

# 3. Consequences of unbalanced hydronic systems

## 3.0 Importance

In designing a system, it is necessary to size the pump to provide sufficient head to ensure a correct flow of water in the highest pressure drop or “least favored” circuit.

The pump head is chosen to obtain a sufficient pressure to each terminal, taking into account pressure drops in the boiler/chiller, pipes, valves and fittings.

Thus, in simple terms, in any multi-circuit system, the circuits farthest from the pump normally have the lowest differential pressure and the circuit closest to the pump has the largest one.

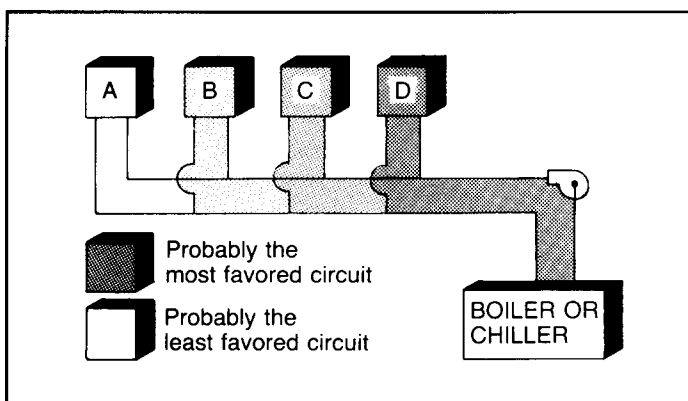


Fig. 15. Hydronic system with favored circuits

### 3.0.1 Low pressure drop of favored circuits

For the low pressure drop circuits, the excess water flow has very little effect on the heat emission since

the emission curve increases only slightly with over-flow. The zone temperature controls will attempt to compensate for any overheating. However, if these are on/off controllers, the room temperature swings continuously around the set point by 3°F to 5°F.

Energy is wasted because the room occupant will set the temperature higher in the winter and lower in the mild season to try to compensate for the hunting and its discomfort.

The situation is not much better with the use of a proportional controller, which normally has to adjust to the correct water flow to obtain the required emission.

As the room temperature increases, the control valve closes slightly. Decrease of the flow has little consequence on the coil emission. The room temperature continues to increase, until finally the control valve is completely closed.

Moreover, the valve characteristic is valid within a certain degree of opening, from approximately 3.3% of its normal maximum flow. If this maximum is three times larger than normal, the control valve can manage to a minimum of 10% of the calculated water flow.

Referring back to Figure 2, 10% of water flow provides 50% of the maximum calculated emission. Under such conditions, a proportional control valve works nearly in “on/off” mode with the same poor results.

### 3.0.2 Low resistant circuits

Without hydronic balancing, the low resistance or favored circuits constitute short circuits which increase the total water flow. According to the Darcy-Weisbach formula, the pressure drop in the pipes varies approximately to the square of the flow:

$$\frac{H_2}{H_1} = \left( \frac{USGPM_2}{USGPM_1} \right)^2$$

Doubling the flow causes the pressure in the pipes to increase four times. This also doubles the velocity, and the pressure drop through the pipe is increased by an inverse proportion to the flow area's cross section, but approximately by four. These changes will first affect the most favored circuits and the expected differential pressure is not detected on the terminals furthest from the pump. The consequences are well known and some circuits in these zones do not deliver the required heat emission.

In a heating system, one solution is to raise the heating water temperature schedule, but areas that were overheated will be even more so.

In a cooling system there is not enough latitude to lower the water temperature. We could increase the pump head and its pumping energy consumption to solve this question but the situation of the favored circuits will be adversely effected.

### 3.1 Energy cost increase with poor room temperature control

We have concluded that an unbalanced system does not permit proper control of the room temperature. It is interesting to calculate the energy cost increase related with a constant deviation of 1°F in the room.

