3.1 The “low ΔT syndrome”

Chillers are sized for certain extreme conditions which depend on the climate relevant for that installation. It is important to realize that, in general, that means that the chillers are oversized since these extreme circumstances occur during less than 1% of the operational time. Effectively we can say that the installation is running in partial load for 99% of the time. When the installation is running in partial load we can experience a phenomenon Low ΔT syndrome which can cause very low chiller efficiencies and fast on-off switching of the chiller. Additionally the low ΔT syndrome prevents the chillers from running in the so-called Max-Cap mode. During Max-Cap the chiller can put out more than its rated capacity at very high efficiencies.

Low DT syndrome occurs when the return supply temperature to the chiller is lower than designed. If the installation is designed for a differential temperature of 6 K but the water fed into the chiller is only 3 K lower than the chilled water supply setpoint, it is easy to understand the chiller can supply maximally only 50% of its rated capacity. It that is insufficient for the situation either the installation will not have enough capacity or an extra chiller needs to be brought online.

Take this example: when the secondary circuit return water temperature is lower than design temperature (due to overflow problems etc.), chillers cannot be loaded at their maximum capacity. If the chillers in the chilled water plant, designed to cool 13°C chilled water return to 7°C, are receiving a design flow rate at 11°C rather than a design temperature of 13°C, the chiller will be loaded at the ratio of:

$$CHL(\%) = \left[\frac{CWRTR - CWSTD}{CWRTD - CWSTD} \right] \text{x } 100\% = \left[\frac{11\text{-}7}{13\text{-}7} \right] \text{x } 100\% = 66{,}6\%$$

Where:
CHL (%) – Percent chiller loading
CWRTR – Real chilled water return temperature (in our case , 11°C)
CWSTD – Design chilled water supply temperature (in our case, 7°C)
CWRTD – Design chilled water return temperature (in our case, 13°C)

In this case, where the low ΔT in the plant (the difference between return and supply chilled water temperature) has been lowered from 6°C (13°C-7°C) design condition to 4°C (11°C-7°C) , the capacity of the chiller has been reduced with 33,4 %.
In many cases the operating efficiency of the chiller can drop 30 to 40 percent when the return chilled water temperature is lower than the design. Contrarily when the DT is increased, the efficiency of the chiller can increase up to 40%.

There are several potential causes of low ΔT syndrome:
• Using three-way control valves:
Three-way valves by their nature bypass the supply chilled water into the return line during part load conditions, causing the chilled water temperature to be lower than design. This exacerbates low ΔT problem (presented in application 2.1.4).
The remedy: Do not use three-way control valves but use a variable flow system with modulating control. If 3-way control valves are unavoidable, application 2.1.3 is recommended to limit overflows in part load conditions.

• Poor 2-way control valve selection with improper system balance:
An improperly sized two-way control valve may allow a higher water flow than necessary. The low ΔT syndrome is worse in partial load due to pressure changes in the system, which results in a high overflow through the control valves. This phenomenon occurs in particular in systems with faulty hydraulic balance as presented in application 2.2.1.
The remedy: 2-way control valves with built in pressure controllers. The pressure control function on the control valves eliminates the overflow problem and therefore eliminates low ΔT syndrome.
• Others such as:
Improper set-point, control calibration or reduced coil effectiveness.

3.2 The "overflow phenomenon"

One of the sources of the well known problems in chilled water systems such as low ΔT syndrome is the overflow phenomenon. In this chapter, we will shortly try to explain what it is and what causes it.

All systems are designed for nominal conditions (100% load). Designers calculate pump heads based on the combined pressure drop in pipes, terminal units, balancing valves, control valves and other elements in the installation (strainers, water meters etc), assuming the installation is operating at maximum capacity.

Consider a traditional system as presented below, fig. 1a, based on application 2.2.1. It is obvious that the coil and control valve located closer to the pump will have a higher available pressure as compared to the one last in the installation. In this application, unnecessary pressure has to be reduced by manual balancing valves, so the manual balancing valves closer to the pump will be more throttled. The system operates properly only with 100% load.

In fig. 1b we see a so-called reverse return system. The idea behind this system is that because the total pipe length for every terminal unit is equal, no balancing is necessary because the available pressure for all units is the same. Please note that if the terminal units require different flows you still need to balance the system with balancing valves. In general we can say that the only proper application of a reverse return system is when we're talking about a constant flow system (3-way valves) and when all the terminal units are of the same size.

