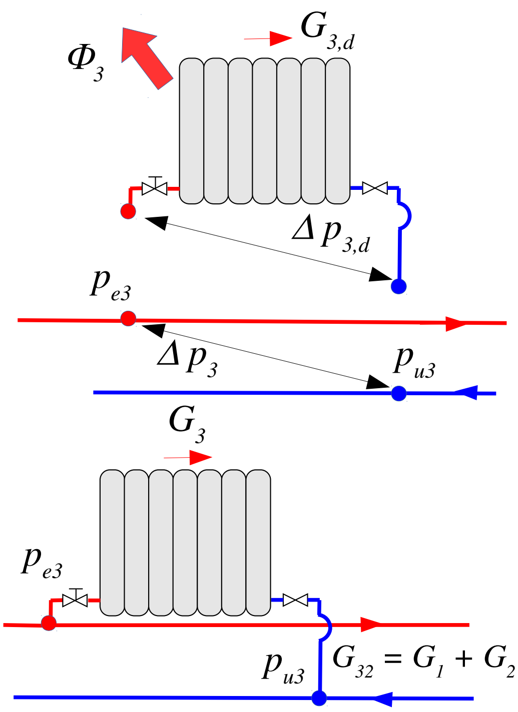
1. compute flow rate as the sum of the one of the two previous terminals
2. size pipe diameter using the pressure loss $\Delta p_3$
3. size the terminal and compute the design pressure loss $\Delta p_{3,d}$
4. balance the flow rate using the available pressure $\Delta p_3$
5. add a pressure loss if the fow rate is too large $\Delta p_{v,3}$

$$G_{32} = G_1 + G_2$$

$$\Delta p_3 = \Delta p_2 + r_{32} \cdot L_{32} + \sum_j \xi_{32,j} \cdot \frac{1}{2} \cdot \rho \cdot u_{32}^2$$

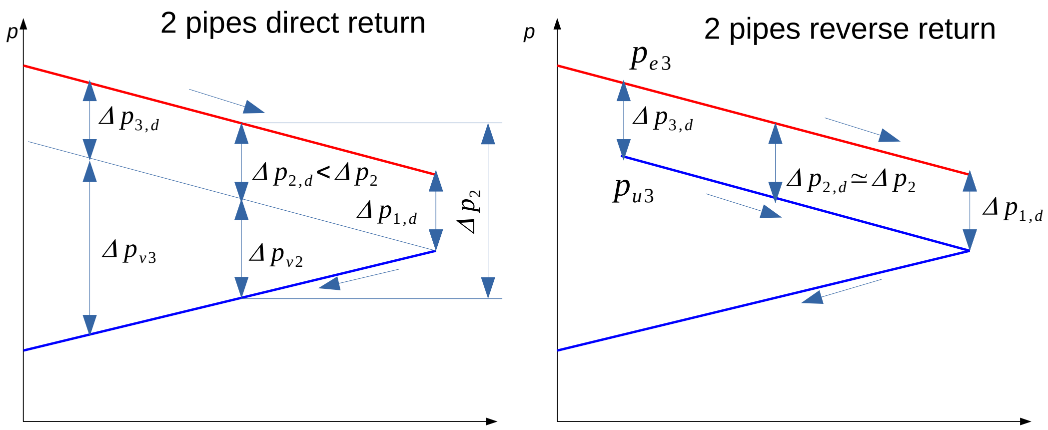
$$G_3 = G_{d,3} \cdot \left( \frac{\Delta p_2}{\Delta p_{3,2}} \right)^{0.525}$$

$$\Delta p_{v,3} = \Delta p_3 - \Delta p_{3,d}$$






# two pipes direct and reverse return



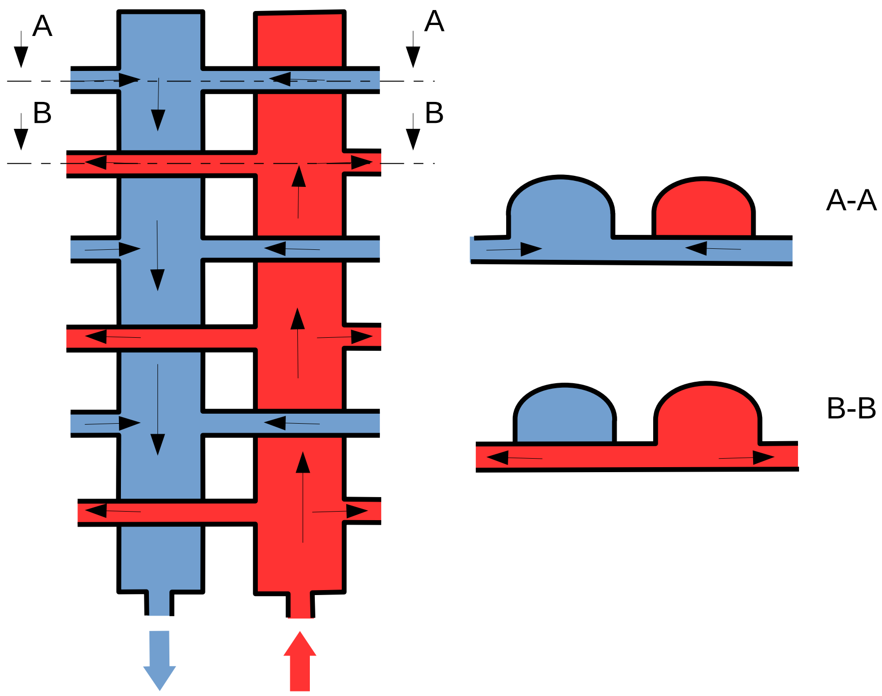
# co-planar manifold

## Caratteristiche

- used in new installations
- terminals connected in parallel
- sizing similar to the two pipes system
- requires balancing for correct function

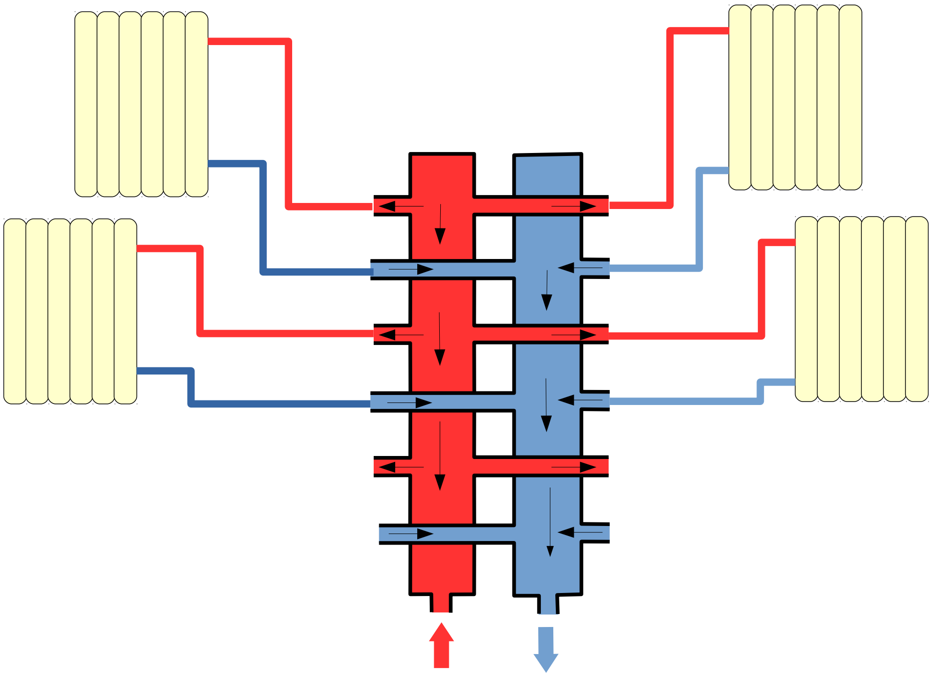


# co-planar manifold



# co-planar manifold

## Plant system

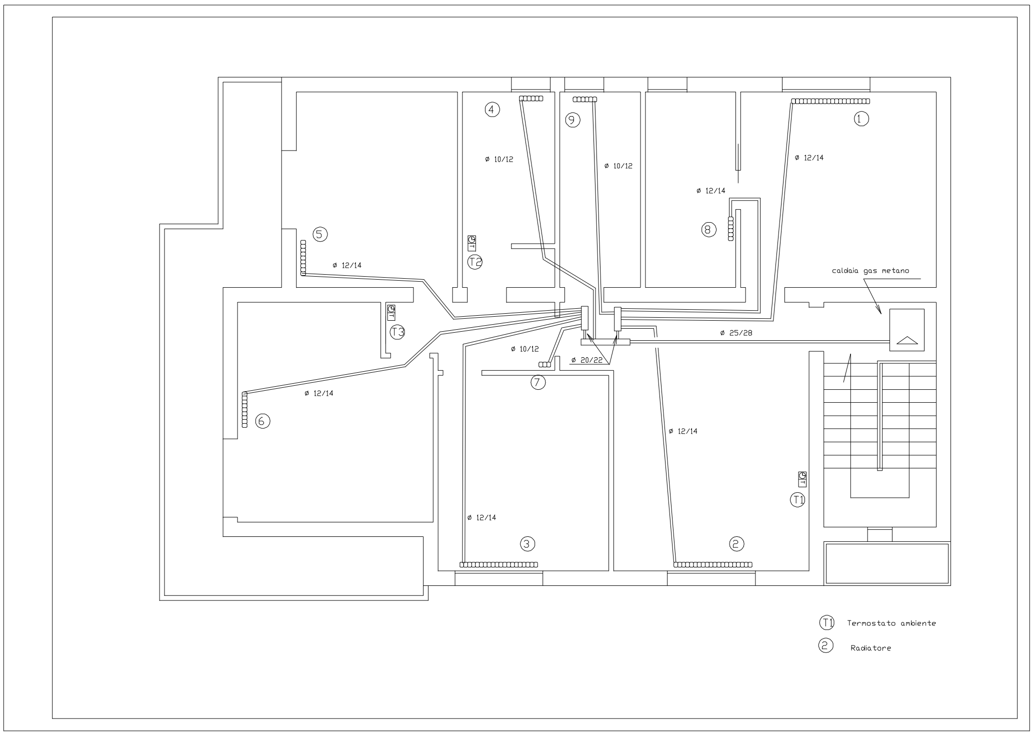




# Co-planar manifold

## example

## co-planar manifold






## co-planar manifold

### Sizing

#### fixed pipe diameter

- Pipe diameter is fixed
- each terminal must be balanced
- different flow rates and different temperature differences
- size the terminals using the mean temperature difference

#### sizing with predefined diameter and temperature difference

- set pipe diameter
- balance each terminal
- compute the additional pressure loss
- the pressure loss can be obtained with a different pipe diameter

# Radiator valves and lockshields

- Radiators are equipped with valves and lockshield
- lockshield can be used to balance water rings
- radiator valves can be either manual or with thermostatic control heads

## manual control

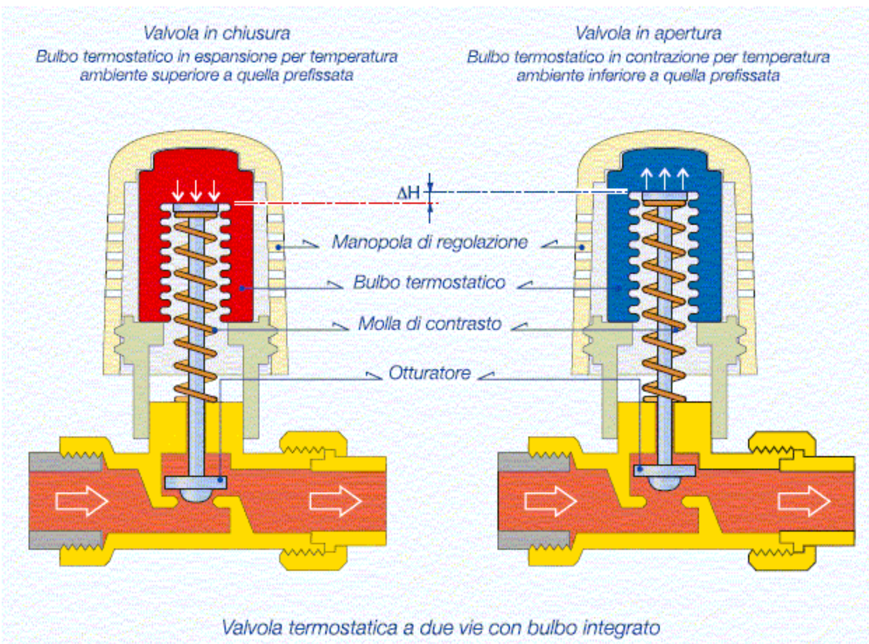
- the position of an obturator sets the pressure loss
- they are used to open or close a circuit, it is not possible to control the temperature

