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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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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

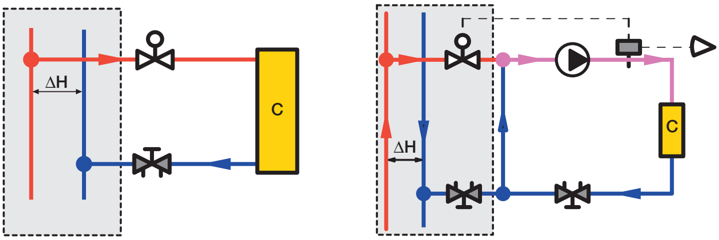

Fig 2.01. Examples of variable flow distribution systems.

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

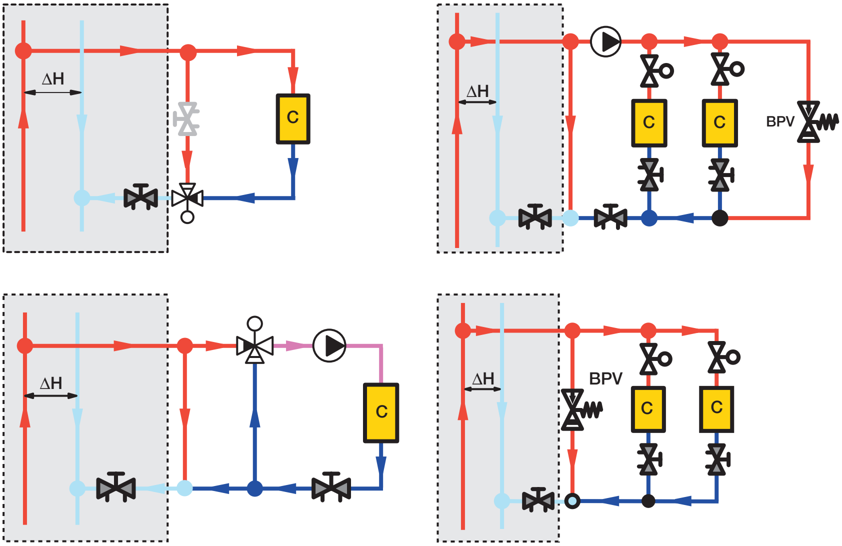

Fig 2.02. Some examples of constant flow distribution.

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.

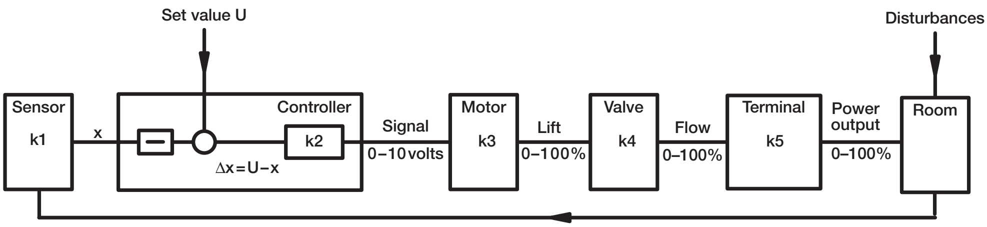

Fig 3.01. The six elements of the control loop interact with each other.

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.

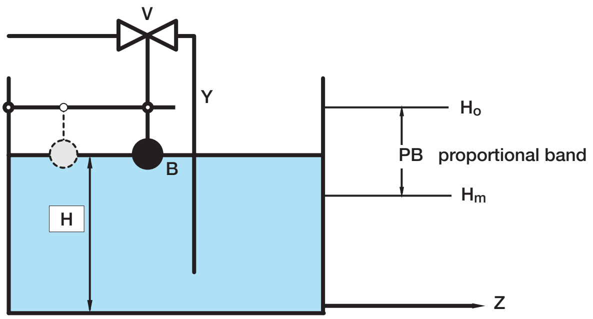

Fig 3.02. Principle of proportional control.

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.

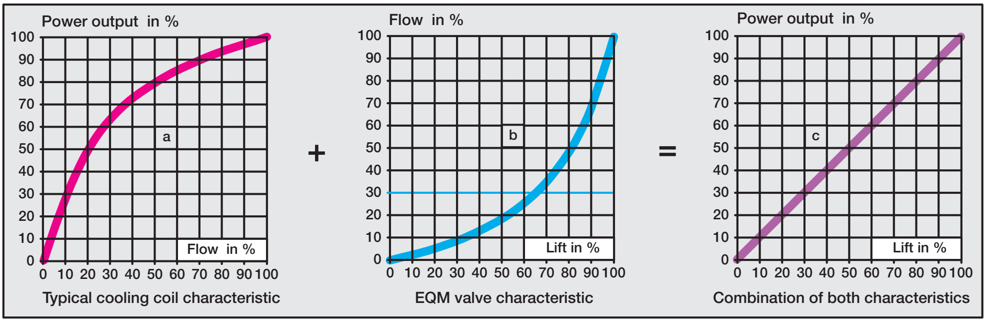

Fig 3.03. Select the control valve characteristic (middle) to mirror this of the terminal (left) to produce a linear relationship power output/lift (right).

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.

