Heating plants

HVAC System design

Marco Manzan

University of Trieste
Department of Engineering and Architecture

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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:

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

θ_m inlet temperature

θ_r outlet temperature

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

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

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

heat power change with temperature

temperature different from the nominal one

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

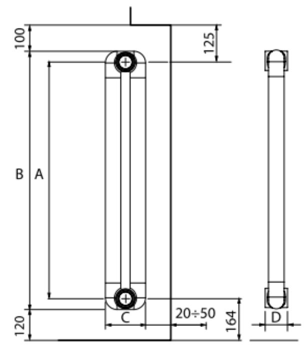
$$\Phi(\Delta\theta_a) = c(\Delta\theta_a)^n \;=\; \Phi_n \left(\frac{\Delta\theta_a}{\Delta\theta_n}\right)^n$$

Example of ta technical information sheet

exampla taken from a technical sheet by FONDITAL

TRIBECA

Standard supply	235 - 335 - 350 - 435 500 - 535 - 600	from 4 to 20 elements
	685 - 700 - 800 - 835	from 4 to 16 elements
	900 - 935 - 1000 - 1135 1200 - 1400 - 1435 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

ρ density

μ 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

Δp_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
- Δp_l takes into account the losses along the pipe and fittings discontinuities

pressure loss

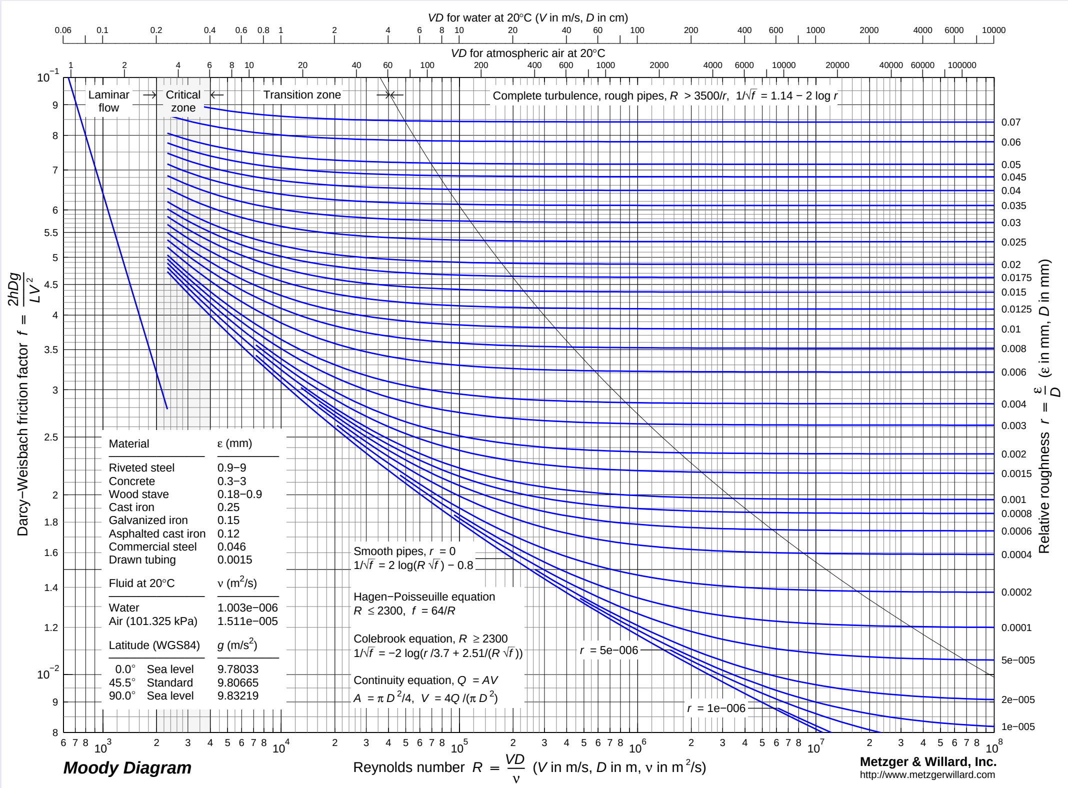
Friction Factor

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

- r [Pa/m] pressure drop per unit length $\frac{\Delta p}{L}$
- L length of the duct
- D diameter of the duct
- ρ 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 ϵ/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

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

- copper
- plastic water pipe

medium roughness

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

- steel
- galvanized steel

high roughness

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

- 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)

