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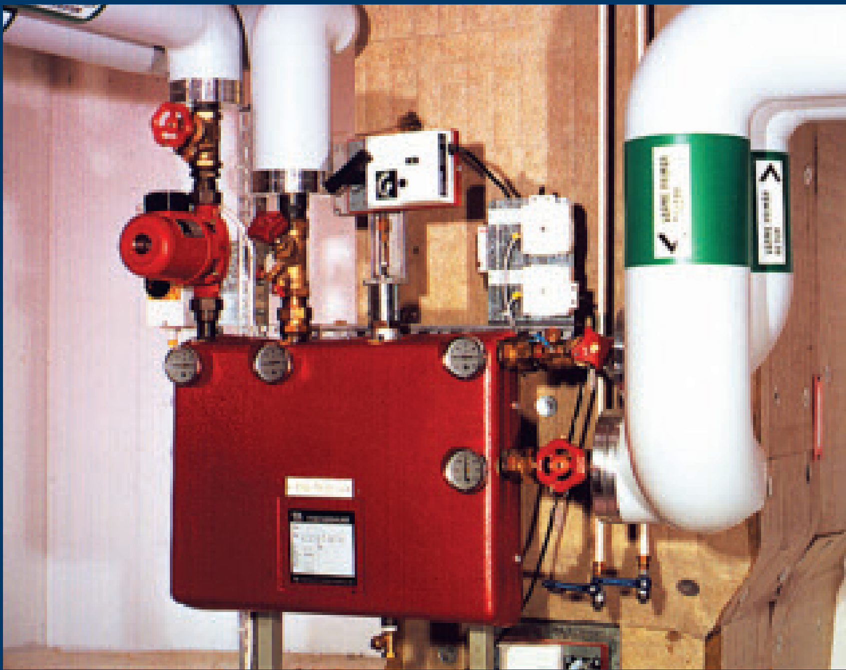

BALANCING OF CONTROL LOOPS

A manual for getting the correct function of 23 control loops used in hydronic heating and cooling systems.

Casselden Place, Melbourne, Australia

Balancing of Control Loops is No. 1 in the TA Hydronics series of publications for HVAC practitioners. Manual No. 2 deals with balancing distribution systems. Manual No. 3 deals with balancing radiator systems and manual No 4 deals with Stabilising differential pressure.

Please note that this publication has been prepared for an international audience. Since the use of language differs somewhat from country to country, you may find that some of the terms and symbols are not the same as those you are used to. We hope this does not cause too much inconvenience.

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Contents

1. 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, particularly after load changes.
- Room temperatures keep swinging, 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 startup after weekend or night set back.

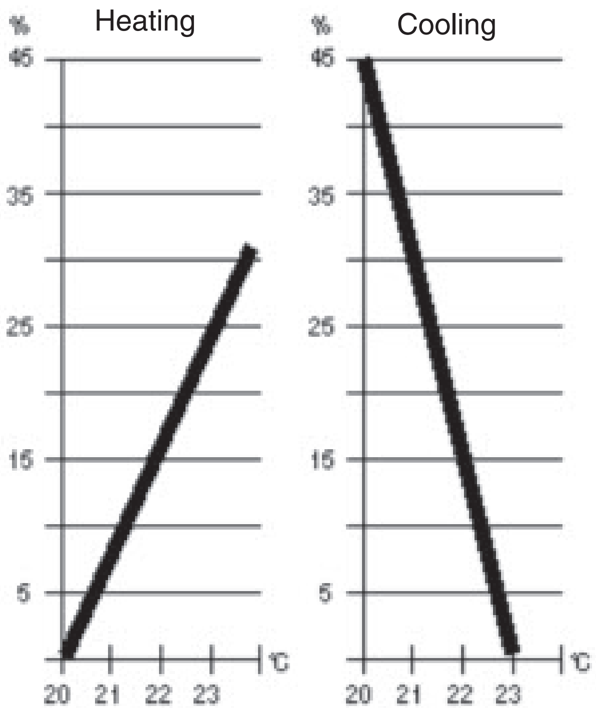
These problems frequently occur because incorrect flows keep controllers from doing their job. Controllers can control efficiently only if design flows prevail in the plant when operating under design conditions.
The only way to get design flows is to balance the plant. Balancing means adjusting the flow by means of balancing valves. This has to be done in three respects:

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. 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 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 control loops. It tells you how to balance 23 common control loops using two-way and three-way control valves.
   For manual about the balancing of distribution systems, please see TA manual No. 2.
   Manual 3 concerns Balancing of radiator systems while manual 4 examines the stabilisation of the differential pressure.

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 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 complaining 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 degrees C increases heating costs by at least 8 percent in mid Europe (12% in the south of Europe). One degree below 23 degrees C increases cooling costs by 15 percent in Europe.

2. The tools you need

Three things are necessary:
Flow measuring and regulating devices, measurement instrument and a balancing procedure.

Flow measuring and regulating devices. These are

Balancing valves which are both variable orifice and regulating valves or
Orifice devices with an independent regulating valve.

There is a great difference between balancing valves of different makes. This translates into an equally great difference in the accuracy of indoor climate control, in energy savings—and in the time, cost and effort required to do an adequate balancing job.

TA, whose products are used worldwide, cater for all the different market requirements and offer both fixed and variable flow measuring devices and regulating valves.

These are some of the distinguishing features of TA products:

STAD
STAD balancing valve
15 to 50 mm

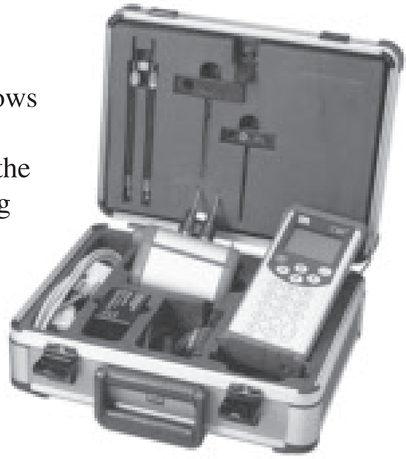
STAF
STAF balancing valve
20 to 300 mm

STAP
STAP Differential pressure controller
15 to 100 mm

Balancing valves and **orifice devices**

- Flow precision for valves better than +/- 10% and for fixed orifices better than +/- 5%.
- Sizes up to 50 mm have four full turns from open to closed position. Larger sizes have eight, twelve or sixteen full turns.
- The valves are available with internal threads, with flanges, with welded or soldered valve ends, with grooved ends and with compression fittings.
- Sizes up to 50 mm are made of Ametal®, probably the only pressure die casting alloy that meets the world's toughest demands for resistance to dezincification.

Differential pressure controller
- Adjustable set point.
- To stabilize the differential pressure across the control valves and/or circuits.



Measurement instrument. Measuring is required in order to really know that design flows are achieved and also to find what differential pressures that are applied in different parts of the plant. It is also a good tool for trouble-shooting and system analyses.

The balancing instrument CBI^{II} from TA has all necessary features to fulfil these demands, eg:

- Measures and documents differential pressure, flow and temperature of STAD, STAF, STAP/STAM and other valves from TA.
- Programmed to calculate presetting values for balancing and also the TA Method and TA Balance.
- Two-way communication with PC.
- Corrects the calculations for antifreeze agents.
- Large storage capacity - can handle 1000 valves and 24 000 values when logging.
- Graphic display making it possible to assign plain-language names for plants and valves.

Proportional relief valve. In variable flow system, a TA BPV valve can be used to perform three distinct functions:

- ensure a minimum flow to protect the pump.
- reduce the temperature drop in pipes.
- limit the differential pressure on the terminal circuits.

The BPV has a shut-off function and preset point of 10–60 kPa.
15 to 32 mm (1/2” to 1 1/4”)

3.1 Variable primary and secondary flows

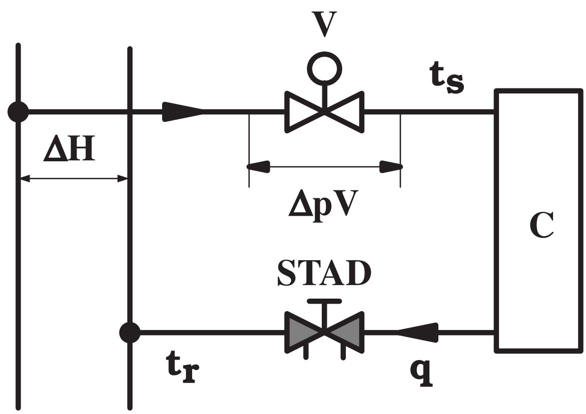


Fig. 1. Control of a variable flow terminal unit

In Fig 1 the two-way control valve controls the coil output by adapting the water flow.

The authority of the control valve $\beta' = \Delta pcV / \Delta H$. The term "authority" is explained in detail in Appendix A and B.

The two-way control valve is selected to create, fully open and design flow, a pressure drop $\Delta pV = \Delta H - \Delta pC - 3$ (kPa)

Moreover this value ΔpV must be higher than $0.25 \times \Delta H_{max}$.

Balancing procedure fig 1

1. Open all control valves fully.
2. Adjust to design flow with STAD. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

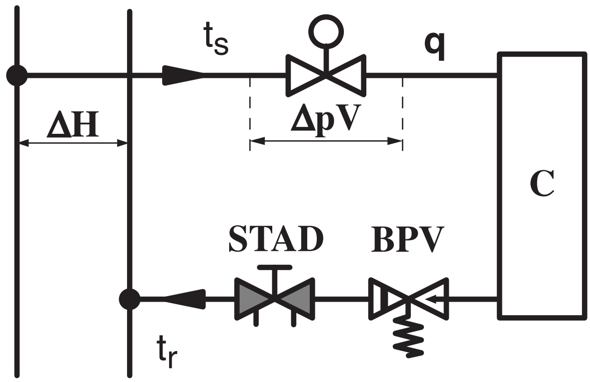


Fig. 2. A modulating relief valve reduces the differential pressure by a constant value regardless of the flow.

When the control valve is oversized, for instance due to the limited choice of Kv values, the primary differential pressure can be indirectly reduced by means of a BPV modulating relief valve. The BPV reduces the differential pressure by a constant value regardless of the flow.

The control valve authority ß' = ΔpcV / (ΔH - ΔpBPV).

Balancing procedure fig 2

1. Open all control valves fully. Make sure all the BPVs are open (minimum setpoint).
2. Adjust to design flow with STAD. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2) and before you proceed to step 3.
3. Find out which handwheel setting of STAD that will create a pressure loss of at least 3 kPa in STAD for design flow. Use the CBI or a TA nomogram to find the correct setting.
4. Readjust STAD according to step 3. The flow in STAD should now be higher than design value.
5. Adjust the setpoint of the BPV until you get back to design flow in STAD. Measure the flow in STAD as you adjust the BPV.

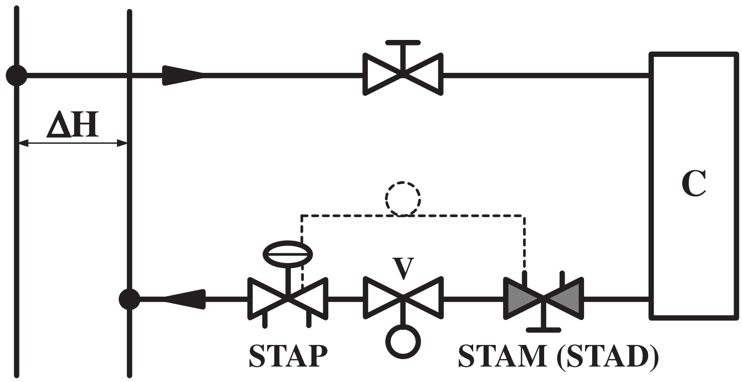


Fig. 3. A Δp controller keeps constant the differential pressure across the control valve.

Depending on the design of the plant, the differential pressure available on some circuits can vary dramatically with the load. In this case, to obtain and maintain the correct control valves characteristic, the differential pressure across the control valves can be maintained practically constant with a Δp controller as represented on figure 3.

The differential pressure across the control valve ”V” is detected on one side by connecting the capillary downstream the measuring valve STAM. The pressure on the other side is connected directly to the acting membrane by an internal connection in the STAP.

When the differential pressure across the control valve increases, the STAP closes proportionally to compensate.

The control valve ”V” is never oversized as the design flow is always obtained for the valve fully open and its authority is and remains close to one.

All additional differential pressure is applied to the STAP. The control of the differential pressure is quite easy in comparison with a temperature control and a sufficient proportional band can be used to avoid hunting.

As the flows are correct at each terminal, no other balancing procedure is required. If all control valves are combined with STAP, then balancing valves in branches and risers are not necessary but for diagnosis purposes.

Balancing procedure fig 3

1. Open fully the control valve ”V”.
2. Preset the STAM (STAD) to obtain at least 3 kPa for design flow.
3. Adjust the set point ΔpL of the differential pressure controller STAP to obtain in STAM (STAD)the design flow.

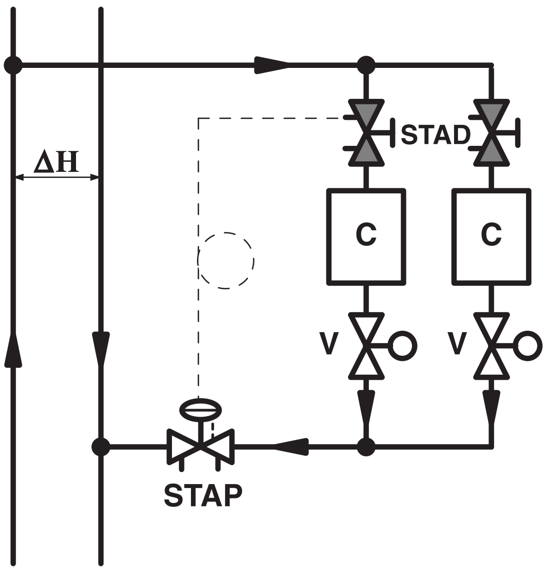


Fig. 4. A differential pressure controller STAP stabilises the differential pressure across a set of terminal units.

When several small terminal units ”C” are close to each other, it can be sufficient to stabilise the differential pressure on the whole set according to figure 4.

The supply pressure is transmitted to the STAP by means of a capillary mounted on the entering of the balancing valve of the first circuit.

When the differential pressure ΔH increases, the valve STAP shuts to compensate. Each control valves ”V” is chosen to create, fully open and at design flow, approximately the same pressure drop as its coil unit.

Balancing procedure fig 4

1. Keep the set point of the STAP as it comes from the factory.
   The control valves ”V” are fully open
2. Balance the terminals of the branch according to the TA Balance method (Handbook 2) which does not depend on the differential pressure ΔH available.
3. Adjust the set point of the STAP to obtain the design flow through the balancing valve STAD of the first circuit. The flows will be automatically correct in the other circuits.

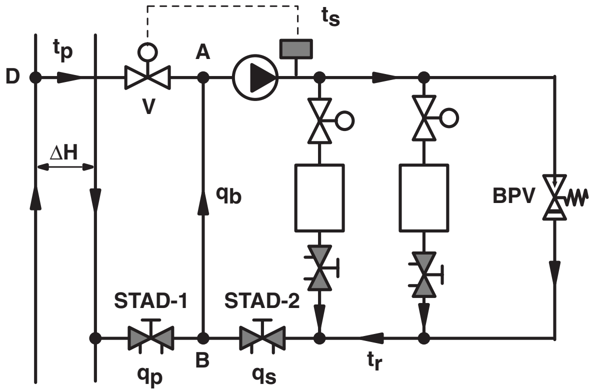


Fig.5. A secondary pump creates a sufficient differential pressure. The secondary water temperature is necessarily different from that of the primary.

If the differential pressure ΔH is too low to give a reasonable authority for the coil control valves, a secondary pump can create a sufficient differential pressure.

The solution in Fig 5 can also be used when the primary differential pressure is too high.

The secondary water temperature ts can be constant or variable, but is necessarily different from the primary temperature tp. In heating ts < tp, while in cooling ts > tp.

At low loads, the differential pressure across the secondary tends to increase. When this pressure exceeds a certain value, the BPV opens to allow a minimum flow to protect the pump. This flow also limits the temperature drop in the pipes so that the necessary water temperature is obtained throughout the secondary network.

Balancing procedure fig 5

The secondary.

1. Open all control valves fully. Close the BPV.
2. Balance the coils in the secondary system with STAD-2 as the Partner valve (see TA manual No 2).
3. Set the BPV on the maximum allowed Δp for the coil control valves.
4. Close the coil control valves.
5. Set the BPV to obtain the minimum pump flow (see Appendix C).

The primary.

1. Open the control valve V.
2. If the primary flow is unknown, calculate it using the formula on page 15.
3. Adjust the primary design flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

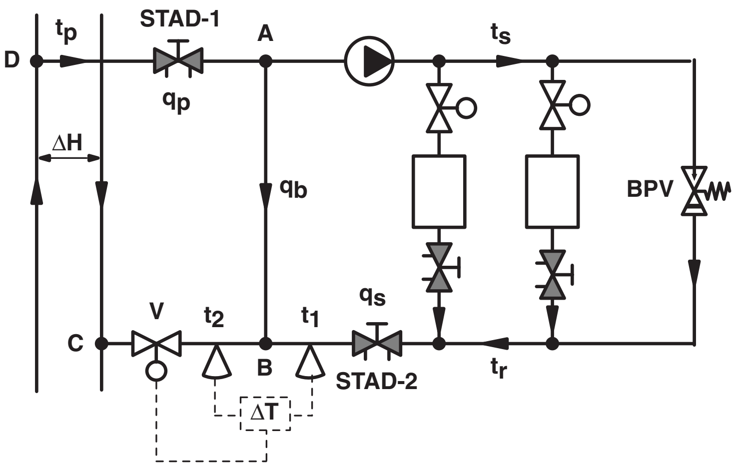


Fig.6. A differential temperature controller maintains a minimum flow qb in the bypass, such that ts = tp.

If the secondary water temperature must be equal to that of the primary, the circuit in Fig 6 (heating only) or the circuit in Fig 7 (both heating and cooling) may be used.

To obtain ts = tp, the flow qb through the bypass must be greater than zero. A ΔT controller acts on the primary control valve V to ensure a minimum flow qb in the right direction. The ΔT controller keeps t2 slightly higher than t1. Normally, the setpoint of the ΔT controller is between 1 and 2 degrees.

Balancing procedure fig 6

The secondary.

1. Open all control valves. Close the BPV.
2. Balance the coils in the secondary system with STAD-2 as the Partner valve (see TA manual No 2).
3. Set the BPV on the maximum allowed Δp for the coil control valves.
4. Close the coil control valves.
5. Set the BPV to obtain the minimum pump flow (see Appendix C).

The primary.

1. Open the control valve V.
2. If the primary flow is unknown, calculate it using the formula below.
3. Adjust to primary design flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

$$qp = 1.05\ qs$$

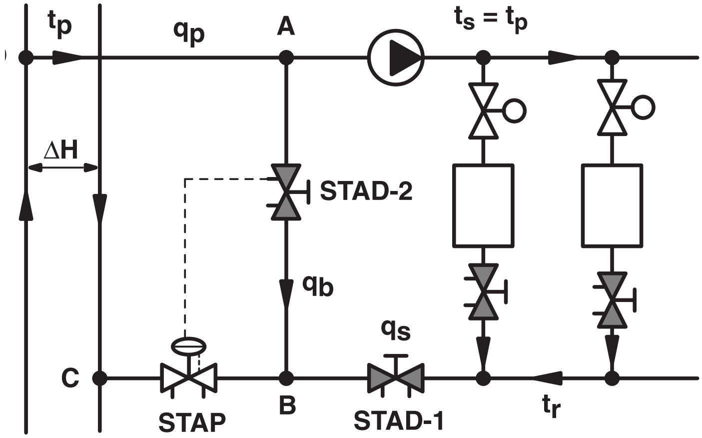


Fig. 7. A differential pressure controller keeps the flow constant in the bypass, ensuring a constant differential pressure across this bypass.

The circuit in Fig 7 may be used in cooling plants where ΔH is too low to give a sufficient authority for the coil control valves, and where ΔH varies greatly.

The control valve STAP maintains a small and constant flow in the bypass, regardless of variations in ΔH. This small flow is measured by means of STAD-2. When ΔH increases, the STAP closes correspondingly, ensuring a constant differential pressure across the balancing valve STAD-2.

Balancing procedure fig 7

1. Open all control valves.
2. Set STAD-2 to create, for 5% of design flow qs, a pressure loss corresponding to the selected setpoint of the Δp controller. Use the CBI, or a TA nomogram, to find the correct setting for STAD-2.
3. Balance the secondary circuit where STAD-1 is the Partner valve (see TA manual No. 2).

3.2 Variable primary flow and constant secondary flow

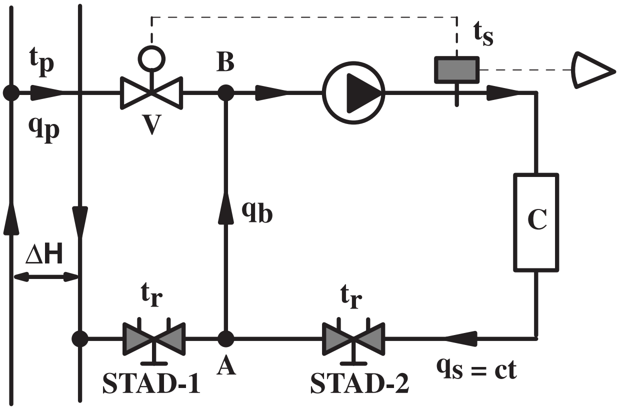


Fig. 8. Control of coil output for a coil supplied at constant flow.

This circuit is frequently used both in heating and in cooling. The coil supply temperature ts is adapted to the power need through control of the primary flow.

If, under design conditions, ts must be equal to tp, the maximum flow qp in the primary must be equal to or greater than the secondary flow qs. Otherwise, the installed power can not be transmitted to the secondary since the design value tsc can not be obtained. Primary and secondary flows must be compatible. These flows are adjusted by the balancing valves STAD-2 and STAD-1.

A floor heating example: Assume that tsc = 50°C, which is well below the tp = 80°C. The control valve must then be selected for a relatively small flow. For a return temperature trc = 45°C, the formula below shows that the primary flow will be only 14% of the secondary flow. If the control valve is selected for this flow, it can operate over its entire range. The limit of 50°C for the circuit supply temperature will not be exceeded at the maximum valve opening. If the secondary pump fails, the primary flow passes through the bypass, preventing overheating in the circuit.

Balancing procedure fig 8

1. Open control valves fully.
2. Adjust to secondary design flow with STAD-2.
3. If the primary flow is unknown, calculate it using the formula below.
4. Adjust the primary flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

$$q_p = q_s \frac{t_s - t_r}{t_p - t_r} = q_s \frac{50 - 45}{80 - 45} = 0.14\ q_s$$

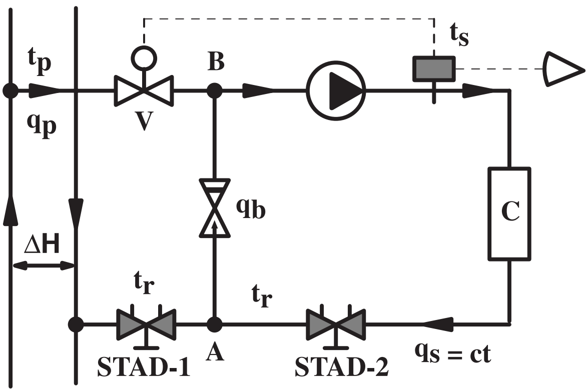


Fig. 9. A check valve in the bypass allows a certain waterflow through the coil C even if the secondary pump fails.

This is essentially the same circuit as in Fig 8. However, a check valve is added to prevent circulation in direction BA in the bypass.

If the circuit is used in district heating and the primary control valve is oversized, the check valve prevents heating of the return water. If the circuit is used for a heating coil in contact with outside air, the check valve eliminates the risk of freezing due to secondary pump failure.

Note that it's impossible to obtain a primary flow greater than the secondary flow.

Balancing procedure fig 9

tsc equal to tp:

1. Close the control valve V.
2. Adjust to secondary design flow qsc with STAD-2.
3. Open the control valve V.
4. Adjust the primary flow to the same flow qsc with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

tsc not equal to tp:

1. Close the control valve V.
2. Adjust to secondary design flow with STAD-2.
3. If primary flow is unknown, calculate it using the formula below.
4. Open the control valve.
5. Adjust the primary design flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

$$q_p = q_s \frac{(t_s - t_r)}{(t_p - t_r)}$$

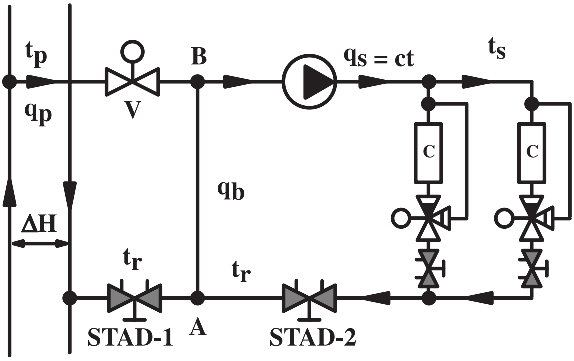


Fig. 10. A primary constant flow distribution converted to primary variable flow.

It's common to convert existing constant water flow distribution systems into variable flow in large plants. There are three reasons: 1) The supply water temperature can then be kept constant without having to keep in service all production units at all loads. 2) A variable distribution flow means reduced pumping costs. 3) The plant can be designed with a diversity factor.

Normally, the secondary side continues to work with constant flow.

After conversion, we cannot work with ts=tp. When the valve V is completely open, we can get ts=tp with flow reversal in the bypass. Since the demand is satisfied in this situation, there is no signal to make the two-way valve close. It remains open and we are back to a constant flow distribution system. To avoid this, ts has to be adjusted so that ts < tp in heating and ts > tp in cooling.

The primary flow will vary as a function of the load:

$$qp = \frac{P}{1 + \frac{(tsc - trc)}{(tp - trc)}\left(\frac{P}{100} - 1\right)} \%$$

P is the load in percent of design power.

Now assume that tp=6°C, tsc=8°C and trc=12°C. For P=50%, we get qp=75%. Thus the flow demand is 75% for a power demand of 50%.

Before converting the constant flow system into a variable flow system, the flow demand was 100% for a power demand of 50%.

This conversion does not change really the primary into a true variable distribution as the flow in % remains higher than the power in %.

Balancing procedure fig 10

1. Balance the three-way valve circuits (see TA manual No 2). STAD-2 is the Partner valve.
2. If the primary flow qp is unknown, calculate it using the formula below.
3. Open the control valve V.
4. Adjust the primary flow qp with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

$$qp = qs \frac{(ts - tr)}{(tp - tr)}$$

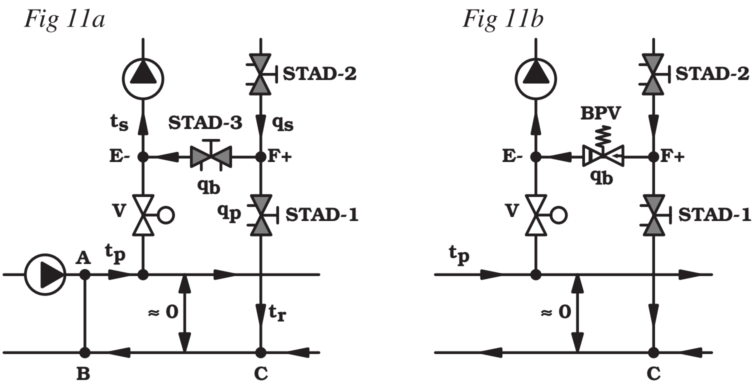


Fig. 11. Secondary pumps induce flow through the distribution network.

If the distribution network is a low pressure loss passive circuit, circulation may be induced by secondary pumps.

The balancing valve STAD-3 creates a certain differential pressure between F and E. This pressure generates the primary flow qp in the control valve V, through FCB and AE. The differential pressure ΔpcEF is obtained for qb= qsc-qpc. This implies that qsc has to be greater than qpc. When the control valve V is closed, the bypass flow qb=qsc and ΔpEF is at its maximum. This is also the differential pressure applied across the closed control valve V. To obtain a good authority for this valve, it's important to avoid big changes in ΔpEF. This means that qpc has to be as small as possible compared to qsc. Consequently, this system can only be considered if there is a large difference between ts and tp, as for example in floor heating.

The flow through the bypass is given by this formula:

$$qb = qs \frac{(tp - ts)}{(tp - tr)}$$

Assuming that the secondary flow qs is more or less constant, the control valve authority ß' = ΔpcV/ΔpEFmax.

Example: Floor heating with tp = 80°C, ts = 50°C, tr = 45°C and qs = 100. At full load, qb = 100 (80-50)/(80-45) = 85.7. At this flow, the balancing valve STAD-3 in the bypass must create a differential pressure that compensates for the pressure loss of the two-way valve (8 kPa for instance) and of the primary circuit (5 kPa), a total of 13 kPa. When the two-way valve is closed, at zero load, the flow qb changes to 100 (assuming that the increase in the pressure loss EF has little effect on the flow qs) and the pressure loss in the STAD-3 becomes ΔpEFmax = 13 x $(100/85.7)^2$ = 18 kPa.

The control valve authority is therefore ß' = 8/18 = 0.44.

The STAD-3 can be replaced by a modulating relief valve BPV (Fig 11 b) which keeps a constant differential pressure EF. In the floor heating example, this improves control valve authority from 0.44 to 0.61.

Balancing procedure fig 11

STAD-3 in the bypass (Fig 11a):

1. Open all control valves fully.
2. Set the STAD-3 to create a pressure loss ΔpEF = ΔpcV+ pressure drop in the primary circuit (8+5=13 kPa in our example) for a flow rate qb = (qsc - qpc) in the bypass. Use the CBI, or the TA nomogram, to find the correct handwheel setting for STAD-3.
3. Set the STAD-1 to create a pressure loss of 3 kPa for primary design flow. Use the CBI, or a TA nomogram, to find the correct handwheel setting for STAD-1
4. Close the control valve V. Adjust to design flow with STAD-2.
5. If the primary flow qpc is unknown, calculate it using the formula below.
6. Open the control valve V. Readjust STAD-3 to obtain qp= qpc measured in STAD-1.

$$q_p = q_s \frac{(t_{sc} - t_{rc})}{(t_p - t_{rc})}$$

BPV in the bypass (Fig 11b):

1. Open all control valves fully.
2. Set the STAD-1 to create a pressure loss of 3 kPa for qp=qpc. Use the CBI, or a TA nomogram, to find the correct handwheel setting for STAD-1.
3. Open STAD-2. Adjust the BPV to obtain design flow in STAD-1.
4. Adjust STAD-2 to obtain design flow in the secondary.

3.3 Constant primary flow and variable secondary flow

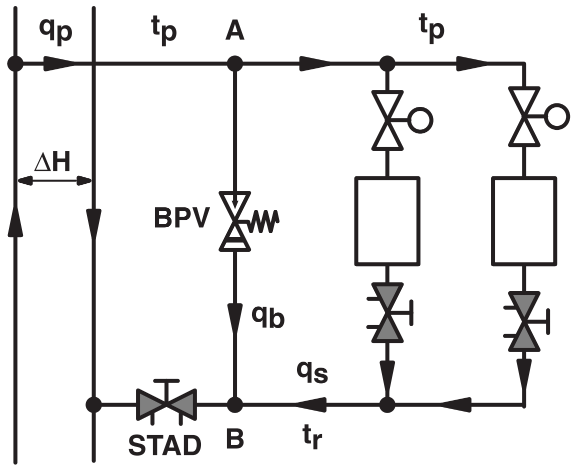


Fig. 12. A modulating relief valve BPV stabilizes the differential pressure applied to small units.

When the available differential pressure on the primary is too high for the secondary, the circuit in Fig 12 may be used.

The setpoint of the BPV can be selected within a range from 8 to 60 kPa. This makes it possible to ensure good working conditions for the coil control valves (good authority) regardless of variations in the differential pressure ΔH. The BPV ensures a constant differential pressure between A and B. STAD creates a pressure loss of (ΔH - ΔpBPV).

Balancing procedure fig 12

1. Open all control valves. Close all BPVs.
2. Balance the coils against each other, the branch against other branches and the riser against other risers (see TA manual No 2). Do this before you proceed to step 3.
3. Close the control valves of this branch.
4. Reduce the BPV setpoint slowly until you get back to 2/3 of the design flow in STAD.
   (See also handbook 4 - appendix 5.5 for complementary explanations).

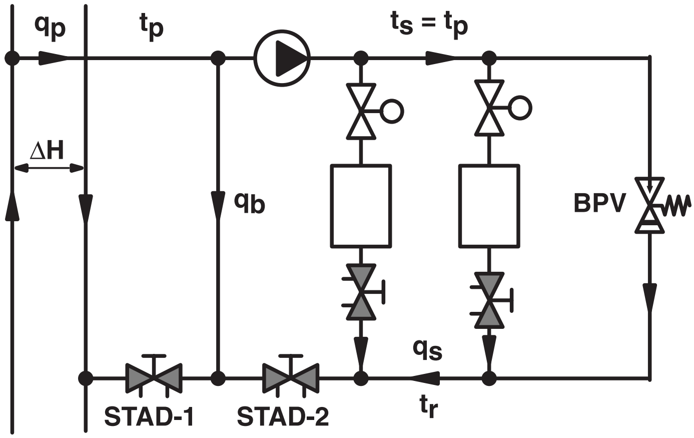


Fig. 13. Reducing or increasing the differential pressure applied across coils by means of a secondary pump.

When the primary differential pressure is too high or too small for the secondary, the circuit in Fig 13 presents a possible solution. In this circuit, a relief valve is used to obtain a minimum flow to protect the pump. STAD-1 is essential to avoid short-circuiting the primary system.

Balancing procedure fig 13

1. Open all control valves. Close all BPVs.
2. Balance the coils against each other with STAD-2 as the Partner valve (see TA manual No 2).
3. Set the BPV on the maximum permitted differential pressure for the coil control valves.
4. Close the coil control valves of this branch.
5. If necessary, reduce the BPV setpoint until you obtain the minimum pump flow (see Appendix C).
6. Adjust to primary design flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No. 2).

3.4 Constant primary and secondary flow

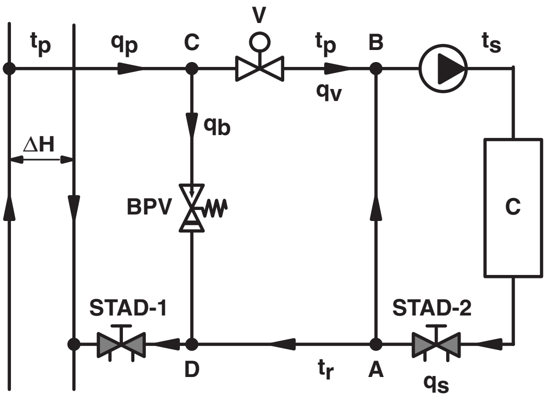


Fig. 14. Constant primary and secondary flows.

A coil is supplied at constant flow. The supply water temperature is modified by the two-way control valve V. This temperature has to be adjusted so that ts < tp in heating and ts > tp in cooling. The BPV keeps the differential pressure CD constant. This is the design pressure loss for the control valve V, which has an authority close to 1 after balancing.

Balancing procedure fig 14

1. Open all control valves. Close all BPVs.
2. If the primary flow is unknown, calculate it using the formula below.
3. Adjust the primary flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2) and before you proceed to step 4.
4. Close the control valve V.
5. Measure the flow in STAD-1. Reduce the BPV set point slowly until you get back to 2/3 of the design flow in STAD-1.
6. Adjust the secondary design flow with STAD-2.

$$qp = qs \frac{(tsc - trc)}{(tp - trc)}$$

(See also handbook 4 - appendix 5.5 for complementary explanations).

4.1 Variable primary flow and constant secondary flow

Passive primary network

A passive primary network is a distribution network without a pump. The secondary pump pressurizes both the primary and the secondary.

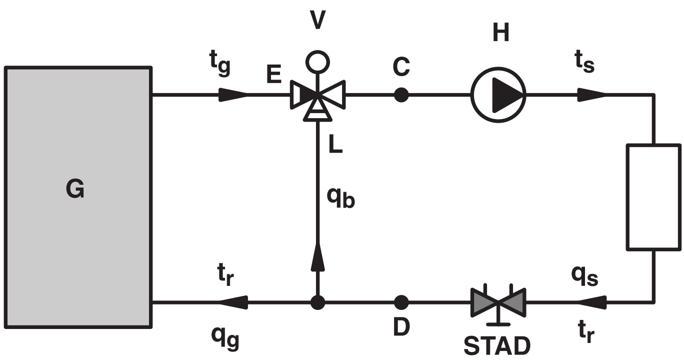


Fig. 15. Mixing circuit associated with a production unit.

Fig 15 shows a circuit controlled by a three-way mixing valve. The primary circuit consists of an exchanger, a bypass line or a boiler that can either accept zero flow or be equipped with a bypass pump that generates a minimum flow. The three-way valve should be selected for a pressure loss at least equal to that in G, and at least 3 kPa.

Balancing procedure fig 15

1. Open the three-way valve completely.
2. Adjust to design flow with STAD.

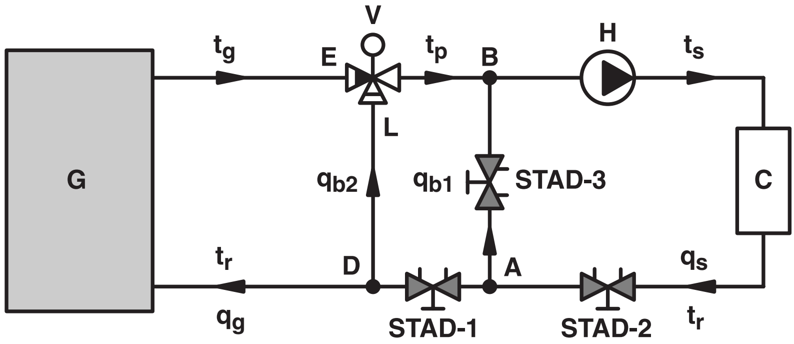


Fig. 16. Mixing circuit with intermediate bypass.

When the flow q_s in the circuit is greater than the design flow through the production unit, a bypass AB insures compatibility between the flows.

The pressure drop created by the STAD-3 , for a water flow $q_{b1} = q_{sc} - q_{gc}$, is the necessary differential pressure to compensate the pressure drops in the STAD-1 + G + the 3-way valve.

The pressure drop created by the three-way control valve for the design flow q_{gc} must be equal or higher than the design pressure drop in G and accessories with a minimum of 3 kPa.

Balancing procedure fig 16

1. Open the three-way control valve ”V”.
2. Calculate the design flow q_{b1} required in the STAD-3 and the flow q_{gc} in STAD-1 with formula below.
3. STAD-3 and STAD-1 are balanced according to the TA Balance method (See Handbook 2 - version 2).
4. Adjust the flow q_s with the STAD-2.

$$q_{gc} = q_{sc} \frac{(tsc- trc)}{(tg- trc)} \qquad q_{b1} = q_{sc} - q_{gc}$$

Active primary network

An active primary network is a distribution system with its own pump.

The primary pump creates a differential pressure which forces the water flow through the secondary circuits.

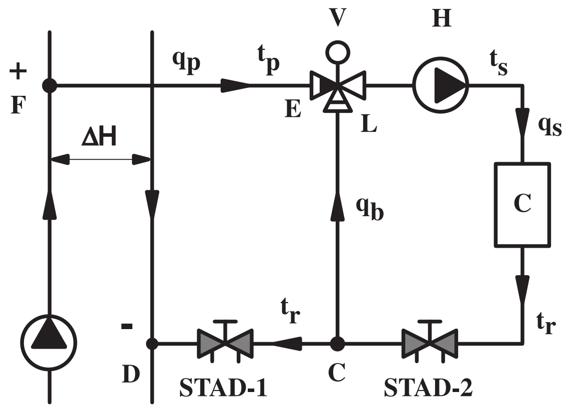


Fig. 17. Mixing valve with differential primary pressure and compensation balancing valve.

The three-way valve in Fig 17 is supplied by a primary differential pressure ΔH. This pressure may disturb the function of the three-way valve. The water flow qb in the bypass may reverse and cancel the mixing function of the control valve.

To prevent this, the balancing valve STAD-1 has been installed. The pressure loss in STAD-1 should be ΔH for design flow qpc.

The design pressure loss across the three-way valve must be at least equal to ΔH to give an authority of 0.5. This pressure loss has to be covered by the secondary pump.

Balancing procedure fig 17

1. Close the three-way valve.
2. Adjust to secondary design flow with STAD-2
3. Open the three-way valve.
4. Continue to measure the flow with STAD-2. Adjust STAD-1 to obtain the same flow as in step 2. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2).

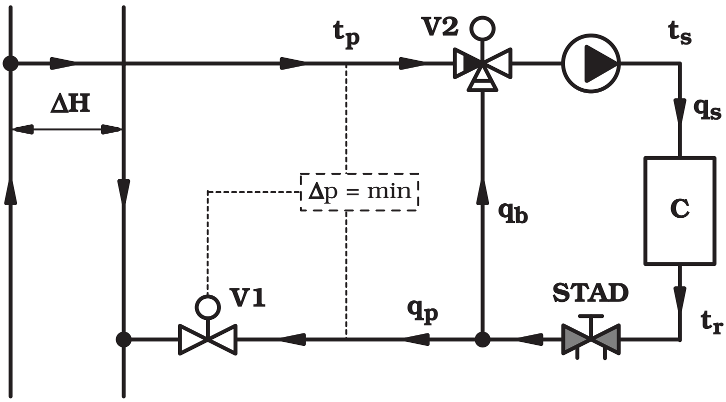


Fig. 18. Elimination of the primary differential pressure by means of a pressure controller.

In some plants the three-way valves do not work satisfactorily because of a too high primary differential pressure. Sometimes a differential pressure controller is installed to eliminate or reduce this pressure to a reasonable value, as shown in Fig 18.

This is an expensive solution. However, it may be considered if the differential pressure controller is used to supply several 3-way valves and when a variable flow system is required in the distribution. If a constant primary flow is accepted, the design of figure 20 is better.

Balancing procedure fig 18

1. Close the three-way valve.
2. Adjust to secondary design flow with STAD.
3. Adjust the setpoint of the differential pressure controller to a value as low as possible.

4.2 Variable primary and secondary flows

Passive primary network

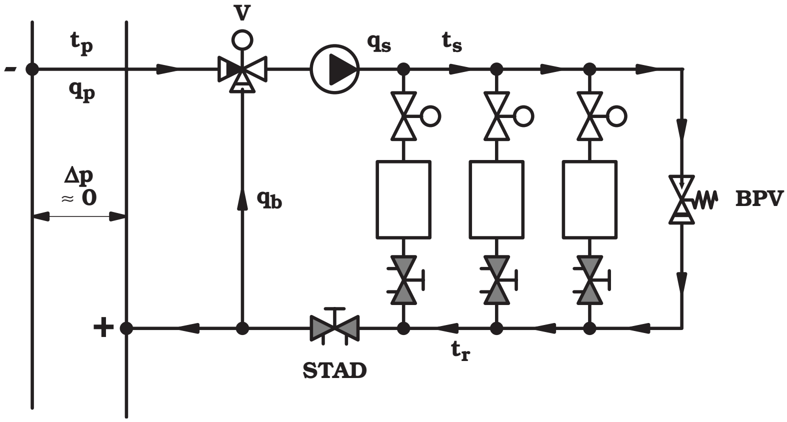


Fig. 19. The three-way valve prepares the water temperature in the distribution system.

The three-way valve controls the secondary water temperature. The two-way control valves make the fine tuning of the energy supply by adapting the flow to the demands.

The three-way valve has an authority close to 1. At low loads, the modulating relief valve BPV ensures a minimum pump flow, and also reduces the temperature drop in the pipe line.

Note: Below a certain flow, a three-way valve will work with laminar rather than turbulent flow. Then the three-way valve temporarily loses its basic characteristics and the control loop becomes difficult to stabilize. Thus, the minimum flow controlled by the BPV must be high enough to create a pressure loss of at least 1 kPa in the three way valve.

Balancing procedure fig 19

1. Open all control valves. Close the BPV.
2. Balance the secondary system (see TA manual No 2) with STAD as the Partner valve.
3. Close all two-way control valves.
4. Set the BPV to obtain the minimum pump flow (see Appendix C).

4.3 Constant primary and secondary flows

Active primary network

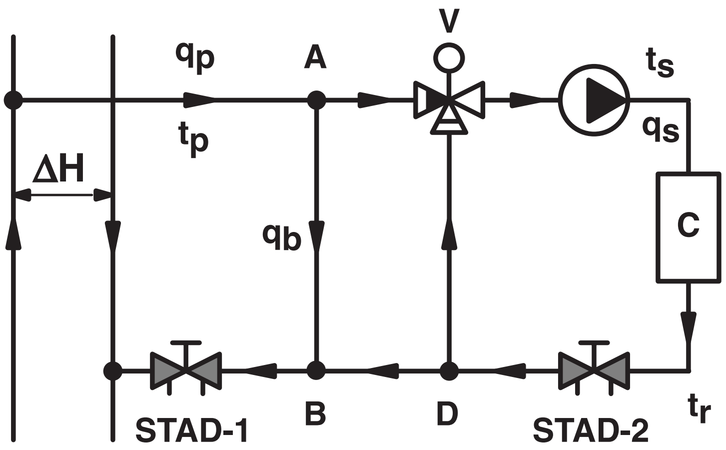


Fig. 20. The balancing valve STAD-1 and the bypass AB, eliminate the primary differential pressure across the three-way valve.

If the primary flow may be constant, it's simple to avoid a too high differential pressure on the primary of a mixing three-way valve. It's just to install a bypass AB and to compensate the primary differential pressure with the balancing valve STAD-1. The authority of the three-way valve will then be close to 1.

Balancing procedure fig 20

1. Open the three-way valve.
2. Adjust to secondary design flow with STAD-2.
3. If the primary flow qp is unknown, calculate it using the formula below.
4. Adjust the primary flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No. 2).

$$q_p = q_s \frac{(t_{sc} - t_{rc})}{(t_p - t_{rc})}$$

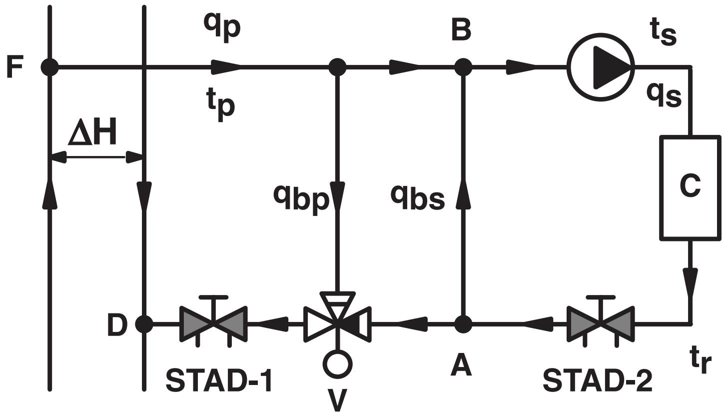


Fig. 21. When tsc is not equal to tp, it is better to place the bypass on the secondary side.

When the design temperature tsc is not equal to tp, the circuit in Fig 21 is often preferable to that in Fig 20.

The flow in the control valve is lower in Fig 21 than in Fig 20 (qp instead qs), thus allowing the use of a smaller three-way valve.

The authority of the three-way valve is close to 1.