## thermostatic control

- the opening of the obturator is controlled by a thermostatic head
- when the room temporature approaches the set value the obturator closes
- this can lead to unbalanced plants



# Thermostatic valve head



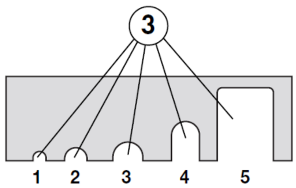
Valvola termostatica a due vie con bulbo integrato

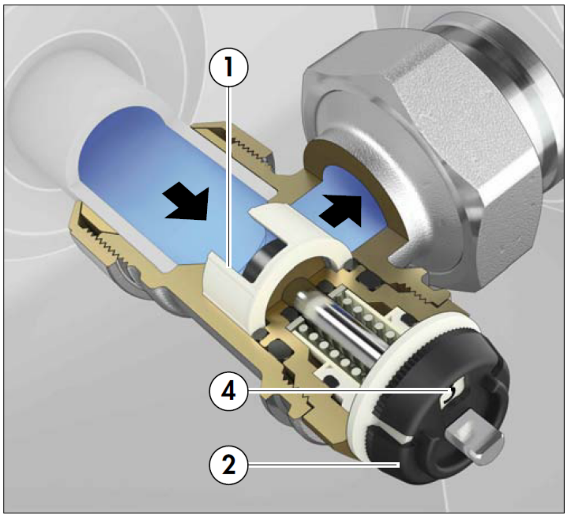
# preset valves

**Operating principle**

The convertible radiator valves are equipped with an internal device (1) for pre-setting the head loss hydraulic characteristics. Specific passage cross sections (3) can be selected by means of the control nut (2), in order to generate the required resistance to the motion of the medium.
Each passage cross section determines a specific Kv value for the creation of the head loss, which corresponds to a setting position on a graduated scale (4).
Depending on the position in the system, the valve can be pre-setted so as to obtain an immediate balancing of the hydraulic circuit, valid for both manual and thermostatic operation.

**Flat development of the internal (1) pre-setting device**



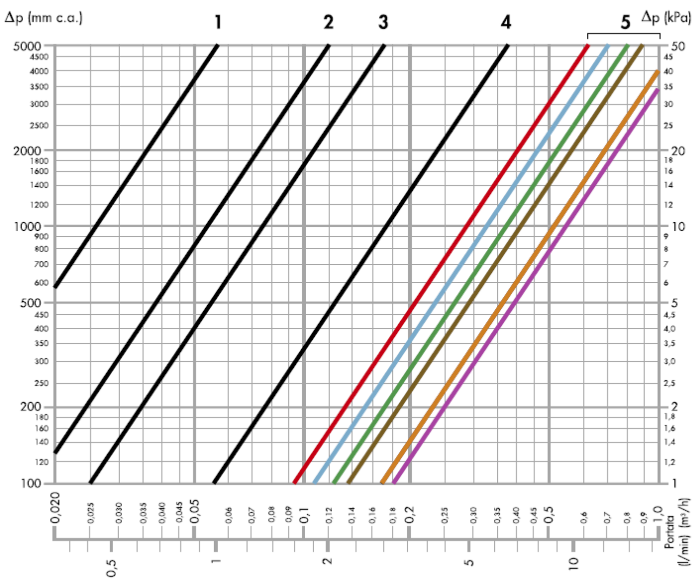

fonte Caleffi



# Characteristic pressure loss diagram

## Manual control

**Valvole termostatizzabili preregolabili con manopola manuale**



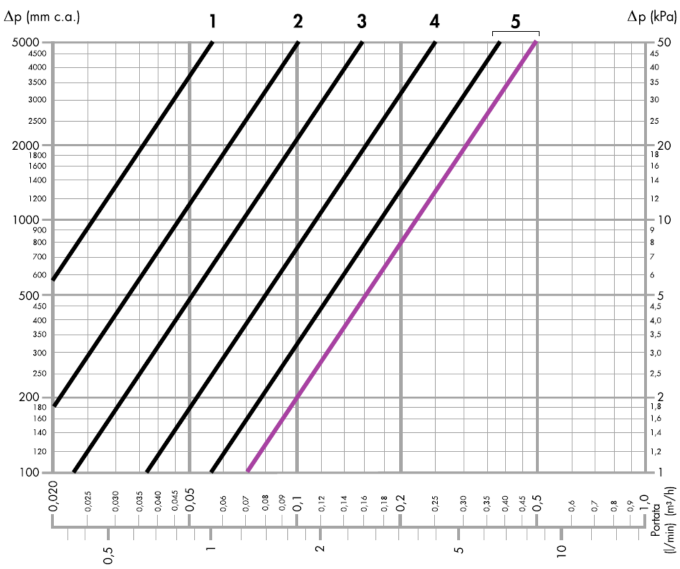

| Posizione di preregolazione | Kvs (m³/h) 3/8" squadra | 3/8" diritta | 1/2" squadra | 1/2" diritta | 3/4" squadra | 3/4" diritta |
|---|---|---|---|---|---|---|
| 1 | **0,08** | 0,08 | 0,08 | 0,09 | 0,12 | 0,12 |
| 2 | **0,17** | 0,17 | 0,17 | 0,19 | 0,22 | 0,22 |
| 3 | **0,25** | 0,25 | 0,25 | 0,27 | 0,41 | 0,41 |
| 4 | **0,55** | 0,55 | 0,55 | 0,56 | 0,95 | 0,93 |
| 5 | 1,30 | 0,90 | 1,40 | 1,00 | 1,80 | 1,70 |

fonte Caleffi

# Characteristic pressure loss diagram

## Thermostatic control

**Valvole termostatizzabili preregolabili con comando termostatico banda proporzionale 2K**




| Posizione di preregolazione | Kv ($m^3/h$) (Banda proporzionale 2K)** 3/8" squadra | 3/8" diritta | 1/2" squadra | 1/2" diritta | 3/4" squadra | 3/4" diritta |
|---|---|---|---|---|---|---|
| 1 | **0,08** | 0,08 | 0,09 | 0,09 | 0,12 | 0,12 |
| 2 | **0,15** | 0,15 | 0,16 | 0,16 | 0,20 | 0,20 |
| 3 | **0,22** | 0,22 | 0,23 | 0,23 | 0,32 | 0,32 |
| 4 | **0,35** | 0,35 | 0,36 | 0,36 | 0,50 | 0,50 |
| 5 | 0,50 | 0,50 | 0,55 | 0,55 | 0,72 | 0,72 |

fonte Caleffi



# Sizing with predefined pipe diameter and temperature difference

**procedure**

1. for each circuit compute the flow rate
2. define pipe diameter and fluid velocity $u_j$
3. compute pressure losses, do not consider the pressure loss of valves
4. with preset valves, add the pressure loss with full open valve
5. for each circuit determine the required pressure loss
6. find the set position of the valve

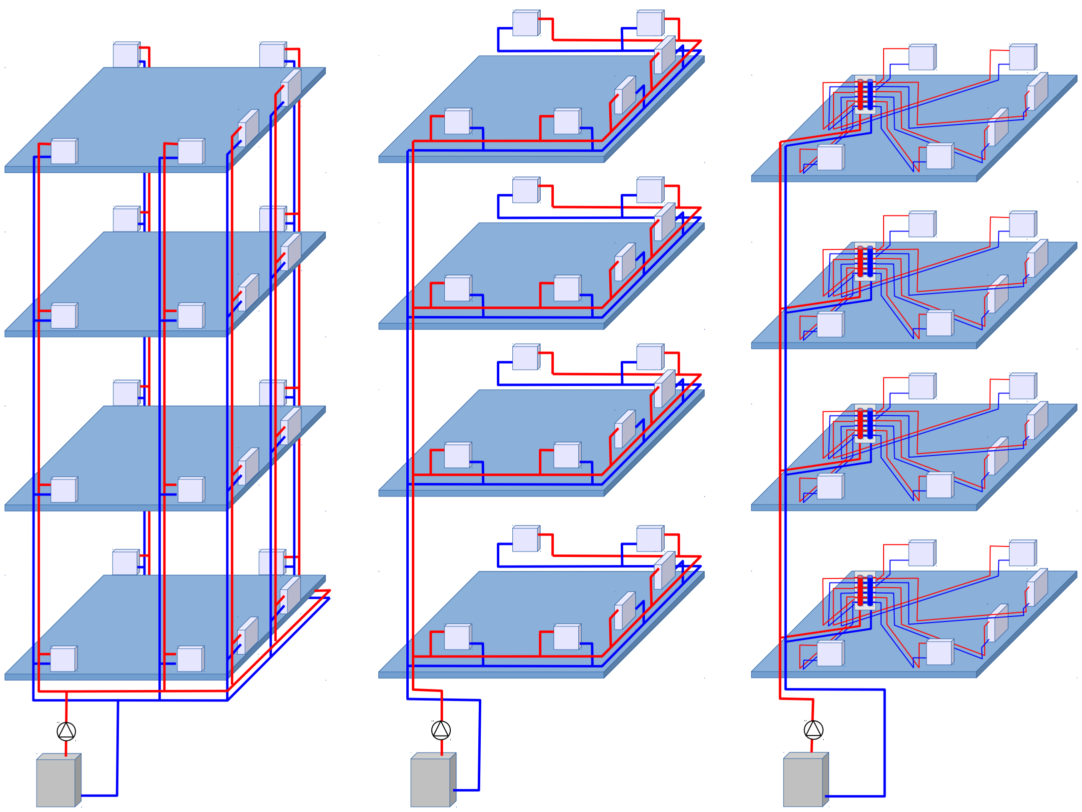
# Sizing with predefined pipe diameter and temperature difference

## procedure

1. $G_j = \frac{\Phi_j}{c_w \cdot \Delta\theta \rho}$ design flow
2. $u_j = \frac{G_j \cdot 4}{d_j^2 \cdot \pi}$ fluid velocity
3. $\Delta p_{c,j} = r_j \cdot L_j + \sum_k \frac{1}{2} \cdot \rho \cdot u_j^2$ circuit pressure loss
4. $\Delta p_{tot,max} = \Delta p_{c,max} + \Delta p_V$ maximum pressure loss
5. $\Delta p_{V,j} = \Delta p_{tot,max} - \Delta p_{c,j}$ pressure loss for each valve
6. $k_{v,j} = \frac{G_j}{\sqrt{\Delta p_{V,j}}}$ using a diagram $\Delta p_{V,j}$



# Vertical distribution plants



column two pipes co-planar manifold

## radiant systems

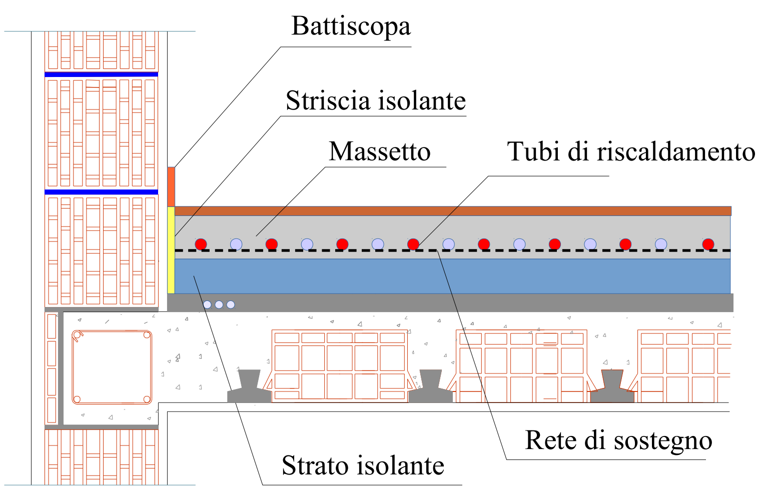
Inlet temperature for heating $30 \div 45^o C$, Can be used for cooling during summer season Can be installed in:

- **floor** for heating and cooling, the preferred solution in domestic homes.
- **wall** heating and cooling, furniture problems
- **ceiling** ideal solution for cooling
- **standards** the system is described in standards of the series UNI EN 1264



## heated floor

- pipes embedded in concrete slab.
- pipes in plastic material must be fixed during installation
  - metal net with clips
  - on preformed insulation material