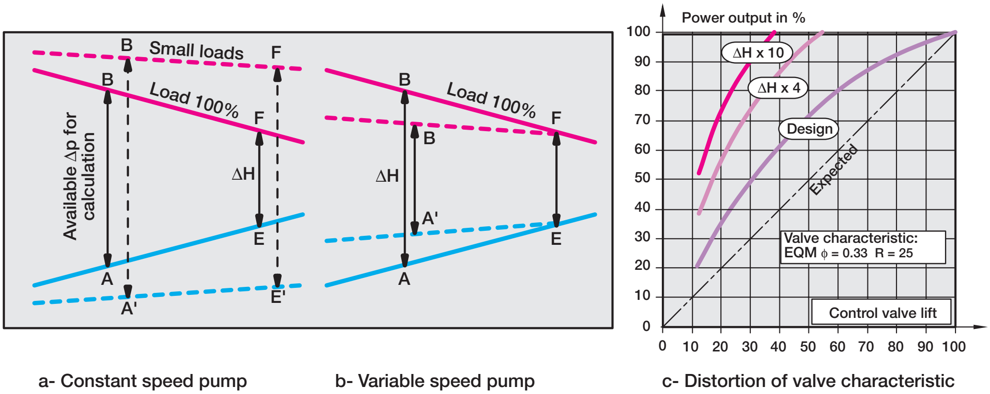

Fig 3.04. At design, control valve authority = 0.25. When the average load of the plant changes, differential pressure ΔH on the circuit also changes and distorts the control valve characteristic.

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!

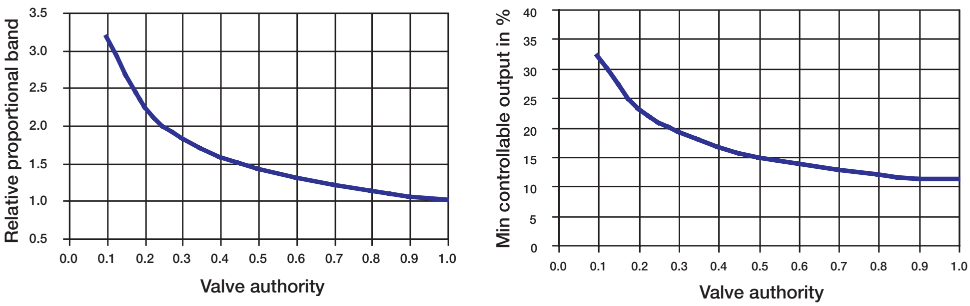

Fig 3.05. To obtain stable control, the required relative proportional band increases when the authority decreases. Below a certain minimum load control becomes unstable.

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).

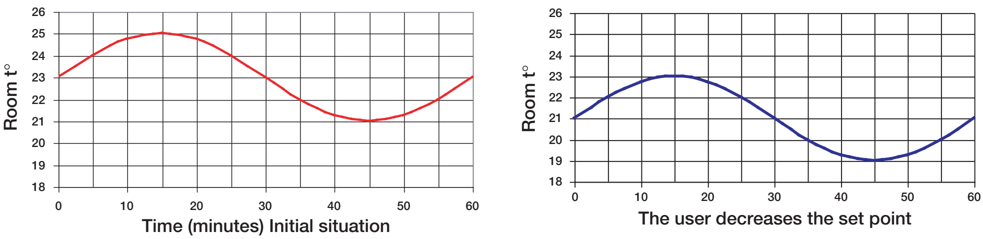

Fig 3.06. To compensate for overheating, the user decreases the set point of the room thermostat.

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).

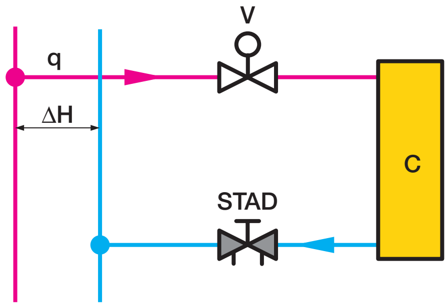

Fig 3.07. A two-way control valve.

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…

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

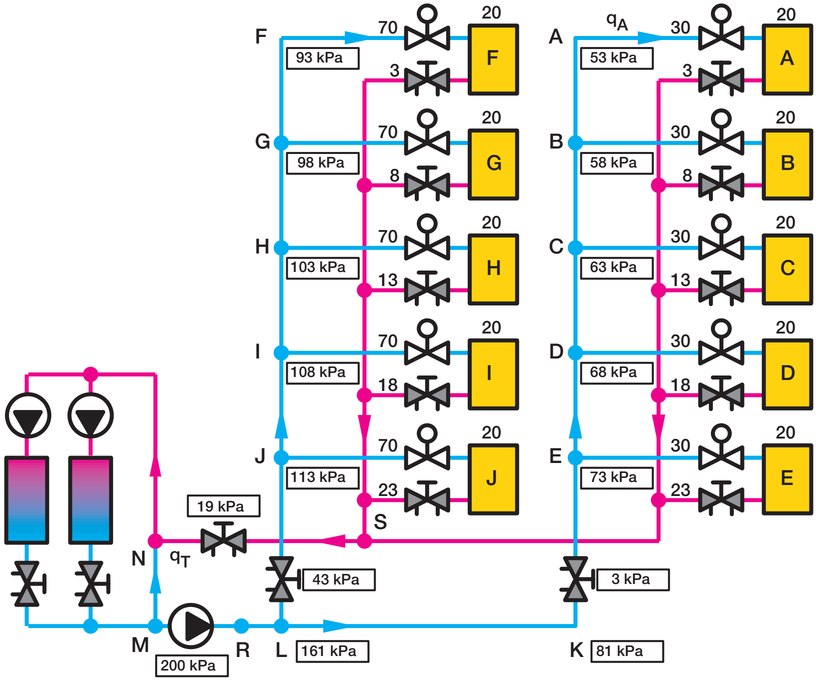

Fig 3.08. Cooling plant working at design condition.

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?

