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BALANCING OF RADIATOR SYSTEMS

A manual for the design, balancing and troubleshooting of hydronic radiator heating systems.

Opera House, Gothenburg, Sweden

“Balancing of radiator systems” is the third manual in the TA series of publications about hydronic design and balancing. The first manual deals with balancing control loops, the second with balancing distribution systems and the fourth with hydronic balancing with differential pressure controllers.

This publication has been prepared for an international audience. Because the use of language and terminology differs from country to country, you may find that some terms and symbols are not those you are used to. We hope this will not cause too much inconvenience.

Written by Robert Petitjean. Warm thanks to TA experts in hydronic balancing: Bjarne Andreassen, Eric Bernadou, Jean-Christophe Carette, Bo G Eriksson and Peter Rees for their valuable contributions.

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— 3rd edition —

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Further information is available in our general book on balancing “Total hydronic balancing”.

Why balance?

Many property managers spend fortunes dealing with complaints about the indoor climate. This may be the case even in new buildings using the most recent control technology. These problems are widespread:

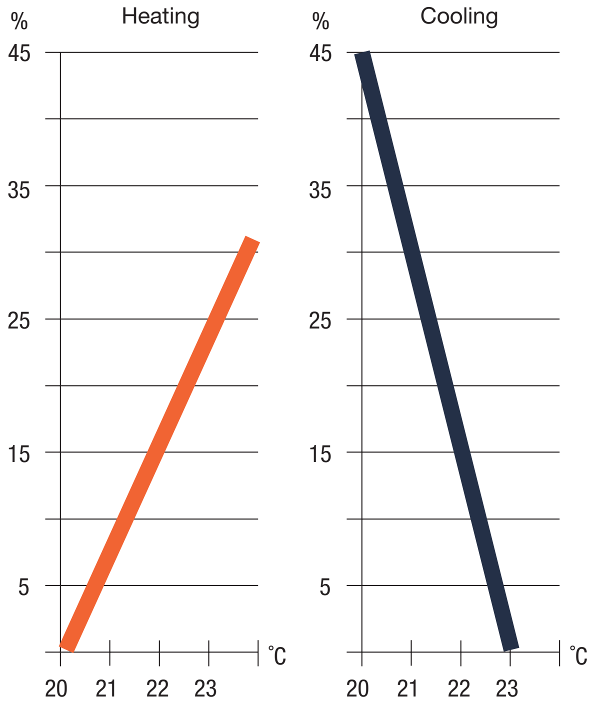
- Some rooms never reach the desired temperatures.
- Room temperatures oscillate, particularly at low and medium loads, even though the terminals have sophisticated controllers.
- Although the rated power of the production units may be sufficient, design power can't be transmitted, particularly during start-up after weekend or night setback.

These problems frequently occur because incorrect flows keep controllers from doing their job. Controllers can control efficiently only if design flows prevail in the plant when operating at design condition.

The only way to get design flows when required is to balance the plant. Balancing means adjusting the flows at correct values at design condition. Avoiding underflows at design condition makes sure that underflows will be avoided in all other normal conditions. Balancing is necessary for three reasons:

1. The production units must be balanced to obtain design flow in each boiler or chiller. Furthermore, in most cases, the flow in each unit has to be kept constant when required. Fluctuations reduce the production efficiency, shorten the life of the production units and make effective control difficult.
2. The distribution system must be balanced to make sure all terminals can receive at least design flow, regardless of the total average load on the plant.
3. The control loops must be balanced to bring about the proper working conditions for the control valves and to make primary and secondary flows compatible.

This manual deals with the balancing of radiator distribution systems.
Other manuals available are:
Manual 1: Balancing of control loops.
Manual 2: Balancing of distribution systems.
Manual 4: Balancing with differential pressure controllers.



Percentage increase in energy costs for every degree C too high, or too low, relative to average building temperature.

Why is the average temperature higher in a plant that is not balanced? During cold weather it would be too hot close to the boiler and too cold on the top floors. People would increase the supply temperature in the building. People on the top floors would stop complaning and people close to the boiler would open the windows. During hot weather the same applies. It is just that it would be too cold close to the chiller, and too hot on the top floors. One degree more or less in a single room rarely makes any difference to human comfort or to energy costs. But when the average temperature in the building is wrong, it becomes costly.

One degree above 20 °C increases heating costs by at least 8 per cent in mid Europe (12 per cent in the south of Europe). One degree below 23 °C increases cooling costs by 15 per cent in Europe.

1. Balancing of radiator systems

1.1 Overflows cause underflows

The design flow must pass through each radiator at design condition, which requires individual local adjustment.

At first sight, there would appear to be no advantage in balancing a heating plant equipped with thermostatic valves as their function is to adjust the flow to the correct value. Hydronic balancing should therefore be obtained automatically.

This would be more or less true in normal operation provided that all control loops are stable. However, unbalanced radiators create major distortions between flows. Let us consider two radiators on the same branch, one 500 W and the other 2500 W. The installer usually installs the same thermostatic valves on all radiators. The radiator headloss is normally negligible and the flow is limited mainly by the thermostatic valve. Flows will therefore be the same for both radiators. If this flow is right for the 2500 W radiator, it is five times the design value for the 500 W radiator.

As if that were not enough to create problems in a plant, other distortions are added. For example, thermostatic valves left at the maximum set point, will keep them open permanently. If the maximum flow is not limited, these overflows create underflows in other parts of the plant where the required room temperature cannot be obtained.

Restarting the plant, every morning after night setback, is a serious problem as most thermostatic valves are open. This creates overflows getting unpredicted pressure drops in some pipes, consequently reducing the flows in unfavoured circuits. These circuits do not receive sufficient water until the favoured thermostatic valves are at their nominal lift. This causes the plant to have a non uniform start-up, which makes management by a central controller difficult and also makes any form of optimisation practically impossible.

Figure 1.1 represents a branch with four radiators. The pressure drops in the pipes between each radiator are one kPa at design flow. The available differential pressure is 9 kPa for the first radiator and 6 kPa for the last one.

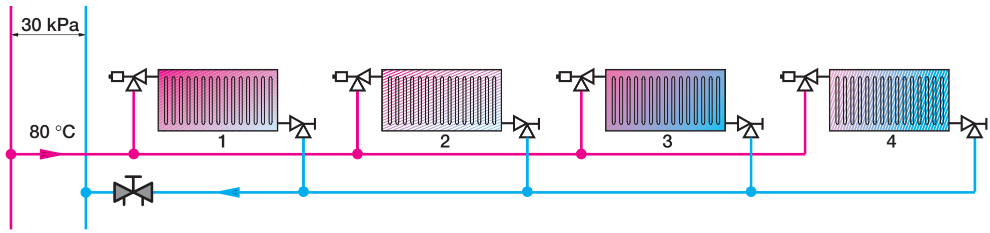


Fig 1.1. Branch with four radiators.

The presettings of the thermostatic valves have been chosen to obtain the design flow in each radiator. The branches and risers are also balanced.

The results are in table 1.1.

Branches and risers balanced – Thermostatic valves balanced					Total flow
Radiators	1	2	3	4	l/h
Kv thermostatic valve	0.04	0.15	0.25	0.14	
Flow (l/h)	11	43	65	33	152
Heat output (W)	255	1000	1512	765	
Room t° in °C	20	20	20	20	

Table 1.1. Results obtained when the plant is fully balanced.

Now, let us consider the case of a plant where the risers and branches are balanced, but the radiator valves are not preset. The total flow in the branch is correct, but the radiators are not working at design flow. The results are shown in table 1.2.

Branches and risers balanced Thermostatic valves not balanced and fully open					Total flow
Radiators	1	2	3	4	l/h
Kv thermostatic valve	0.8	0.8	0.8	0.8	
Flow (l/h)	66	45	30	11	152
Flow (%)	600	105	46	33	
Heat output (W)	290	1006	1270	573	
Heat output (%)	114	101	84	75	
Room t° in °C	24.1	20.2	15.2	12.4	

Table 1.2. Risers and branches are balanced, but not the thermostatic valves.

The first radiator receives 6 times its design flow. This increases the heat output by only 14%. That means that the necessary time to reach the design room temperature, after a night setback, is not reduced significantly. If the thermostatic valve 1 is set at the correct value, the flow in the first radiator will be reduced after a certain time allowing the two last radiators to finally receive their design flow. Start-up is then much longer than expected.

If the thermostatic valve of the first radiator is maintained fully open, radiators 3 and 4 will never obtain their design flow and the room temperatures obtained at design condition are given in table 1.2 (12.4 °C for room 4).

Another possibility is to preset all thermostatic valves without any balancing valves in the branches and risers. In this case, the balancing procedure is very difficult as all circuits are interactive. Therefore, all the excess of differential pressure has to be taken away by the thermostatic valves, which can be noisy. Moreover, the valve's maximum Kv is so small that the risk of clogging is high.

1.2 Overflows in distribution

An overflow in the distribution creates one or several undesired mixing points and the actual supply water temperature is lower than expected.

An overflow in the distribution, particularly during the morning start-up, creates an incompatibility problem between production and distribution. Let us consider two typical examples.

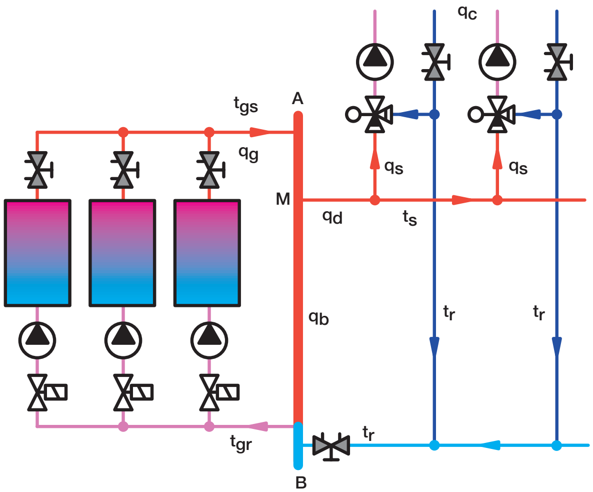


Fig 1.2. Several circuits are connected to the heating plant through a decoupling bypass.

In Fig 1.2, if the distribution flow q_d is greater than the production flow qg, the difference circulates in the bypass in the direction BM. A mixing point is therefore created at M causing a drop in the supply water temperature. The maximum supply water temperature in the distribution is lower than the maximum temperature obtained in the boilers. The plant therefore has difficulties at start-up as the installed power cannot be transmitted. In some plants, the problem is solved by installing additional boilers, which can increase the production flow q_g, making it compatible with the distribution flow q_d. This type of solution is very expensive both in capital cost and in operation as the seasonal efficiency drops.

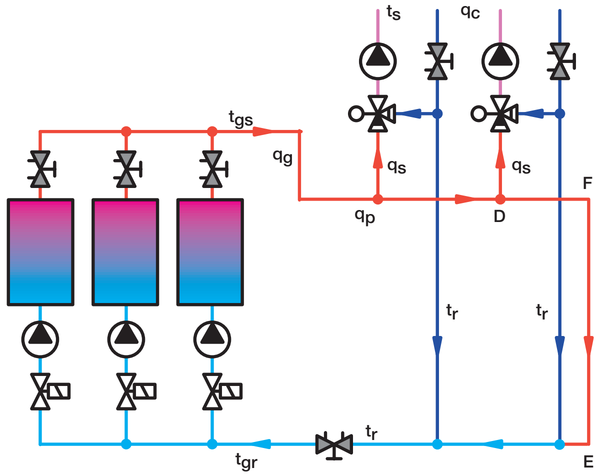


Fig 1.3. Distribution through a closed loop.

In Fig 1.3, circuit overflows cause circulation in pipe FE from E towards F with a mixing point created in D. The last circuit is then supplied from its own return. This circuit works under particularly bad conditions and becomes the plant's "nightmare" circuit.

Placing a non-return valve in pipe EF would appear to solve the problem, but it actually creates another problem as the loop is open and boiler pumps go into series with circuit pumps, making some control loops unstable.

In conclusion, the most efficient and easiest solution is to correctly balance production and distribution ensuring their flow compatibility.

2. Radiator valves

2.1 General

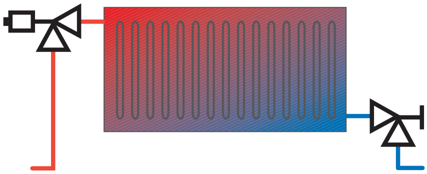


Fig 2.1. Radiator valves on the inlet and the outlet.

Radiator valves have several functions. One of them is to isolate the radiator on the inlet and the outlet. An important function is also to adjust the flow to the required value. This function can be achieved either by the valve on the inlet or by the return valve.