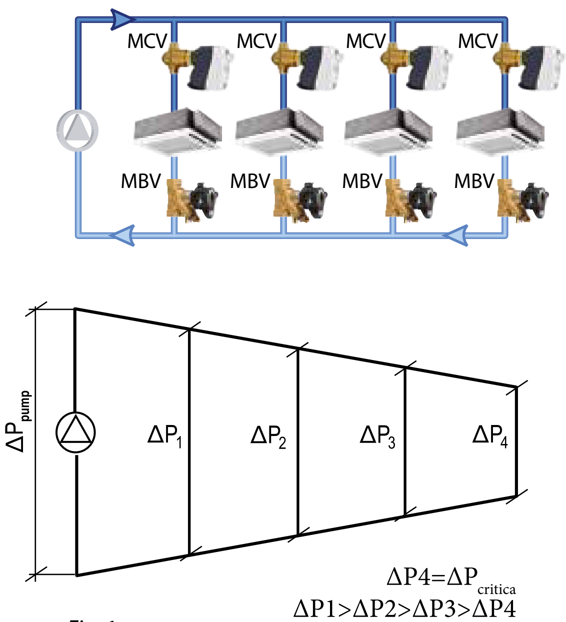

Fig. 1a
Direct return system (not recommended system)

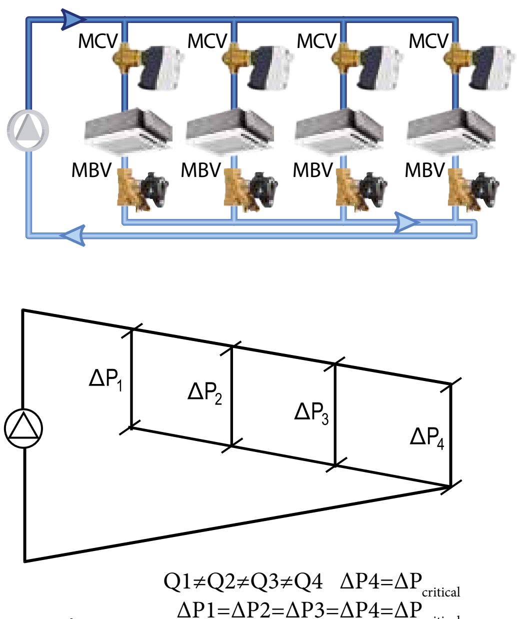

Fig. 1b
*Variable flow **static** FCU control*

To control flow across each coil, two-way control valves are used. Consider the situation in partial load (i.e. coils 2 and 3 are closed).