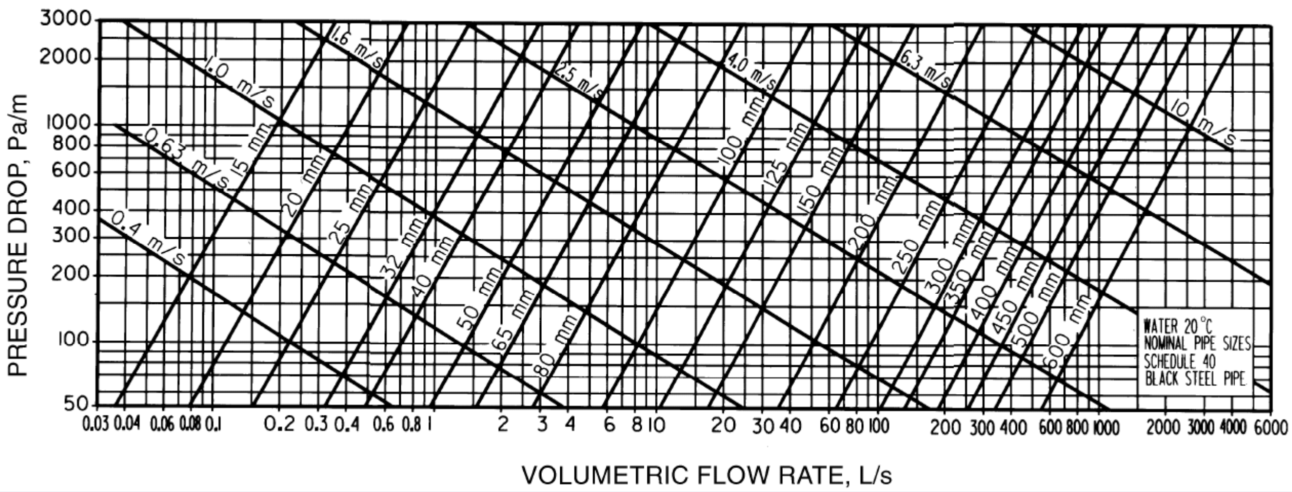
$$F_a = 0,07\ Re^{-0,13}\ D^{-0,14}$$

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

Friction chart

from ASHRAE

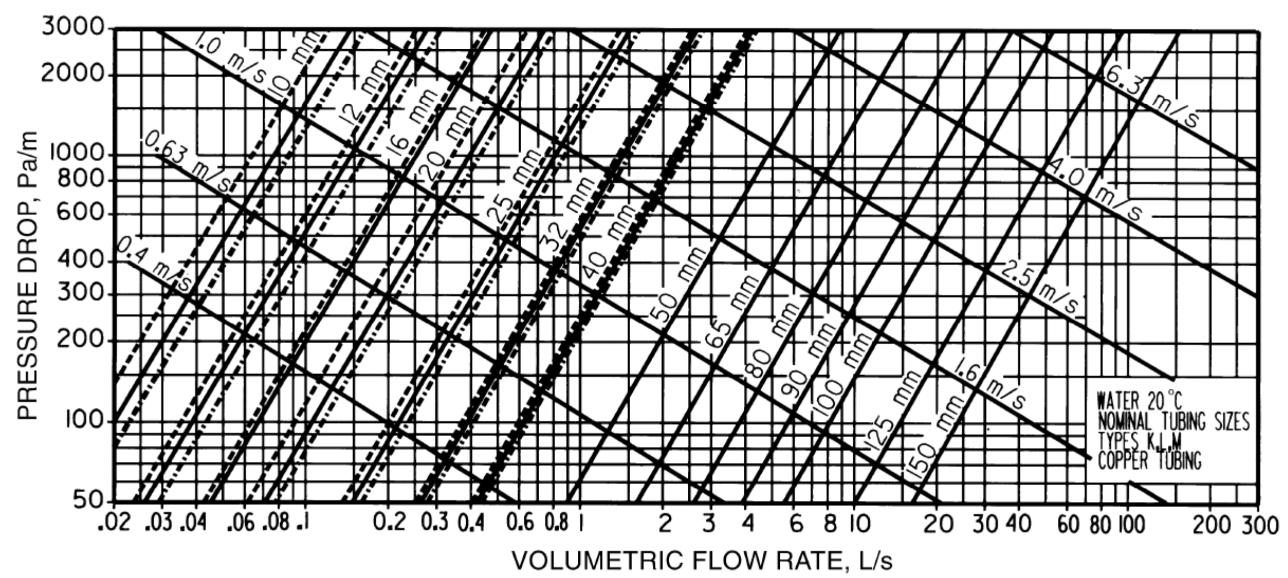
steel pipes



Friction chart

from ASHRAE

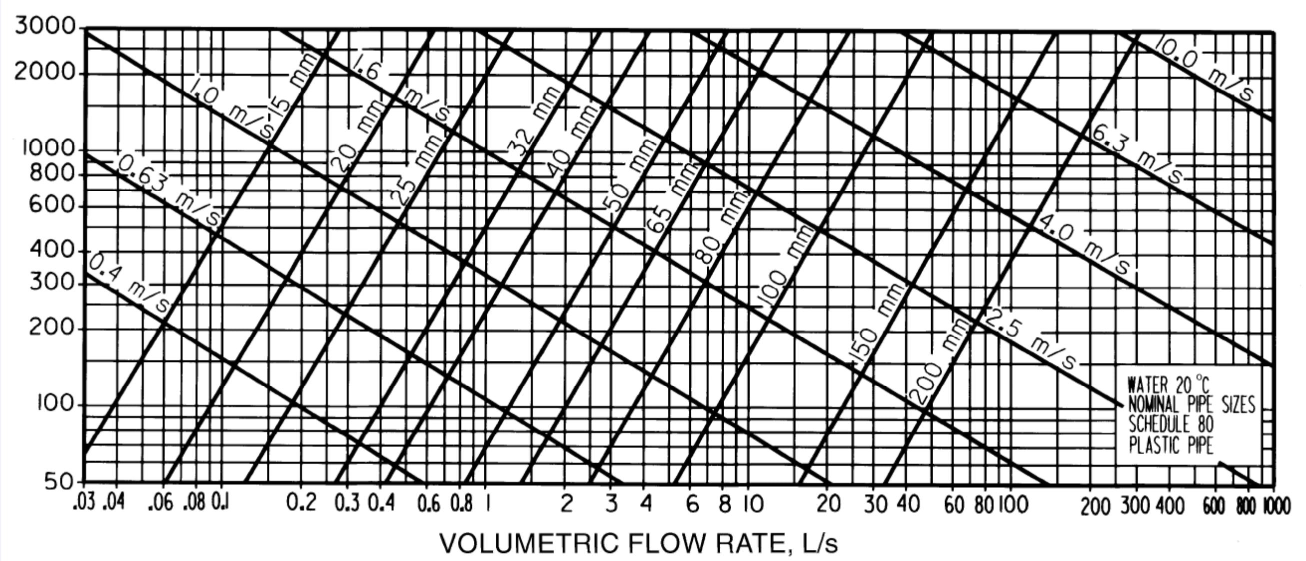
copper pipes



Friction chart

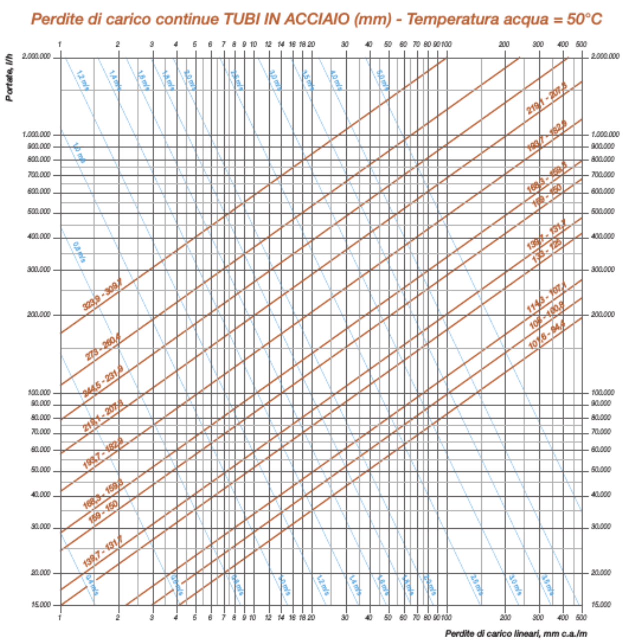
from ASHRAE

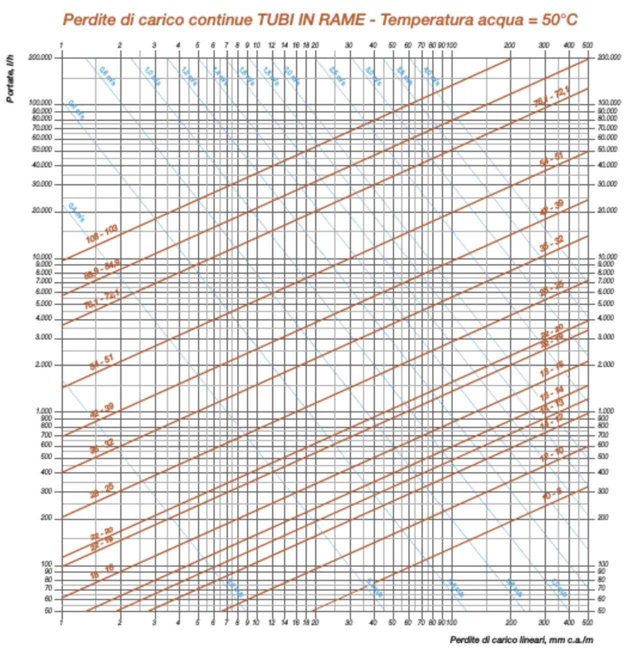
plastic pipe



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

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

ξ 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

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

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 gpm obtained with $\Delta p = 1\ \mathrm{psi}$.

gpm gallon per minute

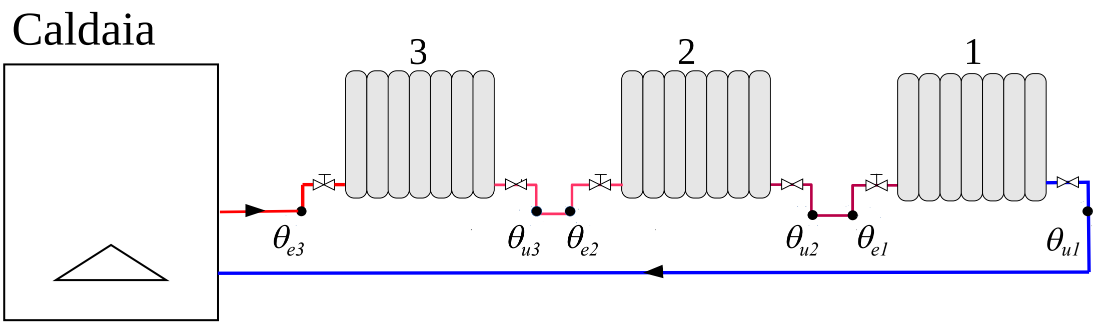
psi pounds square inch, 1 psi $=$6894.8 Pa