Balancing procedure fig 21

1. Open the three-way valve.
2. Adjust to secondary design flow with STAD-2.
3. If the primary flow qp is unknown, calculate it using the formula below.
4. Adjust the primary flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No. 2).

$$q_p = q_s \frac{(t_{sc} - t_{rc})}{(t_p - t_{rc})}$$

4.4 Constant primary flow and variable secondary flow

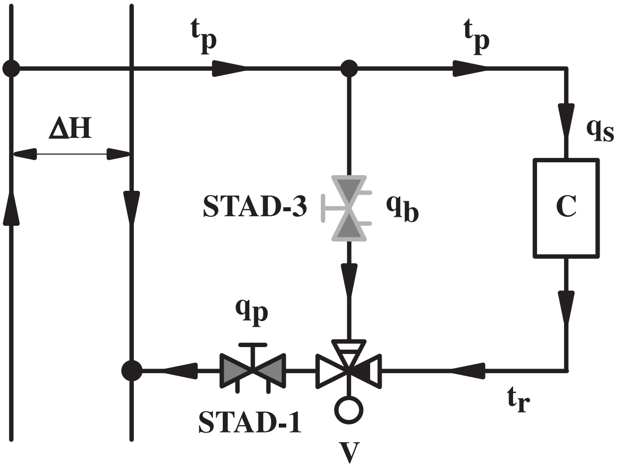


Fig. 22. A three-way mixing valve in a diverting circuit.

The three-way valve used as a mixing valve in a diverting circuit can supply the coil at variable flow and constant water supply temperature, while keeping primary flow constant. This way, the three-way valve eliminates interactivity between circuits on the primary side.

The three-way valve should create a design pressure loss equal to or greater than the pressure loss in circuit C to ensure an authority of at least 0,5.

Note: The most important balancing valve is the STAD-1. STAD-3 can be omitted if $\Delta pC < 0.25\ \Delta H$.

Balancing procedure fig 22

1. Open all three-way valves.
2. Adjust to design flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2) and before you proceed to step 3.
3. Close the three-way valve.
4. Measure the flow in STAD-1. Adjust to design flow with STAD-3.

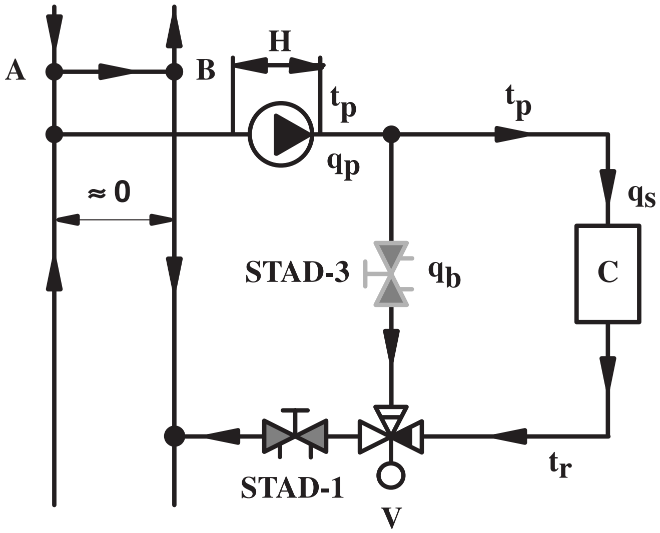


Fig. 23. Circuit in diversion on a passive distribution.

If the distribution system is passive (no active differential pressure), there is a need for a separate pump. This pump can be common to several circuits.

Note: The most important balancing valve is STAD-1. STAD-3 can be omitted if $\Delta pC < 0.25$ H.

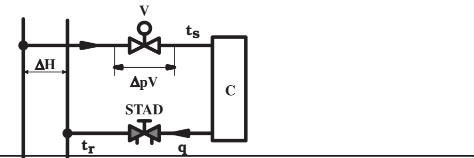
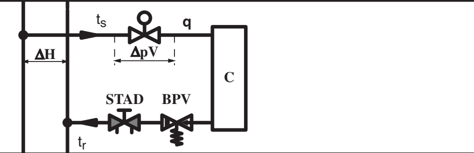
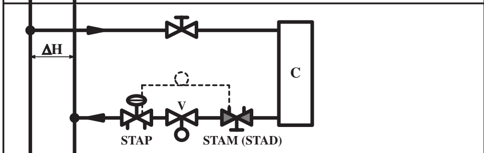
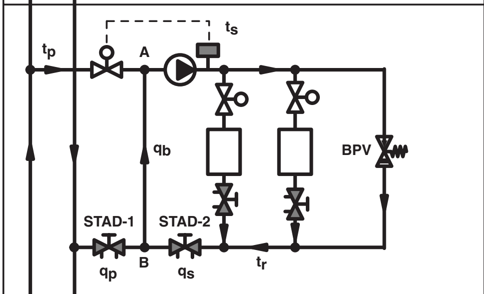
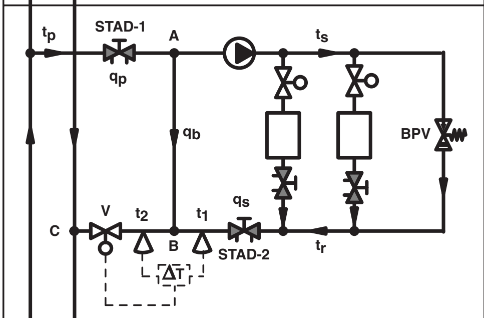
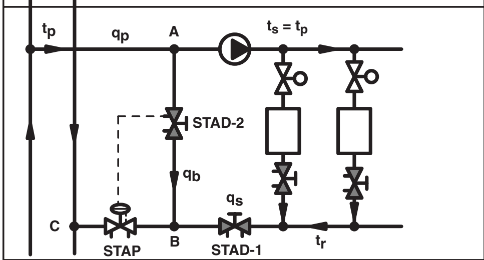
Balancing procedure fig 23

1. Open all three-way valves.
2. Adjust to design flow with STAD-1. Do this as part of the balancing procedure for the entire primary system (see TA manual No 2) and before you proceed to step 3.
3. Close all three-way valves.
4. Measure the flow in STAD-1. Adjust to design flow with STAD-3.

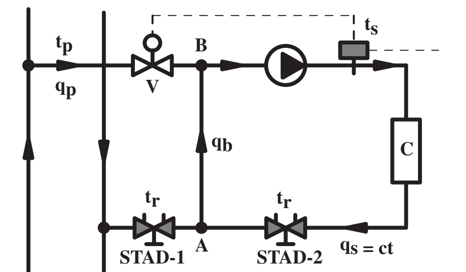
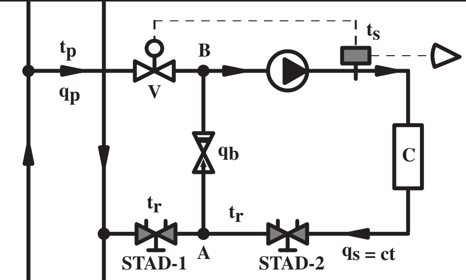
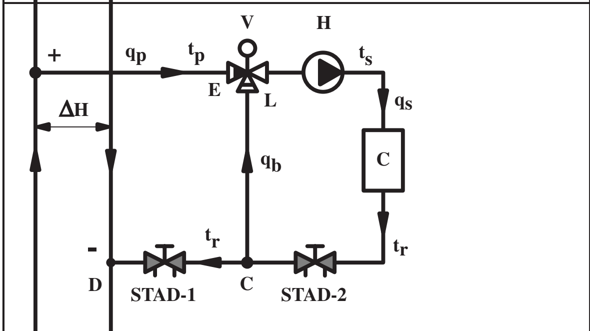
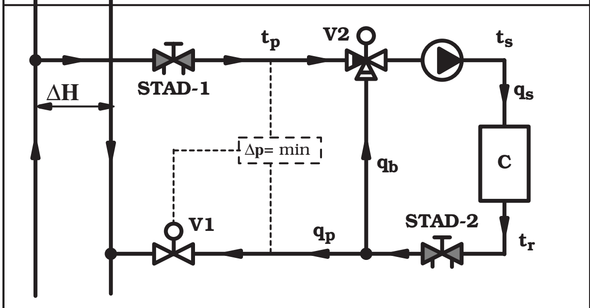
Variable primary water flow			
Variable secondary water flow		Constant secondary water flow	
2 way 3 way	5 19	2 way 3 way	8 - 9 - 10 17 - 18
Constant primary water flow			
Variable secondary water flow		Constant secondary water flow	
2 way 3 way	12 - 13 22	2 way 3 way	14 20 - 21

Same functions obtained with two way or three way control valves.

5.1 Active primary network

	1	$\Delta pV > \Delta H/2$ * $\Delta p_{STAD} = \Delta H - \Delta pV - \Delta pC$ $\beta' = \Delta pV / \Delta H$
	2	$\Delta pV > (\Delta H - \Delta p_{BPV})/2$ * $\Delta p_{STAD} > 3$ kPa $\Delta p_{BPV} = \Delta H - \Delta pV - \Delta pC - \Delta p_{STAD}$ $\beta' = \Delta pV / (\Delta H - \Delta p_{BPV})$
	3	$\Delta pV >$ Min STAP set point ≥ 10 kPa $\Delta p_{STAM\ (STAD)} \geq 3$ kPa β' close to one
	5	$q_s < q_p$ $\Delta pV > \Delta H/2$ * $\Delta p_{STAD-1} = \Delta H - \Delta pV$ $\beta' = \Delta pV / \Delta H$
	6	$t_s = t_p$ $q_s < q_p$ $\Delta pV > \Delta H/2$ * $\Delta p_{STAD-1} = \Delta H - \Delta pV$ $\beta' = \Delta pV / \Delta H$
	7	$t_s = t_p$ $\Delta pV > \Delta H/2$ * $\Delta p_{STAD-1} = \Delta H - \Delta pV - \Delta p_{STAD-2}$ $\beta' = \Delta pV / \Delta H$

Variable primary and secondary water flows.
Variables are represented by their design values - Values recommended (*).

	**8**	$q_s < q_p$ $\Delta pV > \Delta H/2$ * $\Delta p_{STAD-1} = \Delta H - \Delta pV$ $\beta' = \Delta pV / \Delta H$
	**9**	$q_s < q_p$ $\Delta pV > \Delta H/2$ * $\Delta p_{STAD-1} = \Delta H - \Delta pV$ $\beta' = \Delta pV / \Delta H$
	**17**	$\Delta pV > \Delta H$ * $\Delta p_{STAD-1} = \Delta H$ $\beta' = \Delta pV / (\Delta pV + \Delta pH)$
	**18**	$\Delta pV1 > \Delta H/2$ * $\Delta pV2 > 3$ kPa * $\Delta p_{STAD-1} = \Delta H - \Delta pV1$ $\beta'_{V1} = \Delta pV1 / (\Delta H - \Delta p)$

Variable primary water flow and constant secondary water flow. Variables are represented by their design values - Values recommended (*).

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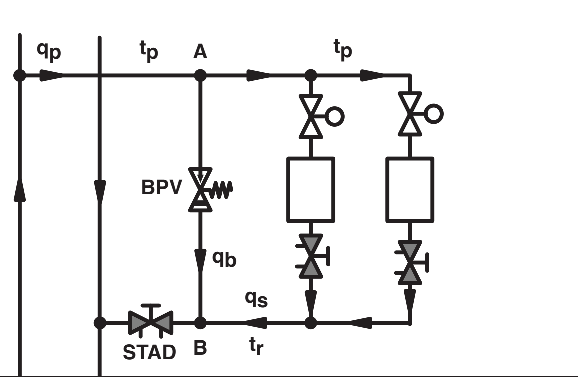
<td>
</td>
<td>12</td>
<td>$t_s = t_p$

$\Delta p_{STAD-1} = \Delta H - \Delta p_{BPV}$</td>
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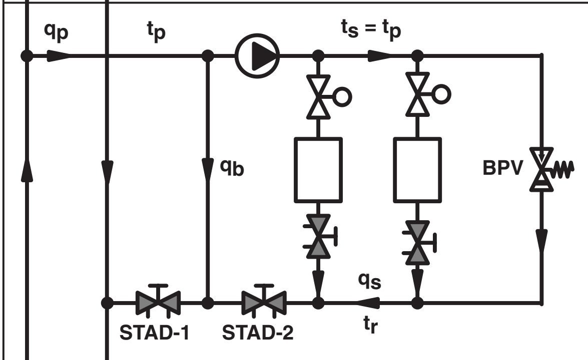
<td>
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<td>13</td>
<td>$t_s = t_p$

$\Delta p_{STAD-1} = \Delta H$</td>
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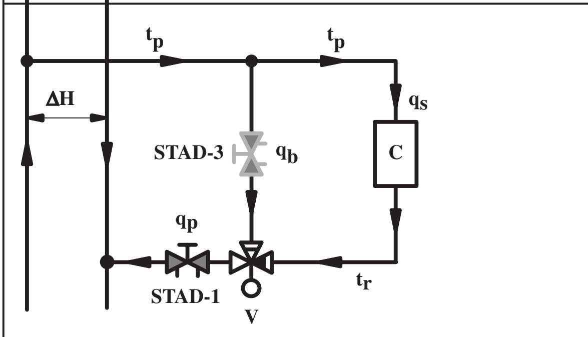
<td>
</td>
<td>22</td>
<td>$t_s = t_p$
$\Delta pV > \Delta pC$ *
$\Delta p_{STAD-3} = \Delta pC$
$\Delta p_{STAD-1} = \Delta H - \Delta pV - \Delta pC$
$\beta' = \Delta pV / (\Delta pV + \Delta pC)$</td>
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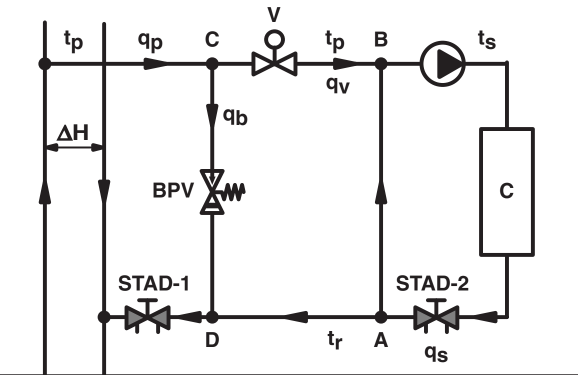
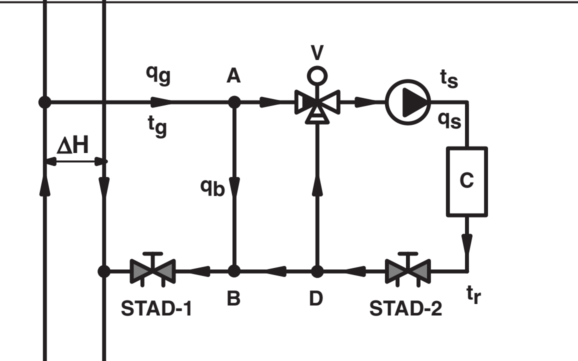
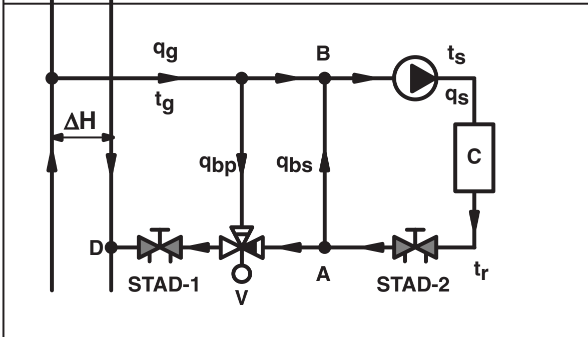
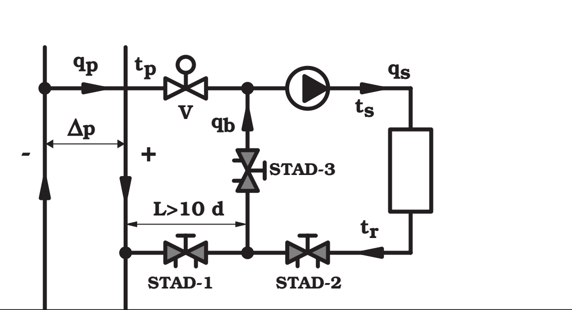
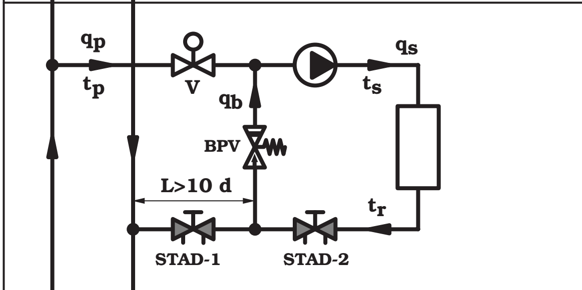
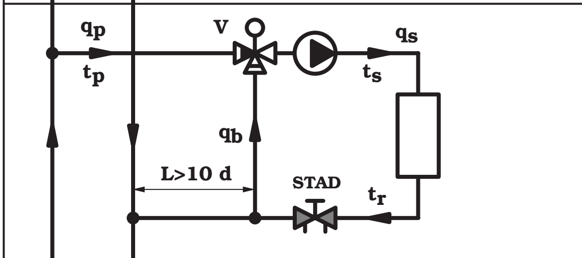
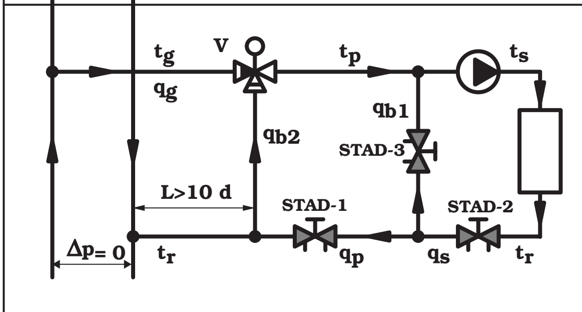
Constant primary water flow and variable secondary water flow. Variables are represented by their design values - Values recommended (*).

	**14**	$q_s > q_p$ $\Delta pV > 8$ kPa $\Delta p_{STAD-1} = \Delta H - \Delta p_{BPV}$ $\beta' = \Delta pV / \Delta p_{BPV}$
	**20**	$\Delta pV > 3$ kPa * $\Delta p_{STAD-1} = \Delta H$ $\beta' = 1$
	**21**	$\Delta pV > 3$ kPa * $\Delta p_{STAD-1} = \Delta H - \Delta pV$ $\beta' = 1$

Constant primary and secondary water flows.
Variables are represented by their design values - Values recommended (*).

5.2 Passive primary network

	(11a)	$q_p < q_s$ $\Delta p_{STAD-3} =$ $\Delta p1 + \Delta pV + \Delta p_{STAD-1}$ $\Delta p_{STAD-1} \geq 3$ kPa * $\Delta pV \geq \Delta p_{STAD-3} / 2$ * $\beta' = \Delta pV/\Delta p_{STAD-3}max$
	(11b)	$q_p < q_s$ $\Delta p_{BPV} =$ $\Delta p1 + \Delta pV + \Delta p_{STAD-1}$ $\Delta p_{STAD-1} \geq 3$ kPa * $\Delta pV \geq \Delta p_{STAD-3} / 2$ * $\beta' = \Delta pV/\Delta p_{BPV}$
	(15)	$\Delta pV > \Delta p1$ * $\beta' = \Delta pV / (\Delta pV + \Delta p1)$
	(16)	$q_p < q_s$ $\Delta pV > \Delta p1$ * $\beta' = \Delta pV / (\Delta pV + \Delta p1)$

Variable primary water flow and constant secondary water flow. Variables are represented by their design values - Values recommended (*).

Appendix A
The authority of two-way control valves

A.1 The incomplete definition of valve authority

The static characteristic of a control valve is defined for a constant differential pressure across the valve. But this pressure is rarely constant in a plant. Therefore, the real characteristic of a control valve is not the same as the theoretical one.

When the control valve is fully open, the differential pressure Δpmin is equal to the available differential pressure minus pressure losses in terminal unit, pipes and accessories.

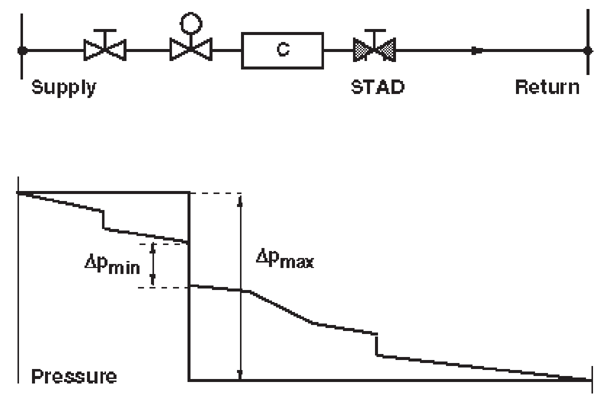
When the control valve is closed, pressure losses in the other elements disappear since the flow is zero. The entire available differential pressure ΔHmax=Δpmax is then applied across the control valve.

But the control valve is sized based on Δpmin since it is at this pressure loss the design flow is to be obtained (fully open valve).

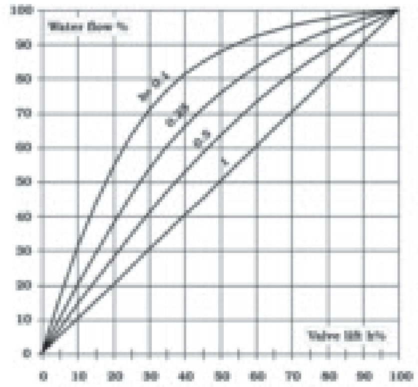
When the valve is near its closed position, the real flow is higher than the theoretical since the differential pressure is greater than Δpmin. The theoretical characteristic is distorted. The degree of this distortion depends on the ratio Δpmin/Δpmax.

This ratio is the control valve authority.

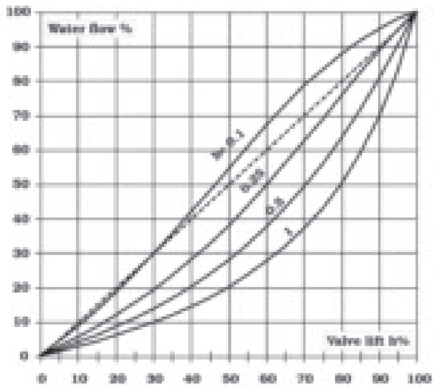
$$\beta = \frac{\Delta p_{min}}{\Delta p_{max}}$$



The differential pressure applied to the control valve depends on its degree of opening.



Distortion of a linear valve characteristic as a function of its authority.



Distortion of the "EQM" characteristic of a valve as a function of its authority.

The lower the authority, the bigger the distortion of the theoretical valve characteristic.

Consider a valve with a linear characteristic, designed to obtain the design flow exactly at full opening, but with a fairly low authority of 0.1. At a 10% valve lift, the flow in the circuit is about 30%.

Suppose that the terminal unit is a heating coil with a design temperature drop of 10K. Then 30% of the water flow produces 80% of the design power.

The final result is that the coil output is 80% of the design power at a control valve lift of only 10%. Under these conditions there is little hope to obtain stable control. The situation would be even worse if, for the same authority, the control valve was oversized!

An authority of 0.5 is acceptable since it does not greatly deform the valve characteristic. In other words, the pressure loss at design flow in a fully open control valve must be equivalent to at least half the available differential pressure.

Note that the design flow does not appear in the definition of valve authority.

The curves in the figures above are plotted assuming design flow when the control valve is fully open. But that is rarely the case in practice, since it is difficult to avoid a certain degree of oversizing.

When a control valve is oversized, Δpmin is reduced, assuming constant Δpmax. Thus, the control valve authority is also reduced. The theoretical valve characteristic will be heavily distorted and control becomes difficult at small loads.

However, an oversized control valve can have a good authority. If the differential pressure applied to a circuit is doubled, Δpmin and Δpmax increase in the same proportion and the authority remains unchanged, although there's now an overflow in the circuit.

What, then, will happen to the valve authority in a circuit exposed to a variable differential pressure?

Then Δpmax and Δpmin will vary simultaneously in the same proportions. The valve authority ß thus remains constant.

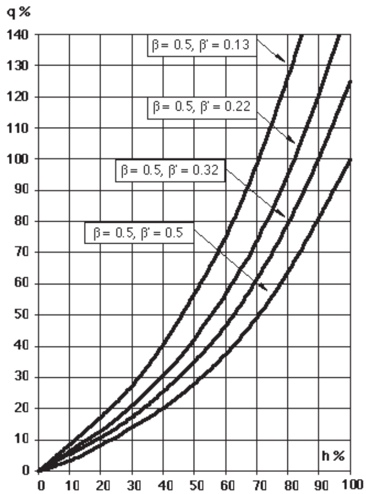
However, the valve characteristic is deformed, despite the fact that the authority ß is the same.

Therefore, the authority as defined above, does not give enough information about the real distortion of the valve characteristic.

A.2 The correct definition of valve authority ß'

We get a more coherent definition of the authority if we relate the pressure loss in the control valve for design flow to the maximum pressure loss in the valve:

$$ß' = \frac{\Delta p \text{ across the fully open control valve and design flow}}{\Delta p \text{ valve shut}}$$



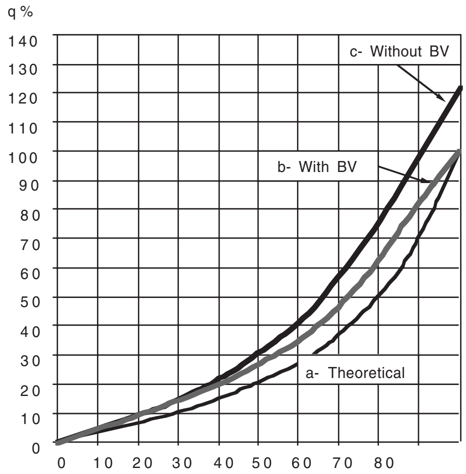
Flow as a function of valve lift when the circuit supply pressure varies with constant authority ß.

The figure shows that the authority ß' takes into account the distortion of the valve characteristic. This is not true for the authority ß according to the conventional definition.

The two authority factors relate to each other according to the formula below (Sq is the overflow factor):

$$ß = (Sq)^2 \,.\, ß'$$

$Sq \geq 1$ with the valve open. When the maximum flow is equal to design flow, ß= ß'.



Influence on the control valve characteristic through limiting the maximum flow by a balancing valve.

Can a balancing valve be put in series with a control valve?

A control valve with exactly the calculated Kv-value is usually not available on the market. Consequently, the installed valve is more or less oversized. During start up after night setback, when most control valves are open, the overflow in favoured units creates underflows in others. It's therefore essential that the flow through the control valve is limited by a balancing valve.

The figure above shows how this type of limitation influences the control valve characteristic. Without a balancing valve, the overflow in the fully open control valve will be 22% and its authority ß = 0.5 according to the conventional definition of valve authority. But this is a rather misleading information about the authority, since the flow is wrong.

The authority ß' = 0.34 indicates the real distortion of the valve characteristic.

The authority ß' is the same with or without the balancing valve, and depends mainly on the initial choice of control valve.

By installing a balancing valve, we can obtain the correct water flow under design conditions and **improve** the control valve characteristic.

A.3 Sizing of control valves

The Kv factor.

A control valve creates a complementary pressure loss in the circuit to limit the water flow to the required value. The water flow depends on the differential pressure applied to the valve:

$$q = Kv \sqrt{\frac{\Delta p \times 1000}{\rho}}$$

Kv is the valve flow factor.

ρ is the density. For water, ρ =1000 kg/m3 at 4°C and ρ = 970 at 80°C.
q is the liquid flow in m3/h.

Δp is the differential pressure expressed in bars.

The maximum Kv value (Kvs) is obtained when the valve is fully open. This value corresponds to the water flow expressed in m3/h, for a differential pressure of 1 bar.

The control valve is selected so that its Kvs value will give design flow for the differential pressure available when the valve is working under design conditions.

It is not easy to determine the Kvs needed for a control valve since the available differential pressure for the valve depends on many factors:

- The actual pump head.
- Pressure losses in pipes and accessories.
- Pressure losses in terminal units.

These pressure losses also depend on the precision with which balancing is done.

During the design of the plant, the theoretically correct values for pressure losses and flows for various components are calculated. But components with exactly the specified properties are rarely available. The installer must normally select standard values for pumps, control valves and terminal units.

Control valves, for instance, are available with Kvs values which increase in a geometric progression, called a Reynard series:

Kvs: 1.0 1.6 2.5 4.0 6.3 10 16 .....

Each value is about 60% greater than the previous value.

It is unusual to find a control valve that creates exactly the desired pressure loss for design flow. If, for instance, a control valve that creates a pressure loss of 10 kPa for the design flow is needed, it may occur that the valve with the nearest higher Kvs value creates a pressure loss of only 4 kPa, while the valve with the nearest lower Kvs value creates a pressure loss of 26 kPa for design flow.

Δp (bar), q (m3/h)	Δp (kPa), q (l/s)	Δp (mm WG), q (l/h)	Δp (kPa), q (l/h)
$q = Kv\sqrt{\Delta p}$	$q = Kv\sqrt{\Delta p}$	$q = 10\ Kv\sqrt{\Delta p}$	$q = 100\ Kv\sqrt{\Delta p}$
$\Delta p = \left(\frac{q}{Kv}\right)^2$	$\Delta p = \left(36\frac{q}{Kv}\right)^2$	$\Delta p = \left(0.1\ \frac{q}{Kv}\right)^2$	$\Delta p \approx \left(0.01\frac{q}{Kv}\right)^2$
$Kv = \frac{q}{\sqrt{\Delta p}}$	$Kv = 36\ \frac{q}{\sqrt{\Delta p}}$	$Kv = 0.1\frac{q}{\sqrt{\Delta p}}$	$Kv = 0.01\frac{q}{\sqrt{\Delta p}}$

Some formulas involving the flow, Kv and Δp (ρ = 1000 kg/m^3).

In addition, pumps and terminal units are often oversized for the same reason. This means that control valves frequently have to work near their closed position, resulting in unstable control. It is also possible that these valves periodically open to a maximum, definitely during start up, creating an overflow in their unit and underflows in other units. We should therefore ask the question:

What to do if the control valve is oversized?

We have already seen that we usually can't find exactly the control valve that we want.

Take the case of a 2000 watt coil, designed for a temperature drop of 20 K. Its pressure loss is 6 kPa for the design flow of 2000 x 0.86/20 = 86 l/h. If the available differential pressure is 32 kPa and pressure losses in pipes and accessories are 4 kPa, the difference is 32 - 6 - 4 = 22 kPa which must be applied across the control valve.

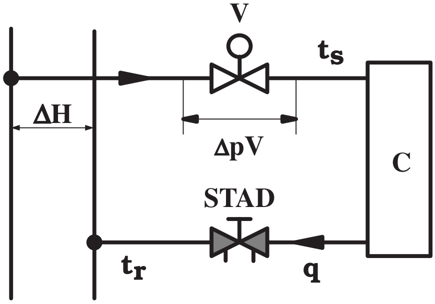
The required Kvs value is 0.183.

If the lowest available Kvs value is 0.25, for example, the flow will be 104 l/h instead of the desired 86 l/h, an increase by 21%.

In variable-flow systems, differential pressures applied to terminal units are variable since pressure losses in pipes depend on the flow. Control valves are selected for design conditions. At low loads, the maximum potential flow in all units is increased and there is no risk of creating underflows in some units. Under design conditions, and when maximum load is required, it's essential to avoid overflows.

a- Flow limitation by means of a balancing valve in series

If the flow in the open control valve under design conditions is greater than the required value, a balancing valve in series can be used to limit this flow. This does not change the real authority of the control valve, and we even improve its characteristic (see figure page 41). The balancing valve is also a diagnostic tool and a shut-off valve.



A balancing valve limits the flow in the control valve without changing its authority ß'

b - Reduction of the maximum valve lift

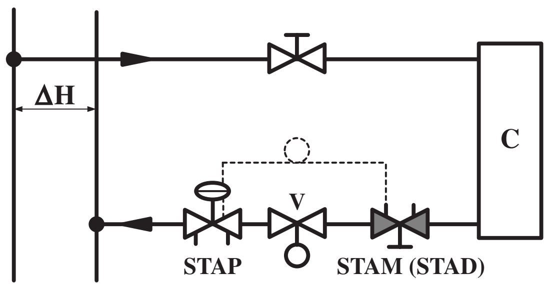
To compensate for oversizing of a control valve, the valve opening may be limited. This solution can be considered for Equal percentage characteristic valves, since the maximum Kv value can be significantly reduced with a reasonable reduction in the maximum opening. If the degree of opening is reduced by 20%, the maximum Kv value is reduced by 50%.

In practice, balancing is carried out by means of balancing valves in series with fully open control valves. The balancing valves are then adjusted in each circuit successively in order to give a pressure loss of 3 kPa at the design flow.

The control valve lift is then limited to create 3 kPa in the balancing valve. Since the plant is and remains balanced, the flow is therefore actually obtained under design conditions.

c- Flow reduction using a Δp control valve in series.

The differential pressure across the control valve can be stabilized according to figure below.



A Δp controller keeps constant the differential pressure across the control valve

The set point of the differential pressure control valve STAP is chosen to obtain the design flow for the control valve fully open. In this case, the control valve is never oversized and its authority is kept close to one. Balancing procedure is described on page 10.

Some rules of thumb.

When two-way control valves are used on terminal units, most control valves will be closed or almost closed at low loads. Since water flows are small, pressure losses in pipes and accessories are negligible. The entire pump head is applied across the control valves, which must then be able to resist this pressure. This increase in the differential pressure makes control difficult at small flows, since the real valve authority ß' is reduced greatly.

Assume that a control valve is designed for a pressure loss of 4% of the pump head. If the plant works at small flow, the differential pressure across this control valve increases from 4 to almost 100%. The differential pressure is thus multiplied by 25. For the same valve opening, all flows are then multiplied by 5 ($\sqrt{25} = 5$).

The valve is forced to work near its closed position. This may result in noise and hunting (under these new working conditions, the valve is oversized five times).

This is why some authors recommend that the plant is designed so that the design pressure loss in the control valves is at least 25% of the pump head. Then, at low loads, flow oversizing of control valves does not exceed a factor 2.

It's not always possible to find control valves capable of resisting such large differential pressure without generating noise. It is also difficult to find valves small enough to satisfy the above criterion when low power terminal units are used. Then, differential pressure variations in the plant should be limited, for example by the use of secondary pumps.

Taking this additional concept into consideration, the sizing of a two-way control valve must satisfy the following conditions:

1. When the plant operates under normal conditions, the flow in the fully open control valve must be the calculated flow. If the flow is greater than this, a balancing valve in series will limit the flow. An authority of 0.30 is then acceptable for a PI type controller. If the authority is lower, the control valve should be replaced for a smaller one.

2. The pump head should be such that the pressure loss in the two-way control valves can be selected to be at least 25% of this pump head.

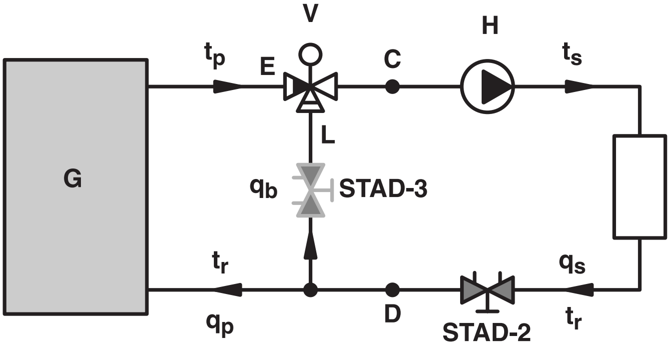
For On-Off controllers, the authority concept is meaningless since the control valve is either open or closed. Its characteristic is therefore not very important. In this case the flow is limited, with no fundamental restriction, by a balancing valve in series.

The authority of three-way control valves

B.1 In mixing function

A three-way valve used in mixing can supply a circuit at constant flow and variable inlet water temperature.

The primary water at temperature tp is mixed with the return water at temperature tr in the necessary proportion to obtain the required mix temperature ts.



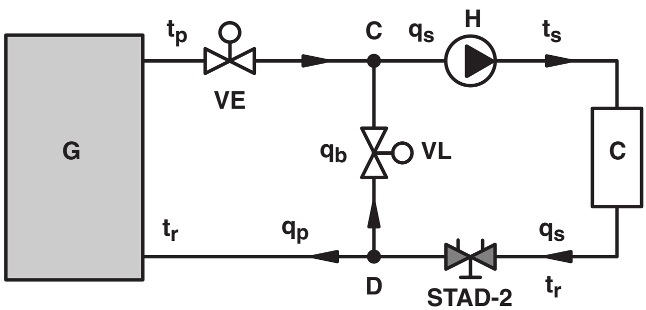
Three way valve in mixing function.

When port E opens, port L closes in the same proportion. The third common port remains open. When port E is closed, the three-way valve is closed and no energy can be extracted from the primary. The temperature ts is then equal to tr which gradually reaches the mean temperature of the room.

The balancing valve STAD-2 can adjust the flow to the required value. In principle, a hydraulic resistance equal to that of G must be created in the bypass by means of STAD-3 in order to give the same water flow qs regardless of whether the three-way valve is open or closed. In this case, the three-way valve is balanced.

The authority of the three-way valve.

We will replace the three-way valve by two two-way valves working in opposition. We will then obtain the same mixing function.



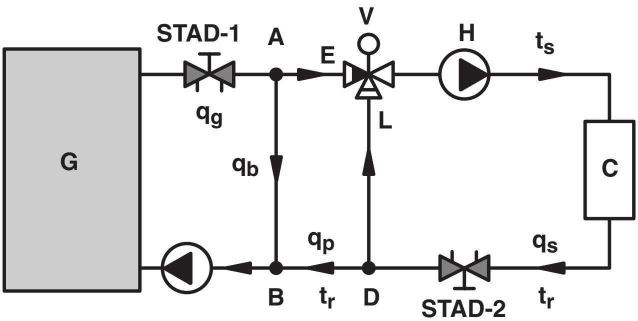
A three-way valve can be represented by two two-way valves working in opposition.

The valve VE represents the control port. Its pressure drop for the design flow = ΔpV. If the circuit flow qs is constant, the pump pressure head H is constant, as are pressure losses in the circuit. The result is that the pressure difference ΔpDC is constant. This pressure difference is applied to valve VE when it is closed. By definition, the valve authority is given by the ratio Δp (valve open) to Δp (valve closed). Thus:

$$ß' = \frac{\Delta pV}{\Delta pDC} = \frac{\Delta pV}{\Delta pV + \Delta pG}$$

This authority is equal to 0.5 or more if ΔpV > or = ΔpG. This means that the pressure loss across the three-way valve must be at least equal to the pressure loss in the variable flow circuit G, including the pipes.

The circuit below gives a constant flow in the production unit and the three-way valve authority is close to 1.



A bypass AB and a primary pump can give a constant production flow and a three-way valve authority close to 1.

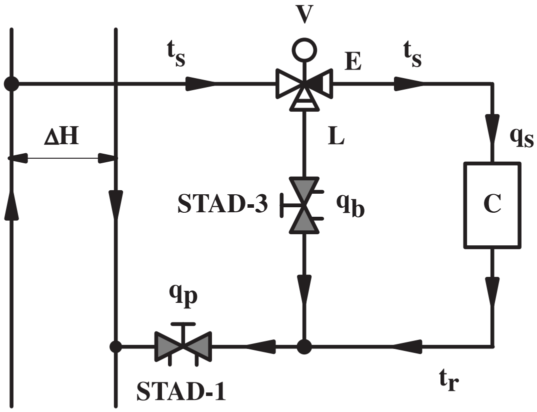
In fact, the three-way valve draws from and discharges into the bypass AB which actually forms a virtual production unit with no pressure loss. In this case the authority of the three-way valve is:

$$ß' = \frac{\Delta pV}{\Delta pV + \Delta pDBAE}$$

Since ΔpDBAE is low, the authority of the control valve is close to 1.

B.2 In diverting function

When used in diverting function, a three-way valve can supply a circuit at variable flow and constant water supply temperature while keeping the primary flow practically constant.



Diverting valve mounted in a diverting circuit.

The primary flow is transferred through port E or bypassed through port L. In principle, it's constant. The balancing valve STAD-1 located in the constant-flow pipe, limits the flow by creating a constant pressure loss.

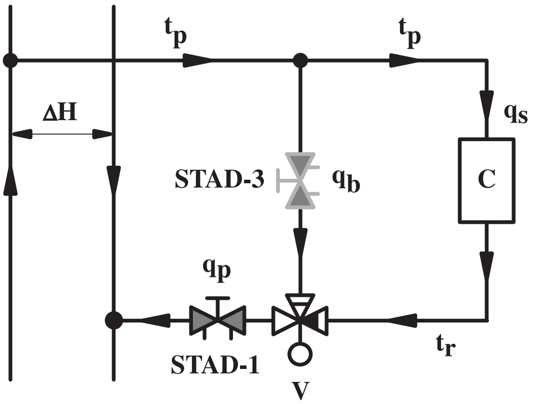
Since the three-way valve in a distribution circuit is used to keep the primary flow constant to avoid interactivity between circuits, it would be logical to take whatever action is necessary to ensure that this purpose is actually satisfied.

This is done by placing a balancing valve STAD-3 in the bypass in order to create a pressure drop equivalent to that of C for the same flow. In this way, the primary flow is unchanged if port E or port L is fully open since the hydraulic resistances in series with these ports have the same value.

The most important balancing valve is this STAD-1. STAD-3 can be omitted if ΔpC is lower than 0.25 times ΔH.

Note:

Three-way valves are usually designed to perform a mixing function: two inputs and one output. Their use in diverting function with one input and two outputs will generate water circulation through the valve in the direction opposite to the planned. For some valves, this reversal may lead to a significant increase in the noise level and a valve chattering phenomenon.



Diverting circuit using a three way mixing valve

This is why a diverting function with a three-way mixing valve is obtained by placing the valve in the return circuit as shown in the figure. This then provides the same function, while respecting the water circulation direction through the valve.

In both cases the valve authority is:

$$ß' = \frac{\Delta pV}{\Delta pV + \Delta pC}$$

To obtain an authority of at least 0.5, the pressure loss in the three-way valve must be equal to or greater than the pressure loss in the terminal unit C.

How to set the BPV to ensure the minimum pump flow

In some cases a BPV modulating relief valve is installed to ensure a minimum flow to protect the pump, as in the figure.

If this minimum flow is for instance 10% of design flow, the pressure loss in the balancing valve STAD-2 is only 1% of the pressure loss at design flow. Normally, this is a far too low value to allow precise measurement. So how can we measure such a small flow as qsmin?

The following method can be used:

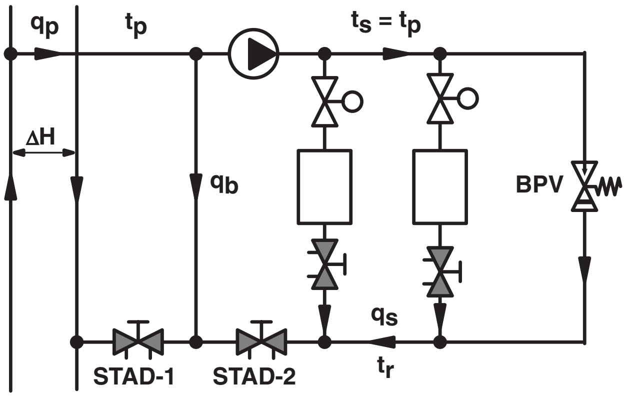
a) Find out which handwheel setting of STAD-2 that will create 3 kPa for the minimum pump flow qsmin, for instance 10% of design flow. Use the CBI, or a TA nomogram, to find the correct setting.

b) Adjust STAD-2 to this setting temporarily. Close the two-way control valves.

c) Open the BPV slowly until you obtain the minimum pump flow qsmin in STAD-2.

d) Reopen STAD-2 to its preset position.

When the coil control valves close and the flow qs goes below the specified minimum flow qsmin, the BPV opens. The BPV then bypasses a flow qsmin for as long as the flow qs in the coil control valves remains below qsmin.



This method is only applicable if the flow measurement device is of the variable orifice type, as the STAD balancing valve.

Definitions

Authority: See Appendix A two-way valves and Appendix B three-way valves in this manual.

Automatic: Anything that executes specific operations without human intervention.

Balancing: Measurement and control process to obtain the required flows in hydraulic circuits.

Circuit: A number of hydraulic components connected by piping forming a continuous and closed path through which a fluid, normally transporting energy, can circulate.

Compatibility: Two circuits are hydraulically compatible if water flows in each circuit are matched to obtain the required temperatures.

Control loop: A closed loop including a sensor, controller, actuator and a controlled system, in order to keep the controlled physical variable at a set value.

Design value: The plant is calculated in certains 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).

Differential pressure: The pressure difference measured between two points.

EQM: Equal percentage valve characteristic modified to avoid discontinuity of flow near the shut position.

Indoor climate: The indoor climate in a room is defined by a set of physical variables (ambient temperature, radiating surface temperatures, circulating air speeds, relative humidity) which in combination give a sensation of comfort or discomfort.

Instability: A control loop is said to be unstable if the controlled variable permanently oscillates without finding an equilibrium position. Except at extreme loads (zero or maximum), an On-Off controller is essentially unstable.

Interactivity: Two circuits are said to be interactive when variation of the water flow in one modifies the water flow in the other.

Interface: The point at which two circuits meet, and where there is generally an energy exchange. The two circuits are generally distinguished by calling one the primary circuit and the other the secondary circuit. In principle, energy is transferred from the primary circuit to the secondary circuit under normal operating conditions.

Pressure drop: The loss of pressure determined by friction in pipes or in any other element through which a fluid circulates.

Pump head: Differential pressure generated by a pump and applied to a circuit to create a forced circulation of water or another fluid. It is normally expressed as a head of liquid.

Relief valve: An automatic pressure relieving valve that opens in proportion to the increase in pressure over the setpoint. It may perform one, two or three of the following functions: (1) stabilize the differential pressure across the control valves, (2) ensure a minimum flow to protect the pump, and (3) limits the temperature drop or rise in the pipes.

Set value: Used in a control loop and selected, normally by the User, to achieve a given purpose. The controller is required to maintain this physical variable as close as possible to the set value, despite the various disturbances which may influence the controlled system.

Temperature drop or rise: The difference in fluid temperature, measurable between the supply pipe and the return pipe for a terminal or for a production unit, or more generally any temperature difference between two points in the plant.

Terminal (terminal unit): Any device which directly or indirectly transmits heat or cold into a room (radiator, heating or cooling coil).

Total balancing: A general concept designed to produce optimum indoor climatic conditions by applying a dynamic procedure to the hydraulics, which is one of the optimization factors; this procedure includes five steps:
1. Make sure that the control concept is compatible with the hydraulic design.
2. Choose suitable controllers and control valves with correct characteristics.
3. Make sure that control valves always operate under reasonable working conditions.
4. Obtain required flows in terminal units under design conditions and potentially at least these flows under other conditions.
5. Guarantee flow compatibility at all interfaces.

Total pressure: Sum of the static pressure and the dynamic pressure at the point considered.

Valve characteristic: This is the relation set up between the water flow through the valve and the valve lift, assuming that the differential pressure across the valve remains constant. The flow and the lift are expressed as a percent of their maximum value.

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2

BALANCING OF DISTRIBUTION SYSTEM

The most efficient methods for balancing waterflows in distribution systems for heating and cooling systems.

Franz Josef Spital, Austria

"Balancing of Distribution Systems" is Manual No. 2 in the TA Hydronics series of publications for HVAC practitioners. Manual No. 1 deals with balancing control loops. Manual No. 3 deals with balancing radiator systems. Manual No. 4 deals with stabilising of differential pressure.

Please note that this publication has been prepared for an international audience. Since the use of language differs somewhat from country to country, you may find that some of the terms and symbols are not the same as those you are used to. We hope this does not cause too much inconvenience.

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— 2nd edition —

Content

This manual deals with the hydronic balancing methods.
For balancing of control loops, please see TA manual N° 1.
For balancing of radiators systems, please see TA manual N° 3
For stabilising the differential pressure across the circuits, please see TA manual N° 4.

1. Why balance?

(More about in appendix F)

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, particularly after load changes.
- Room temperatures keep swinging, particularly at low and medium loads even though the terminals have sophisticated controllers.
- Although the rated capacity of the production units may be sufficient, design capacity can't be transmitted, particularly during start-up after weekend or night set back.

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 under design conditions. The only way to get design flows is to balance the plant. Balancing means adjusting the flow by means of balancing valves. This has to be done in five respects:

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. 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 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.