2.1.1 WHEN THE INLET VALVE IS USED ONLY TO ISOLATE

When the manual valve on the inlet is only used for shut-off function, its oversizing is not so important. The limitation of the flow at design value is obtained with the valve on the return, which takes the majority of the differential pressure available. This return valve must have a profiled cone to get adequate authority on the flow in the range of adjustment. The presetting of the return valve is made according to the expected available differential pressure and the required design flow.

The water flow depends on the differential pressure across the valve and its Kv according to the equation:

$$q = 100\ Kv \sqrt{\Delta p} \qquad \text{(q in l/h and } \Delta p \text{ in kPa)}$$

The Kv value depends on the degree of opening of the valve. When the valve is fully open the specific Kv obtained is called the Kvs.

The correct valve can be determined with the help of a nomogram, (see e.g., Fig 4.2).

2.1.2 WHEN THE INLET VALVE IS USED TO ISOLATE AND ADJUST THE FLOW

This manual valve must be provided with a profiled plug to obtain a progressive restriction of the flow when shutting the valve. This progressiveness only works if the valve is not oversized and thus has a sufficient authority.

The Kvs of the valve in the inlet is chosen to obtain approximately the design flow for the valve 75% open. When the available Kvs is too high, one solution is to limit the degree of opening of this valve to the correct Kv. Another possibility is to install in the inlet a double regulating valve where the shut-off and regulating functions are independent.

2.2 What is a thermostatic valve?

A thermostatic valve is a self-acting automatic valve controlled by an expanding element. Depending on the difference between the temperature set point and the room temperature, the valve gradually opens or closes.

All the thermostatic valves on the market have a total lift of several millimetres. However, starting from the valve in shut position, a decrease in the room temperature of 2K will open the valve about 0.5 mm. This part of the lift, where the control valve normally works, is called the nominal lift.

Fig 2.2 shows two relations between the water flow and the room temperature. Curve a is for an unlimited flow thermostatic valve. Curve b is for a thermostatic valve with flow limitation. This limitation is obtained with an adjustable resistance in series with the active port of the valve.

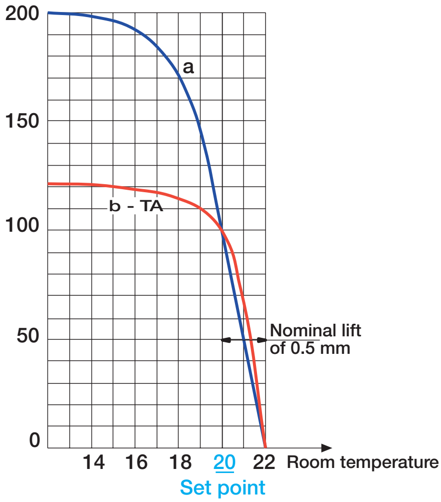


Fig 2.2. Relation between the water flow and the room temperature for a thermostatic valve supplied at constant differential pressure. a - unlimited flow valve. b - valve with flow limitation.

In practice, the thermostatic valve reacts gradually to the changes in the room temperature, unless the temperature starts to develop in the other direction. In this case the plug of the valve does not move until the room temperature varies by a value that exceeds the hysteresis (normally around 0.5K). This phenomenon sometimes gives the impression of a stable heat transfer from the radiator whereas the control loop can become unstable in the longer term.

Thermostatic valves allow the achievement of the correct temperature in each room individually. They compensate a possible oversizing of the radiator and reduce heat output when other sources of heating (lamps, people, sun, etc.) compensate for part of the heat losses. In this respect, thermostatic valves provide the user with more flexibility, improve the comfort and save energy.

2.3 Thermostatic valves and the supply water temperature

Controlling the heat output from a radiator with a thermostatic valve is quite difficult when the supply water temperature is maintained constant during the whole heating season. For this reason, the supply water temperature is normally variable and depends, for instance, on outdoor conditions.

As an example, we can take a radiator permanently supplied at 80 °C throughout the entire heating season. The thermostatic valve is assumed to be correctly sized to give design flow at nominal opening (80/60 conditions). The minimum outdoor design temperature is –10 °C.

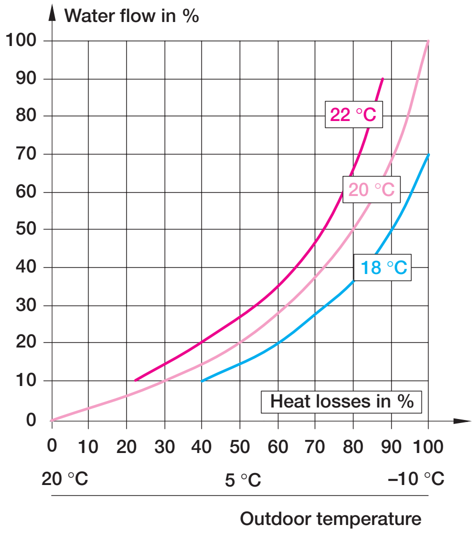


Fig 2.3. The necessary water flow to maintain the room temperature constant as a function of outdoor conditions. The supply water temperature is assumed to be constant and equal to 80 °C.

Fig 2.3 shows the necessary water flow in the radiator as a function of outdoor conditions, in order to obtain room temperatures of 18, 20 and 22 °C.

Around the mean winter temperature (t_e = 5 °C), a water flow variation of 4% will change the room temperature by 2K. To obtain a precise room temperature within ± 0.5K, the water flow must be controlled with an accuracy of ± 1%. As 50% of the load corresponds to 20% of the flow, the lift of the valve has to be set at an opening of 0.1 mm (20% of the nominal lift of 0.5 mm), with a precision of ± 0.005 mm (1% of nominal lift)! Obviously this is impossible, and the thermostatic valve cannot find a stable degree of opening. It then works in on-off mode with oscillations in the room temperature. When the thermostatic valve is open, the heat output is much higher than necessary, creating a transitory increase of room temperature before the thermostatic valve reacts.

This is why the thermostatic valve is normally used with a central controller that modifies the supply water temperature to suit requirements. These are determined by an outdoor sensor or by a temperature sensor located in a reference room (not fitted with thermostatic valves) or by a combination of the two. The thermostatic valve corrects residual variations as a function of the conditions specific to each room.

In conclusion, the heat output of a radiator cannot be controlled only by varying the flow. Basic control is obtained by controlling the supply water temperature according to general needs.

2.4 Is the thermostatic valve a proportional controller?

The thermostatic valve theoretically behaves like a proportional controller. In practice, working conditions are not always favourable and the thermostatic valve often works as a temperature limiter. In this case a small proportional band gives better results. Even then, it may sometimes give the impression of behaving proportionally as it moves into intermediate, temporarily stable positions, as a function of its hysteresis.

A proportional controller gradually opens or closes the control valve in proportion to the deviation between the controlled value and its set point.

Fig 2.4 shows a level controller. The operation obtained is similar to that of a thermostatic valve if it is assumed that the water level represents the room temperature. The flow Z corresponds to the heat losses, and the supply flow Y corresponds to the radiator's heat output.

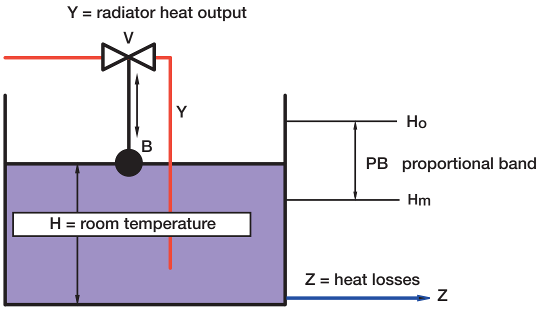


Fig 2.4. Analogue representation of a thermostatic valve.

When the level decreases, the float B goes down and opens valve V proportionally to the level reduction. A balance is obtained when the supply water flow Y equals the flow Z.

When Z = 0, the level rises to H_o at which valve V is closed. When Z reaches its maximum value, a stable situation is obtained when valve V is fully open. The float is then in the H_m position. The level therefore takes on stable values between H_o and H_m depending on the amplitude of disturbances.

This difference H_o–H_m is called the proportional band. It is the level variation necessary to change the control valve from maximum opening to closing.

If this proportional band is reduced to increase the control accuracy, there is a risk of reaching a critical value at which the control loop becomes unstable. A small proportional band results in a large variation of the flow Y for a small change in the level. This flow variation may then be larger than the disturbance that caused the change in level, thus creating a reverse disturbance larger than the initial disturbance. The level then oscillates continuously.

Reconsider Fig 2.2. The valve is fully closed for a room temperature of 22 °C, and fully open for t_i = 14 °C. The proportional band is therefore 8K. However, for a thermostatic valve with presetting, the design flow is obtained in practice for a room temperature variation of 2K and it is normal practice to arbitrarily assume that the proportional band of the thermostatic valve is 2K. We would like to clarify that this 2K is not really the proportional band of a thermostatic valve. The proportional band has to represent the range of room temperature modifications where linearity is obtained between the room temperature and the water flow. This control is expressed in terms of % of flow per K of the room temperature deviation. *Therefore, we have adopted, for thermostatic valves, a specific definition for the proportional band: it's the double of the deviation in room temperature which changes the water flow from 0 to 50% of the design value (100% being obtained for a deviation of 2K). This definition concerns the assembly consisting of thermostatic valve + radiator + return valve (if there is one).*

If we consider now that a return valve or an internal restriction in the thermostatic valve is used to obtain the correct flow at nominal lift, the resulting curves for some settings are shown in Fig 2.5.

The set point is chosen so that the flow is less than 100% when the room temperature exceeds 20 °C.

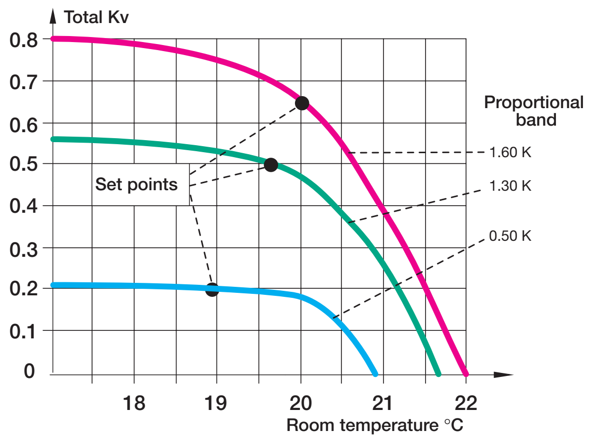


Fig 2.5. A regulating valve or an internal restriction in the thermostatic valve modifies the resulting Kv = f (ti) curve and the practical proportional band.

Table 2.1 shows the variation of the proportional band, for three setting examples, when a restriction reduces the maximum flow.

Kv2	0.65	0.50	0.20
Kv_{max}	0.80	0.56	0.20
BP	1.60	1.30	0.50
SP	20.00	19.7	18.90
ST	22.00	21.7	20.90

Kv2 = (Kv at ΔT 2K) Kv at nominal lift corresponding to a deviation of 2K, this Kv corresponds to the design Kv.
Kv_{max} = Kv obtained with the valve fully open.
PB = Proportional band
SP = Set point adopted for a required room temperature of 20 °C.
ST = Room temperature at which the valve is completely shut.

Table 2.1. Variation of the proportional band of one thermostatic valve under the effect of a restriction in series.

The real curve between Kv and room temperature also depends on the hysteresis of the valve as well as the variation of the microclimate around the thermostatic head.

In any case, a proportional band of less than 1K will almost certainly make the valve work in on-off mode. This is not a serious problem if oscillations of the room temperature are practically not perceivable, which is the case in balanced plants where the water temperature is controlled as a function of outdoor conditions. In well-insulated buildings a narrow proportional band gives more accurate control of room temperature despite the on-off behaviour of the control loop, which also contributes to reduced energy consumption. When working with wide proportional bands, we may get a stable control. However, as the temperature is slowly moving within a large span, we don't save as much energy as possible. Effectively, we don't take all the benefit of internal heat or sun energy.

The question is the following: when a thermostatic valve works to compensate for an internal emission, is it better to have a small or a large proportional band?

Let us consider two different thermostatic valves, set at 20 °C, with the same Kv but with a proportional band of 2 and 1K respectively.

With a proportional band of 2K, the room temperature can increase to 22 °C before the radiator emission will be stopped. If the proportional band is only 1K the radiator will be already isolated for a room temperature of 21 °C. It is then possible to save more energy when working with small proportional bands.

2.5 Should a plant be hydraulically balanced with all thermostatic valves fully open?

The answer is yes and the thermostatic valves must have a saturated characteristic.

