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The Need for Balancing Valves in a Chilled Water System

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In chiller systems, pumps, pipes... and terminal units are selected to meet a specific maximum cooling load. If this capacity cannot be obtained because the system is unbalanced, then the owner has not the full return on his investment.

Hydronic balancing insures that design flows can be obtained

If the supply water flow is correct, any zone flow greater than its design will create less flow available to other zones. This results in remote circuits with less than design flow which are not able to meet their required load.

Hydronic balancing makes the chiller capacity available to all terminals.

In **Figure 1**, a cooling system is shown with two chillers piped in parallel, providing their supply in a production loop. A distribution pump draws chilled water from this loop at tee M, delivers it to the terminal units and returns it at tee N. A two way modulating control valve varies the flow through each terminal in response to a local temperature/humidity controller. A balancing valve limits the flow to each terminal.

If the system is not balanced, the distribution flow "q_s," may be greater than chiller flow "q_g " and a reverse flow "q_b" will occur in the bypass MN to mix with the chiller supply at M. This increases the supply water temperature "t_s" and consequently reduces the chiller capacity available to the distribution pumping system and the terminals. Hydronic balancing avoids this situation. The balancing investment (balancing valves + flow adjustments) typically costs less than one percent of the total HVAC cost. This allows the maximum chiller capacity installed to be fully distributed to all terminals, thereby returning the investment to the owner in terms of building performance and satisfied occupants.

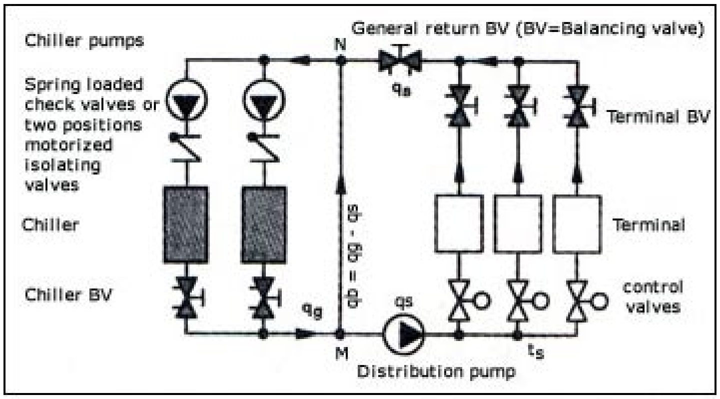

Fig 1: Variable Flow Chilled Water System .

Balancing valves have several functions.

Each terminal is shown with a balancing valve on its return side to act as a shut off valve, a flow measuring devices and a flow adjusting valve. Since a shut off valve is recommended on both supply and return sides of the terminal piping, the balancing valve does not add to the installation labour cost. The balancing valve gives the opportunity to measure the terminal flow accurately in the commissioning process and is an important diagnostic tool during the life of the system. In addition, the balancing valve is provided with a mechanical memory to permit re-opening the valve to its correct position after servicing the terminal.

Manual balancing valves have to be balanced, which is a beneficial constraint, because the balancing procedure gives the possibility to detect most of the hydronic abnormalities during the commissioning operation.

Control valves are not self balancing

In variable flow of water distribution, some believe that two way control valves solve the balancing problem as they automatically provide the required flow in each terminal unit. This is true if the control valves are correctly sized, if the control loop is stable, if the set point of the thermostat is not at an extreme value and if the terminal units are selected for the maximum load required. A lot of "ifs" to meet this statement!

The typical characteristics of a cooling terminal and control valve are represented at **Figure 2** (See **reference A**).