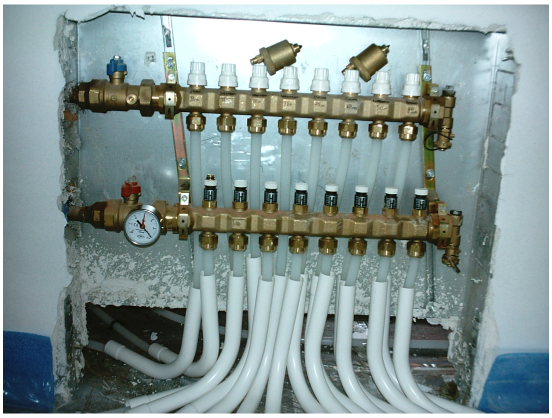
# Heated floors

## examples



# heated floors

## Manifold



## operative temperature

$$\theta_i = \frac{\theta_{Ai} + \theta_{mr}}{2}$$

dove $\theta_{Ai}$ internal temperature, mean radiant temperature $\theta_{mr}$

$$\theta_{mr} = \left[\sum_{j=1}^{n} (\theta_{sj} + 273)^4 \cdot F_j\right]^{\frac{1}{4}} - 273,15$$

dove

| | |
|---|---|
| $n$ | number of walls |
| $\theta_{sj}$ | temperature of j-th wall or ceiling |
| $F_j$ | form factor of j-th wall |
| $A_j$ | Area of j-th wall |

$$\theta_{mr} \approx \frac{\sum_{j=1}^{n} \theta_{sj} \cdot A_j}{\sum_{j=1}^{n} A_j}$$



## specific thermal output

$$q = B \cdot \prod_i a_i^{m_i} \cdot \Delta\theta_H$$

| | |
|---|---|
| $q$ | specific thermal output |
| $B =$ | system dependent coefficient |
| $a_i,\ m_i$ | parameters of the floor |
| $\Delta\theta_H$ | mean logarithmic temperature difference |

# temperature difference between the heating medium and room temperature

the mean logarithmic temperature difference is expressed as:

$$\Delta\theta_H = \frac{\theta_V - \theta_R}{ln(\frac{\theta_V - \theta_i}{\theta_R - \theta_i})}$$

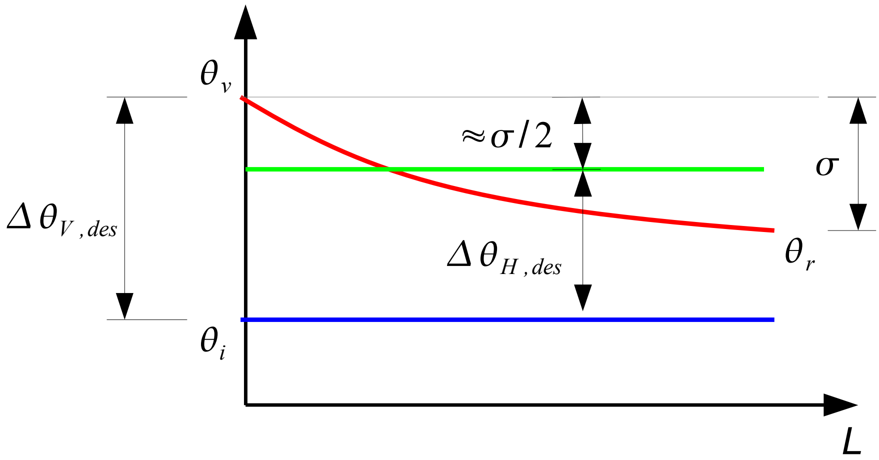
con

$\theta_V$ inlet water temperature

$\theta_R$ outlet water temperature



# explanation of the terms

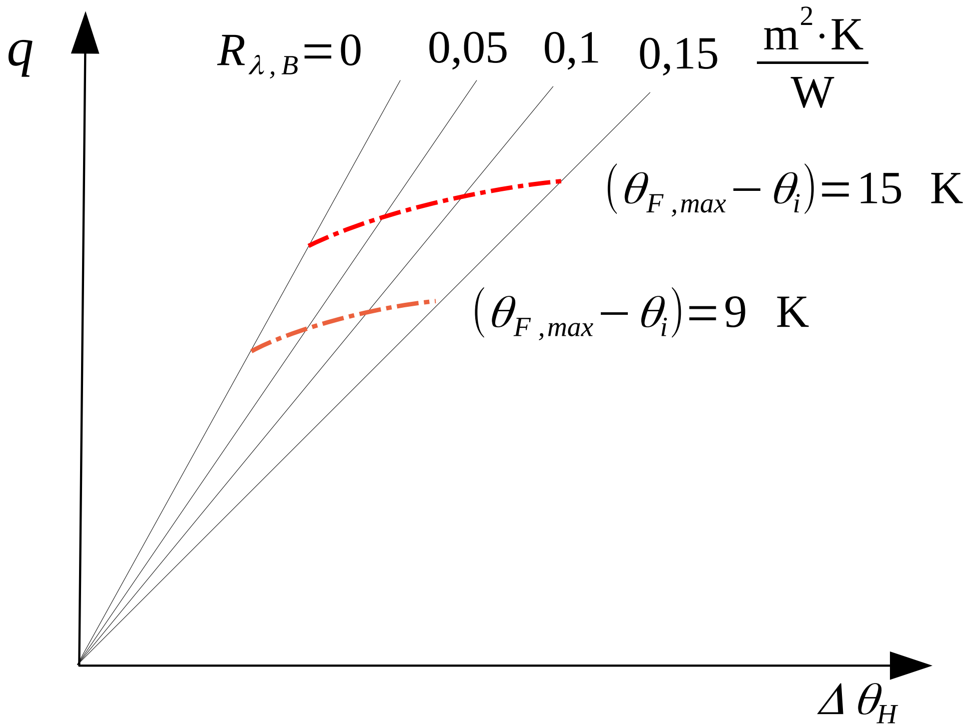


## Parameters driving the heat output

- Il **Pipe spacing**, T;
- Lo **screed thickness**, $S_V$, measured from the plane of pipes
- La **screed conductivity**, $\lambda_E$;
- La **heat resistance of floor covering**, $R_{\lambda B}$;
- Il **pipe external diameter**, D, including sheathing for $O_2$ migration:
- **additional conductive elements**, $K_{WL}$;
- Il **contact resistance between screed and pipe**.

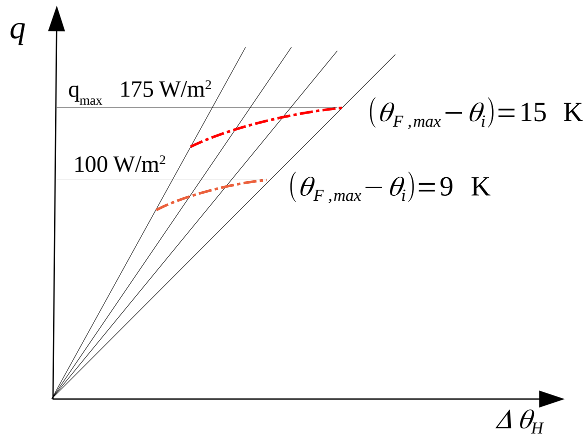


## specific thermal output diagram



# Limits of specific thermal output

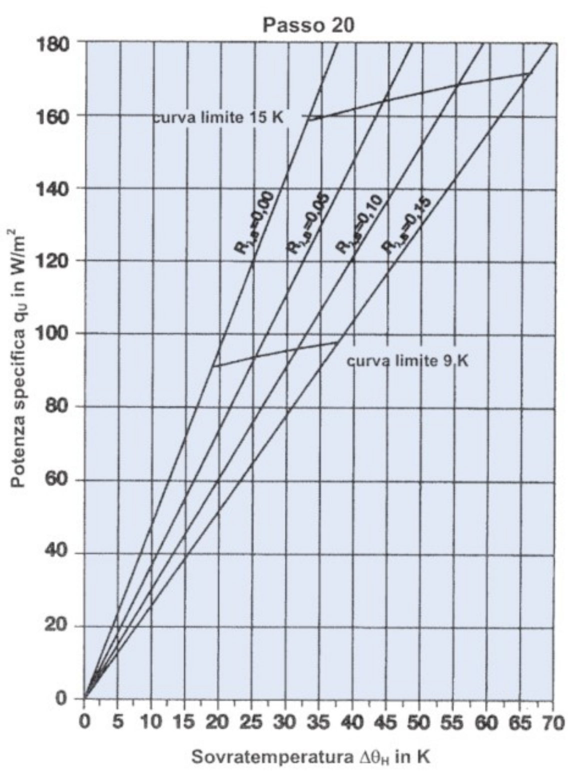
- standard prescribes the **maximum floor temperature** $29^oC$,
- typically the maximum permissible specific thermal output is about $100\ \mathrm{W/m^2}$.
- in peripheral areas the maximum temperature could reach $35^oC$
- the maximum permissible specific thermal output is $175\ \mathrm{W/m^2}$
- usually specific thermal output reaches $80 - 90\ \mathrm{W/m^2}$.

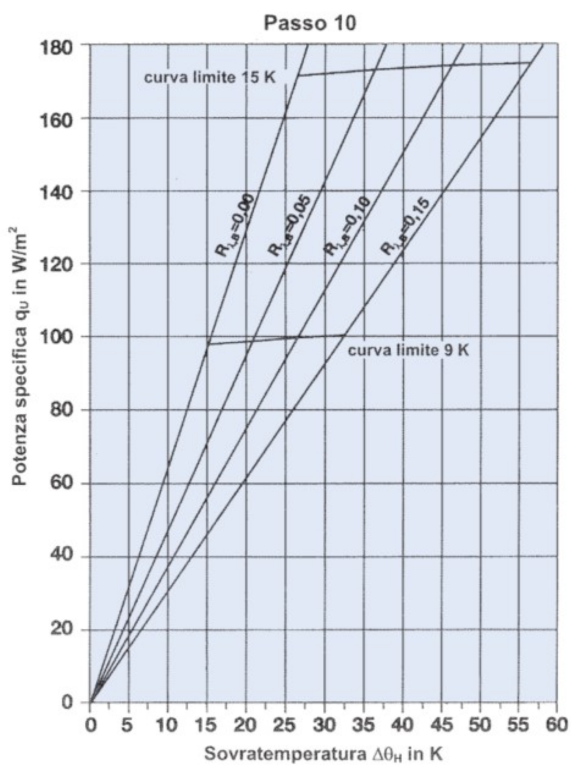




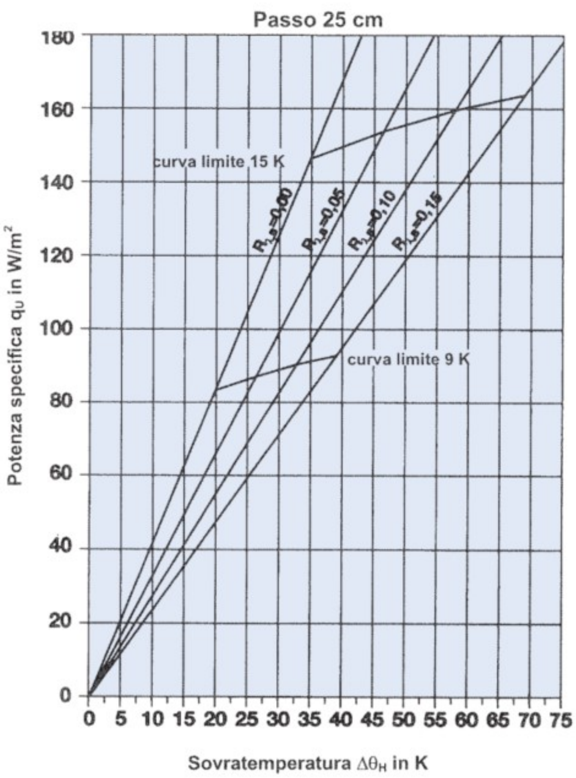

# Example of design thermal output

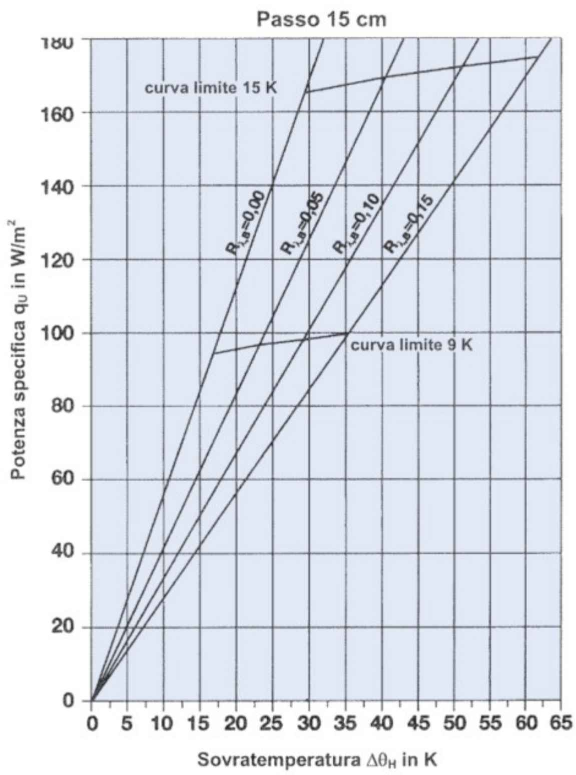
from Buderus



# Example of design thermal output
from Buderus






# Sizing

for each environment the specific thermal output is computed as:

$$q = \frac{\Phi_{Nf}}{A_f}$$

$\Phi_{Nf}$ heat output required for each environment

$A_f$ surface of environment

with peripheral area

$$A_F = A_{F\ permetrale} + A_{F\ calpestabile}$$

$$q = \frac{A_{F,perim}}{A_F} q_{perim} + \frac{A_{F,calp}}{A_F} q_{calp}$$

## water temperature

design temperature difference

$$\Delta\theta_H = \frac{\theta_V - \theta_R}{ln(\frac{\theta_V - \theta_i}{\theta_R - \theta_i})}$$

design temperature difference

$$\sigma = \theta_V - \theta_R$$

design input temperature: $\theta_{V,des}$
input temperature difference

$$\Delta\theta_{V,des} = \theta_{V,des} - \theta_i$$



## determination of design flow temperature

**$\sigma/\Delta_H < 0,5$**

$\sigma = 5$ K $\Longrightarrow \Delta\theta_H > 10$ K

$$\Delta\theta_{V,des} \leq \Delta\theta_{H,des} + \frac{\sigma}{2}$$

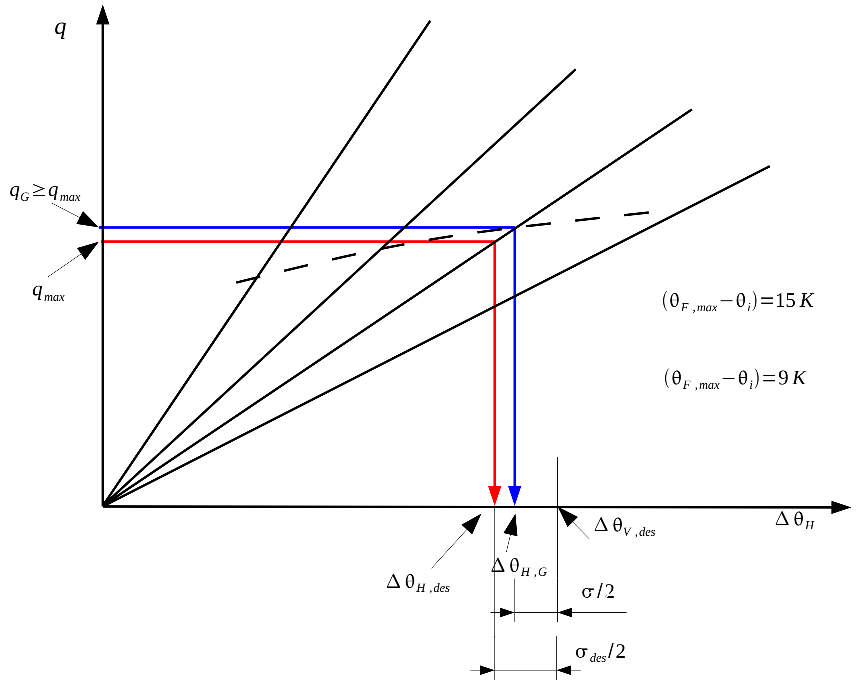
**$\sigma/\Delta\theta_H \geq 0,5$**

$\sigma = 5$ K $\Longrightarrow \Delta\theta_H \leq 10$ K

$$\Delta\theta_{V,des} \leq \Delta\theta_{H,des} + \frac{\sigma}{2} + \frac{\sigma^2}{12\Delta\theta_{H,des}}$$

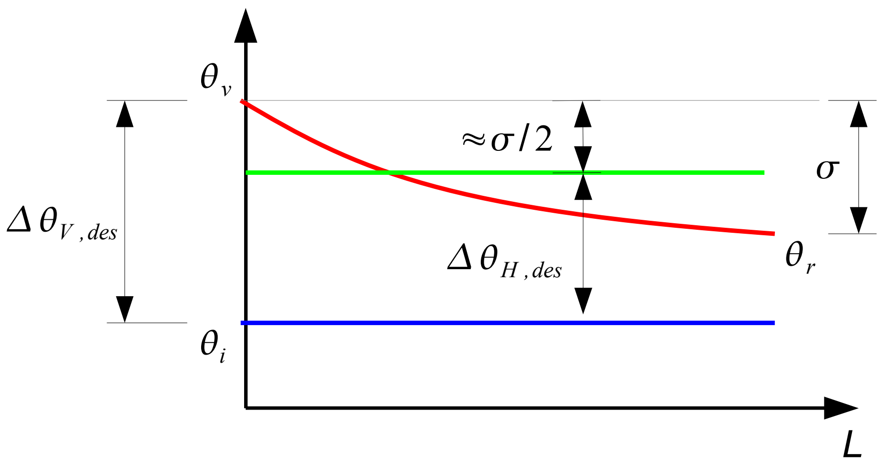
,

# Determination of design temperature difference






# Explanation of terms



# Direct formula

$$\Delta\theta_H = \frac{\theta_V - \theta_R}{ln(\frac{\theta_V - \theta_i}{\theta_R - \theta_i})}$$

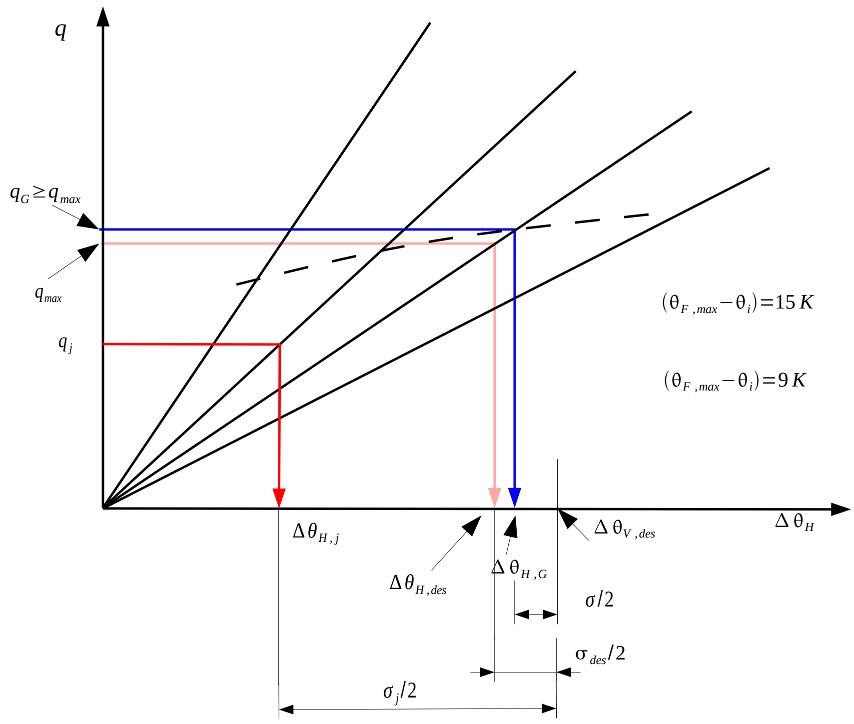
$$\Delta\theta_H = \frac{\sigma}{ln(\frac{\Delta\theta_{V,des}}{\Delta\theta_{V,des} - \sigma})}$$

$$\frac{\Delta_{V,des}}{\Delta\theta_{V,des} - \sigma} = e^{\frac{\sigma}{\Delta\theta_H}}$$

$$\Delta\theta_{V,des} = \frac{\sigma}{1 - e^{-\frac{\sigma}{\Delta\theta_H}}}$$



# determination of design temperature for other rooms



## determination of design temperature for other rooms

- inlet temperature is the same
- the temperature difference is different
- in other rooms different pipe spacing can be selected

**$\sigma_j/\Delta_{H,j} < 0,5$**

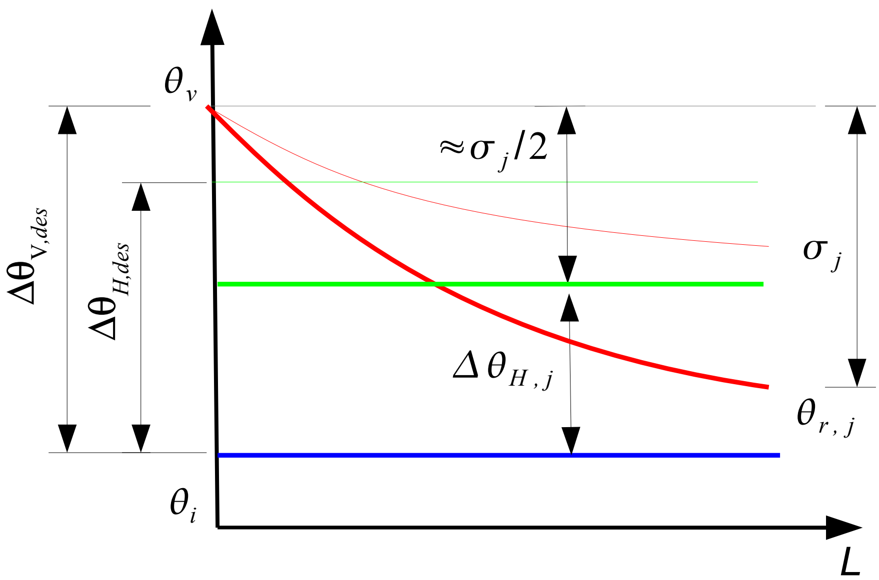
$$\frac{\sigma_j}{2} = \Delta\theta_{V,des} - \Delta\theta_{H,j}$$

**$\sigma_j/\Delta_{H,j} \geq 0,5$**

$$\sigma_j = 3\Delta\theta_{H,j}\left[\left(1 + \frac{4(\Delta\theta_{V,des} - \Delta\theta_{H,j})}{3\Delta\theta_{H,j}}\right)^{\frac{1}{2}} - 1\right]$$



## explanations of the terms



## water flow rate determination

for the generic j-th room the heating floor must provide the thermal output $\Phi_{Nf,j}$:

$$\Phi_{Nf,j} = \dot{m}_{H,j} c_w (\theta_V - \theta_R)_j - q_{u,j} \cdot A_F$$

dove:

- $m_{H,j}$ mass flow for the j-th room
- $c_w$ water specific heat capacity
- $q_u$ downward specific heat flow rate
- $A_F$ floor area

mass flow rate is obtained as

$$\dot{m}_{H,j} = \frac{A_F q_j}{c_w \sigma} \left( 1 + \frac{R_o}{R_u} + \frac{\theta_i - \theta_u}{R_u q_j} \right)$$



## Hot boiler room
### Devices

**Boiler Room $\Phi > 35$ kW**

- reduced specifications nella *Raccolta R* collection from ISPESL (now INAIL)
- Devices:
  - Security devices
  - protection devices
  - control devices

# devices following collection R ISPESL

## safety devices

- pressure relief valve
- Thermal relief valve
- Gas shutoff valve

## protection devices

- control thermostat
- emergency shutdown thermostat
- emergency pressure shutdown
- low water pressure shutdown