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

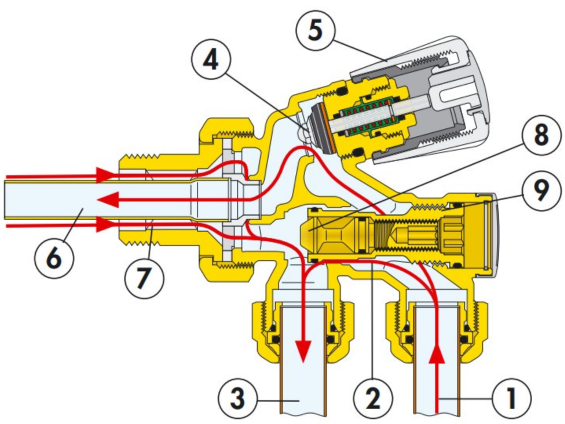
- low installation cost
- requires special attention in connecting radiators
- four way valves or bypass
- temperature drop computed on the whole ring
- the temperature of the radiator changes along the ring
- requires high flow rates to minimize the temperature differences

bypass

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

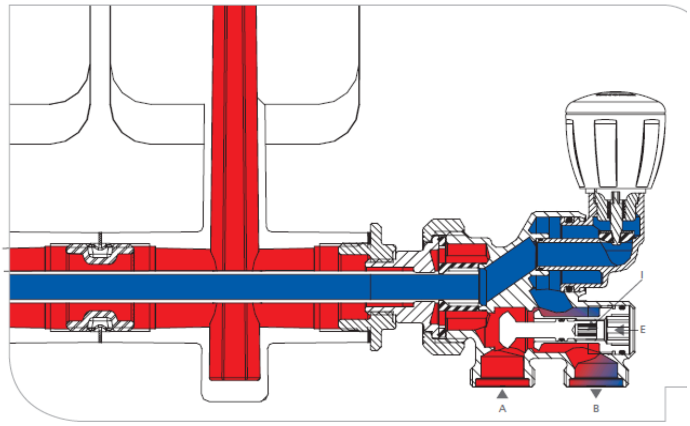
four way valve

- 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.

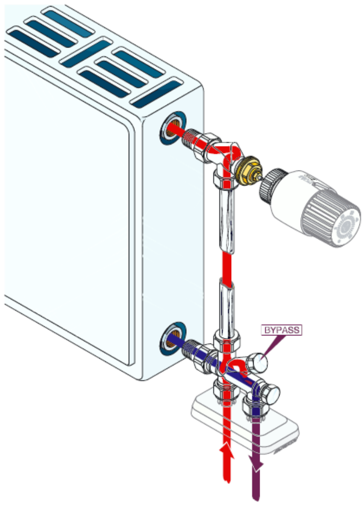


from Caleffi

four way valve



Giacomini



IVAR

pipe sizing

each circuit is analyzed at once:

1. heat Φ_A heat exchanged along the whole ring it is the sum of the heat exchanged by each Φ_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

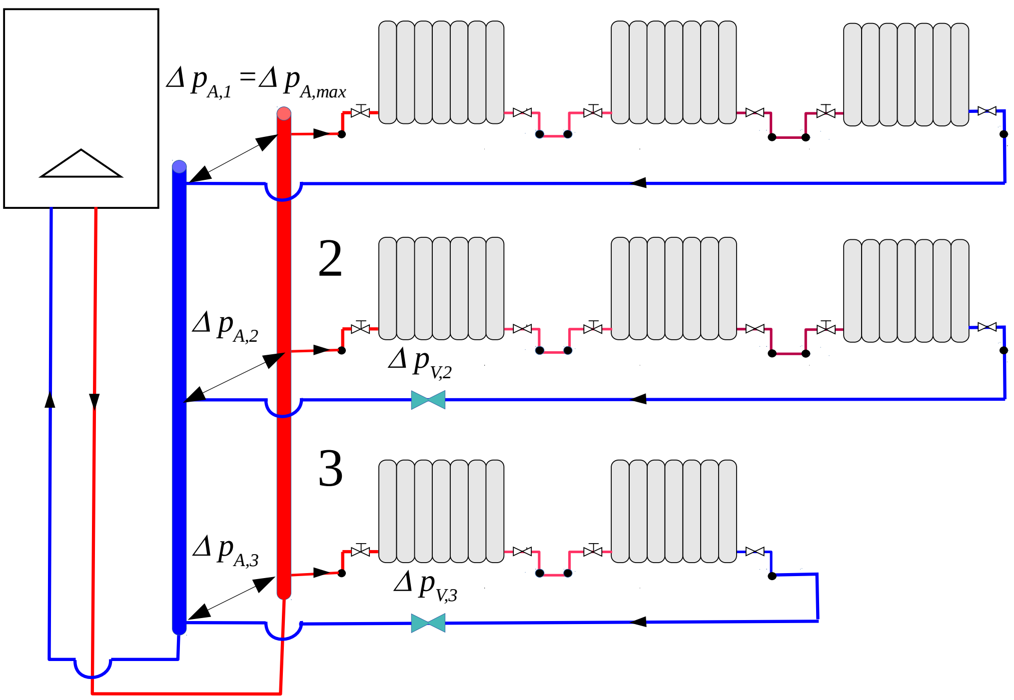
pressure loss

once sized the pipes compute totale loss:

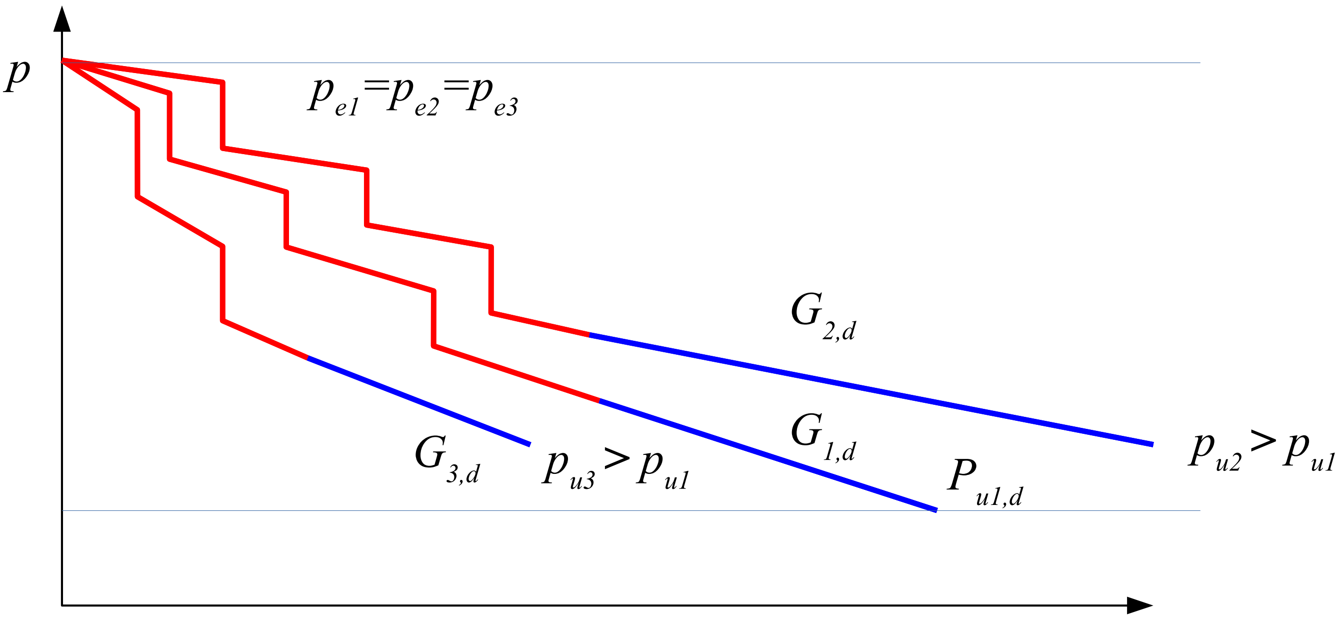
$$\Delta p_A = r_A \cdot L_A + \sum_i \Delta p_i + \sum_j \xi_j \cdot \rho \cdot \frac{v_A^2}{2}$$