The non-linear characteristic of the cooling coil, makes control difficult with a proportional control valve. At small loads, for a small increase of valve lift, the cooling output increases rapidly. To make the control better, an equal % control valve must be used as shown in **Fig 2b**. For 50% of valve lift, the water flow is 20% (**Fig 2b**). 20% of flow gives 50% of cooling effect (**Fig 2a**). Finally, 50% of valve lift gives 50% of cooling effect (**Fig 2c**). However, the non-linear characteristic of the terminal is really compensated by the equal % control valve characteristic. The design flow must be just obtained when the control valve is fully open (**Fig 2b**)

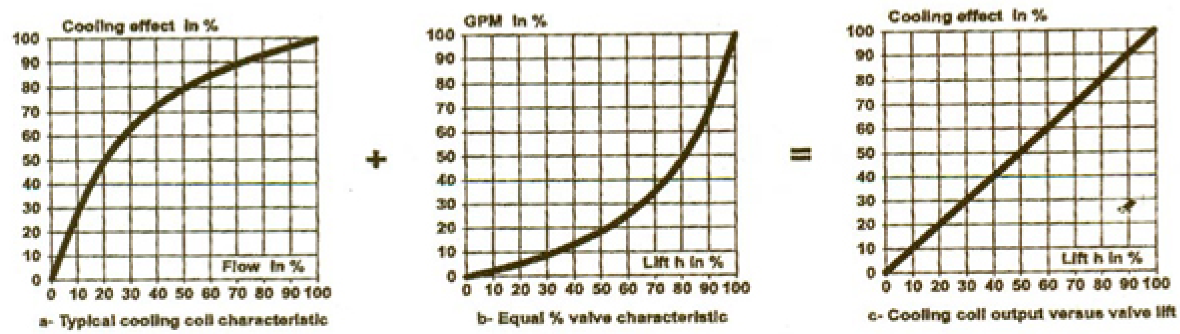

Fig 2: Combination of cooling coil characteristic and equal % control valve to obtain theoretical linear cooling transfer.

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How to obtain the design flow when the control valve is fully open?

To obtain the terminal's design flow at design condition, the pressure drop of the control valve at full open position must be equal to the available differential pressure ΔH less the design pressure drop in the terminal coil (8ft) and the piping components (1 ft) (See **Fig 3a**). In the design stage, who knows the available differential pressure ΔH on each circuit? It is doubtful the designer provides this. Also what is the pressure drop for the terminal coil, which will not be chosen until the contractor is selected? And we may not find, on the market, the correctly sized control valve calculated for this terminal.

In example **Fig 3a**, the pressure drop in the control valve, for a design flow of 25 GPM, must be ideally equal to (27-8-1 =) 18 ft. The corresponding[2] Cv should be 9. On the market, this Cv may not be available and a value of Cv = 12 is selected. This valve (Cv=12) creates a pressure drop of 9 ft for the design flow. This oversized valve gives a flow[3] of 31 GPM (Curve 1 **Fig 3b**). With a balancing valve set for a pressure drop of (27-8-9 =) 10ft, the correct design flow is obtained for the wide open control valve.

The valve characteristic (Curve 2 **Fig 3b**) is closer to the theoretical equal % characteristic. This example shows why the control valves may be oversized and why balancing valves are required. At full load, all the control valves will be fully open. If the piping system is unbalanced, it may not be able to provide the design flow to each terminal. Remember if the flow is greater than the design flow for some terminals, it will result in less flow available for other terminals.

Balancing for the design flow condition guarantees adequate flow in each terminal

Often, the system has to work at full load. If the system is never fully loaded, the chillers, pumps, etc. are oversized and the system is not correctly designed. When the system is balanced, it's not necessary to oversize, reducing the equipment investment and the operating costs. In starting-up each morning after night shut down, the cooling tonnage installed is required but may not be properly disturbed. Any mixing at M (**Fig 1**) keeps the chilled supply water temperature higher than its design.

The room temperatures slowly reduce to their set point, first for the circuits at design flow or greater. These controllers are reducing their flows. Design room temperatures will eventually be reached, but not at the same time for all terminals. If the plant has to be started ½ hour earlier than normal, in comparison with an 8 hours normal day, the energy usage increases by 6.25% per day, i.e. 0.5 x 100/8, which can exceed the distribution pumping costs.