4. Balancing with manual balancing valves gives the possibility to detect most of the hydronic abnormalities and to determine the pump oversizing. The pump head can be adjusted at the correct value, optimising the pumping cost.

5. When the plant is balanced a central controller or optimiser can be used as all rooms react the same way. Moreover, when the average room temperature deviates from the design value, due to absence of balancing, a costly uncomfort may be the result as explained hereafter.

(More about in appendix F)

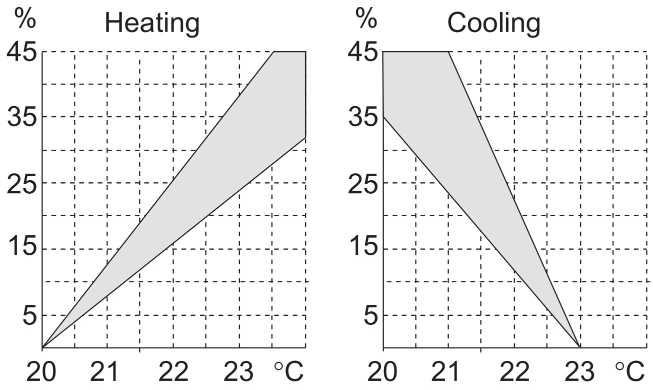


Fig 1.1: Percentage increase in energy costs for every degree too high or too low, relative to design comfort room temperature.

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 complaining 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 percent in mid Europe (12% in the south of Europe). One degree below 23°C increases cooling costs by 15 percent in Europe (Fig 1.1).

A HVAC system is designed for a specific maximum load. If the plant cannot deliver full capacity in all circuits because it is not balanced for design condition, the investments for the entire plant are not realised. Control valves are fully open when maximum capacity is required and thus cannot manage this situation. Furthermore, control valves are generally oversized and they cannot contribute to balancing. Hydronic balancing is thus essential and represents typically less than two percents of the total HVAC system.

Each morning, after a night set back, full capacity is required to recover the comfort as soon as possible. A well-balanced plant does this quickly. If a plant starts up 30 minutes quicker, this saves 6 percent of the energy consumption per day. This is often more than all distribution pumping costs.

But an important consideration is to compensate for pump oversizing. Balancing valves adjusted with the Compensated Method or the TA Balance Method reveal the degree of pump oversizing. All the overpressure is shown on the balancing valve closest to the pump. Corrective action (e.g. reduce the pump speed or trim the impeller) can then be taken.

Hydronic balancing requires the correct tools, up to date procedures and efficient measuring units. A manual balancing valve is the most reliable and simple product to obtain the correct flows in design conditions. It also allows the flows to be measured for diagnostic.

Three things are necessary:

Flow measuring and regulating devices,
measurement instrument
and a balancing procedure.

Flow measuring and regulating devices

These are:
Balancing valves which are both variable orifice and regulating valves or
Orifice devices with an independent regulating valve.

There is a great difference between balancing valves of different makes. This translates into an equally great difference in the accuracy of indoor climate control, in energy savings—and in the time, cost and effort required to do an adequate balancing job.

TA, whose products are used worldwide, cater for all the different market requirements and offer both fixed and variable flow measuring devices and regulating valves.

These are some of the distinguishing features of TA products:

STAD
STAD balancing valve
15 to 50 mm

STAF
STAF balancing valve
20 to 300 mm

STAP
STAP Differential pressure controller
15 to 100 mm

Balancing valves and orifice devices

- Flow precision for valves better than +/- 10% and for fixed orifices better than +/- 5%.
- Sizes up to 50 mm have four full turns from open to closed position. Larger sizes have eight, twelve or sixteen full turns.
- The valves are available with internal threads, with flanges, with welded or soldered valve ends, with grooved ends and with compression fittings.
- Sizes up to 50 mm are made of Ametal®, probably the only pressure die casting alloy that meets the world's toughest demands for resistance to dezincification.

Differential pressure controller

- Adjustable set point.
- To stabilize the differential pressure across the control valves and/or circuits.

Measurement instrument

Measuring is required in order to really know that design flows are achieved and also to find what differential pressures that are applied in different parts of the plant. It is also a good tool for trouble-shooting and system analyses.

The balancing instrument CBI from TA has all necessary features to fulfil these demands, eg:

- Measures and documents differential pressure, flow and temperature of STAD, STAF, STAP/STAM and other valves from TA.
- Programmed to calculate presetting values for balancing and also the TA Method and TA Balance.
- Two-way communication with PC.
- Corrects the calculations for antifreeze agents.
- Large storage capacity - can handle 1000 valves and 24 000 values when logging.
- Graphic display making it possible to assign plain-language names for plants and valves.

Proportional relief valve

In variable flow system, a TA BPV valve can be used to perform three distinct functions:

- ensure a minimum flow to protect the pump.
- reduce the temperature drop in pipes.
- limit the differential pressure on the terminal circuits.

The BPV has a shut-off function and preset point of 10–60 kPa. 15 to 32 mm (1/2” to 1 1/4”)

Balancing of the water flows shall be carried out prior to the finetuning of control equipment.
Prepare the balancing carefully. This gives a better final result at less time input.

Acquire drawings and study them for a sound understanding of the principles of the plant's function. Make an on-site inspection to avoid losing time over practical problems such as searching for a key to a locked room or trying to find a non-existent balancing valve.

For detailed instructions, see Appendix E.

3.1 Plan the balancing at your desk

Study the plant drawing carefully

Make an initial study of the plant drawings in order to understand design and operation principles. Identify control loops, distribution system and balancing valves. Divide the plant in modules as explained in section 3.2.

In a four-pipe distribution system, you should prepare separate drawings for the heating circuit and the cooling circuit. Sometimes it may be a good idea to draw up a circuit scheme of principle with all details eliminated that do not concern the balancing work.

Select a suitable balancing method

When you adjust the flow with a balancing valve, the pressure loss changes in the valve and pipe line. Then the differential pressure across other balancing valves also changes. Hence, each flow adjustment disturbs the flow in already adjusted valves. In other words, the circuits are interactive. The main difference between different balancing methods is how they compensate for interaction between circuits. Some methods do not compensate at all. This means the balancer will have to set the same balancing valve several times until the flow finally converges towards the desired flow. Other methods compensate directly or indirectly. Three such methods are the Proportional Method, the TA Compensated Method and the TA Balance Method, which are described in this manual.

The Compensated Method is a further development of the Proportional Method, giving better results with less time input. The TA Balance is the easiest method requiring only one setter and one measuring unit to balance a complete installation.

However, none of these methods can be used to balance distribution systems designed according to the reverse return principle. In that case, you must use an iterative method. That is: go through the entire plant several times and adjust the flows ”by ear” until they correspond reasonably well with design flows, or calculate, manually or by computer, the correct preset values for the balancing valves.

TA manual No. 1 ”Balancing of Control Loops” provides efficient step-by-step methods for the balancing of 23 control loops for two-way and three-way control valves.

3.2 Divide the plant into modules

Theory and practice

In theory, it is sufficient with one balancing valve per terminal unit to create the correct repartition of flows in the distribution system. But this requires that the preset value for all the balancing valves are calculated, that these calculations are made correctly, and that the plant is realised according to the drawings.

If you change one or several flows, all other flows are more or less affected, as previously mentioned. It may require a long and tedious series of corrections to get back to the correct flows.

In practice, it is necessary to divide larger systems into modules and install balancing valves in such a way that readjusting only one or a few balancing valves can compensate a flow adjustment anywhere in the system.

The law of proportionality

The terminals in the figure 3.1 form a module. A disturbance external to the module causes a variation in the differential pressure across A and B. Since the flow depends on the differential pressure, the flows in all terminals change in the same proportion.

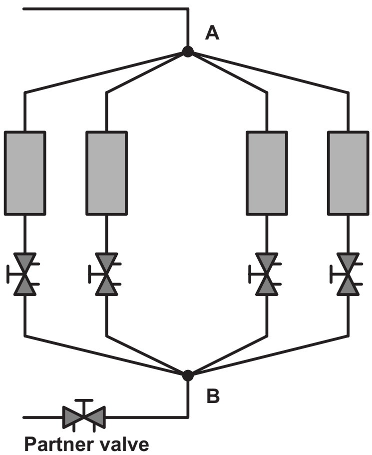


Fig 3.1: An external disturbance has the same effect on each terminal in the module.

The flow through these terminals can therefore be monitored through measurement of the flow in just one of them, which can serve as a reference. A balancing valve common to all terminals can compensate for the effect of the external disturbance on the terminal flows in the module. We call this common valve the Partner valve.

However, terminals are normally connected as in figure 3.2. The water flow through each terminal depends on the differential pressure between A and L. Any modification of this pressure affects the flow in each terminal in the same proportion.

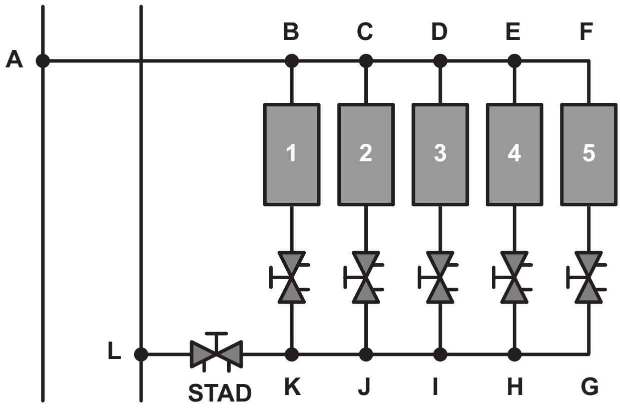


Fig 3.2: A branch with several terminals forms a balancing module. STAD is the Partner valve, which can compensate for external disturbances on the circuits already balanced.

But what happens if we create a disturbance that is internal to the module, for instance by closing the balancing valve of terminal 3?

This will strongly influence the flows in pipes lines CD and IJ, and thus the pressure loss in these pipe lines. The differential pressure between E and H will change noticeably, which will affect the flows in terminals 4 and 5 in the same proportion.

The fact that terminal 3 is closed has little effect on the total flow in the pipe lines AB and KL. The pressure losses in these pipe lines change very little. The differential pressure between B and K is changed only somewhat and terminal 1 will not react to the disturbance in the same proportion as terminals 4 and 5. Thus, the law of proportional flow change does not apply for internal disturbances (as shown in figure 3.3).

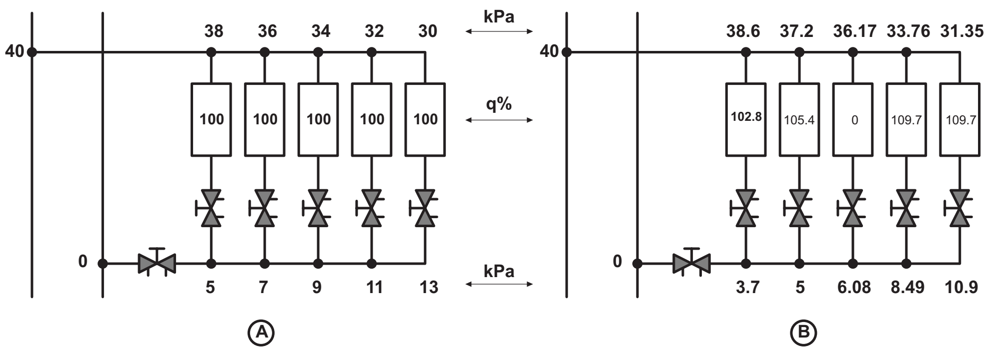


Fig 3.3: At an internal disturbance, the flows do not change in proportion to each other.

However, the water flows change in proportion in a module only if all the pressure drops depend on the flow q according to the same relation everywhere in the module. This is not true in reality because for the pipes the pressure drop depends on $q^{1.87}$, while it depends on q^2 in valves. For low flows, the circulation can become laminar and the pressure drop becomes linearly proportional to the flow. The law of proportionality can be used only to detect deviations around design values. This is one of the reasons why the most accurate balancing method is the Compensated Method described in chapter 5 as the design flows are maintained during the balancing process of each module.

A module can be a part of a larger module

When the terminals on a branch are balanced against themselves, you may see the branch as a "black box", i.e. a module. Its components react proportionally to flow adjustment external to the module. The Partner valve can easily compensate such disturbances.

In the next step, the branch modules are balanced against each other with the riser balancing valve as the Partner valve. After this, all modules on the riser form a larger module, whose flow can be adjusted with the riser's balancing valve. Finally, the risers are balanced against each other with each riser as a module and the balancing valve on the main pipe line as the Partner valve.

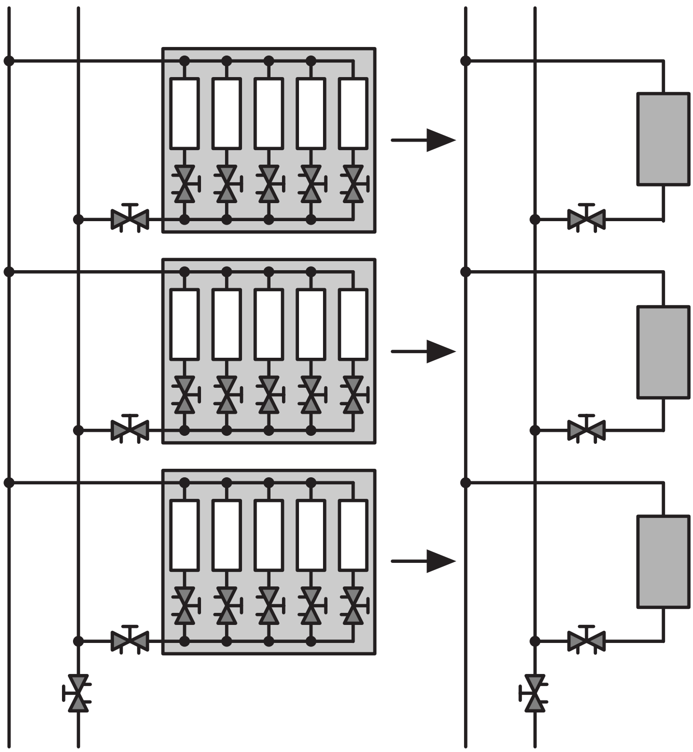


Fig 3.4: Each branch on a riser forms a new module.

What is optimum balancing?

Figure 3.5 shows two modules. The numbers indicate the design pressure loss in each terminal and the pressure loss in each balancing valve. Both modules are balanced. In both cases the differential pressure on each terminal is the required one to obtain design flow. The pressure losses are differently distributed between the balancing valves of the terminals and the Partner valve.

Which balancing is the better of the two?

Optimum balancing means two things: (1) that the authority of the control valves is maximised for exact control, and that (2) pump oversizing is revealed so that pump head and thereby pumping costs can be minimised. Optimum balancing is obtained when the smallest possible pressure loss is taken in the balancing valves of the terminals (at least 3 kPa to allow precise flow measurement). Any remaining excess pressure is taken in the Partner valve.

Balancing to obtain pressure losses as in (b) in the figure is thus the best, since the pressure loss is then the lowest admissible in all balancing valves on the terminals to obtain the design flows. Note that optimum balancing is only possible when the required Partner valves are installed.

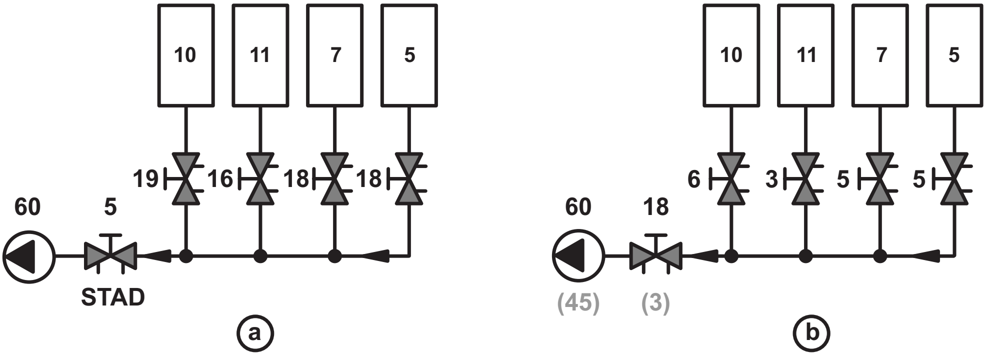


Fig 3.5: A set of terminals can be balanced in many ways, but only one is the optimum.

The Partner valve reveals the excess of differential pressure. The pump speed for instance can be decreased correspondingly and the partner valve reopened. In example "b", the pressure drop in the Partner valve and the pump head can be reduced both by 15 kPa, decreasing the pumping costs by 25%.

Where balancing valves are needed

The conclusion is that balancing valves should be installed to split the system in modules that can be balanced independently of the rest of the plant. Thus, each terminal, each branch, each riser, each main and each production unit should be equipped with a balancing valve.

It is then simple to compensate for changes relative to the drawings, for any construction errors, and for oversizing. This saves time and allows optimum balancing. Furthermore, the plant can be balanced and commissioned in stages, without having to rebalance when the plant is completed.

The balancing valves are also used for troubleshooting and shut-off during service and maintenance.

Accuracy to be obtained on flows

We have considered the advantages of hydronic balancing. Before studying balancing procedures, the precision with which flows have to be adjusted has to be defined.

In practice, the flow adjustment precision to be achieved depends on the precision to be obtained on the room temperature. This precision depends also on other factors such as the control of the supply water temperature and the ratio between the required and installed coil capacity. Some specifications stipulate a required water flow accuracy of between +0 and +5%. There is no technical justification for this severity. This requirement is even more surprising that little care appears to be taken about the actual temperature of the supply water to remote units. Particularly, in the case of variable flow distributions, the supply water temperature is certainly not the same at the beginning and at the end of the circuit, and the influence of this water temperature is not negligible. Furthermore, water flows are frequently calculated based on required capacity and they are rarely corrected as a function of the really installed capacity. Oversizing of the terminal unit by 25% should normally be compensated by a water flow reduction in the order of 40%. If this is not done, there is no point in adjusting the water flow to within 5% of accuracy while the required water flow is defined with an initial error of 40%.

An underflow cannot be compensated by the control loop, and has a direct effect on the environment under maximum load conditions; it must therefore be limited. An overflow has no direct consequence on the environment since in theory, the control loop can compensate for it. We may be tempted to accept overflows, especially when they have little effect on the room temperature. This would neglect the pernicious effect of overflows. When the control valves are fully open, for example when starting up the plant, the overflows produce underflows elsewhere and it is impossible to obtain the required water temperature at high loads, due to incompatibility between production and distribution flows. Overflows must therefore also be limited. This is why it is logical to penalise underflows and overflows with the same factor and to adopt a general precision rule in the form ±x%.

Fortunately, when the flow is situated close to the design value, it has no dramatic effect on the room temperature. By accepting a deviation of ± 0.5°C on the room temperature at full load due to water flow inaccuracy, the value of x, with a certain safety factor, is in the order of:

$$x = \frac{\pm 100\,(t_{sc} - t_{ic})}{(t_{sc} - t_{rc})(t_{ic} - t_{ec} - a_{ic})} \quad \text{with}$$

t_{sc} : Design supply water temperature.
t_{ic} : Design room temperature.
t_{rc} : Design return water temperature.
t_{ec} : Design outdoor temperature.
a_{ic} : Effect of internal heat on the room temperature.

Examples:

Heating- t_{sc} = 80°C; t_{rc} = 60°C; t_{ic} = 20°C; t_{ec} = -10°C; a_{ic} = 2°C; x = ±10%.
Cooling- t_{sc} = 6°C; t_{rc} = 12°C; t_{ic} = 22°C; t_{ec} = 35°C; a_{ic} = 5°C; x = ±15%.

Variations in the differential pressure across a circuit change the flow in the circuit terminals in the same proportion. This fundamental principle is the basis for the Proportional Method.

The Proportional Method is shortly described hereafter as the Compensated Method 'chapter 5' or the TA Balance Method 'chapter 6' progressively replace it. For more information, please see the TA Handbook "Total Hydronic Balancing"- second edition 1997- section 5.4.

We will just examine step by step the balancing of one branch of one riser.

1. Measure the flow in all terminals on the selected branch, with the branch balancing valve (STAD-1.2.0) fully open.

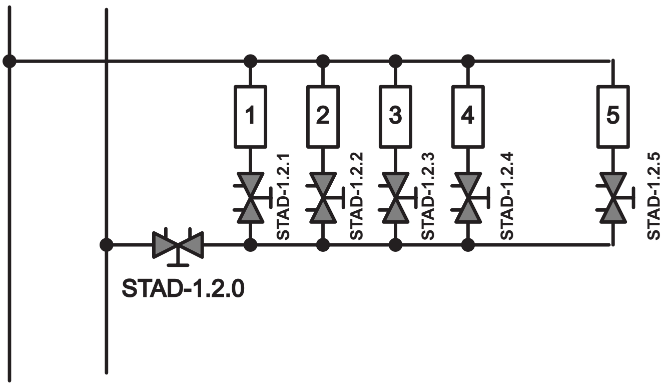


Fig 4.1: Balancing of terminals on a branch.

2. For each one of the terminals, calculate the flow ratio λ: measured flow/design flow. Identify the terminal with the lowest flow ratio λmin. Call it "index unit". If the terminals have the same pressure loss for design flow, terminal 5 normally has the lowest flow ratio since it receives the smallest differential pressure. If the terminals do not have the same pressure loss, any of them may have the lowest flow ratio.

3. Use the balancing valve of the last terminal on the branch as the Reference valve (STAD.1.2.5 in the figure 4.1).

4. Adjust the Reference valve STAD-1.2.5 so that λ5=λmin. Lock STAD-1.2.5 to this setting (screw the inner spindle to stop). Connect a CBI for continuous flow measurement.

5. Set STAD-1.2.4 so that λ4=λ5. This will change the flow ratio λ5 somewhat. If the setting of STAD-1.2.4 changes the flow in the Reference valve by more than 5%, then STAD-1.2.4 must be readjusted so that λ4 becomes equal to the new value of λ5. Lock STAD-1.2.4 to this setting.

6. Adjust the flow in all terminals on the branch. Work your way against the pump according to step 5 above. When STAD-1.2.2 is adjusted, the flow ratio λ5 changes, but λ3 and λ4 remain equal to λ5. Terminal 3, 4 and 5 therefore remain balanced relative to each other. This is the reason why the last terminal is used as the reference. When all terminals are balanced relatively to each other, it is possible to adjust the Partner valve STAD-1.2.0 so that λ5=1. All the other flow ratios λ4, λ3, λ2 and λ1 would then become equal to 1. However, do not carry out this operation since it will be done automatically as you perform the very last balancing operation for the plant.

7. Repeat the same procedure for all branches of the same riser.

Note: Instead of controlling the flow ratio to the Reference valve (circuit 5), it can be done on the last balancing valve adjusted. For instance, after setting the balancing valve of circuit 2, the new flow ratio for all balancing valves of circuits 3-4 and 5 is the same and can be measured on the balancing valve of circuit 3 instead of going to the reference (circuit 5). This can save time for the balancer who has to use two CBI (CBIa and CBIb).

When the circuit 3 is set, the CBIa remains on it. The balancer goes to circuit 2 and adjusts it, with CBIb, for the correct flow ratio. He goes back to circuit 3, he measures the new flow ratio, and removes the CBIa. He readjusts now the flow of circuit 2 and, without removing the CBIb put on this circuit, he goes to circuit 1 with CBIa and so on...

Remember that proportional balancing is only valid when the flow ratios remain close to one (see remark at the end of section 3.2), this condition is fulfilled only with the Compensated Method.

The Compensated Method is a further development of the Proportional Method, with three main advantages:

Staged commissioning: *You can balance the plant in stages as construction goes on, without having to rebalance the entire building when it is completed.*

Quicker commissioning: *It reduces time consumption significantly since it is not necessary to measure the flows in all balancing valves and calculate all flow ratios. It also requires just one flow adjustment at each balancing valve.*

Pumping costs may be minimised: *When balancing is finished, you can read off the pump oversizing directly on the main balancing valve. The pump head may be reduced correspondingly. Frequently, large energy savings can be made, particularly in cooling plants.*

5.1 A development of the Proportional Method

The Compensated Method is based on the Proportional Method, but is further developed in one essential aspect: Using the Compensated Method, *the flow ratios are automatically kept equal* to 1 throughout the balancing process of a module (see remark at the end of section 3.2).

a) Staged commissioning

- The plant may be divided in modules. This means that the plant can be commissioned in stages, as construction goes on, and no rebalancing of the entire building is required after completion.

b) Quicker commissioning

- No first scan to measure the flows in all branches and risers. No calculation of flow ratios to determine the starting point of balancing.
- Balancing can start at any riser (although you should close the risers you are not balancing).
- No worrying about causing a too high flow for the main pump. No worrying about the differential pressure being too small to produce measurable flows.
- Only one flow adjustment at each balancing valve is required.

c) Pumping costs may be minimised

- The Compensated Method automatically minimises pressure losses in the balancing valves. The main balancing valve reveals any oversizing of the main pump. The pump may often be exchanged for a smaller one.
- The set point of a variable speed pump can be optimised.

5.2 Reference valve and Partner valve

When the flow is adjusted by a balancing valve, pressure losses change in the valve and pipe line, thereby changing the differential pressure across other balancing valves. Flow adjustment in one balancing valve thus changes the flow in valves that have already been adjusted. This often makes it necessary to adjust the same balancing valve several times over.

The Compensated Method eliminates this difficulty. The flow in each balancing valve is only adjusted once. The method assumes that it is possible to measure the flow disturbance occurring when a balancing valve is adjusted, and that the disturbance can be compensated in some way.

The disturbance is detected on the balancing valve furthest away from the pump, in this module. This balancing valve is called the Reference valve.

A balancing valve acting on the total branch flow, called the Partner valve, compensates for the disturbance. With this valve, the differential pressure across the Reference valve can be reset to its initial value each time a disturbance occurs.

The method begins by adjusting the flow to design value in the Reference valve, according to a particular procedure presented below. The result is a certain differential pressure ΔpR (Fig 5.1), which is to be monitored continuously. The Reference valve is then locked to this setting.

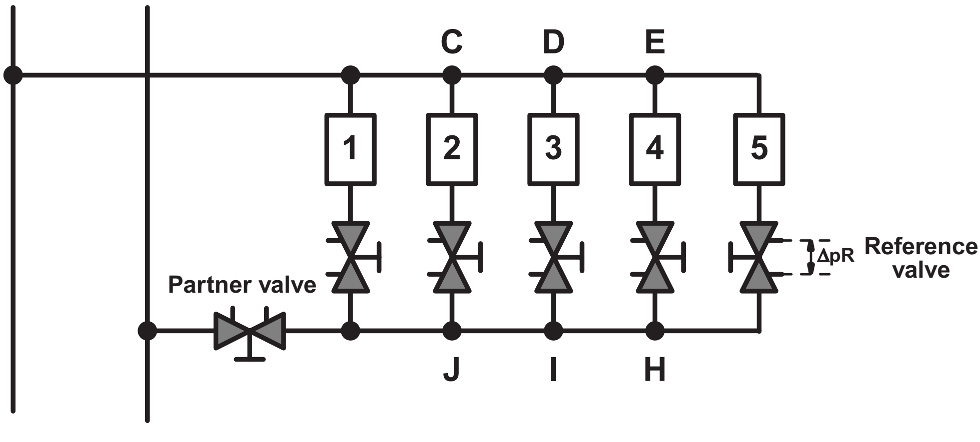


Fig 5.1: The Reference valve is always located in the module and on the terminal furthest away from the pump. The Partner valve determines the total flow in the branch.

Since the flow is now correct, the pressure losses are also correct in terminal 5, its balancing valve and accessories. The differential pressure ΔpEH is therefore correct and we may proceed to adjust the flow in terminal 4.

When the flow in terminal 4 is being adjusted, ΔpR changes slightly in the Reference valve, whose setting is locked. This is an indication of the disturbance from the flow adjustment in terminal 4.

ΔpR must be readjusted to its initial value with the Partner valve. In other words, design flow must be readjusted in the Reference valve by compensating on the Partner valve.

Since the flows in terminal 4 and 5 are now at their design values, the differential pressure ΔpDI across terminal 3 is also equal to the design value. The flow in this terminal may therefore be adjusted.

Adjustment of the flow in terminal 3 creates a disturbance, which is detected at the Reference valve and compensated on the Partner valve. The readjustment of design flow in terminal 5 automatically brings the differential pressure ΔpEH and the flow in terminal 4 to design value.

This procedure works well regardless of the number of terminals on a branch. Adjustments must be carried out by working towards the pump, beginning at the Reference valve. The same procedure is then applied for balancing of risers. The last branch on the riser furthest away from the pump is used as the reference, and the riser's balancing valve becomes Partner valve.

5.3 Setting the Reference valve

Select ΔpR as small as possible but big enough to meet the following two conditions:

• **Minimum of 3 kPa to obtain sufficient measurement accuracy.**
The CBI balancing instrument indicates flow for differential pressures down to 0,5 kPa. However, to decrease the relative influence of the pressure pulsation in the plant on the flow measurement, we recommend $\Delta pR > 3$ kPa.

The Kv value may be calculated for a pressure loss of at least 3 kPa using the formula:

$$Kv = 5{,}8 \times q \text{ (m3/h) or } Kv = 21 \times q \text{ (l/s)}$$

Another and simpler way, is to let the CBI calculate the correct setting of the Reference valve.

• **The pressure drop in the valve fully open and at design flow.**
If the pressure loss is greater than 3 kPa for design flow and the balancing valve fully open, it is obviously not possible to set the Reference valve to create 3 kPa. This represents the second condition on the ΔpR: at least as high as the pressure loss across the fully open balancing valve at design flow. In this case, the balancing valve on the reference is just fully open.

When a suitable ΔpR is selected, preset the Reference valve to create ΔpR for design flow. Use the CBI or a nomogram to find the correct handwheel setting. Then lock the handwheel.

To obtain the selected ΔpR, and thus design flow, adjust the Partner Valve. This is always possible since the other risers are closed and the pressure loss in the main pipe line is small. The available differential pressure is thus higher than normal. The surplus will be taken in the Partner valve.

If the pressure losses differ substantially between the terminals, please refer to section 5.8.

5.4 Equipment needed

Two CBI balancing instruments are needed to measure differential pressures and flows in the balancing valves.

5.5 Balancing terminals on a branch

Select any riser, for instance the one closest from the pump. This ensures a sufficient differential pressure for the selected riser. Select any branch in the riser you have selected. Normally, you do not have to shut any of the other branches of this riser. However, if some branches are provided with a bypass line, which can create short circuits, the flow in these branches has to be limited or these branches isolated.

1. Determine which is the handwheel position of the Reference valve that will give design flow at the selected ΔpR (normally 3 kPa). Use the CBI or a nomogram to find the correct handwheel position.
2. Adjust the Reference valve to this position and lock the valve (turn the inner spindle down to stop).
3. Connect one CBI to the Reference valve.
4. Balancer (1) adjusts the Partner valve to obtain the selected ΔpR in the Reference valve. Information about current value of ΔpR is transmitted to Balancer (1) from Balancer (3) by means of a walkie-talkie for instance. This operation gives design flow in terminal 5. If the selected ΔpR cannot be reached, the cause may be that non balanced terminals on the branch are passing a too high flow. Shut as many of them as required to obtain the selected ΔpR.

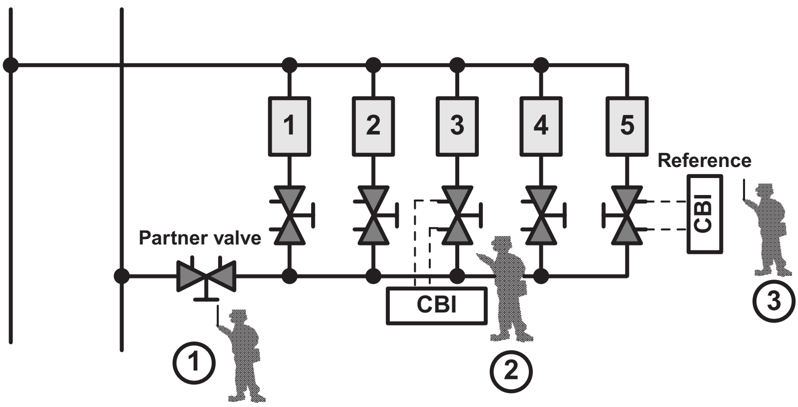


Fig 5.2: Balancing of terminals on a branch.

5. Balancer (2) now adjusts the flow to design in terminal 4 by using the CBI computer function. It calculates which handwheel position that will give design flow. During the whole procedure, balancer 1 continuously readjusts the partner valve to maintain ΔpR to its initial value.

6. Balancer (2) adjusts the flows in each terminal by working successively towards terminal 1, according to step 5 above. All terminals on the branch are now balanced relative to each other, independently of the current differential pressure applied on the module.

Note: Let us suppose working with two balancers (1 and 2) and two CBI (CBIa and CBIb). When adjusting terminal 3 for instance, with CBIa, the balancer can check the change of the flow in terminal 4 (CBIb) instead of going to the reference (terminal 5). He communicates with balancer 1 to readjust the flow at terminal 4, takes back the CBIb put on this terminal and eventually he readjusts the flow at terminal 3. He leaves the CBIa put on terminal 3 and goes with CBIb to terminal 2, following the same procedure and checking the flow evolution at terminal 3. Repeat proceedure for all valves.

5.6 Balancing branches on a riser

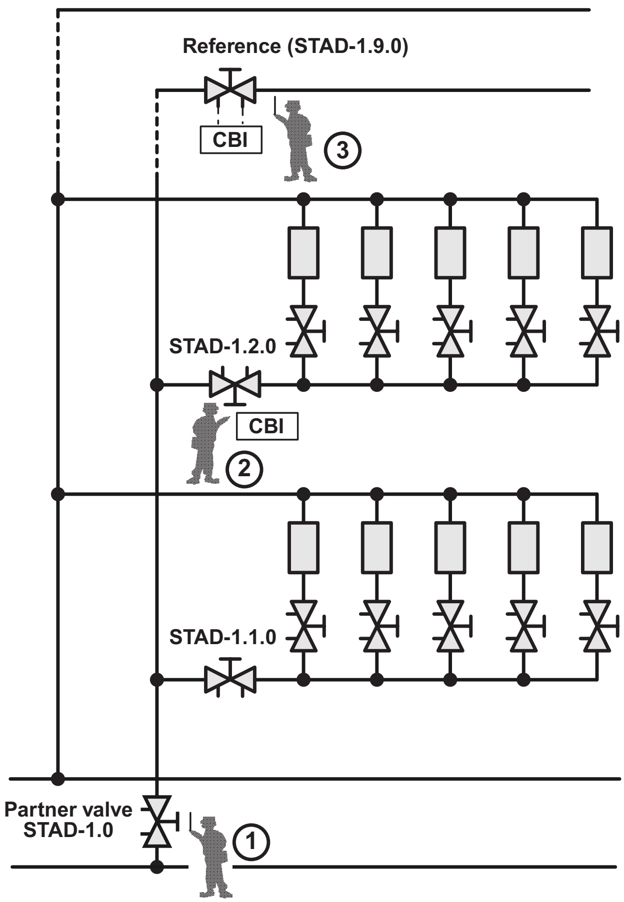


Fig 5.3: Balancing branches on a riser.

1. Find out the handwheel position for the Reference valve STAD-1.9.0 that will give design flow for the selected ΔpR, normally 3 kPa. Use the CBI or a nomogram to find the correct position.
2. Adjust the Reference valve to this handwheel position and lock the valve (turn the inner spindle down to stop).
3. Connect one CBI to the Reference valve.
4. Balancer (1) adjusts the Partner valve to create the selected ΔpR in the Reference valve. This then gives the design flow in the reference branch. If the selected ΔpR cannot be obtained, the cause may be that some branches on the riser are passing a too high flow. Then close as many branches as required to obtain the selected ΔpR.

5. Balancer (2) now adjusts to design flow in branch 1.2.0 by using the CBI computer function. It calculates the handwheel position that will give design flow. During the whole procedure, balancer 1 continuously readjusts the partner valve to maintain the flow in the reference to its initial value.
6. Balancer (2) adjusts the flows in each branch by working successively towards branch 1.1.0 according to the procedure in step 5 above. All branches on the riser are now balanced relative to each other independently of the current differential pressure available on the riser.

5.7 Balancing risers on a main pipe line

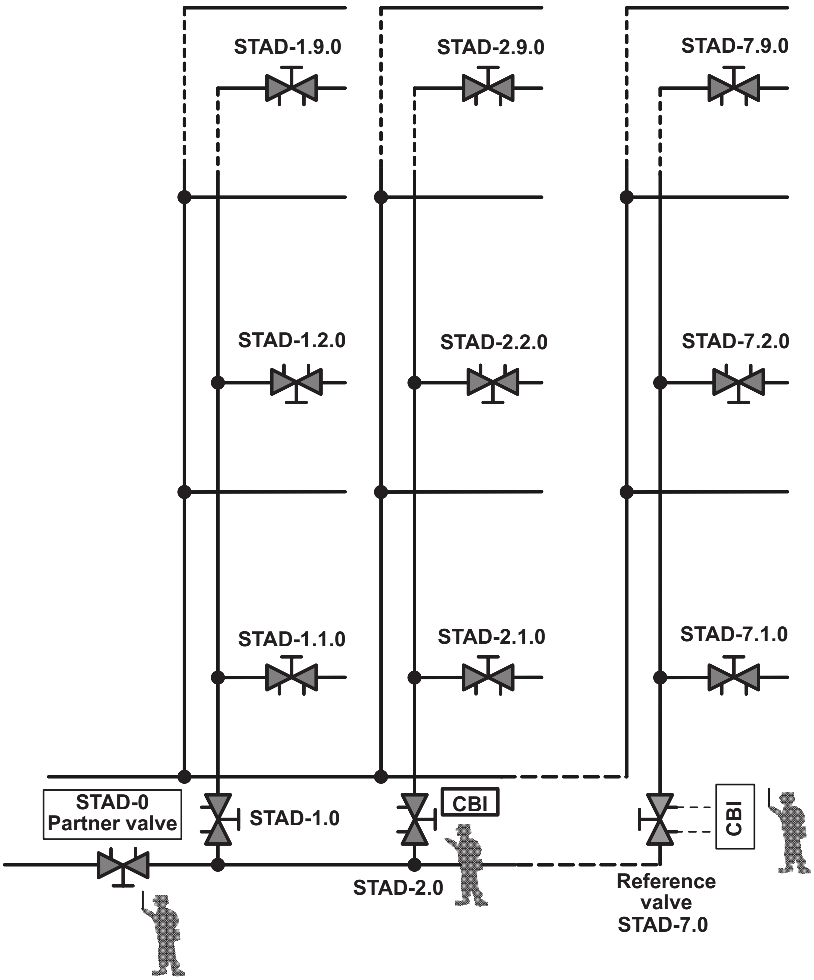


Fig 5.4: Balancing of risers.

The balancing procedure is exactly the same as for balancing of branches on a riser. The Reference valve is now STAD-7.0 and the Partner valve is STAD-0.

When balancing of risers 7.0, 6.0, 5.0 etc., is completed, the entire plant is balanced for design flows and the remaining pressure loss in STAD-0 reveals the pump oversizing. If the excess pressure is large, it may be profitable to change the pump for a smaller one.

When using a variable speed pump, the STAD-0 is not necessary. The maximum speed is adjusted to obtain the correct design flow in the Partner valve of one riser. All the other flows will be automatically at design value.

5.8 Setting the Reference valve when pressure losses differ substantially between the terminals

If the terminals pressure losses differ substantially, a ΔpR of 3 kPa in the Reference valve may not be sufficient to give the necessary differential pressure for the other terminals. This problem is solved in the Proportional Method by using the same flow ratio for the Reference valve as the flow ratio measured in the index circuit. But the Proportional Method often overestimates the ΔpR and balancing is not optimised (unnecessarily high pressure loss in the balancing valves). A way to achieve a suitable value for ΔpR is presented below.

The branch in figure 5.6 has terminals with different pressure losses.

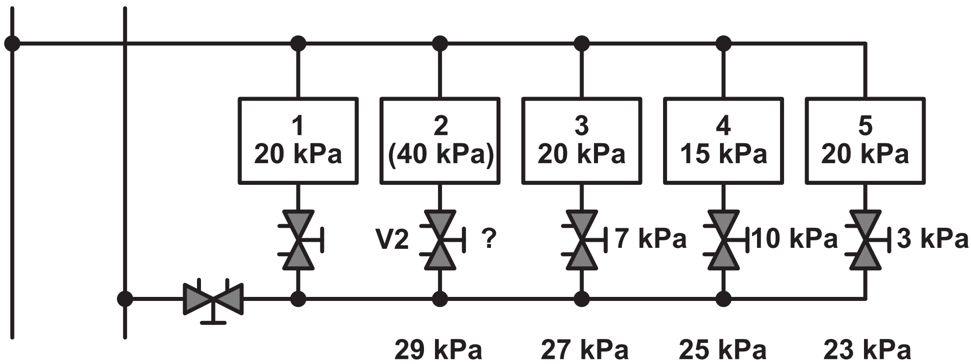


Fig 5.6: If 3 kPa is selected for the Reference valve, the differential pressure may be too low for the index circuit, here terminal 2.

Select ΔpR based as recommended in section 5.3, normally 3 kPa. We call this preliminary value ΔpRo. Proceed with balancing according to the Compensated Method.

When you reach the index circuit, you will note that it is impossible to obtain design flow since the differential pressure is only 29 kPa, while it would take more than 40 kPa to obtain design flow. Perform the following steps:

1. Shut the balancing valve (V2) of the index circuit and readjust the correct flow in the reference with the Partner valve. Measure the differential pressure across V2. Call this value Δpo.
2. Preset V2 so that its pressure drop will be approximately 3 kPa for design flow.
3. Open the partner valve to obtain the design flow in the index circuit.
4. Measure the flow in the reference circuit. Calculate the flow ratio λ = flow measured/design flow.
5. The new value of ΔpR to be set on the Reference valve is given by the formula:

$$\text{New } \Delta pR = \Delta pRo + \Delta po \text{ x } (\lambda^2 - 1)$$

6. Preset the Reference valve to obtain this pressure loss for design flow, and *rebalance the entire branch.*

Compared with fig 5.6, the result of this procedure is given in fig 5.7.

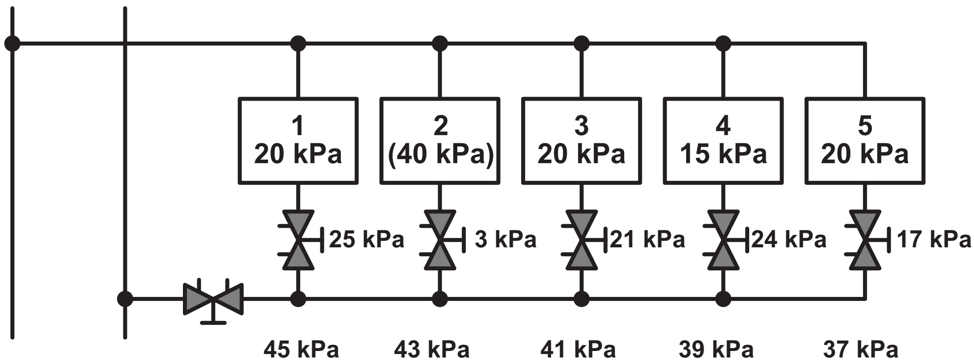


Fig 5.7: Differential pressure across circuits and pressure losses in balancing valves and terminals.

The TA Balance Method is a computer program built into the CBI balancing instrument with the same three main advantages of the Compensated Method plus the possibility for one man and one CBI to balance an entire system:

These advantages are the following:

***Staged commissioning:** You can balance the plant in stages as construction goes on, without having to rebalance the entire building when it is completed.*
Quicker commissioning: It reduces time consumption significantly since it is not necessary to measure the flows in all balancing valves and calculate all flow ratios. It also requires just one flow adjustment at each balancing valve.

***Pumping costs may be minimised:** When balancing is finished, you can read off the pump over-sizing directly on the main balancing valve. The pump pressure may be reduced correspondingly. Frequently, large energy savings can be made, particularly in cooling plants.*

***One man and one instrument:** After having carried out pressure and flow measurements, the program calculates the correct settings of the balancing valves in order to achieve the desired flows.*

The program assumes that the plant is divided into modules. Let us remember that a module is created out of several circuits connected in direct return to the same supply and return pipes. Each circuit has its own balancing valve and the module has a common balancing valve called the Partner valve.

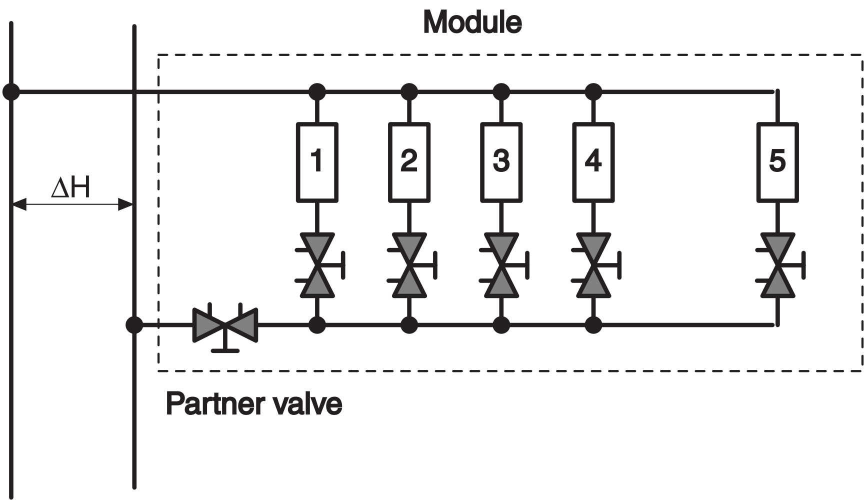


Fig 6.1: A module is created of several circuits connected to the same supply and return pipes.

6.1 Preparing the procedure

During the measurements, the differential pressure ”ΔH”, at the inlet of the module, should be constant. The value of this ”ΔH” is not important unless there is insufficient pressure to obtain good measurements. For this reason, the risers or modules not yet balanced, which can create big overflows, have to be isolated. To be sure that the pressure drops in the balancing valves will be sufficient to obtain a correct measurement, set the balancing valves on 50% opening (STAD = 2 turns), or at the precalculated positions if any. The Partner Valve of the module to be balanced must be fully open during the procedure.

The TA Balance Method demands that the valves be numbered according to the figure 1. The first valve after the Partner valve must be number one, with following valves being numbered successively (See Fig 6.1). The Partner valve is not numbered.

6.2 The procedure

Measure one module at a time.

CBI gives directions on the display of each step of the procedure.

For each valve in the module, in any order, the following procedure is applied:

1. Enter the reference number, type, size and current position (e.g. 1, STAD, DN 20, 2 turns).
2. Enter the desired flow.
3. A flow measurement is then automatically performed.
4. Shut the valve completely.
5. A differential pressure measurement is automatically performed.
6. Reopen the valve to its original position.
7. When all the balancing valves in the module have been measured, the CBI requires the measurement of the Δp across the Partner valve in the fully shut position.

When all these procedures have been carried out, the CBI calculates the correct handwheel setting for the balancing valves within the module. Adjust the balancing valves to these settings.

The CBI has ”discovered” the index circuit (the circuit requiring the highest differential pressure) and has given the index balancing valve the minimum pressure drop that is necessary to measure the flow correctly. This value is normally 3 kPa, but can be changed if you want. The settings of other balancing valves are calculated automatically to obtain a relative balancing of the elements in the module. These settings do no depend on the current differential pressure ΔH applied on the module.

At this moment, the design flows are not yet achieved. This will happen when the Partner valve has been adjusted to its correct flow. This operation is carried out later on in the procedure.

6.3 Balancing the modules of a riser between themselves

When all the modules in one riser have been balanced individually, these modules are balanced between themselves. Each module is now looked upon as a circuit whose balancing valve is the Partner valve in the module. The balancing procedure consists of calculating the setting of the Partner valves of modules 1, 2 and 3 of the riser, using the TA Balance Method.

