A GUIDE TO UNDERSTANDING HVAC COMPONENTS AND CONTROL SYSTEMS

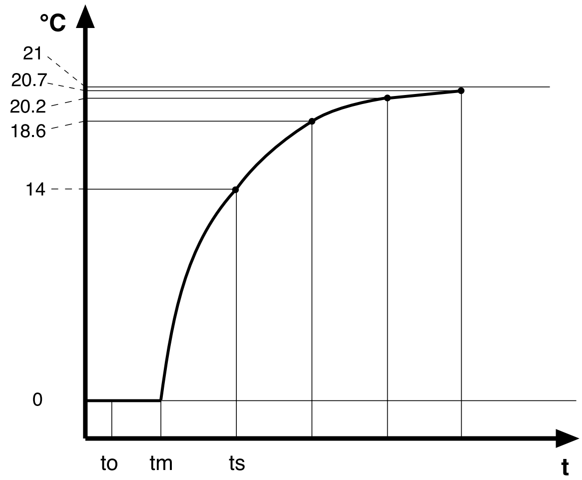
1. GENERAL

1.1 Time constant (ts)

The response following an adjustment to a desired value (e.g. ambient temperature = magnitude controlled) is never immediate.
This behaviour is determined by the “Time Constant”: that is, by the time necessary for the value measured (e.g. ambient temperature) to acquire 2/3 of the total adjustment. The same length of time (i.e. another “Time Constant”) is necessary to acquire 2/3 of the remaining third of the total adjustment, and so on.

Example:

A thermometer which indicates 0 °C is immersed in a bowl containing water maintained at a constant temperature of 21 °C and with a stop watch we note the time that elapses until the thermometer indicates 14 °C (that is, the time necessary to acquire 2/3 of 21 °C).
This time represents the Time Constant (ts) of the thermometer under examination.
Now we know that the same time must elapse for the thermometer to indicate about 18.6°C (that is, to acquire 2/3 of the remaining 7 °C and so on. After five Time Constants about 99.3% of the total variation will be acquired (in practice, we consider the difference to be acquired after 4 Time Constants).



Legend :
t = time
°C = measurement unit of physical magnitude
ts = Time Constant
DT = Dead Time (start of measurement)
to = start of variation

Fig. 1 Representation of Time Constant

When in the technical specifications for a detector the Time Constant value is given, e.g. 10 seconds, we know that this is the time required to measure 2/3 of the variation that has taken place.

The Time Constant depends on the material, weight and the measuring element so that it is specific for the model of a particular manufacturer and is different, as a rule, for models by other makers.

We meet the Time Constant in all detectors measuring any physical variable (temperature, humidity, pressure, etc) and also in the plant components (heat emitters, etc).

In the HVAC field it is important to have available detectors with different Time Constants to meet various requirements:

– detectors with Time Constant of *10.. 25 minutes* for measuring the outside temperature for compensating and similar controllers.
Reason = Shorter times are no advantage (on the contrary, they can lead to unstable control) since changes in external temperature do not have an immediate effect on the room temperature.

– detectors with Time Constant of *20…40 seconds* are used for measuring the temperature of the discharge air temperature in air-conditioning plants.
Reason: the air temperature responds immediately to a change in valve position so the temperature detectors must also react quickly.

– detectors with Time Constant of *3…10 minutes* are acceptable for measuring ambient temperature (10 minutes for heating by radiators or similar, 3 minutes with air handling plants).
Reason : the ambient temperature requires time to adapt to the new situation.

– detectors with Time Constant of 5…10 seconds are necessary for measuring the temperature of DHW circuits (mixture of DHW from the boiler or storage tank with cold water from mains supply).
Reason: the control system is critical because there are simultaneous variations in temperature and flow.

1.2 Dead Time (DT)

This is the time that elapses from the moment a variation occurs (to) to the start of the measurement (intervention of control system):

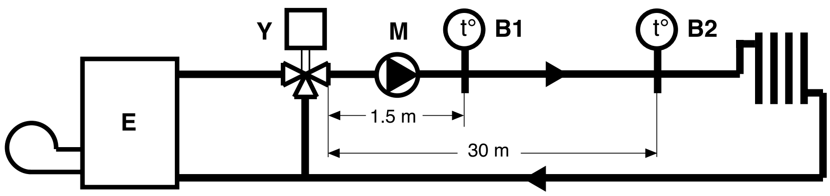


Fig. 2 Example of Dead Time
Considering a water flow speed in the plant of 0.5 m/s:
– with detector B1 the Dead Time is 1.5 m ÷ 0.5 m/s = 3 seconds (insignificant value)
– with detector B2 the Dead Time is 30 m ÷ 0.5 m/s = 60 seconds (excessive value for control)

1.3 Accuracy of control

The control system in a plant has to ensure that, during the transitory period, the response to a variation in the variable controlled has minimum oscillations and that subsequently the desired value (W) is restored.

Since the difficulties in plants to be controlled are always the same (Time Constant (ts), difficulty index ”λ“ and transmission ratio “Ks”) the only solution is to use controllers with suitable control behaviour, and choose measurement detectors with Time Constants suitable for the plant in question and to site them so they measure exactly the variable to be controlled.

2. CONTROL MODES

The traditional control modes are :
- Proportional (P)
- Integral (I)
- Proportional/Integral (PI)
- Derivative (D)
- Proportional/Derivative (PD)
- Proportional/Integral/Derivative (PID)

In practice, those used in HVAC plants are: P, PI and, exceptionally, PID.

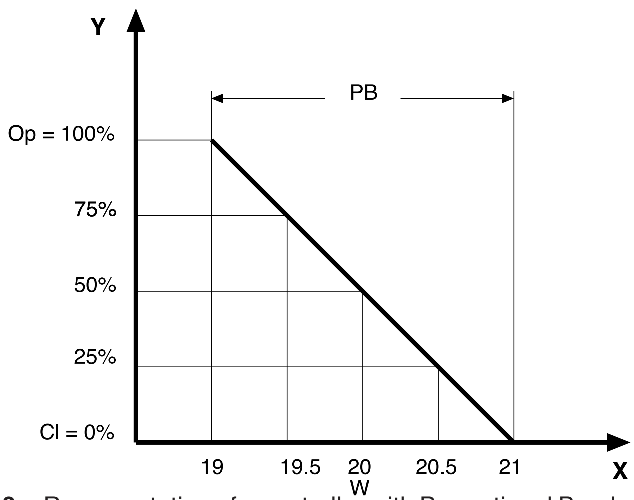
2.1 Proportional control (P)

The actuator (motorised valve, damper actuator, etc) takes up positions proportional to the deviation from the desired value (W).

As a result, the control signal (Y) of a proportional controller depends, within the range of the Proportional Band, **only** on the amount of deviation (Wx) of the variable controlled from the desired value (W): that is to say, the control is directly proportional to the amplitude of the deviation.

2.2 Parameters of proportional control

Proportional Band (PB)
Represents the variation range of the variable controlled when the actuator makes the complete run from open to closed and vice versa.
In the Proportional Band range, to each position of the actuator corresponds a single, and therefore easily identifiable, value of the variable controlled (temperature, humidity, etc).



Legend :
Y = control signal
X = actual measurement
W = desired value
PB = Proportional Band (± 1 °C)
Op = completely open
Cl = completely closed

Fig. 3 Representation of a controller with Proportional Band of ±1 °C (PB = 2 °C total).
The desired value (W = 20 °C) corresponds only to the position of 50% (half run) of the actuator; in other positions the temperature values are different.

In proportional controllers, according to their use, the desired value (W) can be positioned in the PB range :
- in the centre (A), that is at 50% = half run of actuator; typical of controllers with 1 output
- at an extremity (B), that is at 0% = actuator closed; typical of controllers with 2 outputs
- at an extremity (C), that is at 100% = actuator open; for special uses (optimisation)

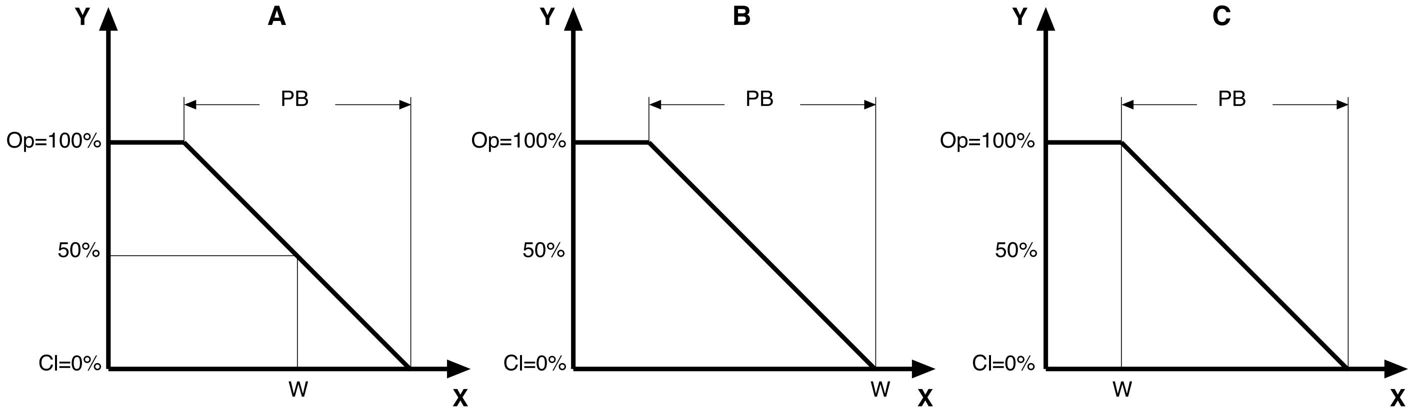


Fig. 4 Positions of desired value (W) in Proportional Band
A = desired value in the centre
B = desired value at extremity and increasing
C = desired value at extremity and decreasing

Permanent deviation (ΔWx) of control

Permanent control deviation (ΔWx = X-W) is defined as all the values, in the Proportional Band range, different from the desired value (W).

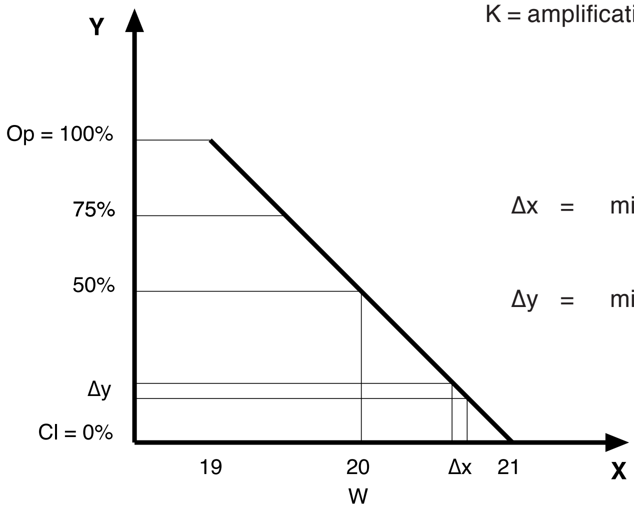
Example :

If, for a period of time, the load (plant demand) remains constant at the value corresponding to valve 75% open (see Fig. 3) the temperature also remains constant at 19.5 °C, that is, with a deviation ΔWx of 0.5 °C (19.5 - 20 = - 0.5).

Amplification or sensitivity factor (K)

The amplification factor is the minimum variation in the variable controlled for which the actuator changes its position: in other words, it is the sensitivity of the controller.

$\Delta Y = K \times \Delta X$ where : Y = control signal
X = value of variable controlled
K = amplification factor of controller



Δx = minimum adjustment value of measurement (= $\frac{1}{20\%}$ of PB)

Δy = minimum adjustment value of measurement (= $\frac{1}{20\%}$ of total run)

Fig. 5 Proportional Band of ±1 °C with amplification factor K = 20

Note : In microprocessor-based controllers the “K” factor is replaced by the *resolution + neutral zone* the value of which (fixed or adjustable) does not change when the value of the Proportional Band (PB) changes and therefore what follows is valid only for analogue controllers.

Example :

Let us consider a controller with K = 20. This means that the minimum value of the control signal (Y) is 1/20 of the total run of the actuator (100% ÷ 20 = 5%), corresponding to an analogue 1/20 of the variation in the variable controlled (temperature, humidity, pressure, etc) in the Proportional Band range.

The minimum control signal is constant, whatever the value of PB set, since it depends on the design of the controller (in our example this is 5% of the total actuator run).

vice versa

The absolute value of the minimum change in the variable that causes the minimum control signal depends on the PB set:

- *with PB of 2 °C Δx = 0.1 °C (2 : 20)*
- *with PB of 10 °C Δx = 0.5 °C (10 : 20)*

2.3 Considerations on proportional controllers

The proportional controller:

- responds promptly to changes in the variable controlled or in the desired value
- is easy to use; the only parameter to set is the PB
- controls the various values of the PB set = permanent deviation, only one operating condition (valve position) corresponds to the desired value
- to reduce the amount of deviation permanently you have to reduce the PB
- however, extremely small PBs can, at limit, convert proportional modulating control into 2-position (On-Off) control.

Example :

If we consider a controller for regulating the temperature of discharge air having a K = 20 and we set a PB of ± 0.25 °C (total = 0.5 °C) this means, as we have seen above, that the valve would change its position for each deviation of = 0,025 °C (0.5 ÷ 20 = 0.025 °C),so that the control becomes On-Off.

Note: Control is considered stable if, when there is a minimum adjustment to the value of the variable controlled, the actuator moves to the new position without uncertainty; on the other hand, control is unstable if the actuator has difficulty in positioning itself.

2.4 Commissioning

Practical rules

1) set a PB suitable for the plant; as a guide, 1...2 °C for ambient temperature control, extract or mixed air, 2...5 °C for discharge air from conditioners.

2) adjust *slowly* on the setting scale of the controller, positioning the actuator in correspondence with the *desired* value, in the PB range (50%; 100% or 0%).
The position found is that of the value measured in that moment by the detector, usually different from that required in operation; for example, when commissioning for ambient control the temperature measured is 18 °C, while that desired in operation should be 21 °C.

3) Provoke a small variation by modifying the value set (18 °C) and observe the response of the actuator:

- if, after a short run, it stops in a new position or, at the most makes a couple of oscillations before taking up a position = stable operation, therefore PB set correctly.
- repeat the previous operation but modifying the temperature in the opposite direction in respect of the value measured..

vice versa :

- if the run is excessive, at the limit of one of the extremes = instable operation, PB set too small.
- increase the PB gradually, repeating at each new value the operations in point **3** until you obtain a stable operation.

In other words, when in operation the actuator should make brief runs followed by pauses.

Warning :

An excessive increase in PB stabilises control but "exaggerates" the permanent deviation and leads to excessively long operating periods at values markedly different from the one desired.
Accordingly, if in order to stabilise the operation it is necessary to set a high PB value, this means that a proportional controller is not a suitable choice.
An example of this is fixed-point control of temperature or relative humidity of discharge air in an air handling unit.

2.5 Use

General

- plants in which the variable controlled is not subject to continuous and sudden variations (load unstable over time)
- plants in which, under certain conditions, operation at values different from the desired one is acceptable (permanent deviation).
- plants handling large volumes (storage) or at constant flows.

Examples :

- *control of ambience (temperature, relative humidity, etc) in which the variable controlled is not subject to sudden variations, thereby permitting narrow proportional bands (PBs) to be set and accordingly small permanent variations.*
- *control of secondary temperature at constant flow by control of motorised valve of primary circuit of a heat exchanger.*

2.6 Uses not recommended

- direct control of DHW circuits, control of valve for mixing hot water from storage tank or boiler with cold water from mains.

Reason : the measurement is influenced by the temperature variations but above all by sudden and continuous changes in flow due to discontinuous withdrawal of water.

- in the control of temperature, humidity, etc, of discharge air in an air conditioner or fan coil.

Reason : small adjustments to the valve position have an immediate effect on the treated air.

Warning :

Special care is necessary when sizing motorised valves controlled by a proportional controller:

– size the valves for the real flow and pressure drop (see section 5)
– oversized valves (diameter larger than necessary) contribute to operational instability .
– use seat valves having equal percentage; the use of slipper, ball, etc valves with linear characteristics is NOT recommended.

2.7 Integral control (I)

Integral control acts on the actuator with a speed proportional to the amount of deviation from the desired value. There is no direct relationship between the deviation and the actuator position as there is with proportional action,

The ratio between actuator speed and deviation (e,g, 1 mm/minute for 0.1 °C) is defined as Integration Ratio (KI).

Integral action gives rise to a control signal (Y) for the time the deviation lasts and becomes progressively smaller until it disappears when the *desired value is reached*.

If the deviation is not annulled, the actuator continues to operate until it reaches one of the extremities of its run.

In the HVAC field, the exclusively integral controller is not used; its use is confined to control in plants with rapid response, without inertia and with slow load variations.

On the other hand, controllers with combined integral and proportional action are in to common use.

2.8 Proportional/Integral controllers (PI)

PI controllers exploit the advantages represented by the rapid response of the proportional controller, in relation to the amount of deviation, and the independence of load of the integral controller.
In the presence of a variation in the variable controlled:

– proportional action intervenes immediately; its control signal adjusts the position of the actuator according to the amount of the deviation and to the proportional band set.

– when the proportional action ends, integral action takes over with a control signal, repeating (Tn) the correction made by the proportional action to cancel out the permanent deviation from the desired value left by the proportional action.

The integral action ends when the desired value is reached,

2.9 Parameter of the integral controller

Integral Time (IT)

Integral Time (IT) is the time required for the integral action to repeat a control signal having the same value as the signal produced immediately before by the proportional action.

Note: At the plant, the response of the PI controller to a deviation from the variable controlled can be recognised by:
– the first control signal continuous over time (proportional action)
and
– repeated control pulses becoming progressively shorter separated by pauses becoming progressively longer (integral action) as the residual deviation from the desired value diminishes.

2.10 Parameters of the proportional/integral (PI) controller

In PI controllers there are two parameters that affect control:
– PB = proportional band
– IT = integral time

These parameters can have values that are:

– fixed = set by the manufacturer, as is usual for compensated heating controllers
– adjustable = for controllers to be used in conditioning, fan coil, etc plants.

2.11 Commissioning

Good practice

When commissioning PI controllers with adjustable PB and IT, proceed as follows:
– set Integral Time at maximum permitted value.
– set PB at a suitable value for plant, as described for controllers having only proportional action (Section 2.4).
– reduce the IT gradually until the system appears stable: that is, a small adjustment to the desired value or to the variable controlled correspond small values in the control signal; that is, the desired value is quickly re-established.

Considerations :
– *PB and IT less than optimal ones* = unstable control, control signal excessively long following a deviation.
– *with small PB* = controller acts as if only integral action functioning: that is, it takes a long time to eliminate the permanent deviation typical of the proportional component (in practice, the initial continuous control signal corresponding to P is lacking).
– *with large IT* = the controller tends to act only with proportional action (the permanent deviation takes an excessively long time to correct).