Balancing the radiators in a circuit results in obtaining correct flows in each radiator at design condition. At intermediate loads, flows and pressure drops in pipes are reduced, differential pressures increase and each radiator can at least obtain its design flow.

Some consider that all thermostatic valves should be set to their nominal opening before balancing a plant. This appears logical as flows are normally determined in these conditions. The thermostatic heads should then be replaced by graduated caps for setting the valves at their nominal lift.

Whenever the plant is started up, after the night setback, thermostatic valves are opened beyond their nominal opening and will be in overflow, creating underflows in other parts of the plant. The purpose of balancing is thus not achieved.

This situation is difficult in the case of an unlimited flow thermostatic valve such as that shown in Fig 2.2a. Moreover, overflows are permanent on valves for which the thermostatic head has been removed.

Thermostatic valves with an interchangeable plug, allowing the achievement of the right Kv, normally don't have a flat enough curve to solve the problem.

The problem is related to the big difference in flows between the valve fully open and the valve at nominal lift (Fig 2.2a). Solving this problem is quite simple: the valve characteristic has to be saturated. It means that the flow will not significantly increase beyond the nominal opening (Fig 2.2b). This is obtained with a resistance in series with the thermostatic valve (Fig 2.5). In this case, the flow/opening curve beyond the nominal lift is so flat that the plant can be balanced with all thermostatic heads removed.

This discussion demonstrates the necessity to balance the plant with all thermostatic heads removed and to use thermostatic valves with a small difference between the design and the maximum flows, this means with a saturated characteristic as shown on Fig 2.5.

However, in the case of an occupied building, the operation of removing all thermostatic heads and replacing them after balancing is a difficult operation. Sending circulars asking occupants to carry out this operation is not a particularly reliable method. Some installers prefer to do the balancing during the heating season; they reduce the hot water temperature significantly the day before, inciting occupants to fully open their thermostatic valves.

2.6 Accuracy to be obtained on the flow

In most cases the flow has to be adjusted with an accuracy of ± 10%.

In section 1 we considered the disadvantages of hydronic unbalances in plants. Before studying balancing procedures, we must define the precision with which flows have to be adjusted.

In practice, flow adjustment precision depends on the required room temperature precision. This precision also depends on other factors such as control of the supply water temperature and the relation between the required and installed capacity. There is no point in imposing a very high accuracy on the flow if the supply water temperature is not controlled with an accuracy producing equivalent effects on the room temperature.

An underflow cannot be compensated by the control loop, and has a direct effect on the room temperature under maximum load conditions; it must therefore be limited. An overflow has no direct consequence on the room temperature since in theory, the control loop can compensate for it. However, when the control valve is fully open, for example, when starting up the plant, this overflow produces underflows in other units and makes distribution incompatible with production. Overflows must therefore also be limited.

Table 2.2 compares the influence of the flow on the room temperature under well-defined design conditions.

Design			Allowable deviation in % of design water flow, for a room temperature accuracy of 0.5K	
t_{ec}	t_{sc}	t_{rc}	– 0.5K	+ 0.5K
0	90	70	– 15	+21
	82	71	– 24	+44
– 10	93	82	– 21	+34
	90	70	– 12	+15
	90	40	– 4	+ 4
	80	60	– 10	+13
	80	50	– 7	+ 7
	80	40	– 4	+ 5
	60	40	– 8	+ 9
	55	45	– 15	+ 20
– 20	90	70	– 10	+13
	80	60	– 9	+10
	80	40	– 4	+ 4
	75	45	– 5	+ 6
	70	45	– 6	+ 7
	60	40	– 7	+ 8
	55	45	– 13	+17

Table 2.2. Variations of the flow q in the radiator to modify the room temperature by 0.5K at full load.

Let us take an example for a heating plant working with the following design conditions: Supply water temperature t_{sc} = 80 °C, return t_{rc} = 60 °C and room temperature t_{ic} = 20 °C. The design outdoor temperature is tec = –10 °C. A room temperature variation of 0.5K can be obtained by reducing the water flow by Δq = 10%.

The water flow adjustment accuracy must be better when the plant is working with a relatively high thermal effectiveness Φ.

For a rough conclusion, we can see that water flows have to be controlled with an accuracy of ± 10 to ± 15%. Concurrently, the water temperature has to be controlled with an accuracy of ± 1 to ± 1.5K.

We may be tempted to accept overflows, especially when they have little effect on the room temperature. This would neglect the pernicious effects of overflows which create underflows elsewhere making it impossible to obtain the required water temperature at high loads, due to incompatibility between production and distribution flows (see section 1.2).

3. Radiators

3.1 Nominal and design conditions

A radiator heat output is to be defined for a given room temperature (20 °C), water supply and return temperatures, for example 75 and 65 °C. The temperatures of 20, 65 and 75 °C are nominal values of the room temperature and water temperatures. They are identified by the subscript "n" (for example t_{in} = nominal room temperature). Nominal values are actually catalogue values used by manufacturers; they determine the conditions under which the power of a unit is defined. According to European norms EN442, the nominal power of a radiator is valid for a supply water temperature of 75 °C, a return water temperature of 65 °C and a room temperature of 20 °C. But, normally, a radiator does not work in these conditions. Then, the required design water flow in the radiator must be determined in each particular case. It is obviously meaningless to try to adjust the water flow in a radiator if this flow is not correctly determined.

The plant is calculated in certain conditions with specific values for the controlled variables, outdoor conditions, supply and return water temperatures. Those values, used to calculate the plant, are the design values; they are identified by a subscript "c" (values used for calculations).

3.2 Selection of a radiator not working in nominal conditions

Radiator heat output in catalogues refers to nominal conditions, for example, water supply temperature t_{sn} = 75 °C, return temperature t_{rn} = 65 °C and a room temperature t_{in} = 20 °C. How is a radiator selected if it does not work in these conditions?

The real transferred power P is related to the nominal power P_n as follows:

$$P = P_n \times \left(\frac{(t_s - t_i)\ \ (t_r - t_i)}{(t_{sn} - t_{in})\ \ (t_{rn} - t_{in})} \right)^{n/2}$$

No subscript: present conditions *Subscript n: nominal conditions.*
t_s = *Supply water temperature.*
t_r = *Return water temperature.*
t_i = *Room temperature.*
n = *This exponent for radiators is normally taken = 1.3*

This formula expresses the influence of the temperature geometric average between the radiator and the room. This formula is translated into a graph on figure A1 in appendix A where some specific examples are explained.

> *Example:* What is the nominal power 75/65 of a radiator which has to deliver 1000 W in a room at 22 °C when the actual supply and return water temperature are respectively 72 and 60 °C respectively?
>
> In Fig A1 (Appendix A), join $t_s - t_i = 72 - 22 = 50$ to $t_r - t_i = 60 - 22 = 38$ to find $Sp = P_n/P = 1.18$. The nominal power to install is 1000 x 1.18 = 1180 W.

This formula is theoretical as it assumes that the water flow is distributed uniformly in the radiator.

Heat output is also affected by a window sill above the radiator that may reduce heat output by 35%. A radiator close to a window generates hot air circulation and supplementary heat losses through the window, reducing the energy really transmitted in the room. The nominal power of a radiator is determined in favourable conditions, which are not always reproduced in practice. A coefficient of security remains necessary when a radiator is selected.

3.3 Emission of a radiator as a function of the water flow

The required water flow in the radiator can be calculated using the following equation:

$$q = \frac{0.86 \times P}{\Delta T}$$

q: flow in l/h   P: heat output in W   ΔT: temperature drop in K

For a 1000 W radiator and a design temperature drop of 20K, the required water flow is $0.86 \times 1000 / 20 = 43$ l/h.

However, when the flow varies, the water temperature drop also varies which makes the relation between the flow and heat output non-linear.

Fig 3.1 shows this relation for a supply water temperature of 80 °C and various temperature drops ΔT_c.

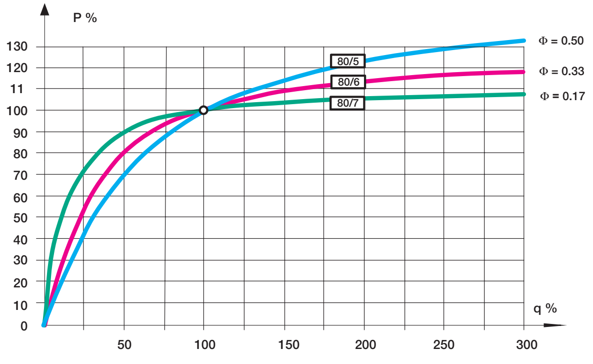


Fig 3.1. Heat output, in a room at 20 °C, as a function of the water flow, for a radiator (n = 1.3) and for different water ΔT_c values. t_{sc} = 80 °C.

At the origin, the gradient of the “heat emission/flow” curve is the inverse of the thermal effectiveness “Φ” of the radiator. This thermal effectiveness is defined as follows:

$$\phi = \frac{\Delta Tc}{\Delta To}$$

ΔT_C = design water temperature drop.
ΔT_O = water temperature drop at zero load = $t_{SC} - t_{iC}$

For 80/60 design conditions, the thermal effectiveness Φ is: (80 – 60) / (80 – 20) = 0.33 and the increased power at the origin is 1/0.33 = 3% power per % of flow.

At design condition, an overflow in the radiator does not significantly increase the emitted heat, particularly when the thermal effectiveness is low.

3.4 Selection of the design water temperature drop

For a supply water temperature between 70 and 90 °C, it’s quite common to design the plants for a ΔT = 20. This magic value has been adopted for many years and translated into local units (20 °C in continental Europe and 20 °F (11 °C) in the UK and USA, for instance). However, to reduce the return water temperature in district heating or when using condensing boilers, a higher ΔT is adopted. The design ΔT depends mainly on the habits in each country, but it can be optimised according to each specific plant.

Radiators working with a low water temperature drop ΔT_C have a strongly saturated response curve P% = f (q%). Flow variations therefore have little influence on the maximum emission. However, these radiators become difficult to control at low loads since the emission is very dependent on the flow in this zone.

The use of a high ΔT_C can reduce water flows, pumping costs, pipe diameters and losses. Control of the radiator is also improved. However, the maximum power becomes more sensitive to the water flow, requiring a precise hydronic balancing of the plant.

A high value of ΔT_C reduces heat exchanges, thus requiring the use of radiators with larger surface areas. For example, the use of a ΔT_C of 30K instead of 20K reduces the heat exchange approximately by 16%.

The optimum ΔT depends on each plant. Increasing the ΔT reduces the water flows, the sizes of the pipes and accessories, the pumping costs and the heat losses in pipes but radiator surfaces have to be increased. The optimum ΔT can therefore be calculated for each plant.

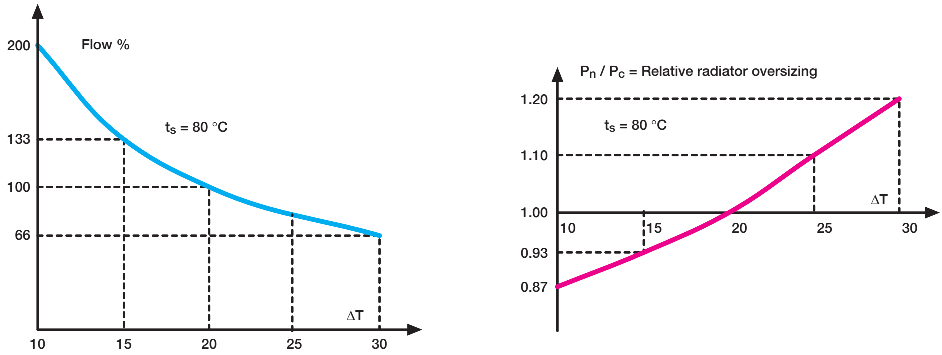


Fig 3.2. When the designed ΔT is increased, the water flow decreases but the required radiator surfaces increase (t_s = 80 °C).

3.5 Existing plants

How to compensate for oversized radiators after an improvement of the insulation of the building.

Existing plants can be treated in the same way as new plants. However, improvements may have been made to the building, considerably reducing heat losses. Radiators will then be oversized with respect to the initial conditions.

If the thermal insulation was improved uniformly, the heat output of the radiators is adjusted to new conditions by reducing the water supply temperature.

An example of a calculation: Take, for instance, a radiator with a nominal power of 1200 W in conditions 75/65. The design power required is, for instance, 1000 W for a supply water temperature of t_s = 80 °C in a room temperature of t_i = 20 °C. What flow should the radiator have?

The nominal oversizing factor is $S_{pn} = P_n /P$ = 1200/1000 = 1.2.