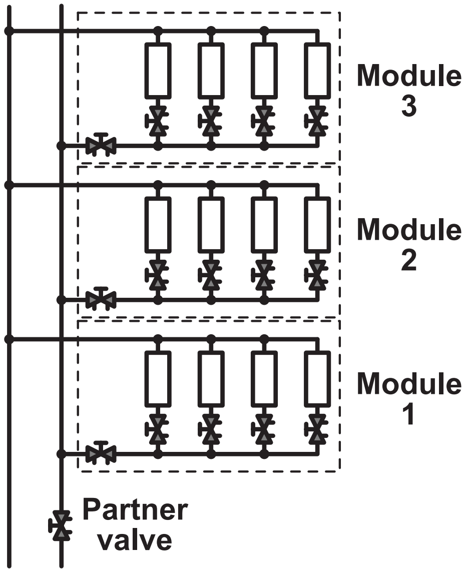


Fig 6.2: The riser module is created of modules 1, 2 and 3 when these are calculated and set.

This riser module should now be measured and calculated in the same way as described earlier.

6.4 Balancing the risers between themselves

When all risers have been balanced individually, they constitute a module. The Partner valve of this module is the main balancing valve associated with the pump.

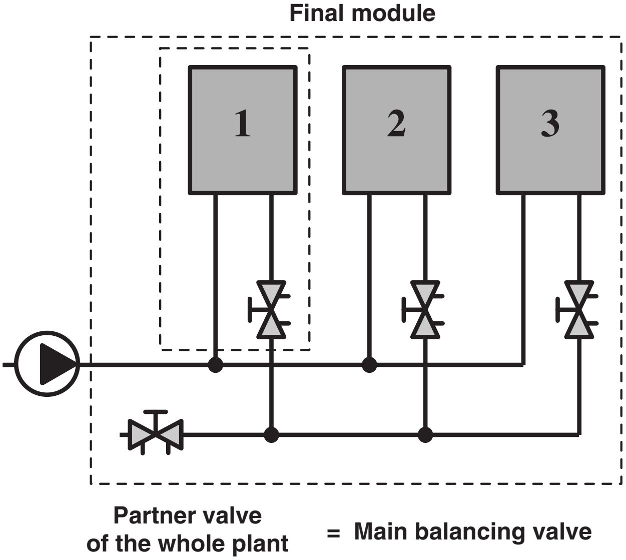


Fig 6.3: All the risers constitute the final module

In this new module, the risers are balanced between themselves following the same procedure.

Finally, the total flow is adjusted with the main balancing valve. When this operation is completed, all circuits in the plant will have the desired flows. To verify this, flow measurements can be done on some balancing valves.

Printout via a PC provides a list of settings and verified data if these values have been stored in the CBI.

All the overpressure is located in the main balancing valve. If this overpressure is important, the maximum pump speed can be reduced (variable speed pump), or with a constant speed pump, the impeller may be changed to reduce the pump head to save pumping costs. In some cases, the pump oversizing is so high that the pump is changed for a smaller size.

With a variable speed pump, the main balancing valve is not necessary. The maximum speed is adjusted to obtain the design flow in the Partner valve of one of the risers. All the other flows will be automatically at design value.

Notes:

1. During the measurements in one module, external disturbances (isolation of an other riser ...) have to be avoided. They may create some errors in the mathematical model elaborated by the CBI and some deviations in the flows obtained with the settings calculated.

2. When measuring the differential pressure across a balancing valve fully shut, remember that the mechanical protection of the CBI will intervene automatically when this differential pressure is higher than 200 kPa.

3. TA Balance Method is generally the fastest balancing method, as it requires only one engineer using this very simple procedure. However, in comparison with the Compensated Method, the engineer has to visit more times each balancing valve (to make the measurements). Consequently, if the balancing valves are very difficult to reach, the Compensated Method can be sometimes more economical.

7.1 Variable flow system with balancing valves

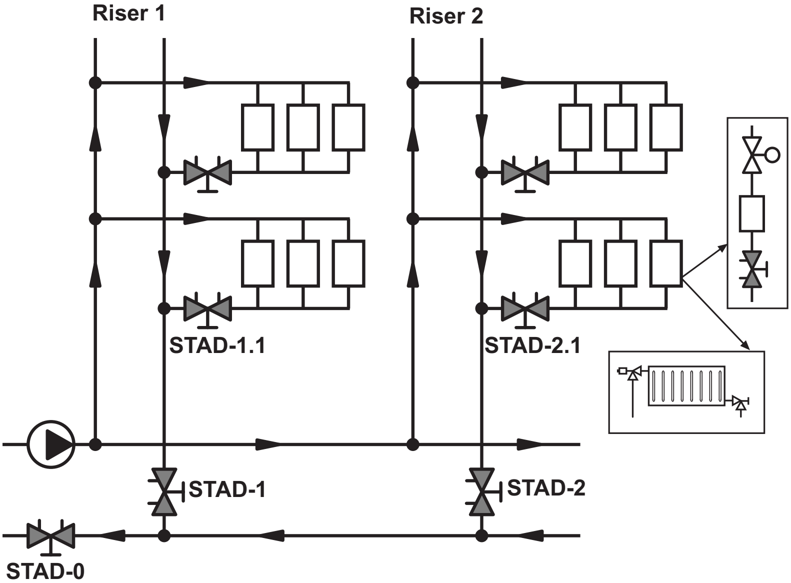


Fig 7.1: General example of a hydronic distribution.

The system is divided in modules.

STAD-1.1 is the Partner valve of the first branch of the first riser.

STAD-1 is the Partner valve of the riser module and STAD-0 is the main Partner valve.

When the terminal units are radiators, the thermostatic valves are preset based on a pressure drop of 10 kPa for design flow. Hydronic balancing is normally done before installing the thermostatic heads.

To balance this typical system, we recommend the Compensated Method (chapter 5) or the TA Balance Method (chapter 6). The main balancing valve STAD-0 shows the pump-oversizing and suitable adjustment of the pump is made accordingly. If the pump is a variable speed pump, STAD-0 is not required; the speed of the pump is adjusted to obtain the design flow in the balancing valve of one of the risers.

7.2 System with BPV and balancing valves

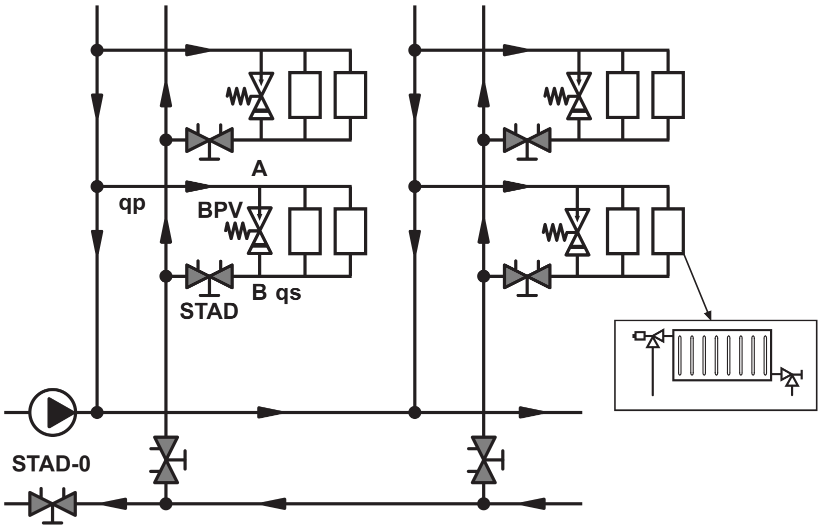


Fig 7.2: On each branch a pressure relief valve keeps constant the differential pressure AB.

This system is mainly used in heating plants with radiators.

On each branch, serving several radiators or terminal units, the balancing valve is associated with a pressure relief valve BPV.

If some terminal control valves shut, the differential pressure AB has the tendency to increase. If this differential pressure increases above the set point of the BPV, the BPV starts to open. The increasing flow in the BPV creates a sufficient pressure drop in the balancing valve STAD to keep approximately constant the differential pressure across A and B. Without a balancing valve, the BPV, open or shut, will be submitted directly to the differential pressure between supply and return riser pipes. The BPV cannot alone stabilise the secondary differential pressure, it must be associated with a balancing valve.

The radiator valves are preset based on a pressure drop of 10 kPa for design flow. The plant is balanced as for figure 7.1 with all BPV fully shut. When the plant is fully balanced, the setting of the BPV is chosen equal to the 10 kPa adopted for the thermostatic valves plus 5 kPa, that means 15 kPa. There are other ways to set the BPV but the method suggested above is the simplest.

Example:
The primary differential pressure available is 40 kPa. During the balancing procedure a pressure drop of 27 kPa was created in the branch-balancing valve to obtain the correct water flow of 600 l/h in the branch. That means a differential pressure of 40 - 27 = 13 kPa between A and B at design condition. The radiator valves have been set for a differential pressure of 10 kPa, but to obtain the total correct flow, this differential pressure of 10 kPa has to be situated in the middle of the branch, with more than 10 kPa at its beginning (13 kPa).

Now let us consider that some thermostatic valves shut, decreasing the secondary flow qs. The table below gives some values showing the evolution of the flows and differential pressure.

Secondary flow qs	Flow in BPV	Δp AB	Primary flow qp
600	0	13.0	600
576	1	15.0	577
562	14	15.1	576
400	162	16.5	562
100	430	18.9	530
0	525	20.6	525

Table 7.1: When the thermostatic valves shut, the BPV opens progressively.

As the primary flow has only decreased from 600 l/h to 525 l/h, the primary differential pressure of 40 kPa remains practically unchanged.

The BPV starts to open when ΔpAB reaches the set point of 15 kPa. When all thermostatic valves are shut, the differential pressure ΔpAB reaches 20,6 kPa instead of more than 40 kPa without the BPV.

The main balancing valve STAD-0 shows the pump-oversizing and suitable action on the pump is made accordingly. If the pump is a variable speed pump, STAD-0 is not required; the speed of the pump is adjusted to obtain the design flow in the balancing valve of one of the risers.

7.3 System with STAP on each riser

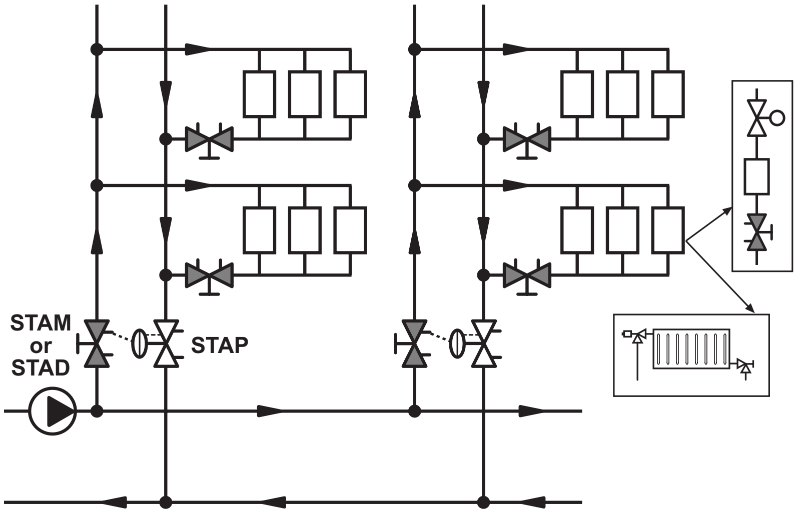


Fig 7.3: A controller STAP stabilises the differential pressure on each riser.

For large systems, the pump head may be too high or variable for some terminals. In this case, the differential pressure is stabilised at the bottom of each riser, at a suitable value, with a STAP differential controller.

Each riser is a module that can be considered independent from the others for the balancing procedure. Before starting the balancing of one riser, its STAP should be put out of function and fully open to be sure of obtaining the required water flows during the balancing procedure. An easy way to do it is to shut the drain on the STAM or STAD in the supply and to purge the top of the membrane (Plug a CBI needle in the top of the STAP).

When the terminals are radiators, the thermostatic valves are first preset at design flow for a differential pressure of 10 kPa.

When each terminal has its own balancing valve, the terminals are balanced against themselves on each branch before balancing the branches against themselves with the Compensated Method or the TA Balance Method.

When a riser is balanced, the set point of its STAP is adjusted to obtain the design flow that can be measured with the STAM (STAD) valve situated at the bottom of this riser. The risers are not balanced between themselves.

Note:
Some designers provide a pressure relief valve (BPV) at the end of each riser to obtain a minimum flow when all control valves are shut. Another method is to provide some terminal units with a three-way valve instead of a two-way control valve. Obtaining this minimum flow has several advantages:

1. The flow of water in the pump does not drop below a minimum value.
2. When the water flow is too low, the pipes heat losses create a higher ΔT in the pipes and the circuits remaining in function cannot deliver their full capacity if required as their supply water temperature is too low in heating or too high in cooling. A minimum flow in the circuit reduces this effect.
3. If all the control valves shut, the differential control valve STAP will also shut. All the return piping of this riser decreases in static pressure as the water is cooling down in a closed area. The differential pressure across the control valves will be so high that the control valve that reopens first will be extremely noisy. The minimum flow created avoids such a problem.

The setting of the BPV is done according to the following procedure:

- The STAP being in normal operation, all the branches of the riser are isolated.
- The STAM(STAD) is preset to obtain at least a pressure drop of 3 kPa for 25% of design flow.
- The BPV is set to obtain 25% of the riser design flow measurable at STAM(STAD).
- The STAM(STAD) is then reopened fully and all branches are put again in normal operation.

7.4 System with STAP on each branch

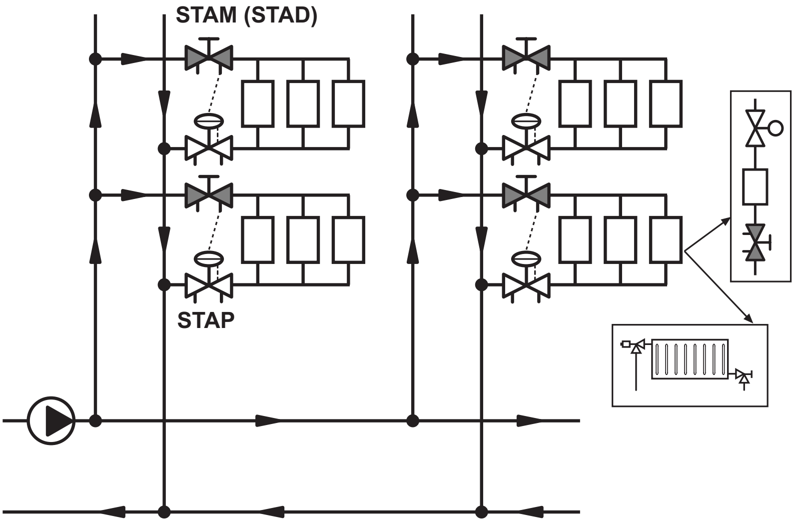


Fig 7.4a: A controller STAP stabilises the differential pressure on each branch.

The differential pressure being stabilised on each branch, the terminals are supplied with a convenient differential pressure. Each branch is balanced independently of the others.

When the terminals are radiators, the thermostatic valves are first preset for a differential pressure of 10 kPa at design flow.

When each terminal has its own balancing valve, they are balanced between themselves using the Compensate method or the TA Balance Method.

When a branch is balanced, the set point of its STAP is adjusted to obtain the design flow that can be measured with the STAM (STAD) valve situated at the supply of the branch

Some designers provide a pressure relief valve (BPV) at the end of each branch to obtain a minimum flow when all control valves are shut. This gives simultaneously a minimum flow for the pump when all terminal control valves are shut. See the note in section 7.3 and also the example hereafter.

It is not necessary to balance the branches between themselves and the risers between themselves.

Example:
It is quite common to provide each apartment of a residential building with one STAP according to figure 7.4b. An On-Off control valve is associated with a room thermostat to control the ambience.

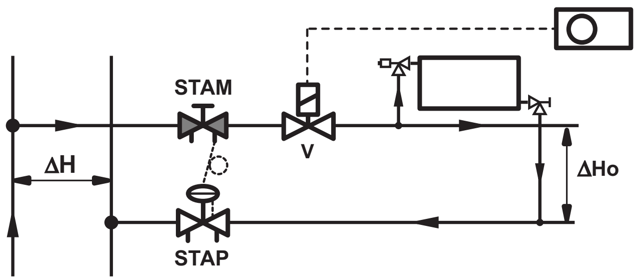


Fig 7.4b: Wrong design with the control valve situated downstream the measuring valve STAM.

When the control valve is situated as in the figure 7.4b, the differential pressure ΔHo corresponds with the differential pressure obtained with the STAP minus the variable pressure drop in the control valve V. So ΔHo is not really well stabilised.

A second problem is the following: When the control valve ”V” shuts, the STAP is submitted to the primary differential pressure ΔH and it also shuts. All the ”secondary” circuit decreases in static pressure as the water is cooling down in a closed area. The Δp across valve ”V” and STAP increases dramatically. When the control valve ”V” starts to reopen, it can probably be very noisy due to cavitations in the valve ”V”. This problem can be solved if the control valve is placed on the return, close to the STAP.

The correct design for the system is shown in figure 7.4c.

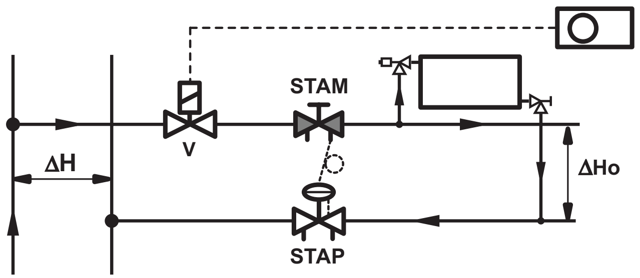


Fig 7.4c: The control valve is situated upstream the measuring valve STAM.

In figure 7.4c, when the control valve shuts, the differential pressure ΔHo drops to zero and the STAP opens fully. The secondary circuit remains in contact with the distribution and its static pressure remains unchanged, avoiding the problem discussed for figure 7.4b. Moreover, the differential pressure ΔHo is much bettered stabilised.

As we can see, a small change in the design of the system can modify dramatically its working conditions.

7.5 System with STAP on each two-way control valve

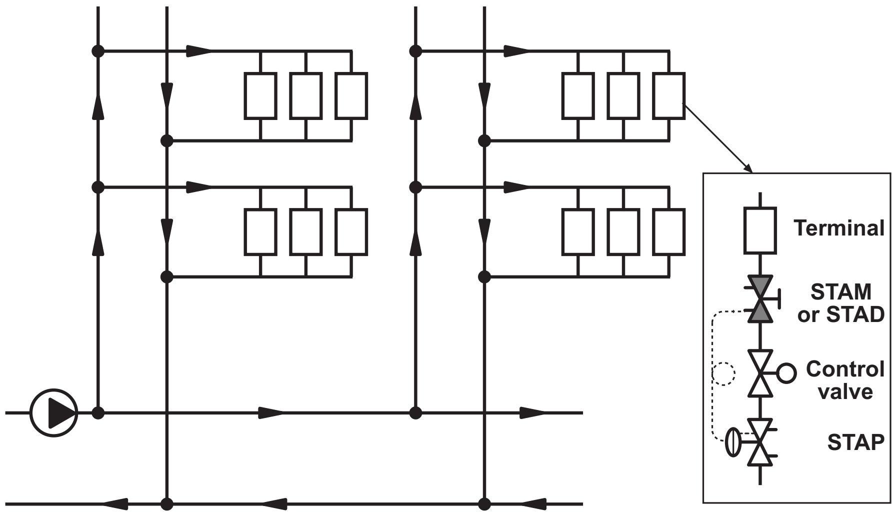


Fig 7.5: The differential pressure is kept constant on each control valve with a STAP.

Each control valve is associated with a Δp controller STAP. From the control point of view, this is the best solution. Furthermore automatic balancing is obtained.

For each terminal successively, the control valve is fully open and the set point of the STAP is chosen to obtain the design flow. Each time the control valve is fully open, the design flow is obtained and the control valve is never oversized. As the differential pressure across the control valve is constant, its authority is close to one.

The balancing procedure is limited to the above description. Terminals, branches and risers are not to be balanced between them as this is obtained automatically.

What happens if only some control valves are combined with STAP and the others are not?

In this case, we are back to figure 7.1 with balancing valves installed on branches and risers. The complete balancing is made with the STAPs fully open. Please note that a STAD is recommended in this case instead of a STAM. This STAD is used as a normal balancing valve during the balancing procedure. When the plant is balanced, the procedure for each STAP successively is as follows:

- The STAD coupled with the STAP is reopenned and preset to obtain at least 3 kPa for design flow.
- The set point of the STAP is adjusted to obtain the design flow across its control valve fully open, the flow being measured by means of the balancing valve STAD.

7.6 Constant flow distribution with secondary pumps

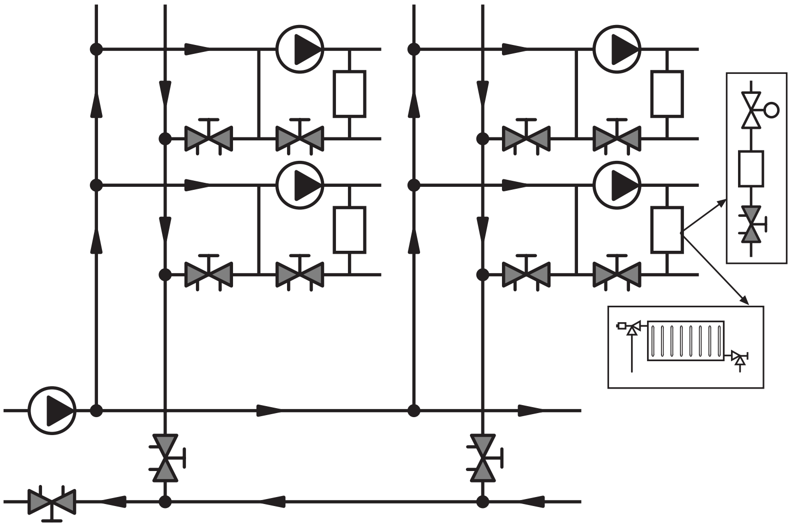


Fig 7.6: Constant flow distribution in the primary side and variable flow in the secondary circuits.

When there is just only one production unit, a constant flow distribution is the most suitable choice. The head of the primary pump has just to cover the pressure drops in the production unit and the primary distribution pipes. Each circuit is provided with a secondary pump.

To avoid interactivity between the primary pump and the secondary pumps, each circuit is provided with a bypass line.

Each circuit is balanced independently of the others.

The primary circuit is balanced separately as for system 7.1 with the following remark: to avoid a short circuit with extreme overflows, it is recommended that all balancing valves on the primary distribution are set to 50% opening before starting the balancing procedure.

7.7 Constant flow distribution with three-way valves

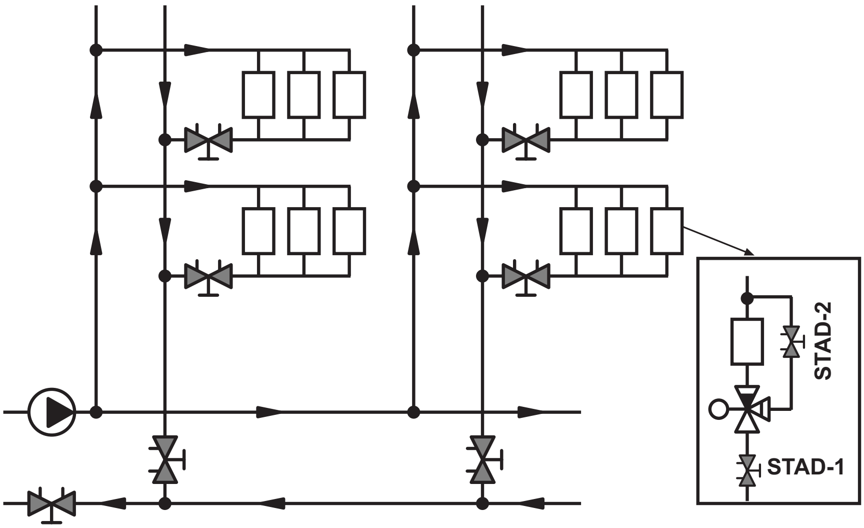


Fig 7.7: The primary flow is maintained constant with a three-way valve in diverting function on each terminal.

The balancing of this system is the same as for figure 7.1. For each three-way valve, a balancing valve STAD-1, in the constant flow, is essential for the balancing procedure. The balancing valve STAD-2 in the bypass has normally to create the same pressure drop as for the coil. In this case, the water flow will be the same when the three-way valve is fully open or fully shut. However, this balancing valve STAD-2 is not necessary when the design pressure drop in the coil is lower than 25% of the design differential pressure available on the circuit.

7.8 Domestic hot water distribution with balancing valves

In domestic hot water distribution, temperature of the water in the pipes drops significantly when consumption is low or zero. As a result, people have to wait a long time to obtain hot water when required. Moreover, below 55°C, the bacteria (Legionella) proliferate dangerously.

To keep the water hot, a permanent circulation is maintained in pipes to compensate for heat losses. A circulation pump is therefore installed guaranteeing a minimum flow q_1 in the loop (Fig 7.8a)

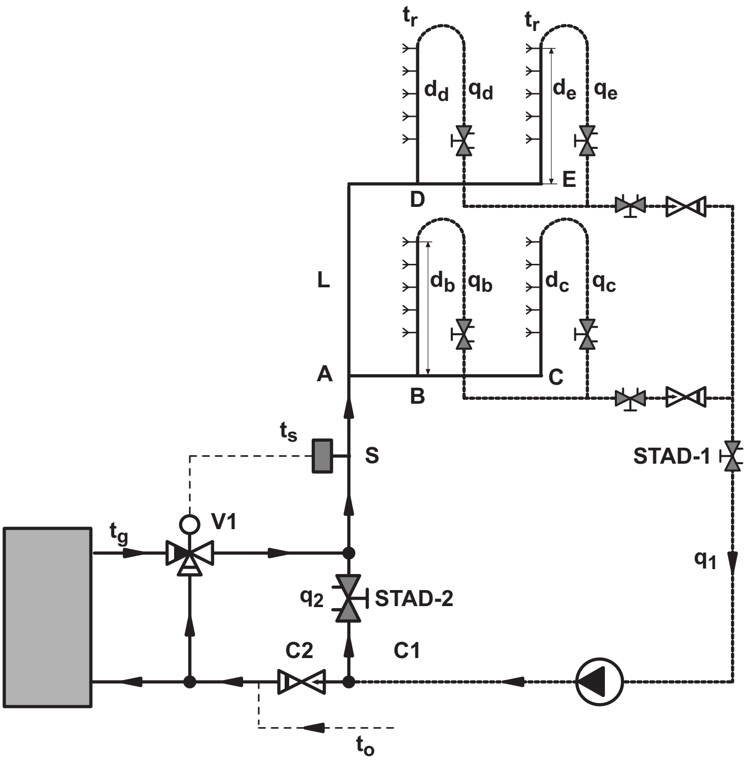


Fig 7.8a: A circulation pump maintains the temperature of water distribution.

Determination of circulation flows

If we accept the most unfavoured user is supplied at a temperature of ΔT below the water supply temperature ts, we can calculate the minimum circulation flow q_1.

$$q_1 = \frac{0.86\ P_m}{\Delta T}$$

where:

P_m: Heat losses in Watt of the **supply** pipes.
Pipes concerned: $\Sigma L + \Sigma d = [SA+AC+AE] + [d_b+d_c+d_d+d_e]$.

ΔT: Admissible temperature drop (5K).

q_1 : In l/h.

For a ΔT of 40K between the water and the ambience, the heat losses are situated around 10W/metre, independently of the pipe diameter. This is valid if the thickness of the insulation in mm (λ=0.036) equals 0.7 x external pipe diameter (without insulation).

Obviously the best procedure is normally to calculate flows according to the insulation installed. A much better estimation can be done using the following empirical formula:

$$P = \frac{\Delta T}{40}\left(3 + \frac{5\ de}{3.5 + \frac{0.036\ I}{\lambda}}\right)$$

with P in W/m, de external pipe diameter in mm (without insulation)

I = thickness of the insulation in mm, λ in W/m.K.
For ΔT = 40 and λ = 0.036 (Foam glass), this formula becomes:

$$P = \left(3 + \frac{5\ de}{3.5 + \ I}\right)$$

with de < 100 mm.

If the distribution is well balanced, a wrong estimation of the total flow does not seem dramatic. If the flow is reduced by 50%, and for a supply water temperature of 60°C, the most unfavoured user will have 51°C instead of 55°C. In this case however, the risk of proliferation of legionella increases.

Hereafter, in the examples, we will consider the following hypothesis:
ts = 60°C, tr = 55°C and P = 10 W/metre. Consequently:

$$q_1 = \frac{0.86 \text{ x } 10}{(60\text{-}55)}(\Sigma L + \Sigma d) = 1.72\ (\Sigma L + \Sigma d)$$

The total flow being known, we have to calculate the flow in each branch. Starting from point S (Fig7.8a) where the temperature sensor is located, the water temperature at the inlet of branch A can be calculated.

$$t_A = t_S - \frac{0.86\ P_{SA}}{q_1}$$

with P_{SA} = heat losses section SA.

For the first branch, the pipes heat losses are $Z_{AC} = P_{AC}+Pd_b+Pd_c$. So we can calculate successively the temperatures at the nodes and the required flows as shown hereafter.

$q_{AB} = \dfrac{0.86\ Z_{AC}}{t_A - 55}$	$t_B = t_A - \dfrac{0.86\ P_{AB}}{q_{AB}}$	$q_b = \dfrac{0.86\ P_{db}}{t_B - 55}$
$q_{BC} = q_{AB} - q_b$	$t_C = t_B - \dfrac{0.86\ P_{BC}}{q_{BC}}$	$q_c = \dfrac{0.86\ P_{dc}}{t_C - 55}$

The flow $q_{AD} = q_1 - q_{AB}$, so we can calculate t_D and the second branch as above. This systematic and simple procedure can be used even for complicated systems.

Knowing the flows, the plant can be balanced normally, using the Compensated Method or the TA Balance Method.

For a rough estimation of the pump head, the pressure losses in the supply pipes can be neglected. Considering just the return pipes, we suggest H [kPa]= 10+0,15 (L_{SE}+de) +3 kPa for each balancing valve in series (3 in this example). If L_{SE}+de = 100 metres for example, H = 10+15+9 = 34 kPa. In this formula we consider 10 kPa pressure drop for the exchanger, check valve and accessories and a pressure drop in the return pipes of 0.15 kPa/m.

Considering just the branch AC in figure 7.8a, but with 4 distribution circuits, we can use the above formulas to calculate the flows. These formulas can be translated in another form, more suitable for a systematic calculation. This other form is explained based on an example hereafter.

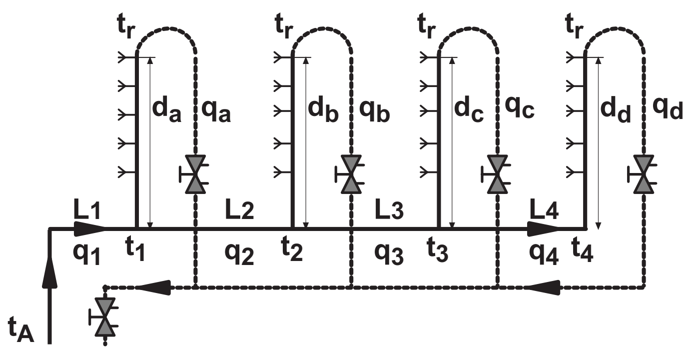


Fig 7.8b: One branch of the distribution with 4 circuits.

Following lengths of pipe (in metres) have been adopted:

L_1	L_2	L_3	L_4
40	25	20	35
d_a	d_b	d_c	d_d
10	9	11	12

Pipe lengths in metres.

The temperature at the supply of the branch is tA and the expected return temperature is t_r. For instance $t_A = 59°C$ (considering 1°C loss between S and A in figure 7.8a) and $t_r = 55°C$.

For a $\Delta T = t_A - t_r = 4K$, and heat losses per metre of pipe equals 10 W/m in average, the total flow q_1 is:

$$q_1 = 0.86 \times 10\ (\Sigma Li + \Sigma di)/(t_A - t_r)$$

so

$$q_1 = 2.15\ (40+25+20+35+10+9+11+12) = 348\ l/h.$$

$$\text{and } t_1 = (t_A - 8.6\ L_1/q_1)$$

In order to obtain a more convenient formula, let us transform it in the following way:

$t_1 = 8.6((t_A - t_r)/8.6 - L_1/q_1)+t_r$. We call $(t_A - t_r)/8.6 = \lambda$ and $D_1 = \lambda - L_1/q_1$

Finally $t_1 = 8.6\ D_1 + t_r$. In this example $\lambda = 0.465$.

$D_1 = \lambda - L_1/q_1$	$q_a = d_a / D_1$	$q_2 = q_1 - q_a$	$t_1 = 8.6\ D_1 + t_r$
$D_2 = D_1 - L_2/q_2$	$q_b = d_b / D_2$	$q_3 = q_2 - q_b$	$t_2 = 8.6\ D_2 + t_r$
$D_3 = D_2 - L_3/q_3$	$q_c = d_c / D_3$	$q_4 = q_3 - q_c$	$t_3 = 8.6\ D_3 + t_r$
$D_4 = D_3 - L_4/q_4$	$q_d = d_d / D_4$		$t_4 = 8.6\ D_4 + t_r$

Formulas used.

These formulas can be extended the same way for more circuits. We have used them to calculate the flows. *Calculations of the temperatures are not necessary but are given for information.*

D_1=0.465-40/348=0.351	q_a=10/0.351= 29	q_2=348-29=319	t_1=8.6x0.351+55=58.0
D_2=0.351-25/319=0.272	q_b= 9/0.272= 33	q_3=319-33=286	t_2=8.6x0.272+55=57.3
D_3=0.272-20/286=0.202	q_c=11/0.202= 54	q_4=286-54=232	t_3=8.6x0.202+55=56.7
D_4=0.202-35/232=0.051	q_d=12/0.051=232	q_4 is obviously = qd	t_4=8.6x0.051+55=55.4

Numerical calculations.

Let us point out that the last circuit requires 67% of the branch flow while the first circuit requires only 8%. On the contrary, if the distribution is not balanced, the first circuit will receive more flow than the last circuit.

A rough estimation of the required pump head is:

$$H=10+0.15\ (40+25+20+35+12)+3x3=39\ kPa.$$

7.9 Domestic hot water distribution with TA-Therm

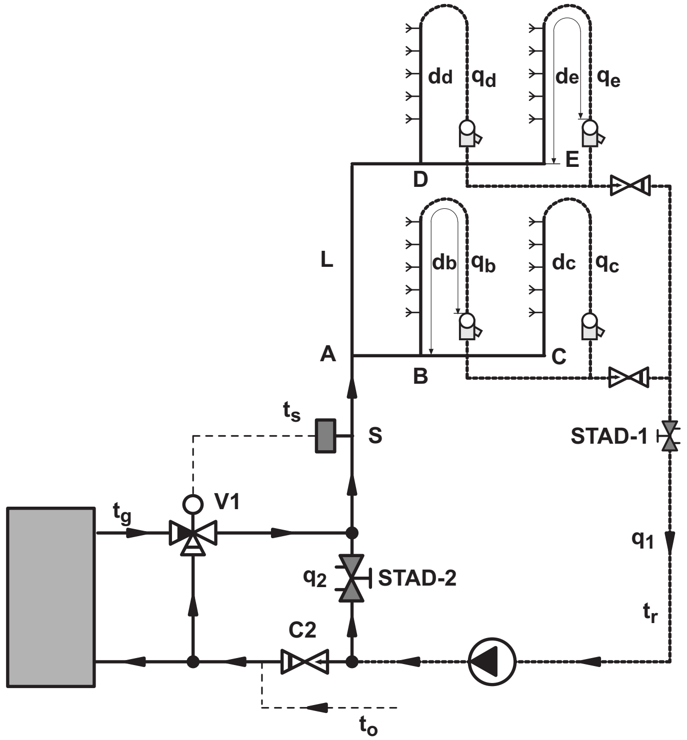


Fig 7.9: The return temperature of each branch is maintained automatically.

The return of each circuit is provided with a thermostatic valve (TA-Therm) that maintains the return water temperature at an adjustable value. A thermometer may be incorporated in the TA-Therm to measure the temperature obtained. The circulation flows are calculated (See figure 7.8b) to size the return pipes and the pump. For the most remote circuits, the pump head is roughly estimated as follows (for TA-Therm with a Kv=0.3):

$$\text{Circuit } q_e : H = 10 + 0.15\,(SE+d_e) + (0.01\,q_e\,/0.3)^2 + 3$$
$$\text{Circuit } q_c : H = 10 + 0.15\,(SC+d_c) + (0.01\,q_c\,/0.3)^2 + 3$$

The highest value of H is adopted.

The Kv of 0.3 given above corresponds with a deviation of 2°C, of the water temperature, relatively to the set point of the TA-Therm.

The Presetting method

The presetting method requires that the designer calculates the correct preset values for all balancing valves and notes them on the drawing. The advantage of this method is that it is rather simple for the installer to preset all balancing valves in conjunction with the installation.

The pressure losses at design flow are determined for each terminal and accessories (control valve, pipeline, valves and bends). The pressure losses between the pump and the least favoured circuit are summed up, giving the necessary pump head.

A pump with the nearest available standard pump head is than selected to meet the flow demand in the least favoured circuit. The difference between the head of the selected pump and theoretically necessary pump head is an excess pressure applied to the system. If it is significant, it should be eliminated in some way. In variable-flow systems, control valves may be resized to take up as much as possible of the excess pressure. The remaining difference can be compensated in balancing valves.

Preset values and flows are noted on the plant drawings. This considerably simplifies the task when balancing the plant.

Since the presetting method is applied on the drawing board, corrections will be necessary when the plant is completed. Plants are rarely installed exactly according to the drawings. Changes affect the flows. Real flows and changes relative to drawings must be noted in the final balancing report.

Recalculation of flows when terminals are oversized

When capacity demands are known, the flows for different terminals are also defined, provided that plant ΔT is selected. Use these formulas to calculate the flow:

$$q = \frac{0.86\ P}{\Delta T_c}\ (l/h) \quad \text{or}$$

$$q = \frac{0.86\ P}{4186\ \Delta T_c}\ (l/s)$$

But the terminals do not necessarily work with the design supply temperature. Neither is it to be taken for granted that terminals with the exact design output are installed in the plant. A terminal with a smaller capacity is rarely selected, but rather with the nearest higher standard value relative to the design requirements.

The capacity of a terminal is defined by the manufacturer under nominal conditions (subscript ”n”). Assume that a terminal is working under other conditions than the nominal conditions, for instance at another supply temperature, and that it is oversized a little. If we know the current supply temperature and the oversizing, we may recalculate the flow to see which flow is really required. This required flow is normally given by the manufacturers.

Use this formula for radiators:

$$t_r = t_i + \frac{(t_{sn} - t_{in})\ (t_{rn} - t_{in})}{(t_s - t_i)\ (P_n / P_c)^{2/n}} \qquad \text{with}$$

t_r = return water temperature (t_{rn} for nominal condition)
t_s = supply water temperature (t_{sn} for nominal condition)
t_i = room temperature (t_{in} for nominal condition)
P_c = Capacity in watt required for the radiator.
P_n = Capacity in watt, in nominal condition, really installed

If n in (2/n) is not given by radiator manufacturer, use n = 1.3.

Example:
A radiator shall give a design output of P_c = 1000 Watt at a room temperature of t_i = 22°C. The supply temperature is t_s = 75°C. The capacity of the installed radiator is P_n = 1500 W, defined for a supply temperature of t_{sn} = 80°C, a return temperature of t_{rn} = 60°C and a room temperature of t_{in} = 20°C.

What should be the flow in the radiator?

If we insert the values above in the formula, the return temperature t_r = 46°C. The real temperature drop is then ΔT = 75-46, that is 29K, and the flow q = 0.86x1000/29, that is 30 l/h.

Sizing of balancing valves

A balancing valve that is bigger than necessary does not only cost more. It also has to be adjusted close to its shut position, which may give poor flow accuracy.

The best operating range for a balancing range is between 50 and 100% of the maximum valve opening. Therefore, select the balancing valve so that the required pressure loss is obtained within this range, for the design flow.

At pressure losses below 3 kPa, the measurement accuracy is reduced because of disturbances before the balancing valve from pump, control valves, bend, etc. The formulas below can be used to size the balancing valve when the Δp to create is known.

$$Kv = \frac{0.01 \text{ x } q}{\sqrt{\Delta p}} \qquad q \text{ (l/h)}, \quad \Delta p \text{ (kPa)}$$

$$Kv = \frac{36 \text{ x } q}{\sqrt{\Delta p}} \qquad q \text{ (l/s)}, \quad \Delta p \text{ (kPa)}$$

Example:

A balancing valve has to create a pressure drop of 15 kPa for a flow of 2000 l/h. According to the formula above, Kv = 5.16.

The balancing valve having the nearest Kvs (table below) above 5.16 is the STAD20.

When the pressure drop required is unknown, selection may be done according to the table below

Table of selection								Pressure drop in pipes in Pa/m				Velocity in pipes in m/s			
STAD DN	Kvs	Water flow l/h		Water flow l/s		Valve open Δp in kPa		Valve size + 1		Valve size		Valve size + 1		Valve size	
								Min	Max	Min	Max	Min	Max	Min	Max
10	1,47	100	430	0,028	0,119	0,5	8,6	17	390	76	1332	0,14	0,59	0,23	0,97
15	2,52	350	750	0,097	0,208	1,9	8,9	62	244	268	1085	0,27	0,57	0,48	1,04
20	5,7	650	1600	0,181	0,444	1,3	7,9	61	312	184	990	0,31	0,77	0,49	1,21
25	8,7	1300	2400	0,361	0,667	2,2	7,6	55	167	213	664	0,36	0,66	0,62	1,15
32	14,2	2000	3800	0,556	1,056	2,0	7,2	57	183	119	391	0,41	0,77	0,55	1,04
40	19,2	2800	5700	0,778	1,583	2,1	8,8	33	119	104	390	0,35	0,72	0,57	1,15
50	33,0	4500	11000	1,250	3,056	1,9	11,1	19	77	100	408	0,23	0,57	0,57	1,39
STA-DR															
15	2,00	200	450	0,056	0,125	1,0	5,1	21	97	96	438	0,15	0,34	0,28	0,62
20	2,00	200	600	0,056	0,167	1,0	9,0	7	53	21	167	0,10	0,29	0,15	0,46
25	4,01	600	1200	0,167	0,333	2,2	9,0	13	48	53	193	0,16	0,33	0,29	0,57
STAF		m3/h		l/s											
65	95,1	10	25	2,78	6,94	1,1	6,9	38	208	84	467	0,52	1,30	0,72	1,79
80	120	18	38	5,00	10,56	2,3	10,0	31	125	113	463	0,56	1,17	0,94	1,98
100	190	33	60	9,17	16,67	3,0	10,0	34	105	96	297	0,67	1,22	1,02	1,85
125	300	55	95	15,28	26,39	3,4	10,0	35	97	89	251	0,77	1,33	1,12	1,94
150	420	90	150	25,00	41,67	4,6	12,8	24	63	90	235	0,74	1,24	1,26	2,09
200	765	150	270	41,67	75,00	3,8	12,5	20	60	63	189	0,78	1,41	1,24	2,22
250	1185	270	420	75,00	116,67	5,2	12,6	25	58	60	138	1,00	1,55	1,41	2,19
300	1450	400	650	111,11	180,56	7,6	20,1	29	71	53	131	1,16	1,88	1,48	2,40

Selection of balancing valves to avoid oversizing when the pressure drop required is not known.

Example:
A balancing valve must be chosen for a water flow of 2000 l/h. The pressure drop required is not known. The flow being situated between 1300 and 2400 l/h, a STAD25 is selected.

For 2000 l/h, the pressure drop in a steel pipe DN25 is 530 Pa/m (See figure C1). As this pressure drop is too high, the selected size of the pipe is probably DN32.

It is also possible to choose a STAD32 to have the same diameter as the pipe. To obtain at least 3 kPa in a STAD32 for 2000 l/h, the STAD32 has to be set on position 3.45 (86% opening), (above 80% open is normally acceptable).

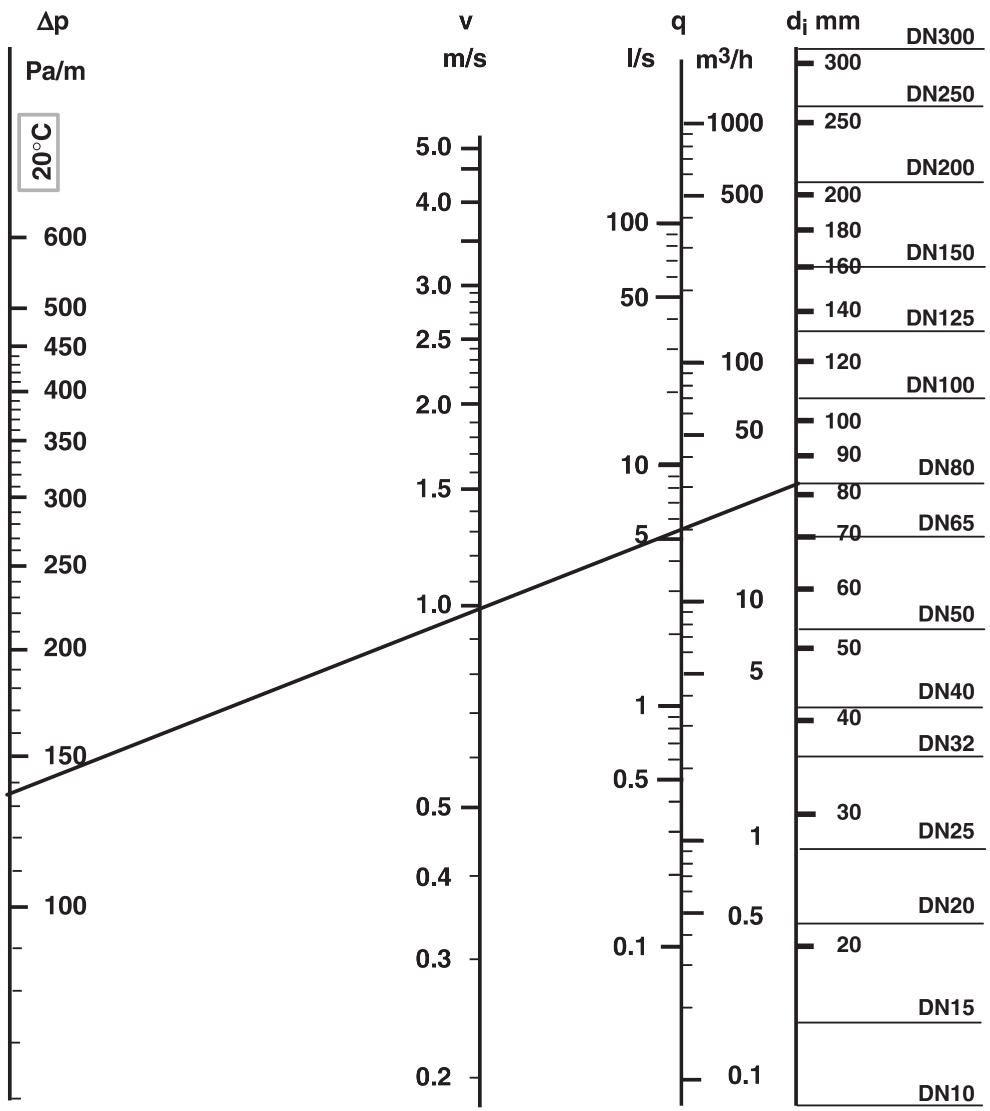


Fig C1: Pressure drops and velocities (steel pipes with a roughness of 0.05 mm) for water at 20°C.

This diagram gives the possibility to check if the size of the balancing valve chosen is compatible with the size of the pipe. Generally, the size of the pipe is the same or one size above the size of the balancing valve.