Δp_A	total loss of the ring
r_A	pressure loss for unit length
L_A	pipe length of ring
Δp_i	pressure loss for each emitter
ξ_j	pressure loss coefficient
v_A	fluid velocity

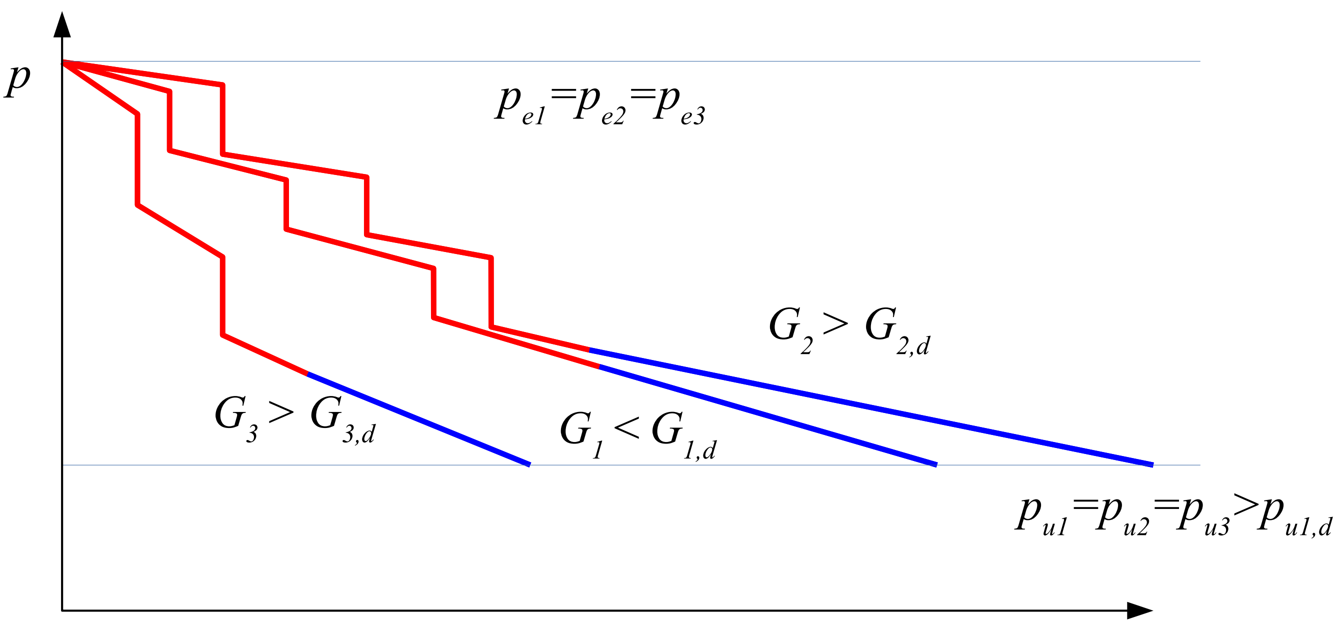
layout with additional circuits



One pipe circuits in parallel
Design pressure distribution

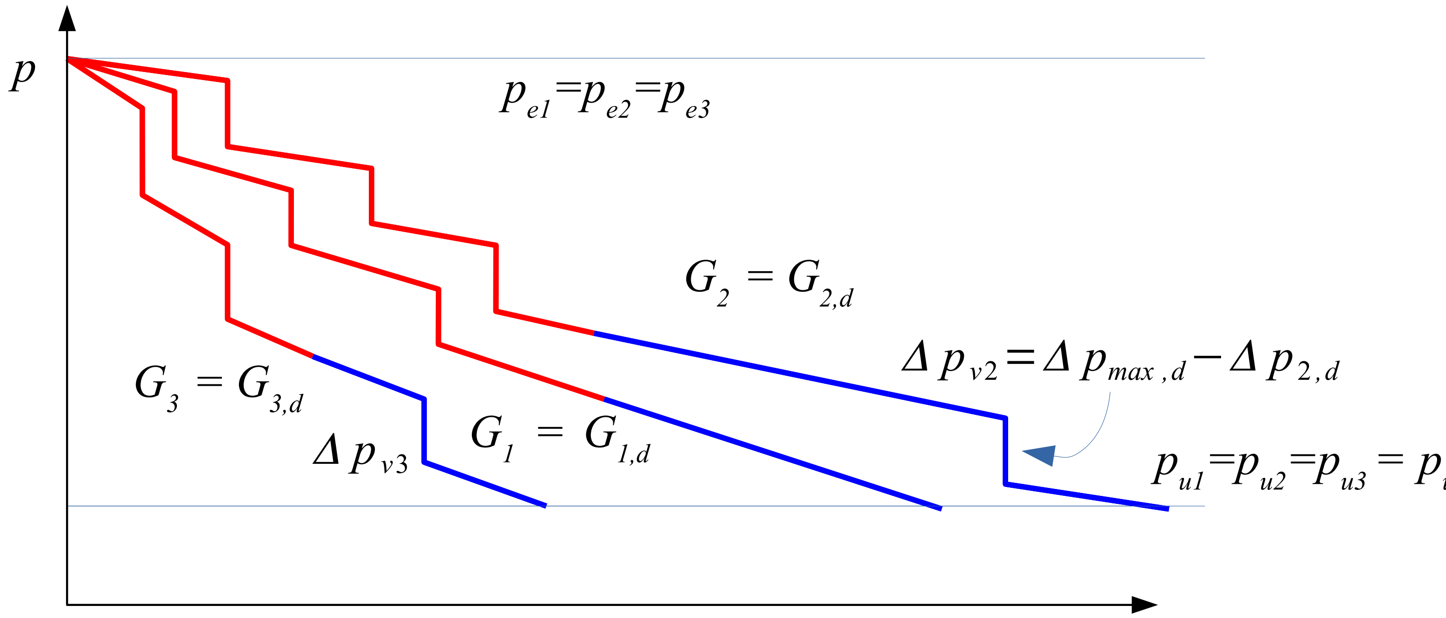


One pipe circuits in parallel
Design pressure distribution



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 ΔP_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