### 3.1.1 In heating

For a room temperature set-point at 70°F and a minimum calculated outdoor temperature (OA) = 20°F, overheating the room 1°F increases the heat loss by the following:

Since the heat loss is

$$UA = \text{Constant} = \frac{Q}{(\text{Room} - \text{outdoors})}$$

$$= \frac{100\%}{(RT-OA)} = \frac{100\%}{(70 - 20)} = 2\% \text{ max. heat loss}$$

$$Q = U A (\text{Room Temp.} - \text{Outdoor Temp.})$$

In this case, overheating the room by 1°F increases the energy consumption by about 2% of the maximum heat loss. If we assume the yearly mean energy consumption is approximately 40% of the maximum, then the mean consumption is increased by:

$$2.0 \times \frac{100}{40} = 5\% \text{ for each } 1^\circ\text{F overheating}$$

### 3.1.2 In cooling

Following the same calculation with room at 75°F and outdoor air at 100°F increases the energy consumption by approximately 10% for a 1°F overcooling. This does not include the additional energy for latent cooling.

These calculations are rough approximations and have to be adapted according to actual local conditions using, for instance, the ASHRAE BIN load method (Ref: B-3, chapter. 28) and taking into consideration the real internal loads.

These calculations give an idea about the amount of energy that can be saved with stable control of the room temperature, which will depend on the balancing of the system.

## 3.2 Pumping cost estimation

The formula for estimating required pump brake horsepower is:

$$\begin{aligned} &\text{Pump brake horsepower} \\ &= \frac{\text{USGPM} \times \text{Ft Head} \times \text{S.G.}}{3960 \times \text{Pump Eff}^y} \end{aligned}$$

Since 1 HP = 2545 BTU/Hr  
Water S.G. = 1.0

The equation now becomes:

$$\begin{aligned} &\text{Pump brake horsepower} \\ &= \frac{\text{USGPM} \times \text{Ft Head} \times 2545 \text{ BTU/Hr}}{3960 \times \text{Pump Eff}^y} \end{aligned}$$

If we assume a pump efficiency of 75% and motor efficiency of 85%, the pump energy can be expressed as:

$$\begin{aligned} &\text{Pump Energy} \\ &= \frac{\text{USGPM} \times \text{Ft Head} \times 2545 \text{ BTU/Hr}}{3960 \times 0.75 \times 0.85} \\ &= \text{USGPM} \times \text{Ft Hd in BTU/Hr} \end{aligned}$$

since the maximum heat delivered by the pumping system is:

$$Q = \text{BTU/Hr} = 500 \times \text{USGPM} \times \Delta T$$

and if we consider that seasonal consumption is approximately 40% of the maximum, then the % pump energy of the total seasonal energy is:

$$\begin{aligned} &\frac{\text{Pump Energy}}{\text{Average Energy Consumption}} \\ &= \frac{\text{USGPM} \times \text{Ft Head}}{(40/100)(500) \text{USGPM} (\Delta T)} = \frac{\text{Ft Head}}{2 (\Delta T)} \% \end{aligned}$$

The above formula is for estimating purposes only to provide a "rule of thumb."

The percentage pump energy is directly increased in relation to pump head and inversely to the water temperature change.

Examples:

### Heating:

With Ft = 80 Ft and  $\Delta T = 20^\circ\text{F}$  we obtain pump energy cost = 2% of energy consumption

### Cooling:

With Ft = 30 Ft and  $\Delta T = 10^\circ\text{F}$  we obtain pump energy cost = 4% of energy consumption

On the cost side, one BTU consumed by an electrical motor costs more than one BTU delivered by a coil. From the cost point of view these results have to be multiplied by at least a factor of 2.

In heating, this consumption is not a total loss since a part of this energy contributes to heating the water.

In cooling, however, we have to add additional energy with the chiller and this generates additional cost.

Pumping costs are substantial but can be decreased by proper hydronic balancing to reduce the water flow to the correct value, making it possible to reduce the pump head and motor size.

## Pumping Costs

$$= \frac{\text{USGPM} \times \text{Ft}}{3960} \times \frac{1}{\text{Motor eff'cy}} \times \frac{1}{\text{Pump eff'cy}}$$

$$\times \frac{0.746 \text{ KW}}{1 \text{ HP}} \times \frac{8760 \text{ hrs}}{1 \text{ Yr.}} \times \frac{\$0.10}{\text{KWH}}$$

Pumping costs expressed in \$/year, for a pump running the whole year, (assuming motor efficiency of 85% and pump efficiency of 75%) can be estimated with the following formula:

### Pumping costs

$$= \frac{\text{USGPM} \times \text{Ft}}{3960} \times \frac{1}{0.85} \times \frac{1}{0.75} \times \frac{0.746 \text{ KW}}{1 \text{ HP}}$$

$$\times \frac{8760 \text{ Hours}}{1 \text{ Year}} \times \frac{\$0.10}{\text{KWH}}$$

$$= 0.254 \times \text{USGPM} \times \text{Ft Hd in } \$/\text{year}$$

Example:

Pump Head = 80 Ft and Flow = 300 USGPM gives a cost of \$6200/year.