Hydronic balancing allows each control valve, when fully open, to obtain its design flow. This remains true when the system is not working at the design condition, with a total average water flow lower than design. Since a lower average flow means less pressure drops in the pipes and accessories, the available differential pressure and flow on each circuit can only increase.

$$^{2}Cv = \frac{1.5 \times GPM}{\sqrt{\Delta p(ft)}} = \frac{1.5 \times 25}{\sqrt{18}} = 9$$

$$^{3}GPM_2 = GPM_1\sqrt{\frac{\Delta H_2}{\Delta H_1}} = 25\sqrt{\frac{18 + 18 + 1}{9 + 8 + 1}} \approx 31$$

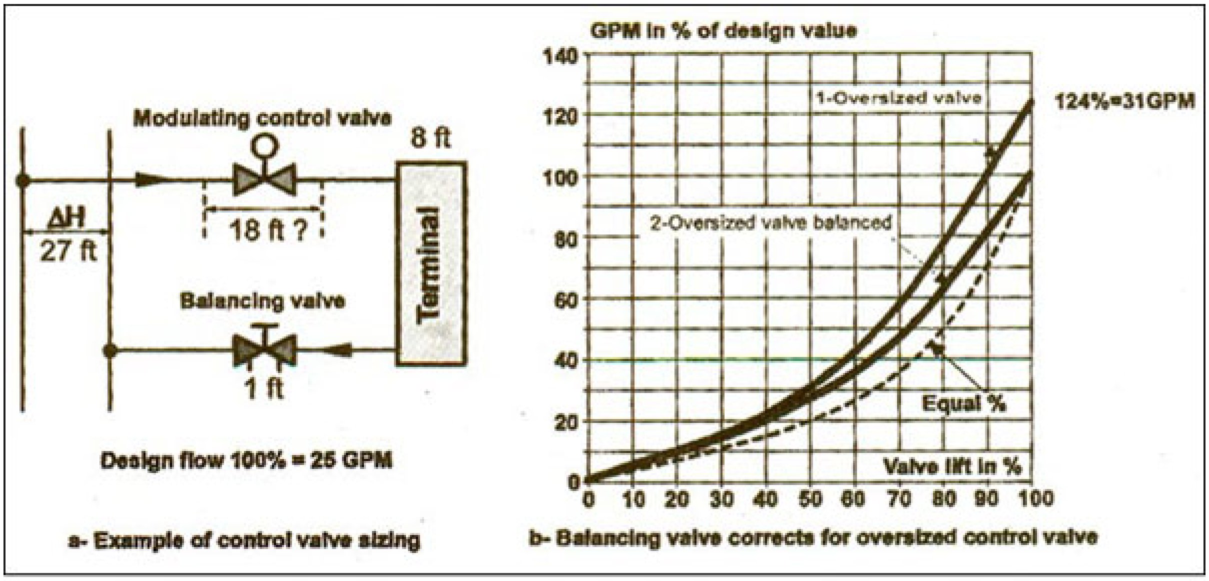

Fig 3: The balancing valve improves the control valve characteristic.

Energy cannot be saved by opening existing balancing valves.

If the balancing valve is adjusted properly, it creates the minimum drop to obtain the design flow for the terminal at design condition. If after commissioning, the balancing valve is opened fully, the flow will increase. In example **fig 3b** (curve 1), the flow and the pumping energy increase 24% of design[4]. The control valve will have to close further, resulting in a shift of control point. The friction head energy is not saved that way, it will just be transferred from the balancing valve to the control valve. It is then obvious that the balancing valves do not create added pressure drops.

Excess pump head is revealed by the pump discharge balancing valve.

One author has said: "*In a plant, an analysis of pressure losses revealed a pressure drop of 50 to 56 psi across a balance valve installed on the discharge of he secondary pumps. This resulted in a energy loss of over 900, 000 kWh per year*."

Thanks to the balancing valve for showing that the pump is oversized. This is to say: "Don't accuse at thermometer if it is too hot in the room". Knowing this pump is oversized, the correct action is to trim or replace the pump impeller or the pump itself, re-open the pump balancing valve to eliminate any additional pumping cost. If only the balancing valve is eliminated, the excess pump head must be absorbed by the control valves or automatic flow limiters without saving any pumping energy. From the point of view, manual hydronic balancing allows the minimum pumping cost in the distribution system.