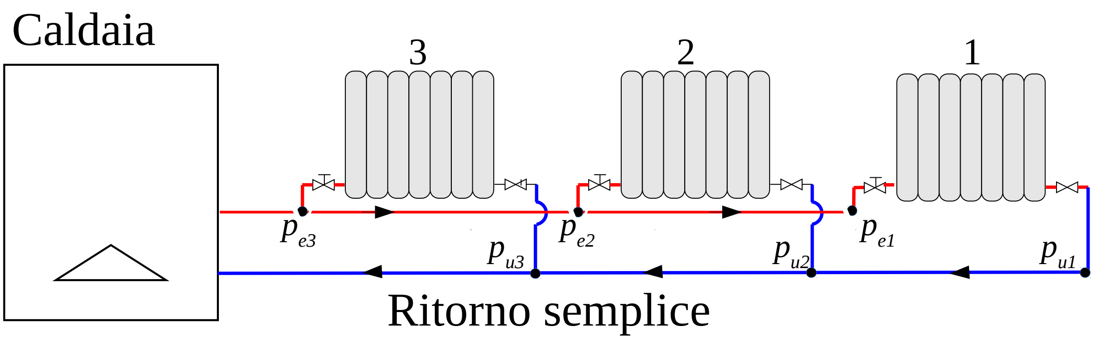
Direct Return

- 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

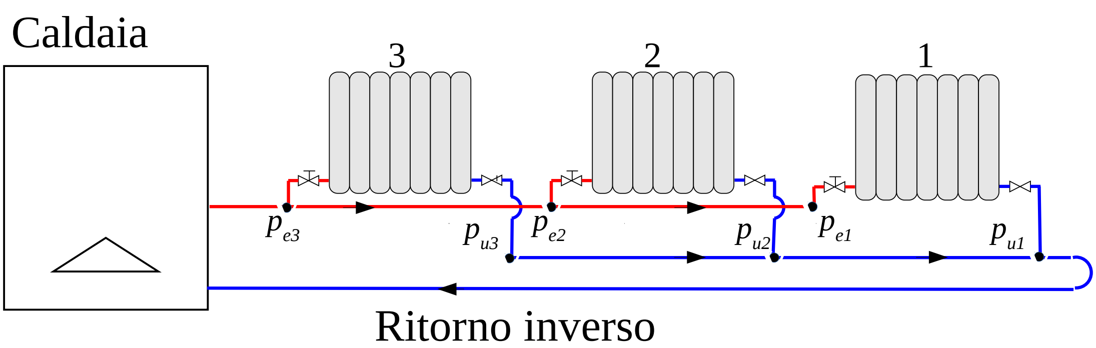


differential pressure

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

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

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

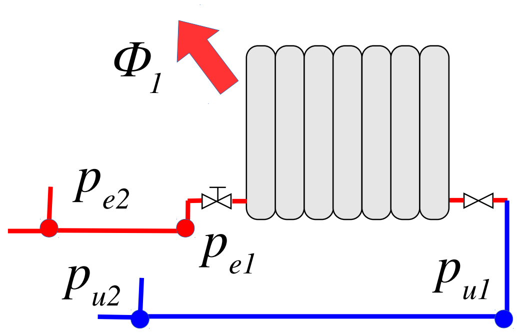
sizing and balancing

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

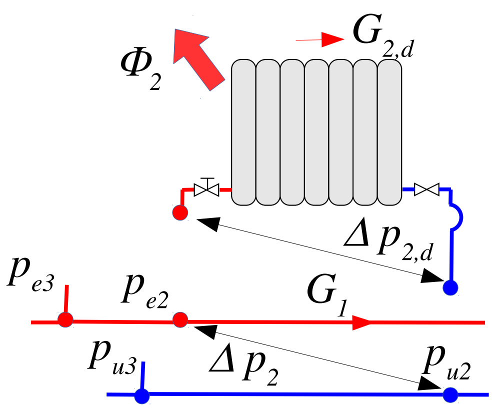
second terminal

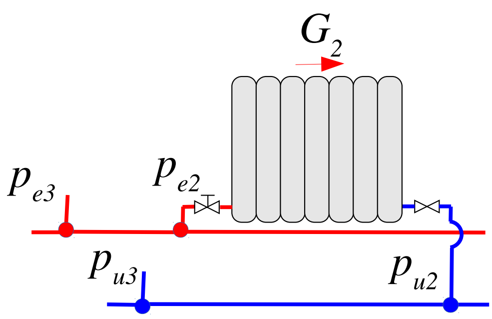
1. compute available pressure difference
2. size the terminal computing the design pressure loss $\Delta p_{2,d}$
3. balnce the system using the available pressure difference Δp
4. if the new flow rate is too large, add an additional pressure loss $\Delta p_{v,2}$

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

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

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





two pipe direct return

sizing and balancing

third terminal

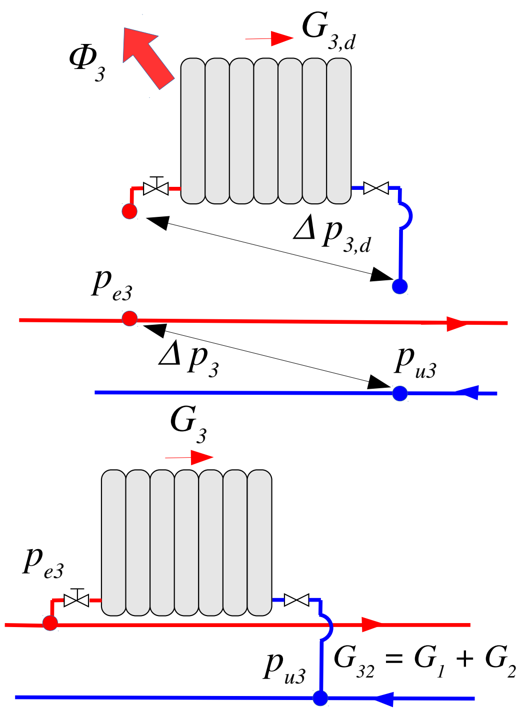
1. compute flow rate as the sum of the one of the two previous terminals
2. size pipe diameter using the pressure loss Δp_3
3. size the terminal and compute the design pressure loss $\Delta p_{3,d}$
4. balance the flow rate using the available pressure Δp_3
5. add a pressure loss if the fow rate is too large $\Delta p_{v,3}$

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

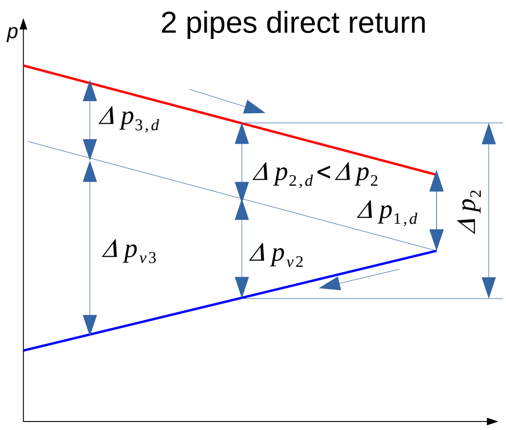
$$\Delta p_3 = \Delta p_2 + r_{32} \cdot L_{32} + \sum_j \xi_{32,j} \cdot \frac{1}{2} \cdot \rho \cdot u_{32}^2$$

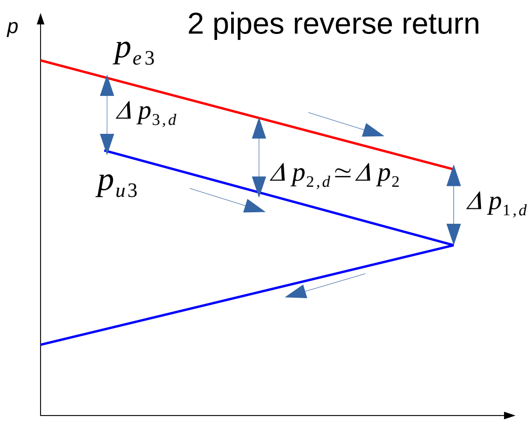
$$G_3 = G_{d,3} \cdot \left(\frac{\Delta p_2}{\Delta p_{3,2}} \right)^{0.525}$$