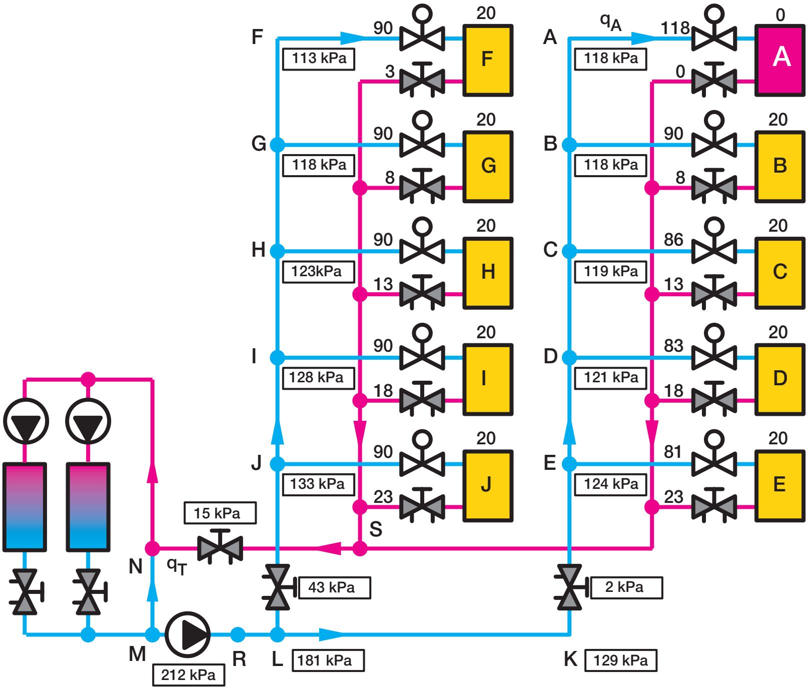

Fig 3.09. The control valve of terminal A closes.

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.

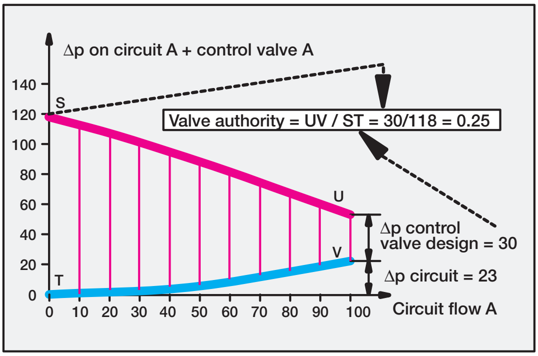

Fig 3.10. When the valve of circuit A closes (Fig 3.09), differential pressure applied on this circuit increases from 53 to 118 kPa while the rest of the plant remains at design flow.

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.

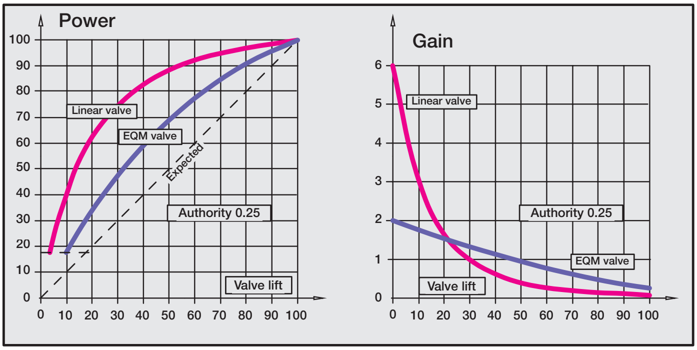

Fig 3.11. Power and gain versus valve lift at design condition for terminal A.

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)

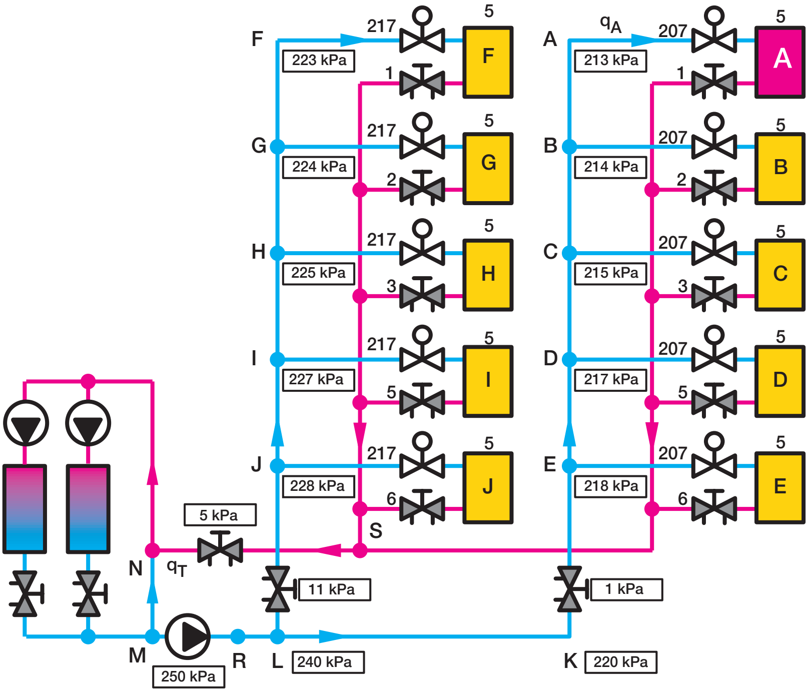

Fig 3.12. Total flow q_T = 50% of design.

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.