Example:

Pipe DN 80 and water flow 20 m^3/h: Velocity = 1m/s and Δp = 135 Pa/m

With this pipe, STAF65 and STAF80 are normally accepted.

Installation of balancing valves

In order to ensure accurate flow measurement to have balancing valves, it is normally sufficient with a straight pipe line of five pipe diameters before the balancing valve, and two pipe diameters after the valve.

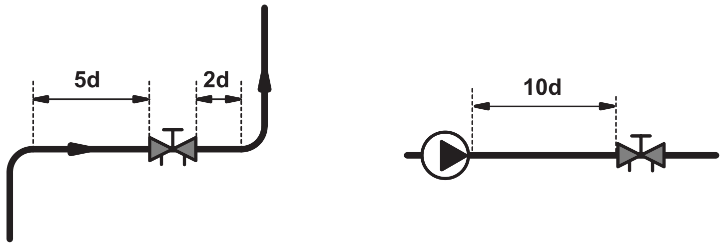


Fig D1: Straight pipe line before and after a balancing valve.

If the balancing valve is installed after something that creates strong disturbances, as for instance a pump or a control valve, we recommend a straight pipe line of minimum 10 pipe diameters before the balancing valve. Do not install anything in this pipe line that can create disturbances (like temperature sensors).

In supply or return?

Hydraulically, it makes no difference whether the balancing valve is located in the supply or in the return pipe. The supply water flow is of course the same as the return water flow.

However, it is customary to place balancing valves in the return pipe, particularly when the balancing valve contains a draining device located in such a way that the adjacent terminal can be drained. It is always preferable to install it so that the flow tends to open the valve (figure below) since this gives a more precise flow measurement and reduces the risk of noise.

In practice, balancing valve may be installed at the most accessible location, as long as turbulence before the valve is avoided.

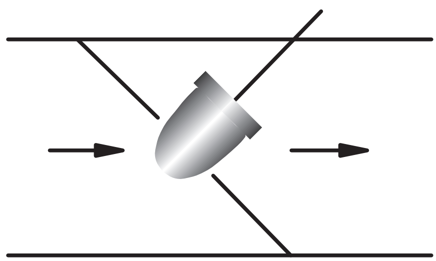


Fig D2: The flow tends to open the valve.

Detailed instructions for the preparation work

It happens that balancers have to waste time searching for simple items like a key to get into a room where a balancing valve is located, or to find a ”lost” balancing valve in a false ceiling, or to access practically inaccessible pressure tapping points.

A preliminary on-site inspection may save a lot of unnecessary work time, particularly in large plants. Such an inspection may involve the following:

- Check the drawings so that all flows are clearly noted for all balancing valves. Check also that the total flow corresponds to the partial flows. In a branch, for example, the sum of the terminal flows must be the same as the total flow in the branch.
- Check that the drawings show the plant as built. If necessary, correct principle circuit schemes and flows.
- Identify all balancing valves, and make sure they are accessible. Check their size and label them.
- Check that the piping is cleaned, that all filters are cleaned and that the piping is deaired.
- Check that all non-return valves are installed in the correct direction and that they are not blocked.
- If the terminals are oversized, check whether the flows have been recalculated (see Appendix B).
- Pressure losses in pipes vary by 20% between 20°C and 80°C. It is therefore important that balancing is carried out with the same temperature everywhere in the system.
- Charge the batteries for the CBI balancing instrument, and check that you have all other tools available and in good shape.

Just before you start

- Prepare report forms and the necessary equipment.
- Check that the static pressure is sufficient.
- Check that all shut-off valves are in the correct position.
- In radiator systems with thermostatic valves, you should remove the thermostats so that the valves open.
- Check all pumps for proper rotation. In the case of variable speed pump, check that the pump is running on full speed.

General design recommendations

The design of a hydronic plant depends on its characteristics and working conditions. However, for any variable flow distribution system with direct or reverse return, constant or variable speed pump, modulating or on-off control, the following recommendations are relevant:

1. Balance the plant hydraulically in design conditions. This ensures that the installed capacity can be delivered. There is no difference if modulating or on-off mode has been selected for control of the terminal units, they must be fully open.

2. Use either the Compensated Method or the TA Balance computer program for balancing of the plant. This avoids any scanning of the plant and significantly reduces labour costs. These two methods reveal pump oversizing and make it possible to reduce pumping costs.

3. Select modulating two-way control valves carefully based on:
   a) Correct characteristic (normally equal percentage).
   b) Correct size: the control, when fully open and at design flow, must take at least 50% of the available circuit differential pressure under design conditions.
   c) The control valve authority should not drop below 0.25.

4. If the last condition 3c cannot be fulfilled for some circuits, a local differential pressure controller is installed in these circuits to improve the control valve authority and decrease the risk of noise.

5. When using a variable speed pump, locate the differential pressure sensor to achieve the best compromise between the desire to minimise pumping costs and limit the differential pressure variations across all the control valves.

More about ”Why balance?”

Hydronic balancing - a necessity for good control

In theory, modern HVAC systems can satisfy the most demanding requirements for indoor climate and operating costs. In practice, however, not even the most sophisticated controllers always perform as promised. As a result, comfort is compromised and operational costs are higher than expected.

This is often because the mechanical design of the HVAC plant does not meet some conditions necessary for stable and accurate control. Three important conditions are:

1. The design flow must be available at all terminals.
2. The differential pressures across the control valves must not vary too much.
3. Flows must be compatible at system interfaces.

F.1 The design flow must be available at all terminals

Common problems

These problems are typical indications that condition number one (i.e. that the design flow is not available at each terminal) is not met:

- Higher than expected energy costs.
- Installed capacity is not deliverable at intermediate and/or high load.
- Too hot in some parts of the building, too cold in other parts.
- Long delay before the desired room temperatures are obtained when starting up after night setback.

Obtaining the correct flows

The power transmitted by a terminal unit depends on the supply water temperature and the water flow. These parameters are controlled to obtain the required room temperatures. Control is only possible if the required water flows are available.

Some people, however, seem to think that it is sufficient to indicate design flows on the drawings in order to obtain them in the pipes. But to obtain the required flows, they must be measured and adjusted. This is why specialists are convinced that hydronic balancing is essential. The discussion is limited to the question: how to do it? Is it, for instance, possible to obtain a correct flow distribution by sizing the plant carefully? The answer, in theory, is yes. But in practice it’s just a dream.

Production units, pipes, pumps and terminals are designed to cover the maximum needs (unless the plant is calculated with a diversity factor). If a link of the chain is not properly sized, the others will not perform optimally. As a result, the desired indoor climate will not be obtained and the comfort will be compromised.

One might think that designing the plant with some security factors would prevent most problems. However, even if some problems are solved that way, others are created, particularly on the control side. Some oversizing cannot be avoided, because components must be selected from existing commercial ranges. These generally do not fit the calculations made. Moreover, at design stage, the characteristics of some components are not known since the contractor will select them at a later stage. It is then necessary to make some corrections taking also into account the real installation, which frequently differs somewhat from the initial design.

Hydronic balancing enables the required flows to be obtained, compensates for oversizing and justifies the investments made.

Distribution systems with constant flow

In a distribution system with constant flow (Figure F.1a), the three-way valve is calculated to create a pressure drop at least equal to the design pressure drop in the coil ”C”. This means a control valve authority of at least 0.5, which is essential for good control. If the pressure drop in the coil plus the pressure drop of the control valve is 20 kPa and the available differential pressure ΔH is 80 kPa, then the balancing valve STAD-1 must take the difference of 60 kPa away. If not, this circuit will experience an overflow of 200%, making control difficult and disturbing the rest of the plant.

In figure 1b, the balancing valve STAD-2 is essential. Without it, the bypass AB will be a short circuit with an extreme overflow, creating underflows elsewhere in the plant. With STAD-2, the primary flow q_p is measured and adjusted to be a somewhat higher than the secondary design flow qs measured and adjusted with STAD-3. If qs > qp, the water flow reverses in the bypass AB, creating a mixing point on A. The supply water temperature will increase in cooling and decrease in heating and the design capacity will not be obtainable on the terminal units.

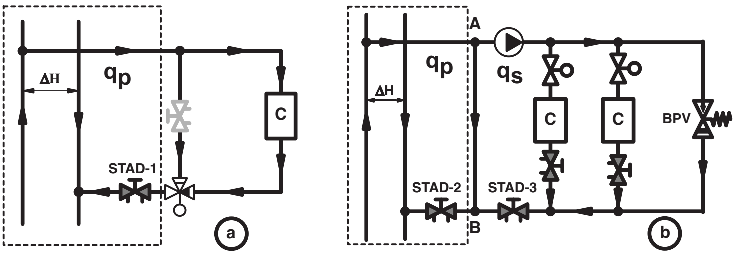


Fig F.1: Examples of circuits in constant flow distribution systems.

Balancing ensures correct flow distribution, prevents operational problems and lets controllers really control.

Distribution system with variable flow

In a distribution system with variable flow, underflow problems occur essentially at high loads.

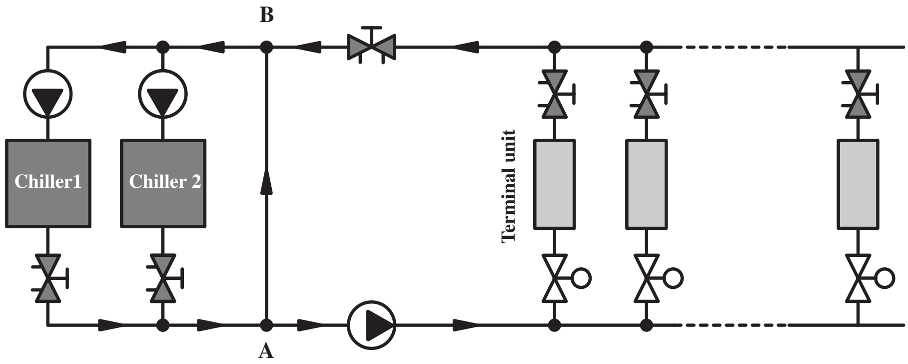


Fig F.2: Example of a variable flow distribution system.

At first glance, there appears to be no reason to balance a system with two-way control valves on the terminals, since the control valves are designed to modulate the flow to the required level. Hydronic balancing should therefore be obtained automatically. However, even after careful calculations, you find that control valves with exactly the required Kvs are not available on the market. Consequently, most control valves are oversized. Total opening of the control valves cannot be avoided in many situations, such as during start up, when big disturbances occur, when some thermostats are set at minimum or maximum value or when some coils have been undersized. In these cases and when balancing valves are not in place, overflows will result in some circuits. This will create underflow in other circuits.

Using a variable speed pump will not solve this problem since all the flows will change proportionally when the pump head is modified. Attempting to avoid overflows this way will simply make the underflows more significant.

The entire plant is designed to provide its maximum capacity at maximum load. It is then essential that this maximum capacity is available when required. Hydronic balancing, made in design conditions, guarantees that all terminals can receive their required flow, thus justifying the investments made. At partial loads, when some control valves close, the available differential pressures on the circuits can only increase. If underflows are avoided in design conditions, they will not occur in other conditions.

Morning start up

In distribution systems with variable flows, morning start-up after each night time setback is a serious consideration since most control valves are driven fully open. This creates overflows, which produce unpredictable pressure drops in some of the piping network, starving the terminals in the less favoured sections of the system. The unfavoured circuit will not receive adequate flow until the favoured spaces have reached thermostat set point (if these set points have been reasonably chosen), allowing their control valves to begin to throttle. Start up is therefore difficult and takes a longer time than expected. This is costly in terms of energy consumption. A non-uniform start-up makes management by a central controller and any form of optimisation practically impossible.

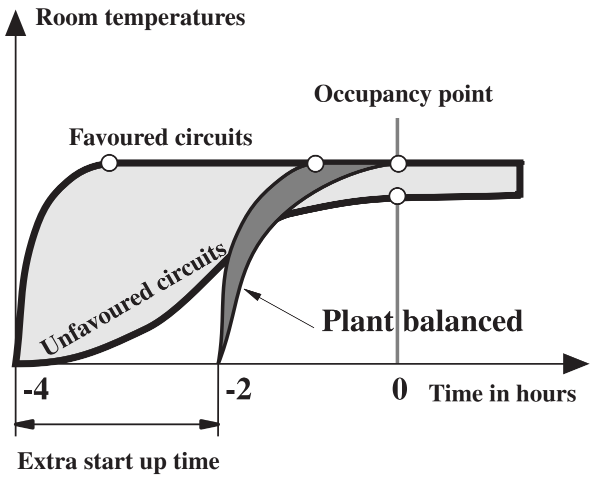


Fig F3: An unbalanced plant has to start up earlier, increasing the energy consumption.

In a distribution system with constant flow, underflows and overflows remain both during and after start up, making the problem much more difficult.

The tools required for balancing

To balance a plant, the required tools must meet following conditions:

- The flow must be measurable with an accuracy of around ± 5%. The balancing procedure makes it possible to check if the plant works as designed, to detect faults and to decide upon measures to correct them.
- The flow must be easy to adjust, thus making the plant flexible.
- The balancing device must guarantee a long-term reliability. It must be resistant to aggressive water.
- During flushing, the balancing devices should not have to be removed and should not require the use of special filters.
- The setting position must be easy to read and be protected by a hidden memory. Full throttling range should require at least four full turns of the handwheel to enable sufficient resolution of the setting.
- A balanced cone should be available for big sizes to reduce the torque required to set the valve against high differential pressures.
- A shut-off function must be included in the balancing valve.
- A measuring instrument must be available, so that flows can be measured easily, without having to use diagrams. The instrument should incorporate a simple balancing procedure and the possibility to print a balancing report. The instrument also enables the evolution of flows, differential pressures and temperatures to be registered for diagnostic purposes.

F.2 Stabilisation of the differential pressures

The control valve characteristic

The characteristic of a control valve is defined by the relation between the water flow through the valve and the valve lift at constant differential pressure. Water flow and valve lift are expressed as a percentage of their maximum values.

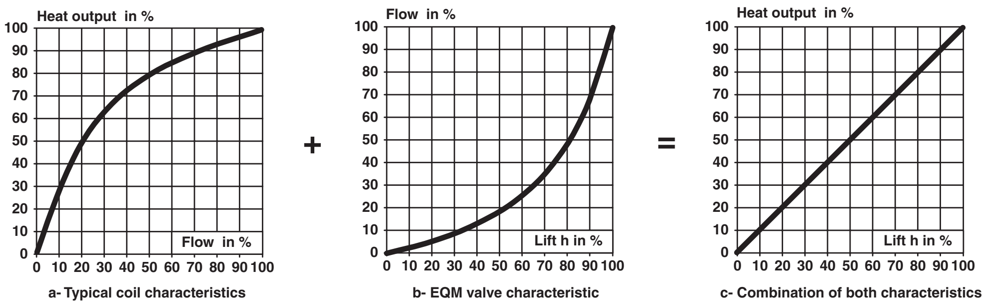


Fig F4: Adopting an inverse non-linear characteristic for the control valve compensates non-linearity of a coil characteristic.

For a valve with linear characteristic, the water flow is proportional to the valve lift. Due to the non-linear characteristic of the terminal unit (Figure F4a), opening the control valve slightly can significantly increase the emission at small and medium loads. The control loop may therefore be unstable at small loads.

Choosing a control valve characteristic to compensate for the non-linearity can solve this problem. This helps ensure that emission from the terminal unit is proportional to the valve lift.

Let's say that the output of the terminal unit is 50 percent of its design value when supplied by 20 percent of its design flow. The valve may then be designed to allow only 20 percent of the design flow when it is open 50 percent. When the valve is 50 percent, 50 percent of the heat output are obtained (Figure F4c). If this holds true for all flows, you can obtain a valve characteristic that compensates for the non-linearity of a typical controlled exchanger. This characteristic (Figure F4b) is called equal percentage modified ”EQM”.

To obtain this compensation, two conditions must be fulfilled:

- The differential pressure across the control valve must be constant.
- The design flow must be obtained when the control valve is fully open.

If the differential pressure across the control valve is not constant, or if the valve is oversized, the control valve characteristic becomes distorted and the modulating control can be compromised.

The control valve authority

When the control valve closes, the flow and the pressure drop are reduced in terminal, pipes and accessories. The difference in pressure drop is applied to the control valve. This increase in the differential pressure distorts the control valve characteristic. The control valve authority can represent this distortion.

$$ß = \text{Valve authority} = \frac{\Delta pVc \text{ (Pressure drop in the control valve fully open and design flow)}}{\Delta p \text{ valve shut}}$$

The numerator is constant and depends only on the choice of the control valve and the value of design flow. The denominator corresponds with the available differential pressure ΔH on the circuit. A balancing valve installed in series with the chosen control valve does not change any of these two factors and consequently does not affect the control valve authority.

The control valve is chosen to obtain the best possible authority. However, the control valve calculated is not available on the market. This is why most of the control valves are oversized. By using a balancing valve, the design flow may be obtained when the control valve is fully open. With the balancing valve, the characteristic obtained is closer to the required characteristic, improving the control function (Fig F6b).

If the balancing valves are well adjusted, they just take away the local overpressures, due to the non-homogeneity of the plant, to obtain the design flow in all coils in design conditions. If afterwards the balancing valves are fully open, the control valves are obliged to shut further. The friction energy can not be saved that way, it will just be transferred from the balancing valves to the control valves. It is then quite obvious that balancing valves do not create supplementary pressure drops.

Moreover, if the pump is oversized, the control valves will create overflows when fully open and take away this overpressure when operating. The pump oversizing will never be detected that way while a balancing procedure will reveal the overpressure, which can be compensated by set-up correctly, the variable speed pump for example.

In some exceptional cases, it’s possible to find control valves with adjustable Kvs, but the problem is to adjust the Kvs at the correct value. This is impossible if the flow is not measurable and if the plant is not balanced to obtain the design differential pressure on each circuit. Balancing valves are then required anyway.

Differential pressure changes with the average load in the plant

In a direct return distribution (Fig F5a), the remote circuits experience the highest variations in differential pressure. At low flows, when the control valve is subjected to almost all the pump head, control valve authority is at its worst.

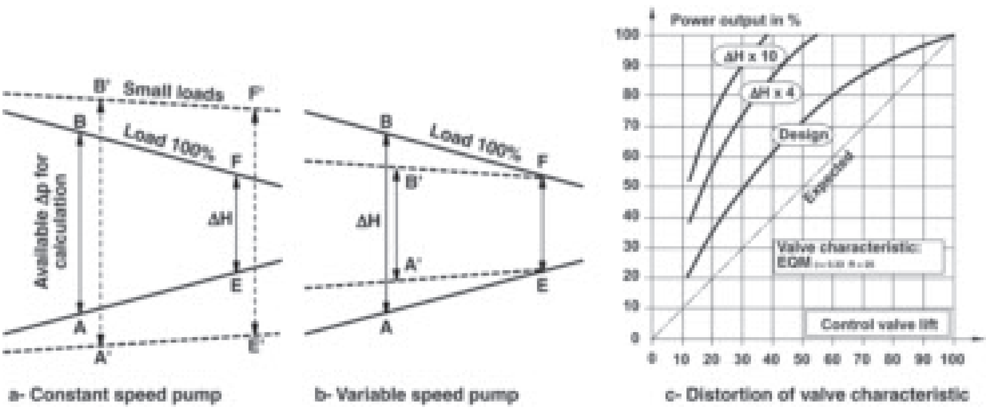


Fig F5: The control valve authority is 0.25 in design condition. When the average load of the plant changes, the differential pressure ΔH on the circuit increases dramatically. This further distorts the control valve characteristic.

With a variable speed pump, it is common to keep the differential pressure constant close to the last circuit (Fig F5b). Then, the problem of varying ΔH is reported to the first circuit.

Locating the differential pressure sensor for the variable speed pump near the last circuit should, in theory, reduce pumping costs. This however causes problems for the circuits close to the pump. If the control valve has been selected according to the available ΔH in design condition, then the circuit will be in underflow for smaller ΔH. If the control valve has been selected based on the minimum ΔH, then, at design condition, the circuit will be in overflow and the control valve will have a bad authority. As a compromise, the differential pressure sensor should preferably be located at the middle of the plant. This can reduce differential pressure variations by more than 50 percent compared to those obtained with constant speed pump.

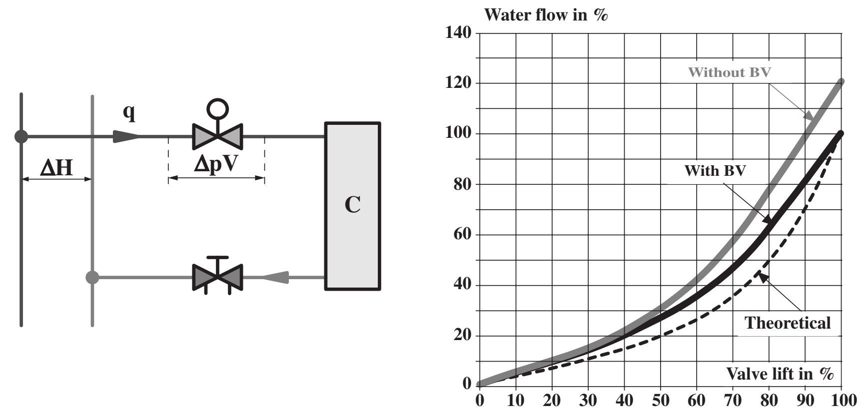
Figure F5c shows the relation between the heat output and the valve lift for EQM control valves selected to obtain the correct flow when fully open and a valve authority of 0.25. When the available differential pressure applied on the circuit increases, the control valve characteristic is distorted so much that it causes hunting of the control loop. In this case, a local differential pressure controller can be used to stabilise the differential pressure across the control valve and keep the valve authority close to 1 (Fig F7a).

Selection of modulating control valves

A two-way control valve is well sized when:

1. The design flow is obtained through the control valve when fully open under design conditions.
2. The control valve authority is and remains sufficient, that is, generally above 0.25.

The first condition is necessary to avoid an overflow, which creates underflows in other circuits, when the control valve is open and remains so for a relatively long period. This occurs (1) during start-up, such as each morning after a night set back, (2) when the coil has been undersized, (3) when the thermostat is set on minimum value in cooling, and (4) when the control loop is not stable.



a- How to calculate the pressure drop that must be taken across the control valve to obtain the design flow?

b- Balancing valve allows design flow. Characteristic obtained is closer the theoretical EQ% than without

Fig F6: If the control valve is oversized, a balancing valve improves the control valve characteristic.

To obtain the design flow at design condition, the pressure drop in the control valve when fully open and at design flow, must be equal to the local available differential pressure ΔH, minus the design pressure drop in the coil and accessories (Fig F6a).

Now, assume that this information is available (!) before selecting the control valve. For a flow of 1.6 l/s, what is available on the market? One control valve that creates a pressure drop of 13 kPa, another that creates 30 kPa and a third that creates 70 kPa. If 45 kPa must be created in the fully open control valve, then such a valve is not available on the market. As a result, control valves are generally oversized. A balancing valve is then needed to obtain the design flow. The balancing valve improves the control valve characteristic without creating any unnecessary pressure (Fig F6b).

Once the control valve has been selected, we must verify if its authority ΔpVc / ΔHmax is sufficient. If it is insufficient, the plant design must be reselected to allow a higher-pressure drop across a smaller control valve.

Some designs to solve local problems

Providing separate solutions for special cases usually results in better operating conditions than forcing the rest of the system to respond to abnormal conditions.

When control valve selection is critical or when the circuit is subjected to major changes in ΔH, a local differential pressure controller can stabilise the differential pressure across the control (Fig F7a). This is generally the case when the control valve authority can drop below 0.25.

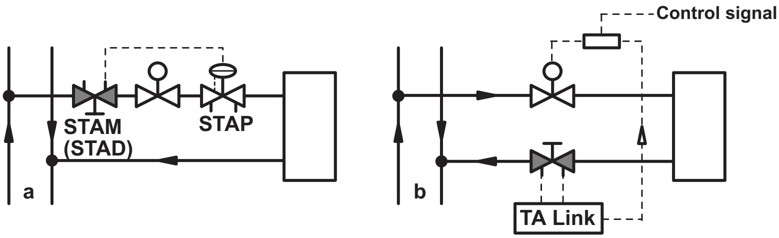


Fig F7: Limitation of the flow across a terminal unit.

The principle is simple. The membrane of the STAP differential pressure controller is connected on the inlet and the outlet of the temperature control valve. When the differential pressure increases, the force on the membrane increases and shuts STAP proportionally. STAP keeps the differential pressure on the control valve almost constant. This differential pressure is selected to obtain the design flow, measurable at STAM, when the control valve is fully opened. The control valve is never oversized and valve authority is close to 1.

All additional differential pressure is applied to STAP. The control of the differential pressure is quite easy in comparison with a temperature control and a sufficient proportional band is used to avoid hunting.

Combining local differential pressure controllers with a variable speed pump ensures the best conditions for control. The comfort is improved with substantial energy savings. The risk of noise is reduced considerably. For economic reasons, this solution is normally reserved for small units (pipe size lower than 65 mm).

For larger units, for which the differential pressure varies widely, the maximum Kvs can be limited by using a differential pressure sensor connected to a balancing valve (Fig F7b). When the differential pressure measured corresponds to the design flow, the control valve is not permitted to open furthermore.

If the plant has been calculated with a diversity factor, the maximum flow allowed is reduced during start up to obtain a homogeneous flow distribution. The set point of the maximum flow can also be changed according to the requirements of priority circuits.

When terminal units are controlled with on off or time proportional control valves, limitation of the differential pressure can help reduce noise and simplify balancing. In this case, a differential pressure controller keeps the differential pressure constant across a set of terminal units (Fig F8).

This solution also works for a set of small units controlled by modulating control valves.

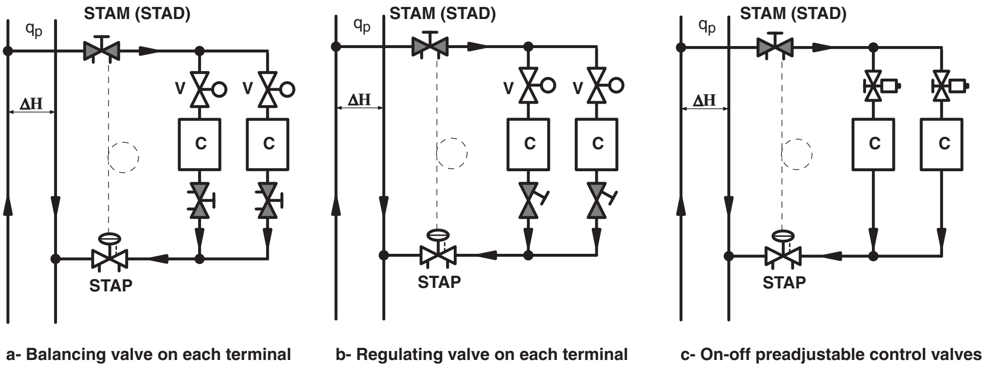


Fig F8: The STAP keeps the differential pressure constant across a set of terminal units.

These examples are not restrictive; they just show that using specific solutions can solve some particular problems.

Keeping the differential pressure constant in heating plants

Variable flow distribution

In a radiator heating plant, the radiator valves are generally preset considering that the available differential pressure ΔHo equals 10 kPa.

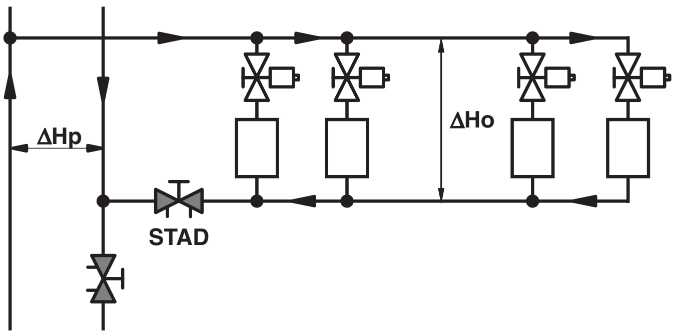


Fig F9: Each radiator valve is adjusted as if it was subjected to the same differential pressure of 10 kPa.

During the balancing procedure, the STAD balancing valve is set to obtain the right total flow in the branch. This justifies the presetting and the 10 kPa differential pressure expected is obtained at the centre of the branch.

In radiator systems with available differential pressure over 30 kPa, there is a risk of noise in the plant, especially when air remains in the water. In this case, you should use STAP to reduce the differential pressure and to keep it constant (Fig F10).

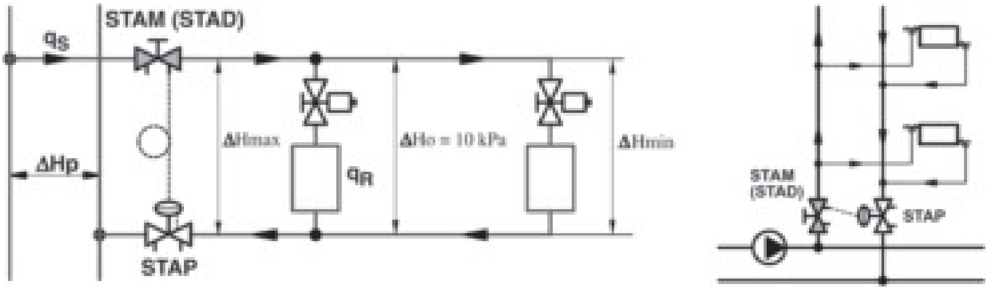


Fig F10: A STAP keeps the differential pressure at the circuit inlet constant.

STAP keeps the differential pressure constant on each branch or small riser. The branch water flow (qs) is measured with the STAM (STAD) measuring valve. This combination relieves the thermostatic valves of excess differential pressure.

Constant flow distribution

The supply water temperature, in a residential building, is adjusted with a central controller according to the outdoor conditions.

The pump head may be high, which can cause noise in the thermostatic valves. If there is no restriction on the return water temperature, a constant flow distribution may be used.

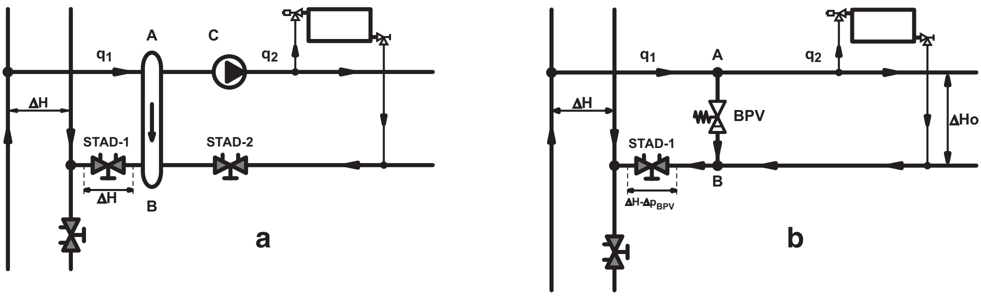


Fig F11: Each apartment receives a differential pressure less than 30 kPa.

One solution is to provide each apartment with a bypass line AB and a balancing valve STAD-1 (figure F11a). This balancing valve takes away the available ΔH. A secondary pump with a pump head less than 30 kPa, serves the apartment. When the thermostatic valves close, the Δp across the thermostatic valves is acceptable and does not create noise in the plant. The secondary design flow must be slightly lower than that of the primary flow to avoid a reverse flow in the bypass, which would create a mixing point at A and decrease the supply water temperature. This is why another balancing valve STAD-2 on the secondary is necessary.

Another solution is to install BPV, a proportional relief valve, for each apartment (Fig F11b). This eliminates the need for a secondary pump and for the balancing valve STAD-2. BPV works with one STAD balancing valve STAD-1 to stabilise the secondary differential pressure. The BPV is set to suit the requirement of the radiator circuit. When the thermostatic valves close, the differential pressure between A and B increases beyond the set point. The BPV then opens and bypasses a supplementary flow creating a sufficient pressure drop in the balancing valve to keep almost constant the differential pressure between A and B.

Let us suppose that the balancing valve STAD-1 is not installed. If the primary differential pressure ΔH increases, BPV will open, increasing the primary flow q_1. The resistance of the pipes between AB and the riser being negligible, the differential pressure across AB remains Practically equal to ΔH. Consequently, to stabilise the secondary differential pressure, the BPV must be coupled with a balancing valve STAD-1 that creates a sufficient pressure drop.

F.3 Flows must be compatible at system interfaces

To give value for investment made

Production units, pumps, pipes and terminal units are designed to provide a certain maximum load even if a diversity factor has been considered. If this maximum load cannot be obtained because the plant is hydraulically unbalanced, we don't give value for investment made.

If the system never requires the maximum capacity installed, it means that the chillers, pumps ... are oversized and the plant is not correctly designed. When the plant is well balanced, it's not necessary to oversize, which reduces the investment and the running costs.

It is quite obvious that overflows in some parts of the plant create underflows in other parts. Unfavoured circuits are not able to provide their full load when required. However another problem will occur. At full load, the supply water temperature will be lower than expected in heating and higher in cooling due to incompatibility between production and distribution water flows.

Example in heating

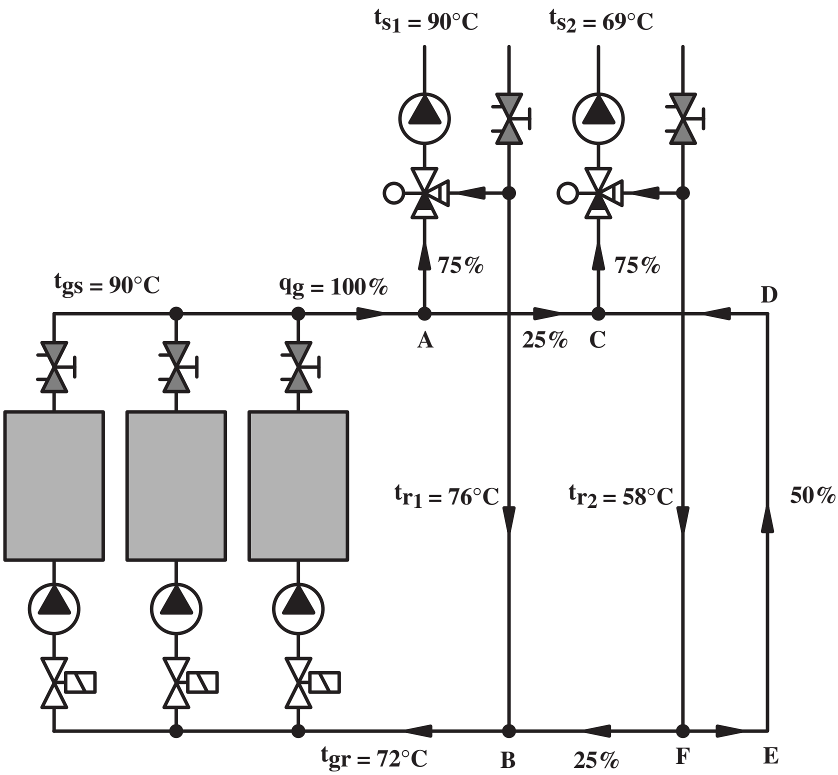


Fig F12: Two circuits are in overflow.

Figure F12 shows a heating plant with three boilers working in sequence. The distribution loop has a low resistance in order to avoid hydraulic interference between the boilers and between the circuits. For this reason any hydraulic resistance has to be avoided in the bypass ”DE”. A check valve between D and E, for instance, will put the secondary pumps in series with the primary pumps, disturbing heavily the function of the three-way mixing valves.

If the two circuits are identical, they have each to take 50% of the total flow. Assume that they take 75% instead. On point ”A”, the first circuit takes 75% of the total flow. It remains 25% for the second circuit. The second circuit takes 75% flow but receives only 25%. It will take 50% from its own return. At ”C”, 25% of hot water is mixed with 50% of the return water from circuit 2. For this circuit, the maximum supply water temperature is 69°C. In design conditions, with an outdoor temperature of -10°C, as long as the first circuit takes its maximum flow, the room temperatures in circuit 2 cannot exceed 14°C. When the room set point of circuit 1 is reached, its three-way control valve starts to shut.

The supply water temperature of the second circuit increases to a maximum of 80°C with an available capacity 10% below the design value. In these conditions, the maximum room temperature will be 17°C for the second circuit. Increasing the pump head of the second circuit to ”solve” the problem will make it worse.

Start up is much longer than expected and the capacity installed is not completely transmittable. To avoid this problem, the total maximum flow absorbed by the circuits must be equal or lower than the maximum flow provided by the production units.

We might think that it would be sufficient to reduce the secondary pump head, in one way or another, to limit the flows. Attempting to avoid overflows this way will simply make the underflows in unfavoured units more significant. Consequently it remains necessary to balance the terminal units between themselves. If the overflow in the circuit is the result of no balancing, we can imagine that some circuits will receive only 50% of their design flow. For these circuits, the situation is worse. The supply water temperature is 10°C lower than design and the flow is also reduced.

Balancing investment represents typically less than one percent of the total HVAC costs, allowing the maximum capacity installed to be transmittable, valorising all the investments made.

Example in cooling

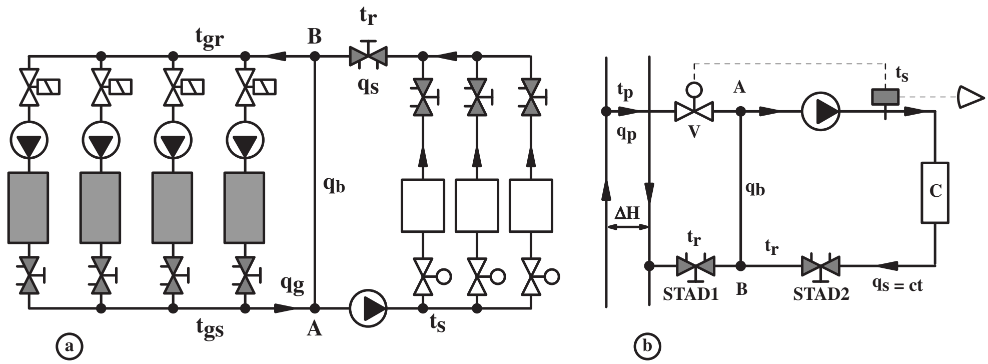


Fig F13: Examples in cooling.

Fig F13a represents a chilled water plant with four chillers. If the distribution circuit is not balanced, the maximum flow q_s may be higher than the production flow q_g. In this case, the flow q_b in the bypass reverses from B to A, creating a mixing point at A. The supply water temperature ts is then higher than design and the maximum capacity installed is not transmittable.

Fig F13b represents a terminal unit working at constant flow with a two-way valve controlling injection. If the flow in the terminal unit is too high, the flow q_b is always in the direction B to A. The supply water temperature ts is always higher than design and the maximum design capacity is never obtained in the terminal unit.

For both examples, an overflow of 50% in the distribution or through the coil will increase the supply water temperature from 6°C to 8°C.

Troubleshooting and system analysis

Hydronic balancing prevents overflows in certain circuits from causing underflows in others, detects possible oversizing of pumps and verifies that the plant does provide the functions and performances intended by the designer.

G.1 Common problems

Symptoms	Common and incorrect counter measures	Typical and usually neglected root causes	Correct solutions	Advantages
Too hot in some parts of the building, too cold in other parts	Increase pump head (main or secondary pump head	Overflow in some circuits create underflow in others No balancing of circuits downstream of Δp controllers	Balance distribution system using STAD/STAF balancing valves	• Correct room temperatures at optimised energy cost • Flows are verified and documented • Possibility to measure and log the flow and detect problems • Manual balancing identifies hydronic problems which can be solved at an early stage • Shortest possible start-up time after night setback • Possibility to minimise pump energy consumption • Avoid noises due to abnormal pump head • Stable and accurate temperature control • Low energy cost • Risk of noise reduced • With the control of the Δp across the control valves, with STAP, their authority remains close to one and the control valves are never oversized
Long time before all rooms reach correct temperature after night setback	Increase supply temperature (heating). Reduce when possible supply temperature (cooling) Cancel the night setback function Install additional boilers or chillers	Overflow in some circuits create underflow in others Distribution flow higher than production flow (compatibility between flows not obtained)	Balance distribution system using STAD/STAF balancing valves Make sure that production and distribution flow are compatible by adequate balancing	
Abnormal pump energy consumption		Pump is oversized and pump oversizing not detectable	Balance the plant with STAD/STAF to reveal pump oversizing Adjust consequently the pump speed or trim the impeller or change the pump Install secondary pumps to overcome special high losses	
Temperature controlled fluctuates Noise from control valves	Modify control software although it is a hydronic problem Replace correctly sized but hunting control valve by smaller ones	Plant not balanced Control valve authority is too low Wrong control parameters Control valve oversized or submitted to variable Δp.	Balance the plant using STAD/STAF balancing valves. Size the control valve in a better way. Adopt the correct characteristic and suitable control parameters Limit Δp variations using STAP Δp controllers	

G.2 Quick troubleshooting

CBI instrument makes it easier to put your finger on problems in hydronic systems. CBI measures and registers differential pressures, water flows and temperatures using balancing valves.

Here is a list of some typical faults that CBI can help you identify:

- Wrong flow in pipes and terminals.
- Too high or too low supply water temperature.
- Incorrect air temperatures.
- Production and distribution water flows are not compatible.
- Interactivity between production units.
- Abnormal pressure drops in elements with the possibility to detect blocked filters and clogged terminals.
- Shutoff valves that should be open but are closed.
- Improperly connected check valves.
- Dented pipes.
- Too high or too low available differential pressure across a circuit.
- Distribution pump oversized or undersized.
- Wrong rotation direction for a pump with a three-phase motor.
- Unstable rpm in a pump.
- Interactivity between circuits with sometimes reverse flows in pipes.
- Unstable control on terminals.
- Oversized control valve and possibility to calculate its authority.
- .....

One of the many benefits of manual balancing is that you detect many of these types of faults during the balancing procedure. It is much less expensive to correct faults at this stage than, for instance, after the false ceiling has been installed and tenants have moved into a building.

G.3 Accurate system analysis

To be able to correct especially tricky operating problems, it is sometimes necessary to perform an accurate system analysis. As a basis for such analysis, it is useful to know how differential pressure, flow and temperature vary over time at strategic points in the plant.

CBI can help. Connect the instrument to the plant and let it collect data for a while. Then connect CBI to a computer and print out the data in the form of easy-to-grasp chart that you can analyse at your leisure back in the office.

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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.

Production: Sandberg Trygg AB, Sweden.

— 3rd edition —

Contents

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:

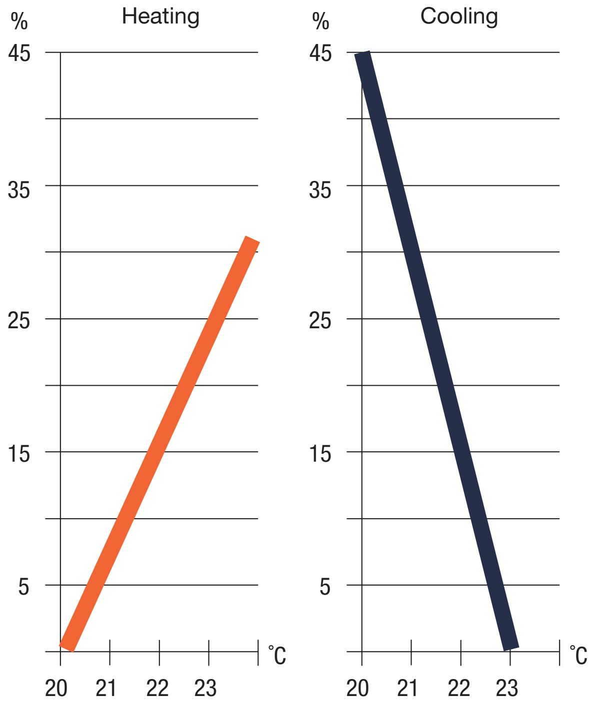
- Some rooms never reach the desired temperatures.
- Room temperatures oscillate, particularly at low and medium loads, even though the terminals have sophisticated controllers.
- Although the rated power of the production units may be sufficient, design power can't be transmitted, particularly during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

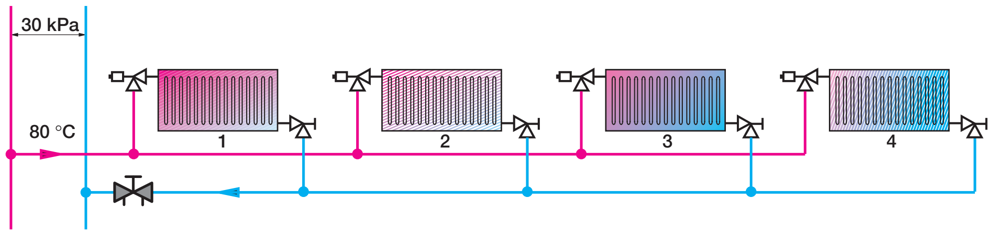


Fig 1.1. Branch with four radiators.

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.

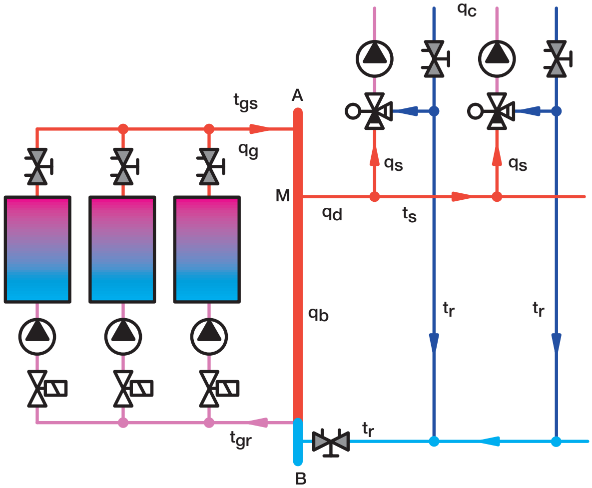


Fig 1.2. Several circuits are connected to the heating plant through a decoupling bypass.

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.

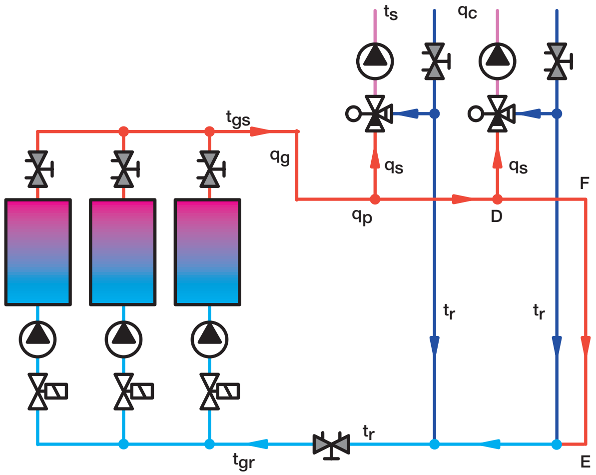


Fig 1.3. Distribution through a closed loop.

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

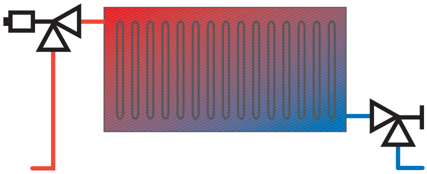


Fig 2.1. Radiator valves on the inlet and the outlet.

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.

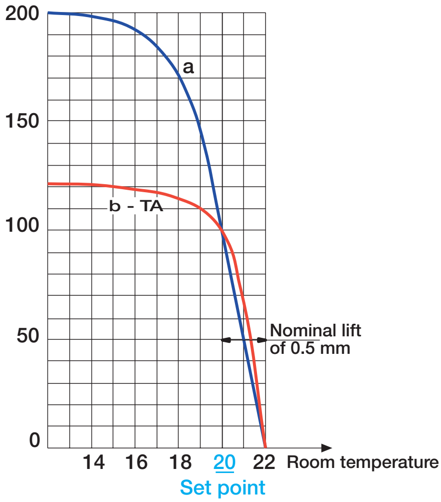


Fig 2.2. Relation between the water flow and the room temperature for a thermostatic valve supplied at constant differential pressure. a - unlimited flow valve. b - valve with flow limitation.

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.