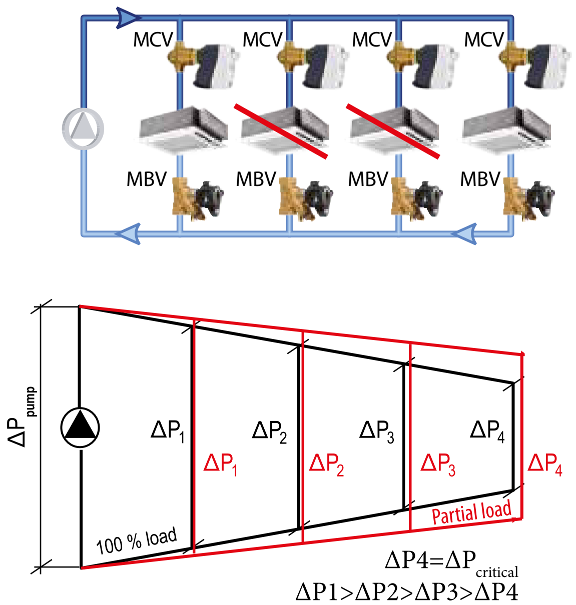

Fig. 2a
Partial load – direct return system

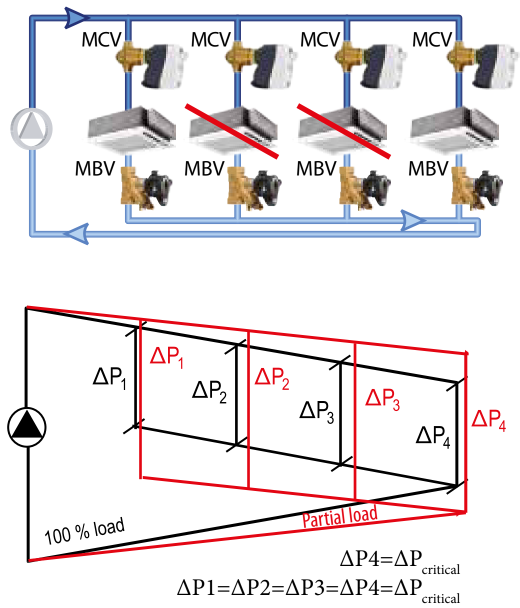

Fig. 2b
*Variable flow **static** FCU control*

Due to a lower flow in the system, the pressure drop in the pipe system decreases, providing a higher available pressure in the still open circuits. Since manual balancing valves (MBV) with fixed, static, settings were used to balance the system, the system becomes unbalanced. Consequentially a higher differential pressure across the 2-way control valves causes overflows across the coils. This phenomenon appears in direct return systems as well in reverse return systems. This is the reason why these applications are not recommended, as the circuits are pressure dependent.

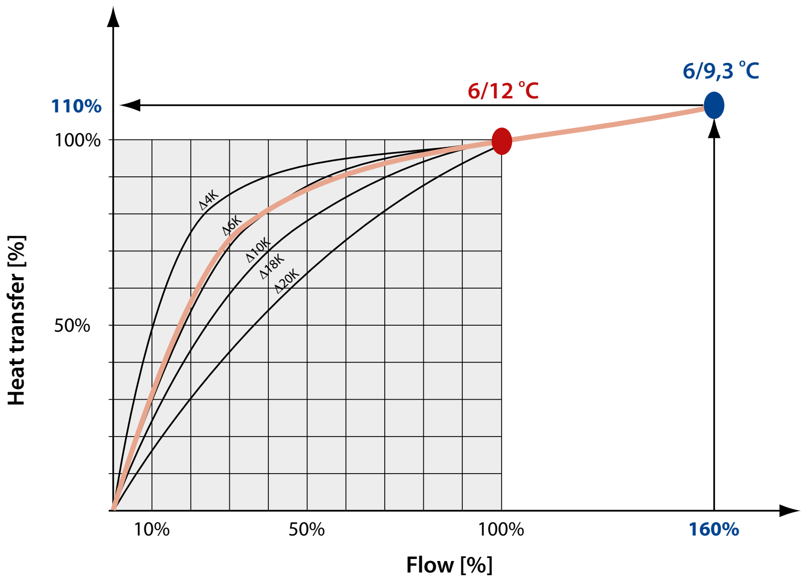

Fig. 3
Terminal unit emission characteristic

The traditional FCU is usually designed for a DT of 6 K. The 100% emission is achieved at 100% flow across the unit at a supply temperature of 6ºC and a return 12ºC. The overflow across the unit has little influence on the emission. However another phenomenon is more critical for proper chilled water system functionality. Higher flow across the units has an incredible influence on heat/cool transfer which means that the return temperature never achieves the design temperature. Instead of the design temperature of 12ºC, the real temperature is much lower, for example 9,3ºC. The consequence of a lower return temperature from the FCU can be low DT syndrome.

For variable flow systems it is not recommended to use fixed speed pumps as they worsen the overflow problem. In fig. 4 this can be seen clearly. The figure represents the pump curve and the differently colored areas represent the pressure drops in the system. The red area represents the pressure drop across the control valve. If we let the pump follow its natural curve we see that with a decreasing flow, the differential pressure will rise. If you compare the differential pressure at 50% of the load you can see that the available pump head is much higher (P_1) than the pump head at full load (P_{nom}). All the extra pressure will have to be absorbed by the control valve. This will cause overflows in the system, as well as a serious deformations of the characteristic of the valve.

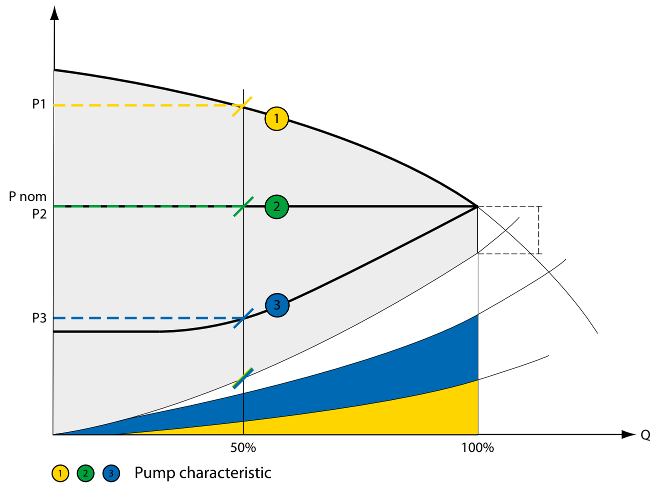

Fig. 4
Different pump characteristics

Today the commonly used Variable Speed Drives (VSD) with pressure transmitters can modify the pump characteristic in accordance with flow and pressure changes in the water system. The nominal flow at 100% load and the above mentioned pressure drop in the system determine the pump head which is equal to the nominal pressure, Pnom. We can see that a constant differential pressure results in a much better situation at partial load, the differential pressure across the control valve will increase much less than when the natural curve of the pump is followed. Please note however that the pressure across the control valve will still rise considerably.

