6.
How to balance

6.1 The purpose of balancing

When the flow through a valve is changed, either manually or automatically, the flows in adjoining branches are affected since there is a change in the pressure of all branches.

The purpose of the balancing valve is to limit the maximum flow in each branch, riser or main to the flow required by the terminal design. It is not always possible to select the exact flow through the size of the control valve so that the exact flow needed is obtained. Therefore, the balancing valve provides the fine tuning required.

There are four principal methods to balance hydronic systems:

1. Temperature drop method
2. Preset valve method
3. Reverse return method
4. Proportional method

6.1.1 Designing the system

- Divide the system into headers, risers, branches and units (coils, offices, apartments, etc.).
- Size pipework, using normal ASHRAE design procedures (see ref: B-3).
- List the flow rates of all units, branches, risers and headers on drawings.
- Provide properly sized Armstrong CBV balancing valves (see paragraph 6.2.1) in the return piping of all headers, risers, branches and units.
- Size the Armstrong pump based on total flow and pressure drop requirements (Ft Hd).
- Specify that the piping contractor and pump manufacturer reduce pump impeller diameter to meet "as-balanced" conditions.
- Specify that the proportional balancing method shall be used in the balancing procedure.
- Finally, and very importantly, specify that the system must be balanced by a qualified balancing contractor, require a certified report to verify the balancing result, and obtain approval by the Designing Engineer or Commissioning Agent.

6.1.2 Preparation for balancing

- Study the piping drawings and identify headers, risers, branches and units. Also, check that waterflows are specified on the drawings for all Armstrong CBV balancing valves.
- Remove and clean all strainers for full flow conditions.
- Open all isolating and automatic control valves providing full flow through the coils. Put all balancing valves in fully open or preset position. If a balancing valve does not produce any pressure drop when fully open, close that valve until a one foot pressure drop is recorded.
- Vent air from the system before balancing.
- If two-way or three-way control valves are used in the system, check to see if an adjustable balancing valve is installed in the return line and for three-way valves also in the by-pass line, to ensure proper balancing of the control valve.
- Check pump operation for proper rotation. Record pressure gauge readings, flow and motor current

draw when main balancing valve is fully open. If necessary, adjust main balancing valve to a position where the pump motor is not overloaded.

Equipment required:

a. One or preferably two Armstrong meters designed for reading the pressure drop across the balancing valves
b. Armstrong CBV Flow/Pressure Curves or Circular Slide Rule
c. For larger systems, the actual balancing should be carried out by a team of at least two people, using a walkie-talkie type communication system

6.2 Valve Coefficient Cv and Kv

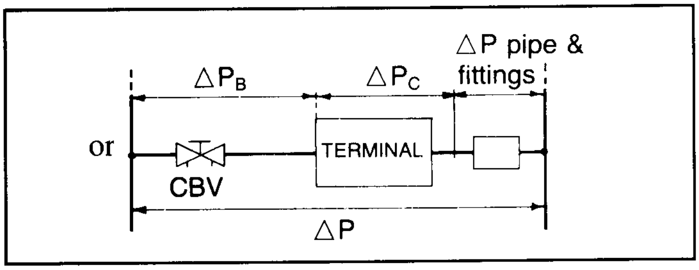


Fig. 28. Terminal pressure components

A balancing valve creates a supplementary hydraulic resistance in a hydronic circuit to limit water flow to the calculated value. Each passive element in a hydraulic circuit creates a pressure drop according to the following general equation.

$$\Delta P = R \times G^n \times d$$

R = Hydraulic resistance of this element
d = Density of the liquid, with d = 1 for water at 60°F
G = The flow of liquid in USGPM

For turbulent flows, the coefficient "n" is generally assumed to be 2, although for steel pipes "n" is generally accepted as 1.85

For a valve, n = 2 and $\text{USGPM} = \sqrt{\dfrac{\Delta P(\text{PSI})}{(R \times d)}}$