2.12 Derivative action (D)

The derivative component produces a control signal based on the speed and it does this *only at the moment* when the variable controlled deviates from the desired value.
As a consequence, derivative action is not active when there is no variation of the measurement irrespective of the fact that the variation is constant over time at a different value from that desired.

In short, its action can be interpreted as a warning of the control signal which serves to neutralise the Dead Time but is not able to annul the deviation from the desired value.

Advantage: when Dead Time is less than the response time of the system controlled, the Time Constant that recovers 63.2% of the total variation in variable controlled.

Useless : – with Dead Time ≥ that of the response of the system
– variable with small variations over time

2.13 Parameter of derivative action

Derivative Time (*Tv*)

Derivative Time, expressed in measurement units per time in seconds or minutes (e,g, 2°C/s) is, in practice, the duration time of the action.

2.14 Use

When a plant is in a “Balanced” situation the derivative action is inactive (since there are no variations) so controllers with derivative action cannot be used to control such plants.
On the other hand, the derivative mode in combination with P and PI action is decisive in the control of plants having long Dead Times.

Note : PD action is not discussed further since it has no application in the control of HVAC plants.

2.15 Proportional/integral/derivative action (PID)

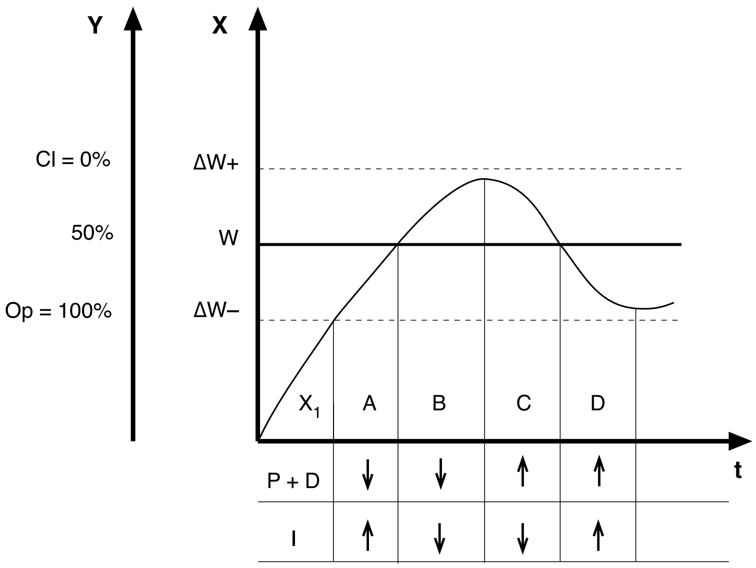
PID controllers use all the three actions creating a control signal on the basis of the speed (D) of the deviation, its magnitude (P) which continues over time (I) until the deviation is cancelled.
When a variation in the variable occurs:

– first the derivative action (D) intervenes with an immediate signal that progressively decreases over the time (Tv) set.
– next the proportional action (P) intervenes, its control signal depending on the PB set and proportional to the deviation.
– finally, the integral action (I) intervenes and this, over time (Tn), produces a control signal with pause/pulses until the desired value is restored.

Note : In digital systems the control actions undergo modifications such as:

– limiting the integral action so that the actuator, if the deviation persists for a long time, assumes the extreme positions.
– excluding the action (I) when the desired value is modified (set point) and in some cases limiting the proportional action.

Operation of a PID controller



X = variable controlled (temperature, etc)
X_1 = lower value of Proportional Band
Y = control signal
W = desired value (PB middle value)
ΔW– / ΔW+ = Proportional Band
t = time
P = Proportional action
I = Integral action
D = Derivative action
↑ = tends to open
↓ = tends to close

Behaviour of the actions:

X_1 = value of variable controlled less than lower value of the Proportional Band, actuator open.

A = variable (e.g. temperature) increasing; **P** and **D** tend to close (the measurement is within the Proportional Band); vice versa, the **I** tends to open as long as the magnitude is less than the desired value (W).

B = variable increasing in respect of desired value (W); **P**, **D** and **I** act together towards closure (the **I** tends to close in order to cancel the residual deviation).

C = variable decreasing; **P** and **D** act towards opening; the **I** tends to close until the magnitude is larger than the desired value (W).

D = variable decreasing in respect of desired value (W); **P**, **D** and **I** act towards opening (the **I** tends to open provided the variable is less than the desired value).

2.16 Commissioning

– set the controller for Proportional mode only, entering the times Tn at maximum and Tv at minimum (excluding the two **I** and **D** modes) and setting a PB so as to obtain a stable control condition.

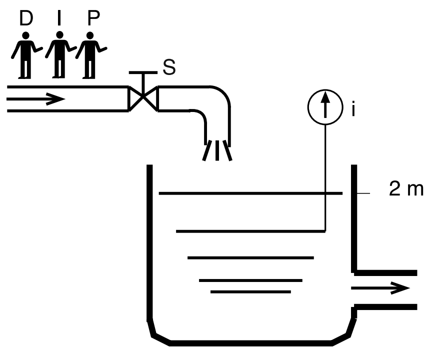
– next reduce Tn and increase Tv, bearing in mind that with :

XP and /or Tn less than the optimal = unstable control

Tv too small = weak control signal when there is a variation in the magnitude controlled; in practice, derivative action almost zero (however, preferable to an excessive Tv value)

Tv too large = control signal immediately excessive in respect of amount of deviation from set-point and so control becomes unstable.

How control modes work : three scenarios



P - I - D = operators

S = sluice gate valve

i = level indicator

→ = withdrawal of fluid

Fig. A

The P, I, and D modes of control can be illustrated by the example shown in Fig. **A**, in which the level of a tank is controlled by **three** operators who, for this purpose, have available the sluice gate valve (S) and the level indicator (i)

The operators know :

– the value of the level to be maintained, which we shall suppose to be 2 m (= pre-set or desired value)

– the total run, from open to closed, of the sluice gate valve (S) which we shall suppose to require 8 complete turns of the handwheel (= valve actuator run).

Proportional action (P)

The control of the water level is entrusted to an operator who is *methodical* by nature and who knows from experience that :

– if the level falls by 0.1 m he must open the sluice gate valve (S) by two turns.

therefore

– a fall in level of 0.2 m (twice the above) requires 4 turns of the sluice gate valve, and so on.

It follows that the sluice gate valve will have to be fully open for a fall of 0.4 m in the water level (= total range of Proportional Band).

From these considerations we can understand his method of operating :

– for a fall in level (= increase in load) shown by the level indicator (= measurement detector) of 0.1 m (= minimum detectable value).

– he opens the sluice gate by two turns without taking into account the desired value.

From this moment, until he notes a new variation in the level, he takes no further action.

By his intervention he has stopped the fall in level since he has introduced the same volume as that withdrawn; nevertheless, he has to admit that the level is below the desired value of 2 meters (= permanent deviation).

By virtue of his *methodical* nature, he does not consider it essential to restore the level to the desired value; he is satisfied if the level is stabilised at a value included in the variation range which experience has taught him (= Proportional Band).

His method of working is the same as that of a controller in the Proportional mode (P) in that: :

– he adjusts the sluice gate valve to positions corresponding to the amount of variation (= deviation).

– he does not bother to restore the desired value

– he accepts that, over time, the level can have values different from the desired one provided they are within a certain range (= Proportional Band).

Intergral action (I)

Imagine that we are giving the same task to an operator who has an ingenuous nature.
When the level indicator shows a fall he starts to open the sluice gate valve gradually, slowing down as the speed of the measuring instrument pointer falls (= speed of actuator according to the variation).

At a certain point the indicator *stops* (= volume input equal to that withdrawn) but the level will inevitably be below the desired value..
This situation does not satisfy our operator - he may be ingenuous but he is not incapable – and so he continues to open the sluice gate valve slowly until the pointer moves to the desired value.

At this point our operator stops, but the level does not - it continues to increase, obviously without a variation in the liquid withdrawn, since now the volume admitted is greater than that released, and so exceeds the desired value and causing the intervention of our operator who gradually closes the valve.

It is easy to imagine that when the level is restored to the desired value the sluice gate valve will have been closed more than necessary, with the consequence that the level decreases further thereby obliging our operator to open the sluice gate valve once again.

This cycle will be repeated continually in exactly the same way as would a controller with exclusively Integral control action in which the speed in operating the sluice gate valve is proportional to the deviation of the level from the desired value.

Derivative action (D)

The control of the level of the tank is entrusted to an operator who is energetic but not very attentive.

Following a sudden increase in withdrawal from the tank, he is quick open the sluice gate by an amount based solely on his estimate of the *speed* with which the level decreases.

He then looks at the level indicator and if he does not notice variations, irrespective of the absolute value of the tank level… he takes no action.

His task starts and ends in the time in which he notes a variation in the position of the level indicator.

If the level does not change, even if its value remains different from that required, this does not worry our operator who continues to take it easy

Unfortunately for him, his rest period does not last long because in order to stop the fall in level he has opened the sluice gate to permit a larger flow than that withdrawn because of the delay in taking the measurement due to the Dead Time and the Time Constant of the measuring instrument (= response of the plant controlled).

It follows - he notes with surprise - that the level starts to rise,compelling him to take action and close the sluice gate, and he does this, as always, only on the basis of how quickly the increase has taken place.

Presumably, unless there were a change in the amount withdrawn from the tank (= constant withdrawal), he would have closed the sluice gate too much …obliging him to open it again.

He has done all this without obtaining the desired value and stabilising the system.

In practice, his efforts are useless but his *energy* proves useful for neutralising the Dead Time (provided it is shorter that of the process reaction), naturally in combination with the two operators we looked at previously.

Proportional/Derivative action (PD)

Considering that none of the three operators we have looked at satisfy us completely, let us see if we can gain some advantage by combining their efforts.
For the first combination we shall entrust the two operators (P) and (D) with maintaining the level of the tank.

When a change in the level occurs, operator (D) acts immediately to prevent any further change in the level (for him, as we know, the desired value is not an imperative condition)..

Next, operator (P) takes charge of the situation, and, in accordance with his way of reasoning, positions the sluice gate at a value (opening or closing) proportional to the amount of the variation.

The two actions combined give rise to :

– a recovery of the Dead Time and of the general plant delays through the intervention of operator (D)

but do not :

– eliminate the permanent deviation typical of the way of working of operator P

To conclude, we have improved the control, but only for specific situations rarely met in heating/DHW plants..

Proportional/Integral action (PI)

In this case the task is given to the operators with (P) and (I) action.

When there is a change in the level the methodical operator (P) positions the sluice gate so as to introduce or reduce a volume of water equal to that removed…here his task ends.

Now it is the turn of his colleague (I), with his fixed idea of the set (or desired) value, who, when he notices that the level is not the desired one, adjusts the sluice gate (opening or closing it) and then pauses to check the result.

After which :

– if the adjustment is sufficient (this never happens) … he takes a rest

If it is not

– he makes a series of progressively smaller adjustments with increasingly longer pauses between each as the prescribed (or desired) value is reached.

The task of operator (I) takes place over a period of time and consists in fine tuning to eliminate the difference left by his colleague.

The two operators integrate perfectly: in fact, PI action is the basis of very many controllers both in the domestic and industrial fields.

Proportional/Integral/Derivative action (PID)

This method of controlling the level differs from the previous one by:

– the use of three operators

– the presence of the energetic operator (D)

The presence of the operator (D) means that his *rapid* and brief intervention, usually of a magnitude greater than that of the deviation, permits recovering the delay times in acquiring the measurement.

This intervention *reduces* the work of colleagues (P) and (I) who act after him as described in the previous paragraph

The action of operator (D), following a deviation from the required value, recovers the desired value in a shorter time, and so is recommended for circuits or systems with sudden load variations.

The only difficulty … so to speak … is that of integrating the work of the three operators (= setting of controller); if the three do not work well together … then the advantages may be nullified.

3. CONTROL VALVES

3.1 General

Valves are classified on the basis of the way they are constructed and their use, for example: valve body, internal plug, type of movement, working pressure, etc.

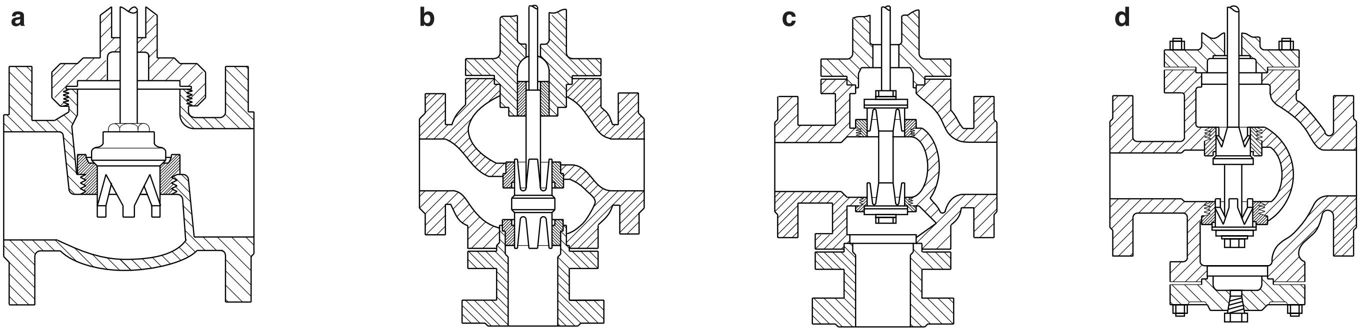
• Classification of valves according to movement of plug::
– slipper valves (rotary movement)
– ball valves (rotary movement)
– seat valves (reciprocating movement)

• Classification of valves according to number of ports :

2-port valves :
– butterfly valves
– ball valves
– seat valves with one seat
– seat valves with two seats

3-port valves :
– slipper valves
– ball valves
– mixing seat valves (two inputs, one output
– diverting seat valves (one input, two outputs)

4-port valves :
– slipper valves
– seat valves (in fact, these have three ports with incorporated by-pass)



Seat valves :
a = single seat valve
b = 3-port mixing valve
c = 3-port diverting valve
d = double-seat valve

Slipper valves :
e = 3-port valve (mixing or diverting)
f = 4 port valve

Fig. 6 Cross-sections of valves

3.2 DN = Nominal diameter (in mm or inches)
Nominal diameter = dimension of port or ports in the body of the valve

3.3 PN maximum pressure of valve body (kPa or bar)
Maximum pressure permitted by valve body; this depends mainly on the material used in the manufacture of the components (body,seat, plug, packing gland, etc)
The PN of valves used in HVAC plants are: PN 6, 10, 16, 25 and 40, with a prevalence of PN 6 and PN 10.
The PN of the valve must be greater than, or equal to, the sum of the static and dynamic pressures of the plant.

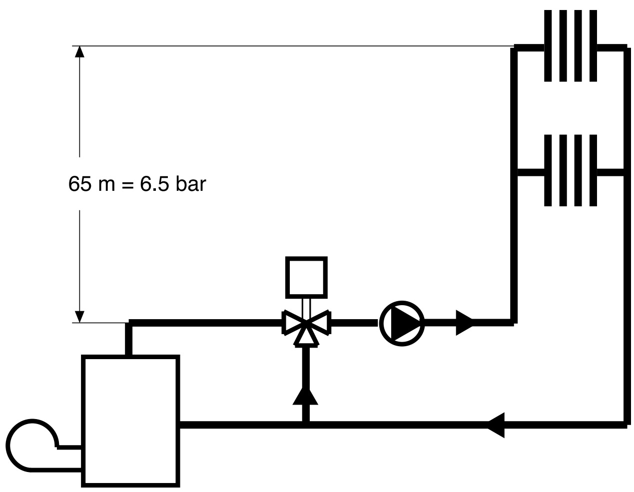


Fig. 7 Valve installed in the boiler plant of a building with heat emitters sited up to a height of 65 m, a situation requiring the use of **PN10** valves

3.4 Δpv = pressure differential permitted by valve body (kPa or bar)
Maximum pressure differential (or pressure drop) permitted with valve completely open; that is, pressure difference between upstream and downstream of the valve in plant

This value can be appreciably less than that of the PN and of the power available to the actuators
The value is indicated by the manufacturers and determined by manufacturing constraints.

Example :

For a series of PN10 valves use is limited to a pressure of 100 kPa (1 bar) because the seat is contoured directly in the cast iron body and higher working pressures can be the cause of erosion with consequent let by when closed.

Accordingly, even if the actuators permit use at higher pressures this is not feasible because of the way the valve body is constructed.

3.5 Kvs = flow rate (m^3/h)

Flow rate = feature of diameter of a valve of a particular series: 2- or 3-port slipper, seat, etc.

The Kvs indicates the flow *in m^3* of water, in the temperature range 5…30°C, which flows in one hour with valve completely open and constant pressure drop of 100kPa (1 bar).

The Kvs value is obtained and indicated by the manufacturers and serves :
– for sizing the valve
– for comparing valves from different manufacturers

Example :
Valve PN 16 - DN 40 :
– from manufacturer **A** = Kvs 25
– from manufacturer **B** = Kvs 32

Note: Countries that do not use the metric system use the term “Cv” instead of “Kvs”

The conversions are as follows :
– Kvs = 0.856 x Cv
– Cv = 1.167 x Kvs

3.6 Kvr = minimum adjustable value (m^3/h)

Minimum adjustable value of Kvr, that is, the minimum flow that still conforms to the manufacturer’s specification below which it functions as ON-OFF.

Values between 1…3% of total Kvs are the norm for seat valves for use in HVAC plants.

Example :
Valve with Kvs 10 and Kvr of 2% = means that 200 l/h cannot be controlled according to the specification of valve.

3.7 Δp max. pressure difference permitted by actuator (kPa or bar)

Maximum pressure difference permitted by the actuator mounted on valve body

Value that **cannot** be greater than :
– Δpv indicated by manufacturer
– PN of valve

this depends :
– on power of actuator
– with equal actuator power, on valve diameter

and is limited in the plant :
– by the noise caused by the high speed of the fluid flow
– by kettling, consequence of the excessive speed of the fluid which, in its passage between the plug and seat causes a vacuum (with consequent change in state from water to vapour) followed by an increase in pressure after the seat.

The kettling phenomenon can be recognised by :
– the unacceptable level of noise, particularly when the valve is near closure
– the vibrations of the valve stem and, in extreme cases, of the connecting pipework

and can cause :
– removal of material from inside the valve body, damaging the seat and plug (and so reducing the life of the valve)
– damage to the actuator

Nota : In the HVAC plants with which we are concerned it is practically impossible to come across the kettling phenomenon.
It can occur in district heating substations when the pressure between upstream and downstream of the control valve, in the absence of a differential pressure control, can be10m or more.