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



two pipes direct and reverse return



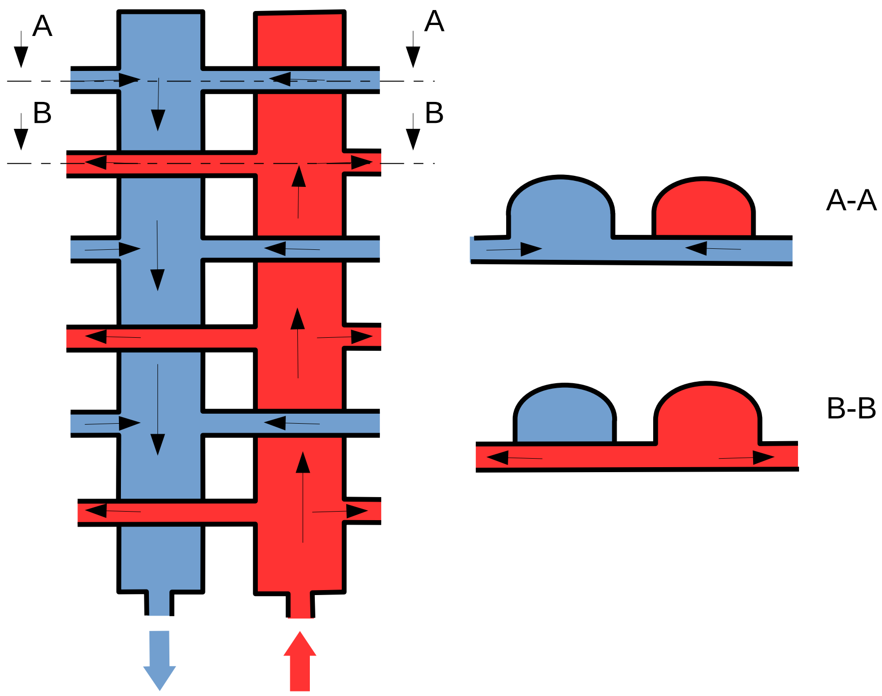


co-planar manifold

Caratteristiche

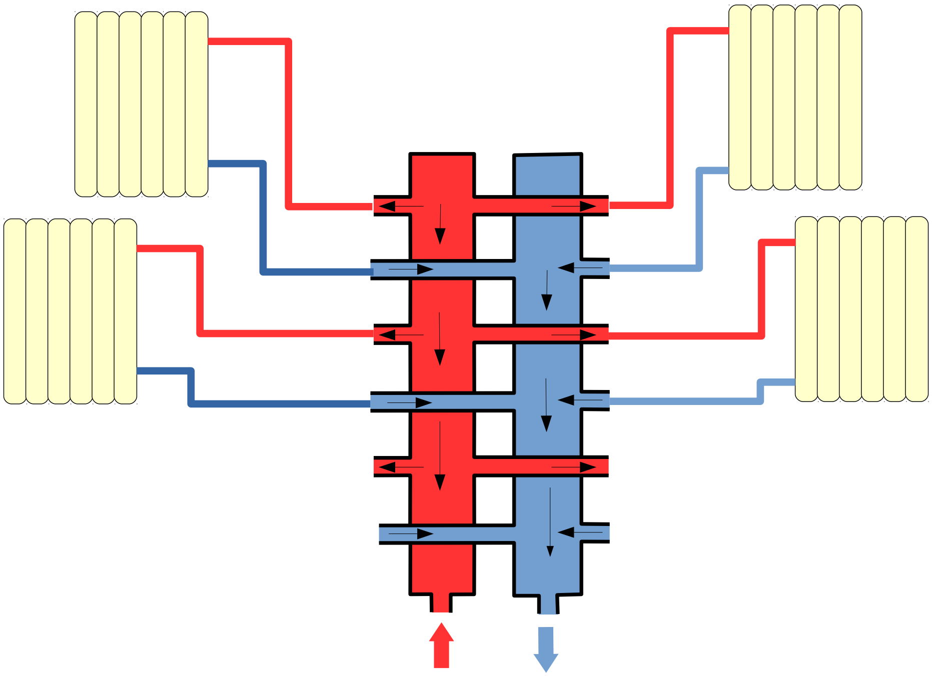
- used in new installations
- terminals connected in parallel
- sizing similar to the two pipes system
- requires balancing for correct function

co-planar manifold



co-planar manifold

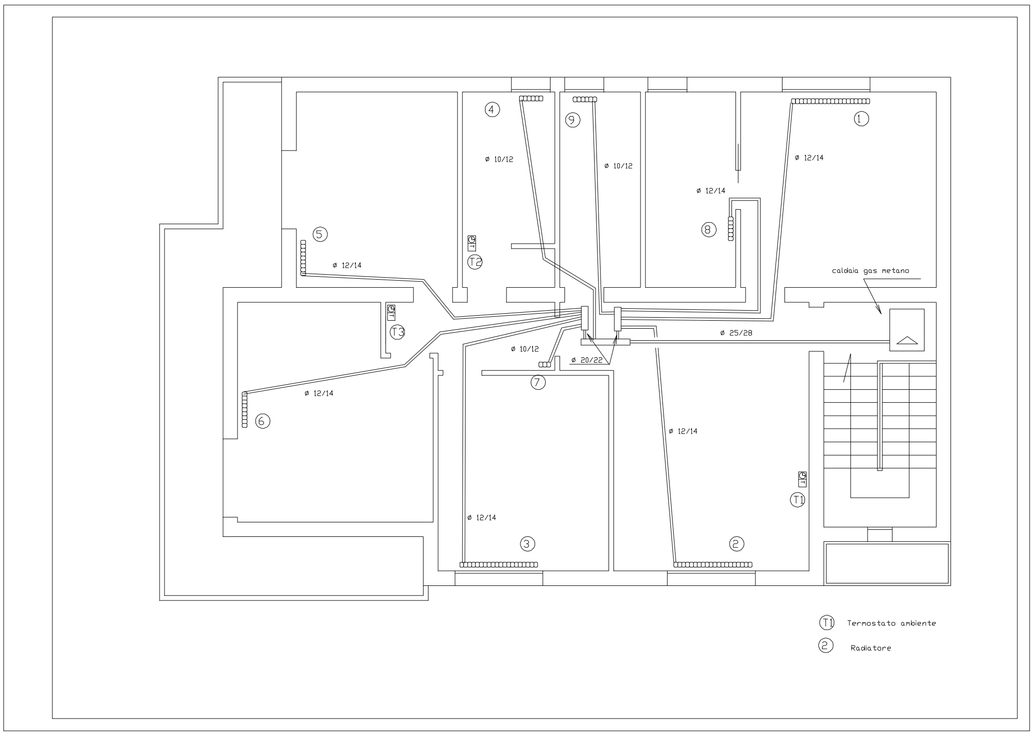
Plant system



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

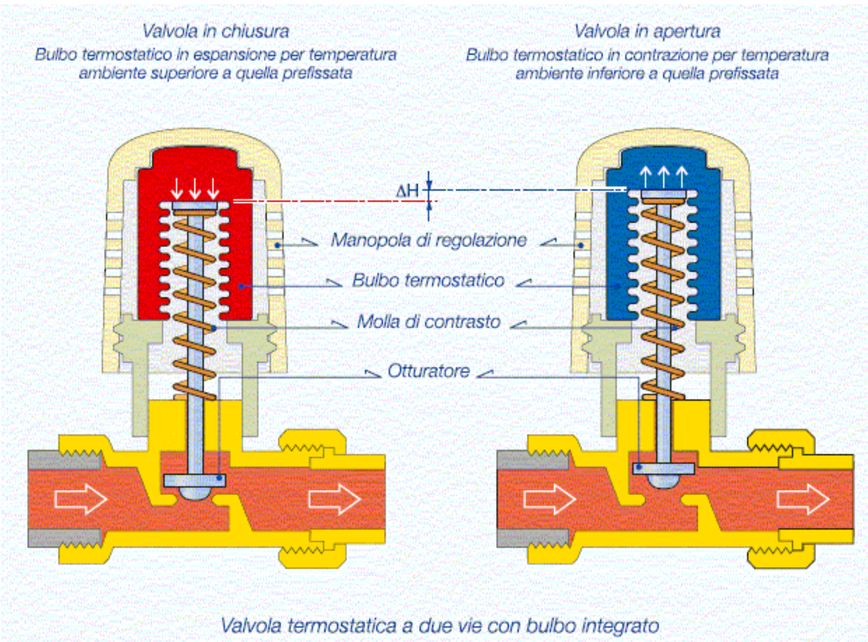
manual control

- 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

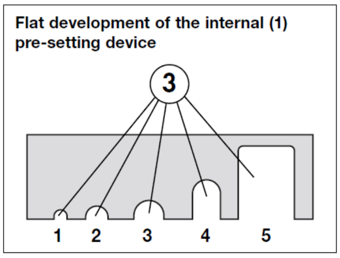
Thermostatic valve head

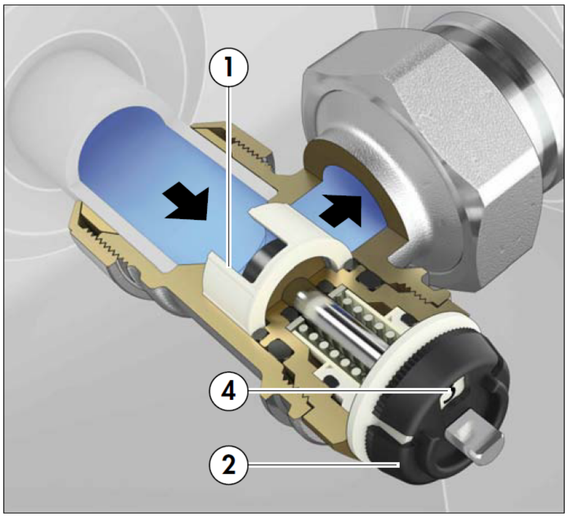


preset valves

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.