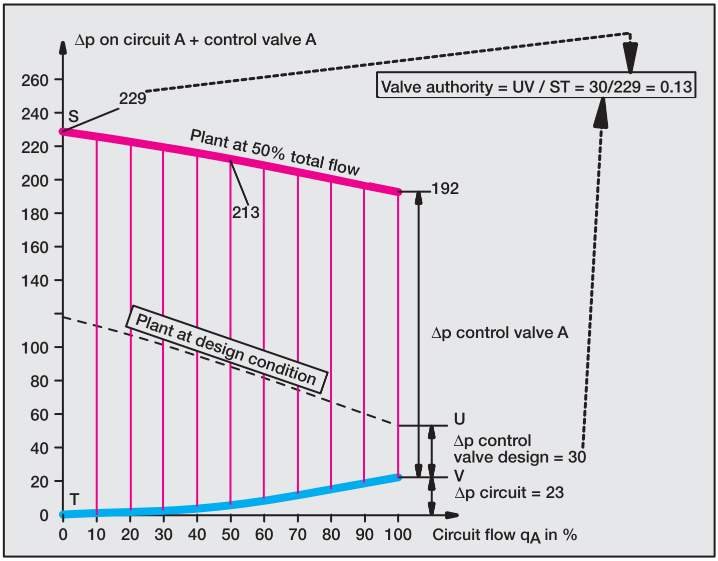

*Fig 3.13. The entire plant works uniformly at 50% of total design flow.
The control valve A opens from 0 to design flow.*

With a linear valve, 78% of maximum power is already obtained for the valve lift at 20% (Fig 3.14A).

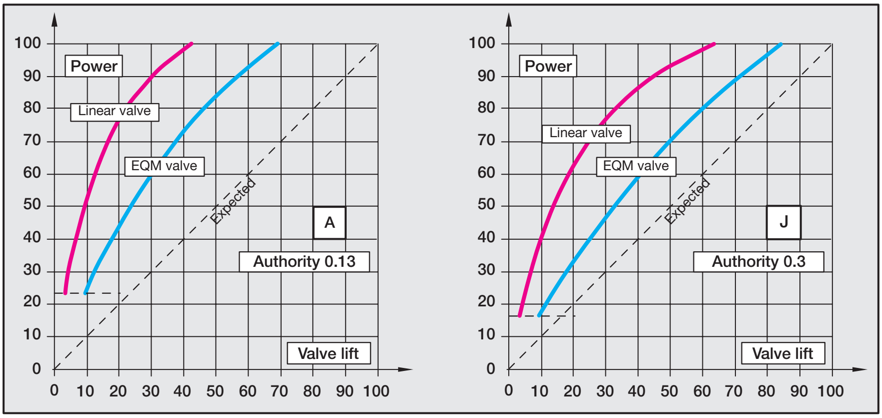

Fig 3.14. Power versus valve lift for terminals A and J – q_T = 50%.

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

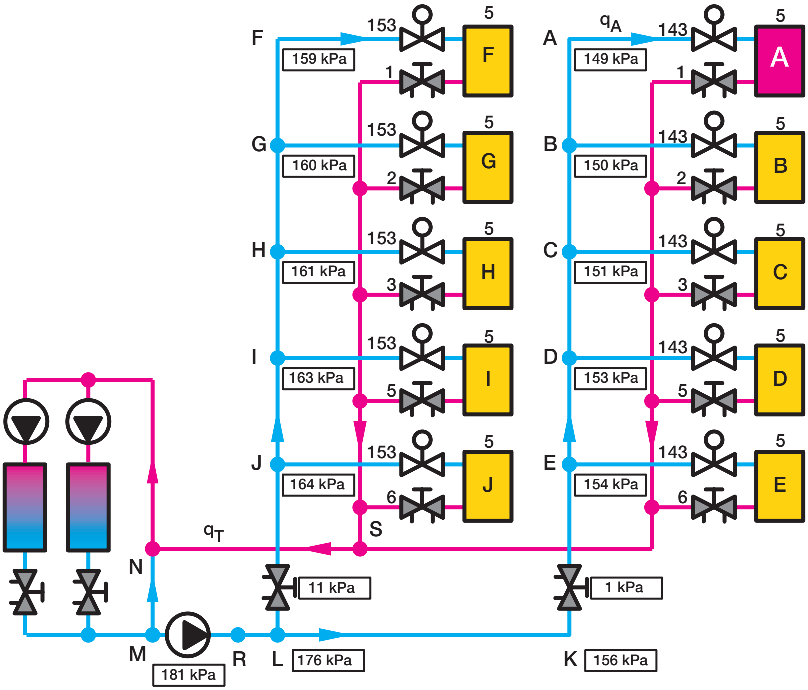

Fig 3.15. The variable speed pump works with a constant pump head and q_T = 50%.

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.

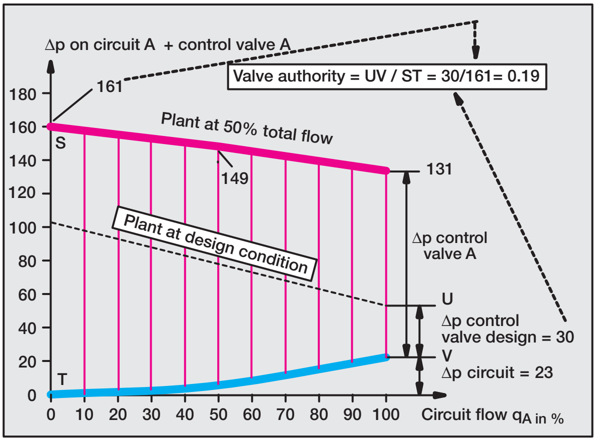

Fig 3.16. Evolution of the Δp across the control valve A when the plant works on average at 50% flow.

Other ways of controlling the variable speed pump are examined in appendix 5.3.