3.8 Kvo value for basic feature (0.01...0.02)

Value for defining mathematically the basic feature of the valves (linear, equal percentage, etc)

3.9 Let by (0.05...2% of Kvs)

Permitted loss when valve closed. The values are indicated by the manufacturer

In HVAC plants these values are tolerated.

3.10 Operating curve

Relation between the Kvs and the valve run: that is, the variation in flow in relation to the valve opening.

The valves that concern us in the HVAC field can be divided into :
- valves with linear operating curve
- valves with equal percentage operating curve
- valves with linear/equal percentage operating curve

Valves with linear operating curve are :
- slipper, ball, butterfly and similar

Valves with equal percentage operating curve are :
- 2-port seat used in HVAC plants (in the field of industrial processes valves with linear and special operating curves are also used)

Valves with equal percentage/linear[1)] operating curve are :
- 3-port seat characterised by equal percentage throughport and by-pass linear

or
- by-pass equal percentage/linear

1) This design solution has the advantage of maintaining practically constant the flow through the port which is always open; and the disadvantage (if one can call it that) of being obliged to use throughport with equal percentage feature for control.

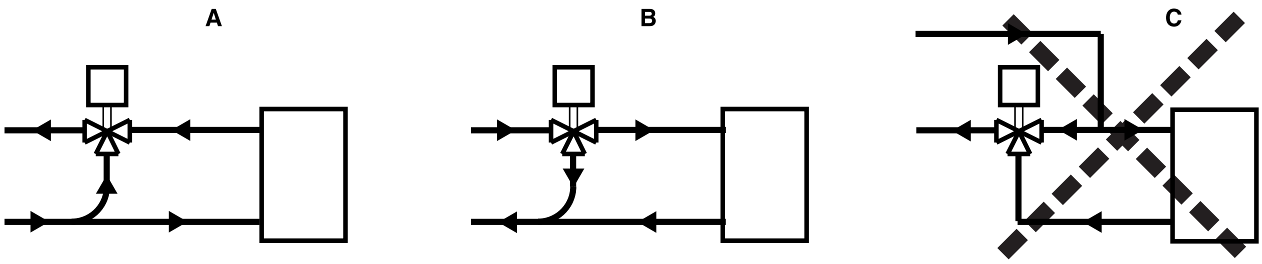


Fig. 8 Mounting 3-port seat valve (throughport equal percentage, by pass linear) for control of batteries and primary of heat exchangers.
A = as mixing valve (2 inputs)/1 output) – preferred mounting
B = as diverting valve (1 input/2 outputs) – permitted mounting
C = mounting **not permitted** (the control port is that of by pass!)

Linear performance curve :

Valves in which to equal absolute values of run in respect of total correspond equal percentage changes in Kvs

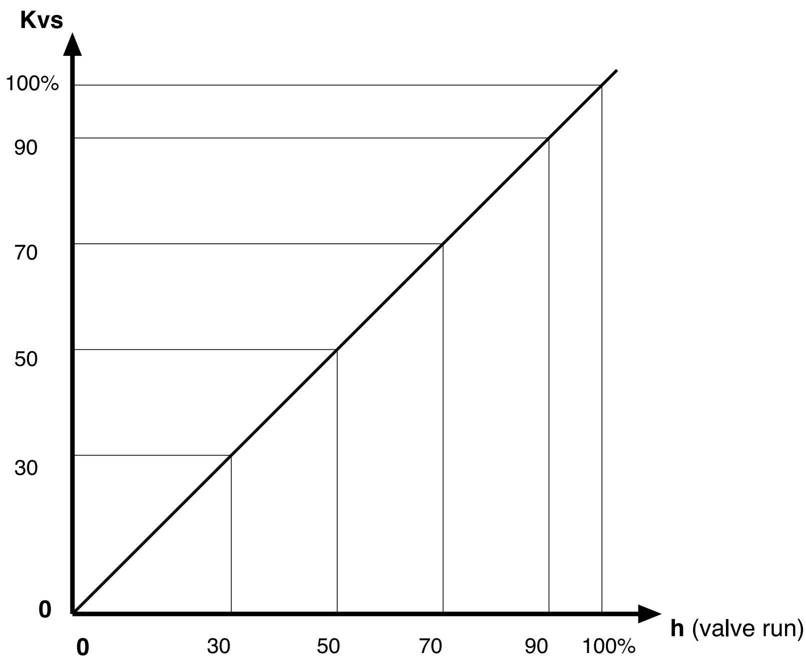


Fig. 9 Linear performance curve of the base valve (in practice, the curve does not end at 0, zero, but at the value of the Kvo)

Example :

Variation in the valve run from 30% to 50% (= 20% in absolute value) gives rise to a variation of the Kvs identical to that caused by a change in the valve run from 70% to 90% (= 20% in absolute value)

Equal percentage performance curve :

Valves in which to equal variations, *in absolute value*, of the run in respect of the total always corresponds the same *absolute percentage* of Kvs variation in respect of the previous Kvs value.

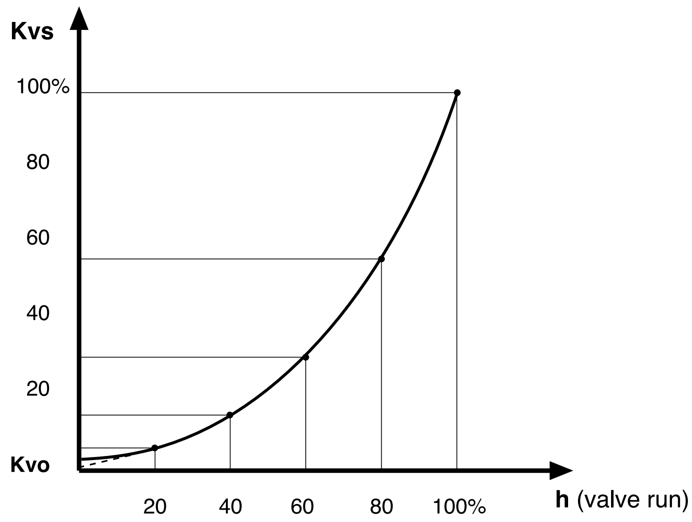


Fig. 10 Basic equal percentage performance curve of the valve: in fact, near to closure the run becomes linear ( ) to diminish the let by.

Example :

Adjustment of the valve run from 30% to 50% (= 20% in absolute value) gives rise to a variation in Kvs of 85%[1)] of that corresponding to the 30% run.

Similarly, a variation in the valve run from 70% to 90% (= 20% in absolute value) changes the Kvs by 85% in respect of that relative to a 70% run of the valve.

1) The curve of the equal percentage performance curve is obtained from a mathematical equation Ngl logarithm $\frac{Kvs}{Kvo}$
The percentage shown in the example (85%) refers to Ngl = 3.3
The Ngi values considered by manufacturers are in the range 3…3.91 and give origin to various more or less concave curves.

The analysis of the equal percentage curve Ngl = 3.3 shows that:

– the valve run from 0…60% (approx) gives rise to a change in flow (Kvs = flow) of 0…30%, that is, at the start of valve opening *to large runs correspond small variations in flow*

– the remaining flow (from 30% to100%) is obtained during the remaining valve run from 60% to 100% (= 40% in absolute value), that is, *to small runs correspond large variations in flow.*

Equal percentage / linear performance curve:

The combination of the two performance curves is used in 3-port seat valves so that the flow in the mixing port (always open) is as constant as possible for the whole run.

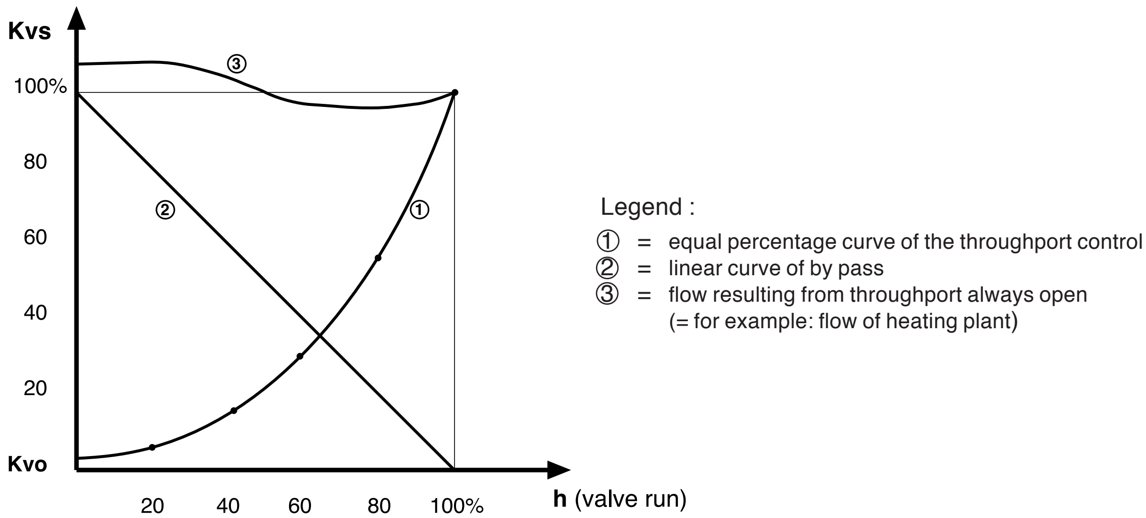


Fig. 11 Equal percentage/linear performance curve modified by Pv 0.5 authority of 3-port seat valve installed in plant

The performance curve of the valves is a determining factor in the control of plants as we shall see in the section "Hydraulic circuits"; however, with present-day microprocessor-based digital controllers we have reached efficiency levels that compensate to a large extent for any errors in the choice and sizing of valves in the plants..

3.11 Sv = control capacity = $\frac{Kvs}{Kvr}$ (30...100)

Control capacity, defined as the ratio between the Kvs with valve completely open and that (Kvr) still adjustable with valve nearly closed..
In practice it is a parameter that defines the quality of the valve.

For the valves which interest us the Sv values are in the range 30...100

Example :

Valve with Kvs = 10 and Sv = 50 the non-adjustable flow (i.e. flow which does not respect the valve operating curve) is equal to:

$$Kvr = \frac{Kvs}{Sv} = \frac{10{,}000\ l/h}{50} = 200\ l/h$$

3.12 Pv = valve authority = $\frac{\Delta pv}{\Delta pvc}$ (0.1...1)

Valve authority derives from the relation between the pressure drop with the valve completely open and with nominal flow and the pressure difference between the upstream and downstream with valve closed.
With the control valve inserted in a hydraulic circuit the pressure of interest to it varies according to the position between open and closed.

This is due to resistance by the plant components such as: gate valves, pumps, DHW/heating/auxiliary circuit components, etc, which cause pressure drops that increase to the power squared of the increase in flow.

As a result, to each change in the control valve run corresponds a change in the flow, and consequently in the pressure, which *deforms* the basic operating curves (linear or equal percentage).

The extent of the deformations to the operating curves depends also on the Pv authority :

– *with 0.1* : the *equal percentage* curve tends towards a straight line (in practice, becomes linear)

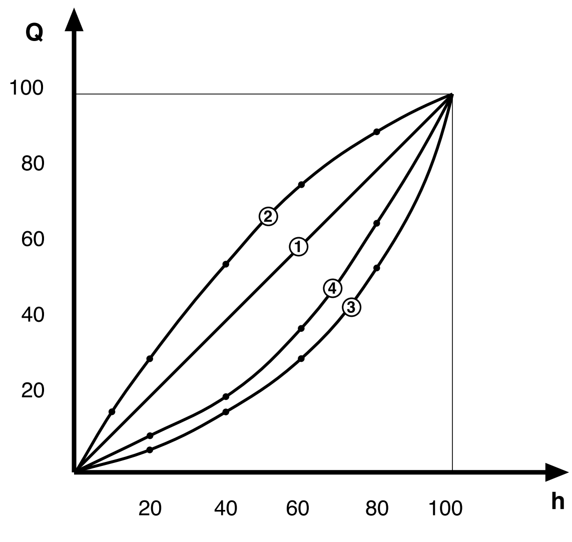
vice versa

the *linear* curve bends, but in the opposite direction to the equal percentage one (in practice, cannot be used)

– *with 1* : both the *equal percentage* and the *linear* curves tend towards the basic form established by the valve manufacturer.

The authority of the valves on the plant varies also in relation to their sizing :

– diameter larger than that "acceptable" = Pv authority less than 0.5
– diameter equal to that "acceptable" = Pv authority equal to 0.5
– diameter less than that "acceptable" = Pv authority greater than 0.5



Legend :
1 = basic linear curve (Pv1)
2 = linear curve modified by Pv 0.4 authority
3 = basic equal percentage curve (Pv1)
4 = equal percentage curve modified by Pv 0.5 authority

Fig. 12 Example of basic performance curves for the valves modified in plants by Pv authority; defined also as dynamic

Note : as demonstrated in the section "Valves and plants", for "acceptable" diameter we mean the diameter that causes a pressure drop with the valve completely open at least equal to that of the section of the circuit with variable flow which affects the control port of the valve, a condition in which the Pv is equal to 0.5

4. VALVES AND PLANTS

4.1 Plant curves

The power released by the batteries, radiators, heat exchangers, etc never has a linear trend i.e. a trend proportional to the flow or to the temperature of the incoming fluid.

Diagrams 13, 14 and 15 give an idea of the power released (Q) in relation to the flow (V); the values are only indicative; in practice they are within a range the width of which depends on:

– the temperature difference between the primary fluid which releases heat or cold (primary of heat exchangers, fluid which passes through the batteries) and that of the secondary which is heated or cooled (secondary of heat exchanger, down-stream air of the batteries).

– the greater the temperature difference, the more accentuated is the curve of energy released.

– with variable flow the graph is more curved than those with constant flow.

– the curve tends towards a straight line in compensated control of a heating plant or heat exchangers.

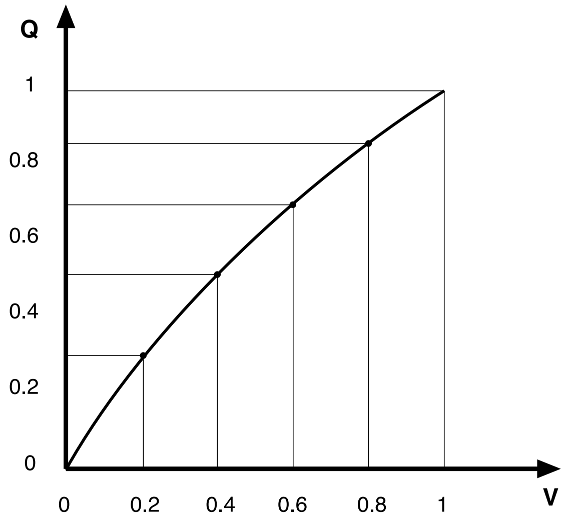


Fig. 13 Heating with compensated control = constant flow with temperature varying according to that outside

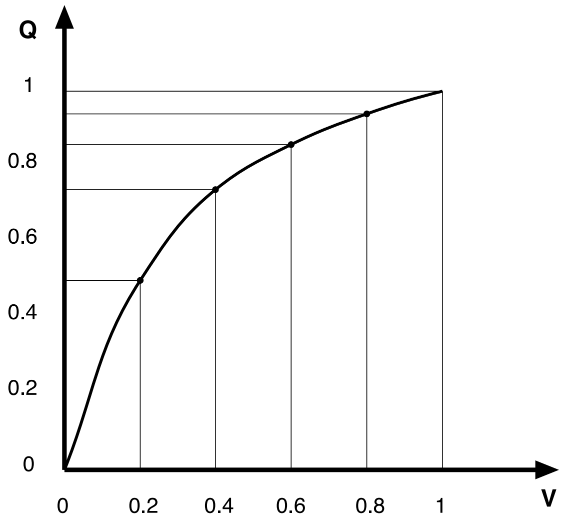


Fig. 14 Battery or heat exchanger with variable flow and constant temperature

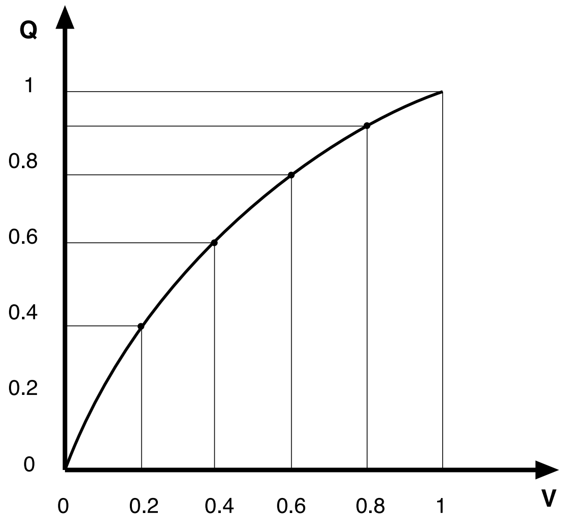


Fig. 15 Battery or heat exchanger with constant flow and variable temperaturee

From an inspection of Fig, 14, batteries or heat exchangers, with variable flow, practically all the control valves, of the batteries and of the heat exchangers we can deduce that:

– to a flow of 20% corresponds a power release of about 55%

therefore

– to obtain the release of the remaining 45% of power the flow has to increase from 20% to 100% (80% in absolute value)

For optimal operation of the control system it is desirable that the relation between the desired power requested by the DHW/ heating/auxiliary circuits and the positions of the control valve take place according to a curve with a linear tendency so that to equal adjustments of the valve run correspond proportional variations in the power released.

To achieve this the performance curve of the valve has to be in opposition to those of the power released by the DHW/heating/ auxiliary circuits so that the resultant of the two curves tends to a straight line.

When the above conditions are not met the control can be:

– unstable at the minimum requests for power as a result of the excessive *steepness* of the curve

– slow in responding to the maximum and unexpected calls for power because the curve is too *flat*

4.2 Use of the valves

The operation of the control system is made easier when, besides being correctly sized, the valves used have performance curves suited to the plant in question:

Central heating plants with compensated control:

– 3- or 4-port slipper valves, that is, valves having linear performance curves

or (even if not indispensable)

– 3-port seat valves having equal percentage throughport and linear by pass

Injection plants, in particular for circuits inserted in distribution manifolds

– 3-port seat valves with performance curves as above

or (taking maximum care in sizing)

– 3-port slipper valves (4-port slipper valves NOT recommended)

Heat exchangers and batteries for use both with variable flow and constant temperature (represents almost all of the applications) and with constant flow and variable temperature

– 3-port seat valves having operating curves as above

or

– 3-port seat valves with equal percentage performance cerve

Note: for pre-heating batteries in air-handling plants (particularly if large ones), sited in extremely cold areas, injection circuits (“d” or “f” in figure 16). are to be preferred.

Reason : by decreasing flow the distribution of the water in the battery ranges is no longer “uniform”; as a result the same happens to the temperature of the air coming from the battery with the result that, because of stratification, unwanted interventions by Frost Protection take place.