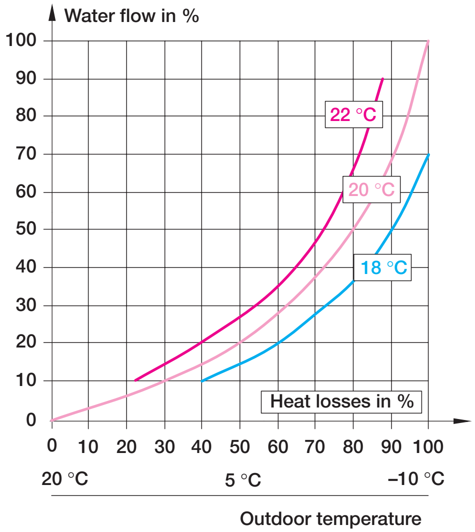


Fig 2.3. The necessary water flow to maintain the room temperature constant as a function of outdoor conditions. The supply water temperature is assumed to be constant and equal to 80 °C.

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.

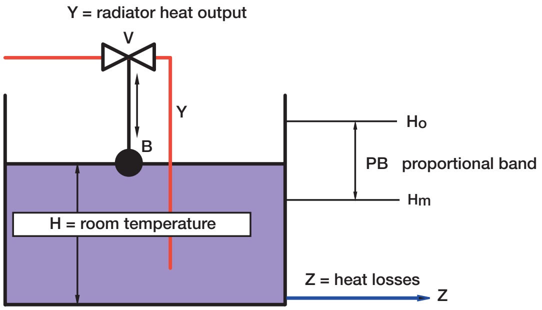


Fig 2.4. Analogue representation of a thermostatic valve.

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.

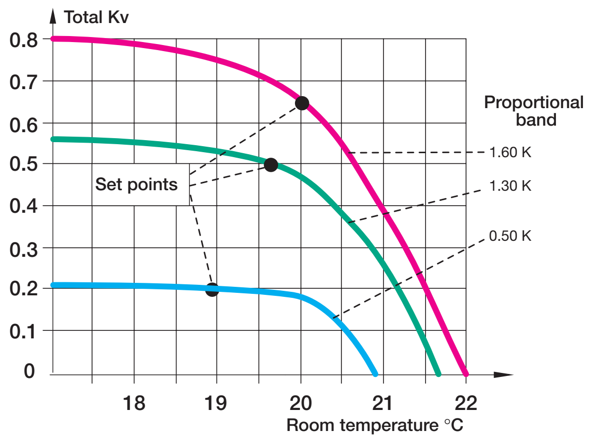


Fig 2.5. A regulating valve or an internal restriction in the thermostatic valve modifies the resulting Kv = f (ti) curve and the practical proportional band.

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.

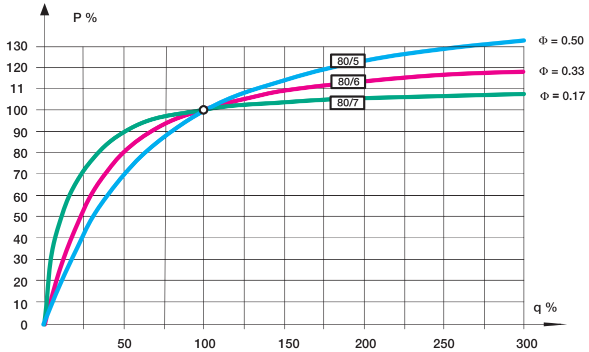


Fig 3.1. Heat output, in a room at 20 °C, as a function of the water flow, for a radiator (n = 1.3) and for different water ΔT_c values. t_{sc} = 80 °C.

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

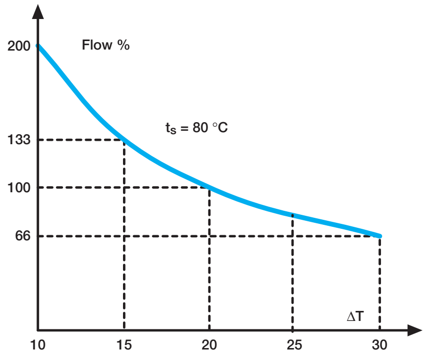
For a supply water temperature between 70 and 90 °C, it's quite common to design the plants for a ΔT = 20. This magic value has been adopted for many years and translated into local units (20 °C in continental Europe and 20 °F (11 °C) in the UK and USA, for instance). However, to reduce the return water 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.



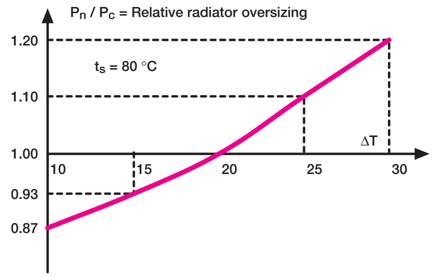


Fig 3.2. When the designed ΔT is increased, the water flow decreases but the required radiator surfaces increase (t_s = 80 °C).

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.

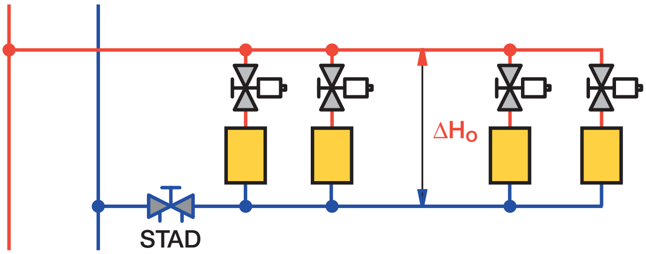


Fig 4.1. Each radiator valve is adjusted as if it were subject to the same differential pressure ΔH_O.

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 $\Delta H_o = 8$ 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 $\Delta H_o = 2$ 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.

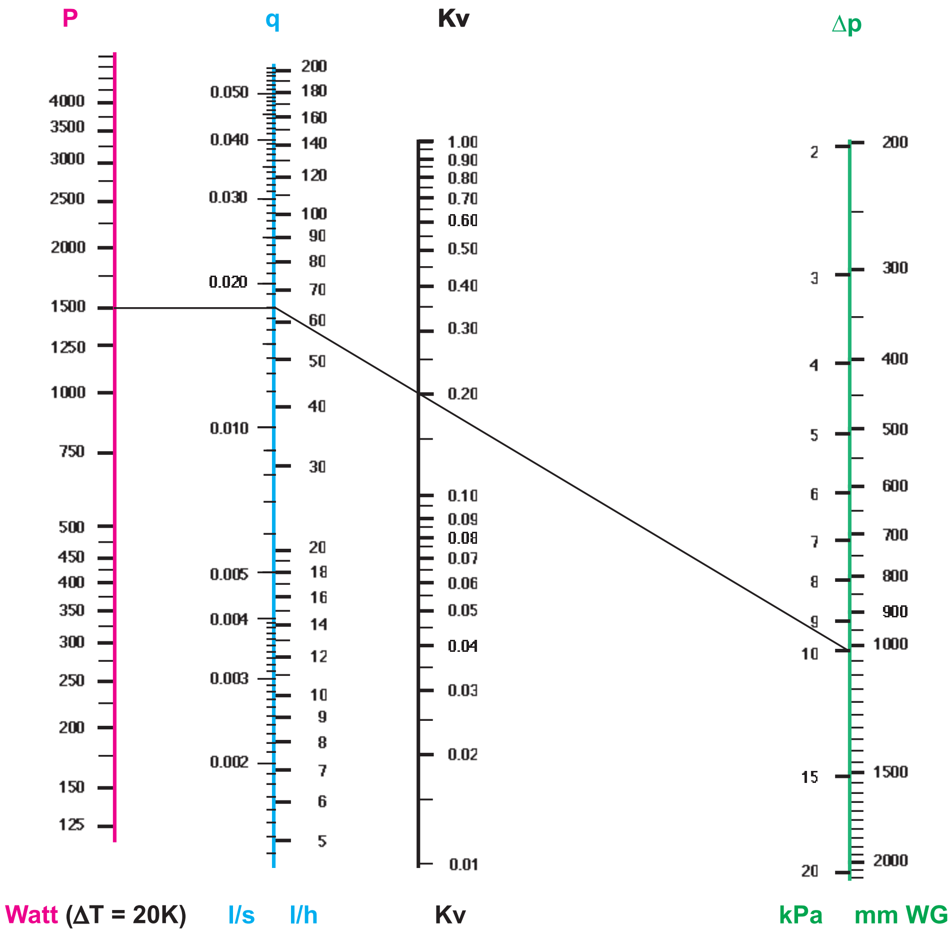


Fig 4.2. Determining the Kv of a thermostatic valve.

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.

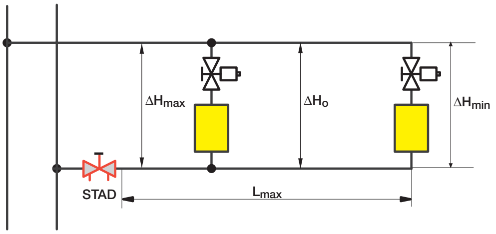
> *Example:* A 2000 W radiator working with a $\Delta T = 20K$ is supplied at a differential pressure of 10 kPa. The thermostatic valve has a Kv = 0.5. What Kv should be set at the return valve?
>
> Referring to Fig 4.2, it can be seen that the radiator flow is 86 l/h. At this flow, the pressure drop 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.



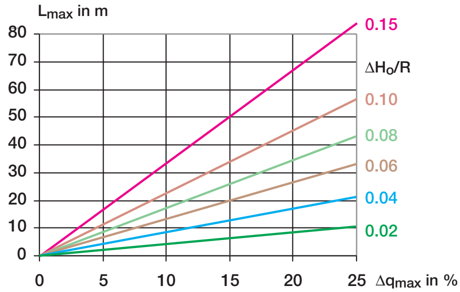


Fig 4.3. When all valves are calculated based on the same $\Delta p = \Delta H_o$, the circuit length should not exceed a given value (ΔH_o in kPa and R (pressure drop in pipes) in Pa/m).

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.

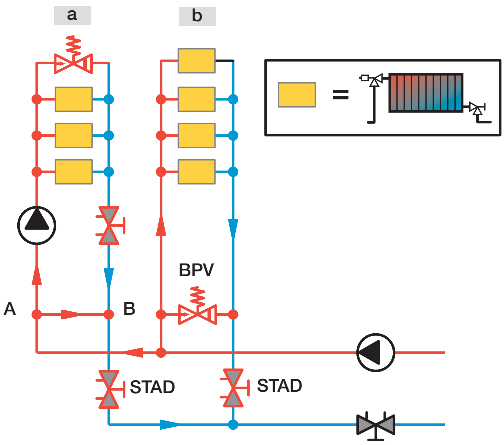


Fig 4.4. Two circuits with radiators are designed to give a constant primary flow.

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.

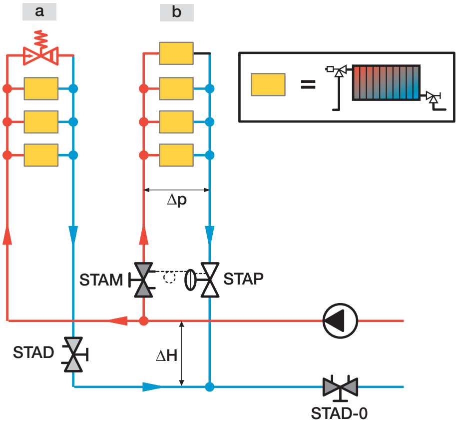


Fig 4.5. Two circuits supplied at variable primary flow.

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.

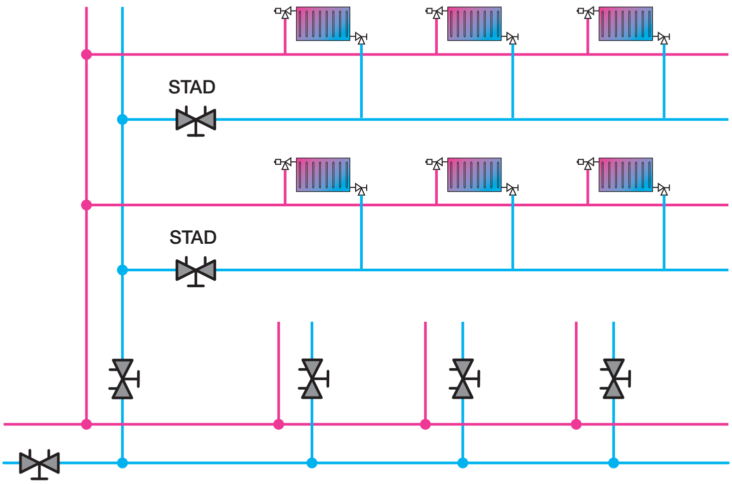


Fig 4.6 The installation is balanced using the TA method.

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.

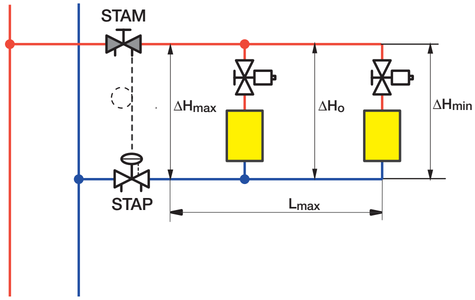


Fig 4.7. A controller stabilises the differential pressure at the circuit inlet.

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).

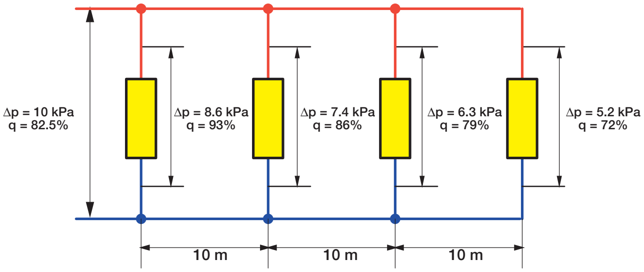


Fig 4.8. Δp = 10 kPa at the inlet of the branch.

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.

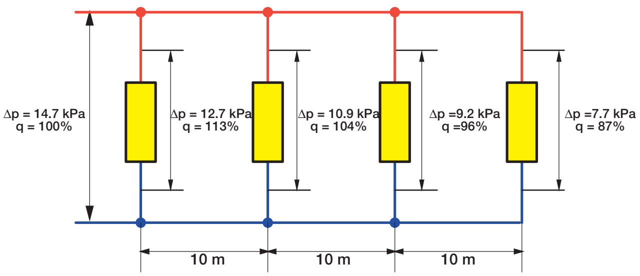


Fig 4.9. The set point of the Δp controller is adjusted to obtain the correct design flow in the branch.

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.

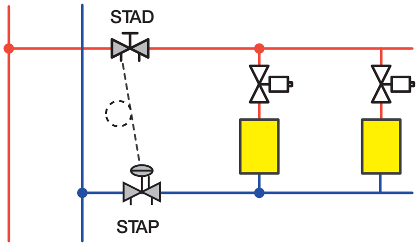


Fig 4.10. The pressure drop in the balancing valve is included in the total Δp controlled by the STAP.

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

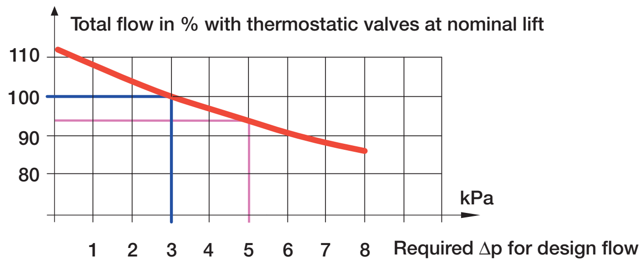


Fig 4.11. If the required Δp at design condition is 5 kPa instead of the 3 kPa expected, the deviation in flow is only 7%.

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.

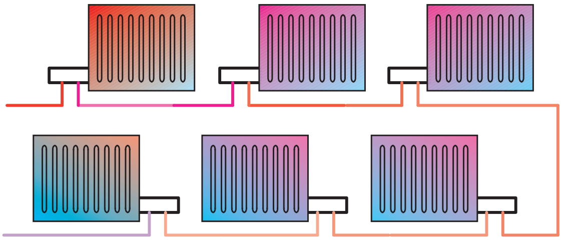


Fig 5.1. Radiators with one-pipe distribution.

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.

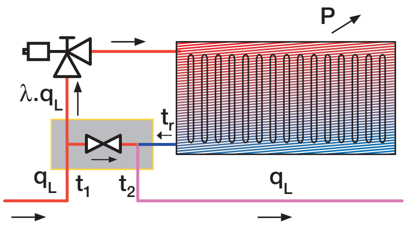


Fig 5.2. A bypass diverts part of the loop flow.

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.

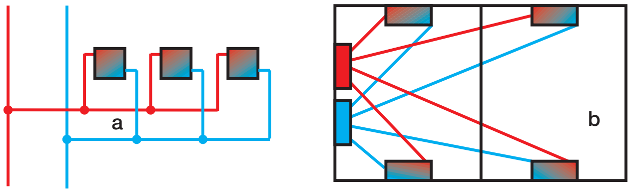


Fig 5.3. Emission from pipes in a two-pipe distribution.

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.

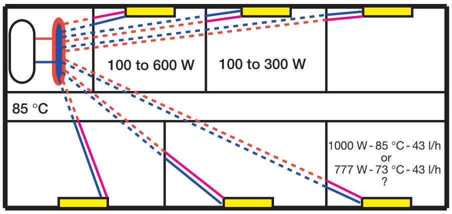


Fig 5.4. In a two-pipe system, the pipes emission in a room depends mainly on other rooms.

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.

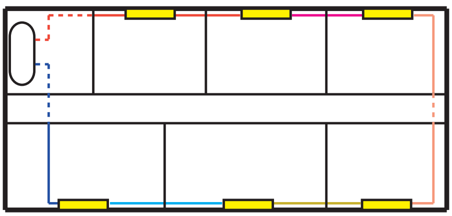


Fig 5.5. In a one-pipe distribution the pipes emission can be deducted from the heat losses to calculate the radiators.

5.2 One-pipe valves

5.2.1 CONSTANT BYPASS – VARIABLE KV

Several methods of installation are possible.

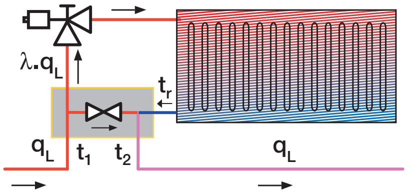


Fig 5.6. Distributor with fixed bypass.

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

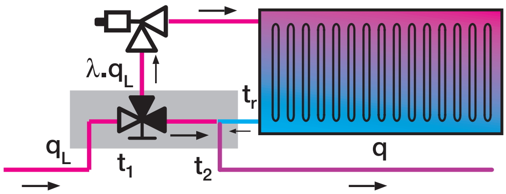


Fig 5.7. Distributor with variable bypass.

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.

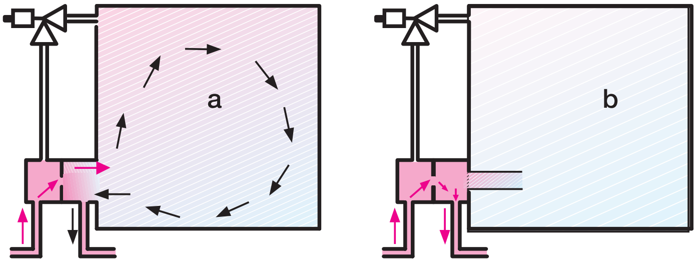


Fig 5.8. With the thermostatic valve closed, double circulation can take place in the radiator return connection. An inserted tube can practically eliminate this phenomenon.

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 ΣP = 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

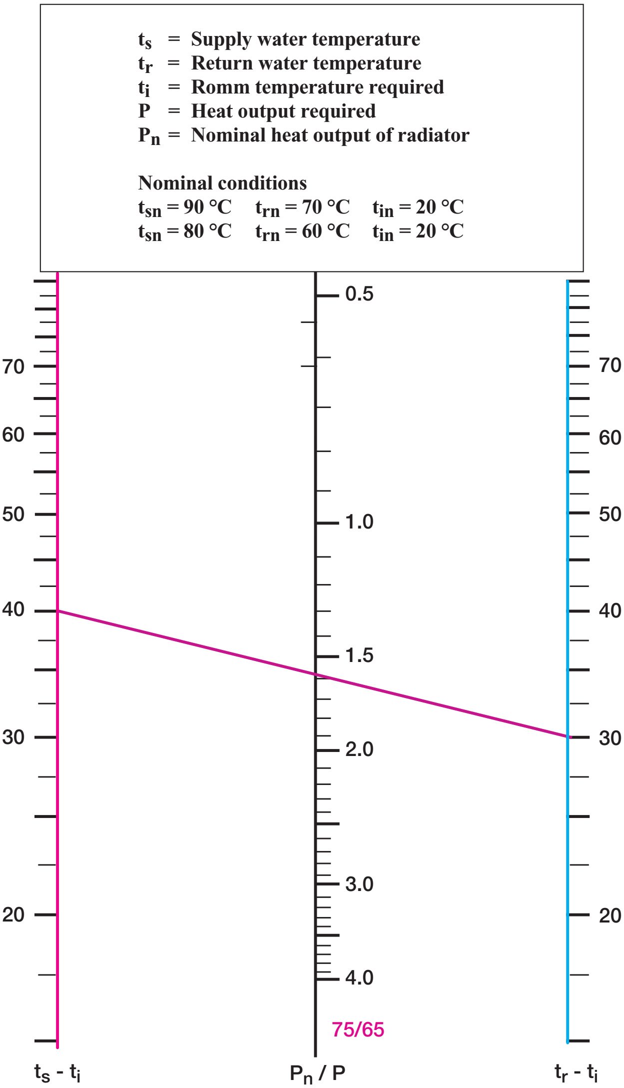


Fig A1. The power of a radiator not working in nominal condition (n = 1.3).

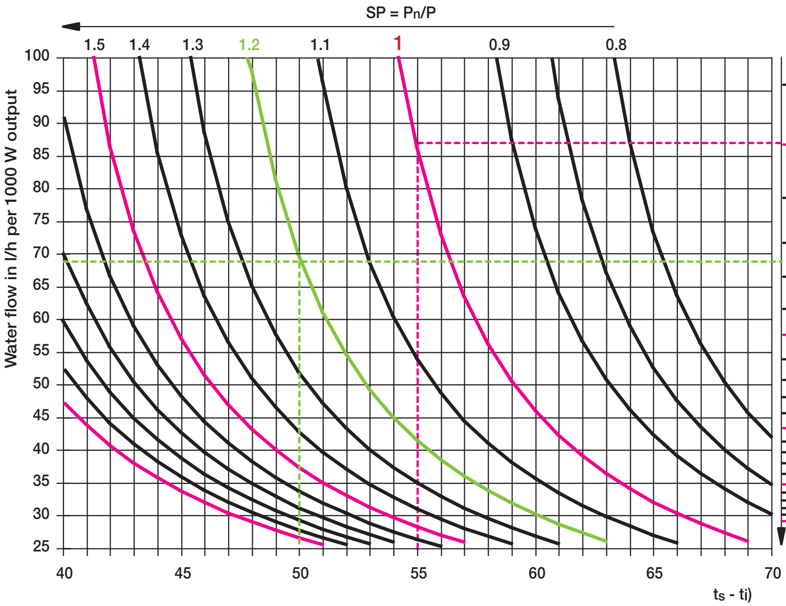


Fig A2. Water flow in l/h per 1000 W is a function of the oversizing factor Sp = P_n/P and the supply water temperature t_s (nominal condition t_{sn} = 75 °C, t_{rn} = 65 °C).

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.

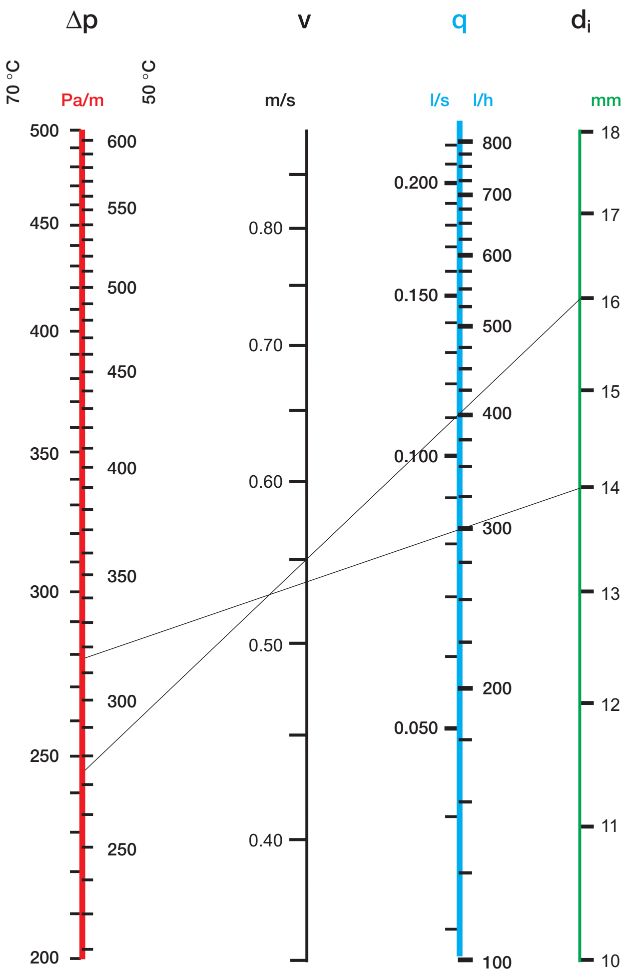


Fig B1a. Pressure losses in pipes with roughness less than 0.0045 (smooth steel, copper, polyethylene, etc.)- di is the inside diameter of the pipe in mm.

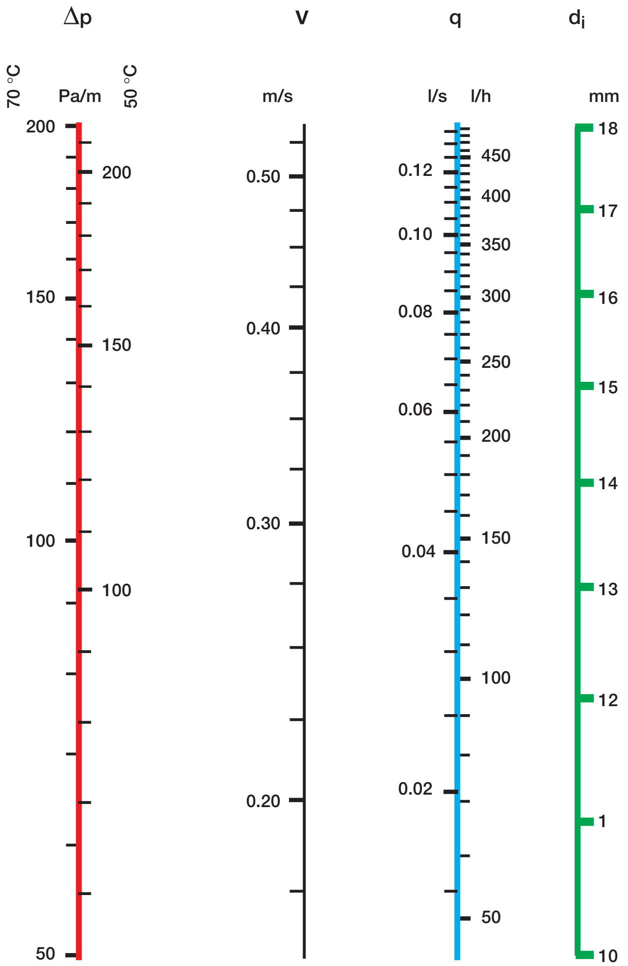


Fig B1b. Pressure losses in pipes with roughness less than 0.0045 (smooth steel, copper, polyethylene, etc.)- di is the inside diameter of the pipe in mm.

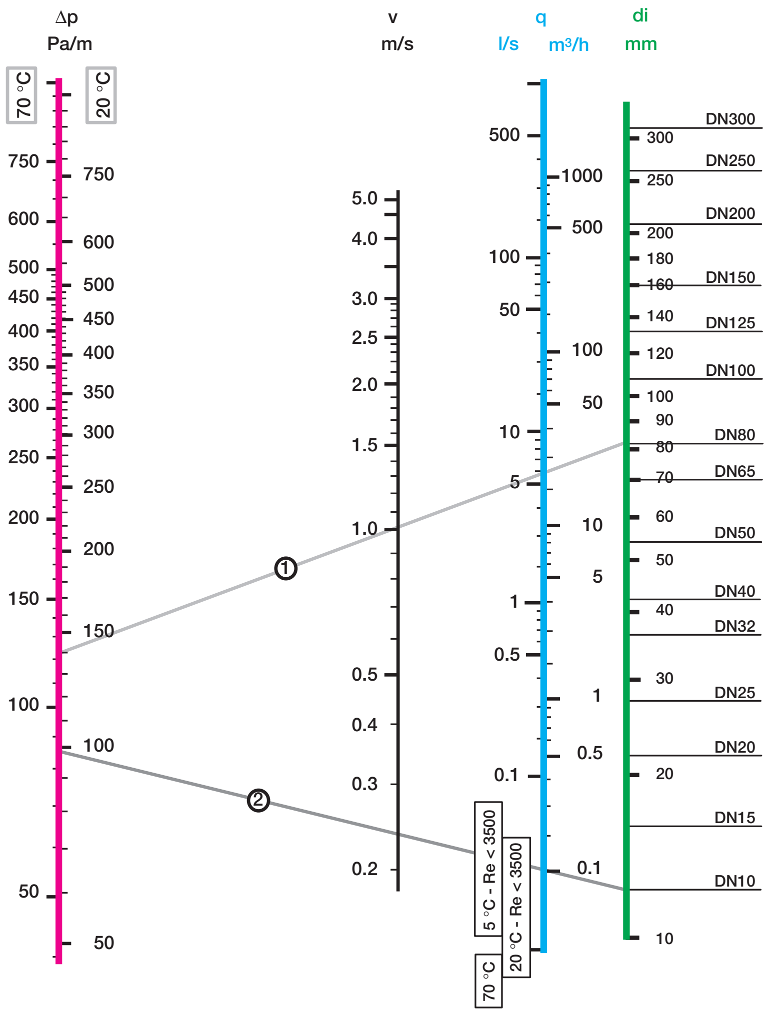


Fig B2. Pressure drops and velocities (pipes with a roughness of 0.05 mm) for water at 20 and 70 °C.

Example 1: Pipe DN80 (di = 82.5 mm) and water flow 20 m³/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 m³/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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HYDRONIC BALANCING WITH DIFFERENTIAL PRESSURE CONTROLLERS

Why and when variable flow HVAC systems require differential pressure control.

Leeds City Office Park, England

“Hydronic balancing with differential pressure controllers” is the fourth 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 third with balancing radiator systems.

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 —

Contents

1. Introduction

In theory, modern HVAC systems can satisfy the most demanding indoor climate requirements at low operating costs. In practice, however, not even the most sophisticated controllers always perform as promised. As a result, comfort is compromised and operational costs are higher than expected.

This is often because the mechanical design of the HVAC plant violates one or several of these three conditions:

1. Design flow must be available for all terminals when needed.
2. Differential pressure across control valves must not vary too much.
3. Flows must be compatible at system interfaces.

The second condition mainly concerns variable flow distribution systems. In such systems, differential pressure across control valves is variable. The circuits are also hydraulically interactive. Large variations in differential pressure will cause unstable modulating control. Since the circuits are interactive, disturbances in one part of the building will propagate to other parts.

To enable accurate and stable control of variable flow systems, it is often necessary to stabilise differential pressure. The best way is to use differential pressure control valves. They keep differential pressure variations within reasonable limits, and also make circuits independent of each other.

Differential pressure control valves give the following benefits:

1. They enable accurate and stable modulating control.
2. They minimise noise from control valves (on-off or modulating).
3. They simplify balancing and commissioning.

This handbook explains more in detail why it is important to use differential pressure control in variable flow HVAC systems. The handbook also provides methods for balancing.

2. Types of distribution system

In HVAC plants, water distribution can be obtained at constant or variable flow. Each type of distribution has advantages and disadvantages.

2.1 Variable flow

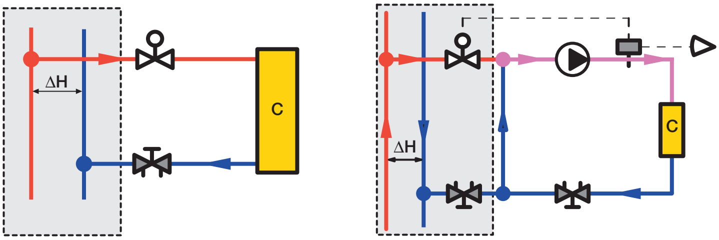


Fig 2.01. Examples of variable flow distribution systems.

In variable flow distribution systems, control is mainly obtained using two-way control valves.

Advantages

- Pumping costs depend on the product of pump head and water flow. The more the control valves reduce the flow, the more energy is saved. This is particularly interesting in cooling where distribution pumping costs, at constant flow, represent between 6 and 12% (section 5.4) of the average energy consumption of the chillers.
- The plant can be designed with a diversity factor. This can be the main reason for converting from constant flow distribution to variable, allowing a plant expansion using the same pipe work.
- Since full load is uncommon, pipes may be designed with higher pressure drops, reducing investments costs.
- Production and distribution flows are compatible ensuring a constant supply water temperature at all loads. This is important in cooling plants essentially for dehumidification.
- The return water temperature can be minimised in heating and maximised in cooling. This is important in district heating/cooling and when using condensing boilers.

Disadvantages

- Differential pressures across circuits are essentially variable. This affects the control valve authority and the stability of control loops working in proportional or PI/PID mode.
- Sizing a two-way control valve is not easy as it depends on the available ΔH for the circuit. This value is generally unknown and is essentially variable.

- Circuits are interactive. When one control valve closes, the differential pressure on other circuits increases. The associated control valves must close to compensate. If one or several loops are unstable, control problems can spread to all circuits.
- With a normal load of 50%, flow is reduced to 20% and differential pressures available on all circuits increase dramatically, decreasing drastically the authority of the control valves with a risk of hunting.
- A minimum flow must be obtained to protect the pump, requiring specific solutions.

2.2 Constant flow

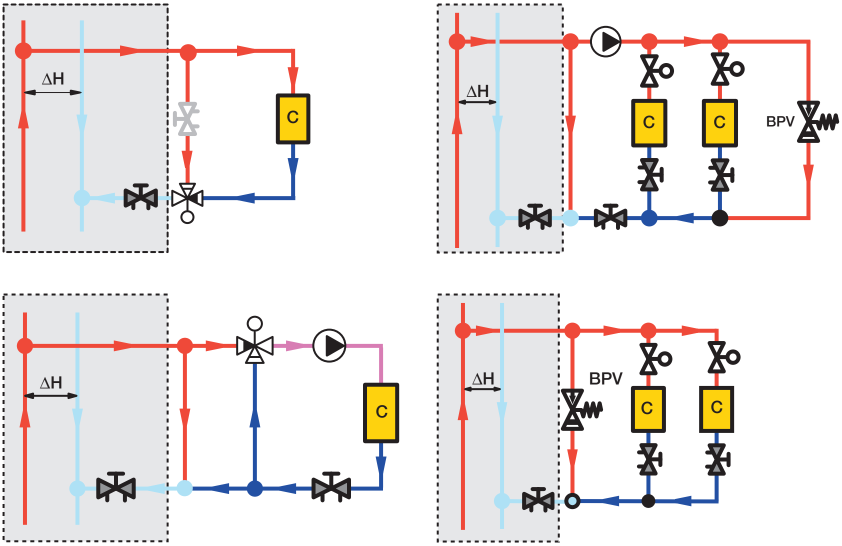


Fig 2.02. Some examples of constant flow distribution.

Advantages

- Pump head is constant, pressure drops in distribution pipes are also constant and circuits are not interactive. Consequently, each circuit receives a constant differential pressure and working conditions are maintained at all loads, which is favourable for the control loops.
- Sizing the control valves is easy. The sizing of a three-way valve in a diverting circuit is based on the same pressure drop as for the terminal unit at design condition. This pressure drop is normally well known. The control valve authority is constant and, in some cases, can be close to one.
- Supply water temperature is more uniform throughout the plant.

Disadvantages

- Pumping costs do not decrease with load.
- Design of the whole distribution system must take into account that all terminals work at maximum flow all the time. Designing the plant with a diversity factor is not possible.
- Return water temperature is not minimised in heating or maximised in cooling. District heating/cooling companies do not appreciate this. In heating, a higher return water temperature is not suitable for condensing boilers.
- When several production units work in sequence, production and distribution flows are not compatible at partial loads. The difference in flows creates a mixing point and the supply water temperature cannot be maintained constant, which will cause problems in cooling systems.

The choice between constant and variable flow distribution systems depends on the use of the plant and the importance attached to specific advantages and disadvantages.

3. Why differential pressure controllers are useful

3.1 Ensure accurate and stable modulating control

3.1.1 THE CONTROL LOOP

3.1.1.1- Elements of a control loop

In heating and air conditioning plants, control loops generally affect a temperature or a flow in order to act on the system in which we want to control one physical value (e.g., temperature).

How effective the control loop is, depends on the combination of six interactive elements that form the loop.

1. The sensor detects what is to be controlled, such as room temperature or supply water temperature.
2. The controller compares the measured value with its set value. According to the difference between these two values, the controller reacts depending on its nature (e.g., on-off or PID) and controls the valve motor.
3. The motor activates the valve in accordance with instructions received from the controller.
4. The two- or three-way valve controls the flow and consequently the quantity of energy to be transmitted to the system to compensate for disturbances.
5. The terminal unit transmits this energy to the controlled system.
6. The controlled system is, for example, a room in which the sensor is located.

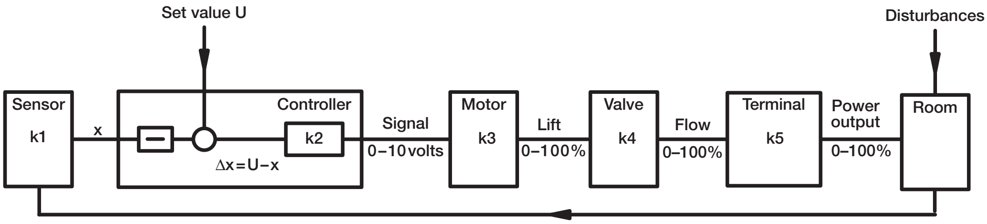


Fig 3.01. The six elements of the control loop interact with each other.

The controller is the brain of the control loop. To obtain a stable "marriage", the control principle chosen must suit the system design

Controllers can be divided into two classes: discontinuous and continuous (modulating controllers). On-off controllers belong to the class of discontinuous controllers. In heating, if it is too hot, the control valve is fully closed, and fully open when it is too cold. Regardless of whether the valve is open or closed, too much or too little heat is always supplied and the controlled value cannot reach a stable state. It continuously oscillates between a maximum and a minimum. Better comfort can be achieved with modulating control based on proportional mode, which also decreases energy consumption.

3.1.1.2 Proportional control

A proportional controller opens or closes the control valve in proportion to the difference between the controlled value and the set value. The control valve finds stable positions corresponding to an energy balance. The supply air temperature and room temperature thus stabilise, significantly improving comfort.

Figure 3.02 shows a level control loop using a proportional controller. Level H must normally be held constant by action on the supply flow Y, which is designed to compensate for disturbances Z.

When level H drops, float B goes down and proportionally opens the control valve V. The system finds a state of equilibrium when flows Y and Z are equal.

When Z = 0, the water level increases until it reaches level H_o, making Y = 0.

When Z = max, a stable equilibrium is achieved with the float at H_m, which is obtained when the control valve is fully open.

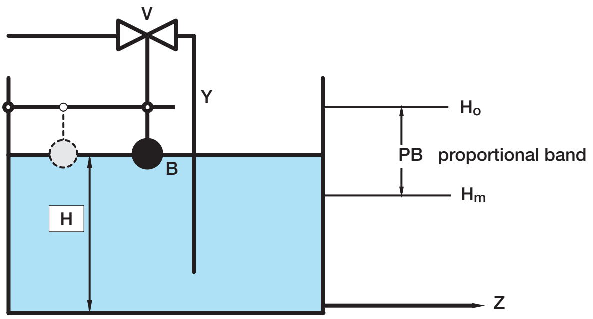


Fig 3.02. Principle of proportional control.

This therefore gives stable values to the level between limits H_o and H_m.

The H_o–H_m variation of the controlled value, which determines the valve setting between the closed position and the fully open position, is called the proportional band (PB). Equilibrium level conditions, depending on the disturbance Z, are always located within this band.

Moving the float towards the lever arm rotation point reduces the proportional band. In doing this, the level variation necessary to fully open the valve is reduced. However, a small level difference then produces a large variation of flow Y, and a stronger reaction than the disturbance. This then creates a larger inverse disturbance. The loop becomes unstable and operates in on-off mode, with poor performance.

The system shown in figure 3.02 is similar to a room temperature control system where:

Z = Heat losses / gains.

Y = Coil emission.

H = Room temperature.

In this case, the proportional band is equivalent to the room temperature variation necessary to move the control valve from the closed position to the open position.

In cooling, with a set value of 23°C and a proportional band of 4°C, room temperature will be 25°C at full load and 21°C at no load.

A proportional band of 4°C means that the valve opens 25% when the temperature increases by 1°C. The gain k2 of the controller corresponds to this value of 25% / °C.

However, the real physical input in the room is the power delivered by the terminal unit. The effective gain between the deviation in the room temperature and the power output of the terminal is:

k2 (controller) × k3 (motor) × k4 (valve) × k5 (terminal) = k. (See Fig 3.01).

If gain "k" is too high, the control loop is unstable. If gain "k" is too small, control is not accurate.

This gain should be chosen as high as possible without getting unstable control.

It is important to maintain gain "k" as constant as possible, to avoid unstable function in some conditions and inaccuracy in others.

Gain k4 (Fig 3.01) defines the ratio between the flow and the valve lift. It depends on:
1. The valve characteristic
2. The sizing of the control valve
3. The differential pressure (Δp) on the control valve

As shown in 3.1.1.3 below, the non-linearity of the terminal unit can be compensated by a suitable characteristic of the control valve to maintain the product of k4.k5 constant. However, gain k4 changes with the differential pressure across the control valve. There is no compensation for that and the only solution is to stabilise this differential pressure with a local controller.

3.1.1.3 Control valve characteristic

The relation obtained between the water flow and the valve lift, at constant differential pressure, defines the characteristic of a control valve. These two magnitudes are expressed as a percentage of the maximum values. For a valve with a linear characteristic, the water flow is proportional to the valve lift.

At small and medium loads, due to the non-linear characteristic of terminal units (Fig 3.03a), a slight opening of the control valve can significantly increase the emission. There is therefore a risk that the control loop may be unstable at low loads.

The control valve characteristic is chosen to compensate for the non-linearity of the terminal unit, so that power output from the terminal unit is proportional to the valve lift.

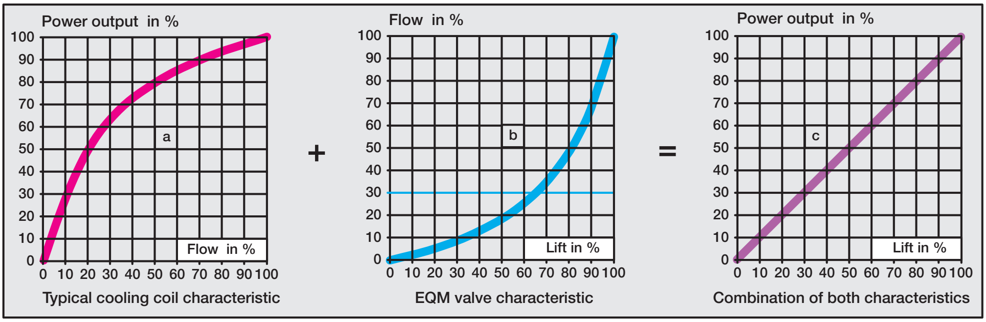


Fig 3.03. Select the control valve characteristic (middle) to mirror this of the terminal (left) to produce a linear relationship power output/lift (right).

If the output of a terminal unit is 50% of its design value when supplied with 20% of its design flow, the valve may be designed such that it allows only 20% of design flow when it is 50% open. 50% of power output is then obtained when the valve is 50% open (Fig 3.03c). Extending this reasoning to all flows, we can obtain a valve with a characteristic that compensates for the non-linearity of the controlled terminal unit. This characteristic (Fig 3.03b) is called equal percentage modified, EQM.

To obtain this compensation, two conditions must be fulfilled:
- Differential pressure across the control valve must be constant.
- Design flow must be obtained when the control valve is fully open.

Once a control valve has been selected and its design flow is given, differential pressure across the control valve, when fully open, can be calculated and is well defined.

If differential pressure across the control valve is not constant, or if the valve is oversized, the control valve characteristic is distorted and the modulating control can be compromised.

3.1.1.4 Control valve authority

When the control valve closes, the flow and pressure drop are reduced in terminals, pipes and accessories. This results in a higher differential pressure across the control valve. This distorts the control valve characteristic. This distortion of the control valve characteristic is represented by its authority β, which is defined as:

$$\beta = \text{valve authority} = \frac{\Delta pV_c \text{ (Pressure drop in control valve fully open and design flow)}}{\Delta p \text{ valve shut}}$$

The numerator is constant and depends only on the choice of the control valve and the value of design flow. The denominator corresponds to the ΔH available on the circuit. A balancing valve installed in series with the chosen control valve does not change either of these two factors and has consequently no influence on the control valve authority.

In a direct return distribution (Fig 3.04a), the remote circuits experience the highest ΔH changes. The worst control valve authority is obtained when distribution works at small flows, or in other words when the control valve is subjected to nearly all the pump head.

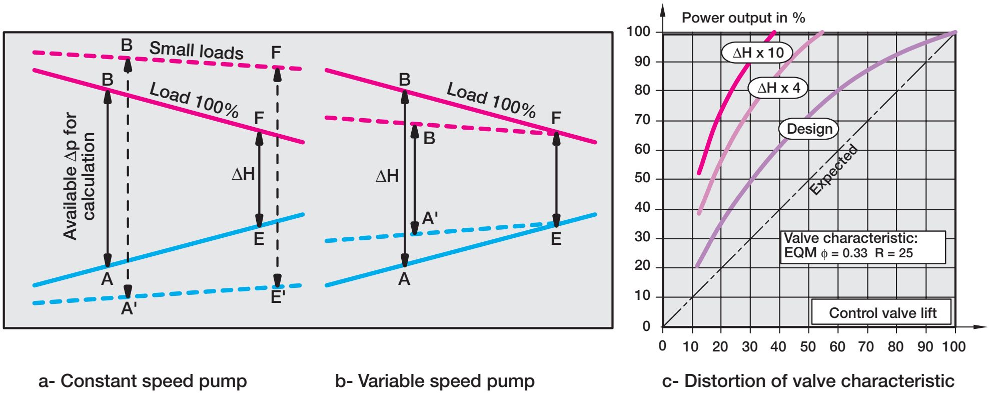


Fig 3.04. At design, control valve authority = 0.25. When the average load of the plant changes, differential pressure ΔH on the circuit also changes and distorts the control valve characteristic.

If a variable speed pump maintains constant differential pressure close to the last circuit (Fig 3.04b), the problem of variable ΔH is reported on the first circuit (more about that under section 5.3).

Figure 3.04c shows the relation between the power output and valve lift for an EQM control valve. This valve has been chosen to obtain, fully open, the design flow with an authority of 0.25. *When the ΔH applied on the circuit increases, the control valve authority decreases and the characteristic of the control valve may become so bad that it may create hunting of the control loop.*

As a rule of thumb, we size two-way control valves fully open and at design flow for a pressure drop at least equal to 25% of the maximum ΔH (generally the pump head) that can be applied across the control valve. To be able to select higher pressure drops in the control valves, the design ΔH must be high enough. This condition is not always fulfilled, because it will increase the necessary pump head and consequently pumping costs.

Example: In the table below, three cases are examined for the same plant. The pump head is selected to cover the need of the most demanding circuit. In the first case, pump head is 180 kPa and the minimum authority of the control valve is only 45/180 = 0.25. To obtain an authority of 0.5 (second case), the Δp of both the pump and the control valve must be increased by 90 kPa! In this case, the design pressure drop in the control valve will be 135 kPa, requiring a special design. The risk of noise increases. For the third case, a Δp controller stabilises the Δp across the control valve (Fig 4.07). The control valve authority is better than 0.7 and the necessary pump head is minimised.