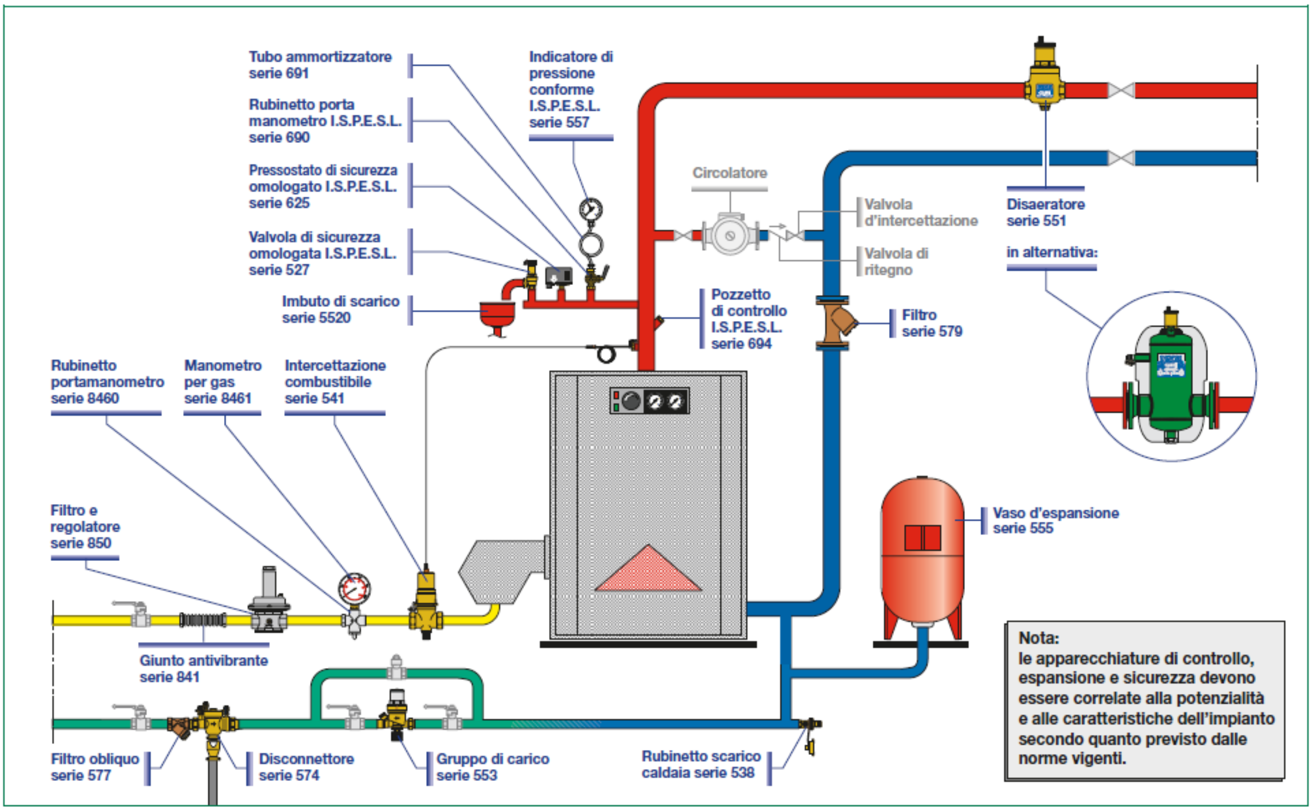
## Control devices

- thermometers
- pressure gauge



# Boiler Room

from Caleffi S.p.A.



## expansion tank

### kind of expansion tank

- the main purpose is to compensate the change of volume
- they may be :
  - open
  - closed

### closed expansion tank

atmospheric with or without diaphragm;

pressurized with or without diaphragm;



## expansion tank

### expansion volume

$$E = \frac{V_A \cdot n}{100}$$

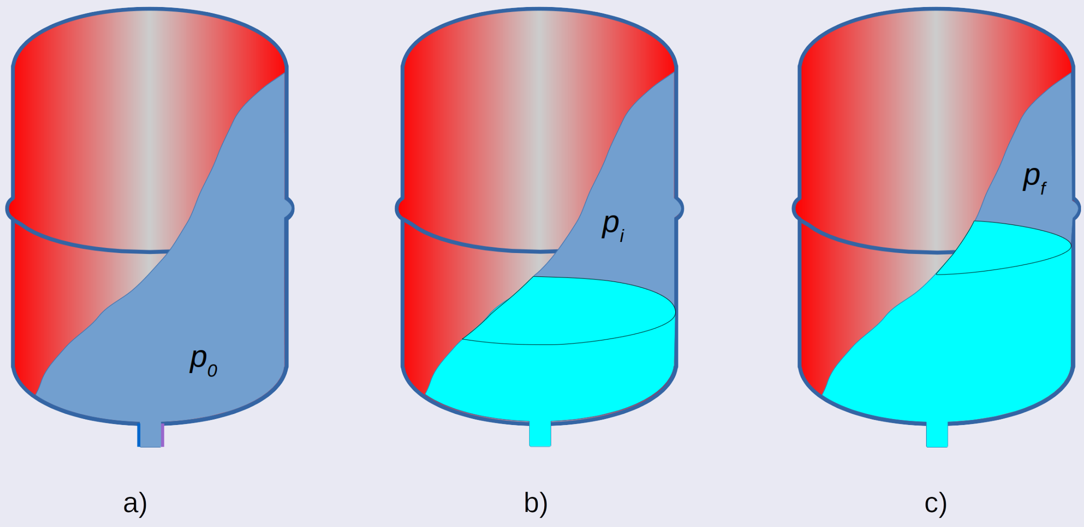
$V_A$ water content

$n$ $= 0,31 + 3,9 \times 10^{-4} \cdot t_m^2$

$t_m$ maximum temperature in °C

# expansion tank

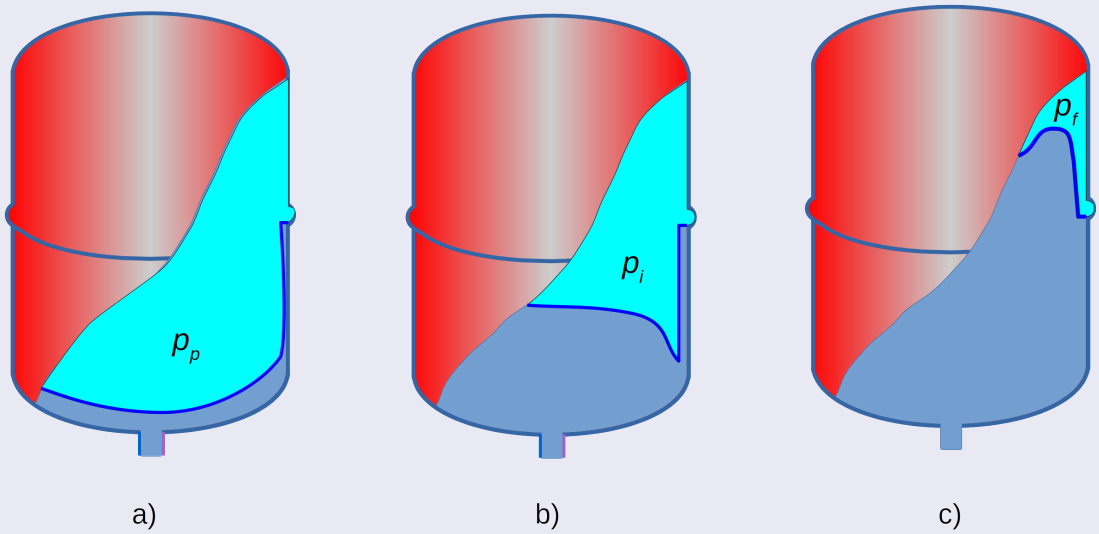
## without diaphragm






# expansion tank

## with diaphgram



## Sizing an expansion tank

### without diaphragm

$$V_v = \frac{E}{\frac{p_o}{p_i} - \frac{p_o}{p_f}}$$

$E$ expansion volume

$p_o$ atmospheric pressure

$p_i$ initial atmospheric pressure

$p_f$ set pressure of pressure relief valve



## sizing expansion tanks

### without diaphragm or bladder

$$p_0 \cdot V_0 = p_i \cdot V_i = p_f \cdot V_f$$
$$E = V_i - V_f$$
$$V_i = V_0 \cdot \frac{p_0}{p_i}$$
$$V_f = V_0 \cdot \frac{p_0}{p_f}$$
$$E = V_0 \cdot \left(\frac{p_0}{p_i} - \frac{p_0}{p_f}\right)$$
$$V_v = V_0 = \frac{E}{\frac{p_0}{p_i} - \frac{p_0}{p_f}}$$

## sizing expansion tanks

### with diaphragm or bladder

$$V_v = \frac{E}{1 - \frac{p_p}{p_f}} \tag{1}$$

- $E$ expansion volume
- $p_p$ initial pressure, equal to the hydrostatic pressure plus a value to avoid pressure drops ($p_0 = p_{st} + 0.3\ \mathrm{bar}$)
- $p_f$ set pressure of pressure relief valve $p_f = p_{vs} - 0.5\ \mathrm{bar}$



## sizing expansion tanks

### with diaphragm or bladder

$$p_p \cdot V_v = p_f \cdot V_f$$

$$E = V_v - V_f$$

$$V_f = V_v \cdot \frac{p_p}{p_f}$$

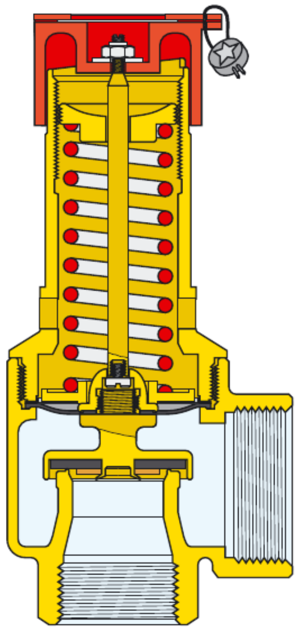
$$E = V_v \cdot \left(1 - \frac{p_p}{p_f}\right)$$

$$V_v = \frac{E}{1 - \frac{p_p}{p_f}}$$

# pressure relief valve

## How it works

- divert flow as the inlet pressure reaches the valve set pressure
- the nozzle discharge water vapour
- latent heat balance the boiler heat input



from Caleffi



# pressure relief valve

## some physics

$$\dot{m}_v \cdot r = \Phi_u$$

$$\Phi_u = \dot{m}_v \cdot r = \frac{\dot{V}}{v_v} \cdot r = \frac{w_{max}}{v_v} \cdot A \cdot r$$

$w_{max}$ nozzle maximum velocity;

$v_v$ water vapour specific volume

$A$ valve area section

$$A = \Phi_u \cdot \frac{v_v}{w_{max} \cdot r}$$

# Pressure relief valve

Raccolta R

$$A = 0,005 \cdot \dot{m}_v \cdot \frac{Q}{0,9 \cdot K}$$

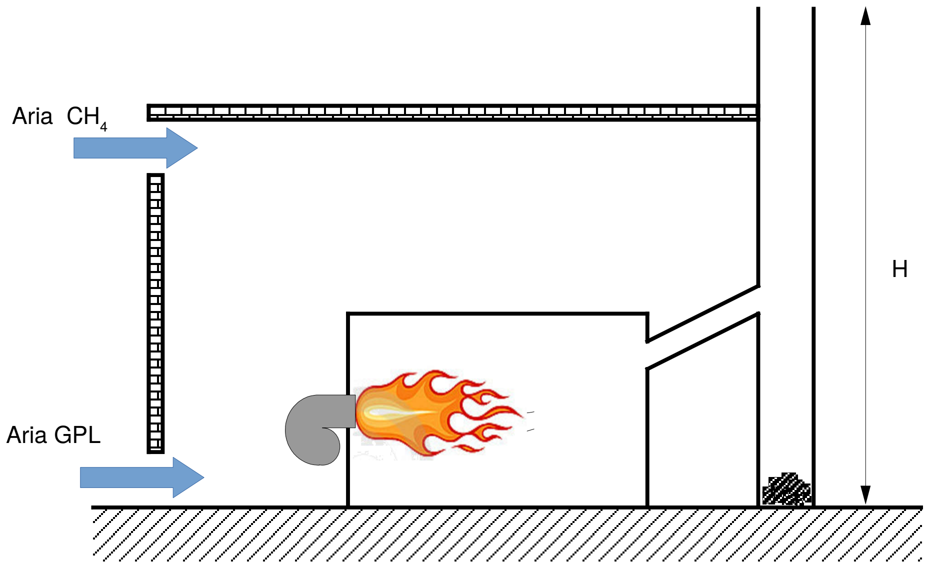
$Q$ discharge capacity [kg/h ]
$A$ minimum nozzle area square centimeters;
$\dot{m}_v$ water vapour mass flow [kg/h];
$F$ pressure factor, from table;
$K$ Valve efflux coefficient from certification.