In an existing building, we hesitate to replace oversized pumps with smaller units. However, if we are sure that a good hydronic balance will give the right flow to each terminal, we can confidently trim impellers or replace pumps with smaller units, resulting in considerable operating cost savings.

The use of variable frequency drives to reduce pump speed will accomplish pumping savings similar to savings obtained by trimming the impeller. The cost of the v-f drive and efficiency of operation makes its use more costly than trimming the pump impeller.

The same situation exists when we design a new building. Instead of assuming (for heating) a water temperature drop of 20°F a larger Δ T can be adopted; for instance 60°F. This reduces the water flow, the pipe sizes and the pressure drops in the terminals.

Pumping costs decrease drastically and the terminal emissions are more sensitive to water flow variations (see Figure 5), which improves the performance of the control loops at any load.

However, since the terminal emissions are more sensitive to the water flow, the system must be well balanced on the hydronic side.

## 3.3 The choice of the water temperature drop

### 3.3.1 Cooling coil installation

In cooling, the water temperature rise is usually selected between 8° and 12°F, with an inlet water temperature of about 42°F to 48°F. Care must be taken not to take too large a water temperature rise because room humidity may increase. In comparison, a 12°F water rise compared to 8°F reduces the water flow by 30% and also the pump head, with substantial energy savings. However, the coil has to be re-selected according to the new working conditions.

### 3.3.2 Heating

In an HVAC installation, when the same pump is used for cooling and heating, this pump is chosen for cooling conditions which generally result in large flows for heating. In this case, the water supply temperature has to be reduced.

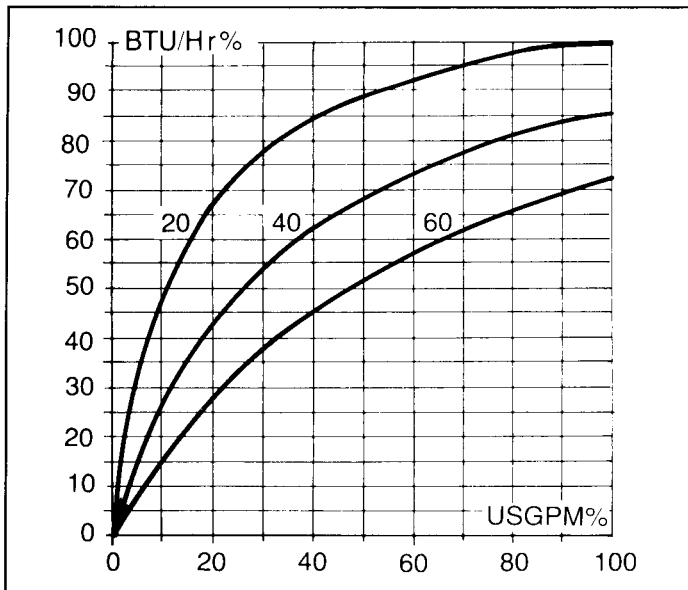


Fig. 16. Heat emission of coil with 20°F, 40°F, and 60°F ΔT and 200°F supply.

If 100% of the water flow represents 100 USGPM for ΔT = 20°F, it represents 43 USGPM for ΔT = 40°F and 24 USGPM for ΔT = 60°F.

### 3.4 Advantages of small water temperature drops

1. Smaller terminal sizes due to turbulence in tubes (velocity increase) and increase of average water temperature.
2. Larger pressure drops in terminals which are less sensitive to changes of the differential pressure applied on the risers. High coil pressure drop serves to purge air from the coil.
3. Minor influence of the water flow on the maximum emission.
4. More uniform air temperature leaving the coil when the modulating control valve is open.

#### 3.4.1 Advantages of higher water temperature drop

##### 1. Smaller pumps and energy consumption

Example: in an existing system with a ΔT of 20°F and an entering water temperature of 190°F. Increase

the EWT from 190°F to 205°F to compensate for a 50% reduction of the water flow which influences the water temperature average and the turbulence coefficient. These values are approximate.