Referring to Fig A1 in appendix A, join $t_s - t_i$ = 80 – 20 = 60 °C to P_n/P = 1.2 to find $t_r - t_i$ = 31.2 °C. Then t_r = 51.2 and ΔT = 80 – 51.2 = 28.8 K. Finally, q = 0.86 × 1000/ 28.8 = 30 l/h.

Fig A2 can also be used. Join $t_s - t_i$ = 80 – 20 = 60 °C to P_n /P = 1.2 to find q = 30 l/h per 1000 W.

In most plants, thermal insulation is not improved uniformly and each radiator has to be treated independently as in section 3.2.

4. Two-pipe distribution

4.1 Balancing of radiators based on a constant Δp

To set the Kv of the thermostatic valves, the differential pressure is considered to be of ΔH_O = 10 kPa, for instance. This differential pressure is automatically obtained after balancing the distribution.

4.1.1 CHOOSING THE DESIGN DIFFERENTIAL PRESSURE

If a flow measurement device is available at each radiator, a standard balancing procedure can be used and a balancing valve on the circuit acts as a partner valve. This can keep previously adjusted radiator flows constant while others are being adjusted (the compensated method). However, thermostatic valves are generally preset according to calculated values.

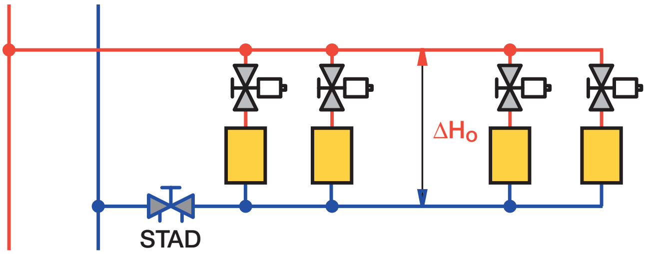


Fig 4.1. Each radiator valve is adjusted as if it were subject to the same differential pressure ΔH_O.

The main pressure drop is in the thermostatic valve with adjustable Kv as the pressure drop in the radiator is normally low. Since some inaccuracy is acceptable on flows, we can assume that each radiator in a branch is subject to the same differential pressure ΔH_O. This differential pressure must not be too high to maintain an adequate cross-section at the valve, thus reducing risks of clogging and noises. This differential pressure must not be too low either, in which case the relative influence of pressure drops in circuit pipes cannot be neglected. Therefore, the differential pressure ΔH_O is generally chosen between 8 and 10 kPa.

Each adjustable thermostatic valve is then preset based on this selected differential pressure ΔH_o. When the balancing valve STAD on the branch is adjusted to obtain a total flow corresponding to the sum of the flows in the radiators, the preliminary settings made are justified. The selected differential pressure ΔH_o is then applied across the hydraulic centre of the circuit. In practice, the first radiator will be in slight overflow and the last radiator will be in slight underflow. These differences depend on the circuit length and on the pressure drops in the pipes and accessories.

Example: A circuit with radiators, each having a design flow of 50 l/h. The pressure drop in the pipes is 2 kPa. Consider ΔH_o = 8 kPa.

Flow in the first radiator is $= 50 \times \sqrt{\frac{8+1}{8}} = 53$ l/h, and in the last =
$50 \times \sqrt{\frac{8-1}{8}} = 47$ l/h

The deviation is ± 6%.
Consider now ΔH_o = 2 kPa and the same pressure drop in the pipes.

Flow in the first radiator is $= 50 \times \sqrt{\frac{2+1}{2}} = 61$ l/h, and in the last =
$50 \times \sqrt{\frac{2-1}{2}} = 35$ l/h

The deviation is –30 to +20%.
This example confirms that ΔH_o should be at least 8 kPa.

4.1.2 PRESETTING THE THERMOSTATIC VALVE

Table 4.1 gives the Kv values to be taken according to the ΔH_O adopted.

Working conditions				Kv valve for ΔHo = 10 kPa
Heat output in (W)		Water flow		
ΔT = 10	ΔT = 20	l/h	l/s	**Kv**
250	500	21.5	0.006	**0.068**
300	600	25.8	0.007	**0.082**
350	700	30.1	0.008	**0.095**
400	800	34.4	0.010	**0.109**
450	900	38.7	0.011	**0.122**
500	1000	43.0	0.012	**0.136**
600	1200	51.6	0.014	**0.163**
700	1400	60.2	0.017	**0.190**
750	1500	64.5	0.018	**0.204**
800	1600	68.8	0.019	**0.218**
900	1800	77.4	0.022	**0.245**
1000	2000	86.0	0.024	**0.272**
1100	2200	94.6	0.026	**0.299**
1200	2400	103.2	0.029	**0.326**
1250	2500	107.5	0.030	**0.340**
1300	2600	111.8	0.031	**0.354**
1400	2800	120.4	0.033	**0.381**
1500	3000	129.0	0.036	**0.408**
1750	3500	150.5	0.042	**0.476**
2000	4000	172.0	0.048	**0.544**
2250	4500	193.5	0.054	**0.612**

Table 4.1. Determining the Kv of a thermostatic valve.

Example: For a 1500 W radiator working with a ΔT of 20K and a differential pressure of 10 kPa, the Kv value of the thermostatic valve must be 0.2. Use the chart in Fig 4.2 to find the Kv value graphically.

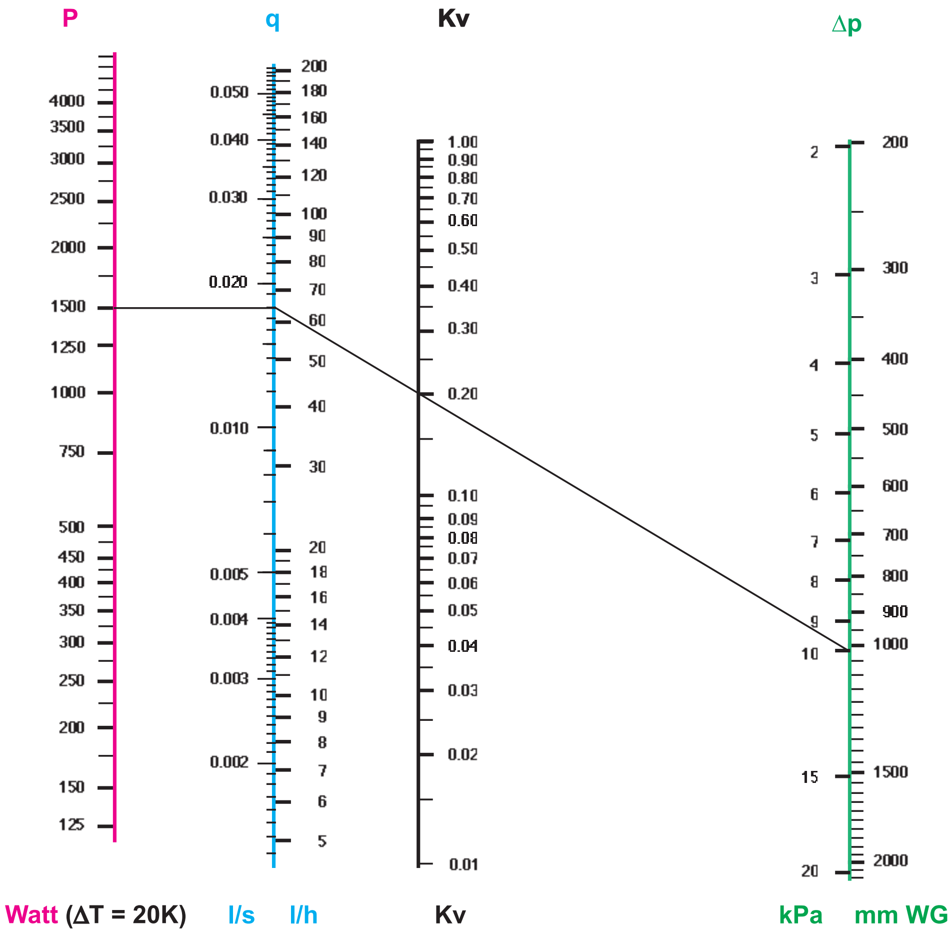


Fig 4.2. Determining the Kv of a thermostatic valve.

For a radiator of 1500 W, the water flow = 64.5 l/h. For a Δp of 10 kPa, Kv = 0.2

4.1.3 NON-PRESETTABLE THERMOSTATIC VALVES

When a thermostatic valve is non-presettable, the adjustment will be made on the return valve. As the thermostatic valve already creates some pressure drop at nominal opening, only the rest of the available Δp is applied across the return valve.

Example: A 2000 W radiator working with a $\Delta T = 20K$ is supplied at a differential pressure of 10 kPa. The thermostatic valve has a Kv = 0.5. What Kv should be set at the return valve?

Referring to Fig 4.2, it can be seen that the radiator flow is 86 l/h. At this flow, the pressure drop in a valve with Kv = 0.5 is 2.96 kPa. The rest is for the return valve: 10 – 3 = 7. Using the same diagram, we find that the Kv must be 0.33 for a flow of 86 l/h and a pressure drop of 7 kPa. If we had neglected the pressure drop in the thermostatic valve, we would have found a Kv of 0.27 for the regulating valve. The flow obtained would have been 75 l/h instead of the predicted 86, representing a deviation of 13%.

The same procedure may be used if pressure drops in other resistances, such as elbows, high resistance radiators, etc., have to be deducted.

4.1.4 LIMITATIONS OF CHOICE WITH THE SAME ΔP FOR ALL RADIATORS

The assumption that the same differential pressure is applied to all radiators has some limits, depending mainly on the required flow accuracy.

Consider the case in Fig 4.3. Valves are preset based on an average differential pressure ΔH_O. The flow will be higher than the design flow at the start of the circuit, and lower at the end. For a deviation of Δq in % of design flow, the maximum allowable length for pipes is determined in Fig 4.3.

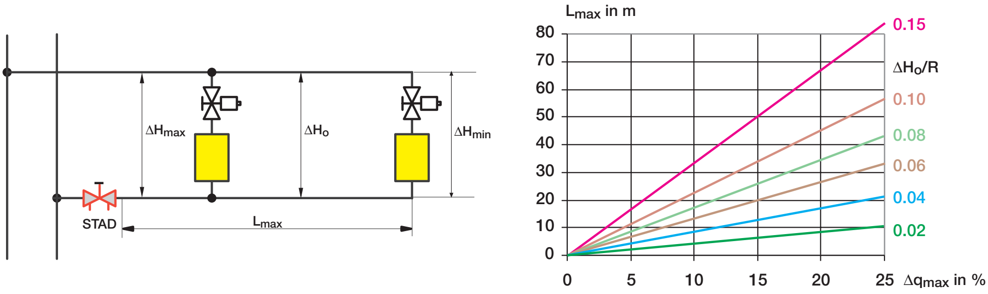


Fig 4.3. When all valves are calculated based on the same $\Delta p = \Delta H_o$, the circuit length should not exceed a given value (ΔH_o in kPa and R (pressure drop in pipes) in Pa/m).

Consider the case of a plant designed for conditions t_{sc} = 80 °C and t_{rc} = 60 °C. A deviation in the room temperature in the order of 1K due to the flow is accepted, which implies a flow precision of $\Delta q = \pm 20\%$. ΔH_o = 10 kPa is adopted, and the pressure drop in the circuit considered equals 100 Pa/m ($\Delta H_o/R = 0.1$). The method described may therefore be used if the distance measured on the pipe between the circuit inlet and last radiator does not exceed 44 metres (See Fig 4.3).

4.2 Presetting based on calculated Δp

If pressure drops in pipes are high, the maximum circuit length is quickly restricted. In this case the differential pressure applied to each radiator must be estimated using the following formula:

$$\Delta H = \Delta H_{max} - \frac{2RL}{1000} = \Delta H_0 + \frac{RL_{max} - 2RL}{1000}$$

ΔH: Differential pressure in kPa available for a thermostatic valve,
R: Pressure drop in pipe in Pa/m,
L: Distance in metres of pipe between the balancing valve of the branch and a radiator.

This ΔH is then calculated, for each radiator, to determine the corresponding Kv.

4.3 Constant or variable primary flow

Primary distribution can be designed for constant or variable flow. This affects the solutions that can be used to obtain the correct differential pressure on the secondary distribution.

Local distribution through thermostatic valves is necessarily a variable flow distribution. However, the primary general distribution may be designed for a constant or a variable water flow.

The advantage of a constant primary flow in the main distribution is that it keeps the pressure drops in the pipes constant. The differential pressure on each circuit is adjusted at the correct value at design condition and does not change with the load. However, the return water temperature is not minimised, which can be a disadvantage in some district heating plants and when condensing boilers are installed.