A terminal coil is not a flow measuring device

The manufacturer does not provide he coil as a flow measuring device. Differences between the catalog and actual pressure drop may vary from 10% to 40%. If a coil is to be used as a flow measuring device, it should be specified with a verified flow vs. pressure drop curve (from the factory) and pressure taps at the inlet and outlet headers. If the coil becomes partially clogged, high pressure drops will be interpreted as a high flow while the actual flow, in reality, may be too low.

Reverse return piping may not solve balancing problems invariable flow distribution.

First let us look at the direct return distribution (**Fig 4**) The first terminal circuit sees a higher differential pressure than the other terminal circuit (ab at first and ef at last, see **Fig 4b**). The control valve is selected according to the available differential pressure ab for first and ef for last. Suppose figure 3 represents the circuit EF in **fig 4a**. The pressure drop ef is 27 ft, including terminal and accessories (9 ft). Then the theoretical pressure drop through the control valve must be 18ft and Cv[5] is 9. If the available differential pressure for the first terminal circuit (ab in **figure 4b**) is 57 ft, the control valve has to be selected for (57-9=) 48 ft pressure drop and its Cv = 5.4. At 50% average flow (**Fig 4c**), the pipe pressure drop decreases and the pump head increases. The differential pressure for the first terminal circuit increases from 57 ft to 65 ft[6] and will require the control valve to work very close to its seat.

Now let us look at reverse return distribution (**fig. 5**). In early systems, reverse return was provided for constant flow distribution serving identical terminal units. It normalizes the differential pressure on all circuits for design flow. At design condition (**Fig 5b**), all terminal circuits see the available differential pressure of 27 ft. At 50% average flow (**Fig 5c**), all terminal circuits are submitted to a differential pressure of 58 ft which requires the control valves to work closer to their seat (with a risk of unstable control). In conclusion, control valves may hunt at small load for remote circuits in direct return while all of them may hunt in reverse return. To reduce changes in differential pressure, a variable speed controller on the pump can be used to keep it constant, for example, between C and D on **figure 5a**. This reduces the pumping cost and improves the quality of control (See references B & C)

Please note that in certain cases, the pump head has to be increased in reverse return design. In figures 4 and 5, if the first circuit requires a differential pressure of 47 ft this condition is satisfied in direct return (**Fig 4b**) since 57 ft is available. It is not he case in reverse return (**Fig 5b**) as only 27 ft is available. Consequently, for the reverse return system, in this case, the pump head has to be increased by 20 ft.

[4] $WPH = \frac{GPM \times \Delta H}{3960}$ - For a constant ΔH, the pump energy increases proportionally with the flow.