3.1.2.5 The use of local Δp controllers

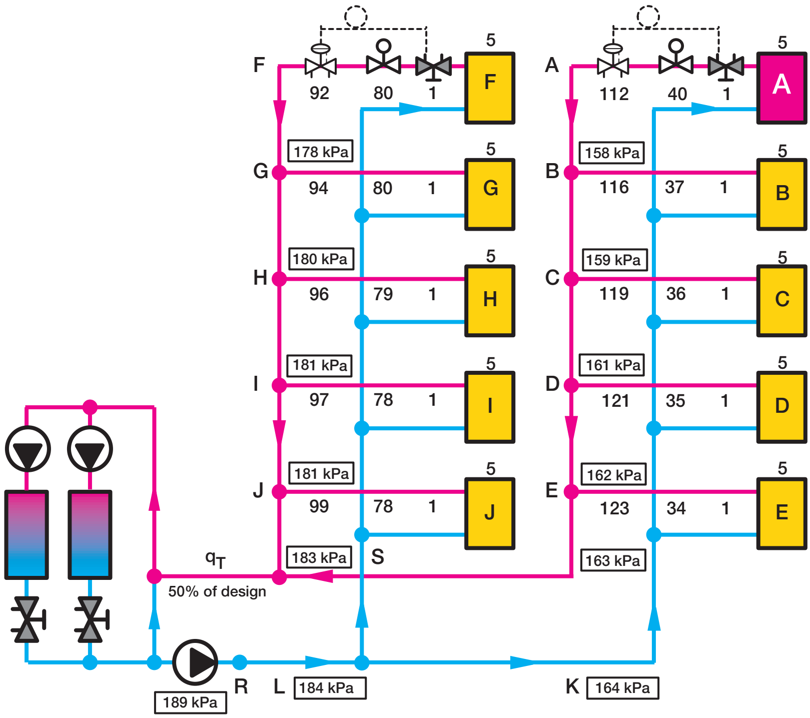

Fig 3.17. A Δp controller stabilises the differential pressure across each control valve. q_T = 50% of design.

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

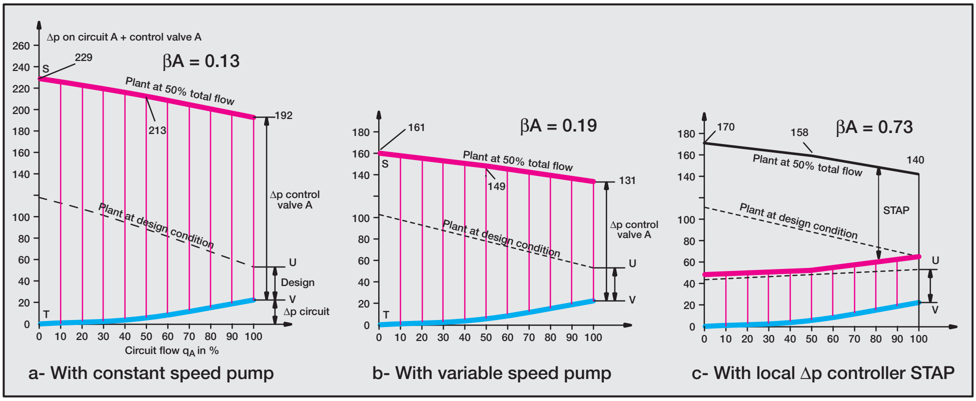

Fig 3.18. Evolution of differential pressure across control valve A, when the plant works with an average water flow of 50% corresponding to an average load of 80%. βA, in the figure, represents the authority of control valve A.

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 substations 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.

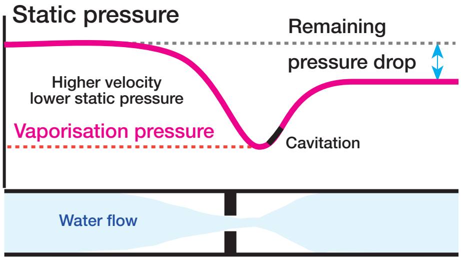

Fig 3.19. When static pressure reaches vaporisation pressure, the liquid starts to boil. When static pressure increases above vaporisation pressure, steam bubbles implode and the valve cavitates.

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.

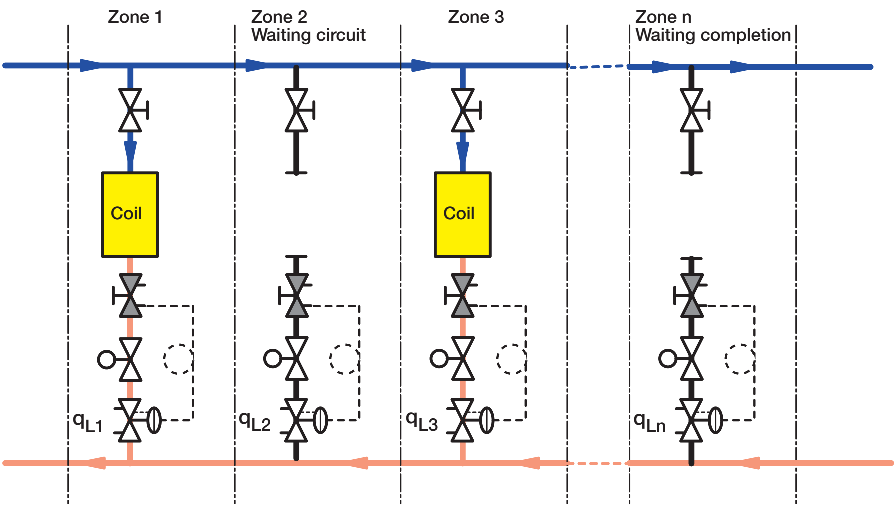

Fig 3.20. When a new circuit is put in service, this does not influence the other circuits.

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.

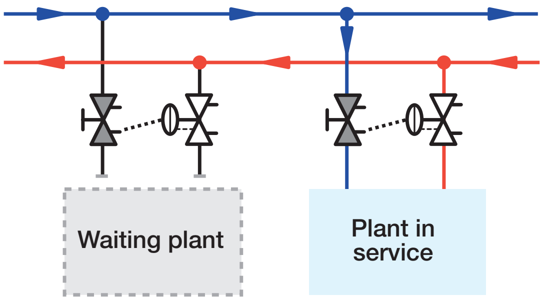

Fig 3.21. An entire new plant can be put in service without disturbing all other plants.