The advantage of a variable flow in the main distribution is that it minimises the pumping costs and reduces the return water temperature when required. However, at small loads, the differential pressure on the circuits increases according to the reduction of the pressure drops in the pipes and accessories when the flow is reduced.

In all cases, the plant has to be balanced to avoid overflows that create underflows in unfavoured sections and incompatibility problems. For a variable flow distribution, balancing is made for design conditions, which guarantees that all circuits will obtain at least their design flow in all working conditions.

4.3.1 ABOUT NOISE

A hydraulic resistance in a circuit creates a pressure drop and a part of the energy is transformed into heat and another part into noise. The risk of noise increases with the differential pressure.

Some norms define the maximum noise level acceptable in a bedroom to be 30 dBA during the night and 35 dBA during the day.

In a presettable thermostatic valve, the differential pressure is taken in the presetter, which limits the flow at design value, and the control port which adjusts the flow to obtain the required room temperature.

During night setback, the supply water temperature is reduced and the control port is fully open. The noise created by the valve in these conditions comes from the presetter.

The geometry of the valve and particularly the design of the presetter are important in order to obtain a “silent” valve.

All tests realised show that noise increases with the water flow. This is another reason to carefully balance the radiators, avoiding overflows.

The risk of noise could be reduced dramatically during night-time by decreasing the pump head, simultaneously reducing the differential pressure and the water flow. This can be obtained, for instance, with a variable speed pump with different settings during night and day.

During the day, when the thermostatic valve has to compensate for internal heat, the control port partly shuts. The differential pressure applied on the control port increases whilst the water flow decreases. The risk of noise is at maximum when the valve is close to closure. Vibrations occur when the valve is connected the wrong way with the flow of water going in the reverse direction.

Noise in a plant can have many causes. A radiator or convector can amplify noise generated by the pump.

Noise can also increase dramatically when the plant is not well vented. Low water temperatures make it more difficult to vent. Increasing the water temperature during venting procedure can be a solution whenever possible.

A too low static pressure in some parts of the plant should also be avoided as the air separates out of the water in a restriction because of the lower local pressure resulting from the high water velocity.

When a thermostatic valve shuts, the pressure drop in the pipes and restrictions decrease. The differential pressure on the thermostatic valve increases which increases the risk of noise. For this reason, it is not a good idea, in big plants, to adjust the flow with just one restriction in series with the thermostatic valve. If this thermostatic valve shuts, the entire pressure drop taken previously in the restriction is transmitted to the control port. It is much better to take parts of the excess differential pressure in balancing valves in the branches and risers and the rest, 10 kPa for example, in the presetter associated with the thermostatic valves. When one thermostatic valve shuts, the water flow and then the pressure drop in the balancing valves in branches and risers do not change much. Consequently, the differential pressure on the thermostatic valve increases just a little.

However, if all the thermostatic valves shut simultaneously, all pressure drops in pipes and restrictions disappear and the thermostatic valves are submitted to the full pump head. If this happens, the control of the supply water temperature has to be reconsidered. For instance, the supply water temperature can be reduced when the total water flow in the plant decreases.

The situation can be more difficult if the pump is oversized and works with a steep curve increasing the pressure at small loads. For this reason, an adjustable and controlled pump head is generally more convenient. The pump head can also be reduced during most of the time and just put at its maximum value during cold seasons. The reduction of the pump head in warmer seasons is compensated by a small increase of the water supply temperature.

When, in extreme circumstances, the differential pressure exceeds the limit defined by the thermostatic valve manufacturer, the differential pressure has to be limited locally. This question will be examined in the next sections.

4.3.2 CONSTANT PRIMARY FLOW

Fig 4.4 shows two different circuits applicable to an apartment.

In principle, the water temperature of the distribution is modified to suit outdoor conditions. A correct distribution of primary flows is obtained by adjustment of balancing valves STAD.

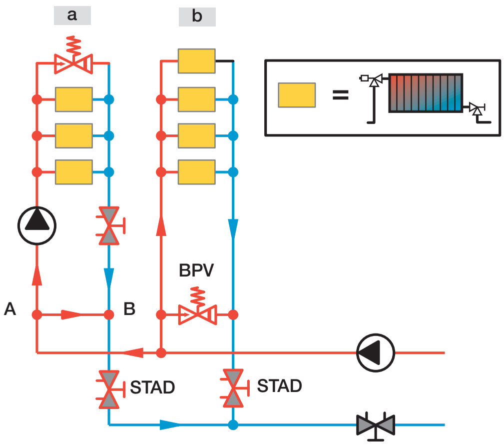


Fig 4.4. Two circuits with radiators are designed to give a constant primary flow.

4.3.2.1 A bypass and a secondary pump minimise the Δp on the branch (Fig 4.4 - circuit a).

This circuit is widely used in some European countries. A bypass pipe AB makes the secondary circuit hydraulically independent of the primary distribution. The high differential pressure in the main distribution network is not transmitted to the circuit. This circuit is provided with a circulating pump which can be controlled by a thermostat located in a reference room. Thermostatic valves are only subject to the relatively low-pressure head of the secondary circulating pump, decreasing the risk of noise considerably.

It is essential that the maximum secondary flow is less than the constant primary flow. Otherwise, at full load, the difference between the two flows will circulate from B to A, creating a mixing point at A. In this case, the supply water temperature will be lower than the design value and comfort is not guaranteed.

A BPV (proportional relief valve) may be installed at the end of the circuit and set at 10 kPa for example.

The principle application of this BPV is to be closed except when the flow through the thermostatic valves drops below a certain value, thereby securing the following:

– A limitation of the maximum Δp on the thermostatic valves.

– A minimum flow for protection of the circuit pump.

– A prevention of large water temperature drops in the pipes. This is the main reason, in this case, to install the BPV at the end of the circuit instead of in the beginning.

4.3.2.2 A BPV stabilises the Δp on the branch (Fig 4.4 circuit b).

A proportional relief valve BPV is placed at the circuit inlet. It gradually opens when the differential pressure across it reaches its set point. Radiator valves have been set based on a given differential pressure, for example 10 kPa.

The BPV is kept shut throughout the balancing procedure.

When balancing is complete, with thermostatic valves open, the BPV set value is reduced until it starts to open. This causes an increase of flow, which can be measured at STAD. The BPV set point is then increased until it closes again. In some plants, the BPVs are set to obtain a small flow at design condition; this greatly reduces the circulation noises in the plant. The explanation for this is related to the pressure waves generated by the pump, which are bypassed by the BPV.

During normal operation, whenever some thermostatic valves close, the pressure drop across STAD is reduced and the differential pressure applied to the BPV increased. The BPV then opens to maintain this differential pressure at its set value. The total primary flow remains practically constant for a constant ΔH.

Note that this function is obtained by the combination of the BPV and STAD. Both elements are essential to keep the total flow and the differential pressure across the circuit constant. The BPV allows a certain supplementary flow through STAD that creates a supplementary pressure drop to compensate an eventual increase of the primary differential pressure ΔH. Without the STAD, the BPV is not operative.

This distribution method is more efficient than the method that uses a secondary pump as shown in the circuit a. The secondary pump is eliminated. The protection against low flows is no longer necessary and the secondary balancing valve is eliminated. Finally, this pump head is chosen according to need and is maintained constant.

4.3.3 VARIABLE PRIMARY FLOW

In order to minimise return water temperatures, a variable flow distribution has to be adopted. This is often essential when the plant is connected to a district heating distribution. Two examples are shown in Fig 4.5.

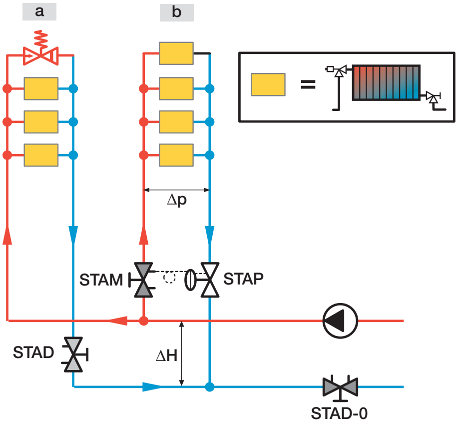


Fig 4.5. Two circuits supplied at variable primary flow.

4.3.3.1 A plant with balancing valves (Fig 4.5 – circuit a).

This is the classic case of a branch or a small riser connected to a main network. Thermostatic valves are preset for a given differential pressure, for example, 10 kPa. The balancing valve STAD is used to obtain the total flow in circuit "a", which at design condition gives the selected differential pressure of 10 kPa at the hydraulic centre of gravity of the circuit, and more than 10 kPa at other loads.

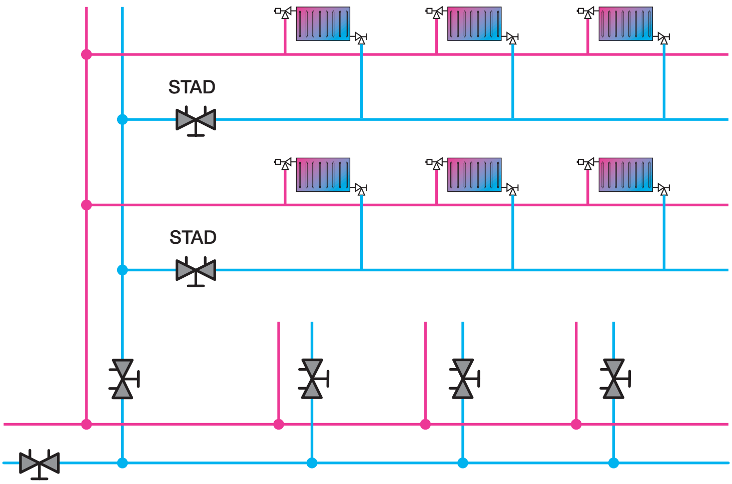


Fig 4.6 The installation is balanced using the TA method.

The installation is balanced using the TA method, with all thermostatic valves open.

Excess differential pressures are mostly resisted in valves on the risers. When a thermostatic valve is closed, the differential pressure across the branch increases only slightly because proportionally this has little effect on the flow in the branch and riser balancing valves. The applied differential pressure becomes equal to the pump head only if all thermostatic valves are closed simultaneously. This situation should however not occur if the supply water temperature is controlled correctly.

If the differential pressure on the thermostatic valves exceeds 30 kPa, the thermostatic valves may become noisy. This problem can be solved by using a BPV at the end of the circuit. This BPV starts to open when the differential pressure exceeds 30 kPa, creating at the same time the minimum flow required to protect the main pump. This minimum flow is also required to avoid too large water temperature drops in pipes, which occur below a certain flow.

4.3.3.2 A Δp controller keeps the Δp constant across a branch (Fig 4.5 circuit b and Fig 4.7).

a- with presettable radiator valves

Differential pressures in large networks are often high, particularly close to the distribution pump. The differential pressure has to be reduced and stabilised to a reasonable value, of 10 kPa for example, to supply each radiator circuit. This reduction is obtained by a self-acting differential pressure controller “STAP”.

It is necessary to have a measuring valve STAM (or STAD) to measure the flow and, if necessary, adjust the set point of the Δp controller to obtain the required branch flow at design condition. Furthermore, this measuring valve is used for isolation and as a diagnostic tool.

The maximum flow in each radiator must always be adjusted to its design value. If balancing is not done, overflows, especially at start-up, make it impossible to obtain a correct distribution of power and the required supply water temperature.

With only Δp controllers, the minimum flow necessary to protect the pump is not generated. This minimum flow has to be created close to the most remote circuits to also obtain this minimum flow in the pipes, avoiding too high a water temperature drop. This minimum flow may be created by some circuits working with constant primary flow (Fig 4.4).

The set point of the Δp controller.

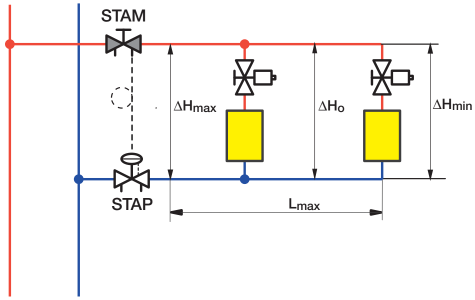


Fig 4.7. A controller stabilises the differential pressure at the circuit inlet.

Let us consider a plant with 4 identical radiators, with a distance of 10 m between each one. Pressure drops in the pipes are of 100 Pa/m. Presettings have been based on a uniform Δp of 10 kPa. We can compare the results when we maintain 10 kPa at the inlet of the branch (Fig 4.8) or if the set point of the Δp controller is adjusted to obtain the correct design flow in the branch (Fig 4.9).