If we replace $\dfrac{1}{\sqrt{R}}$ by a coefficient Cv, we obtain the formula

$$\text{USGPM} = \text{Cv}\sqrt{\frac{\Delta P(\text{PSI})}{d}}$$

for glycol solution density "d" must be used; see Engineering Data and ASHRAE data (ref: B-3)

$\text{USGPM} = \text{Cv}\sqrt{\Delta P(\text{PSI})}$ assuming d = 1 for water

$\text{USGPM} = 0.66\ \text{Cv}\sqrt{\Delta H(\text{Ft})}$ pressure drop in feet of water

The Cv coefficient is a characteristic of the valve which depends mainly on its valve port area. The maximum Cv value is obtained when the valve is fully open and corresponds to flow in USGPM of water at a 1 psi (ΔP) drop across the valve.

The maximum Kv value is obtained for the valve fully open and corresponds to flow in m^3/h of water at a (ΔP) pressure drop of 1 bar across the valve.

$$\text{Kv} = \sqrt{\frac{m^3/h}{\Delta P(\text{bar})}}$$

6.2.1 Balancing valve sizing

Balancing valves must be sized to the flow they are going to control. Most balancing valves are selected by the size of the pipe line in which they are being installed. This can be undesirable as the pipe may be oversized, allowing less friction loss and reducing pumping cost. If in this case a balancing valve of the pipe size was installed, the pressure drop across the valve may be so low that it cannot be read until the valve is between half closed and almost closed.

Using oversized valves makes it possible, with a pressure differential meter attached, to obtain a very small or no pressure drop reading. The balancing contractor may close the valve 50% or so, with no change in the meter reading. This is because there is not sufficient flow through the valve to produce a detectable pressure drop.

It is important to size the balancing valves to the flow they are to control. This often means a valve smaller than the line size.

Minimum flow requires:
1. A one foot pressure drop across the valve to obtain an accurate meter reading;
2. The balancing valve be from 50% to 100% open at the set point for greatest accuracy.

Maximum flow is determined by:
1. The highest velocity acceptable
2. The largest allowable pressure drop

Flow through the pipe
It is recommended in a hydronic heating or cooling system that the pipe be sized on the basis of air and noise control. Air in the system causes noise and it is therefore important that it be removed. For pipes 2" and under, minimum flow velocities should be maintained around 2 Ft/sec, and for pipes over 2" velocities corresponding to a pressure drop of 0.75 Ft/100 Ft are accepted for proper air separation.

Water velocity noise on the other hand occurs when flow exceeds certain limits. For pipe sizes 2" and under 4 Ft/sec, and pipes over 2" in diameter, velocities corresponding to 4 Ft/100 Ft of pipe are recommended. From experience many designers use 7 Ft/sec in smaller pipe sizes and 4 Ft per 100 Ft in the larger pipe (ref: B-3, Chapt. 33).

By utilizing the pipe sizing charts and the balancing valve Flow/Pressure Drop Curves, selecting the proper balancing valve becomes more logical.

Example A:
Flow required (schedule 40 on page 52) is 4 USGPM. The pipe may be 1/2" or 3/4" and since we will have proper air separation and less friction loss, we would select 3/4" pipe. The Flow/Pressure chart indicates that 4 USGPM through a CBV-3/4T balancing valve would have a ΔP of 1.75 Ft fully open or 17 Ft when it is 50% open and would be the best selection.

Example B:
Flow required is 3/4 USGPM and 3/4" type L copper tubing is used. A CBV-3/4S sweat valve would not produce an adequate ΔP of 1 Ft even if it were closed 50%. Selecting the CBV-1/2S in this situation would be the correct selection and we would have a ΔP of 1.6 Ft with the valve 50% open.

The table on page 53 lists Cv(Kv) values and suggested flow limits of the Armstrong CBV Balancing Valves.