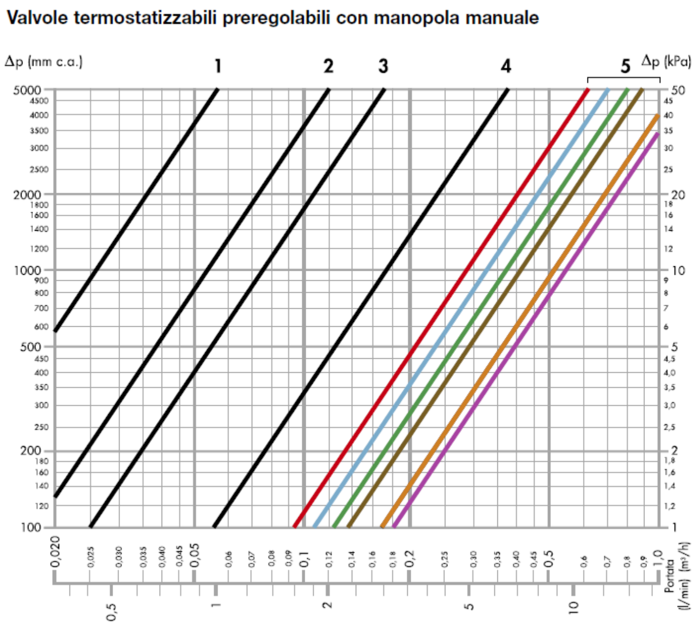




fonte Caleffi

Characteristic pressure loss diagram

Manual control



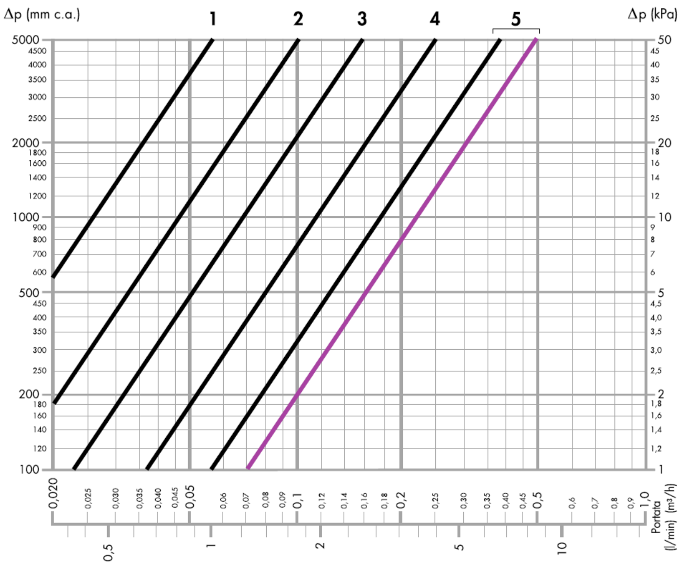
		Kvs (m^3/h)					
Posizione di preregolazione		3/8" squadra	3/8" diritta	1/2" squadra	1/2" diritta	3/4" squadra	3/4" diritta
	1	**0,08**	0,08	0,08	0,09	0,12	0,12
	2	**0,17**	0,17	0,17	0,19	0,22	0,22
	3	**0,25**	0,25	0,25	0,27	0,41	0,41
	4	**0,55**	0,55	0,55	0,56	0,95	0,93
	5	1,30	0,90	1,40	1,00	1,80	1,70

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

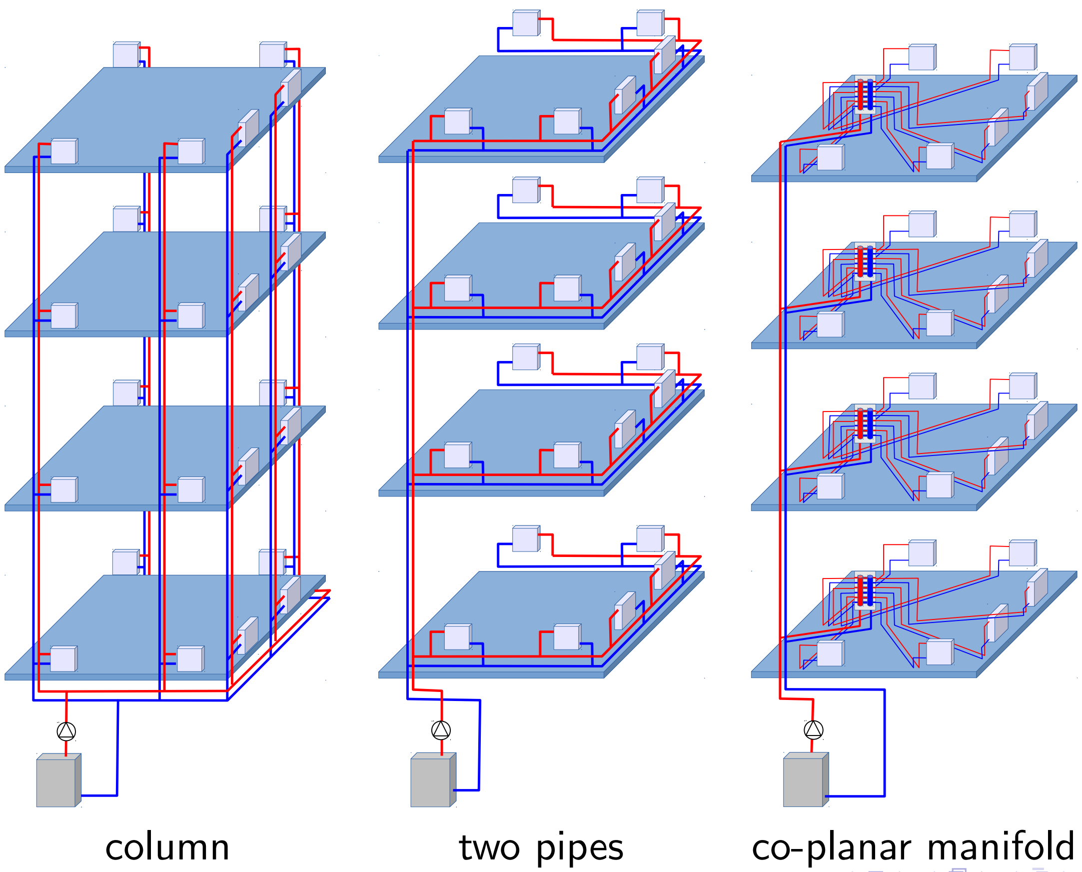
1. for each circuit compute the flow rate
2. define pipe diameter and fluid velocity u_j
3. compute pressure losses, do not consider the pressure loss of valves
4. with preset valves, add the pressure loss with full open valve
5. for each circuit determine the required pressure loss
6. find the set position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