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

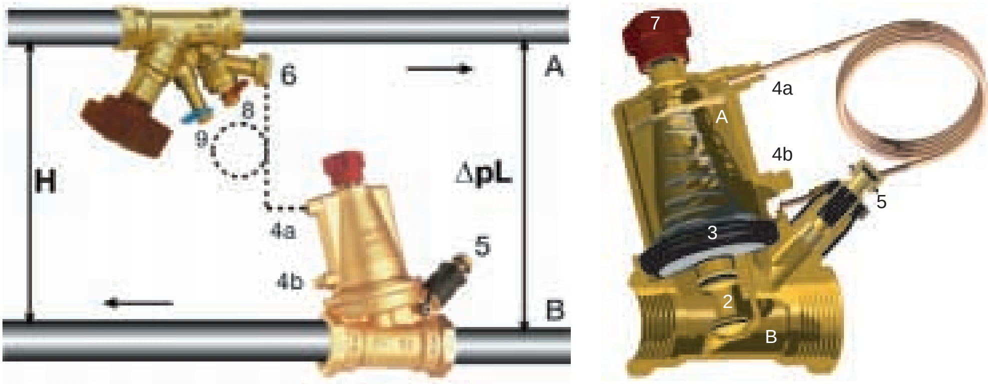

Fig 4.01. STAP stabilises secondary differential pressure ΔpL.

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).

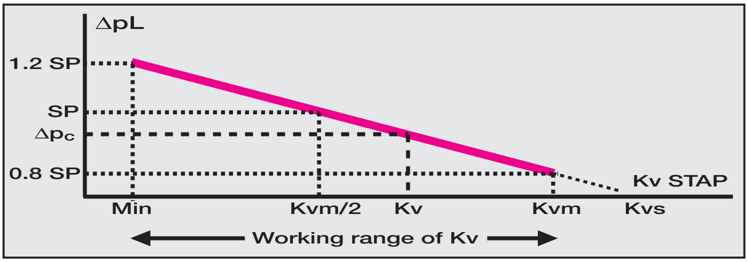

Fig 4.02a. Secondary differential pressure depends on the degree of opening of the STAP.

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.

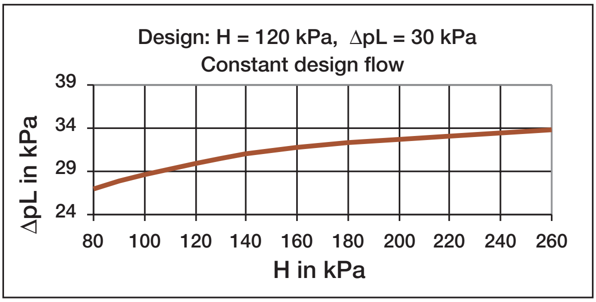

Fig 4.02b. Influence of disturbances on the secondary ΔpL.

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

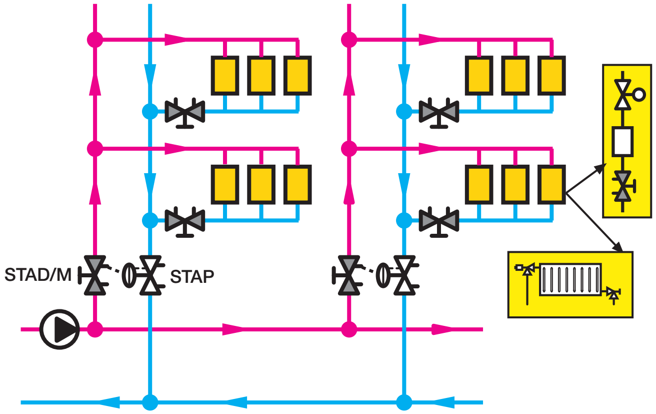

Fig 4.03. A Δp controller STAP stabilises the differential pressure on each riser.

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.

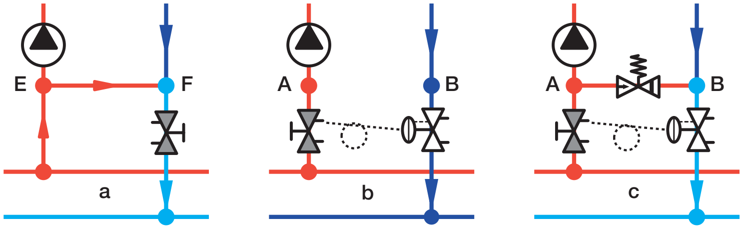

Fig 4.04. How to solve interactivity between primary and secondary pump.

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

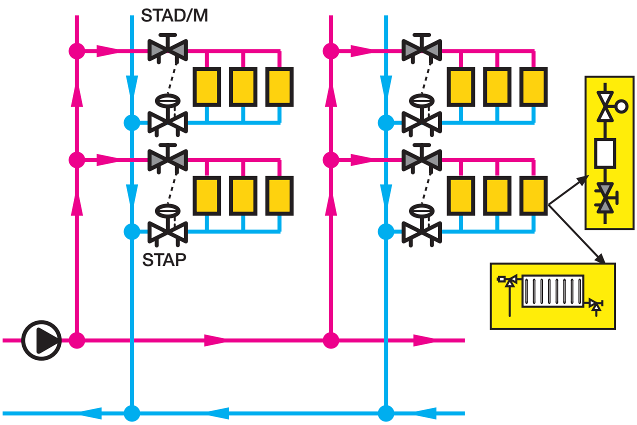

Fig 4.05. A Δp controller STAP stabilises the differential pressure on each branch.

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.