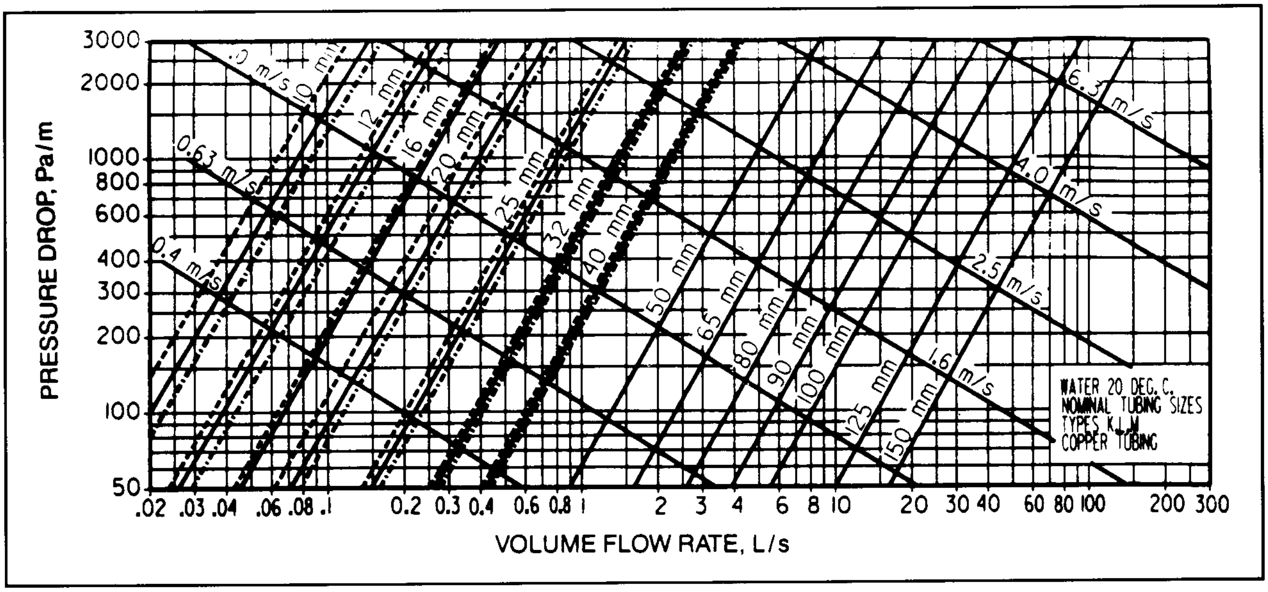


Fig. 29. Friction loss for water in commercial steel pipe (schedule 40)