5. SIZING OF CONTROL VALVES

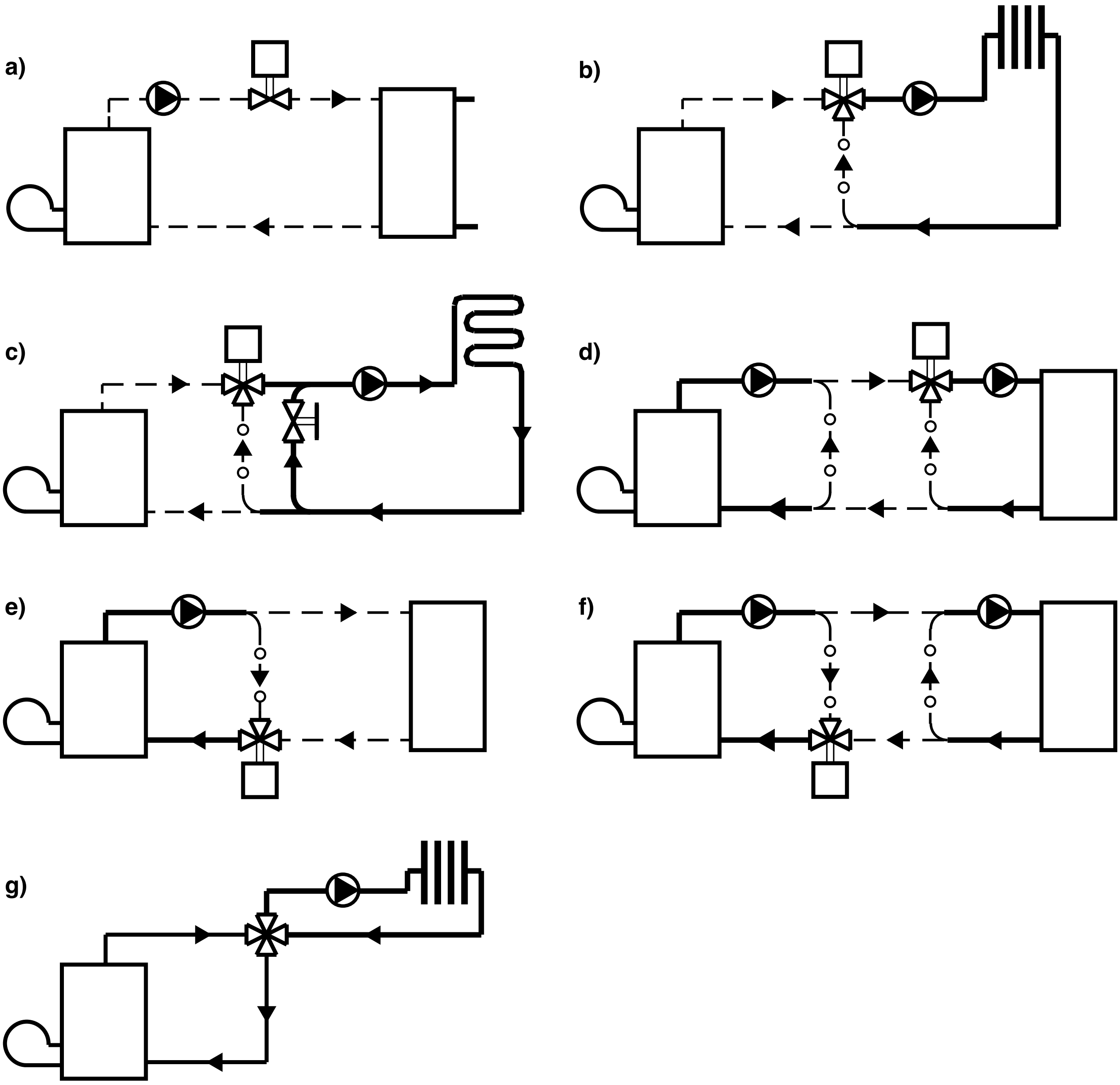


Fig. 16 Plants showing those sections of the circuit with variable flow which, since they involve the control port of the valve, must be taken into account when sizing

Legend of figure 16 :
section of circuit with constant flow
— ○ — section of circuit with variable flow that does not concern the sizing of the control valve
— — — section of circuit with variable flow, the pressure drop of which, with design flow, requires the control valve to be completely open

a) 2-port seat control valve for heat exchangers and batteries having variable flow and constant temperature
b) 3-port mixing control valve for heating plant with traditional heat emitters
c) 3-port mixing control valve for heating plant with panels
d) injection plant with 3-port mixing valve on flow to DHW/heating/auxiliary circuits and primary and secondary pump
e) 3-port control valve (mounted on return) for heat exchangers and batteries with variable flow and constant temperature
f) injection plant with 3-port valve on return of primary circuit
g) 4-port control valve for heating plant having traditional heat emitters

5.1 Sizing criteria

The control valve must be sized so that:

– the pressure drops with the control port completely open, and with the design flow, must be **at least** equal to that of the section of the hydraulic circuit having variable flow concerned with the control port of the valve (see figure 16) essential condition for obtaining a valve authority (Pv) equal to 0.5 :

Reason : the valve authority (Pv). Obtained from the following equation must be greater than or equal to 0.5:

$$Pv = \frac{\Delta pv}{\Delta pv + \Delta pc}$$

Δpv = pressure drop with valve open and with design nominal flow
Δpc = pressure drop of section of circuit with variable flow

Note:The analysis of valve sizing concerns only those with *modulating control*, since for shut-off valves (On-Off control) it is not important and valves with the same diameter as the pipework can confidently be used.

5.2 Data necessary for sizing

The diameter of the control valve can be established once we know:

– the flow

and

– the pressure drop of the section of the circuit with variable flow

or

– the power (or energy)[1)]

and

– the pressure drop of the section of the circuit with variable flow

– the design temperature difference between flow and return

5.3 With the above data we can obtain the valve diameter

– from the specific diagrams both for *water* and for *steam*

– from mathematical equations (more accurate method)

Two diagrams are available :

– *specific* : usually included in the technical data sheets: for each series of valves we can obtain the diameter directly according to the pressure drop

– *general* : both for *water* and for *steam* from which, according to the pressure drop, we can obtain the Kvs and with this, using tables, the valve diameter

5.4 From mathematical equations :

• **for water** hot (maximum 110°C) and cold (without antifreeze)

$$Kvs = \frac{Q}{\sqrt{\frac{\Delta pv}{10}}} \quad \text{from which we obtain} \quad \Delta pv = \frac{Q^2}{Kvs^2} \times 10 \quad \text{and} \quad Q = Kvs \times \sqrt{\frac{\Delta pv}{10}}$$

where Q = m^3/h (design flow with valve open)
Δpv = m WG (pressure drop with control port open)

1) Knowing the power we can obtain the flow:

– **for water** it is necessary to know the temperature difference flow/return from which we can derive the flow $Q = \frac{P}{\Delta t}$

where Q = l/h or m^3/h (design flow)
P = kW or kcal/h (design power or energy)
Δt = °C (temperature difference)

Note : with P expressed in kcal or kcal/h we obtain the flow (Q) in l/h
with P in kW or kWh we obtain a number which multiplied by 860 expresses the flow (Q) in l/h

• **for saturated steam** :

$$Kvs = \frac{Q}{22.4 \times \sqrt{\Delta pv \times P2}} \quad \text{and so} \quad \Delta pv = \frac{Q^2}{(22.4 \times Kvs)^2 \times P2} \quad \text{and} \quad Q = 22.4 \times Kvs \times \sqrt{\Delta pv \times P2}$$

where Q = kg/h (flow of steam with valve open)
Δpv = bar (pressure drop with valve open)[2)]
$P2$ = bar (absolute vapour pressure of steam coming from valve)[3)]

– **for steam** knowing the power we can obtain the flow[4)] from the equation: $Q = P \times 1.6$ or $Q = \frac{P \times 860}{540}$

where Q = kg/h (nominal flow)
P = kW (nominal power
1.6 = average calorific power of a kg of steam

2) The pressure drop to be assumed should be agreed with the plant designer; when this is not possible, assume a value equal to 30% to 50% of the pressure of the incoming steam (however, always below the **critical** pressure drop, the value of which is shown in the diagrams)

3) Absolute pressure = relative pressure + 1 bar (the relative pressure is that which is usually indicated and corresponds to the operating pressure, that is, the pressure read on the manometer of the steam generator).

4) When in place of the flow you know the power (or energy).

6. PRACTICAL EXAMPLES OF VALVE SIZING

Warning : When sizing control valves is our responsibility it is necessary for the pressure drop to be approved by the plant designer or installer. Accordingly, this must be specified in the offer and referred to in the negotiations.

Reason : it is not always possible to know the pressure drop of the section of the circuit with variable flow which serves to size the diameter of the valve and so the customer might not approve of our choice.

Note : In the following examples the pressure drops considered take into account the values acquired under practical conditions; that is, the actual pressure drop of the section of the circuit with variable flow is not taken into account.

6.1 Valves for central heating plants with radiators, convectors, etc and compensated control

3- and 4-port slipper valves (see circuits “b” and “g” in figure 16)

Practical rule : When the design data are not known :

– with plant flow pipes up to DN 80 = valves having same diameter as the pipework

– with plant flow pipes larger than DN 80 = valves having a smaller diameter than that of the pipework

Reasons :

– constructional details of the valve dictate that the pressure drop must be within the range **0.4…0.6** m WG with a maximum limit of **0.8** m WG

– the influence of the section of the circuit with variable flow (boiler) can generally be ignored in respect of the section with constant flow (DHW/heating/auxiliary circuits)

– the plant curve (see section 4.1) tends to linear

– the flow temperature at the heat emitters varies in relation to the outside temperatures according to the climatic curve set on the controller; this operating curve conforms to that of the heat coming from the emitters

Conclusion :

– the sole purpose of the valve is to mix hot water from the boiler with the cooler return water from the heating plant.

This is its only task, so that once in place the valve should not have to be given a thought… as long as it doesn't seize up! (On the other hand, the controller has to be “intelligent”.)

6.2 Sizing with design data available

The diameter of the valve can be determined in two ways according to the data available :

1) if we know :
– the *flow* Q in l/h or m^3/h

2) if we know :
– the *power* P in kW or Kcal

or

–the *energy* E in kW or kcal/h

and

– *the design temperature difference Δt* between flow and return in the heating plant.

6.3 Solution of case 1) using sizing chart 1 for slipper valves

Data available: flow is 37,000 l/h (= 37 m^3/h)

• solution using the sizing chart (flow - pressure drop) for slipper valves

Procedure :

– draw a vertical straight line from value 37m^3/h on the flow axis (abscissa) until it meets the horizontal straight line drawn from the pressure drop scale from 3 to 6 kPa (ordinate)

– the points where it meets the sloping straight lines indicate the Kvs value.

For the 37m^3/h we take into account the Kvs with acceptable pressure drops (see section **6.1**) :
– Kvs 240 gives a pressure drop of 2.5 kPa (0.25 mWG)
– Kvs 170 gives a pressure drop of 4.8 kPa (0.48 mWG)

In the price list of 3-port slipper valves we find the ones corresponding to our Kvs values are :

– 3F DN 100 = Kvs 240
– 3F DN 80 = Kvs 170

Between the two options the choice falls on valve **3F DN 80** with a **CVH** actuator.

6.4 Checking the pressure drop using the mathematical equation : $\Delta pv = \frac{Q^2}{Kvs^2} \times 10$

Substituting the values in the equation we have $\Delta pv = \frac{37^2}{170^2} \times 10 = 0.473$ mWG (4.73 kPa)

i.e. a value practically equal to that found using the chart.

6.5 Solution of case 2) in which we know :

– the power or energy equal to 425 kW (365,500 Kcal)

– the temperature difference = 20 °C

Procedure :

To determine the flow we use the equation $Q = \frac{P}{\Delta t}$ (see note 1) in section **5.4**)

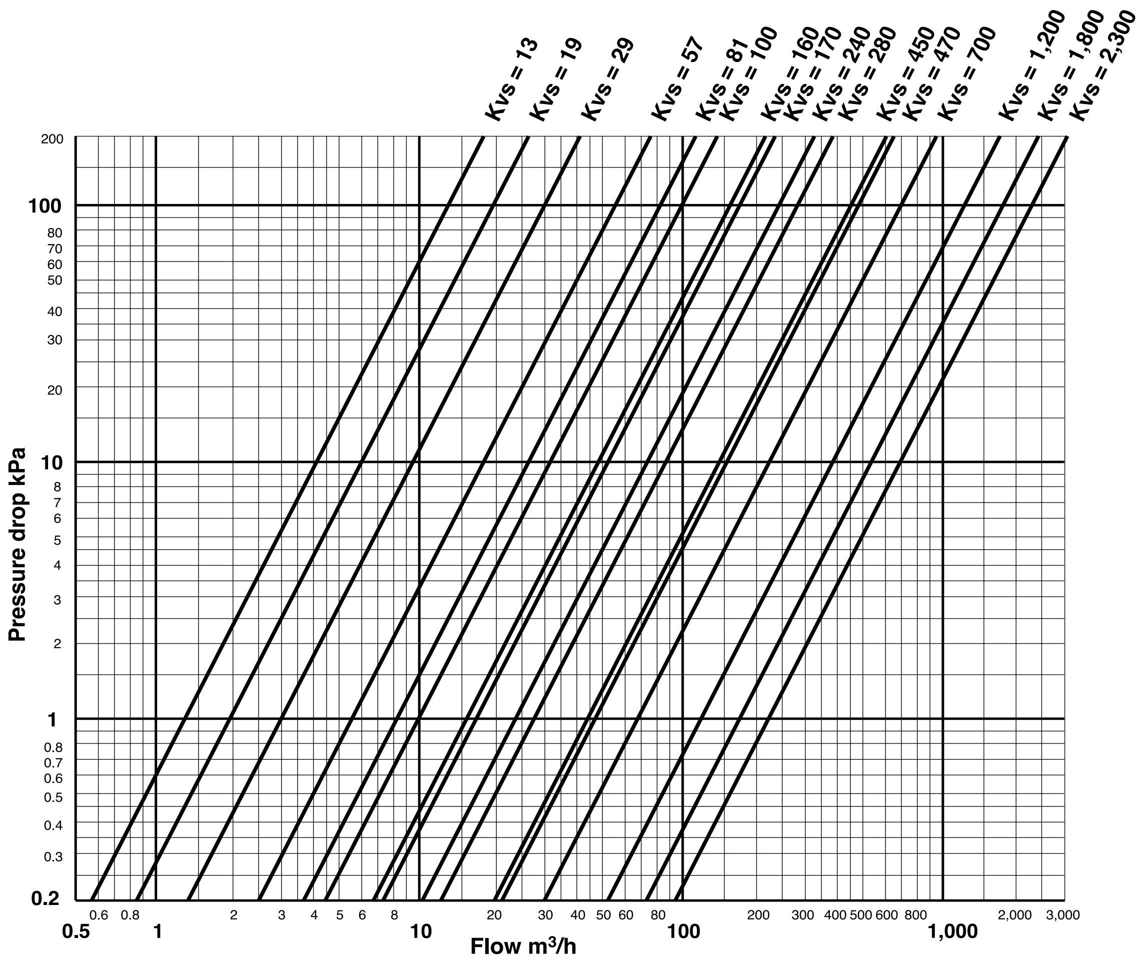
– with power expressed in kW we have : $Q = \frac{425}{20} \times 860 = 18{,}275$ l/h (approx 18.27 m^3/h)

– with power expressed in Kcal we have : $Q = \frac{365.500}{20} = 18{,}275$ l/h (approx18.27 m^3/h)

Having found the flow, we obtain the diameter of the valve by proceeding as described in section **6.3** :

– **3F DN 65** , with Kvs 81 and pressure drop of 5 kPa (0.5 mWG) with **CVH** actuator.

6.6 Check on pressure drop using mathematical equation : $\Delta pv = \frac{18.27^2}{81^2} \times 10 = 0.5$ m W.G. (5 kPa)



Sizing chart 1 General sizing chart for water for 3-or 4-port butterfly and slipper valves

6.7 Valves for heating plants with panels, compensated control and boiler at fixed point (70...80 °C)

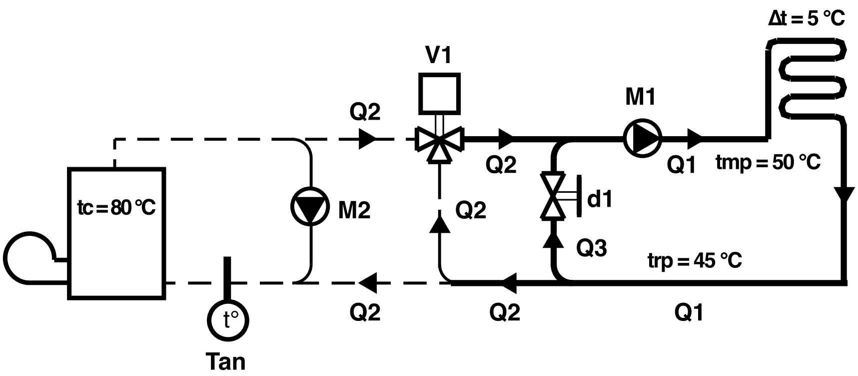
3- and 4-port slipper valves (see circuit “c” in figures 16 and 17)

General :

The control valve of plants with working temperature below that of the heat generator must be sized for a flow smaller than the design flow; the remaining flow is supplied by the by-pass installed downstream of the valve sited between the return and the flow to the DHW/heating/auxiliary circuits.

Reasons :

– a valve sized for the total flow requested by the plant would operate for most of the time near to closure, thereby compromising the accuracy and safety of the control.
– a minimum volume of water from the boiler at 80°C is sufficient to satisfy the variable temperatures in relation to the outside temperature (from 25…45/50°C) requested by the panels plant.
– protection of the plant, even if the valve is “blocked” when fully open; however, the **thermostat with maximum limit** of temperature is always necessary.



Legend :
Q1 = total flow in panels in l/h
Q2 = flow of valve in l/h (= partial of total)
Q3 = flow of fixed by pass in l/h (= partial of total)
tc = operating temperature of boiler
tmp = panels flow temperature
trp = panels return temperature (= tmp – Δt)
d1 = setting valve
V1 = 3-port mixing valve
M1/M2 = pumps
Tan = anticondensing detector

Fig. 17 Plant with panels and fixed by-pass

6.8 Sizing when design data available

The diameter of the valve can be found in two ways according to the data available:

1) if we know :
   – *flow* Q in l/h or m³/of plant with panels
   – design *temperature difference* Δt between flow and return panels
   – *maximum* operating *temperature* :
     of panels
     of boiler (or boilers)
2) if we know :
   – *power* P or *energy* E
   – design *temperature difference* Δt between flow and return panels
   – *maximum* operating *temperature* of boiler

6.9 Solution of case 1) using directly diagram 1 for slipper valves

Data available :
– flow of panels Q1 = 20,000 l/h
– boiler operating temperature = 80 °C
– design maximum temperature panels = 50 °C
– design temperature difference panels = 5 °C

Equation to use to determine the flows (Q2) of the control valve and of by-pass (Q3)

a) $$Q2 = \frac{Q1}{\dfrac{tc - trp}{tmp - trp}}$$ b) $$Q3 = Q1 - Q2$$

Procedure

Substituting in the formula the corresponding values in our example, we have : $Q2 = \dfrac{20{,}000}{\dfrac{80 - 45}{50 - 45}}$ = approx. 2,860 l/h,

We round off this figure to 3,000 l/h (to take into account the start /stop of the burner on the basis of the differential of the boiler control thermostat)..