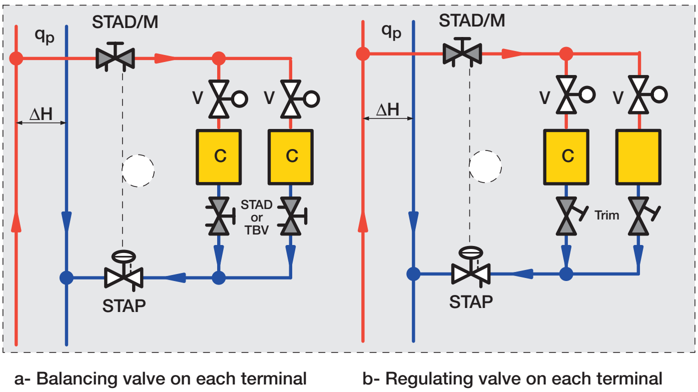

Fig 4.06. One STAP stabilises the differential pressure across a set of terminal units.

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

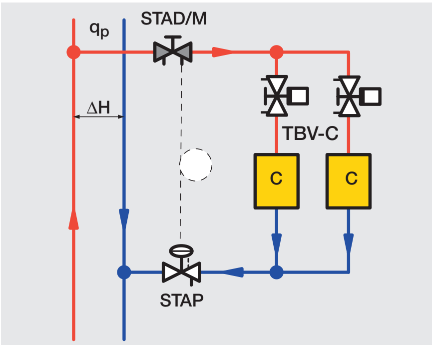

Fig 4.06c. Terminal units are controlled by automatic TBV-C balancing valves.

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.

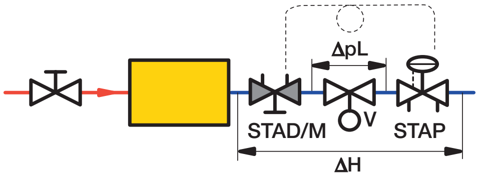

Fig 4.07. A Δp controller stabilises differential pressure across the control valve.

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:

$$\Delta p = \left(0.01 \times \frac{q}{Kvs}\right)^2 \quad (\text{kPa} - \text{l/h})$$

$$\Delta p = \left(36 \times \frac{q}{Kvs}\right)^2 \quad (\text{kPa} - \text{l/s})$$

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.

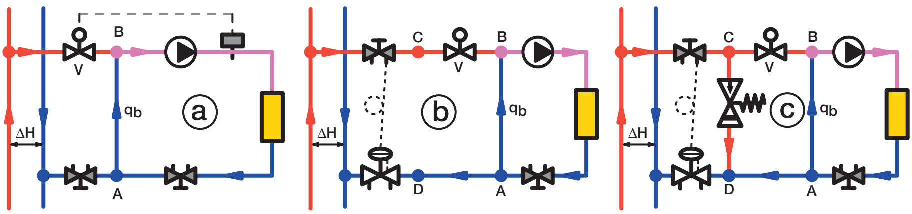

Fig 4.08. A two-way control valve mounted in injection.

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.

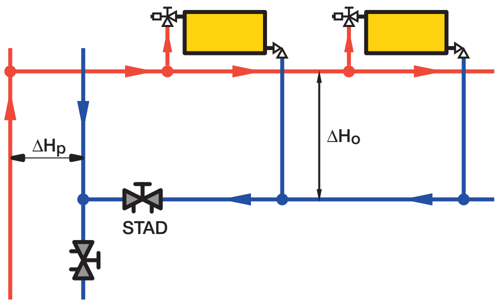

Fig 4.09. Each radiator valve is preset as if it was subjected to the same differential pressure of 10 kPa.

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.

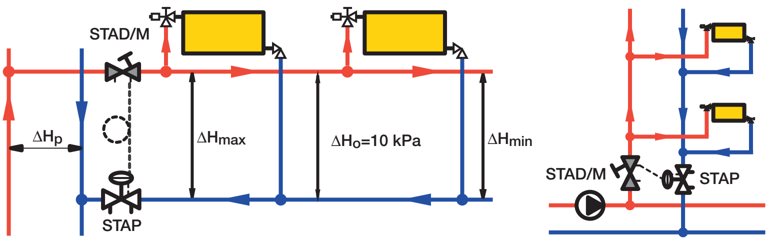

Fig 4.10. A STAP stabilises the differential pressure at the circuit inlet.

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.

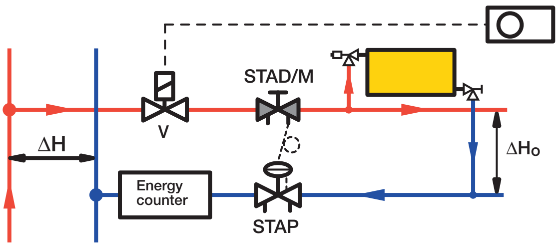

Fig 4.11. One STAP controls the Δp for each apartment.

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.

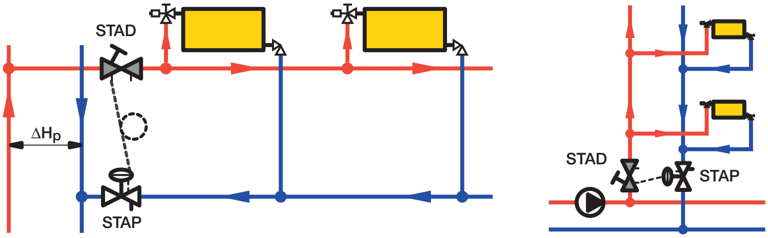

Fig 4.12. The pressure drop in the balancing valve is included in the total Δp controlled by the STAP.

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.

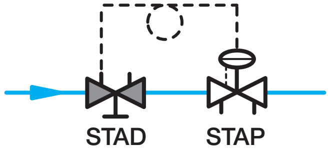

Fig 4.13. STAP as a flow controller.

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!