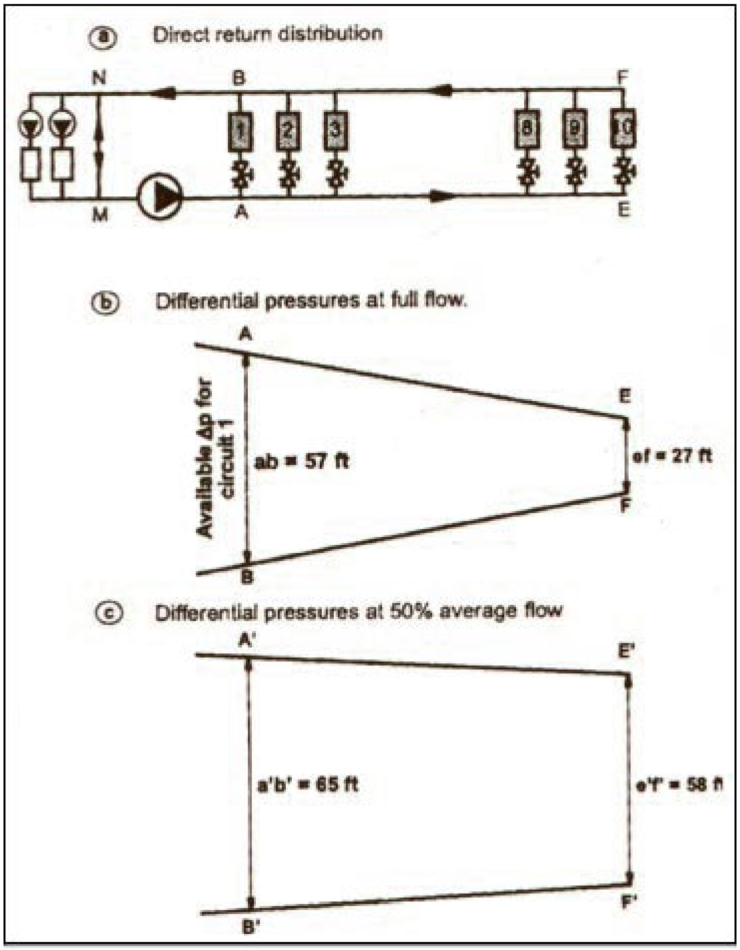

Fig 4: Direct return distribution

Constant speed pump and direct return systems

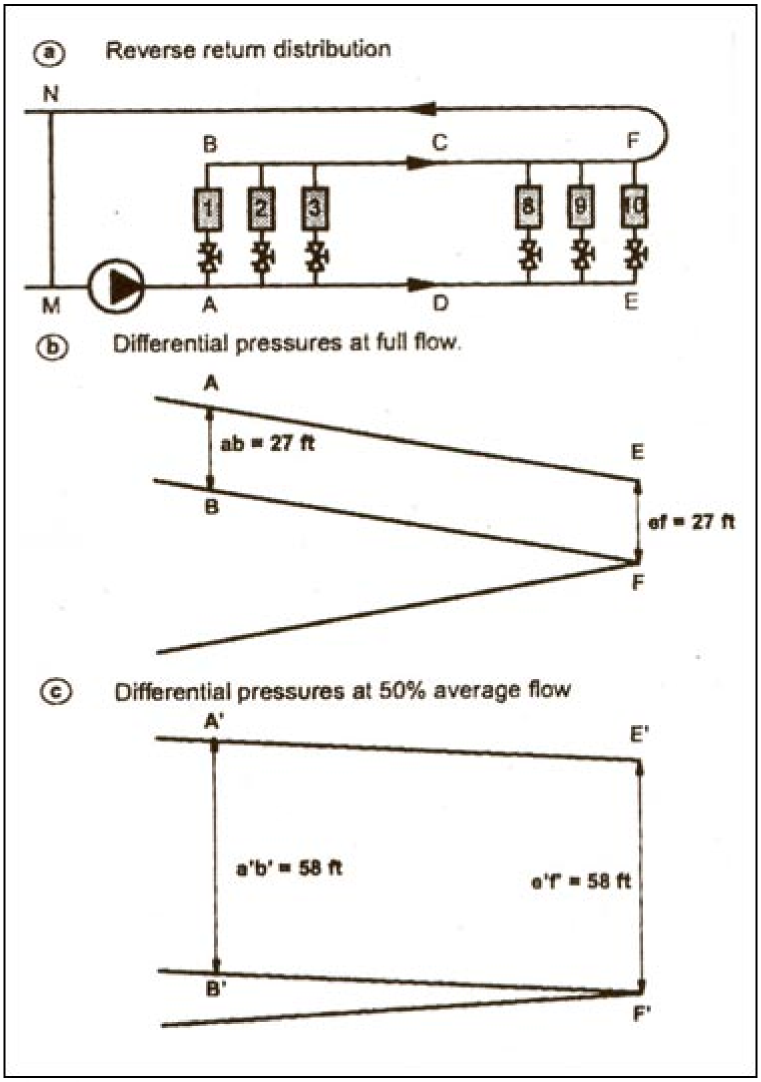

Fig 5: Reverse return distribution

Stability of the terminal control is dependent on the control valves authority (see references D & E) which is defined as:

$$\beta = 100 \times \frac{\Delta p \text{ control valve fully open and at design flow}}{\Delta p \text{ across closed control valve}} \% \quad (7)$$

For good control, it must be at least 25%. In **figure 3**, expected control valve pressure drop is 18ft. The control valve actually selected has a design pressure drop of 9ft.

$$^{5}Cv = \frac{1.5 \times GPM}{\sqrt{\Delta p(ft)}} = \frac{1.5 \times 25}{\sqrt{18}} = 9$$

[6] Since ΔH - $(GPM)^2$, the actual pressure drop in pipes is $(1/2)^2$ (57 - 27) ≈ 7 ft. Then the actual available differential pressure is 65-7=58 ft.