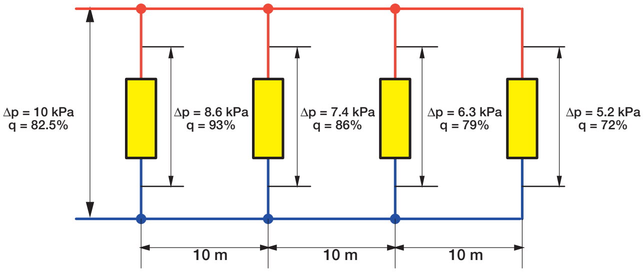


Fig 4.8. Δp = 10 kPa at the inlet of the branch.

In the case of Fig 4.8, all radiators are in underflow. The deviation is then between –7 and –28%, which is normally not acceptable.

With the differential pressure controller STAP, the set point is adjustable. The measuring valve STAM is used to measure and verify the flow and is set to obtain a pressure drop of approximately 3 kPa for design flow. The set value of the differential pressure controller is then chosen to obtain the required flow measurable at the STAM.

In doing this, the set value of the differential pressure controller complies with the adopted preliminary settings.

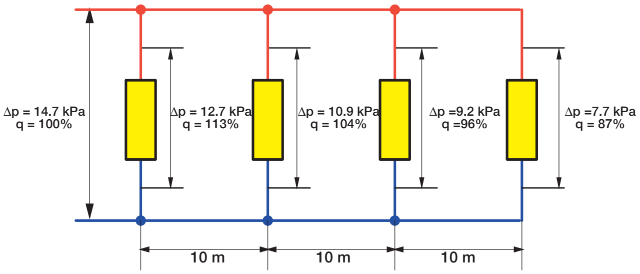


Fig 4.9. The set point of the Δp controller is adjusted to obtain the correct design flow in the branch.

In the example of Fig 4.9, the water flows in the radiators are obtained with a deviation of ± 13%, which is normally acceptable.

b- with non-presettable radiator valves

In some old buildings, the radiator valves are non-presettable and will not be replaced. In this case, it can be sufficient to limit the total flow for each branch. This is conceivable if the radiators are not too different and if the pressure drops in the pipes are small.

The circuit adopted is represented in figure 4.10.

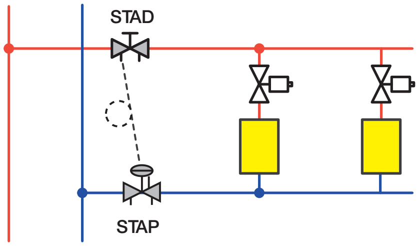


Fig 4.10. The pressure drop in the balancing valve is included in the total Δp controlled by the STAP.

The set point of the STAP is chosen = 14 kPa. The balancing valve STAD is preset for a pressure drop of 11 kPa at design flow.

During start-up, when all thermostatic valves are fully open, the total flow in the branch cannot exceed the design flow by more than 13%. If we consider the extreme case of a branch without any hydronic resistance, all the available Δp of 14 kPa has to be taken in the STAD. That means that the flow in the STAD will be:

$$q = 100 \times \sqrt{\frac{14}{11}} = 113\%$$

If all the thermostatic valves are shut, the pressure drop in the STAD = 0 but the available Δp on the thermostatic valves is limited by the STAP to 14 kPa.

This combination guarantees that the flow and the Δp are limited to the correct values.

When the thermostatic valves are working at design flow, the available Δp = 14 – 11 = 3 kPa.

Other values can be chosen for the set point of the STAP and the presetting of the STAD. If it seems better to obtain an available Δp of 4 kPa for design flow, the STAP is set on 15 kPa instead of 14 kPa, for instance. However the values suggested cover most existing plants.

This is confirmed by figure 4.11

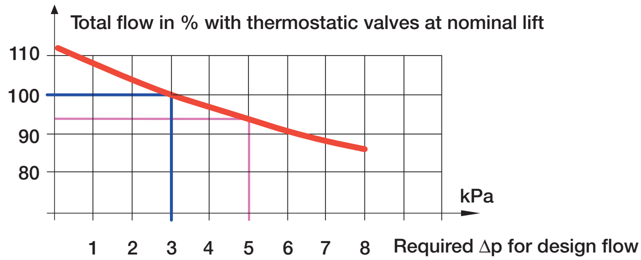


Fig 4.11. If the required Δp at design condition is 5 kPa instead of the 3 kPa expected, the deviation in flow is only 7%.

5. One-pipe distribution

5.1 General

In a one-pipe distribution, radiators are connected in series. Each radiator valve splits the flow into one part being bypassed and one part going through the radiator. The water leaving one radiator valve enters the next one as shown in Fig 5.1.

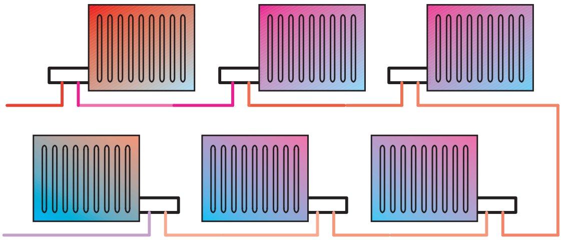


Fig 5.1. Radiators with one-pipe distribution.

If the entire loop flow passes through the radiators, isolation of a single radiator will stop circulation in all radiators in the loop. This is why the bypass is installed at each one-pipe valve as shown in Fig 5.2.

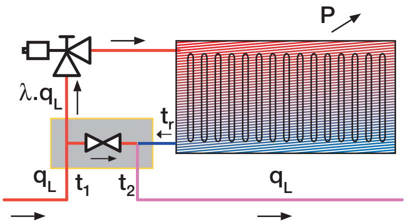


Fig 5.2. A bypass diverts part of the loop flow.

P = Heat emission in W.
q_L = Loop flow in l/h.
λq_L = Water flow in the radiator.
t_1 = Inlet water temperature.
t_r = Outlet water temperature from the radiator.
t_2 = Inlet water temperature for the next radiator.

The differential pressure created by the restriction in the bypass generates the flow through the radiator. This flow is limited by the thermostatic valve.

If P is the heat output of the radiator, the supply water temperature t_2 for the next radiator is calculated as follows:

$$t_2 = t_1 - \frac{0.86 \times P}{q_L}$$

The water temperature in the loop decreases after each radiator; this must be taken into account when choosing nominal radiator power.

5.1.1 ADVANTAGES

– *Reduced pipe lengths.*
When radiators can be spread out throughout the entire loop, the pipe length can be reduced by up to 50%. This also reduces heat emission from pipes, which cannot be controlled by thermostatic valves.

– *Lower labour cost.*
Installation is very fast when pipes are laid out in loops. Many accessories such as tees and elbows are eliminated. The labour saving compared with a traditional two-pipe installation can exceed 40%.

– *More reliable installation.*
Pipes are usually made of copper or soft steel and protected by plastic. The use of cross-linked polyethylene tubes is also becoming quite common. All of these pipes are well protected against corrosion and can be laid in a single operation without any connection in the concrete, making the installation more reliable in the long term.

– *Distribution with practically constant flow.*
Since the flow in the loop is almost constant, the various loops are not interactive.

– *The pipes emission can be deducted from the heat losses to calculate the radiators.*
This advantage will be discussed in section 5.1.3.

5.1.2 DISADVANTAGES AND LIMITATIONS

– *Increased total surface of heating elements.*

The radiators at the end of the loop must be oversized to compensate for the lower water temperature. The total surface of heating elements installed in a one-pipe distribution is therefore sometimes greater than in a two-pipe distribution system.

– *The return water temperature may be higher than in two-pipe systems.*

One-pipe loops have a practically constant flow. When all thermostatic valves are closed, the temperature of the return water is equal to the supply temperature. District heating companies require the lowest possible return temperatures, and therefore do not like one-pipe distributions. This comment should be kept in proportion, since the supply water temperature normally depends on outside conditions. In this case it is only accidental if most thermostatic valves are closed. However, it frequently happens that one-pipe loops work with a lower ΔT than two-pipe loops in order to reduce heating surfaces to be installed. In this case the return is effectively warmer.

– *Interactivity between radiators in the loop.*

Let us consider a loop with four radiators. When closing the first two, the water temperature on the last two will increase. The thermostatic valves on these radiators then have to compensate for a potential increase in emission in the region of 10 to 15%. However closing the last two radiators affects the loop flow by reducing the power of the first two radiators by around 3 to 5%. These interactivity phenomena do not create a real problem and depend on the ΔT used in the loop and the proportion of the loop flow absorbed by the radiators.

When the flow in the radiators is not balanced, the power emitted by the first radiators when the installation starts up may be higher than planned. In this case, the supply water temperature to the last radiators in the loop is too low to be able to supply the required power. It is therefore important to balance the loop by providing the necessary flow to each radiator.

5.1.3 EMISSION FROM PIPES

Connecting pipes emit heat in addition to heat emitted by radiators. This emission is normally ignored when selecting heating elements, but must be considered in calculating the water temperature drop in the loop.

Emission from pipes in the environment depends on the water temperature, the pipe diameter, its degree of insulation, and the pipe location (visible, cast in concrete, etc.).

For a water temperature of 80 °C and a room temperature of 20 °C, the heat losses of a visible plastic tube are of 30 W/m (d_i = 10 mm) and 60 W/m for d_i = 20 mm. For a diameter d_i between 10 and 20 mm, the heat emitted per metre from a visible pipe can be estimated using the following formula:

$$P = (t_s - t_i) \times \frac{d_i}{15}$$

P = Heat losses in W/m.

t_s = The supply water temperature.

t_i = The room temperature.

d_i = Pipe diameter (mm).

If each radiator is connected with 6 metres of pipe (d_i = 15 mm), emission from pipes at 80 °C would theoretically reduce the required heat output for each radiator by 270 W. This relative effect is quite significant for small radiators, but is often ignored in the calculations. Some well-insulated installations provide adequate heating throughout the entire heating season with all radiators closed as the pipes alone provide sufficient heat. In some countries, all connecting pipes, even those inside the apartment, must be thoroughly insulated. This gives better control of the heating power.

When a thermostatic valve closes, emission from the pipe continues and may make the room too warm. However this is limited since the water temperature supply depends on outside conditions. In some very well-insulated installations, a control valve is placed in the loop and isolates this loop when the temperature in a reference room exceeds a limiting value, for example 22 °C.

The problem of uncontrolled emission from pipes is not inherent to the one-pipe system. In a two-pipe distribution as shown in Fig 5.3, this emission is even increased by the presence of two distribution pipes instead of one.

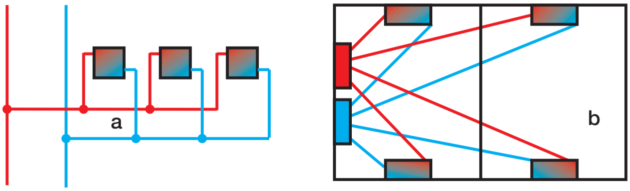


Fig 5.3. Emission from pipes in a two-pipe distribution.

When each radiator is individually connected to the distributors, the closure of a thermostatic valve stops the pipe emission into the room and into the adjacent room where this action is in fact a disturbance.

In a two-pipe distribution as shown in Fig 5.4, the emission of pipes in a room is variable and depends mainly on the room temperature control in other rooms. For this reason, the pipe emission cannot be deducted from the heat losses when calculating the radiators. However, pipe emission must be calculated to determine the real supply water temperature for each radiator.

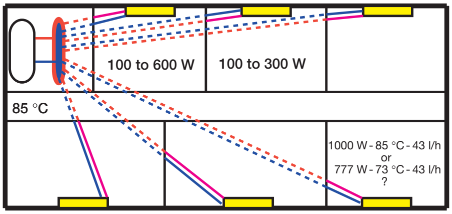


Fig 5.4. In a two-pipe system, the pipes emission in a room depends mainly on other rooms.

The opposite is true for a one-pipe distribution (Fig 5.5); the pipes emission does not influence the function of the radiators in practice. Consequently, this emission can be deducted from the heat losses to calculate the radiators.

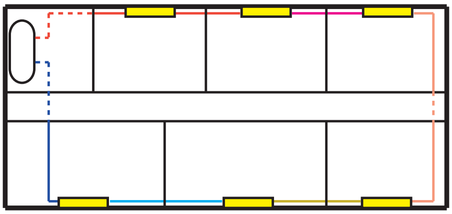


Fig 5.5. In a one-pipe distribution the pipes emission can be deducted from the heat losses to calculate the radiators.

5.2 One-pipe valves

5.2.1 CONSTANT BYPASS – VARIABLE KV

Several methods of installation are possible.