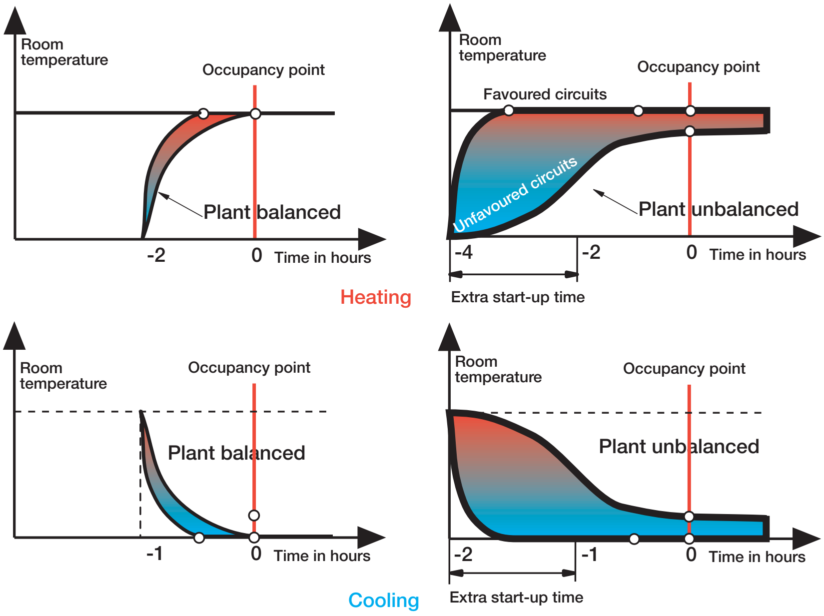

Fig 5.01. An unbalanced heating/cooling plant has to start up earlier, increasing energy consumption.

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.

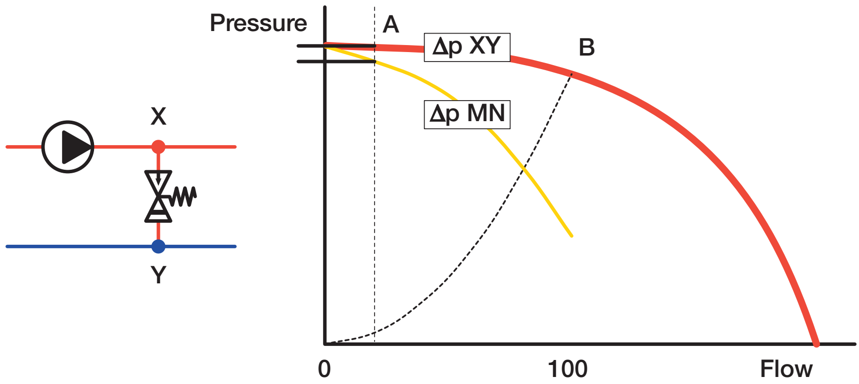

Fig 5.02. The relief valve is situated close to the pump (XY) or on a remote circuit (MN).

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.

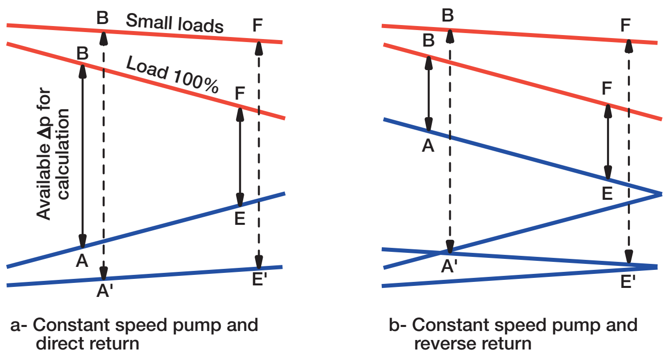

Fig 5.03. Constant speed pump in a direct and a reverse return distribution.

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.

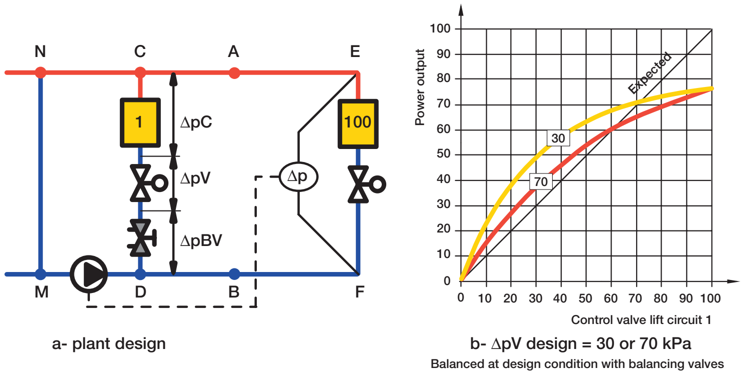

Fig 5.04. Constant Δp is maintained close to the last terminal unit. What happens if all terminals are working at 50% load while terminal 1 has to work at full load?

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:

$100 \times \sqrt{\frac{27}{87}} = 56$ % 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 $\Delta T_c = 10$ K, while in some countries it is quite common for $\Delta T_c = 30$ 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 $H \times q$ (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 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%.

Great Britain
Tour & Andersson Ltd,
Barratt House, 668 Hitchin Road, Stopsley
Luton, Bedfordshire, LU2 7XH, U.K.
Tel: +44 (0)1582 876 232.
Fax: +44 (0)1582 488 678.
www.touranderson.com

Singapore
Tour & Andersson Pte Ltd
171 Chin Swee Road, #05-01 San Centre
SINGAPORE 169877
Tel +65 65 32 06 26.
Telefax +65 65 32 29 19
www.tourandersson.com

Australia
Tour & Andersson
PO Box 154
HIGHETT VIC 3190
Tel +61 3 9553 3366.
Telefax +61 3 9553 3733
www.tourandersson.com

Sweden
Tour & Andersson AB
Norra Gubberogatan 32,
Box 6281, SE-400 60 Göteborg
Tel +46 31 338 73 30.
Telefax +46 31 338 73 49
www.tourandersson.com

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