[7] Δp across closed control valve is the same as the available differential pressure for the circuit. The pressure drop created by the balancing valve these not intervene in any of theses factors and has consequently no influence on the control valve's authority.

So, its authority is 9/27x100=33%. 9 ft is a fixed value as soon as this control valve has been selected. The only variable parameter is the available differential pressure (27 ft). If this differential pressure increase, the control valve authority decreases with risk of control valve hunting.

Let us consider a direct return system with a constant speed pump according to **figure 4a**. Design pressure drop[8] for the control valve in circuit 1 is 40 ft. At 50% average flow, the available differential pressure for the circuit 1 increases from 57 to 65 ft (**Fig 4c**) and the control valve authority from 40/57x100=70% to 40/65x100=62%. This increase in differential pressure does not compromise the control valve performance. However, for the last circuit, the available differential increases from 27 to 58 ft which greatly decreases the control valve authority from 33% to 9/55x100=15%. In reverse return (**Fig 5c**), for all the circuits, the authority will be only 15% (9/58x100).

Control the differential pressure close to the last terminal

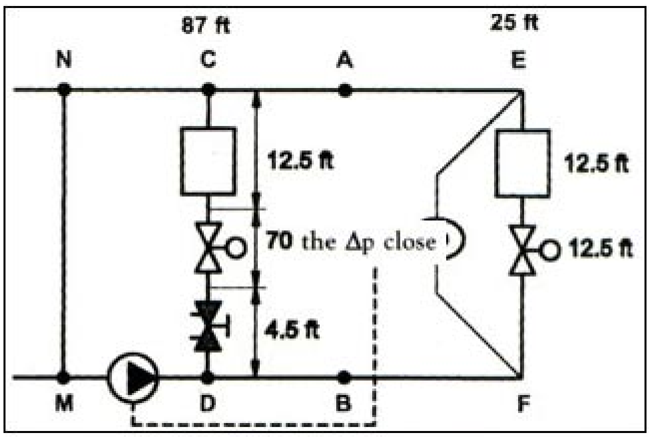

Fig 6: The variable speed pump is controlled by the Δp close to the last circuit

An example of large cooling system with variable speed pump is shown in **figure 6**. The variable speed pump keeps the available differential pressure constant for the last circuit. When all terminals experience nearly the same load at the same time, it is a suitable solution.

The design condition is: Δp coil = 12.5 ft, ΔH CD = 85 ft and ΔH EF = 25 ft. For the last terminal, the design pressure drop for control valve is 12.5ft (authority 50%). For the first terminal, the best available choice is 70 ft (authority 80%) assuming that the valve noise is acceptable and the valve actuator is capable of closing against 87 ft head. If not, the design of the system has to be change. If all the terminal valves are closed and only terminal 1 is in operation, the differential pressure ΔH CD drops from 87 to 25 ft since the differential pressure control is set for ΔH EF = 25 ft. The maximum flow in terminal 1 is reduced. It drops by ratio:

$$GPM_2 = GPM_1\sqrt{\frac{\Delta H_2}{\Delta H_1}} = 100\sqrt{\frac{25}{87}} = 54\%$$

Then the cooling output drops to 80% (**Fig 2a**). The 20% reduction in cooling output, in this extreme case, may be acceptable. The case could be much more dramatic if he system is not balanced. To avoid the reduction in flow for the first terminal, its control valve could be selected based on the minimum available differential pressure (25 ft). The first terminal would be now able to have its design flow in the worst case. However, during start-up, the control valve is fully open with an available differential pressure of 87 ft. In this case, the water flow in the first terminal will reach 187% of its design value[9]. In start-up, most of the control valves are wide open. The pump runs at maximum speed and cannot keep 25 ft at the last terminal.