The diameter of the 3-port motorised valve is obtained by proceeding as described in section **6.3** :
– **3G 1"1/4"** with **CVC** actuator, Kvs 19 and pressure drop of 2.4 kPa (0.24 mWG)

Knowing Q2 we obtain the by-pass flow using equation b) : Q3 = Q1 – Q2
Substituting, we have : Q3 = 20,000 – 3,000 = 17,000 l/h

6.10 Check on pressure drop using equation : $\Delta pv = \dfrac{Q^2}{Kvs^2} \times 10$

Substituting, we have $\Delta pv = \dfrac{3^2}{19^2} \times 10 = 0.25$ mWG (2.5 kPa)

which is a value practically equal to that found using the chart.

6.11 Solution of case 2) when we know :

– heat requirement of the panels = 212 kW (182,000 Kcal)
– boiler operating temperature = 80 °C
– panels design temperature difference = 5 °C (flow 50 °C / return 45 °C)

Equations to use for determining the flows of panels (Q1) and of the valve (Q2):

a) $Q1 = \dfrac{P}{\Delta t}$ b) $Q2 = \dfrac{P}{tc - trp}$

Legend:
P = power or energy
tc = boiler temperature in °C
trp = panels return temperature in°C
Q1 = panels flow
Q2 = valve flow

Procedure

– with P expressed in kW or kWh we have : $Q2 = \dfrac{212}{80-45} \times 860$ = approx 5,200 l/h & $Q1 = \dfrac{212}{5} \times 860$ = approx 36,400 l/h

– with P expressed in Kcal or Kcal/h we have : $Q2 = \dfrac{182{,}000}{80-45} = 5{,}200$ l/h & $Q1 = \dfrac{182{,}000}{5} = 36{,}400$ l/h

The flow (Q2) of the valve, a value we round off to 5,500 l/h (for the reasons given in section **6.9**).
The diameter of the 3-port motorised valve is obtained by the procedure described in section **6.3** :

– **3G 1"1/2"** with **CVC** actuator and with Kvs 29 and pressure drop of 3.5 kPa (0.35 mWG)

Knowing Q2 we obtain the by-pass flow using equation b): Q3 = Q1 – Q2
Substituting, we have : Q3 = 20,000 – 3,000 = 17,000 l/h

6.12 Check on pressure drop using equation : $\Delta pv = \dfrac{Q^2}{Kvs^2} \times 10$

Substituting numbers in the formula : $\Delta pv = \dfrac{5.5^2}{29^2} \times 10 = 0.36$ mWG (3.6 kPa)

a value practically equal to that obtained using the chart.

Note : We can check the result using the technical balance method with the equations :

a) Et = Q2 x tc + Q3 x trp b) $tmp = \dfrac{Et}{Q1}$

Considering the values of case1) (see section **6.9**) we have :

Et = 3,000 x 80 + 17,000 x 45 = 1,005,000 and $tmp = \dfrac{1{,}005{,}000}{20{,}000}$ = approx 50 °C

Considering the values of case 2) (see section **6.11**) we have :

Et = 5,500 x 80 + 30,900 x 45 = 1,830,500 and $tmp = \dfrac{1{,}830{,}500}{36{,}400}$ = approx 50 °C

Comments :

– the by-pass pipework usually has the same cross-section as those of the flow and return of the panels circuit and with the setting valve (d1), mounted on the by-pass pipwork, the flow (Q3) of the by-pass is calibrated.
– the 3-port valve must be mounted **as mixing valve** (2 inputs, 1 output) so that the panels plant flow is constant with any position of the control valve.
– not recommended – better, **not permitted** – mounting the 3-port valve as diverting (1 input, 2 outputs) since the panels plant flow varies from Q1 with control valve open to Q3 with control valve closed
– for the same reason given above, the use of a 2-port valve is **not recommended**
– the return-to-boiler temperature is always below 50°C and so lower than that permitted in traditional boilers to avoid condensation.

– with control valve partially or totally closed (throughport) there is no water circulation in the boiler.

By reducing or cutting off the water circulation, the time constant of the control and safety boiler thermostats increase in respect of the design time constant.

This phenomenon delays taking the measurement of the actual temperature and this occurs precisely in coincidence with the minimum or nil request for heat from the panels installation. However, the energy supplied by the burner is always at the maximum level (except if modulating or in 2 stages) and so the increase in boiler temperature is more rapid than measurement by the thermostats (time constant).

Under these conditions, at best, the burner is switched off when the actual temperature of the boiler is higher than that desired (set on the thermostats); in the worst case the safety thermostat intervenes and blocks *manual resetting*.

– for these last two reasons the presence of the pump (M2) for recycling to boiler and with anticondensing function, is necessary; it should be mounted upstream of the control valve and of the **detector** (**tr**) for minimum return-to-boiler temperature.

The pump (M2), by keeping the water circulating in the boiler even with valve closed, permits the normal intervention by the thermostats for restricting the speed of increase in temperature.

The concept of the function of the recycle pump and of the detector is described in the relevant section

Note : The setting valves must be mounted on the sections of the hydraulic circuit with constant flow.
The detector for measuring minimum return-to-boiler temperature **cannot** be used in the absence of the recycle - to - boiler pump (M2)

6.13 Commissioning (calibrating hydraulic circuit)

The calibrating valve (d1) serves to suit the flow of the by-pass to the value found in sizing the plant.

Procedure

– calibrating valve “d1” open

– 3-port control valve closed

– boiler (or boilers) in operation at design set temperature (80 °C)

– pumps for panels installation (M1) and boiler (M2) in operation

Wait until the boiler (or boilers) section of the circuit becomes stabilised at the design temperature (about 80 °C) and then :

a) open the 3-port control valve rapidly by hand

b) close slowly the calibration valve “d1” until you obtain the desired flow temperature in the panels

c) wait until the panels flow temperature becomes stabilised at the desired value

d) repeat the operations at points a) and b) and, if necessary, adjust the position of the calibration valve “d1”

e) block and seal the calibration valve in order to prevent tampering with the chosen position.

6.14 Use of 4-port valves

In **theory** in the circuits “b”, “c” and “g” of figure 16 you can also use 4-port slipper valves.

Why in theory? What are the reasons for preferring 4-port?

The purpose of using 4-port valves is to obtain a natural circulation of water in the boiler with valve closed, that is, in the absence of a call for heat by the DHW/heating/auxiliary circuits.

When does natural circulation take place?

Natural circulation, as we know, takes place when the following conditions exist :

– *negligible* pressure drops in the hydraulic circuit (corresponding to that of boiler)

– *appreciable* temperature difference between flow and return to boiler

– *large* difference in height between flow and return to boiler.

Do these conditions exist in plants today?

The semi- or completely-pressurised boilers, which for some time have replaced the marine ones in plants, have the following features:

– an appreciable pressure drop to which corresponds a reduced water content

– a negligible (in fact, even with previous boilers) difference in temperature between flow and return since with 4-port valve closed the temperature difference of the natural circulation in boiler is only that due to losses from the flow/return pipe (insulated) of the valve with the boiler.

– the difference in height between flow and return to boiler is practically impossible to achieve in practice since the two attachments are now sited in the upper part of the boiler..

Conclusion :

– none of the conditions necessary for guaranteeing the natural circulation exist

– the use of 4-port valves is **no longer** advisable even if they are still used on small boilers designed for natural circulation.

Notes : – The addition of the recycle pump (P2) does not improve the situation; indeed, a closed 4-port valve can be the cause of further difficulties such as the inversion of circulation in the boiler.

– it is not possible to use the minimum temperature detector for return to boiler on account of the slow or non-existent natural circulation of water in the boiler.

– the control of the panels installation in relation to ambient temperature is not advisable in view of the high inertia of the plant and of the limited movement of the ambient air (limited convective movement) which makes fruitless measurement by the ambient detector.

6.15 4-port valves on the same manifold (fig. 18)

This type of mounting is **NOT recommended** because at point **A** an irregular water circulation occurs when the valve of circuit (1) is closed and that of circuit (2) is open.
At point **A** a part of the return water of circuit (2), especially when the pressure drop of the branch A - B is less than that of branch A - C - B, flows down the vertical pipe and into the open port of the valve of circuit (1) = reverse circulation.

As a result, at point **B** there is a mixture of hot water coming from the boiler and return “cold” water from circuit (2).

This mixture of water at a temperature *below that desired* is the mixture which supplies the maximum call for heat (valve open) of circuit (2), and so is not able to meet the request.

Note: – the parasitic circulation occurs in the 4-port valve of the circuit mounted nearest to the boiler (1 in the diagram)

– mounting a check valve on the **A** length of pipe is effective in preventing parasitic circulation by means of the circuit valve (1), but cancels or prevents the desired gravity circulation in the opposite direction (from **B** to **A**).

– by connecting point A, as indicated in the diagram, you offer a greater resistance to the parasitic circulation and this is often sufficient to eliminate it,

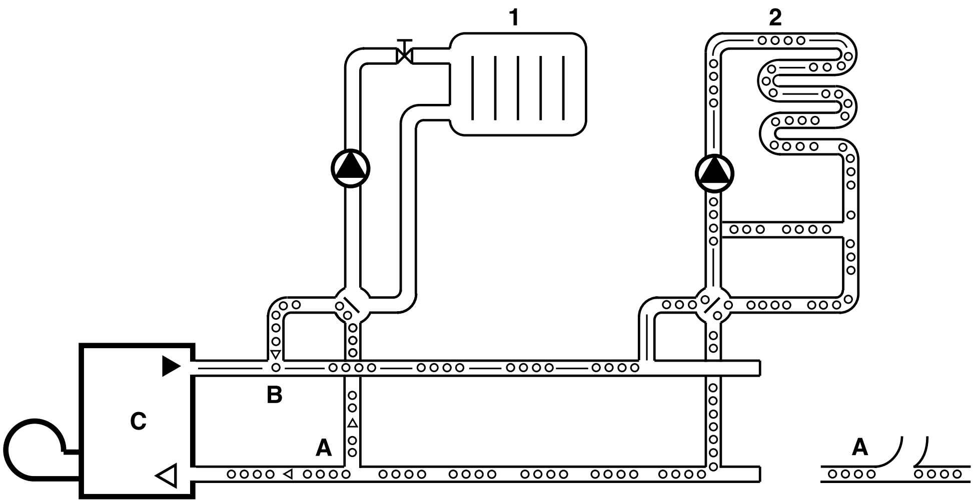


Fig. 18 4-port valves mounted on the same manifold
——— = hot water from boiler
○○○○ = return water from circuit 2

6.16 Control with variable flow in primary of heat exchangers or batteries

2-or 3-port seat valves (see circuit “a” and “e” in figure 16 and figure 19)

General :

– the valve must have an **equal percentage performance** (curve) and must be sized for the total flow required by the DHW/heating/auxiliary circuits.

– the primary flow is variable with either 2-port or 3-port valve (mounted for mixing on return as indicated in circuit “e” , or as diverting)

– the flow in the secondary circuit **must be constant.**

– the maximum differential pressure for the actuator (Δp max, see section **3.7**) to be considered is that for steam (manometer reading)

– the 2-port valve in circuit “a” regulates the flow both in the heat exchanger and in the production circuit (e.g. boiler)

– since the 2-port control valve influences the whole circuit, it should, in theory, have a very high pressure drop when open (sizing).
In practice this is not possible for several reasons, e.g.: economic (cost of pump), noise, kettling. Accordingly, as we shall see, it assumes the role of the heat exchanger and of the batteries.

– The control of flow (3-port valves for diverting and/or mixing on return) in large pre-heating batteries with cold outside air is quite frequent the intervention of frost protection even with control valve not completely closed.

Reason : with medium or small loads, on account of the non-linear output of the battery (power output- see section. **4.1**), the valve greaty reduces the flow with the result that the air downstream of the battery becomes stratified.

The variable to be controlled (temperature) must always be always be measured :

– on the secondary of the heat exchanger, flow to the DHW/heating/auxiliary circuits

– on the air downstream of the battery

6.16. 1 Heat exchangers with saturated steam primary and water secondary

2 - port seat valves, figure 19

6.16. 2 Sizing with design data

The diameter of the valve can be obtained in two ways

1) if we know :
   – the *flow* Q in kg/h of steam
   – the vapour[1)] *pressure*
2) if we know :
   – the *power* P or *energy* E
   – the vapour[1)] *pressure*

1) To size the valve we require the **absolute** pressure. Normally, we know the **relative** pressure, that is, the working pressure indicated by the manometer. Absolute pressure = relative pressure + 100 kPa.

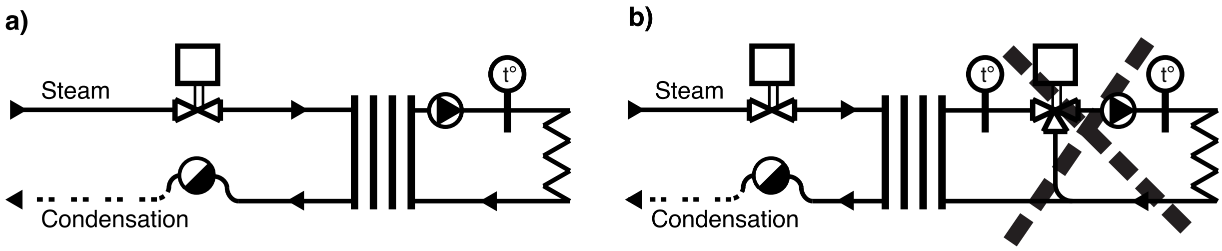


Fig. 19 Steam/water heat exchangers a) secondary with constant flow
b) secondary with variable flow = not permitted

Note : The pressure drop to take into account when sizing a valve has to be 20%…30% of the absolute incoming pressure; in any event, **always below** even if near to the **critical** pressure (see diagram 2)

6.16. 3 Solution of case 1) by means of chart 2 for steam

Data available :
– flow of saturated steam Q = 250 kg/h
– operating pressure (relative) Pr = 500 kPa (5 bar)

Procedure :

a) on the scale of absolute steam pressure **Pv**, from the value 600 kPa (absolute pressure = 500 kPa + 100 kPa), draw a vertical straight line to meet the sloping straight line showing the pressure drop **Δp** of 150 kPa (30% of absolute pressure)

b) on the flow scale **Gv** of the steam at 250 kg/h draw a vertical straight line

c) from the point of the pressure drop, found in a), draw a horizontal straight line to meet the vertical straight line from the scale **Gv** (shown as b))

d) the meeting point identifies the Kvs value of the valve, and with the values in the example corresponds to:
– KVS = 4 to which corresponds a valve DN15 mm

Having established the diameter of the valve we must check that the motorised 2-port valve is suitable :
– for steam at 500 kPa
– for the steam temperature (500 kPa = 150 °C approx)
– for the difference of maximum operating pressure Δp max of the actuator (500 kPa)

In the case in question this is **RV 2A 15** with actuator **CLC 01…** or **CLC…** according to the run time and the power supply voltage.

Note : The point found in **d)** may fall between two Kvs values, e.g. if, in the case under examination, the flow were 300 kg/h, the corresponding Kvs would fall between the two Kvs values 4 and 6.3; accordingly, it is necessary to check the corresponding pressure drops (drawing the horizontal straight lines until they meets the sloping straight lines showing the pressure drops **Δp**)

The new values are :
– for Kvs 4 = 200 kPa
– for Kvs 6.3 = approx 85 kPa

The choice would be for valve DN 15 with Kvs 4

Warning :
– clearly, in this situation, the final decision must be agreed with the client; that is, the recommendation in section **6** always holds good.
– oversized valves for steam operate for long periods near to closure: this is a critical situation for control owing to the drastic reduction in vapour pressure (caused by the valve) which makes it difficult to get rid of the condensation.
– the control of the steam is influenced by the efficiency of the discharge of the condensate; faulty dischargers, sloping and/or undersized return network for the condensate are often the causes of instability in the desired value (swings in the action of the valve with long “waiting” times when closing.