**discharge pressure values from 0,5 to 12,5 bar**

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| p | 0,50 | 0,60 | 0,70 | 0,80 | 0,90 | 1,00 | 1,10 | 1,20 | 1,30 | 1,40 | 1,50 | 1,60 | 1,70 |
| F | 2,47 | 2,32 | 2,19 | 2,07 | 1,97 | 1,87 | 1,79 | 1,71 | 1,63 | 1,57 | 1,51 | 1,45 | 1,40 |
| p | 1,80 | 1,90 | 2,00 | 2,10 | 2,20 | 2,30 | 2,40 | 2,50 | 2,60 | 2,70 | 2,80 | 2,90 | 3,00 |
| F | 1,35 | 1,31 | 1,26 | 1,22 | 1,19 | 1,15 | 1,12 | 1,09 | 1,06 | 1,03 | 1,01 | 0,98 | 0,96 |
| p | 3,10 | 3,20 | 3,30 | 3,40 | 3,50 | 3,60 | 3,70 | 3,80 | 3,90 | 4,00 | 4,20 | 4,40 | 4,60 |
| F | 0,93 | 0,91 | 0,89 | 0,87 | 0,85 | 0,84 | 0,82 | 0,80 | 0,79 | 0,77 | 0,74 | 0,71 | 0,69 |
| p | 4,80 | 5,00 | 5,20 | 5,40 | 5,60 | 5,80 | 6,00 | 6,20 | 6,40 | 6,60 | 6,80 | 7,00 | 7,20 |
| F | 0,67 | 0,65 | 0,62 | 0,61 | 0,59 | 0,57 | 0,56 | 0,54 | 0,53 | 0,51 | 0,50 | 0,49 | 0,48 |
| p | 7,40 | 7,60 | 7,80 | 8,00 | 8,20 | 8,40 | 8,60 | 8,80 | 9,00 | 9,50 | 10,0 | 10,5 | 11,0 |
| F | 0,46 | 0,45 | 0,44 | 0,43 | 0,43 | 0,42 | 0,41 | 0,40 | 0,39 | 0,37 | 0,36 | 0,34 | 0,32 |
| p | 11,50 | 12,00 | 12,50 | | | | | | | | | | |
| F | 0,32 | 0,30 | 0,29 | | | | | | | | | | |



# Boiler Room



# Centrifugal pumps

- Pumps circulate water
- in a closed circuit the pumps must only take into account the losses of the circuit
- in a closed circuit the height is not taken into account
- the choice of the pump derives from the size of the system and the losses



# Centrifugal Pumps

## Characteristic parameters

- volumetric flow $[\mathrm{m^3/s}]$ or mass flow $[\mathrm{kg/s}]$
- head as height or pressure
- power
  - power to fluid $P_i$
  - shaft power $P$
  - electrical power $P_e$
- net positive suction head NPSH [m]
- hydraulic efficiency $\eta = \frac{P_i}{P}$
- electrical efficiency $\eta_e = \frac{P}{P_e}$
- global efficiency $\eta_g = \eta \cdot \eta_e = \frac{P_i}{P_e}$

# pump laws

## absorbed power

$$P = \frac{\dot{m} \cdot g \cdot \Delta z}{\eta} = \frac{q_v \cdot \rho \cdot g \cdot \Delta z}{\eta}$$
$$P = \frac{\dot{m} \cdot v \cdot \Delta p}{\eta} = \frac{q_v \cdot \Delta p}{\eta}$$

## pump laws

$$\frac{q_{v1}}{q_{v2}} = \frac{n_1}{n_2}$$
$$\frac{\Delta z_1}{\Delta z_2} = \frac{\Delta p_1}{\Delta p_2} = \left[\frac{\dot{m}_1}{\dot{m}_2}\right]^2$$
$$\eta_1 = \eta_2$$
$$\frac{P_1}{P_2} = \left[\frac{n_1}{n_2}\right]^3 \text{ se } \eta_1 = \eta_2$$



# pump laws

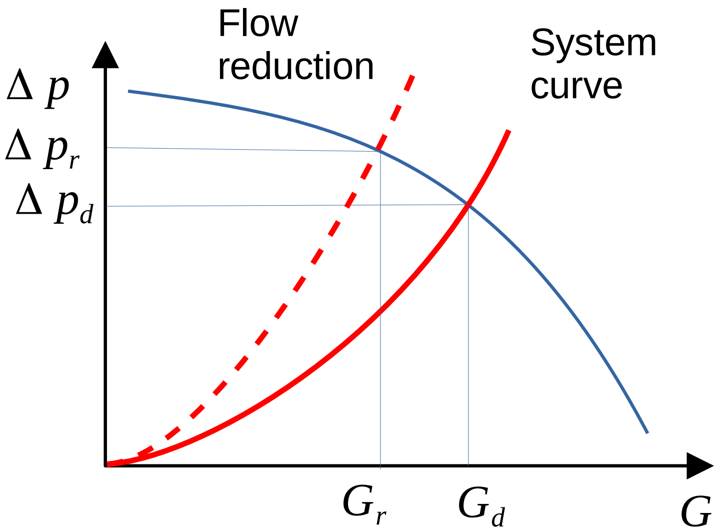
## pump laws, changing diameter

$$\frac{q_{v1}}{q_{v2}} = \frac{D_1}{D_2}$$
$$\frac{\Delta z_1}{\Delta z_2} = \left[\frac{\Delta D_1}{\Delta D_2}\right]^2$$
$$\frac{P_1}{P_2} = \left[\frac{D_1}{D_2}\right]^3$$
$$\eta_1 = \eta_2$$

last two relations are approximations

# Performance curve

## Pumps

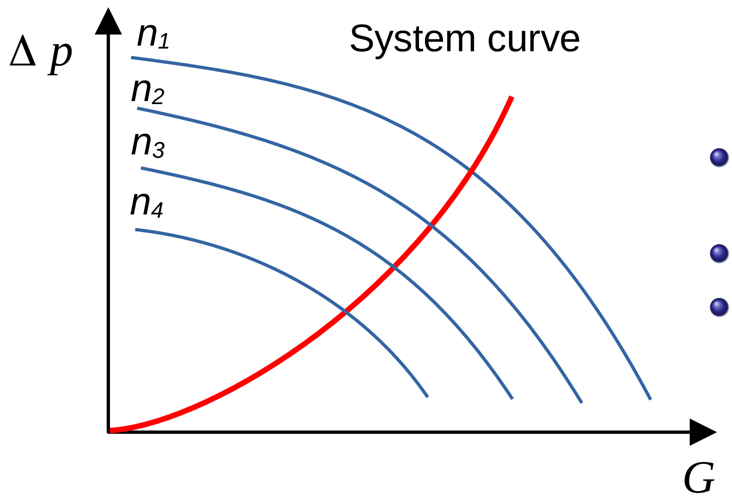



- the characteristic curve describes the pressure-flow relationship
- the operating point is the intersection with the system curve
- varying the system curve, with valves, changes the operating point



# Performance curves

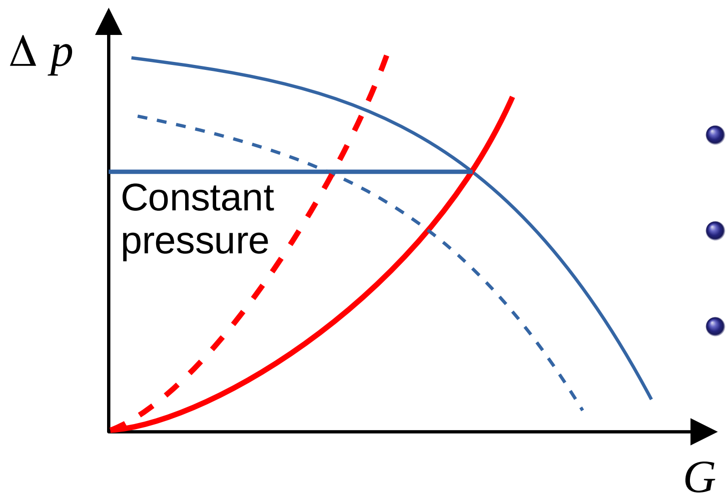
## variable velocities




- varying the pump speed varies the characteristic curve
- also varies the operating point
- decreases the flow rate, decreases the head

# Performance curve

## inverter, constant pressure

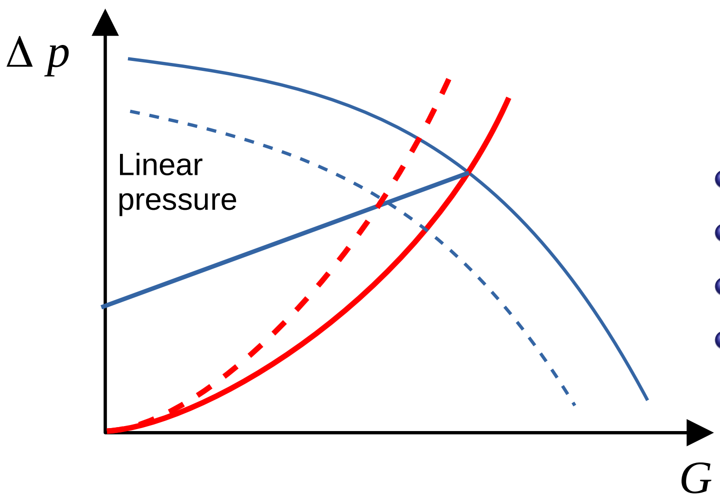



- the pumps speed varies with inverter
- the regulation can be at constant pressure
- the pressure is maintained as the flow rate decreases



# Performance curves

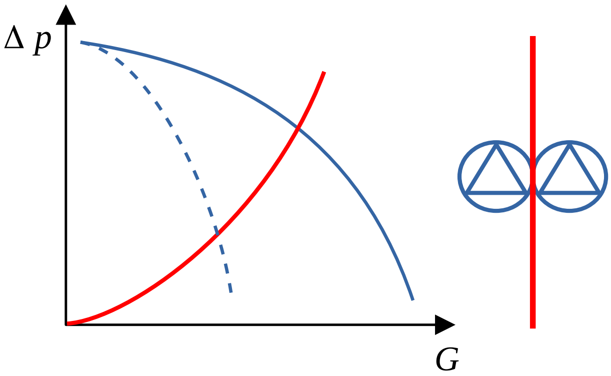
## inverter, linear pressure




- pumo speed changes using inverter
- pressure changes linearly
- pressure is lower at lower flow rate
- reduces energy costs

# coupled pumps

## parallel or alternating operation



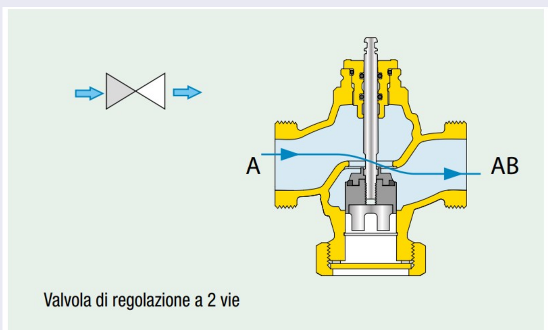

- coupled pumps to increase availability
  - alternating: one pump in standby
  - parallel: with a faulty pump the plant can work with a reduced flow



# Valves

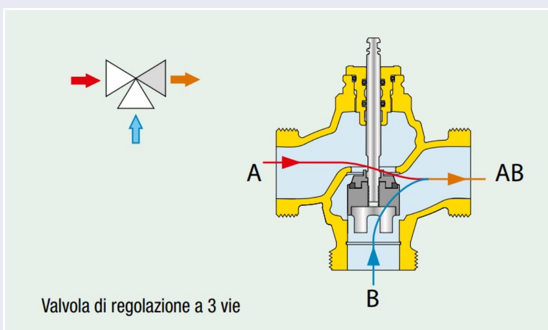
## kind

**2 way**




from Caleffi

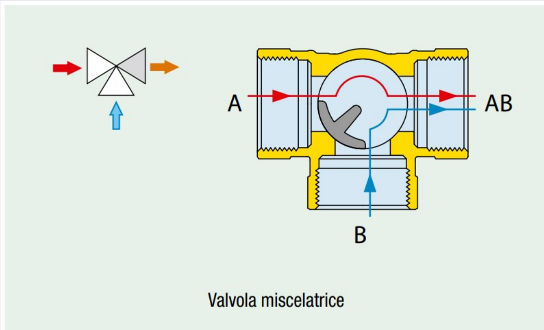
**3 way**




from Caleffi

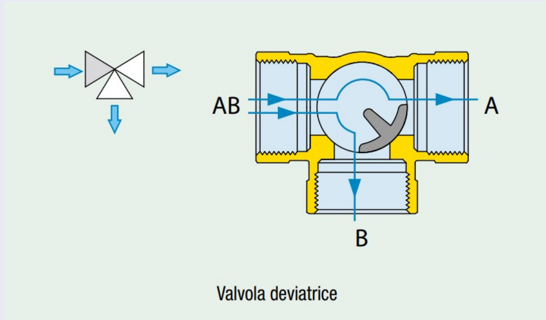
# Valves
## type

### mixing




from Caleffi

### diverting

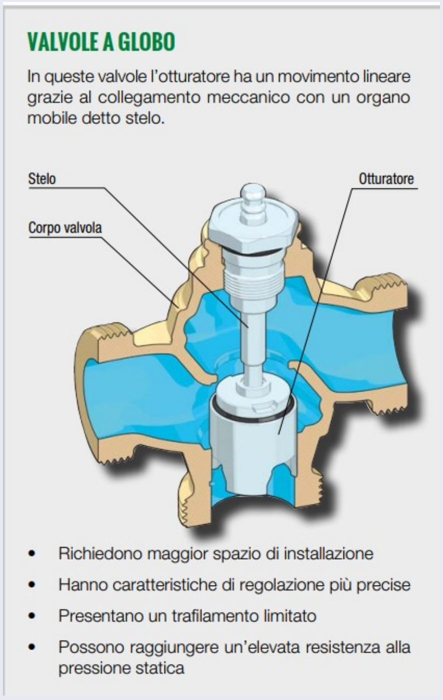



from Caleffi



# Valves
## type

### Globe



VALVOLE A GLOBO

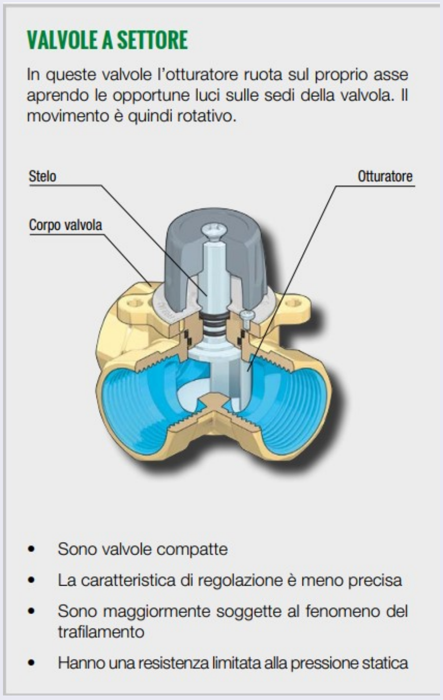
In queste valvole l'otturatore ha un movimento lineare grazie al collegamento meccanico con un organo mobile detto stelo.