	Control valve	Δp controller	Terminal unit	Distribution	Pump head	Authority β
1	45	None	40	95	180	45/180 = 0.25
2	135	None	40	95	270	135/270 = 0.5
3	20	10	40	95	165	20/(1.4x20) = 0.71
	kPa	kPa	kPa	kPa	kPa	See Fig 4.02a

Is differential pressure control essential in all variable flow systems?

Differential pressure control can prevent several operating problems in variable flow systems. These are probably the two most common:

- When a control valve (on-off or modulating) is subjected to a too high differential pressure, it cannot shut and may also produce noise. Differential pressure control can limit the differential pressure locally at a suitable level.
- When a modulating control valve is subjected to large changes in differential pressure, the valve authority may drop so much that temperature control becomes unstable or inaccurate. Differential pressure control can ensure that the control valve authority is sufficient for stable and accurate control.

One way to determine if differential pressure control is essential for accurate and stable control, is then to decide on a minimum acceptable control valve authority (0.25 for example), calculate the valve authority for all control valves, and select differential pressure control if many of the calculated valve authorities are below the accepted level.

Another, and much simpler way, is to compare the design differential pressure that must be available for the most remote circuit with the design pump head. As a rule, differential pressure control is essential in a variable flow distribution system when the ratio C is lower than 0.4 when:

$$C = \frac{\text{Design } \Delta p \text{ across the most remote circuit}}{\text{Design pump head}}$$

Example. Assume that, at design condition, the pump head in a plant is 100 kPa, the differential pressure across the most remote circuit is 40 kPa, and that 25 kPa out of those 40 kPa are applied across the control valve. The design control valve authority is then 25/40 = 0.625. The ratio C is 40/100 = 0.4 and the design pressure drop in the piping is 100 – 40 = 60 kPa. At first sight, this seems like a sufficient valve authority.

But, for an average total flow of 20% in the plant (50% load), the pump head increases to 130 kPa and the pressure drop in the piping is reduced by a factor 25 to 60/25 = 2.4 kPa. The differential pressure across the most remote circuit is now 130 – 2.4 = 127.6 kPa (3.2 times higher than at design condition). The differential pressure across the control valve fully shut is 127.6 kPa (5.1 times higher than the design value). The control valve authority drops to 25/127.6 = 0.2, one third of the authority at design condition!

So, for a plant with C = 0.4, a good control valve authority at design condition (0.625) drops to 0.2 when the average flow is 20% of design. This is why C = 0.4 can be considered the limit below which local differential pressure controllers are essential for accurate and stable control.

What will be the consequences of too low a control valve authority?
For an average load of 50%, which represents the most common situation, water flow is reduced from 100 to 20% (Fig 3.03a). Pressure drops in pipes and accessories become negligible and differential pressures on the circuits increase dramatically. The authority of the control valves is reduced below their design value. The control valves are obliged to work with a small opening, making modulating control quite difficult.

For an authority of 0.1, the minimum load controllable is close to 32% (Fig 3.05 right). Below this load, the control valve works in on-off mode. Note that in most countries, HVAC plants work below this minimum load for more than 35% of both the cooling and heating seasons!

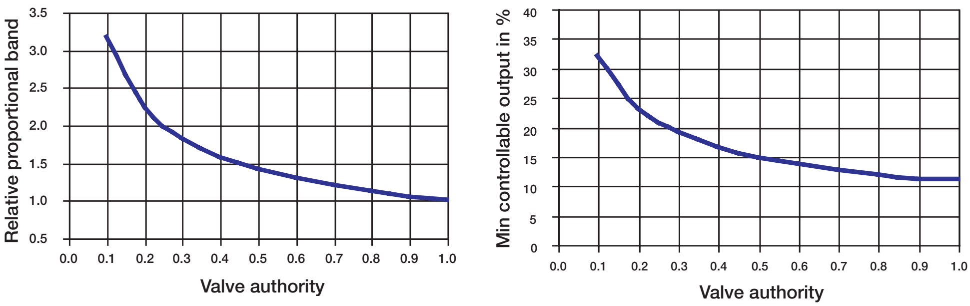


Fig 3.05. To obtain stable control, the required relative proportional band increases when the authority decreases. Below a certain minimum load control becomes unstable.

Let us consider that the required proportional band equals one for an authority of one. Figure 3.05 above shows that this required proportional band increases when the control valve authority decreases. For an authority of 0.1, the relative proportional band has to be multiplied by 3.

If the required proportional band, for instance, is 2°C with a valve authority of 1, it would be 6°C for an authority of 0.1. In this case, the room temperature will be stabilised at ± 3°C around the set value!

We may think that the integral function of most controllers will compensate for these deviations by slowly resetting the proportional controller. However, an integral function can only reduce such a deviation by half after around 30 minutes.

The following example shows why the integral function does not always compensate for deviations.
The sun can scorch the facing of a 50-floor building with up to 10,000 kW. If the sun goes behind a cloud, the heat gain can fall quickly to 1,000 kW, to increase again to 10,000 kW as soon as the cloud disappears. The room temperature will change radically and the integral function has not enough time to intervene. On the contrary, the integral function can create an overshoot, trying to compensate for the first disturbance when the reverse occurs.

Consequently, it is essential to guarantee a good authority for the modulating control valves. This prevents sophisticated PID controllers from just working in on-off mode.

In variable flow, stabilising differential pressure across the control valves with a self-acting differential pressure controller ensures a good authority. This allows the temperature controller to really control in modulating mode.

If the required proportional band is not adopted, the control loop will work in on-off mode.
In cooling, an on-off control cannot stabilise the room temperature, which oscillates between, for instance, 21°C and 25°C. This is obviously quite uncomfortable.

What will the user do to avoid temporary overheating?

He will decrease the set point of the room thermostat by 2°C, increasing energy consumption by as much as 20 to 30% (See section 5.4).

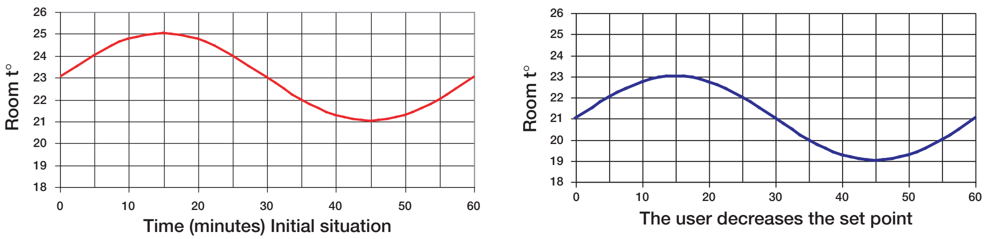


Fig 3.06. To compensate for overheating, the user decreases the set point of the room thermostat.

The final consequence of a low valve authority is a costly and uncomfortable indoor climate!

The authority of the control valve depends basically on how it is sized. It is therefore important to describe how a two-way control valve should be sized.

3.1.1.5 Sizing a two-way control valve

A control valve creates a pressure drop in the hydraulic circuit to limit the water flow to the required value. This pressure drop depends on the flow and the valve coefficient Kv representing indirectly the opening surface of the valve.

For a liquid with a relative density of one, the relationship between the flow, Kv and Δp (in kPa) is shown below:

Waterflow in l/h: $q = 100 \times Kv\sqrt{\Delta p}$ $\Delta p = (0.01 \times \frac{q}{Kv})^2$ $Kv = 0.01 \times \frac{q}{\sqrt{\Delta p}}$

Waterflow in l/s: $q = \frac{Kv}{36} \times \sqrt{\Delta p}$ $\Delta p = (36 \times \frac{q}{Kv})^2$ $Kv = 36 \times \frac{q}{\sqrt{\Delta p}}$

Sizing the control valve involves choosing the most suitable valve for the specific application from the commercially available Kvs (maximum Kv).

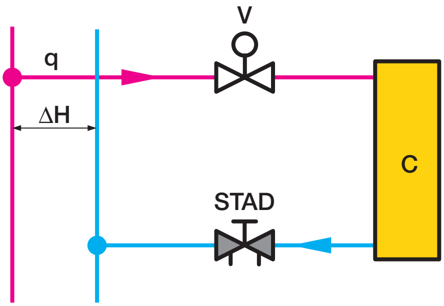


Fig 3.07. A two-way control valve.

Choice of the control valve Kv is based on a pressure drop ΔpV, where

$$\Delta pV = \Delta H - \Delta pC - 3\text{ kPa}$$

ΔH = the differential pressure across the circuit at design condition
ΔpC = the pressure drop in the terminal unit and accessories for design flow
3 kPa = the minimum pressure drop for the balancing valve to allow good measurement of the flow

Based on the theoretical design pressure drop ΔpV, the Kv value of the control valve is calculated using the formulas above. The Kv adopted is the nearest higher value that is commercially available.

The control valve authority is a consequence of its sizing and corresponds to the ratio $\Delta pV_c / \Delta H_{max}$. This value must be ≥ 0.25. If this is not the case, the pump head has to be increased so that we can choose a control valve with a smaller Kv, making this condition possible, or a local Δp controller has to be installed (section 4.2.3) to locally reduce ΔH_{max} for the same ΔpV_c.

Example: Differential pressure $\Delta H = 55$ kPa. For a design water flow of 500 l/h, the pressure drop in the terminal unit C is 10 kPa (including pressure losses in pipes and accessories).

What is the required pressure drop ΔpV in the control valve?
$\Delta pV = 55 - 10 = 45$ kPa minus a minimum of 3 kPa for the balancing valve making $\Delta pV = 42$ kPa.

In this example, $Kvs = 0.01 \times \frac{500}{\sqrt{42}} = 0.77$

Unfortunately, this Kvs does not exist in standard commercial ranges. Available Kvs are for instance:

$$0.1 - 0.16 - 0.25 - 0.4 - 0.6 - 1 - 1.6 - 2.5 - 4\ldots$$

The closest value Kvs = 1 is adopted.

The design pressure drop in the control valve is:

$$\Delta pV_c = \left(0.01 \times \frac{q}{Kv}\right)^2 = \left(0.01 \times \frac{500}{1}\right)^2 = 25 \text{ kPa}$$

The control valve authority is 25/55 = 0.45.

To obtain the design flow with the control valve fully open, the difference 55 – 10 – 25 = 20 kPa is taken away by the balancing valve. This does not change the control valve authority.

The authority calculated above is the design authority, taking into account the oversizing factor (Kvs = 1 instead of 0.77). However, without a local stabilisation of differential pressure, this authority can decrease dramatically due to higher differential pressures effectively applied on the circuit when the plant is working at small average loads.

3.1.2 BEHAVIOUR OF A VARIABLE FLOW DISTRIBUTION SYSTEM – PLANT EXAMPLE

In a variable flow distribution system, differential pressure across the circuits is essentially variable. To see what this means in practice, we will examine different aspects of a simple cooling plant with 10 identical terminal units.
Starting from the plant working at design condition and with a constant speed pump, we will study the evolution of the authority of one of the control valves when the plant is working at partial loads. The plant will be improved by using a variable speed pump in the distribution or local Δp controllers stabilising differential pressure across the control valves.

The steps are as follows:
1- The plant at design condition
2- What happens when one control valve closes?
3- The plant works with a total average flow of 50% (80% of design load)
4- The use of variable speed pumps
5- The use of local Δp controllers
6- Comparison of results

3.1.2.1 Plant at design condition

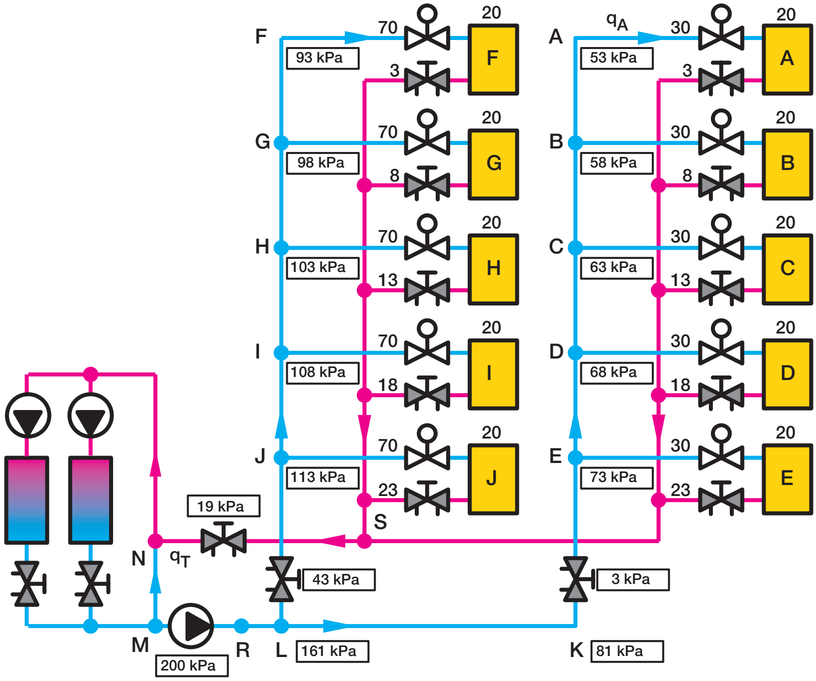


Fig 3.08. Cooling plant working at design condition.

Two chillers with their own pump serve a plant with two risers and identical terminal units. A bypass line MN avoids any interactivity between the chillers and between production and distribution. A constant speed pump generates the water flow in the distribution. Two-way control valves determine the water flow in the terminal units to keep room temperatures constant. Hydronic balancing is obtained with manual balancing valves. All other components are not represented to simplify the illustration.

For design flow, the commercially available control valves have, for example, a pressure drop of 13, 30, 70 or 160 kPa. We consider that the designer has chosen the control valves with a design pressure drop of 30 kPa for the remote riser, while the first riser is equipped with control valves that have 70 kPa pressure drop. When all the control valves are closed, pump head is equal to 266 kPa.

Balancing valves allow design flow in each terminal, avoiding:

- Overflow in some circuits creating underflow in others
- A general overflow q_T making the distribution flow incompatible with production flow. Such an overflow would create a reverse flow in the bypass MN (Fig 3.08) with a mixing point in M and an increase of the supply water temperature, making the maximum power installed not transmittable. This typical phenomenon is examined in detail in Handbook 2.

The purpose of balancing valves is to obtain the correct flows at design condition, guaranteeing that all control valves can obtain at least their design flow under all other conditions. A balancing valve incorporates a shut-off valve with a mechanical memory of the set position, and it is also a diagnostic tool that enables flow measurement.

3.1.2.2 What happens when the control valve of terminal "A" closes?

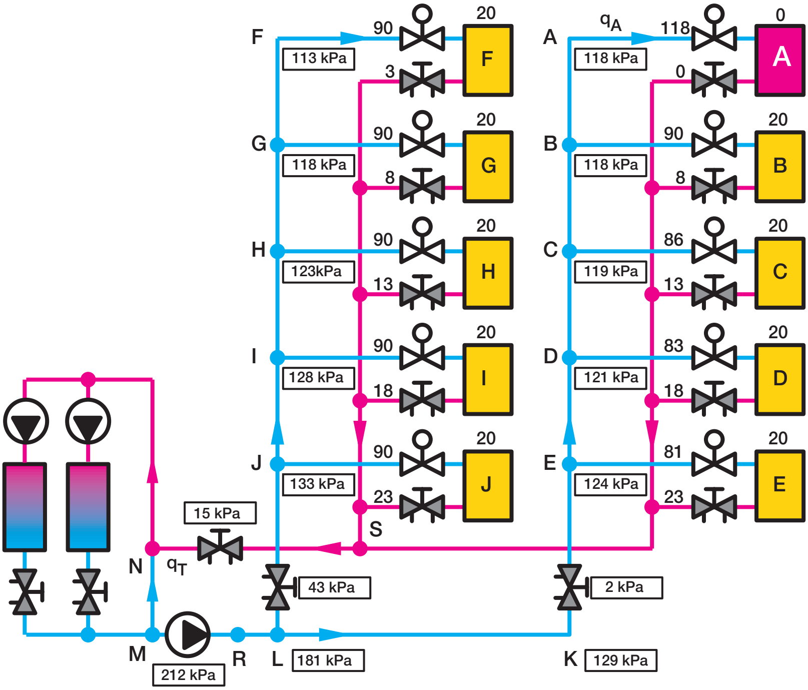


Fig 3.09. The control valve of terminal A closes.

When the control valve of terminal A closes, differential pressure on A increases from 53 to 118 kPa. As the Δp for the control valve fully open at design flow is equal to 30 kPa, the valve authority is 30/118 = 0.25 and not 30/53 = 0.57 (Fig 3.08) as could be expected at first sight. Differential pressures on the other terminals increase dramatically, indicating strong interactivity between the terminal units. This interactivity is important in this example as each circuit represents 10% of the total flow.

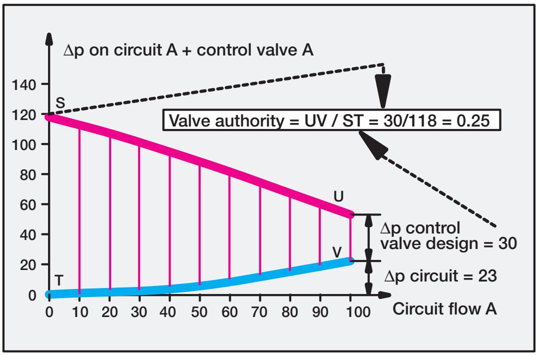


Fig 3.10. When the valve of circuit A closes (Fig 3.09), differential pressure applied on this circuit increases from 53 to 118 kPa while the rest of the plant remains at design flow.

With an authority of 0.25, the relation between valve lift and power output is distorted. For a linear valve, sized exactly, and opened at 20% of maximum lift, power output is already 63% of design value.

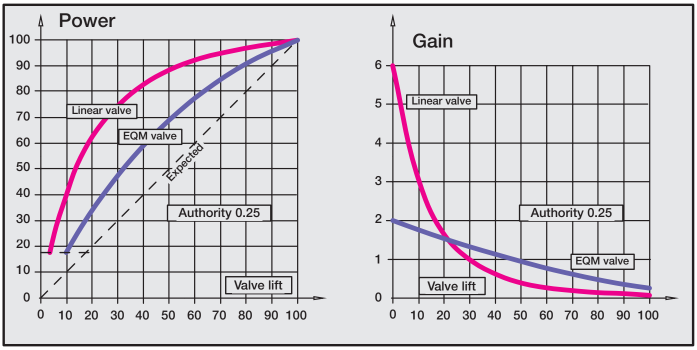


Fig 3.11. Power and gain versus valve lift at design condition for terminal A.

Figure 3.11 left shows the relation between power output and valve lift. For circuit A with a linear valve (Fig 3.11 right), the maximum gain k4 × k5 = 6 (See section 3.1.1.1). To compensate for this high gain situation, the proportional band of the controller has to be multiplied by the same factor (6) dramatically reducing the accuracy of room temperature control.

The EQM characteristic is much better, but an authority of 0.25 is the lowest acceptable as output is already 32% for a lift of 20%. The situation will worsen when the plant mainly works at small average loads. Then, differential pressures across the control valves increase, further reducing their authority.

3.1.2.3 Total average flow of 50% (80% of design load)

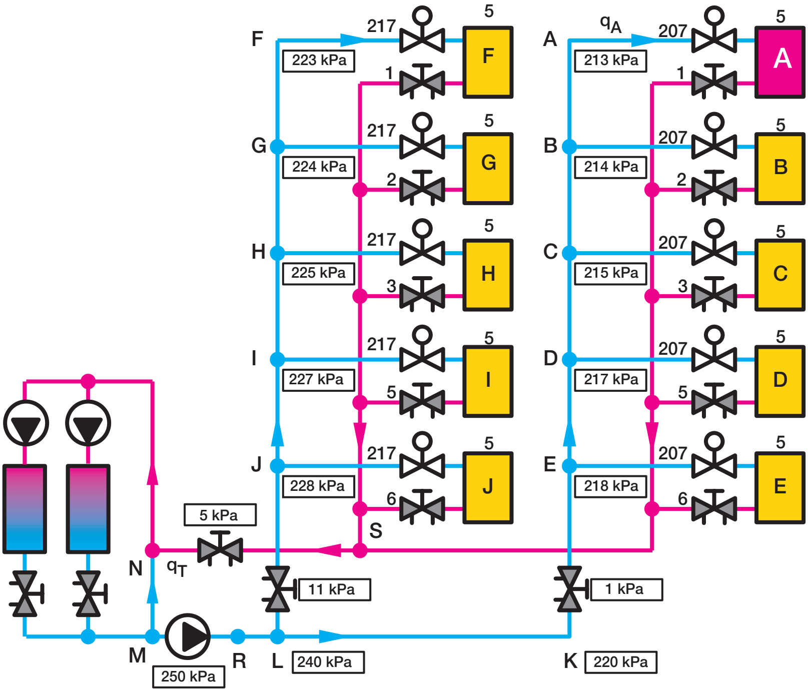


Fig 3.12. Total flow q_T = 50% of design.

When the plant works with a load corresponding to 80% of design value, the total water flow drops already at 50% of design (Fig 3.03a). The pump head increases from 200 to 250 kPa and pressure drops in pipes and accessories decrease by 75%.

This situation is represented in figure 3.12.

Differential pressures across the control valves increase dramatically and their authority decreases. This situation can be observed for circuit A in figure 3.13.

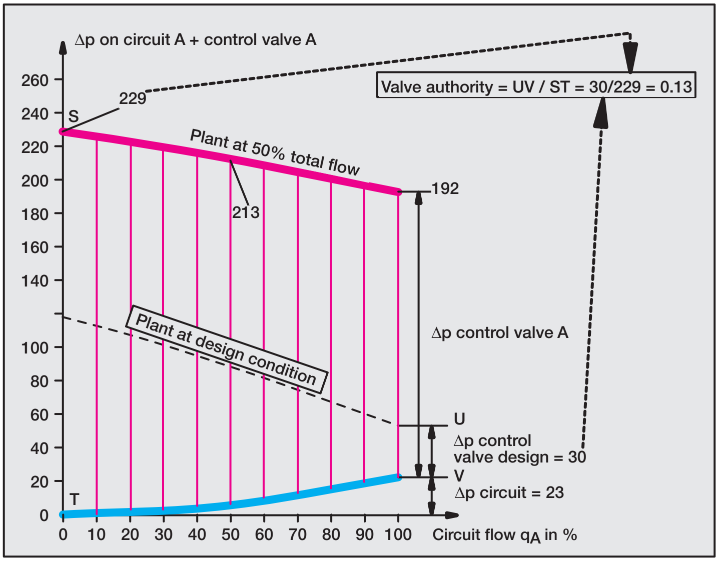


Fig 3.13. The entire plant works uniformly at 50% of total design flow.
The control valve A opens from 0 to design flow.

With a linear valve, 78% of maximum power is already obtained for the valve lift at 20% (Fig 3.14A).

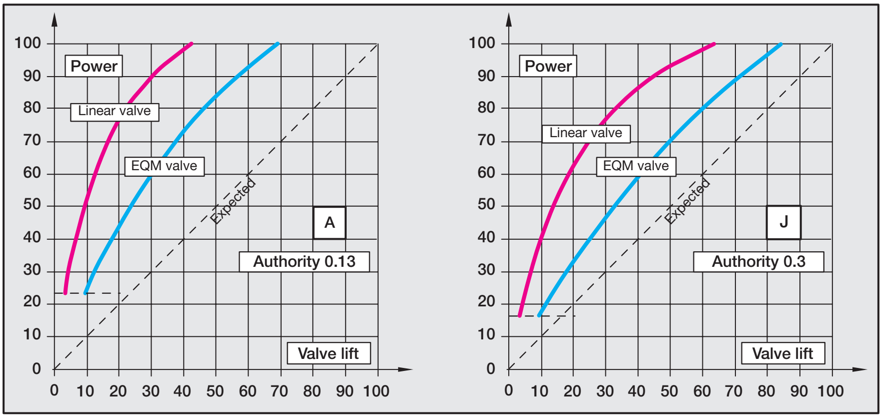


Fig 3.14. Power versus valve lift for terminals A and J – q_T = 50%.

With the EQM characteristic, control of terminal A is much better but remains difficult while terminal J is controllable (Fig 3.14 right).

The control valves have been chosen at the best values possible and it remains difficult to obtain stable control of the terminal units without a substantial increase in the proportional band of the controllers. This does not allow optimum performance. Imagine what would happen if control valves were not well sized.

If all control valves have a suitable characteristic and a design pressure drop at least equal to 25% of maximum pump head, working conditions are good and the plant can be balanced at design condition with balancing valves. Since underflows are avoided at design condition they cannot occur at partial load as differential pressure can only increase when the load decreases. If a minimum authority of 0.25 cannot be obtained, the situation can be improved with variable speed pumps.

In many countries, the average cooling load is around 50% of design with an average water flow of 20%. In these cases, pressure drops in pipes and accessories are only 1/25 of their design values and they become negligible. Therefore, the circuits are directly submitted to the full pump head and all control valves try to modulate the flow, while being almost completely shut.

3.1.2.4 The use of variable speed pumps

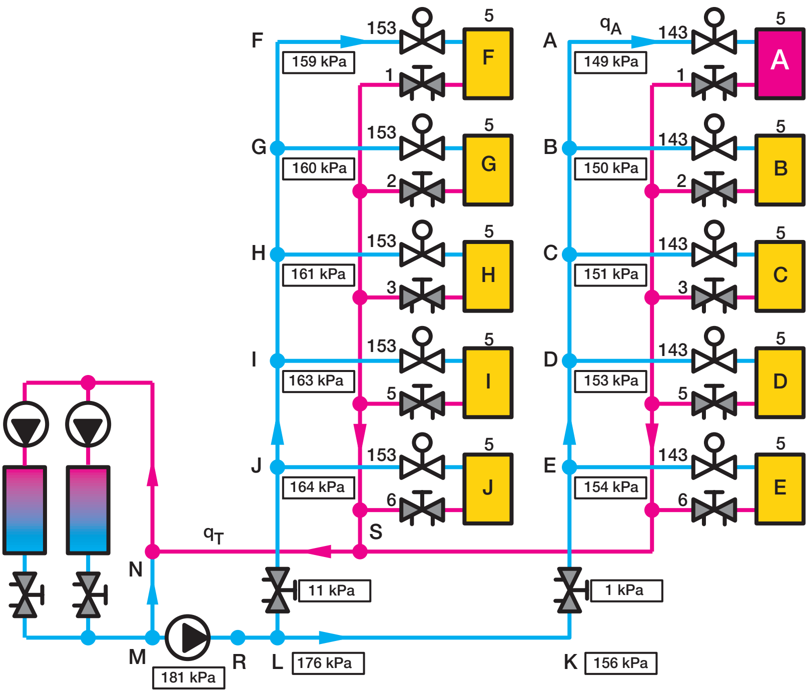


Fig 3.15. The variable speed pump works with a constant pump head and q_T = 50%.

A variable speed pump replaces the constant speed pump of figure 3.08. In this case, the main balancing partner valve is not required.

When the plant works with a total average flow of 50%, the pump head remains unchanged. The pressure drop across the control valve A fully shut decreases from 229 kPa (Fig 3.13) to 161 kPa (Fig 3.16), increasing the control valve authority from 0.13 to 0.19.

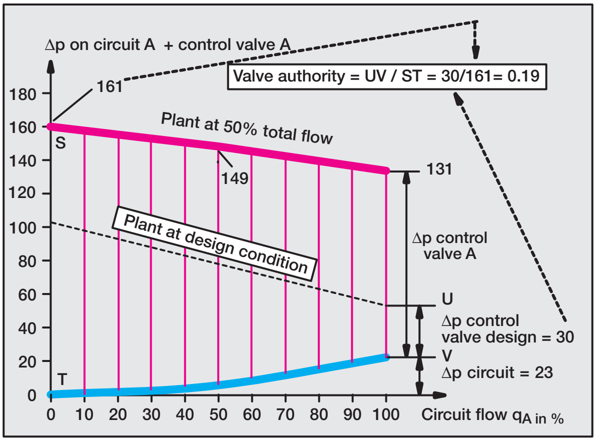


Fig 3.16. Evolution of the Δp across the control valve A when the plant works on average at 50% flow.

Other ways of controlling the variable speed pump are examined in appendix 5.3.

3.1.2.5 The use of local Δp controllers

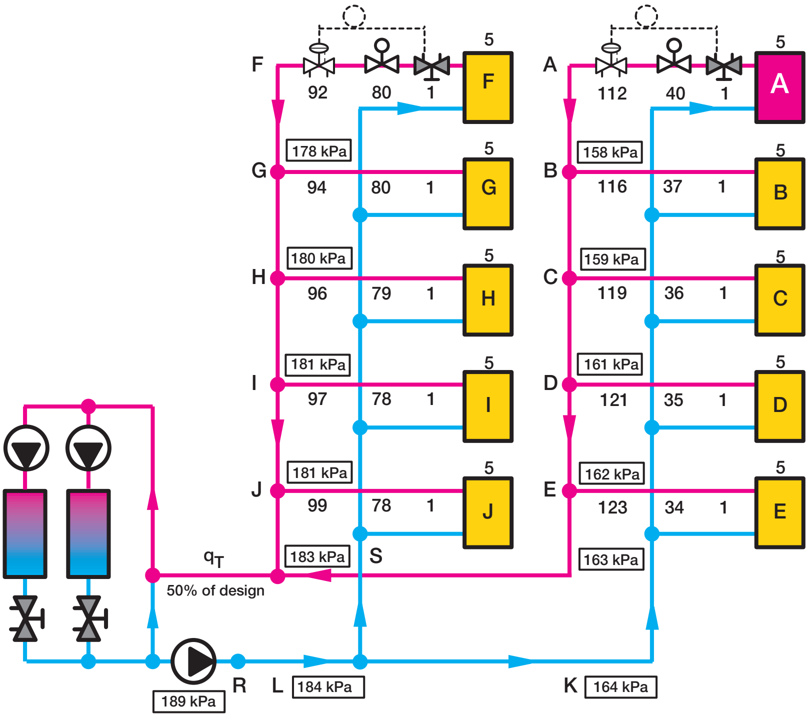


Fig 3.17. A Δp controller stabilises the differential pressure across each control valve. $q_T = 50\%$ of design.

As in previously examined cases, the design Δp across the control valves is 70 kPa for the first riser and 30 kPa for the second riser. In figure 3.17, this differential pressure is stabilised with a local self-acting differential pressure control valve, STAP. The set point of each STAP is chosen to obtain design flow across each control valve fully open. As all control valves provide their design flow when fully open, they are automatically well sized.

To simplify the figure, only circuits A and F are fully represented.

At small average loads, the Δp across the control valves increases a little due to the proportional band of the STAP, but the control valve authority always remains above 0.7 in the worse case.

This is, without any doubt, the best technical solution when differential pressures vary dramatically with the average load. With such a good authority, an equal percentage control valve has practically a linear relation between valve lift and power output. The proportional bands may be adjusted at minimum values, guaranteeing a comfortable indoor climate at the lowest possible cost.

It is possible to adopt the same design pressure drop for all control valves (30 kPa for instance or lower) without fundamentally changing the results.

With STAP on the control valves, and for modulating control of the terminal units, there is no practical difference, from a control point of view, between a constant and a variable speed pump. But variable speed pumps may be used to decrease pumping costs.

3.1.2.6 Comparison of results and conclusions

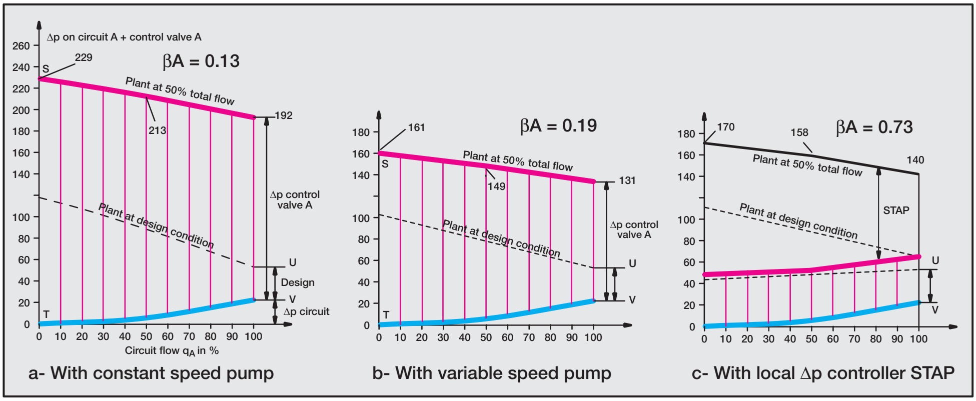


Fig 3.18. Evolution of differential pressure across control valve A, when the plant works with an average water flow of 50% corresponding to an average load of 80%. βA, in the figure, represents the authority of control valve A.

With a constant speed pump, the differential pressure across the control valve varies dramatically from 30 kPa (UV) at design condition to 229 kPa (ST) when the average water flow in the plant q_T equals 50% of design. With a variable speed pump, this differential pressure varies from 30 to 111 kPa, and from only 30 to 41 kPa with a STAP local differential pressure controller.

In HVAC plants where pressure drops in distribution represent more than 60% of the pump head, it is necessary to stabilise differential pressures locally with self-acting differential pressure control valves.

Some users claim they don't see an improvement in comfort when using modulating controllers in comparison with on-off controllers. As explained in this section, this must be because modulating control valves, submitted to large Δp changes, only work in on-off mode!

3.2 Minimise noise from control valves

3.2.1. SOME TYPES OF NOISE

Valves that control water flows often emit noise when the pressure drop exceeds a critical value.

The pressure drop across two-way valves used in district heating/cooling systems can be considerable. Control should be silent, since such valves are frequently installed in sub-stations close to dwellings.

Valve noise can be classified as follows:

- Mechanical noise
- Flow noise
- Cavitation noise

Mechanical noise occurs when parts of the valve start to vibrate, due to the flow. The tendency of a particular valve to vibrate depends on how well the plug and stem are guided in the valve body. Such vibration can rapidly destroy the plug and seat and cause metal fatigue in the stem. With constant differential pressure, this noise increases when the valve opens because the plug is generally freer in this position (with the exception of the full valve opening) but also because water flow increases. With a constant valve opening, noise increases with differential pressure across the valve.

Flow noise is a hissing sound that increases with the turbulence of the flow through the valve. It obviously increases with water velocity depending both on the flow rate and differential pressure across the control valve. If the valve is well designed, this noise level is normally low, but flow noise in piping, bends, cavities and sudden geometrical transitions is not negligible. Quite often it is attributed to the valve.

Air in water is also a very common source of noise. When static pressure decreases, as for instance between the plug and seat of the valve, a part of the air dissolved in the water comes out creating a particularly noisy emulsion. Microbubbles get stuck on metal particles, stick together and rise, causing incorrect flow measurements in measuring units.

Cavitation noise occurs in liquid when the pressure drop across the valve exceeds a critical value. This depends on the fluid, its temperature, the geometrical design of the valve and the local static pressure.

Total pressure at a certain point in the system corresponds to the constant pressure maintained by the expansion tank, the pump head, the pressure drop in pipes and accessories and the level difference between the expansion tank and the point considered. As the liquid passes the valve, water velocity increases and consequently also dynamic pressure, which depends on the square of the water velocity. Static pressure decreases when dynamic pressure increases and the value can drop below atmospheric pressure. If flow velocity is so great that static pressure drops to vapour pressure level, steam bubbles will form and the liquid will start to boil.

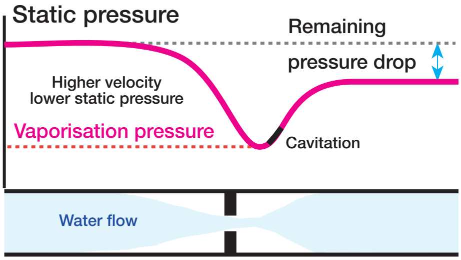


Fig 3.19. When static pressure reaches vaporisation pressure, the liquid starts to boil. When static pressure increases above vaporisation pressure, steam bubbles implode and the valve cavitates.

Once the liquid has passed through the smallest area of the valve plug and seat, the velocity of the flow decreases and static pressure increases. Static pressure rapidly exceeds vapour pressure and steam bubbles implode.

When steam bubbles implode, powerful shock waves are generated which can quickly erode and damage the valve. Apart from reducing the life of the valve, cavitation causes a considerable amount of noise. It may also cause a chirping sound that sometimes reaches 120 dBA with pressure waves above 100 bars!

At the outlet of the control valve, static pressure must theoretically return to the same value as in the inlet, because water velocity and consequently dynamic pressure are the same. However, some energy is lost due to friction in the valve. Static pressure at the outlet is therefore lower than in the inlet. The difference is the pressure drop in the control valve. Energy corresponding to the pressure drop partly becomes noise. The pressure drop equals differential pressure across the control valve.

Common factors that decrease the risk of cavitation are:

- Higher static pressure
- Low differential pressure across the valve
- Low fluid temperature
- Suitable valve design

3.2.2. WHAT CAN BE DONE?

From this short review, it is obvious that differential pressure has to be limited to avoid too high water velocity in control valves.

For a given control valve, the risk of noise also increases with water flow. Therefore, it is essential that this flow is limited at design value by adequate hydronic balancing.

Both modulating and on-off control valves are designed for a certain differential pressure. However, noise can occur even at lower differential pressures due to media conditions and the specific design of the valves. This is particularly unpleasant when the control valves are situated close to tenants, which is mostly the case. For radiator valves designed for a maximum Δp of 3 bars for instance, noise can occur when Δp is higher than 20 to 30 kPa.

At design condition, there is no problem if differential pressures are reasonable. But the plant mainly works at partial loads with an average load of 50% of design. At this load, water flow in the distribution is close to 20% of design (Fig 3.03a). The pressure drops in pipes and accessories are 1/25 of those at design condition and differential pressures on the control valves are much higher.

In this situation, noise in the control valves can dramatically increase particularly when there is air in the water. In appendix 5.3, we show that a variable speed pump can help somewhat but does not solve the problem totally. A practical solution to the noise problem is to use differential pressure control valves to take up the excess pressure and to vent the system.

Since the excess pressure is transferred to the differential pressure control valves, it may appear that we have merely moved the noise problem from one control valve to another. This, however, is not the case for the following reasons:

1- The excess pressure is shared between two control valves.
2- The differential pressure control valve has features specially designed to reduce noise:
   - Balanced plug
   - High time constant preventing the plug from vibrating at audible frequencies.
3- The differential pressure control valve is often located far from tenants and any remaining noise has little or no impact.

For these reasons, a differential pressure control valve efficiently minimises noise from temperature control valves.

3.3 Simpler balancing, commissioning and maintenance

Without differential pressure controllers, circuits in variable flow systems are interactive.

This means, for instance, that flow variations in one unstable control loop will create variable differential pressures for all the other circuits. The control loops in these circuits will then try to compensate for the changes in differential pressures. This will give an impression of unstable control. Attempts to solve the problem by changing control parameters (proportional band, integral and derivative times) do not solve the problem. For the control technician, the situation may quickly deteriorate into a nightmare.

Differential pressure control valves make circuits independent of each other.

When differential pressure controllers protect the branches in a plant, no branch is influenced by disturbances originating in other branches. This simplifies balancing and commissioning. In a large apartment block, for example, not all units are sold or let at the same time. When a local differential pressure controller protects all units, new units can enter service without disturbing those already in use (Fig 3.20)

Another advantage is the possibility of balancing a plant designed with a diversity factor, considering that all units will never work together at full load simultaneously. If fitted with manual balancing valves such a plant requires a special procedure, as it is not possible to obtain design flow simultaneously in all terminal units. Balancing a plant with a diversity factor is easy when differential pressure controllers make the circuits independent.

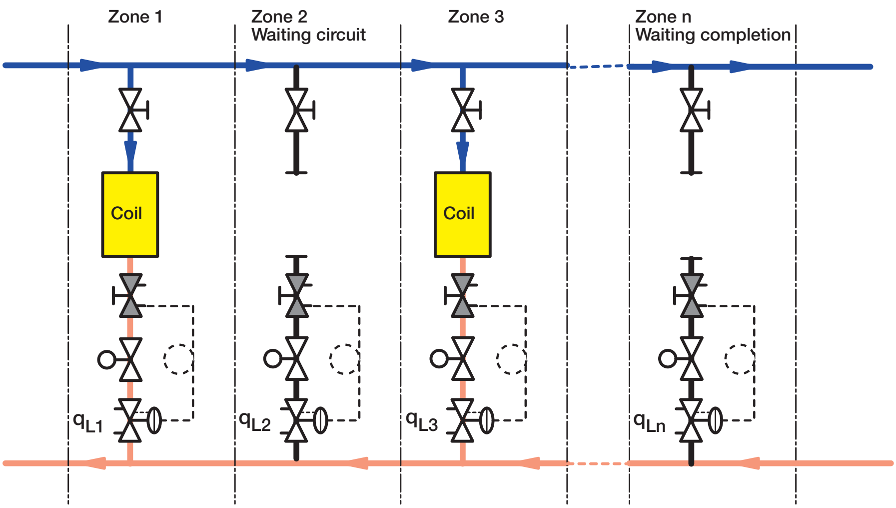


Fig 3.20. When a new circuit is put in service, this does not influence the other circuits.

In brief, making circuits independent of each other using differential pressure controllers gives the following benefits:

1- No balancing devices are needed upstream of the differential pressure control valves (Fig 4.05).
2- One unstable control loop will not cause oscillations in other control loops.
3- Each branch may be balanced independently. No re-balancing of existing circuits is needed should a building be extended.
4- It is easy to balance a plant designed with a diversity factor.

When you install the differential pressure control valve in tandem with a measuring valve, water flows and differential pressures are always measurable. This is essential for diagnostic purposes.

In a district heating or cooling distribution system, a new plant can be put in operation without affecting all the other plants in service. In some cases, the differential pressure controller is associated with a flow limiter. If the required water flow in one building exceeds planned levels, the flow limiter will react to avoid this excess flow. Available differential pressure will decrease creating problems for tenants. If the owner balances his plant carefully these problems can be avoided.

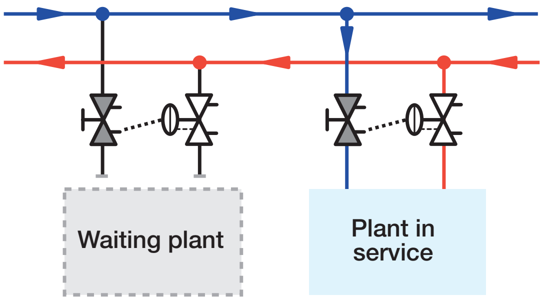


Fig 3.21. An entire new plant can be put in service without disturbing all other plants.

3.4 Benefits of differential pressure control and summary

Symptoms indicating a need for Δp control	Typical but often neglected root causes	Common but incorrect counter-measures	Common and costly consequences
Modulating control valves work in on/off mode. Room tem-peratures oscillate.	Variable differential pressure: Low valve authority means control valve characteristics are distorted.	Widen p-band to get stable control.	Modulating but inaccurate control. Comfort target is not met.
	Interactive circuits: When some control valves close, differen-tial pressure across other control valves increases. These react to compensate and room tempera-ture fluctuates.	Decrease set point of room thermostats in cooling (increase in heating) to counter temporary deviations in room temperature.	Average room temp-erature is lower in cooling and higher in heating. Energy costs increase by 10–15% per degree in cooling and 5–8% in heating.
Control valves (on/off or modulating) on terminals produce noise.	Differential pressures on control valves are too high.	Reduce pump head.	Required power is not available. Comfort target is not met.
Control valves cannot close.	Differential pressures on control valves are too high.	Install more powerful control valve motors.	Unnecessary investment.

Correct remedy	Advantages	Benefits
• Stabilise differential pressure using local differential pressure controllers.	• Stable differential pressures. • Circuits not interactive. • Lower differential pressure across control valves. • Use of cheaper control valves possible.	**In operation:** • Accurate and stable control. • No noise from control valves. • Better comfort, lower energy costs. **In commissioning:** • Simple balancing procedure. • Straightforward commissioning in stages. • Easier to balance a plant designed with a diversity factor.

SUMMARY OF THIS SECTION

Differential pressure controllers mainly do two things:

1- They stabilise differential pressure across control valves.
2- They make circuits independent of each other.

These two functional advantages translate into three clear benefits:

1- Δp control enables accurate and stable modulating control.
2- Δp control minimises noise from control valves.
3- Δp control simplifies balancing and commissioning.

Stable and accurate modulating control

In plants with a variable flow distribution system and relatively high pipe pressure drops, differential pressure control valves are necessary for accurate and stable control. Without them, you either have to accept:

a- unstable control (on-off) although you have invested in expensive PID control,
b- or widen the proportional band until you get stable but inaccurate control

If you choose alternative "a", the controller cannot find a stable level for the room temperature. In cooling, it will oscillate forever between, for instance, 21°C and 25°C. What does the user do? He or she reduces the set point of the room thermostat, often to the minimum set point or at least 2°C below the desired value. Over time, when all users do the same thing, and the average building temperature is 2°C below the expected level, energy consumption will increase by 20 to 30%!

If you choose alternative "b", the controller can find a stable level for the room temperature when the load is above a certain level. But the room temperature may be anywhere between, for instance, 19°C and 25°C, instead of between 21°C and 23°C which is often the specified level. The I-function of advanced controllers will try to bring the stable but incorrect room temperature back to the correct level, but it takes time. If the building is subjected to quick load changes, the I-function is not given enough time to adjust room temperature before the next disturbance. Solar radiation will increase room temperature. This will prompt the user to decrease the set point of the room thermostat. But as soon as the solar radiation disappears, it will be too cold and he will again change the set point, at the same time as he complains about the air conditioning system.

Differential pressure control is often a necessity for accurate and stable modulating control and can prevent energy waste in the region of 15–25%.

Minimises noise from control valves

Noise from control valves increases with differential pressure. This noise on terminal units can be particularly irritating because it is emitted so close to tenants.

At design condition, this is normally not a problem if differential pressures are reasonable. In reality, when the plant is working at, say, an average of 50% of maximum load, the water flow is only 20% of design and the pressure drops in pipes and accessories are 25 times lower. Thus, the differential pressure increases dramatically across each control valve, which can become noisy.

A differential pressure control valve stabilises differential pressure at a correct level across the circuits, minimising noise from all types of control valves.

Simplifies balancing and commissioning

Differential pressure controllers make circuits hydraulically independent of each other. This fact is of fundamental importance for commissioning:

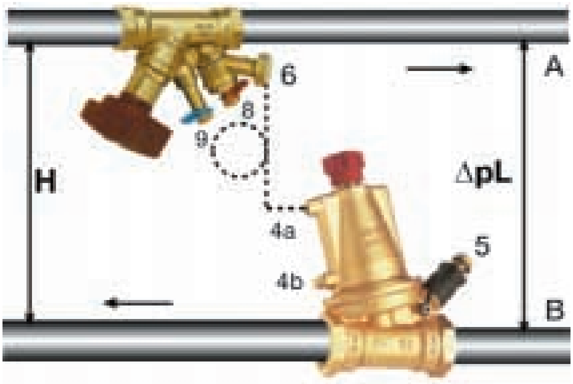
- No balancing valves are required upstream of differential control valves.
- Hydronic balancing is simplified, as the circuits are not interactive.
- A new section may be commissioned without disturbing all other sections already in operation.

4. Applications of the STAP

To enable accurate and stable control, differential pressure across modulating control valves should not vary too much. This can be obtained with a self-acting differential pressure control valve, STAP, in a variable flow distribution system.

The STAP from TA is a self-acting proportional control valve. It can stabilise the supply differential pressure for a control valve, a branch with several terminal units or a riser with several branches.