Some modern pumps come equipped with speed controllers that can modify the pump not only based on the pressure but also on the flow, the so-called proportional control. If the flow is reduced, also the differential pressure is reduced. Theoretically this gives the best results as can be seen at P3 in fig. 4. Unfortunately it is unpredictable where in the installation the flow will be reduced so there is no guarantee that the pressure can be reduced as much as can be seen in Fig. 4. It is therefore strongly recommended to limit the difference between P2 and P3 to prevent parts of the installation from starving in certain situations.

The inescapable conclusion is that over- and underflow problems can not be solved by the pump alone. It is therefore strongly recommended to use pressure independent solutions. Pressure independent Balancing and Control Valves (AB-QM) can take care of pressure fluctuations in the system and will provide the terminal units always with the right flow, under all loads of the system. We definitely recommend using VSDs on the pump since that will result in very big savings. As for the control method we recommend to use fixed differential pressure control which will guarantee enough pressure under all circumstances. If proportional control is wanted than the AB-QM can operate under such conditions but we recommend keeping the difference between P2 and P3 to a minimum to prevent starving of certain parts of the installation during partial load.

3.3 The “underflow phenomenon”

As can be seen from Fig. 1a, the available pressure for the first circuit is much higher than the pressure for the last circuit. In this application the MBVs should take care of this by throttling the excess flow. So, the last MBV should be opened as much as possible and the other MBVs should be more and more throttled the closer they are to the pump.

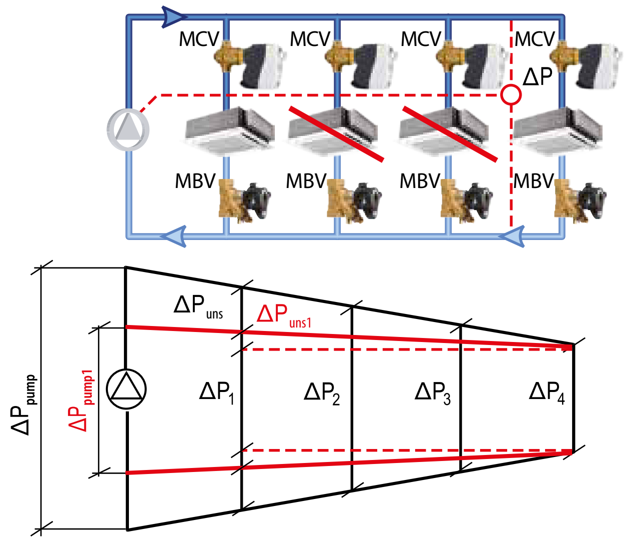

Fig. 5
Direct return system with proportional

A very standard application places the differential pressure sensor controlling the pump at the last terminal unit to minimize pump consumption. We can see what happens when the two middle terminal units are closed. Because the flow in the piping is considerably reduced also the resistance in the system goes down which means that most of the pump head ends up at the end of the installation where the sensor is. This is represented by the red lines in Fig. 5. If you look at the first unit you can see that, even though the pressure for the loop should be the same, it actually gets a much lower differential pressure and therefore too little flow. This can lead to the confusing situation where the installation is operating without problems on full load and when the load is reduced there are capacity problems close to the pump. Needless to say that putting the pump on proportional control will enhance the problems considerably. The pump senses a 50% drop in the flow and will drop the differential pressure accordingly, creating even lower flows in the first terminal unit and a capacity problem at the last terminal unit as well.

An often suggested compromise between creating underflows and minimizing the pump consumption is to put the sensor on a place two thirds of the system. This is however still a compromise and is no guarantee for having the right flow under all circumstances. An easy solution is to mount Pressure Independent Balancing and Control Valves (AB-QM) on every terminal unit and control the pump on constant differential pressure. That way you will maximize the savings on the pump without any under- or overflow problems.