Initially, the heating installation was working 200 days per year with 80 Ft head and 300 USGPM flow.

Pump costs in \$/year

$$= (0.26) \times (\text{USGPM}) \times \text{Ft.} \times \frac{200}{365} = \$3419 \text{ per year}$$

To reduce the flow by 50%, decrease the pump head from 80 to 20 Ft; if we install CBV balancing valves and a new pump, the new conditions may be, for example, 30 Ft and 150 USGPM with a ΔT of 40°F.

New pumping cost:

$$= 0.26 \times 150 \times 30 \times \frac{200}{365} = \$641 \text{ per year}$$

These rough calculations show the potential pump energy savings from using larger ΔT in heating.

##### 2. Better control of a proportional control valve

a) Influence of oversizing:

If we consider a coil which has curve characteristics as presented in Figure 16, an oversized coil of 15% requires the control valve to reduce the water flow and the maximum permissible opening of the valve is reduced.

The control valve now works in a smaller range of the lift which is not favorable for stability.

Nominal WTD	Maximum valve opening allowed
20F	45% water flow
40F	62%
60F	71%

b) Control of small loads (coil according to Figure 16):

A control valve with a rangeability of 30 to 1 (see section 2.7.1, 2.7.2) can control a minimum water flow of 3.3%.

If the valve authority is 0.5 (see section 2.7.1), the minimum water flow = 4.7%. According to the nominal  $\Delta T$  adopted, the corresponding minimum load which can be controlled is given below:

Nominal $\Delta T$	Minimum emission
20F	28%
40F	16%
60F	12%

If we oversize the coil 15% and have an overflow of 50% resulting from bad balancing, we will obtain the following values:

Nominal $\Delta T$	Maximum water flow	Minimum controllable emission value
20F	30% flow	44% of nominal
40F	42%	26%
60F	47%	19%

### 3.5 Specifying the system

In order to correctly specify the control valves, the whole hydronic system must be analyzed.

The objective is to find a compromise between the desire for a low pump head, and the opposing need for a high valve authority. A good compromise is a valve authority  $\beta = .5$ ; it causes only a moderate distortion of the characteristic of the control valve, and therefore makes it possible to provide a good control function, without significantly increasing the need for a high pump head.

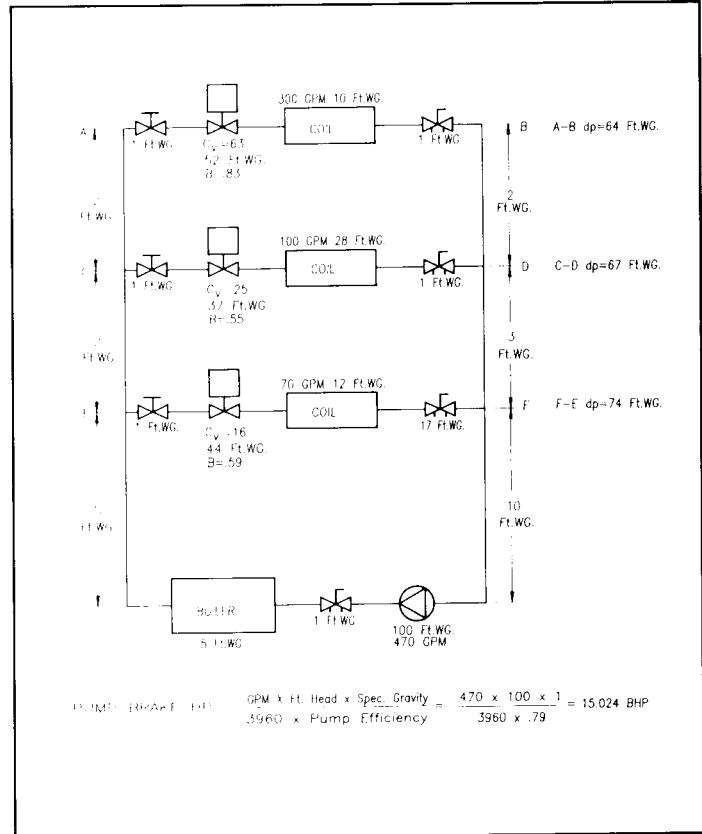


Fig. 17

A way of reducing the necessary pump head is to replace the coil(s) having the highest pressure drop with coils that produce low pressure drops. The impact is quite dramatic, because, for example, a 10 FtWG reduction across the coil will permit a 10 FtWG reduction of the pressure drop over the control valve, so the pump head can be reduced by 20 FtWG. The additional cost for this type of coil is quite small, and only the most resistive coils have to be replaced. A method that often proves to be very effective in reducing the required pump head is the use of a secondary circulating pump at the most resistive circuit(s).