4.1 HOW IT WORKS



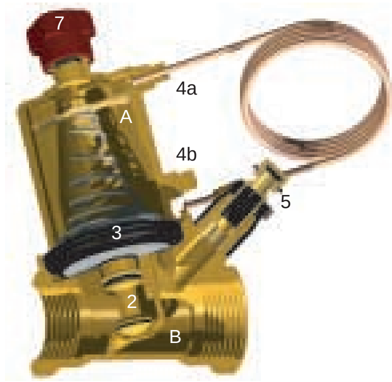


Fig 4.01. STAP stabilises secondary differential pressure ΔpL.

The design of the STAP is based on a spring-membrane combination. The spring pulls the balanced double plug (2) to open the valve. The differential pressure AB is applied on the membrane (3), creating a force against the spring. Pressure A is communicated to the STAP by means of a capillary connected to the drain (6) of the measuring valve STAD/M (STAD or STAM). Pressure B is communicated internally to the other side of the membrane for small sizes and externally for sizes above DN50.

The measuring valve may be omitted or replaced by just a test point in the supply pipe (not recommended if the water flow is not measurable with another feature).

When the force created by the differential pressure AB on the membrane is higher than the spring force, the valve starts to shut proportionally until it finds a new equilibrium position. This creates a supplementary pressure drop in the STAP that limits the increase in the differential pressure ΔpL (the differential pressure across the secondary circuit).

The design force of the spring is modified with an Allen key introduced through the centre of the handwheel (7). This allows the adjustment of the differential pressure (ΔpL) to the required value. The handwheel (7) can also be used to shut the STAP to isolate the circuit when necessary.

The water flow is measured by means of the STAD/M. The ΔpL is measurable between (5) and (4b), (when 4b is equipped with an optional test point) or between (5) and (8).

As STAP is a proportional controller, the ΔpL is not kept absolutely constant. It varies according to the proportional band of the STAP. Figure 4.02a represents the evolution of the ΔpL with the Kv value of the STAP (the degree of opening).

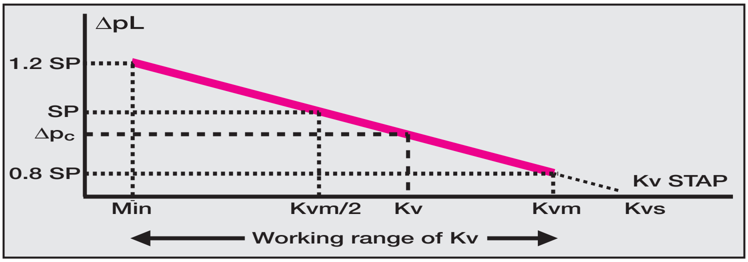


Fig 4.02a. Secondary differential pressure depends on the degree of opening of the STAP.

The Kv of the STAP varies from 0 to a maximum value Kvs. However, the working range is situated between a Kvmin and a Kvm value for which the secondary differential pressure takes stable values at ±20% around the set point (SP). To achieve stable function, a proportional band of 40% to 50% is suitable.

Let us suppose that the design primary differential pressure H is 120 kPa and that the required secondary differential pressure ΔpL is 30 kPa. If H increases from 120 to 220 kPa, the ΔpL will increase from 30 to 33 kPa (+ 10%). The disturbance of 100 kPa has been reduced to 3 kPa on the secondary side. Without the differential pressure controller, the circuit would experience an increase of differential pressure of 100 kPa. In this case, the STAP has reduced the effect of the disturbance by a factor of 33.

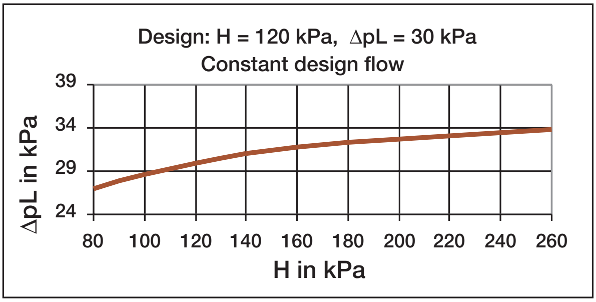


Fig 4.02b. Influence of disturbances on the secondary ΔpL.

To find the correct set point of the STAP, simply measure the water flow with the measuring valve (at design condition) and adjust the set point of the STAP until you arrive at design flow. That's all. There is no need to calculate anything. If this setting is made when the primary H is higher than its design value (Fig 4.02b), the set point obtained is a little lower than normally necessary. The difference is generally negligible but can be corrected when flows are measured for the balancing report.

The set point also corresponds to a certain number of turns of the setting. The relation between the set point and the number of turns is given in technical specifications. This information is useful for faster presetting of the STAP at its required set point, when a measuring valve has not been installed, or when the present differential pressure H is lower than H_{min}. H_{min} is the minimum value of H. For $H<H_{min}$ the design flow cannot be obtained.

Sizing a STAP

A STAP is sized so that its design Kv is close to but lower than $0.8 \times$ Kvm.

For example, let us consider a circuit requiring a secondary differential pressure ΔpL of 30 kPa while primary differential pressure equal to 120 kPa. For the design flow of 2000 l/h, the design pressure drop in the measuring valve is 4 kPa, for instance. The pressure drop to be created in the STAP = 120 – 4 – 30 = 86 kPa. In these conditions, the Kv of the STAP is equal to $0.01 \times 2000/\sqrt{86} = 2.16$.

A STAP with, for instance, a Kvm equal to 3.1 will be suitable for this application.

Sizing a STAD/M

Measuring valves STAD and STAM are chosen to create a pressure drop of at least 3 kPa, fully open and at design flow. This differential pressure of 3 kPa is required to obtain good measurement of water flow.

4.2 Air conditioning

4.2.1 ONE STAP ON EACH RISER

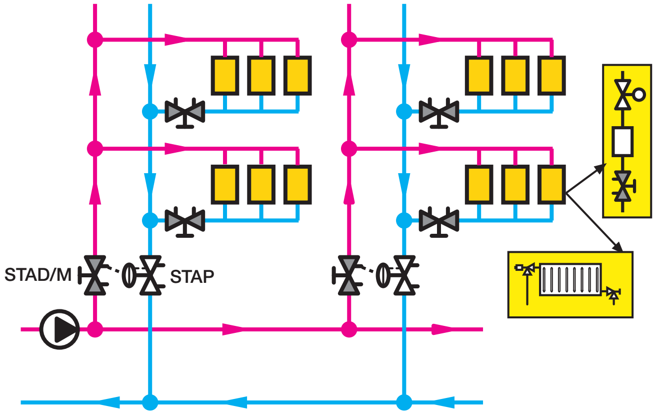


Fig 4.03. A Δp controller STAP stabilises the differential pressure on each riser.

In large systems, the pump head may be too high or vary too much for the terminal control valves. Then, the differential pressure may be stabilised at the bottom of each riser, at a suitable value, with a STAP differential pressure control valve.

Balancing procedure figure 4.03

For the balancing procedure, each riser is a module that can be considered independent from the others. Before starting the balancing of one riser, its STAP should be put out of function and be fully open to ensure obtaining the required water flows during the balancing procedure. An easy way to do this is to shut the drain on the STAD/M (STAM or STAD) in the supply and to purge the membrane chamber using the small venting screw on the bonnet.

1- The terminals are balanced against themselves on each branch before balancing the branches against themselves with the Compensated Method or the TA Balance Method (See handbook 2: "Balancing of distribution system"). STAD/M serves as partner valve.
2- When a riser is balanced, the set point of its STAP is adjusted to obtain the design flow that can be measured with the STAD/M valve situated at the bottom of this riser. Balancing the risers between themselves is not necessary as this is obtained naturally.

Note 1: In heating, and if all control valves of one riser shut, the differential control valve STAP will also shut. The static pressure in the return piping of this riser then decreases as the water cools down in a closed area. Differential pressure across the control valves increases. As a consequence, the control valve that reopens first will be temporarily very noisy. A minimum flow created by a relief valve BPV avoids such a problem.

If the index riser requires a relatively high primary differential pressure, the pump head has to match this need although other risers probably do not need such a high pump head. This increases pumping costs for the whole plant. To reduce these costs, a separate secondary pump for the index riser may be installed.

A bypass EF (Fig 4.04a) avoids any interactivity between the primary and secondary pumps, but it creates a primary constant flow. If variable flow is preferred, another solution is to stabilise the differential pressure upstream of the secondary pump, between A and B, with a differential pressure controller (Fig 4.04b). The necessary pump head of the secondary pump is reduced by the value ΔpAB obtained with the Δp controller.

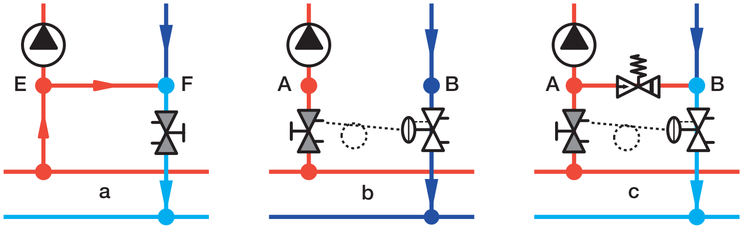


Fig 4.04. How to solve interactivity between primary and secondary pump.

If required (see section 5.2), the minimum necessary flow for the primary pump can be obtained by a proportional relief valve BPV situated between A and B as in figure 4.04c. When the flow is reduced, the differential pressure between A and B increases according to the proportional band of the STAP. The set point of the BPV is calculated to obtain the required minimum flow when required. Another possibility is to isolate the secondary circuit and adjust the set point of the BPV to obtain the minimum flow measurable with the measuring valve.

4.2.2 ONE STAP ON EACH BRANCH

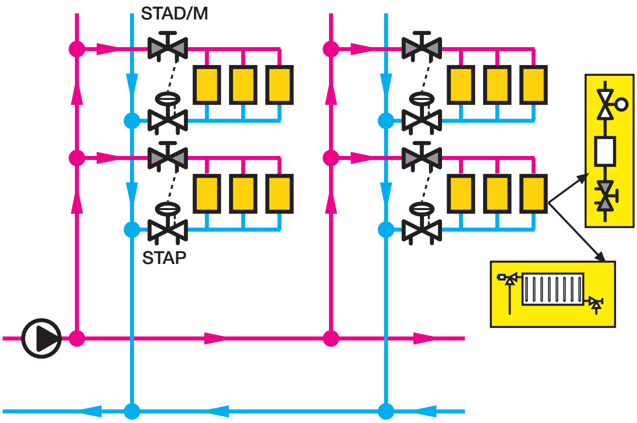


Fig 4.05. A Δp controller STAP stabilises the differential pressure on each branch.

If the differential pressure is stabilised on each branch, the terminals are supplied with a suitable differential pressure. Each branch is balanced independently of the others.

This solution is technically better than one STAP for each riser, because the suitable differential pressure may vary from branch to branch. In addition, changes in differential pressures due to variable pressure drops in pipes in the risers are compensated for automatically.

Balancing procedure figure 4.05

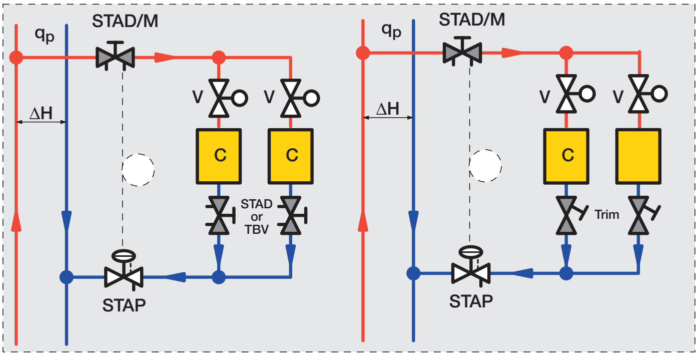
For the balancing procedure each branch is a module that can be considered independently. Before starting the balancing of one branch, its STAP should be put out of function and be fully open to ensure obtaining the required water flows during the balancing procedure. An easy way to do this is to shut the drain on the STAD/M (STAM or STAD) in the supply and to purge the membrane chamber.

1- The terminals are balanced against themselves on each branch with the Compensated Method or the TA Balance Method (See handbook 2: "Balancing of distribution system"). When using the Compensated method, the STAD/M serves as partner valve.
2- When a branch is balanced, the set point of its STAP is adjusted to obtain the design flow that can be measured with the STAD/M valve. Balancing the branches and the risers between themselves is not necessary.

Regarding the minimum flow, the note in section 4.2.1 for risers may be extended to branches.

Examples: In figure 4.06a, each terminal C is fitted with a balancing valve (STAD) or terminal valve (TBV). This is the general case examined in figure 4.05.

In figure 4.06b, each terminal is fitted with a regulating valve (Trim valve or STK). Since regulating valves are not provided with test points, they do not allow flow measurement in the terminal units and the presettings of the regulating valves have to be calculated.



a- Balancing valve on each terminal b- Regulating valve on each terminal

Fig 4.06. One STAP stabilises the differential pressure across a set of terminal units.

Balancing procedure figure 4.06a

1- Open the STAP fully.
   The control valves V are fully open.
2- Balance the terminals of the branch according to the TA Balance method, which does not depend on the differential pressure ΔH available. STAD/M serve as partner valves.
3- Adjust the set point of the STAP to obtain design total flow q_p through the STAD/M.

Balancing procedure figure 4.06b

In the following balancing procedure, we consider that pressure drops in the distribution piping, downstream of the STAP are negligible.

1- For each circuit, the necessary differential pressure (Δp circuit) is the sum of the pressure drops at design flow: Δp circuit = Δp control valve + Δp terminal unit + Δp accessories + Δp regulating valve fully open. Identify the circuit that requires the highest differential pressure (Δp circuit max).
2- For each circuit, calculate the pressure drop to be taken up in the regulating valve: Δp regulating valve = Δp circuit max – Δp control valve – Δp terminal unit – Δp accessories. Adjust each regulating valve to create this pressure drop at design flow. Use a TA nomogram to find the correct setting, or use the TA Select computer program.
3- Adjust the set point of the STAP to obtain the total design flow q_p in the STAD/M.

Example with small units in a branch

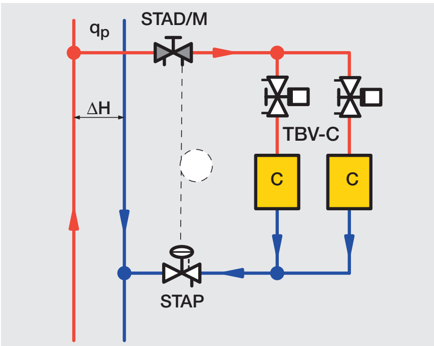


Fig 4.06c. Terminal units are controlled by automatic TBV-C balancing valves.

Figure 4.06c is a typical example of a branch with several units controlled by TBV-C valves. The TBV-C combines five functions in one valve:

- Control
- Stepless presetting from 0 to Kvs, grading from 0 to 10
- Differential pressure measurement
- Flow measurement
- Shut-off

The branch is protected by a controller that keeps the differential pressure stable across circuits. This ensures accurate and stable modulating control. Since the differential pressure is kept at the required level, the risk of noise from control valves is also limited.

Balancing procedure figure 4.06c

Same as for figure 4.06a.

If the examples above are extended to the entire plant, balancing valves in branches and risers are not necessary except for troubleshooting and shut-off.

4.2.3 ONE STAP ON EACH CONTROL VALVE

Depending on the design of the plant, the differential pressure available for a circuit can vary dramatically with the load. To obtain and maintain the correct control valve characteristics, and ensure accurate and stable control, the differential pressure across the control valves can be stabilised with a differential pressure controller as in figure 4.07.

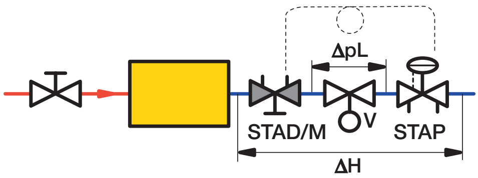


Fig 4.07. A Δp controller stabilises differential pressure across the control valve.

Notes

1- Flow is measured with a measuring valve STAD/M (STAD or STAM), which is an essential tool for troubleshooting.
2- When no measuring unit is required (not recommended), the measuring valve can be replaced by a pressure test point. Then the setting of the STAP is calculated based on the Kvs of the control valve.

In the second case, the given values are: the design flow q and the Kvs of the control valve, which is normally known to an accuracy of ±15%. The theoretical differential pressure that the STAP should create can be calculated using the following formula:

$$\Delta p = \left(0.01 \times \frac{q}{Kvs}\right)^2 \quad (kPa - l/h)$$

$$\Delta p = \left(36 \times \frac{q}{Kvs}\right)^2 \quad (kPa - l/s)$$

The control valve V is never oversized as the design flow is always obtained for the valve fully open. The control valve authority is and remains above 0.7.

As the secondary ΔpL is practically constant, all additional primary differential pressure is taken in the STAP.

Balancing procedure fig 4.07

1- Open the control valve V fully.
2- Preset the STAD/M to obtain at least 3 kPa for design flow.
3- Adjust the set point of the differential pressure controller STAP to obtain design flow.

As the flows are now correct in each terminal, no other balancing procedure is required. If all control valves are fitted with a STAP, then balancing valves in branches and risers are not needed (Figure 3.17) except for diagnostic purposes.

Sizing of the control valve
Sizing the control valve V is not difficult in this case. However, adopting a pressure drop of at least 20 kPa in the control valve is recommended. For a pump head of 250 kPa and without STAP, the design pressure drop in the control valve must be at least equal to $0.25 \times 250 = 63$ kPa. With a STAP, this value may be reduced to 20 kPa. If the design pressure drop in the STAP equals 10 kPa, the pump head may be reduced by 33 kPa, decreasing the pumping costs by at least 13%.

Example with a control valve in injection
Some distribution systems work with a constant flow and a variable supply water temperature. For instance, a constant water flow is required for preheating coils to ensure protection against freezing. For better temperature control, a constant flow in a unit ensures turbulent flow and thus a constant exchange coefficient. In these cases, a three-way mixing valve is normally used to obtain a variable supply water temperature.

When the distribution system is active (primary pump installed), a three-way mixing valve is not permitted as the flow can reverse in its bypass due to the primary differential pressure. When the flow reverses in the bypass of the three-way valve, the mixing function is destroyed. In this case the best solution is to install a two-way control valve in injection as in figure 4.08a.

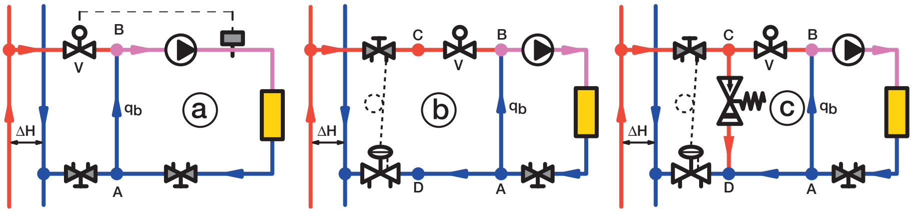


Fig 4.08. A two-way control valve mounted in injection.

If there are great variations in the primary differential pressure ΔH, the authority of the two-way control valve will decrease dramatically and compromise the stability of the control loop. In this case, the best solution is to stabilise the differential pressure ΔpCD across the control valve with a differential pressure controller (Fig 4.08b). If a minimum flow is necessary to protect the primary pump, this can be created by fitting a proportional relief valve (such as the BPV from TA) between C and D. This solution is better than using a manual balancing valve since the minimum flow is generated only when necessary. This reduces the primary flow and therefore the pumping costs.

Some designers specify a non-return valve in the pipe AB to avoid flow from B to A. There are two main reasons for this:

1. For a preheating coil exposed to a low air temperature, the non-return valve allows the primary pump to inject hot water in the coil if the secondary pump fails. This ensures protection against freezing.
2. In district heating, if the two-way control valve is oversized, or if the secondary flow is variable, the water flow in the bypass AB may reverse. This will reheat the return. The non-return valve prevents this reverse flow.

4.3 Radiator heating

In a modern heating plant, thermostatic valves are preset to obtain the required water flows at design condition. These presettings are only valid if the differential pressures on which the presettings are based really are obtained for the radiator valves.

When the plant is working at small average loads, the pressure drops in the piping decrease. The differential pressures across thermostatic valves may then increase substantially. If this differential pressure exceeds 20-30 kPa, there is a risk of noise, especially if air remains in the water. It is therefore also essential to stabilise the differential pressure across thermostatic valves.

This section presents solutions to a few common problems in radiator systems:

- How to obtain the required differential pressure across the thermostatic valves.
- How to make sure this differential pressure is stable at all loads.
- What to do when the thermostatic valves are not presettable.

4.3.1 PRESETTABLE RADIATOR VALVES

To make it simple for the installer, thermostatic valves are generally preset under the assumption that the available differential pressure ΔH_o = 10 kPa. This value is a compromise between two requirements:

- Not too high to maintain a sufficiently large opening of the valve to avoid clogging and noise.
- Not too low so that the relative influence of pressure drops in the piping is low.

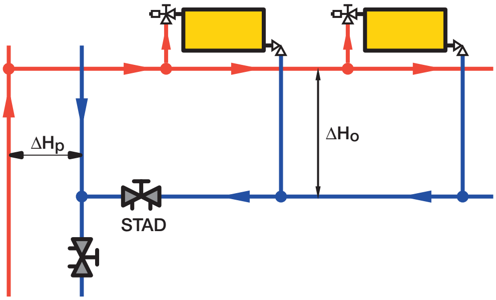


Fig 4.09. Each radiator valve is preset as if it was subjected to the same differential pressure of 10 kPa.

During the balancing procedure for the entire plant, the balancing valve for the branch is set to obtain the correct total branch flow. This justifies the presetting of the thermostatic valves, and the 10 kPa expected will be obtained at the centre of the branch when balancing is completed.

In a variable flow distribution system, differential pressures may increase considerably when the plant works at small loads. Thermostatic valves may become noisy especially if air remains in water. Then, it is advisable to install differential pressure control valves (STAP) as in figure 4.10.

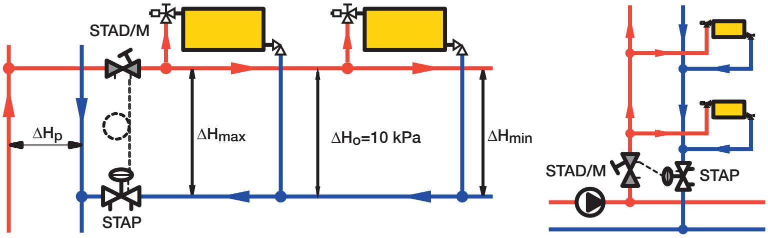


Fig 4.10. A STAP stabilises the differential pressure at the circuit inlet.

A STAP stabilises the differential pressure for each branch or small riser. The flow q_s is measured in the STAD/M (STAD balancing valve or STAM measurement valve).

Balancing procedure figure 4.10

1- Open all thermostatic valves fully, for instance, by removing the thermostatic heads.
2- Preset the thermostatic valves for a constant differential pressure of 10 kPa less the pressure drop in the return valve. Use the result DH to determine the Kv to preset: $Kv = 0.01 \times q/\sqrt{DH}$, with q in l/h and DH in kPa.
3- Adjust the set point of the STAP to obtain total design flow q_s measured in the STAD/M. The expected differential pressure of 10 kPa is then applied across the middle of the circuit.

Note: In reality, the first radiator will be in overflow and the last in underflow. This is acceptable if the pipe length between the STAP and the last radiator does not exceed L = 2200/R (L in m), when R is the average pressure drop in the pipes (in Pa/m). This formula is based on a maximum flow deviation of 10%. For R = 150 Pa/m, L_{max} = 15 metres.

Location of on-off zone valve and energy counter

In some countries, a differential pressure control valve is provided for each apartment. The supply water temperature is adjusted with a central controller according to the outdoor conditions. A room thermostat is often placed in a reference room where the radiator valves are not automatic. The room thermostat controls an on-off valve V as shown in figure 4.11.

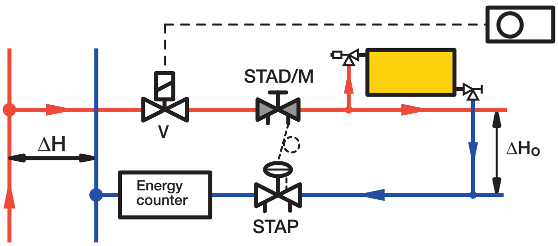


Fig 4.11. One STAP controls the Δp for each apartment.

The on-off control valve and any energy counter are preferably situated in the part of the circuit where the differential pressure is not controlled. This is to prevent their variable pressure from affecting the differential pressure across the radiators. However, the on-off valve and the energy counter may be placed downstream the STAP if the required set point of the STAP does not exceed the maximum value for which thermostatic valves may become noisy.

4.3.2 NON-PRESETTABLE RADIATOR VALVES

In some existing plants, radiator valves are not presettable. Differential pressure controllers can limit the differential pressure for each circuit. But without flow restrictions in the radiator valves, the flow can be several times higher in one or more radiators, and far too low in others, despite differential pressure control.

The best way to solve this problem is, of course, to install presettable radiator valves and balance according to section 4.3.1.

Another solution is to use the balancing valve STAD as a measuring valve and connect the signal pipe from the differential pressure control valve to the test point of the balancing valve upstream of the flow throttling, figure 4.12. The balancing valve is then included in the controlled circuit. Compare figures 4.10 and 4.12 to see how the signal pipe is connected and where the balancing valve is mounted. The flow q_S is measured in the STAD.

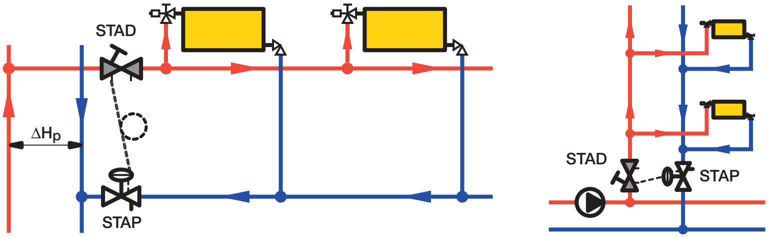


Fig 4.12. The pressure drop in the balancing valve is included in the total Δp controlled by the STAP.

The set point of the STAP is selected to 20 kPa. The balancing valve is adjusted to obtain the total design flow when all thermostatic valves are fully open.

During start-up, all thermostatic valves are fully open and the total flow is correct as it was adjusted at design value with the STAD. When the thermostatic valves close, the available differential pressure is automatically limited to the set point of the STAP (20 kPa), plus the value of its proportional band.

This combination maintains the total flow and the Δp limitation at acceptable values. With this method, the correct distribution of the total flow between the radiators is only achieved if all radiators are identical and close to each other. If this is not the case, it will however significantly improve the performance of a plant with non-presettable radiator valves.

4.4 STAP as a flow controller

In some applications, automatic flow controllers are useful. For instance, to maintain the primary flow of a three-way diverting valve constant, or to serve several cooling circuits for industrial purposes. For these applications specific flow controllers may be used.

A STAP can also solve the problem of flow measurement and adjustment This solution is shown in figure 4.13.

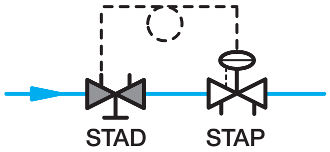


Fig 4.13. STAP as a flow controller.

Selection of the set point should be as close as possible to the minimum value of its range, for instance 10 kPa.

The water flow is measured with the measuring valve STAD whose setting is chosen to obtain the required design flow. In this example, the pressure drop in the STAD corresponds to the set point of the STAP, that means 10 kPa. If the water flow has a tendency to increase, it will increase the pressure drop in the STAD. This differential pressure is communicated to the STAP, which will close a little to restore flow at design value. According to the proportional band of the STAP, the water flow is not maintained absolutely constant. For example, if the differential pressure across the set STAD-STAF increases from 100 to 200 kPa, the flow increases by 7%.

5. Appendices

5.1- Some questions and answers

5.1.1 WHY HYDRONIC BALANCING?

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, particularly after big load changes.
- Room temperatures oscillate, particularly at low and medium loads, even though the terminals have sophisticated controllers.
- Although the rated capacity of the production units may be sufficient, design capacity 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 necessary. The only way to get design flows is to balance the plant. This has to be done 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. Fluctuations reduce production efficiency, shorten the life of the production units and make effective control more difficult.
2. The distribution system must be balanced to make sure all terminals can receive at least design flow, regardless of the total 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.

When the plant is balanced, a central controller or optimiser can be used as all rooms react the same way. Moreover, when the average room temperature deviates from design value, due to the absence of balancing, costly discomfort is generally the result.

During the balancing procedure, most hydronic problems are highlighted and can be corrected before the plant is put in operation. Hydronic balancing devices, able to measure differential pressure and water flows, serve as tools for troubleshooting during the whole life of the plant.

The use of differential pressure control valves for balancing has the added advantages of locally stabilising differential pressure and preventing interactivity problems. These two functional advantages translate into three clear benefits: accurate and stable modulating control, minimal noise from control valves and simplified balancing and commissioning.

5.1.2 WHAT ARE THE COSTS OF DISCOMFORT?

During cold weather, it is too hot close to the boiler and too cold on the top floors. People increase the supply temperature in the building. People on the top floors stop complaining and people close to the boiler open the windows. During hot weather the same applies. It is just that it is too cold close to the pumps, and too hot in other parts.

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 5 to 8%. One degree below 23°C increases cooling costs between 10 and 16%.

5.1.3 IS A WELL-DESIGNED PLANT AUTOMATICALLY BALANCED?

Some people seem to think that it is sufficient to indicate design flows on the drawings to obtain them in the pipes. But to obtain the required flows, they must be measured and adjusted. This is why hydronic balancing is so essential.

Is it possible to obtain correct flow distribution by sizing the plant carefully? In theory, the answer is yes. But in practice, it's just a dream.

Production units, pipes, pumps and terminals are designed to cover maximum needs (unless the plant is calculated with a diversity factor). If a link in the chain is not properly sized, the others will not perform optimally. As a result, the desired indoor climate will not be obtained and comfort will be compromised.

One might think that designing the plant with some costly safety factors would prevent most problems. However, even if some problems are solved that way, others are created, particularly on the control side. Some oversizing cannot be avoided, because devices must be selected from existing commercial ranges. These generally do not suit calculations made. Moreover, at design stage, the characteristics of some elements are not known since the contractor will select them at a later stage. It is then necessary to make some corrections also taking into account the real installation, which frequently differs somewhat from the initial design.

5.1.4 IS A VARIABLE SPEED PUMP SUFFICIENT TO OBTAIN THE CORRECT DESIGN FLOWS?

Let us consider two identical terminal units with a design flow of 1 l/s. One receives 2 l/s while the second receives only 0.5 l/s. Total flow corresponds to 2.5 l/s instead of 2 l/s. The pump head is reduced to obtain the correct total design flow of 2 l/s. When this is done, water flows in the terminals are 1.6 l/s and 0.4 l/s respectively. The plant remains unbalanced, as the terminal units are not working at design flow.

This simple example shows that using a variable speed pump alone will not solve hydronic balancing problems since all the flows will change proportionally when pump head is modified. Attempting to avoid overflows this way will simply make underflows more significant.

5.1.5 IS THE PLANT AUTOMATICALLY BALANCED IF THE TWO-WAY CONTROL VALVES ARE WELL SIZED?

At first glance, there appears to be no reason for balancing a system with two-way control valves on the terminals, since the control valves are designed to modulate flow to the required level. Hydronic balancing should therefore be automatic. However, even after careful calculations, you find that control valves with exactly the required Kvs are not available on the market. Consequently, most control valves are oversized.

Full opening of the control valves cannot be avoided in many situations, such as during start-up, when big disturbances occur, when the supply temperature is too low in heating/too high in cooling, when some thermostats are set at minimum or maximum value or when some coils are undersized. In these cases, and when balancing devices are not in place, there will be overflows in some circuits. This will create underflow in other circuits.

The plant is designed to provide a calculated maximum power

An HVAC system is designed for a specific maximum load. If the plant cannot deliver its full capacity in all circuits because it is not balanced, the total investment in the plant will never be translated into benefits. If this maximum capacity is never required, the plant is not well designed.

Control valves are fully open when maximum capacity is required. They are generally oversized and they cannot contribute to balancing. Hydronic balancing is thus essential and usually represents less than two per cent of the total investment in the HVAC system.

Hydronic balancing, carried out at design condition, guarantees that each terminal can receive its required flow in all conditions. At partial loads, when some control valves close, the available differential pressures on the circuits can only increase. If underflows are avoided at design condition, they will not occur in other conditions.

In conclusion, hydronic balancing enables the attainment of required flows. The maximum power installed can be transmitted, justifying the total investment in the plant.

Morning start-up

Each morning, after a night setback, full capacity is required to recover comfort levels as soon as possible. A well-balanced plant does this quickly. If a cooling plant starts up 30 minutes sooner than normally required, this increases energy consumption by 6% per day. This often represents more than all distribution pumping costs. Imagine the extra cost if the plant has to start 2 hours sooner!

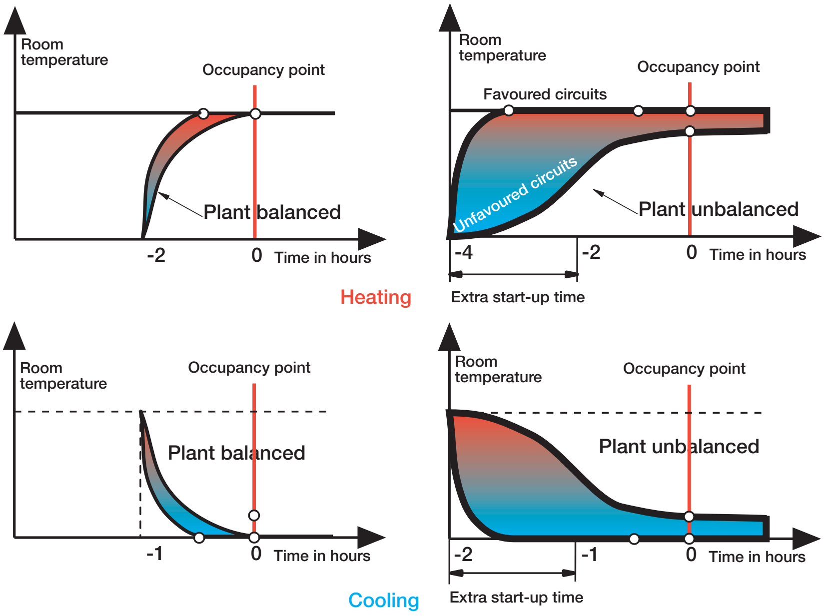


Fig 5.01. An unbalanced heating/cooling plant has to start up earlier, increasing energy consumption.

During morning start-up after each night setback most control valves are driven fully open in variable flow systems. If the plant is not balanced there will be overflows. This produces unpredictable pressure drops in some of the piping network, starving the terminals in the less favoured sections of the system. The unfavoured circuits will not receive adequate flow until the favoured spaces have reached thermostat set point (if these set points have been reasonably chosen), allowing their control valves to begin to throttle. Moreover, distribution and production water flows are not compatible and design supply water temperature cannot be obtained. This increases the time required for all circuits, including the favoured circuits, to reach their design room temperatures. Start-up is therefore difficult and takes longer than expected. This is costly in terms of energy consumption. A non-uniform start-up makes management by a central controller and any form of optimization practically impossible.

In a distribution system with constant flow, underflows and overflows remain both during and after start-up, making the problem even more difficult.

5.1.6 DO BALANCING DEVICES INCREASE PUMPING COSTS?

If balancing valves are well adjusted, they take away local overpressures due to the non-homogeneity of the plant and ensure design flows at design condition. If balancing valves are fully open, control valves are obliged to shut to compensate. Friction energy cannot be saved that way; it will just be transferred from the balancing valves to the control valves. It is then quite obvious that balancing valves do not create unnecessary pressure drops. On the contrary, balancing valves prevent overflows and reduce pumping costs.

5.2 Minimum flow in a variable flow distribution system

In a variable flow system with modulating control valves, water flow drops to 20% of the design value when the load is close to 50% of design, which is quite a common situation. The pump is not able to work below a certain flow; this minimum depends on its design.

To protect the pump, a relief valve may be installed just after the pump as in figure 5.02. But this is not the best location.

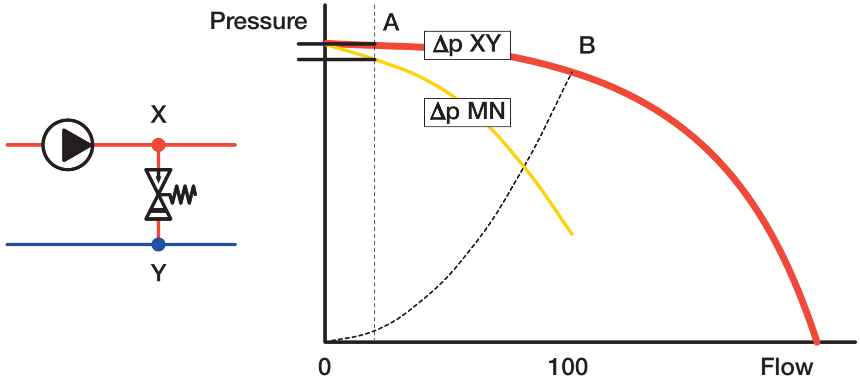


Fig 5.02. The relief valve is situated close to the pump (XY) or on a remote circuit (MN).

In this case, the relief valve has to be set so that it opens when total flow decreases below 20% of design, for instance. The corresponding pressure is situated on point A on the pump characteristic.

If the relief valve is set a little higher than A, it will never open. If it is set a little lower than A, it will open when total flow in the plant is high enough. It is only possible to find a correct set point when the pump curve is very steep. The setting may be corrected after a certain time since old pumps do not have the same characteristics as new ones!

If the relief valve is located far away from the pump, the differential pressure varies quickly with the flow as this differential pressure also depends on the variable pressure drops in pipes and accessories. The set point of the relief valve is consequently easier to adjust (curve Δp MN).

Some designers provide a pressure relief valve (BPV) at the end of each riser (or each branch) to obtain a minimum flow when most of the control valves are shut. Another method is to provide some terminal units with a three-way valve instead of a two-way control valve.

Obtaining this minimum flow has several advantages:

1. Water flow in the pump does not drop below the minimum value.
2. When water flow is too low, heat losses/gains create a higher ΔT and the circuits remaining in function cannot deliver their full capacity if required, as their supply water temperature is too low in heating or too high in cooling. A minimum flow in the circuit reduces this effect.
3. In heating, if all the control valves in one riser shut, the differential pressure control valve STAP will also shut. Static pressure will decrease in this riser as the water cools down in a closed area. The differential pressure across the control valves becomes much higher. As a consequence, the control valve that reopens first will be temporarily very noisy. A relief valve BPV can create a minimum flow to avoid this problem.

5.3 Different ways of controlling a variable speed pump

With a constant speed pump, the pump head increases when the total flow decreases.

With a constant speed pump and a direct return distribution (Fig 5.03a), calculation of the control valve, close to the pump, is based on the design available differential pressure (AB) on the circuit. When the whole plant works at small and average loads, the pump head increases and the pressure drops in pipes decrease. Consequently, the differential pressure available for the circuit increases from (AB) to (A‘B’). This increase does not considerably affect the control valve authority. The situation is quite different for the last terminal which experiences a big change in differential pressure from (EF-design) to (E‘F’), dramatically decreasing the authority of its control valve, with a risk of hunting.

A reverse return distribution (Fig 5.03b) does not solve the problem, as all terminal units will be submitted to big changes in differential pressure.

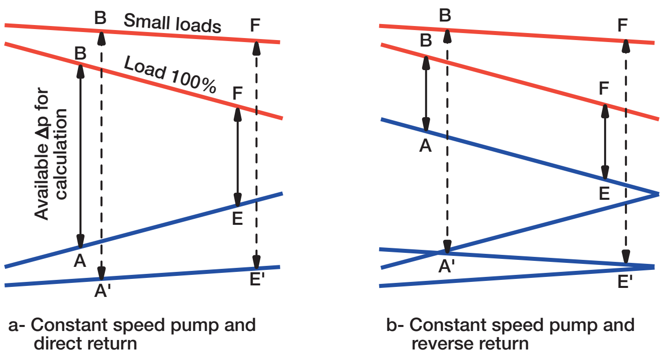


Fig 5.03. Constant speed pump in a direct and a reverse return distribution.

With a variable speed pump, is it possible to decrease the pump head when the total flow decreases?

It is not logical to increase the differential pressure when the control valves are trying to reduce the flow. With a variable speed pump, constant pump head can be obtained. Going further in this direction, the pump head can be reduced when total flow decreases. However, total flow can be reduced to 50% either because all terminals require 50% of design flow or because 50% of the terminals are working at design condition while all the others are off. In the first case the pump head may be reduced. In the second case, the pump head normally needs no change as some units may need design flow. Thus, the way the variable speed pump is controlled is important.

Maintaining constant differential pressure close to the last terminal

Some designers attach an overriding importance to pumping costs, to the extent that the design of the plant appears to be primarily determined by this consideration, without taking into account the effect these choices have on comfort. It is true that pumping costs may be fairly accurately estimated, which is a powerful incentive to take them into account. In a balanced constant flow distribution system, the real pumping costs, in percentage of the production unit's seasonal consumption, are around 2% in heating and 6 to 12% in cooling. These values are reduced in a variable flow distribution system.

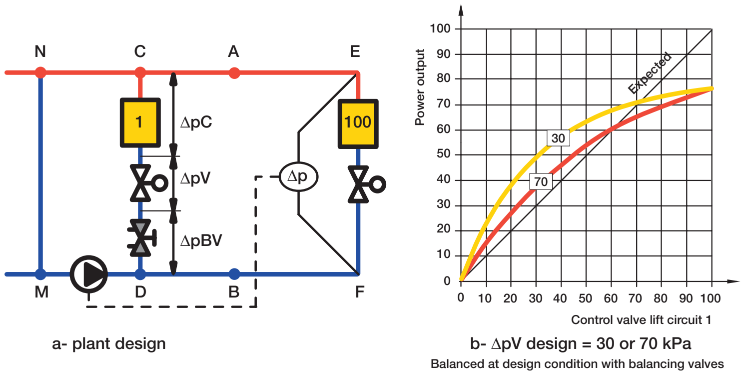


Fig 5.04. Constant Δp is maintained close to the last terminal unit. What happens if all terminals are working at 50% load while terminal 1 has to work at full load?

Figure 5.04a shows what happens in a plant with 100 identical coils. At design condition: ΔpCD = 87 kPa and ΔpEF = 25 kPa. For the terminal EF, the design pressure drop in the control valve is 12.5 kPa (authority 0.5). For the first terminal, the best available choice is 70 kPa (authority 70/87 = 0.8).

If all terminal units work at 50% load, except terminal 1 that needs full flow, the differential pressure ΔpCD drops from 87 to 27 kPa. Design flow cannot be obtained in unit 1 (Fig 5.03b). It drops to:

$$100 \times \sqrt{\frac{27}{87}} = 56\ \%\text{ and the power output to }78\%$$

If this case is considered as exceptional and a reduction of 22% of the power output of circuit 1 is acceptable, then this design is correct.

Imagine now that this situation is not acceptable, which is mostly the case.

To try to solve this problem, the design pressure drop of control valve 1 is selected equal to 12.5 kPa and the balancing valve is cancelled. The circuit can now obtain its design flow with only 27 kPa available. However, during start-up, the control valve is fully open with an available Δp of 87 kPa on the circuit. In this case, water flow in circuit 1 will reach 187% of its design value. The pump runs at maximum speed and cannot maintain the 25 kPa expected close to the last terminal, which will experience a severe underflow. Overflows create underflows in other parts of the plants, generating complaints from tenants.

Moreover, a general overflow will reverse the flow in the bypass line MN, creating a mixing point at M and an increase of the supply water temperature in cooling. This makes each morning start-up quite laborious.

One solution for this circuit is to maintain a constant Δp of 12.5 kPa across the control valve with a local Δp controller. In this case, the flow is always limited to the design value and the control valve authority is and remains close to 0.7 (see figure 4.07).

Controlling Δp in the middle of the plant

In figure 5.04a, differential pressure may also be maintained constant in the middle of the plant (AB instead of EF). Taking the same example as before, the set point would be 56 kPa. When the average load is close to zero, maximum flow obtainable at the first terminal would be 80%, reducing maximum power output by 6%. For the last terminal, the control valve authority will decrease from 0.5 to 0.22. This can be acceptable, but the saving on pumping energy is reduced in comparison with control of the Δp close to the last terminal.

Managing the variable speed pump by using several sensors/controllers

A variable speed pump allows the reduction of the pump head at small loads, decreasing pumping costs. At partial loads, if the Δp sensor is well located, the authority of the control valves can be improved substantially, guaranteeing better room temperature control.

The question is where to install the differential pressure sensor.

Let us take the plant represented in figure 3.08

In this case, the control of the variable speed pump can be optimised by the use of two sensors/controllers; the most demanding one controlling the variable speed pump.
The first sensor/controller set at 113 kPa controls the inlet of circuit J while the second, set at 73 kPa controls the inlet of circuit E.

When the plant works at 50% flow, a pump head of 130 kPa is achieved instead of the 250 kPa obtained with a constant speed pump (Fig. 3.12)

This solution allows a substantial reduction of pumping costs without starving some circuits in certain conditions. The control valve authorities are also improved at small and partial loads.

5.4. Pumping costs compared with the costs of discomfort

Using typical values, for a well-balanced constant flow distribution, these relative pumping costs can be estimated, in % of the seasonal consumption of the production units, using the following formula:

$$C_{pr} = \frac{1.42 \times H}{Sc \times \Delta T_c} \text{ \% when}$$

H: pump head in metres WG **ΔT_c**: ΔT design in K **Sc**: average seasonal load/design load.

In cooling: For $\Delta T_c = 6$, $S_c = 0.8$ and H = 20 m WG, $C_{pr} = 6\%$. If $S_c = 0.4$, $C_{pr} = 12\%$.

In heating: For $\Delta T_c = 20$, $S_c = 0.4$ and H = 10 m WG, $C_{pr} = 1.8\%$.

In a balanced constant flow distribution system, real pumping costs, in percentage of the production unit's seasonal consumption, are around 2% in heating and 6 to 12% in cooling. These values are reduced in a variable flow distribution system.

In an equivalent way, the extra energy cost, due to a constant deviation of the room temperature, is:

In cooling: 1°C too low temperature between 10 and 16%.

In heating: 1°C too high temperature between 6 and 10%.

In most cases 1°C deviation in room temperature costs more than all distribution pumping costs. In conclusion, any action intended to reduce pumping consumption must be taken so that they do not adversely affect the operation of terminal unit control loops.

Increasing the ΔT_c can reduce relative pumping costs. In heating, for instance, some plants are calculated for a ΔT_c = 10 K, while in some countries it is quite common for ΔT_c = 30 K.

Proportional control also allows the reduction of pumping costs. In on-off control, a 50% load is obtained with approximately 50% flow, while with stable proportional control a 50% load with only 20% flow is obtained. (Fig 3.03a).

With a variable speed pump, some say that the potential for saving energy is related to $(\text{flow})^3$. This is too optimistic. Pumping energy depends on the product H × q (pump head × flow). The Δp (= H) of the plant depends on $R \times q^2$ (resistance of the plant × the square of the flow), but R is not constant. It increases to reduce the flow, and finally H is not proportional to q^2.

A better estimation of pumping energy with variable speed pumps is given below:

$$W = \frac{50 \times (2 - a) \times \lambda \times (a + C + \lambda^2 - C\lambda^2) \times \eta_d}{\eta}$$

With W = Pumping costs in % of design

$$C = \frac{\Delta p \text{ design close to the most remote circuit}}{\text{design pump head}}$$

λ = flow ratio $\quad$ η = electrical efficiency × pump efficiency

$\eta_d = \eta$ at design condition

a = 0 when the Δp close to the last terminal is maintained constant

a = 1 when the Δp at the centre of the plant is maintained constant

Example: $\lambda = 0.5$ (50% flow), $C = 0.2$, $\eta = 0.6 \times 0.67 = 0.4$

$\eta_d = 0.84 \times 0.8 = 0.67$. For a = 0, W = 33%. For a = 1, W = 57%.

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