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

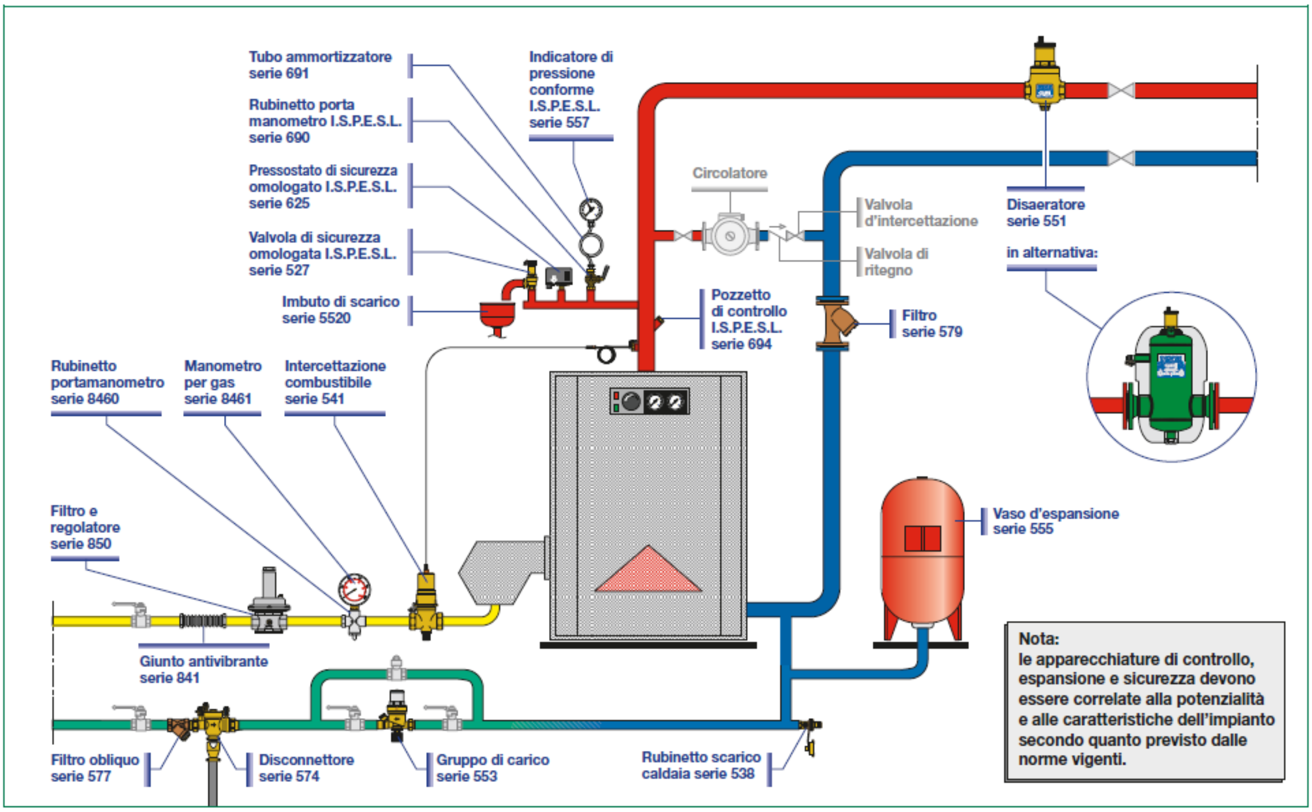
protection devices

- 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

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

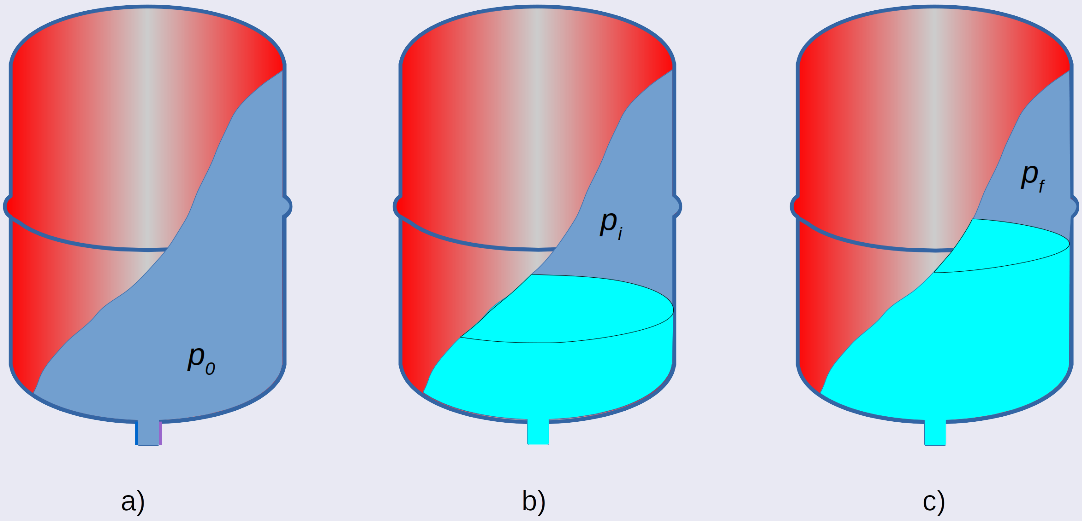
V_A water content

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

t_m maximum temperature in °C

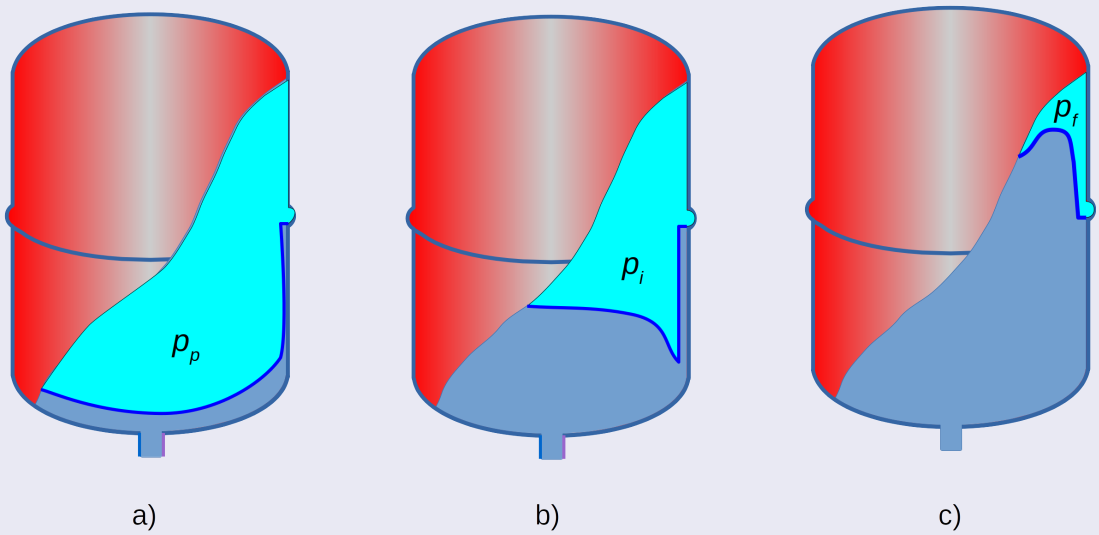
expansion tank

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

with 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

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

E expansion volume

p_p initial pressure

p_f set pressure of pressure relief valve

sizing expansion tanks

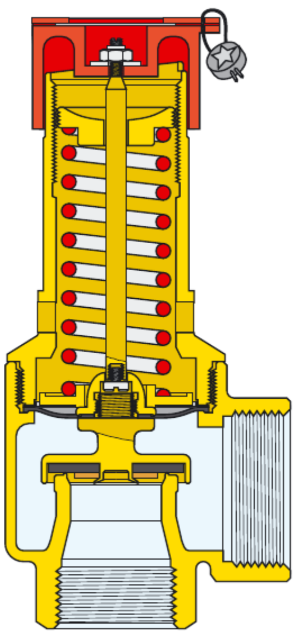
with diaphragm or bladder

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

pressure relief valve

How it works

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