Flow greater than design may cause low flows in other parts of the system generating complaints from occupants. Moreover, if the return main flow is greater than design, the flow in the bypass MN (**Fig 1**) will reverse creating an increase of the supply water temperature at mixing point M. This makes morning start-up a time consuming operation. One solution for terminal 1 is to keep the differential pressure across the control valve constant with a local differential pressure controller (See reference F). (**Fig 7**)

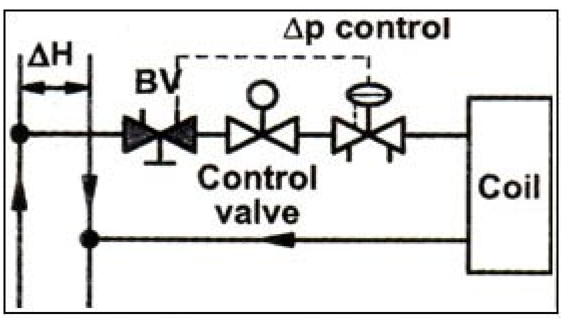

Fig 7: A self acting Δp controller maintains Δp across the control value

The set point of the Ap controller is selected to obtain the design flow for the control valve when fully open. That way, the control valve is always correctly sized and its authority is kept close to 100%.

Control the differential pressure at the middle of the system

When the system can be represented by **figure 6**, the differential pressure may be maintained constant at the middle of the system (AB instead of EF) Taking the same example as before, the set point would be 56 ft. When the average load is close to zero, the maximum flow obtainable at the first terminal would be of 80%10, reducing the maximum power output by only 6% (**fig 2a**). For the last terminal, the control valve authority will decrease from 50% to 22% which is still acceptable.

Control the differential pressure at several locations in the system (See reference G)

Most of the systems cannot be represented by **figure 6** as thee are risers, branches, etc. Control valves are calculated to take a maximum of pressure drop in design condition and the remaining differential pressure is taken up by a balancing valve. So the best authority is obtained and flows over design are avoided. The Δp should be measured in strategic points, for instance on some risers. Each Δp sensor is associated with a controlled whose set point corresponds with the local design value. The variable speed pump is controlled by the highest demand. This gives the guarantee that all terminals can always work at design value when necessary and the control valve authority is generally improved at small loads.

Conclusion

A HVAC system is designed for a certain maximum load. If full load cannot be obtained because the system is not balanced for design condition, the owner will not receive the return for his investment. Control valves cannot handle this situation as they are fully open when maximum load is required. Sizing the two-way control valves is difficult and the valves calculated may not be available on the market. Consequently, they are mostly oversized. Hydronic balancing is then essential and represents typically less than one percent of the total HVAC investment.

Two-way control valves have to be sized properly to achieve stable control. A good rule is to obtain at least an authority of 25% in the worst condition. When this goal cannot be achieved, reconsider at least the design of the circuits concerned by installing, for instance, a local self acting differential pressure controller to maintain the control valves authority close to 100% (**fig 7**).

Pumping costs can also be reduced by increasing the design water ΔT and by using variable speeds pumps with optimum location of the Δp sensor transmitters. Terminal proportional integral (PI) controls require lower flows at intermediary loads than on-off controls (**fig 2a**) and can therefore also reduce the pumping costs.

It is also important to compensate for pump oversizing. The excess pressure is shown on the balancing valves in the pump discharge. When the correct changes to the pump have been made, the pump discharge balancing valve should be fully opened.

Hydronic balancing requires the correct tools, up to date procedures and efficient measuring instruments. The manual balancing valve remains the most reliable and simplest product to obtain the correct flow for design condition and always gives the opportunity to check flows for diagnosing system problems.

$$^{9}GPM_2 = GPM_1\sqrt{\frac{\Delta H_2}{\Delta H_1}} = 100\sqrt{\frac{25}{87}} = 187\%$$

$$^{10}GPM_2 = GPM_1\sqrt{\frac{\Delta H_2}{\Delta H_1}} = 100\sqrt{\frac{56}{87}} = 80\%$$

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