This is especially true if one of the terminals stands out and requires a much higher differential pressure than the other terminals. The addition of a small circulating pump can justify a reduction in the size of the main circulating pump, reducing overall operating costs.

However, if this is done, all the pressure drops and Cv values in the whole system must be re-calculated, based upon the new "most resistive" circuit. Usually the result is a significant reduction of the head, but sometimes the result does not warrant the effort. This depends on how well the available Cv values of the control valves match the new requirements.

### 3.6 Secondary pumps

Secondary pumps at the coils provide many advantages :

- Reduces risk of freezing
- Gives even temperature across the surface of the coil, and thereby a homogeneous air temperature after the coil
- Improves the controllability at low loads
- "Eliminates" the pressure drop across the coil
- Reduces the need for a high pump head at the main circulating pump
- A smaller control valve can be used
- A very high valve authority can be accomplished
- The characteristics of the combination of coil and pump is better than for the coil alone, so the controllability is improved

A coil consists of horizontal tubes, connected between a supply manifold and a return manifold. The water temperature drops as it passes through the tubes so it is colder in the return manifold than the supply manifold. Cold water has a higher density than warm water, so there is a tendency for a water circulation from the bottom of the return manifold back through the lowest tubes to the supply manifold, and then from the top of the supply manifold to the return manifold, causing self-circulation.

Whether or not circulation actually will take place depends upon the total flow across the coil. How-

ever, instead of an even distribution of the flow between the tubes, the tendency to self-circulation will add to the flow in the top tubes, and counteract the flow in the lower tubes.

If no secondary circulation pump is used, and the flow through the coil is low, there will be a large difference in flow between the top and bottom tubes. The temperature drop depends upon the flow, so the bottom of the coil will be cold, and the top warm. There is a risk of freezing in the bottom of the coil, and the uneven temperature distribution across the coil area causes a wide variation of air temperatures when leaving the coil. The air in the upper part of the duct will be warmer than the lower part. This necessitates the use of averaging temperature sensors, and makes it hard to accomplish an accurate temperature control.

A secondary circulation pump increases the flow so much that the tendency to self-circulation will have no significance, and the flow/temperature will be evenly distributed.

The secondary circulating pump eliminates the pressure drop across the coil, and therefore reduces the need for a large head at the main circulating pump. (See section 3.5, "Specifying the System")

Practically the whole differential pressure is applied to the control valve. This will result in relatively small control valves, for which the valve authority will be very high. When a secondary circulating pump is used, the flow through the coil will be constant. The heat emission is controlled instead by the temperature of the circulating water.

This is done by varying the flow of water through the control valve, so when it mixes with the circulating water, the needed temperature is produced. The difference between variable flow and variable temperature is clearly seen when the load is light. With variable flow the differential temperature increases significantly when the load (and flow) goes down.

With constant flow and variable temperature, the differential temperature decreases with the load, but relatively little (from the design temperature to a drop down to zero). It is the large variation in the temperature drop that causes the pronounced curvature when the flow is controlled.

On the other hand, the temperature drop will not change much when the temperature is controlled, so the curvature will therefore not be so pronounced. This is why the controllability is improved when secondary circulating pumps are used.

The above are strong arguments why circulating pumps should be used, especially on preheat coils. Rarely will it be justified to use secondary circulating pumps on terminal re-heat coils. Cooling coils may benefit from secondary circulating coils, but it should be noted that it also can cause problems, because even if the chilled water supply is cool enough to provide dehumidification, at partial loads it is mixed with recirculated water before entering the coil, so the potential for dehumidification is reduced.