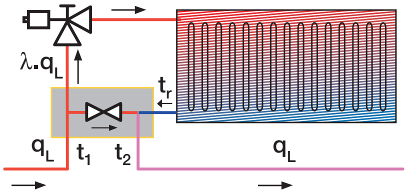


Fig 5.6. Distributor with fixed bypass.

In the case shown in Fig 5.6, the resistance of the bypass is fixed and the proportion of the loop flow passing through the radiator is obtained by reducing the Kv_{max} of the thermostatic valve. For a small radiator, most of the flow passes through the bypass, and the pressure loss in the bypass may be unnecessarily high. In this case, the Kv of the bypass should be increased, and therefore a variable bypass valve should be used.

5.2.2 VARIABLE BYPASS – CONSTANT KV

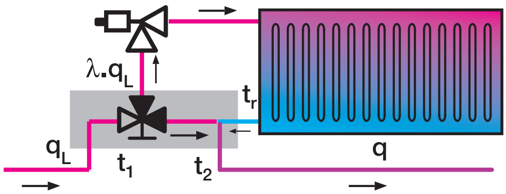


Fig 5.7. Distributor with variable bypass.

In Fig 5.7, a three-way diverting valve distributes flows between the radiator and the bypass at constant total Kv. The pressure loss across the module only depends on the flow in the loop. The adjustment of the flow through a radiator, with the bypass three-way valve, has no influence on the flow loop.

5.2.3 PROTECTION AGAINST DOUBLE CIRCULATION

When the thermostatic valve closes, the bottom of the radiator remains in direct contact with the hot water pipe. This can create double circulation in the return orifice (Fig 5.8a) with uncontrollable emission from the radiator. This is why a tube should be inserted in the radiator as shown in Fig 5.8b, in order to prevent the double circulation phenomenon. This inserted tube cannot be placed on all radiators.

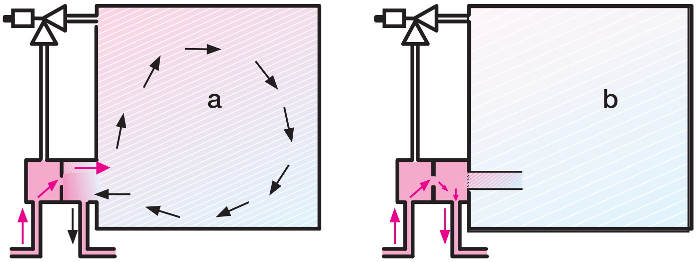


Fig 5.8. With the thermostatic valve closed, double circulation can take place in the radiator return connection. An inserted tube can practically eliminate this phenomenon.

5.3 Proportion of the loop flow in the radiator (λ coefficient)

5.3.1 50% FLOW IN THE RADIATOR (λ = 0.5)

In early versions of the one-pipe system, 50% of the loop flow was designed for the radiator. This magic figure results from the fact that the temperature t_2 of the water supply to the next radiator is equal to the arithmetic mean temperature of the previous radiator. As we saw under Fig 5.2, t_2 is very easily calculated. Based on the average temperature of the radiator, it was then possible to calculate the nominal power to be installed to obtain the necessary emission.

The high overflow in the radiator was considered to improve its emission, and thus make it possible to reduce heating surface areas to be installed. This is not always true as too high a velocity in the inlet creates a suction effect. Cold water in the bottom of the radiator is mixed with the hot water in the inlet decreasing, to a certain extent, the heat output. However, passing 50% of a loop flow of 500 l/h in a 250 W radiator is equivalent to 23 times its nominal flow. Reducing the flow using the thermostatic valve has practically no effect on emission for 96% (22/23) of its nominal lift. Therefore the valve usually works near its closed position, with the risk of being inefficient.

Moreover, when a thermostatic valve closes, the entire flow must pass through the bypass. Therefore the flow in the bypass is doubled, and the pressure loss is quadrupled. This high pressure loss affects the total flow and the emission from the radiators in the loop. To ensure that this will not happen, a lower proportion of the flow can be taken in the radiator. Flow variations in the bypass are then lower and the loop flow is more stable.

5.3.2 CHOICE OF ANOTHER FLOW IN THE RADIATOR

For loop flows of up to 200 l/h, the TA-RSD 801 one-pipe valve has a Kv = 1.2 and the flow in the radiator can be adjusted from 0 to 50% of the loop flow.

For loop flows of more than 200 l/h, the Kv of the TA-RSD 831 is 2.8 and the flow in the radiator is adjustable from 0 to 20% of the loop flow.

In both cases, the Kv of the TA-RSD one-pipe valve is independent of the flow proportion chosen in the radiator.

In order to estimate its required nominal power, the maximum possible flow is generally assumed in the radiator with a minimum value of 10K for the ΔT.

The radiator is selected on this basis but, since the choice is limited, its real nominal power is generally higher than the calculated power. The real flow necessary to obtain the calculated heating power is then determined.

Example: In a 400 l/h loop, a radiator is supplied at a water temperature ts of 82 °C and must emit 850 W into an environment of 20 °C.

Preliminary calculation: 20% of 400 l/h = 80 l/h. For a ΔT of 10K, the flow must be 0.86 x 850/10 = 73 l/h. The lowest of these two flows is used, namely 73 l/h and the ΔT is therefore 10K.

Using the diagram in Fig A1 (in appendix A), it is found that the nominal 75/65 oversizing factor of the radiator is 0.84 (Join $t_s - t_i = 82 - 20 = 62$ to $t_r - t_i = 72 - 20 = 52$). The nominal power 75/65 of the radiator to be installed is therefore $850 \times 0.84 = 716$ W.

Final calculation: The real nominal power of the chosen radiator is 935 W. What flow should be adjusted at the radiator?

The oversizing factor Sp is therefore 935/850 = 1.1.

Using Fig A1 again, join $(t_s - t_i) = (82 - 20) = 62$ to Sp = 1.1 to get $t_r - t_i = 34.5$ therefore $t_r = 54.5$.

The required ΔT is therefore 82 – 54.5 = 27.5K and the flow $q = 0.86 \times 850/27.5 = 26.6$ l/h, namely 7% of the loop flow.

When an installation is being renovated, the nominal power of radiators is given. The flow in the radiator is therefore determined based on the diagram in Fig A1 directly, using the oversizing factor and the supply water temperature. In some new buildings, for aesthetic reasons, all radiators are identical and the correct power has to be obtained by appropriate adjustment of the flow to each radiator.

5.4 The loop flow

In theory, it is better to use the highest possible loop flow. This reduces the temperature drop in the loop and the required heating surface area for the last radiators. Moreover, when thermostatic valves close on the first radiators in the loop, there is not a high increase in the water temperature for the last radiators.

In practice, the loop flow is limited by the available differential pressure in relation to the pipe size, the number of radiators etc.

The total loop power and the supply water temperature are the predominant factors to determine the realistic value for the loop flow (see section 5.4.1). In some cases, when $\lambda_{max} = 0.2$, a large radiator in the loop can be the predominant factor (see section 5.4.2).

5.4.1 BASED ON A GIVEN ΔT

The loop flow is a result of the chosen ΔT and the sum ΣP of powers, emitted by radiators in the loop, according to the following formula:

$$q_L > \frac{0.86 \times \Sigma P}{\Delta T_L}$$

q_L = Loop flow in l/h.
ΣP = Sum of the required heat output in W of the radiators in the loop.
ΔT_L = Temperature drop in the loop (K).

Obviously, the allowable temperature drop reduces with the supply water temperature. The temperature drop in the loop is normally given by $\Delta T_L < 0.25 \times (t_S - 20)$. Substituting this value in the previous formula gives an estimation of the lowest realistic loop flow:

$$q_L > \frac{3.44 \times \Sigma P}{(t_S - 20)}$$

Example: If $\Sigma P = 4\,000$ W and $t_S = 80$ °C, $q_L > 230$ l/h.

5.4.2 BASED ON THE LARGEST RADIATOR IN THE LOOP (WHEN $\lambda_{max} = 0.2$)

The largest radiator in the loop, PM, requires a minimum flow to avoid unreasonable oversizing. Using an effectiveness $\phi = 0{,}43 = \Delta T_R/(t_S - t_i)$ for example: $q_R = 0{,}2 \times q_L = P_M \times 0{,}86/\Delta T_R$ and

$$q_L = \frac{10 \times P_M}{t_S - 20}$$

5.4.3 FINAL CHOICE OF THE LOOP FLOW

The highest water flow from the two methods of calculation is chosen.

5.5 Pressure losses in the loop

For a flow of 300 l/h in a pipe with an inside diameter of 14 mm, the pressure loss for water at 20 °C is 317 Pa/m (32 mm/m) and 278 Pa/m at 70 °C (Nomogram B1 in Appendix B)

One-pipe valves may be converted into equivalent metres of pipe as shown in table 5.1 below.

d_i mm	10	11	12	13	14	15	16	17	18
Kv=1.2	3.79	6.23	9.80	14.87	21.88				
Kv=2.8	0.69	1.14	1.80	2.73	4.02	5.76	8.06	11.05	14.88
1 elbow	0.24	0.26	0.29	0.32	0.35	0.39	0.42	0.45	0.48

Table 5.1. Metres of equivalent pipe for valves with Kv =1.2 and 2.8.
Equivalence for a single elbow. Water temperature 70 °C.

That means, for instance, that a valve of Kv = 2.8 with two elbows has a pressure drop equivalent to 4.72 metres of pipe (di = 14 mm).

Appendices

A. Calculation of radiators in several conditions

t_s = Supply water temperature
t_r = Return water temperature
t_i = Romm temperature required
P = Heat output required
P_n = Nominal heat output of radiator

Nominal conditions
t_{sn} = 90 °C t_{rn} = 70 °C t_{in} = 20 °C
t_{sn} = 80 °C t_{rn} = 60 °C t_{in} = 20 °C

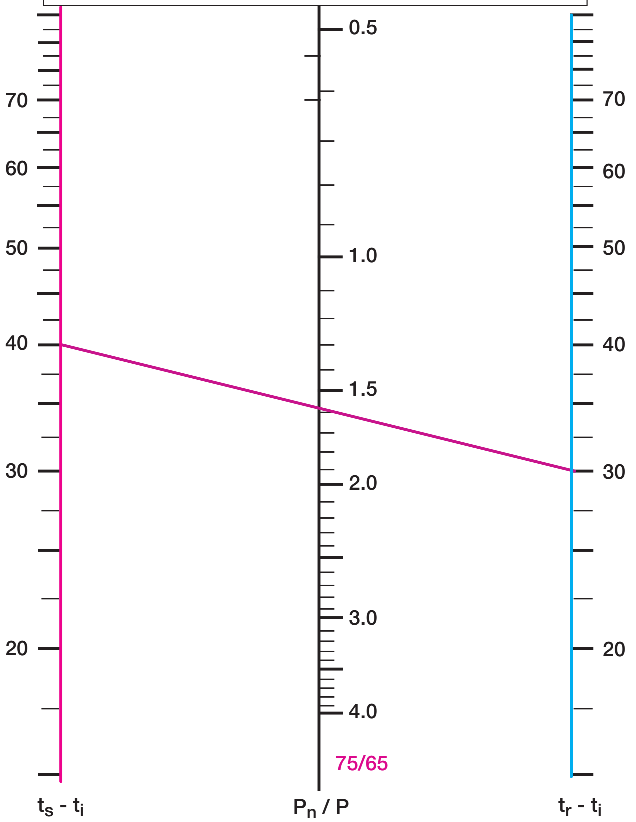


Fig A1. The power of a radiator not working in nominal condition (n = 1.3).

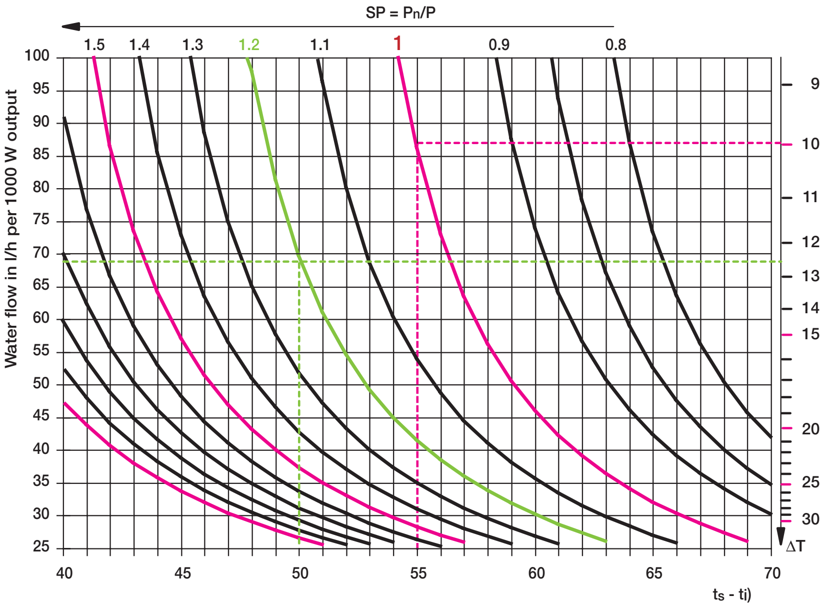


Fig A2. Water flow in l/h per 1000 W is a function of the oversizing factor Sp = P_n/P and the supply water temperature t_s (nominal condition t_{sn} = 75 °C, t_{rn} = 65 °C).