- Richiedono maggior spazio di installazione
- Hanno caratteristiche di regolazione più precise
- Presentano un trafilamento limitato
- Possono raggiungere un'elevata resistenza alla pressione statica

from Caleffi

### Gate



VALVOLE A SETTORE

In queste valvole l'otturatore ruota sul proprio asse aprendo le opportune luci sulle sedi della valvola. Il movimento è quindi rotativo.

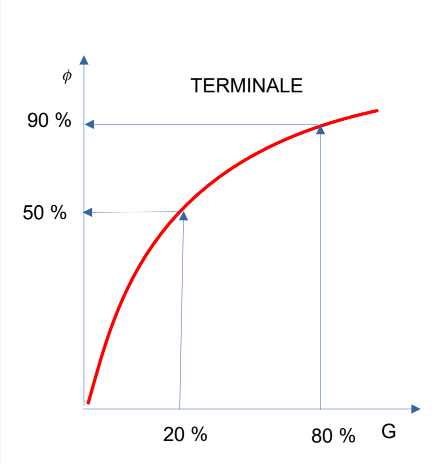


- Sono valvole compatte
- La caratteristica di regolazione è meno precisa
- Sono maggiormente soggette al fenomeno del trafilamento
- Hanno una resistenza limitata alla pressione statica

from Caleffi

# System curve

## not proportional system curve

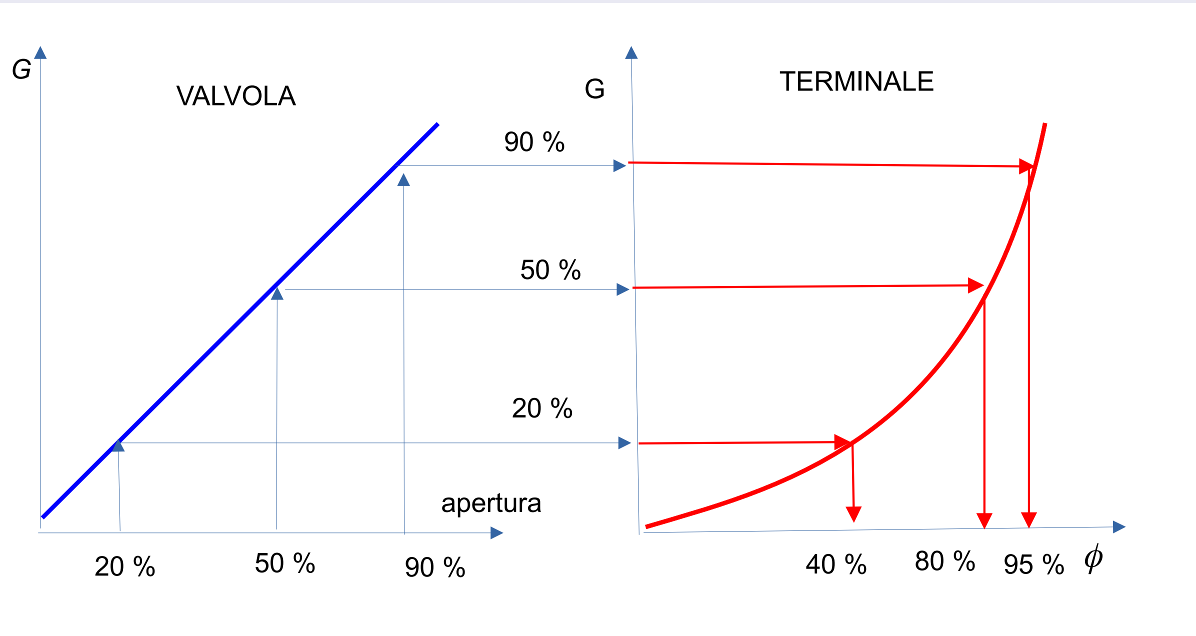



- mass flow and heat release is not proportional
- control problems
- little movement of regulation item result in a large increase of heat released
- linear relationship is sought



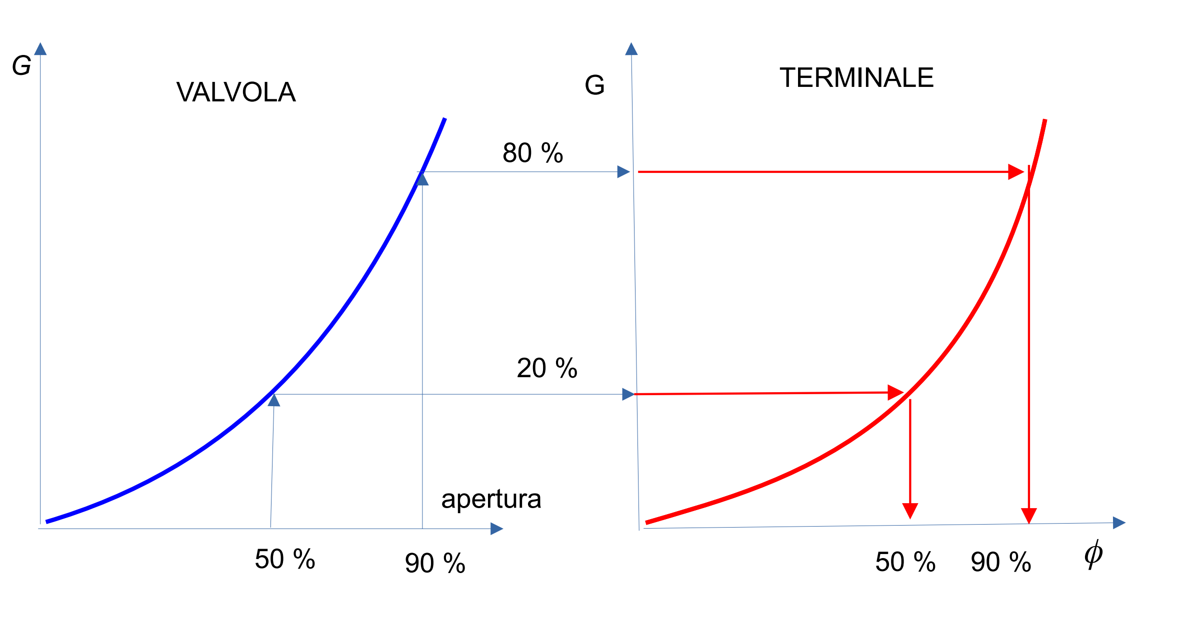
# Terminal control

## linear relationship



# Terminal control

## equipercentual control






# equipecentual valve

## equipercentual law

$$G = G_0 \cdot e^{k \cdot h}$$
$$\frac{dG}{dh} = G_0 \cdot k \cdot e^{k \cdot h} = k \cdot G$$
$$\frac{dG}{G} = k \cdot dh$$

$G$ portata

$h$ posizione dello stelo

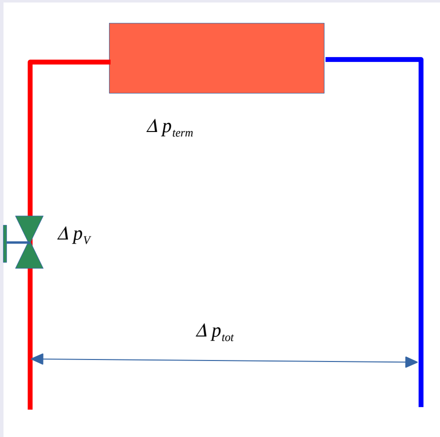
$G_0$ parametro

$k$ parametro

# Autority

plant control

## Autority




- two way valve for controlling flow
- requires a large head loss to control correctly the flow
- valve authority

$$A = \frac{\Delta p_V}{\Delta p_{tot}} = \frac{\Delta p_V}{\Delta p_V + \Delta p_{circ} + \Delta p_{term}}$$

- desirable a sufficient authority $A > 0.5$



# Authority

flow with different authority

- two way valve regulates the flow changing the $k_v$ in a range 0- $k_{va}$ full open valve
- obtained flow changes with the authority

## relation between $G$ e $k_v$

$$\Delta p_{tot} = \Delta p_v + \Delta p_{circ} + \Delta p_{term} = \Delta p_v + \Delta p_{imp}$$

$$\Delta p_{imp} = \left(\frac{G}{k_{imp}}\right)^2$$

$$\Delta p_v = \left(\frac{G}{k_v}\right)^2$$

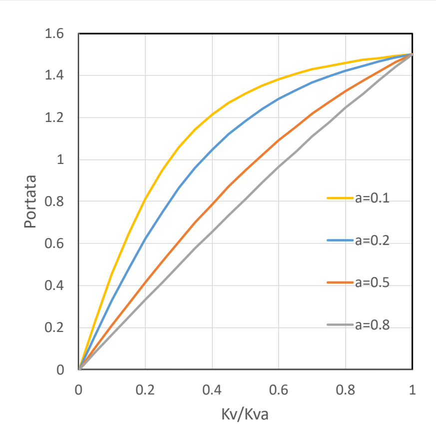
$$\Delta p_{tot} = \left(\frac{G}{k_{imp}}\right)^2 + \left(\frac{G}{k_v}\right)^2 = G^2 \cdot \left(\frac{1}{k_v^2} + \frac{1}{k_{imp}^2}\right)$$

$$G = \sqrt{\frac{\Delta p_{tot}}{1/k_v^2 + 1/k_{imp}^2}}$$

# Authority

water flow and authority

## Authority

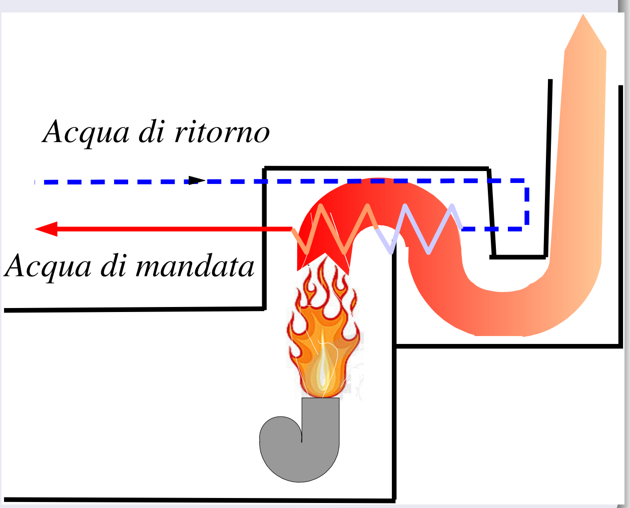



- linear law valve
- changing authority changes the flow
- difficult control with low authority
- desirable $A > 0.5$