6.16. 4 **Check on pressure drop and Kvs using mathematical equation :** $\Delta pv = \dfrac{Q^2}{(22.4 \times Kvs)^2 \times P2}$ (see section **5.4**)

Substituting the values in the equation we have : $\Delta pv = \dfrac{250^2}{(22.4 \times 4)^2 \times 4.5} = 1.73$ bar (173 kPa)

Check on Kvs valve :

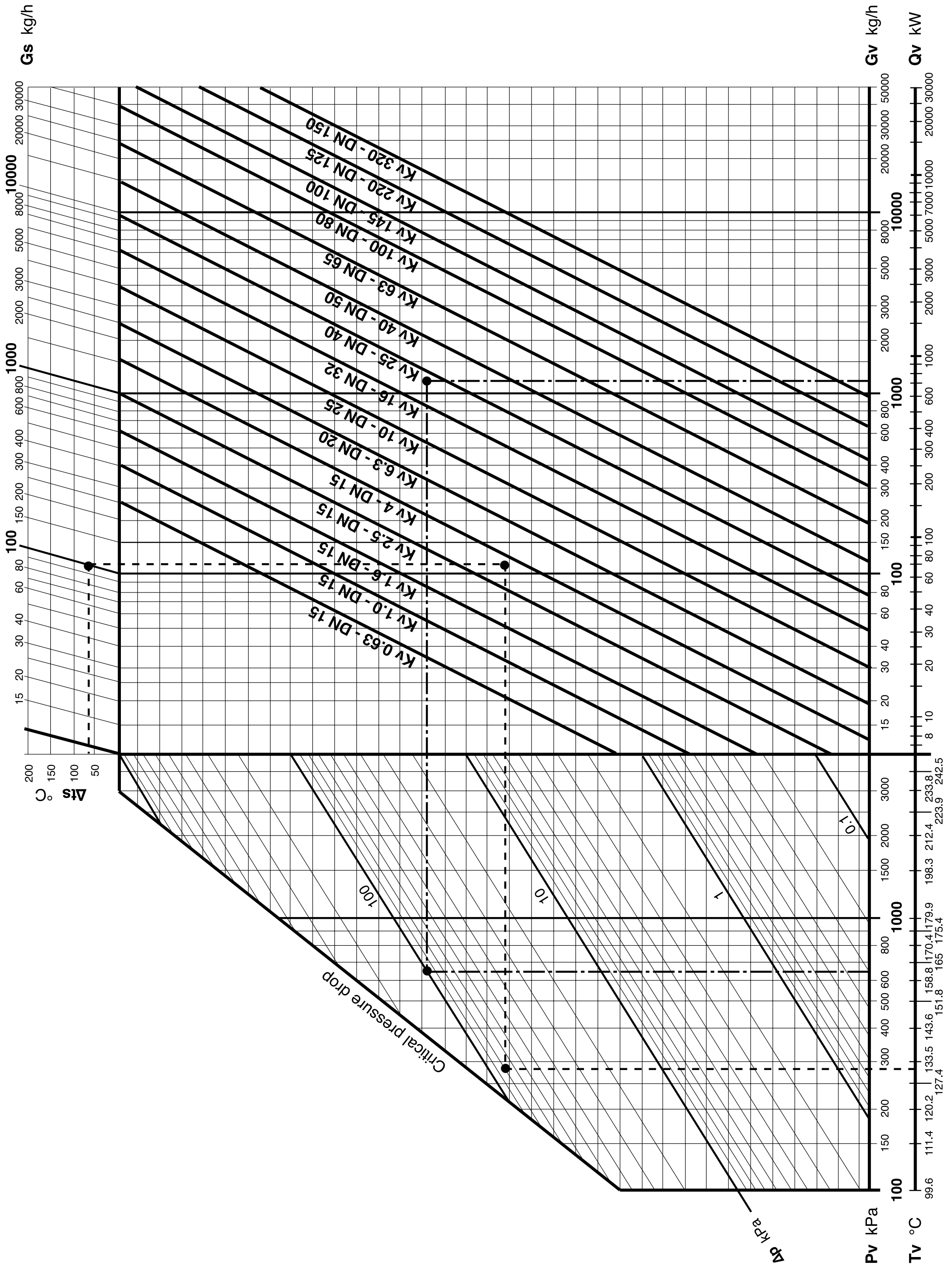
$$Kvs = \frac{Q}{22.4 \times \sqrt{\Delta pv \times P2}} = \frac{250}{22.4 \times \sqrt{1.73 \times 4.5 \text{ approx}}} = 4 \text{ Kvs}$$

6.16. 5 **Solution of case 2) when we know:** – power or energy = 500 kW o (kWh)
– vapour pressure = 500 kPa

Knowing the power (or energy) we can calculate the flow by means of :

Q = P x 1.6 ; 500 x 1.6 = approx 800 kg/h of steam (see section **5.4**)

Having determined the flow we can obtain the diameter by the procedure described in section **6.15.**3 = Kvs between 10 and 16. We shall choose the valve with Kvs 16 = **RV 2A 32** with **CLC 06…** or **CLC 12…** actuator (the actuator with Kvs 10 gives rise to a **critical** pressure drop).



Sizing chart 2 : valves for saturated or superheated steam

6.16.6 Heat exchangers with superheated steam primary and water secondary
2 port seat valves, figure 19

6.16.7 Sizing with design data
The diameter of the valve can be determined in two ways:

1) if we know:
   – *flow* Q in kg/h of steam
   – *pressure* of the steam
   – superheating[1] *temperature*

2) if we know:
   – *power* P or *energy* E
   – *pressure* of steam
   – superheating[1] *temperature*

Note : The only difference, in respect of section **6.16.2** regarding the saturated steam, lies in the need to know the temperature of the superheated steam..

1) The temperature of the superheating of the saturated steam can be indicated:
– directly (50 °C, 100 °C, etc.)
or
– indirectly, for example: steam at 500 kPa at the temperature of 250°C
Certainly we are dealing with superheated steam because saturated steam at 500 kPa is at the temperature of approx 150°C (see diagram 2), and so the superheated steam is derived from: 250°C – 150°C = 100°C.

The difference in respect of the examples in sections **6.16.3** and **6.16.5** is that in this section we are considering in sizing chart 2 only the section concerning superheating.

6.16.8 Solution of case 1) using sizing chart 2 for steam
Data available :
   – flow of superheated steam Q = 100 kg/h
   – relative pressure 180 kPa (absolute pressure = 280 kPa)
   – superheating temperature 59 °C
   – desired pressure drop 80 kPa

Procedure

a) on the absolute pressure scale **Pv**, from the value of 280 kPa draw a vertical straight line to meet the sloping straight line for **Δp** of **80 kPa** pressure drop.
b) on the superheating scale **Δts** draw a horizontal straight line from the value of 59°C to meet the sloping straight line **Gs** at the value of 100 kg/h
c) from the point found in b) draw down a vertical straight line
d) from the point found in a) draw a horizontal straight line; the point at which it meets the vertical straight line found in c) gives the Kvs of the valve.

The point is situated between Kvs 2.5 and 4; since it is nearer to Kvs 4 so we select DN 15.

Having found the diameter of the valve we have to check that the 2-port motorised valve is suitable:
   – for steam at 500 kPa
   – for the temperature of 190 °C
   – for the maximum pressure difference Δp max of the actuator.

In the case in point the suitable valve is:
   – **RV 2A 15** with **CLC…** actuator

Note : if the superheating of the steam were 150°C, the valve diameter would still be DN 15, but on account of the steam temperature (120°C + 150 = 270°C) the required valve is:
   – **RV 3K 15** with **CLC ...**, actuator including the blind flange for closing the by pass.

6.16.9 Solution of case 2) where we know
   – power or energy = 1,560 kWh
   – relative pressure of the steam = 300 kPa
   – superheating temperature = 150 °C

The steam flow we obtain from the equation Q = P x 1.6 Substituting, we obtain :

1,560 x 1.6 = 2,500 kg/h of steam

Proceeding as described in section 6.16.7 we find Kvs 63 with a pressure drop of **100 kPa**

The valve has to be suitable for the temperature of 290°C with actuator for Δp max of 300 kPa such as the :

RV 3K 65 with **CLD 06 ...** or **CLD 12 ...** including the blind flange for closing the by pass.

6.17 Mounting two valves in parallel with control in sequence

Distribute the flow using *two 2-port seat valves* (figure 20):

a) *advisable*, when in the presence of high vapour flows (section **6.16.8**) with appreciable flow variations in the secondary (the circuits not operating simultaneously).

b) vapour pressure higher than that permitted by actuator (Δp max.)

In both cases the controller modulates the control:

to reduce the desired value, *first* the valve with smaller diameter, then the other

vice versa

to increase the desired value, *first* the valve with greater diameter, then the other

Reasons

a) to obtain *precise control* over the whole range of functions:
the valve with the smaller diameter takes control of the minimum requests from the DHW/heating/etc circuits

vice versa

the valve with the larger diameter takes control of the medium and maximum requests from the DHW/heating/etc circuits (e.g. start-up after a plant shutdown)

b) valve (1) open (valve able to function at Δp max of vapour pressure) at point **B** is established a vapour pressure reduced by the pressure drop of the valve (1).

The pressure at point **B** has to be such that the difference between points **A** and **B** is less than that permitted (Δp max) for the operation of the valve (2)

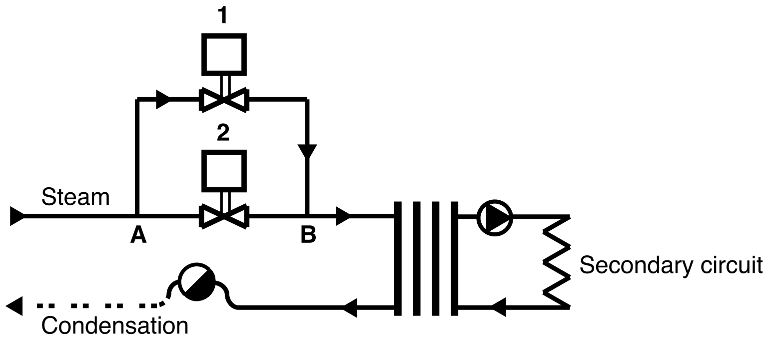


Fig. 20 Parallel mounting of two 2-port seat valves
1 valve of smaller diameter
2 valves of larger diameter

6.18 Water/water (secondary with constant flow) heat exchange batteries, water/air with hot or cold water

3-port seat valves (fig. 21)

General :

The 3-port valve can be used for mixing if mounted on the return or for diverting if mounted on flow (valves 1 and 2 respectively in figure 21).

In both cases the battery and/or the heat exchanger primary have variable flow and constant temperature.

The procedure to follow when sizing the control valve is the same as that described in sections **6.2**...**6.6**, considering that :

– we use diagram 3

– it is important to take into consideration the pressure drop of the section of the circuit controlled by the equal percentage port of the valve (see sections **5** and **5.1**) that is the primary of the heat exchanger or the battery

– when the pressure drops are not known the following values should be employed (always communicating them to the client):

exchangers with tube nest	= min. 1 m ... 2 m WG
exchangers with plates	= min. 2 m ... 3 m WG
batteries with 1 or 2 ranges	= min. 1 m ... 1.5 m WG
batteries with several ranges (cold)	= min. 2 m ... 3 m WG

Note : in all cases **never** with values below 1 m WG

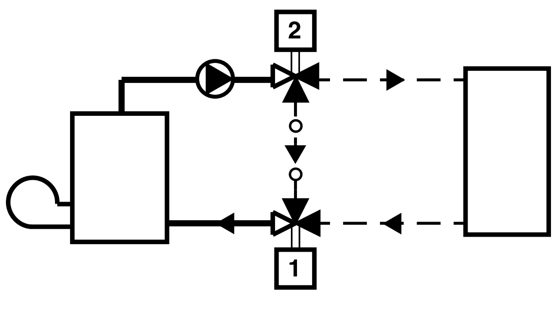


Fig. 21 Mounting 3-port valve for mixing (1) or diverting (2)

6.18. 1 **Sizing with design data**

The diameter of the valve can be found in two ways:

1) if we know :
   – the *flow* Q
   – if necessary, the pressure drop of the section of the circuit
2) if we know:
   – the *power* P or the *energy* E
   – the *difference of flow/return temperature* of the primary of heat exchanger or battery
   – *if necessary the pressure drop* of the section of the circuit

With the data available and using **sizing chart 3** we determine the Kvs of the valve as described in sections **6.3** ... **6.6**.

6.18. 2 **Heat exchangers superheated water/water (secondary with constant flow)**

– the valve has to be suitable for the temperature and PN (see section **3.3**) of the superheated water

– correct the flow only when it is *obtained from the energy or power and the temperature of the superheated water* and is ≥ 120°C, by a **K** factor to consider the lower density of the water and therefore the volume (flow), with equal power, is greater..

The **K** factors for the various temperatures are :

t	120 °C	140 °C	160 °C	180 °C	200 °C	220 °C
K	1.04	1.065	1.085	1.10	1.13	1.16

– mounting two valves in parallel, limited to high flows, is acceptable also in this application,

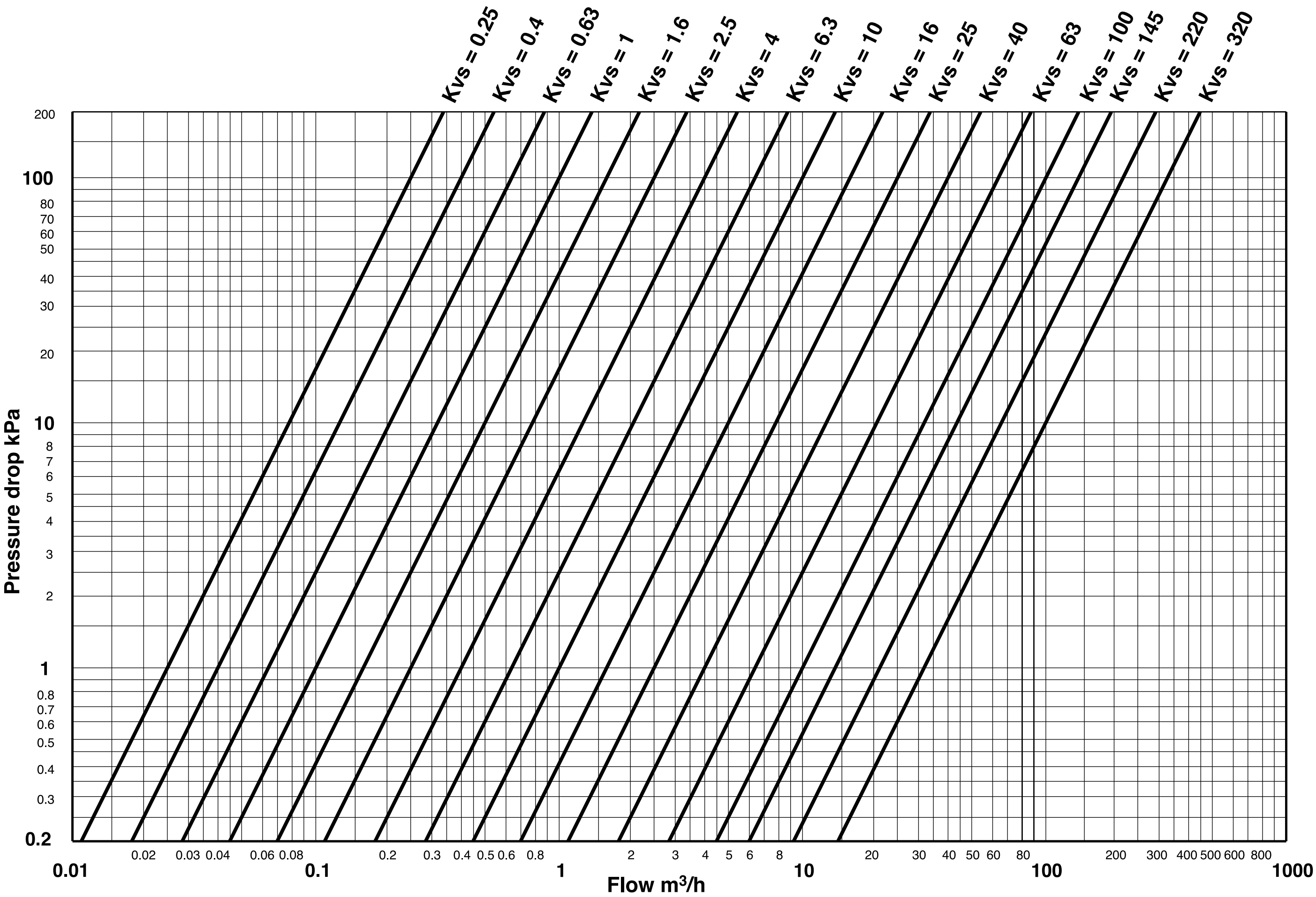
vice versa

– there are no problems for Δp max since the pressure drops for sizing the valves are those regarding the section with variable flow (see section **6.17**).

The static pressure to maintain the desired temperature from overheating influences the whole plant and accordingly downstream and upstream of the valve

The static pressure values necessary to maintain the temperature of the water superheated are:

t	120 °C	130 °C	140 °C	160 °C	170 °C	200 °C
P	100 kPa	200 kPa	300 kPa	500 kPa	710 kPa	1,500 kPa



Sizing chart 3 General diagram for water in relation to 2 -or 3-port valves

6.19 Injection plants with 3-port valves

3-port seat valves [1], see Figure 16, circuits "d" and "f", and Figure 22

1) Slipper valves, excluding 4-port, can also be used (see section **6.15)** but their use demands greater care in constructing the plant.

General :

In the plant, under all load conditions, both with the valve mounted on the return for mixing or on the flow for diverting, the flow is constant both in the primary section (boiler) and in that of the DHW/heating/auxiliary circuits (secondary).

With no load - no demand from consumer circuits - the throughport (2) of the valve is closed, the pump M keeps the flow in the primary section constant and the pump M2 does the same for the DHW/heating/auxiliary circuits section (secondary).

In the sections of the circuit **b - c** and **f - 2** there is no water circulation.

With maximum load, the throughport (2) of the valve is open (by-pass 3 closed), the total flow circulates in the DHW/heating/auxiliary circuits via the section **b - c** and the throughport of the valve **2 - 1**.

In the sections **b - 3** and **f - c** there is no water circulation.

With partial loads, at point "c" there is a mixture of the water coming from the primary, section **b - c** and that from the return from the DHW/heating/auxiliary circuits, via section **f - c**.

At point "g" there is a mixture of water coming from the by-pass **b - 3** with that returning from the DHW/heating/auxiliary circuits, from **f - 2** (the water returning from the DHW/heating/auxiliary circuits corresponds to that introduced at point "c" by the primary pump M1).

In other words, the primary flow (pump M1) divides at point "b", according to the DHW/heating/etc circuits requirements, with one part destined for the DHW/heating/auxiliary circuits and the other returning in the primary.

Similarly, the return flow from the DHW/heating/auxiliary circuits (pump M2) divides at point "f" into the flow destined to point "c" and that returning to primary point "g".

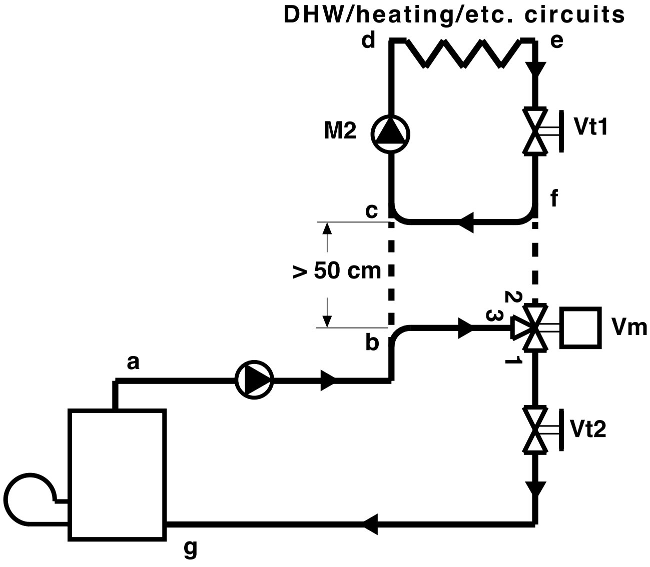
The pump M1 has to have the head to overcome the resistance of the primary circuit and of the 3-port control valve; while the pump M2 takes charge of the pressure drops of the DHW/heating/auxiliary circuits. Only by meeting these conditions can the two sections of the total plant have in constant pressure ratio.

To prevent parasitic circulation (see Figure 22 and section **7**) the length of pipe **b - c** has to be at least **10 times** the diameter of the pipework, and in any case not less than **50 cm**.

Injection plants, in both the primary and secondary branches, must be fitted with a calibration valve (Vt1 and Vt2).

The control valve should be sited as close as possible to the DHW/heating/auxiliary circuits (reduction of Dead Time).

The plant should be constructed **vertically**, as in figure 22, so that it is necessary to have sufficient space available.