Examples: (using Fig A1 and nominal conditions 75/65)

1. The heat losses in a room are 1000 W. t_s = 60 °C and t_r = 50 °C. What nominal radiator power should be installed if the required room temperature is 20 °C?

   Join $t_s - t_i = 60 - 20 = 40$ to point $t_r - t_i = 50 - 20 = 30$, to find $P_n/P = 1.6$. We therefore need to install a 1000 × 1.6 = 1600 W radiator (75/65 conditions) to obtain 1000 W (60/50 conditions).

   The required flow is $P \times 0.86/\Delta T_c = 1000 \times 0.86/10 = 86$ l/h.

   The following formula can also be used:

$$SP = \left(\frac{2475}{(t_s - t_i)\ (t_r - t_i)}\right)^{n/2} = \left(\frac{2475}{(60-20)\ (50-20)}\right)^{1.33/2} = 1.6$$

2. 750 W must be transferred with a radiator supplied at 70 °C. The flow is 43 l/h. What nominal capacity should be installed if the room temperature to be obtained is 22 °C?

   $\Delta T = 0.86 \times 750/43 = 15$K therefore $t_r = t_s - \Delta T = 70 - 15 = 55$ °C. Join $t_s - t_i = 70 - 22 = 48$ to point $t_r - t_i = 55 - 22 = 33$, to find P_n/P = 1,33. We therefore have to install a 750 × 1.33 = 948 W radiator.

3. A radiator has a nominal capacity of 1250 W, whereas losses are only 1000 W. The water supply temperature is 80 °C and the room temperature is 20 °C. What should the water flow be to compensate for this oversizing?

   Join $t_s - t_i = 80 - 20 = 60$ to $P_n/P = 1250/1000 = 1{,}25$, to find $t_r - t_i = 29$. Therefore $t_r = 29 + 20 = 49$ °C.

   The water temperature drop must be $t_r - t_s = 80 - 49 = 31K$ 31K and the water flow to be set $0.86 \times 1000/31 = 28$ l/h, whereas the nominal flow through a 1250 W radiator is $0.86 \times 1250/10 = 108$ l/h.

   For computer, following formula can also be used:

$$q = \frac{0.86 \times P}{(t_s - t_i) - \frac{2475}{(t_s - t_i)} \times Sp_n^{\,-2/n}} = \frac{0.86 \times 1000}{(80 - 20) - \frac{2475}{(80 - 20)} \times 1.25^{-2/1.33}} = 48 \text{ l/h}$$

   This formula has been translated in the graph of figure A2.

4. On a radiator with a nominal capacity of 1000 W (75/65), we measure a water inlet temperature ts of 55 °C and a return temperature t_r of 50 °C. The room temperature t_i is 22.5 °C for an outdoor temperature $t_e = 3$ °C.

   4.1 What is the present heat transfer of the radiator?
   Join $(t_s - t_i) = 55 - 22.5 = 32.5$ °C to $(t_r - t_i) = 50 - 22.5 = 27.5$ °C.
   Then $P_n/P = 1.94$. Therefore $P = 1000/1.94 = 515$ W.

   4.2 What is the present flow compared with the nominal flow?
   Nominal flow $= 0.86\ P_n/\Delta T_n = 0.86 \times 1000/10 = 86$ l/h.
   Present real flow $= 0.86 \times 515/(55 - 50) = 88.6$ l/h.

   4.3 What would the heat losses be for $t_e = 3$ °C if the room temperature was 20 °C?
   Present heat losses $= 515 = k\ (t_i - t_e) = k\ (22.5 - 3)$ therefore $k = 26.4$.
   Losses for $t_i = 20$ °C ger $k\ (20 - 3) = 26.4\ (20 - 3) = 449$ W.

   4.4 What should the return temperature t_r and the water flow q be, to obtain a room temperature of 20 °C?
   For $t_i = 20$ °C, $P_n/P = 1000/449 = 2.23$. Join $(t_s - t_i) = (55 - 20) = 35$ to $P_n/P = 2.23$. To find $(t_r - t_i) = (t_r - 20) = 20.6$ °C.
   Therefore $t_r = 20 + 20.6 = 40.6$ °C and $\Delta T = 55 - 40.6 = 14.4K$.
   To obtain these conditions, the water flow must be: $0.86\ P/\Delta T = 0.86 \times 449/14.4 = 3.2$ l/h.

   4.5 What nominal radiator capacity would have been necessary to work under nominal conditions, if the outdoor design temperature $t_{ec} = -10$ °C?
   Losses at $t_{ec} = -10$ °C: $k\ (20 - (-10)) = 26.4 \times 30 = 792$ W.
   The installed radiator capacity should have been 792 W with a flow of $0.86 \times 792/10 = 68$ l/h.

B. Pressure losses in pipes.

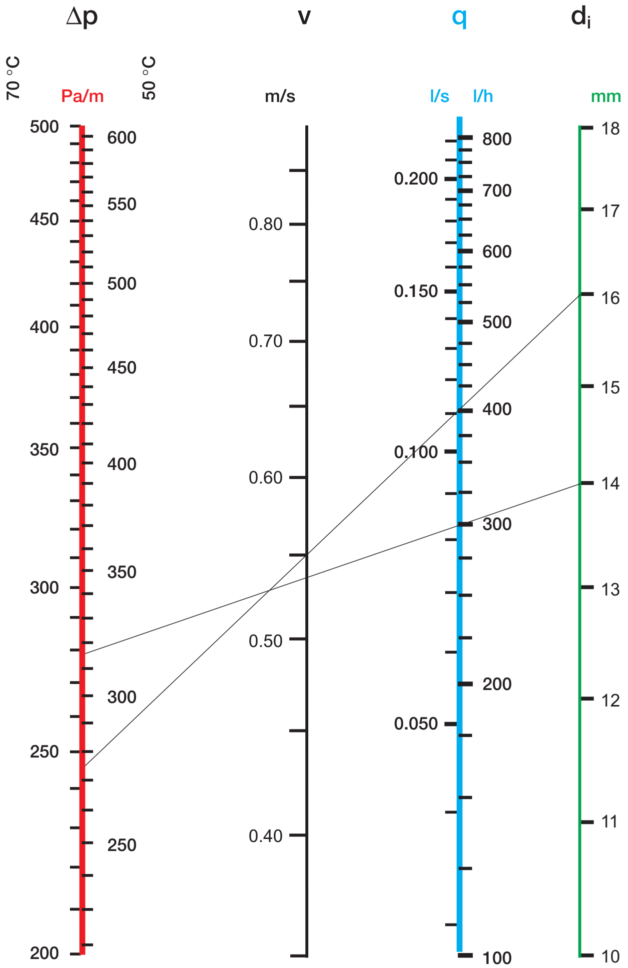


Fig B1a. Pressure losses in pipes with roughness less than 0.0045 (smooth steel, copper, polyethylene, etc.)- di is the inside diameter of the pipe in mm.

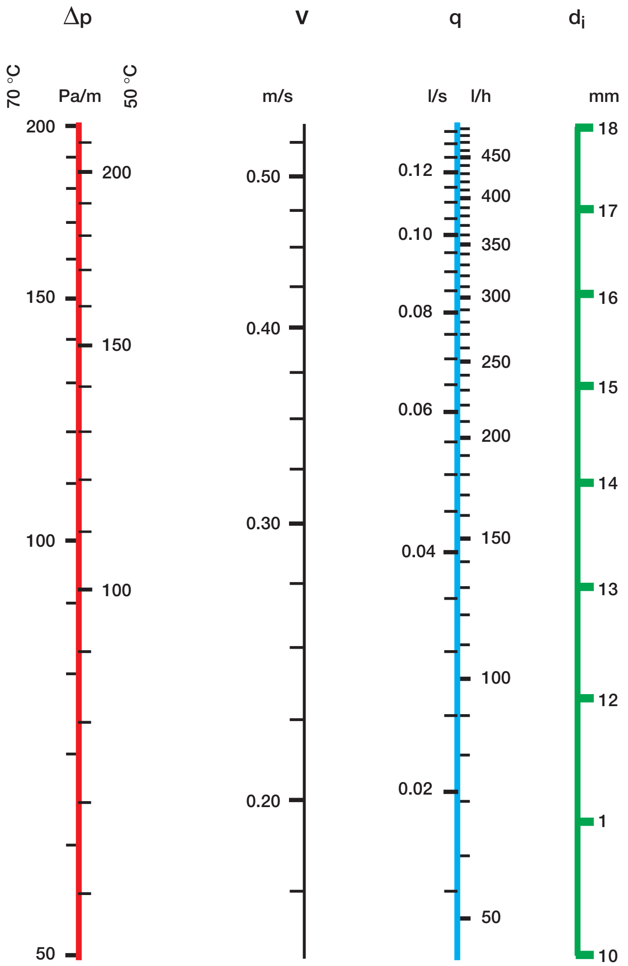


Fig B1b. Pressure losses in pipes with roughness less than 0.0045 (smooth steel, copper, polyethylene, etc.)- di is the inside diameter of the pipe in mm.

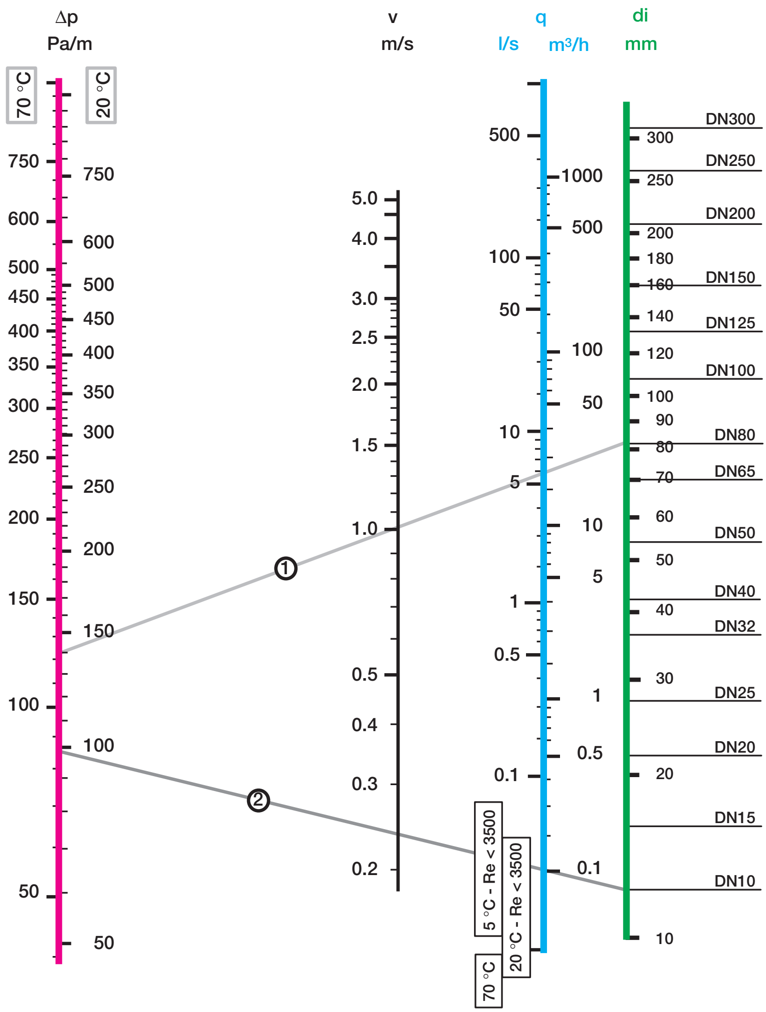


Fig B2. Pressure drops and velocities (pipes with a roughness of 0.05 mm) for water at 20 and 70 °C.

Example 1: Pipe DN80 (di = 82.5 mm) and water flow 20 m^3/h: Velocity = 1 m/s and Δp = 140 Pa/m (at 20 °C).

Example 2: Pipe DN10 (di = 12.5 mm) and water flow 0.1 m^3/h: Reynolds number being below 3500, this nomogram is not applicable in this case, as it was established for turbulent conditions.

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