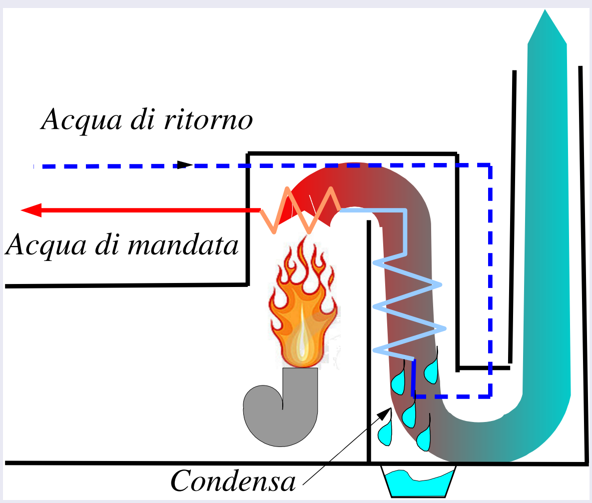


# Boiler

## Non condensing boiler




## condensing boiler



# Condensing boiler

rendimenti $> 1$

## non condensing boiler

$$\eta_{t100} = \frac{\Phi_u}{\dot{m}_c H_i + R} \approx \mathbf{91\%}$$

$\eta_{t100}$ = efficiency at maximum load referred to $H_i$

$H_i$ = lower heating value

$R$ = ventilator and burner power

## Condensing boiler

$$\eta = \frac{\Phi_{u,\ cond}}{\dot{m}_c H_s} \approx \mathbf{90/92\%}$$

$$\eta_{t100} = 98/102\%$$

$\eta$ efficiency related to high heating value $H_s$

$H_s$ Higher heating value *superiore*



# Heat Pumps

## introduction

- possible substitution for gas boilers
- part of the generated heat is a renewable source
- performance can change with external conditions
- to be consider for new buildings, or refurbished buildings
- works better with low temperatures

# Classification with source

| source | Ttype | extraction type |
|---|---|---|
| Internal air | Non-renewable if coming from systems using fossil fuels, with the exception of exhaust air | Cooling and dehumidification of internal exhaust air in recovery systems |
| Rock | Renewable geothermal | Cooling of the subsoil |
| Soil | Renewable geothermal | Cooling of the subsoil |
| Groundwater | Renewable *geothermal* | Cooling of the subsoil |
| Seawater | Renewable *hydrothermal* | Cooling of surface water |
| Lake water | Renewable hydrothermal | Cooling of surface water |
| River water | Renewable hydrothermal | Cooling of surface water |
| Waste water and sewage from technological processes | Non-renewable | Cooling of process water and/or sewage |
| Urban sewage | Similar to renewable | Cooling of urban sewage |



# Heat pumps

## Reference conditions for performance. Heat pumps for heating and domestic hot water production

| cold source | Cold source temperautre | | | | Temperature of hot environment | Temperautre environment | | | T hot environment | |
|---|---|---|---|---|---|---|---|---|---|---|
| Air | -7 | 2 | 7 | 12 | 20 | 35 | 45 | 55 | 45 | 55 |
| Water | | 5 | 10 | 15 | 20 | 35 | 45 | 55 | 45 | 55 |
| Ground/rock | -5 | 0 | 5 | 10 | 20 | 35 | 45 | 55 | 45 | 55 |

## Limit Temperature

Heat Pumps have some limit running temperatures to be considered

$\theta_{H,off}$ The HP is switched off when external temperature reaches this value, usually 20°C

$\theta_{w,off}$ The HP is switched off when DHW reaches this value, usually 60°C

$TOL$ limit temperature of the cold source, this is determined by the producer

$\theta_{H,cut-off,min}$ design temperature of the cold source at which the HP is swithced off

$\theta_{H,cut-off,max}$ design temperature of the hot source at which the HP is swithed off



## Load factor of heat pump

- depends on the temperature of the hot and cold source
- different values for water and air
- maximum temperature for the temperatures
- the load factor can be determined as:

$$CR = \frac{\Phi_{H,hp,out,bin}}{\Phi_{bin,max,H}}$$

## Design evaluations

The power size of HP depends on the load to satisfy

- A. HP satisfies the whole load
- B. The HP can satisfy only partially the load, other heat sources should be considered
- C. other generators are present

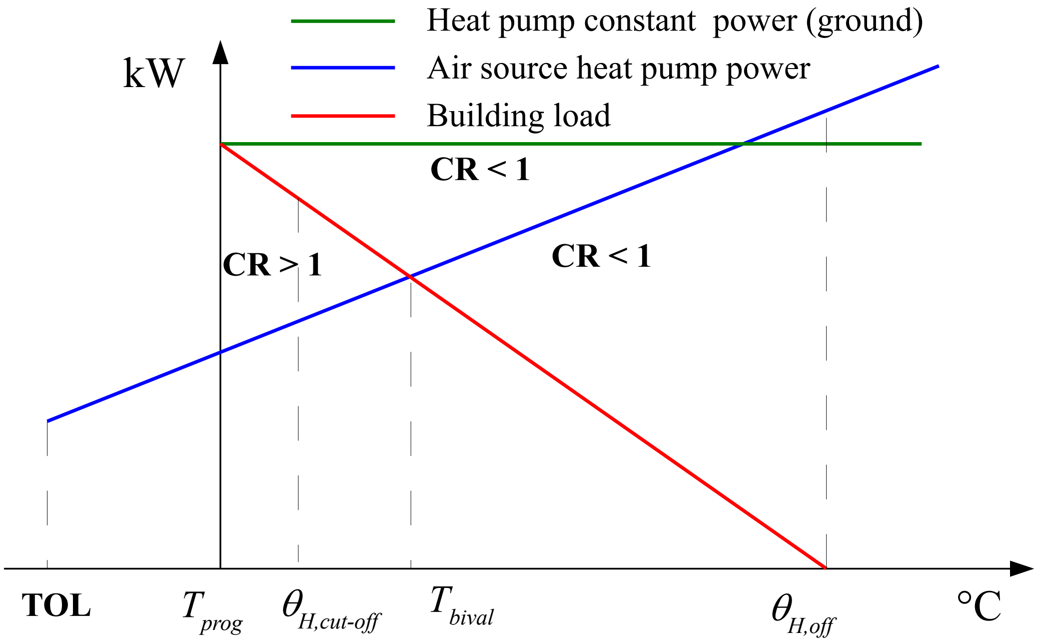
In conditions B and C the HP is in bivalemt working conditions , the temperature at which $CR = 1$ is the bivalent temperature



## HP at balance point

1. alternating operation: the heat pump is deactivated when the bivalent temperature is reached and the integration generator is activated which supplies the entire thermal power up to the design load;
2. parallel operation: the heat pump is not deactivated when the bivalent temperature is reached and the integration generator is activated which supplies the residual amount of thermal power;
3. partially parallel operation: the heat pump is not deactivated when the bivalent temperature is reached and for higher temperatures the integration generator is activated which must supply the residual power. At the temperature $\theta_{H,cut-off,min}$ the heat pump is deactivated and all the required thermal power must be supplied by the integration generator.

## Heat pump working area

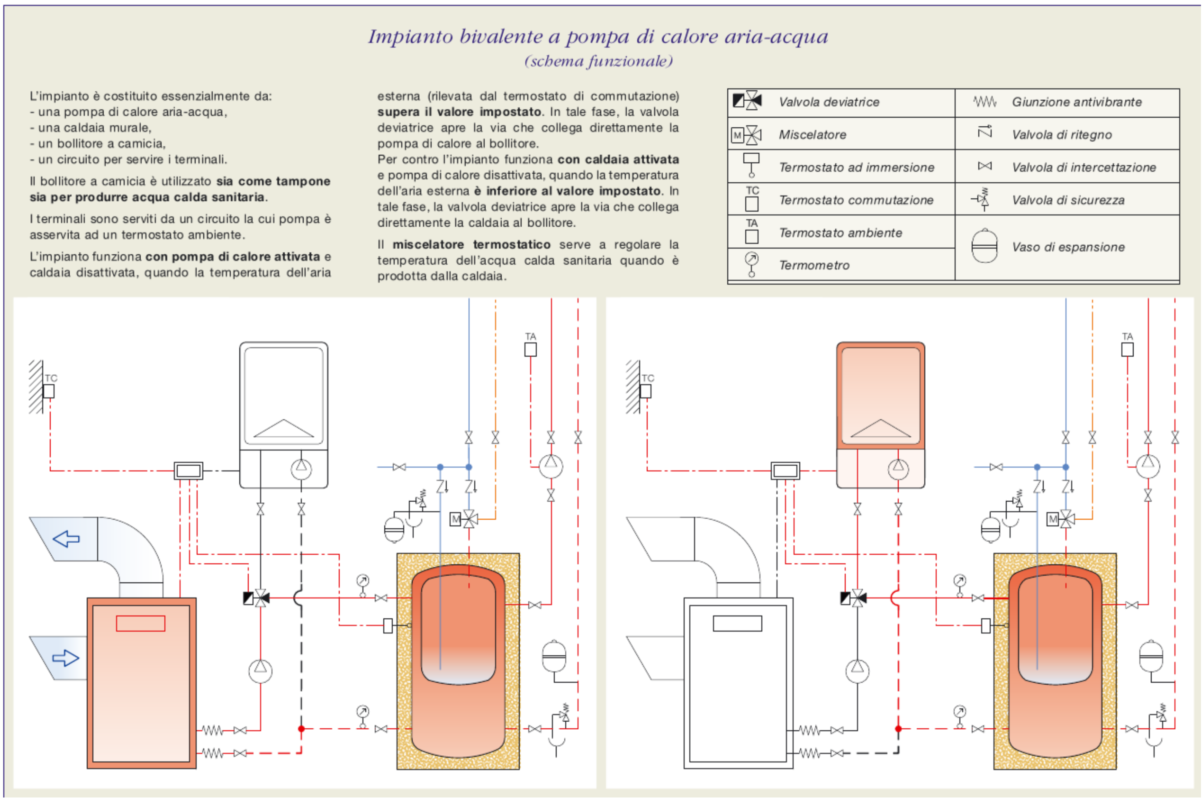





## Possible working conditions

1. $CR > 1$ e $\theta_f > \theta_{cut-off,min}$ HP at full power, but unsufficient to cover the load
2. $CR = 1$ HP at full load, the HP cover the entire load
3. $CR < 1$ partial power, the HP covers the entire load
4. $\theta_f < \theta_{cut-off,min}$, HP swithced off
5. $\theta_f \leq TOL$, HP swithced off

# HP in bivalent condition

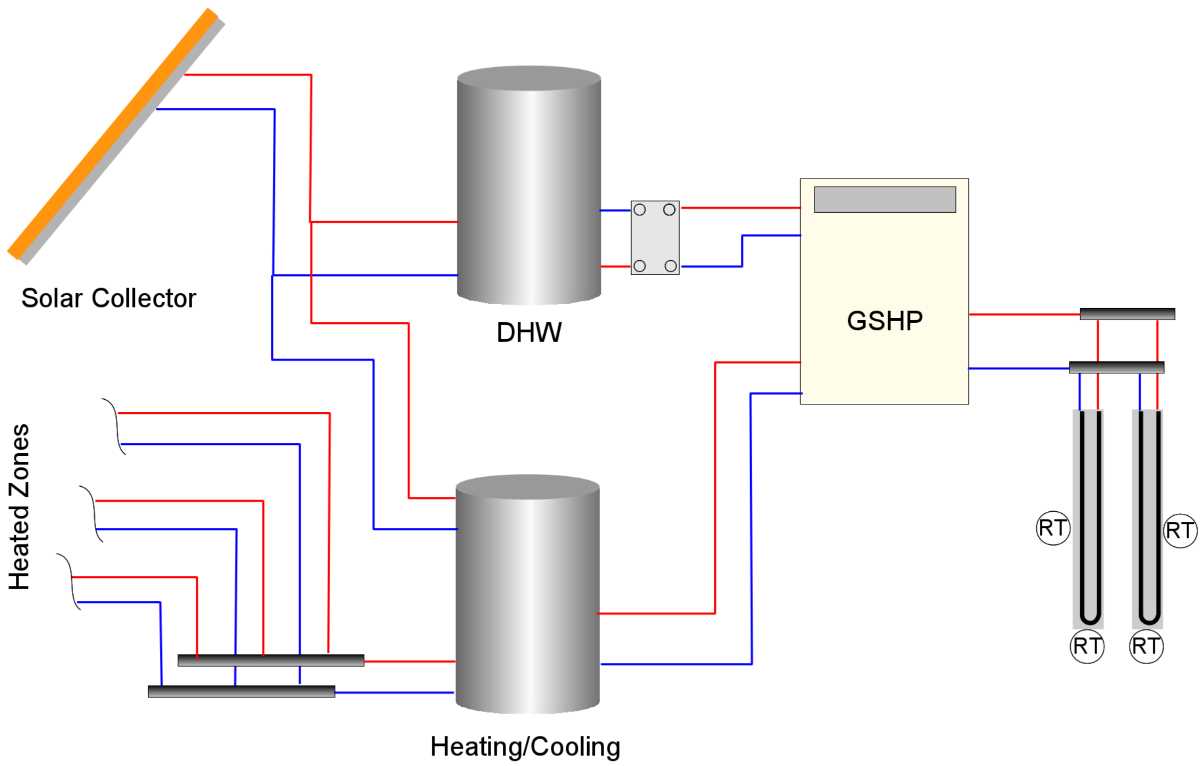



da rivista Idraulica n 33



# Geothermal HP

## building and plant in Trieste



# Geothermal HP scheme

## Building in Trieste