Legend :

Vm = 3-port motorized valve
Vt1 = calibration valve for DHW/heating/etc circuits
Vt2 = calibration valve for primary branch (boiler)
M1 = primary branch pump
M2 = DHW/heating/auxiliary circuit branch pump

Fig. 22 Injection circuit with 3-port mixing valve on return

6.20 Sizing with design data

The diameter of the valve can be determined in two ways:

1) if we know :
 – the *flow* Q in l/h or m³/h

2) if we know :
 –the *power* P in kW or the *energy* E in kWh
 – the *design temperature difference* Δt between flow and return of the DHW/heating/auxiliary circuit

Note : the valve is identified by proceeding for both cases as in sections **6.3** ... **6.6**.

The pressure drop to consider when sizing the valve (see Figure 16) concerns only that for the pipe lengths **b - c** and **f - 2** where the pressure drop is negligible, and so values from 0.4…0.6 m WG (4…6kPa) are sufficient.

6.21 Commissioning (calibrating the hydraulic circuit)

General :

– the calibration valve Vt1 is used to adjust the DHW/heating/etc circuits flow to the design value

– the calibration valve Vt2 is used to adjust the flow of the primary circuit (boiler) so that, with the maximum request from the DHW/heating/auxiliary circuits and with the control valve open, the flow temperature (point **c**) is the design one and accordingly in the length of pipes **f – c** water does not circulate.

Preparation for commissioning

– pumps M1 and M2 in operation
– control valve closed (port 2)
– calibration valves Vt1 and Vt2 open

Wait until the two branches of the plant, both flow and return, are at the same temperature, with the DHW/heating/etc circuits 20…30°C lower than the primary..

The temperatures must remain as stable as possible for the whole duration of the calibration, particularly if there is a temperature difference between the two branches.

Procedure

Open the control valve (port 2) by hand :

– if the return temperature of the primary (point **g**) increases, and is therefore higher than that of the return from the DHW/heating/auxiliary circuits (point **e**), this means that there is water in circulation coming from the primary (point **b**) in the length of pipe from **c** to **f** (the pipework **becomes hot**).

Adjust the calibration valve Vt2 slowly towards closure until the circulation stops

vice versa

– if the flow temperature to the DHW/heating/etc circuits (point **c**) is below that coming from the primary (point **b**), this means that there is in circulation return water from the DHW/heating/auxiliary circuits in the length of the pipe from **f** to **c** (the pipework becomes cooler than under the previous conditions).

Adjust the calibration valve Vt1 slowly towards closure until the temperature at point **c** is at the desired value.

Setting terminated

If it is not possible to make a complete calibration it is preferable for the length **f - c** to be warm (a minimum flow of water coming from the boiler is circulating) rather than the contrary, so that a desired flow temperature to the plant sufficient to meet the maximum request from the DHW/heating/auxiliary circuits is assured.

Note : When the plant includes a manifold from which branch several injection circuits, it is advisable, after the calibration of each circuit, to carry out a further check by repeating the calibration operations.

7. PARASITIC CIRCULATION

7.1 Parasitic circulation in a circuit with mixing valve

In the mixing circuit (figure 23) with 3-port control valve closed (port **2**), when the speed of the water flow is very high (approx 1 m/s), a flow of hot water from the boiler (parasitic) through the by-pass (port **3**) of the valve, can become established.

This circulation of water heats (or keeps hot) the heat emitters with the 3-port valve closed even if the closure is perfect.

Reason :

– on account of its speed, the water returning from the DHW/heating/auxiliary circuits at point "**a**" (link with port 3 of the valve) enters deeply in the return-to-boiler pipe causing a pressure drop.

Because of the above, a part of the hot water, stationary and calm, is drawn into port 3 of the valve and replaced by a similar volume of water returning from the DHW/heating/etc circuits.

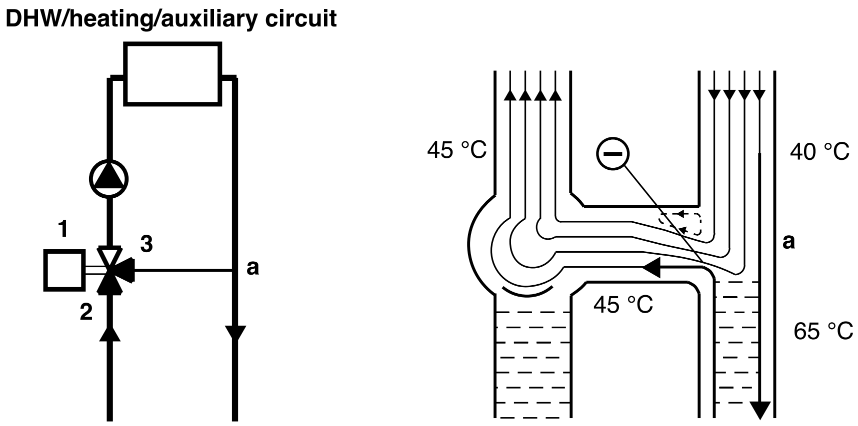


Fig. 23 Parasitic circulation in plants with 3-port mixing valve

7.2 Parasitic circulation in an injection circuit

In the injection circuit in Figure 24 we note that the parasitic circulation phenomenon, for the same reasons given in Section **7.1**, can occur in both the branches (primary and DHW/heating/etc circuits).

At point **b** (flow to DHW/heating/auxiliary circuits) the pipework will be warmer than at point **c** just as point **a** is warmer than point 3 (by-pass of the 3-port control valve).

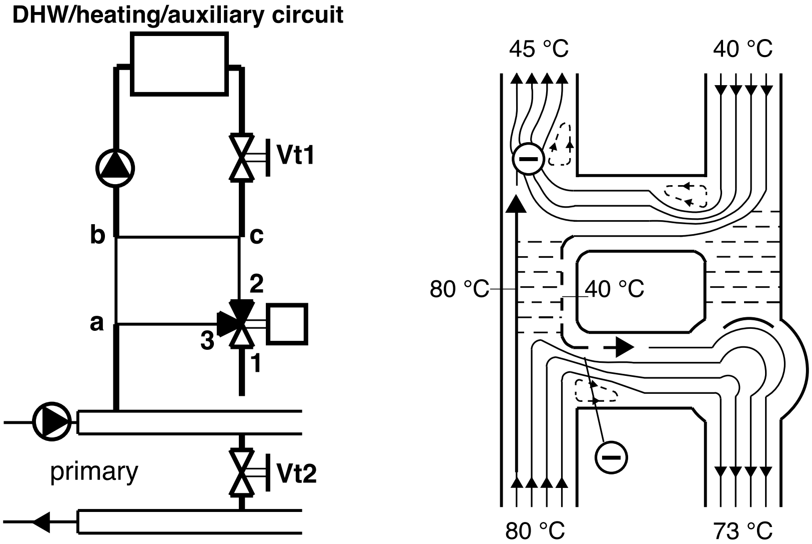


Fig. 24 Parasitic circulation in an injection plant

7.3 Measures for eliminating parasitic circulation

To avoid parasitic circulation the following measures should be considered:

– at design stage :
 assuming water speed not above 0.5 m/s

– at installation stage :
 introducing a “resistance” (pressure drop) consisting of the distance *in height* (D) of the connection pipework – see Figure 25

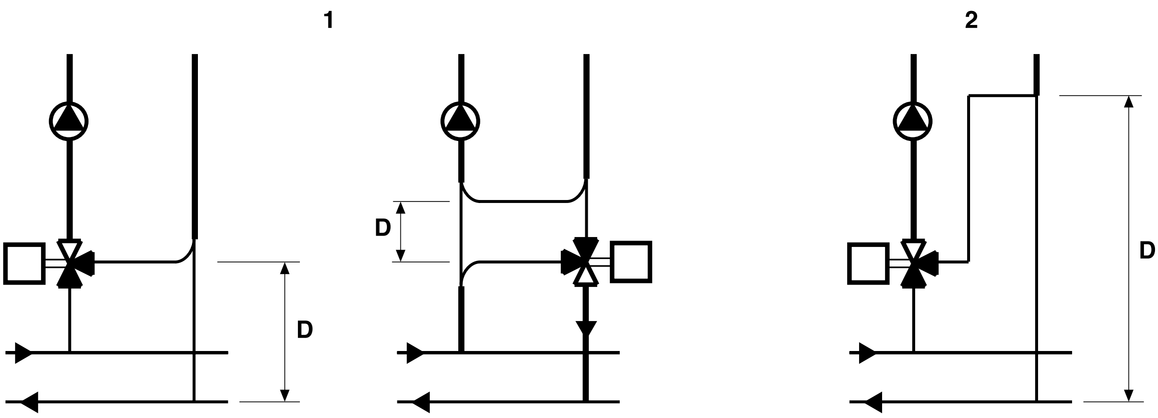


Fig. 25 Measures to take at installation stage
1) D = 10 x diameter of the pipework (minimum 50 cm)
2) construction, when insufficient space for the height D

8. NOTES ON MANIFOLDS

General

When several circuits originate in a single production centre (boilers and/or refrigerators) they are connected in parallel to a flow or return manifold.

8.1 Manifolds without primary pump (Figure 26)

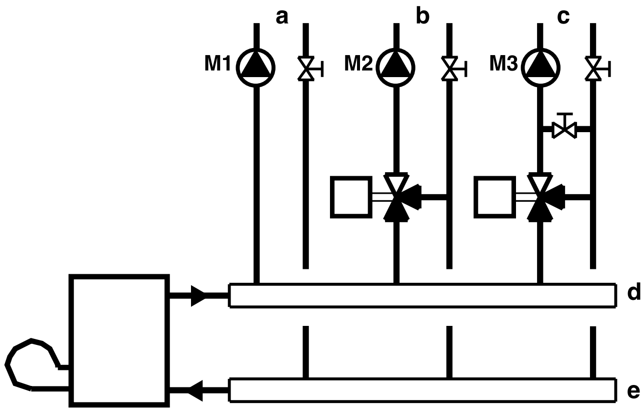


Fig. 26 Manifold without primary pump

a,b,c circuit zones
d flow manifold
e return manifold
a circuit for heat emitters (fan coils, etc)
b circuit for radiators and similar
c circuit for panels

Use

– in plants in which, with maximum demand from the DHW/heating/auxiliary circuits, the sum of the pressure drops of the boiler and of the flow and return manifolds, is less than 20% of the pump with the head below those of the other circuit zones.

– the return-to-boiler flow varies according to the request from the zones (a pump for recycling to boiler and control of the minimum return temperature, with anticondensing function, may be necessary) N:B: *not shown in the diagram*.

– the pumps (M1, M2 and M3) must have head for the pressure drops of their respective circuit (a, b and c) and of the boiler circuit.

–the manifolds must be near to the boiler

8.2 Effects on the control

The variable flow in the boiler changes the pressure ratios between the two manifolds thereby influencing the controls with negative effects on the individual valves.

For this reason, when the pump of circuit (**a**), with two-position control (go/stop), especially if the flow is considerable, control of the other zones can become unstable on account of erroneous circulations.

8.3 Manifolds under pressure with primary pump (Figure 27)

General

The distribution manifolds under pressure are necessary when they are a long way from the heat generators (boilers); or when there are no means of reducing the pressure drop, as indicated in section **8.1**.

The plant must be equipped with the primary pump M4

Use

– the manifolds can be connected to consumer circuits other than those for heating e.g. heating batteries in air conditioning plants, storage calorifiers for DHW, etc.

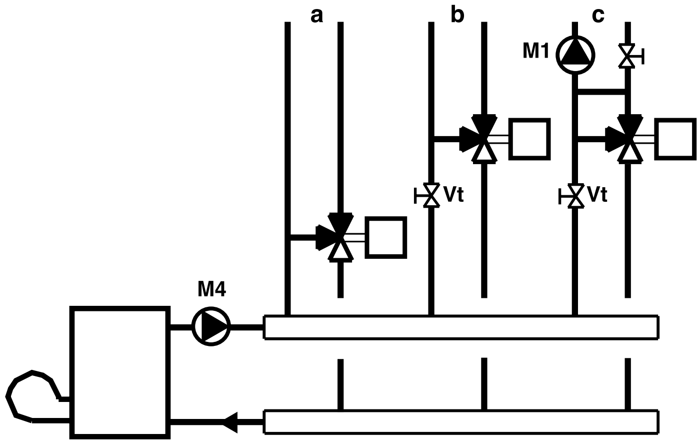


Fig. 27 Manifolds under pressure with primary pump
a) calorifier for DHW (only if the flow is negligible in respect of that of pump M4)
b) circuit for conditioning and fan coil batteries
c) circuit for panels and/or radiators (injection) circuit

Features of circuit

- the flow of the boiler circuit is constant, independently of the positions taken up by the control valves of the circuits connected to the manifolds (flow and return of the flows from the individual zones)
- the primary pump (M4) is sized for the total flow (sum of all the individual DHW/heating/auxiliary circuit offtakes) and must ensure the circulation of the water not only in the two manifolds but also in the secondary circuits (DHW/heating/auxiliary circuits) with mixing valves on the return.
  This means that the head of the primary pump *is defined by the circuit* with the highest pressure drop (in the diagram circuit (a).
- consequently, the head, at the joints of the flow manifold with the other secondary circuits is higher than necessary.
  To avoid a flow greater than that planned in these circuits it is necessary to insert on the flow of each circuit a valve for calibration or for control of the pressure (Vt).
- the injection circuits (c) must be provided with a pump for the secondary branch (M1)
- the return-to-boiler temperature, except at the start-up of the plants after a long shut-down period (e.g. night) is relatively high.

Not suitable

Secondary DHW/heating/etc circuits with control valves on mixing **cannot be used**.

8.4 Manifolds without pressure and with primary pump (Figure 28)

General

As with the collectors under pressure (section **8.3**), the primary pump (M4) ensures circulation in the primary.
The particular design of the circuit includes the by-pass connecting the two manifolds (Vt) in which, with reduced loads, the unused flow from the secondary circuits circulates.
The by-pass is sized for the total flow with a pressure drop of ≤ 0.5 kPa so that the pressure difference between the two manifolds is negligible; accordingly, the boiler circuit has a virtually constant flow and is independent of the load variations in the secondaries.
The secondary circuits must have their own pump sized for the flow of the DHW/heating/etc circuits and for the relative pressure drop including that of the control valve.

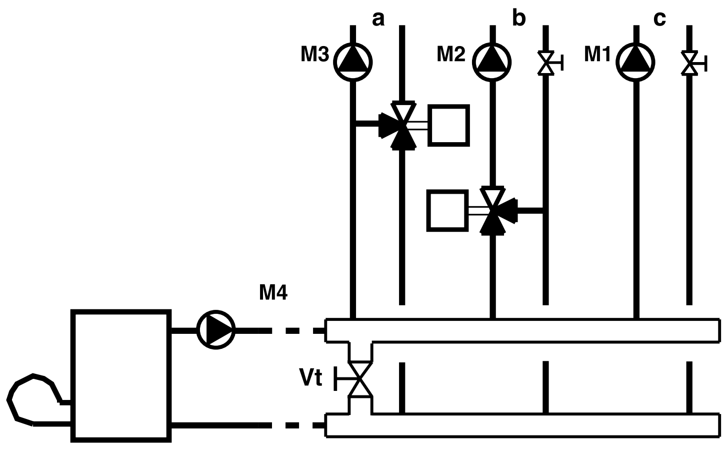


Fig. 28 Manifolds without pressure and with primary pump
a) batteries of air-handling plants and/or heat exchangers
b) heating circuit
c) flow towards other manifolds without pressure

Use

No use limit and so particularly suitable for large plants with several secondary circuits.
Circuits in operation in both winter and in summer (batteries for post-heating, DHW calorifier, swimming pools, etc) must be installed upstream of the by-pass in order to avoid circulation of hot water in all the flow manifold.

Features of the circuit

– each secondary circuit must have a pump
– the primary pump must have a flow 10% greater than that of the total design flow (sum of all the secondary circuits with control valves open)
– flow of boiler circuit practically constant
– no let-by with valves closed so slipper valves can be used for the heating (compensated) circuits
– no danger that the secondary circuits controls will influence each other.
– with the secondary circuits mixing it is not necessary to balance them with calibration valves (Vt)

9. OVERSIZING OF PLANTS

In oversized plants the water circulates through the plant components (radiators and the like, batteries, heat exchangers) too rapidly and does not have sufficient time to give up its heat.

Consequently, the temperature difference between flow and return is less, at maximum load, than that prescribed by design, not to mention what happens at reduced loads.

Heating plants with mixing valves and batteries in air-handling plants with a temperature difference of 5/6 °C with valve open, at maximum load present the control system with real problems.

Reason :
The control valve makes available the maximum thermal power before it is completely open, i.e. the valve operates over a limited range and consequently each minimum adjustment, especially when the demand is low, modifies the flow excessively and this causes instability in the desired value of the magnitude controlled.
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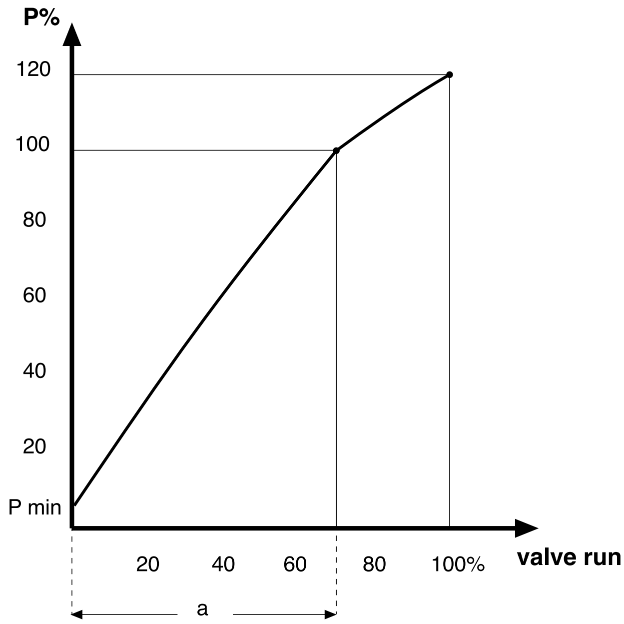

Fig. 29 Behaviour of an oversized plant
a = nominal power of the plant corresponds to 70% of the run
P = power supplied
P min. = minimum non-adjustable power

9.1 Considerations on oversizing

The curve of the plant can be represented on the basis of the relation between flow and head

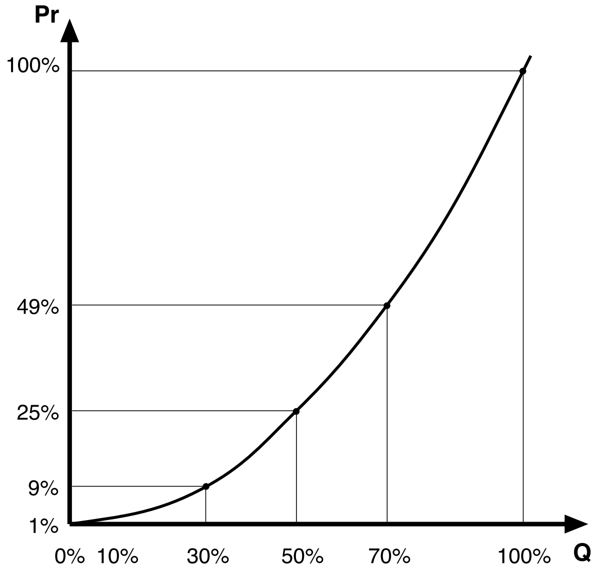

Fig. 30 Curve of a central heating plant

The flow of 70% ($\frac{7}{10}$) of the total is obtained with 49% of the pump head,

50% ($\frac{5}{10}$) with 25% of head

30% ($\frac{3}{10}$) with 9% of head

Q = flow
Pr = pump head

The plant curve can be more or less accented, *steep* for plants with high pressure drops and small flows

vice versa

flat for plants having large flows and small pressure drops .