### **3.6.1 Why a balancing valve is needed in the secondary circuit**

A balancing valve in the secondary circuit is needed to control the circulation so the design flow is accomplished. If the circulated flow is too small, it is obvious that the coil will not be able to provide the full heat emission.

It may seem like a paradox, but this is also the case if the circulated flow is larger than the design flow. The reason for this is that although the control valve is fully open, the flow will be mixed by the water that is recirculated. The temperature of the water entering the coil will be less and consequently the heat emission will be less than the rated value. Only when the circulated (secondary) flow is equal to the rated flow, will the flow in the by-pass line be zero, and no mixing or diverting will take place. The rated

flow and temperature will reach the coil so the rated heat emission is accomplished.

### **3.7 To summarize:**

On/off control is wasteful and should be avoided.

The hydronic system needs to be balanced, so the control valves can be sized properly, which makes it possible to use a small proportional band. This reduces the need to use a higher set-point in order to compensate for the proportional offset.

The performance of advanced control modes, such as PI- and PID-control, will also greatly benefit from balancing and properly sized control valves. No control function can compensate for oversized valves, poor rangeability, actuators with poor resolution or an unbalanced hydronic system.

How can hydronic balancing improve stability and reduce the proportional band?

If we first study the three-way valves, we will find that the balancing valve that is installed in the primary circuit, in series with a three-way control valve, has two functions :

- It controls the primary flow to the desired value, so over-flow is prevented.
- It adjusts the differential pressure across the valve, so it has to be operated over the full stroke to vary the heat emission 0 - 100%

If, for example, a 50% open valve would give the full heat emission, twice as large a proportional band would be needed to get stability.

Also two-way valves need balancing valves to adjust the maximum flow to the desired value.

The greatest stability problem exists when the control valve is almost closed, and the pressure drop across the balancing valve is almost zero. How can



the balancing be of any help to improve the stability? The answer is that by limiting the maximum flow through the different parts of the system, over-flows are eliminated and the hydronic system becomes predictable, so a basis for a correct sizing of the control valves can be established.

If balancing valves are not used, the neighboring (over-sized) control valve can "steal" the flow, if it opens fully. The result is that two or more systems can interact, and continuously disturb each other.

Obviously, balancing valves contribute greatly to improving the possibility for stable control.

Can zone valves or thermostatic radiator valves eliminate the need for balancing?

If the terminals (radiators, convectors or coils ) are equipped with modulating zone valves or thermostatic valves, the flow will automatically be adjusted to whatever is needed to satisfy the load in the room. At first glance it seems as if balancing valves will not serve any purpose in this application, because the flow is controlled anyway.

However, in most installations energy savings are made by reducing the supply of heat during night time (residential buildings) or when unoccupied (commercial buildings).

This results in the indoor temperature dropping, and all the valves open fully. The system will be completely out of balance. The temperature in the rooms served by the favored circuits will drop only slightly, because the reduction in the supply water temperature will be compensated by an increase in the flow, so the expected energy savings will not be realized. The rooms served by the least favored circuits will drop in temperature drastically.

In the morning, when the building is brought back to the normal temperature, the rooms served by the favored circuits will increase in temperature first.

The rooms served by the least favored circuits will not begin to increase in temperature until the control valves in the favored circuit begin to close, and stop "stealing" the flow. In an electromechanical program, the clock will have to be adjusted for a very early start time in order to recover the temperature in the whole building. If a micro-processor based start/stop time optimizer is used, all the adjustments have to be compromised, and the potential for energy savings will not be realized.

On the other hand, if the hydronic system is balanced, and the supply water temperature is reduced, all parts of the building will drop in temperature by the same amount, and the temperature will recover at the same rate in the different parts of the building. This makes it possible to set an electromechanical program clock for a relatively late start time, and an optimizer can be adjusted very aggressively, so the maximum energy savings are realized, still providing the desired temperature in the morning or at the beginning of the occupied period.

The above is equally true for large air handlers. When the system is off, the control valves open fully. Without balancing valves, the distribution of the flow will be off, causing problems similar to those described above. Also, without balancing there will be an undesirable interaction between air-handlers. A valve that suddenly opens in one air-handler can "steal" the flow from the next air-handler, so a disturbance that starts in one place is communicated to the other units. Balancing valves limit the maximum flow, and therefore also the interaction between the different parts of a HVAC system.