In each instance it is always a 2° grade curve.

The point of operation at optimum output of the pump having design flow and head should be situated in the central zone of its operating curve

In the other positions the performance of the pump is less and so the running costs are higher.

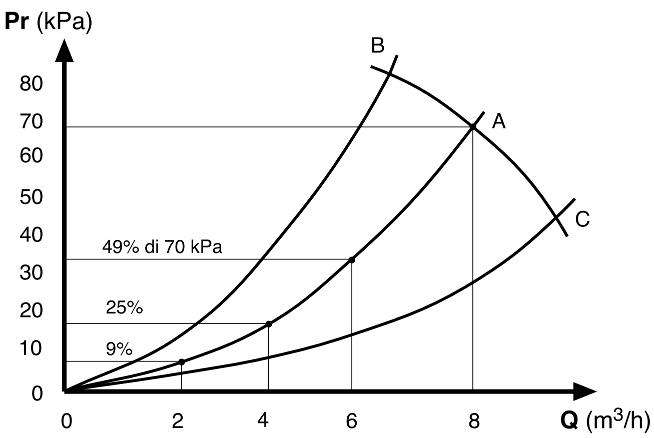

Fig. 31 Curve of a heating plant with :

Q = 8 m^3/h design flow
Pr = 70 kPa design head
A = optimal operating point of the pump
B & C = points of pump operation with lower output :
B = pressure drops larger than the design ones (infrequent case)
C = pressure drops smaller than the design ones

9.2 Determination of the flow of existing plants

Practical method :

In plants in operation for a long time it is difficult to obtain the flow and head data; we can only know for certain that there is a minimum temperature difference between flow and return and that this can be the cause of instability in our control.

Procedure :

- boiler and pump in operation with valve closed (by hand). Wait until the plant cools down so that the temperature is distinctly lower (20…30°C) than that of the boiler
- select a point on the (non-insulated) pipework, downstream of the mixing valve and preferably not very near to the valve
- have available a watch (better a stopwatch)
- place your hand on the non-insulated length of pipe, previously selected, and note the reading on the watch/stopwatch
- **have someone open** manually and rapidly the mixing valve
- note the time elapsing from the moment the valve is opened to that when you become aware of the sudden heating of the length of pipe on which you have placed your hand.

From the equations :

$$v = \frac{d}{t} \quad \text{and} \quad Q = q \times v$$

where :

d = distance, in meters, of the valve from the measurement point on the pipe
t = time in seconds (stopwatch/watch reading)
q = water contained in one meter of pipe
v = speed of water

We obtain the flow of the plant using *tables* showing the water contained in a meter of the pipe according to its diameter

For example, in normal threaded pipes the content is as follows :

D	1/2"	3/4"	1 1/4"	1 1/2"	2"	2 1/2"	3"	4"
l/m	0.20	0.37	1.02	1.38	2.21	3.70	5.14	8.72

D = diameter of pipe
l/m = water contained in one meter of pipe

Example :

Notes: diameter of pipe = 3" (to which corresponds a content of 5.14 l/m)
distance = 4 meters
time elapsed before feeling the hot “wave” with the hand = 7 seconds

$$V = \frac{4}{7} = 0.57 \text{ m/s} \quad \text{therefore} \quad Q = 0.57 \times 5.14 = \text{approx } 3 \text{ l/s} = \mathbf{10{,}800} \text{ l/h}$$

10. CONTROL OF MINIMUM RETURN-TO-BOILER TEMPERATURE

Prevention of corrosion in the boilers

General

As is known, in traditional boilers with gasoil burners, when the temperature of the water in the boiler is below about 50°C, combustion fumes condense on the walls of the boiler with the resultant phenomenon of corrosion and leakage of a liquid. . . which is certainly not colourless!

To obviate this drawback a so-called anticondensing pump (**M4** in figure 32) is mounted on the by-pass pipework between the output and the return to boiler.

The pump increases the temperature of the return-to-boiler water (coming from the plant) with a part of the hot water coming from the boiler.

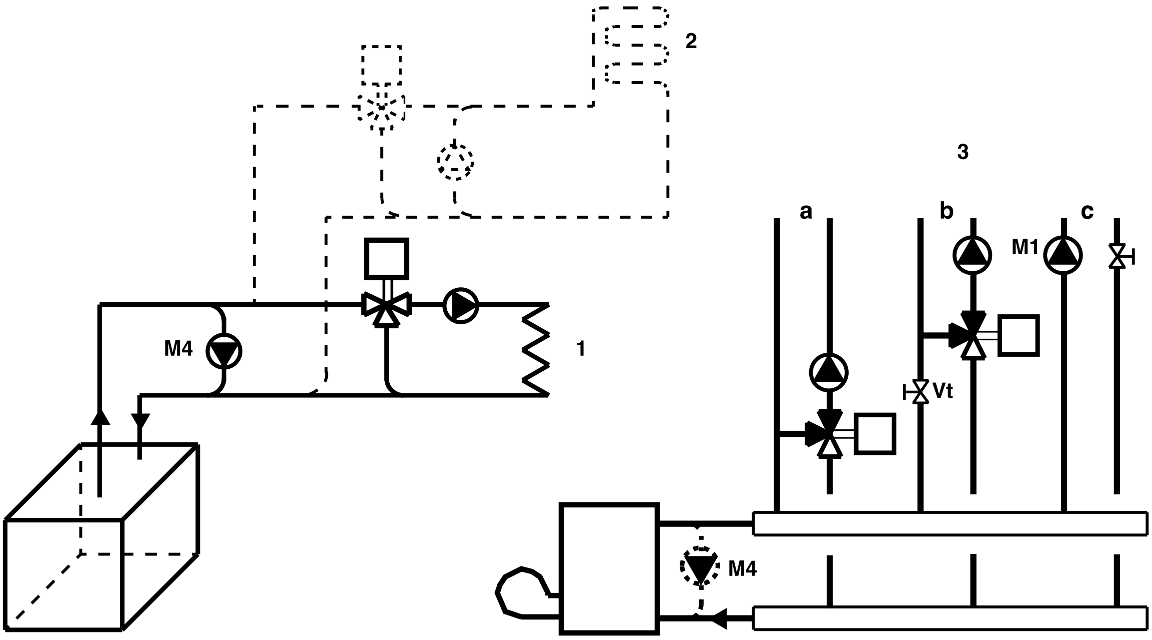

Fig. 32 Heating circuits
1 = plant with radiators, fan coils, etc.
2 = plant with panels
3 = manifold with several outlets

Examining the behaviour of the heating plants in Fig. 32 the following situations can be noted.

10.1 At the start-up of the plants after a prolonged shut-down e.g. at night

At each start-up, and for the whole period while the plant is reaching full capacity, the control valve(s) of the DHW/heating/auxiliary circuits is (are) completely open and as a consequence the volume of cold water circulating in the boiler corresponds to that required by the design.

The **M4** pump cannot increase the return water temperature even if the burner operates without interruptions, since the water coming from the boiler has a very low temperature.

In fact, since the difference in design temperature (Δt°) of the plants, and therefore also of the boiler, lies between 10…20K (mainly at10 K), with a design Δt° of 10K, if the water entering the boiler has a temperature of 20°C the water leaving will have one of 30°C

10.2 Following sudden and substantial variations in load

due to the insertion of a plant zone having intermittent operation (zone “c” of plant **3** in figure 32).

– for the period of time necessary for the circuit zone to reach full capacity the situation can be *the same* as in the previous case.

In this situation, too, the boiler temperature falls immediately with the result that pump **M4** might not be able to maintain the desired temperature of the return-to-boiler water.

In both these situations, in the absence of special measures by the control system for *allocating* correctly the volume of cold water coming from the plant (see section **10.8**) it is not possible to prevent the formation of acid condensation in the boiler.

10.3 With plant inoperative, operation with reduced loads

Behaviour varies according to the type of plant::

– *compensated control* with 3-port mixing valve and *boiler at fixed point* (plant **1** and **2**).
With reduced loads the temperature of the flow to the heat emitters is also low.

As a consequence, the mixing valve withdraws a minimum volume of hot water from the boiler and obviously the volume of cold water returning to the boiler is reduced.

As the flow to the boiler decreases, even without pump **M4**, the possibility increases, following the technical improvements by boiler manufacturers (return on the upper wall of the boiler) that the return water from the plant, mixing with the hot water in the boiler, heats up before coming into contact with the walls washed by the fumes.

However, the reduction in the flow to the boiler compromises the correct operation of the present-day pressurized boilers since it changes the conditions of the boiler circuit (the pressure drop varies with the square of the flow) and causes a rapid increase in the temperature *not controllable* by the boiler thermostats by switching off the burner at a temperature higher than the desired one.

With two-stage or modulating burners the situation improves but the need for a certain amount of circulation in the boiler remains, and in both situations the pump **M4** is required.

– *compensated control* with 3-port mixing valve and *boiler with temperature variable* in relation to utside temperature.

With reduced loads the volume of water the valve withdraws from the boiler is decidedly greater than in the previous example and so,also, is the volume returning to boiler.

Under these conditions it is essential for the variable boiler temperature to be limited to 60°C (minimum temperature limit) and that the pump **M4** is present .

10.4 Control of manifolds in a circuit without control valve

(plant “c” of circuit **3** in fig. 32), the anticondensing function can be undertaken by the pump **M1** provided the following conditions are respected:
pump in circuit “c” always operating, without interruptions, and boiler in operation
indicative flow value of at least 25…30% of plant total (sum of the single flows of the circuits).
If it is not possible to meet the above requirements the pump **M4** must nevertheless be installed.

10.5 Panels installation

In panel installations the return-to-boiler temperature, in all operating situations, is always below 50°C.
This is a further reason for using the circuit described in section **6.7**

10.6 Operation with no load

In the absence of a call for heat by the DHW/heating/auxiliary circuits, the control valve(s) is(are) closed; consequently, without pump **M4** there is no water circulation in the boiler and this accentuates the irregular operation of the boiler thermostats described in section **10.3**.

Conclusions

From what we have seen, the question of the minimum return-to-boiler temperature has to be considered together with that of ensuring a flow to the boiler not below a certain temperature.

For these reasons, rather than an anticondensing pump, we should consider it an **indispensable** pump for recycling in the boiler.

Indicative flow values: about 30% of the total plant flow.

10.7 Calculation of the flow and head of the boiler pump (M4)

When sizing the pump **M4** (for recycling in the boiler) the plant designer must pay special attention because :

– if pump **M4** is undersized, the burner will switch off unnecessarily and so prolong the time required for reaching full capacity.

vice versa

– if pump **M4** if pump M4 is oversized, it will cause an excessive supplementary circulation when the plant operates at full capacity (sections **10.3** and **10.6**), with the risk of an anomalous increase in pressure drop in the boiler branch circuit.

Indicative values can be obtained from the equations:

$$Qa = \frac{Qt}{tc - tan} \qquad \text{and} \qquad Han^{1)} = \Delta pc \, \frac{(Qa)^2}{Q1}$$

1) to this value you must add the continual and accidental losses in the recycle pump pipe branch

Legend :

Qa = flow of recycle pump (**M4**)
Q1 = plant design flow
Qt = power of the boiler in Kcal or energy in Kcal/h
tc = desired boiler operating temperature
tan = desired anticondensing temperature
Han = head of recycle pump (**M4**)
Δpc = pressure drop in boiler circuit

10.8 Measures to adopt for controlling minimum return temperature

10.8. 1 Modulating limit of minimum return temperature

The *simplest and cheapest* control method, under all the operating conditions of the plant, is that of exploiting the anticondensing function, incorporated in the controllers, and using a detector for minimum return temperature.
The compensating controller, with its return detector mounted on the return-to-boiler pipe, when the temperature falls below the desired value (50…55°C), *modulates as required* the closure of the heating circuits valve(s).

With the valves partially closed, the water flow returning to the boiler is reduced, thereby permitting the recycle pump to maintain (or restore) the temperature.

As with start-up, after a night shut-down the minimum temperature detector :

– maintains, or modulates to closure, the heating valve(s) so that water coming from the boiler returns to it.

– when the temperature settles at the desired minimum return value, the valves in the heating circuits start to open progressively, metering over time the volume of cold water returning to the boiler.

This mode of operation permits maintaining the return-to-boiler temperature at the desired value for the whole of the time required for the plant to reach full capacity.

In other words, each time, both at start-up and during plant operation at full capacity, the return-to- boiler temperature decreases in respect of the desired value (50…55°C) the controller(s) consider the desired operating temperature to be that of the minimum detector instead of that of the compensation curve.

The use of a minimum temperature detector ensures that, in all situations, from starting up to full capacity and with reduced or nil loads, the return temperature is at the desired value without noticeable variations.

Note : The modulating limit keeps the return-to-boiler temperature constant **without** increasing the time for reaching full capacity..

10.8. 2 Minimum limit temperature with two positions

The use of a two-position thermostat to control the minimum return temperature is an *unreliable* solution (and possibly a dangerous one).

The thermostat :

– starts the recycle pump M4 when the temperature is decreased by its differential in respect of the desired value

vice versa

– stops the pump when the desired temperature is re-established.

This type of operation does not take into account :

– the condition of always maintaining circulation in the boiler

– the desired temperature during the passage to full capacity and at reduced or nil loads

– the accuracy of the desired temperature which, under the most favourable conditions, will oscillate according to the thermostat differential.

10.9 Siting the detector for the minimum return temperature

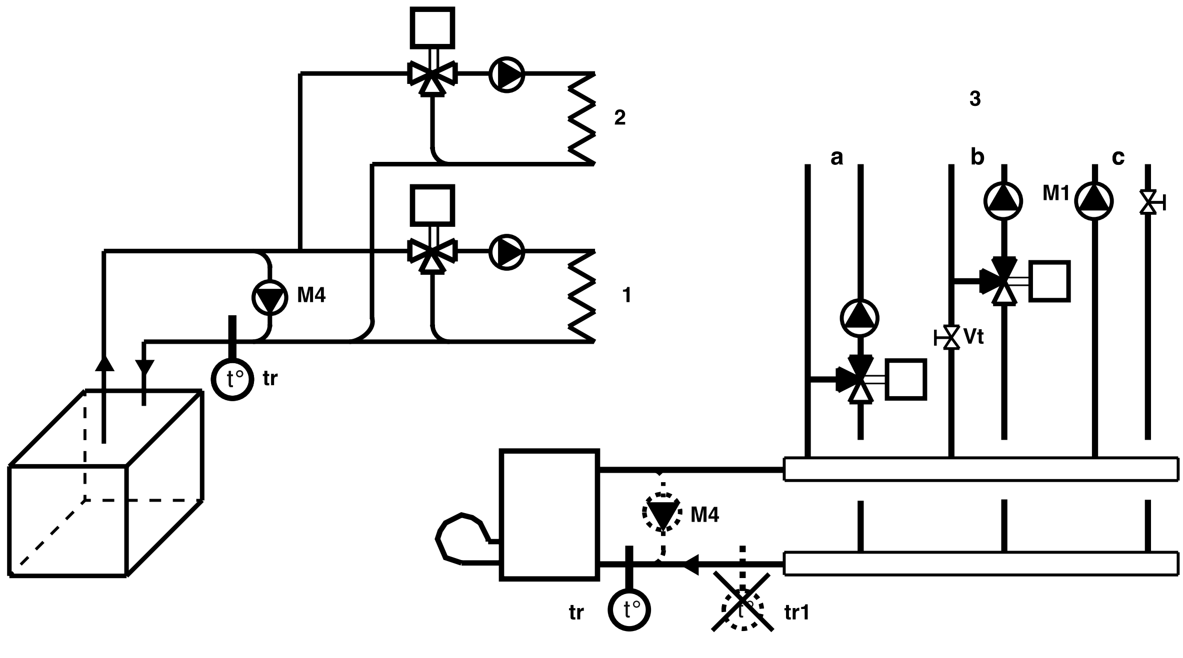

Fig. 33 Heating circuits

Legend :
M4 = recycle pump
tr = anticondensing minimum detector
tr1 = incorrect mounting of anticondensing detector
1 & 2 = radiator circuits
3 = manifold with several derivations

The minimum temperature detector must always be mounted on the length of pipe between the joint with the recycle pump and that of the return to boiler (see fig. 33).
In this position the detector :

– *always* measures the temperature of the mixture of water coming from the boiler with that returning from the plants

vice versa

– the position of detector “tr1” is incorrect (dotted line in fig 33)

reason

It measures *only* the temperature of the return water from the plants which, even if below that desired, may be re-heated by the quota of water coming from the boiler, by means of the recycle pump **M4**.

In short, it can act by modulating the closure of the plant valve(s) even when it is not necessary

10.9. 1 Control of the return of several circuits on the manifold

When there are several DHW/heating/auxiliary circuits it is not essential for the anticondensing detector to act at the same time for all the controllers of the manifold distribution circuits.

In many situations it is sufficient to act on some in order to obtain control of the return-to-boiler temperature without "penalizing" all the DHW/heating/etc. circuits.

In particular, the return-to-boiler minimum detector, in order to reduce the temperature, should not act on :

– the preparation of the DHW (calorifier) (plant “c” in figure 33)
– the controls of the pre-heating batteries (danger of frost)
– the control of hot water in fan coil plants

10.9. 2 Control of return in air circuits

In these applications it is advisable, at each start-up of the plants, to switch off the fans while the plant is reaching full capacity.

Note : when operating at full capacity with 3-port diverting valves the return temperature is practically always high

10.9. 3 Circuit with panels installation and calorifier

In this application, if DHW precedence is requested, with the circuit without anticondensing detector and with a panels installation not executed as described in section **6.7**

At the end of the period with heating blocked (precedence calorifier) the flow temperature to the panels can undergo an *appreciable* increase, followed by oscillations, due to the excessive opening of the valve.

This difficulty does not arise if :

the circuit described in section **6.7** is adopted

and

the return minimum temperature detector (anticondensing) is used