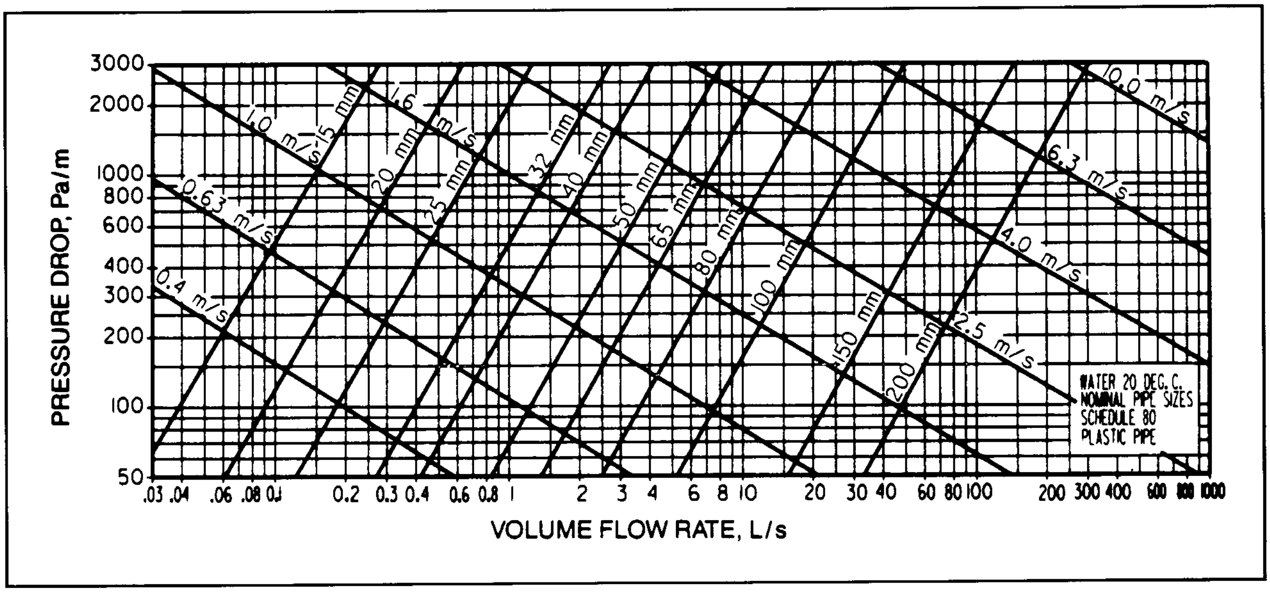


Fig. 30. Friction loss for water in copper tubing (Types K, L, M)

Valve sizing chart

Model	Pipe Size Inches	Fully Open		Minimum Flow @ 1 Ft Pressure Drop		Maximum Recommended Flow		
		C_v	K_v	50% Open GPM	100 % * Open GPM	GPM	Pressure Drop Feet	Velocity Ft/Sec
CBV-1/2 CS	1/2	2.7	3.2	1.3	1.8	6.0	11	6.0
CBV-3/4 CS	3/4	3.7	4.3	1.8	2.3	12.0	20	7.0
CBV-1/2 S or T	1/2	3.3	3.8	.66	2.2	6.0	7.2	6.0
CBV-3/4 S or T	3/4	4.9	5.9	1.2	3.3	12.0	13.5	7.0
CBV-1 S or T	1	14	16	2.6	8.2	18.0	4.5	7.0
CBV-1-1/4 S or T	1-1/4	17	20	3.6	11.0	32.0	9.1	7.0
CBV-1-1/2 S or T	1-1/2	24	28	6.8	16.0	45.0	8.0	7.0
CBV-2 S or T	2	37	43	11.5	24.0	72.0	9.5	7.0
CBV-2-1/2 F	2-1/2	69	59	18.0	46.0	100.0	4.7	7.0
CBV-3 F	3	97	83	19.0	64.0	160.0	6.5	7.0
CBV-4 F	4	255	219	46.0	180.0	275.0	2.8	7.0
CBV-5 F	5	324	279	60.0	220.0	500.0	5.6	7.8
CBV-6 F	6	448	385	100.0	300.0	800.0	7.9	8.8

* If the most resistant circuit in a branch is known it is most accurate to select a balancing valve that would produce one foot of pressure drop when it is fully open.

6.3 The temperature drop balancing method

For all terminals, the relation between the emission and the water flow is:

$$Q = 500 \times USGPM \times \Delta T$$

$$BTU/Hr = 500 \times USGPM \times \Delta T \text{ (water at 40°F)}$$

$$BTU/Hr = 487 \times USGPM \times \Delta T \text{ (water at 170°F)}$$

Between these two formulas, the difference is only 2.6%; to simplify, use the formula:

$$BTU/Hr = 500 \times USGPM \times \Delta T$$

This formula simply establishes that, for the nominal flow, the temperature drop is proportional to the coil emission. When the water flow is changed, the emission and the WTD (ΔT) are also changing. However, for a given emission the ΔT must be calculated for the correct water flow.

Example:
A coil gives a nominal output of Qs = 5000 BTU/Hr in the following design conditions:

EWTs = 200°F
WTDs = 20°F
EATs = 60°F and 154 CFM (LATs = 90°F)
USGPMs = 0.5
("s" is equal to design conditions)

When adjusting the correct water flow, nominal design conditions are not always available. EWT, WTD, and EAT can be measured and the actual emission can be found in the coil's data. If this data is not available, a rough estimation can be made with the following formula, using the above data:

$$Q = Qs\left(\frac{[EST-LAT]\ [EWT-WTD-EAT]}{[EWTs-LATs]\ [EWTs-WTDs-EATs]}\right)^{0.5}$$

(For a radiator, replace LAT and EAT by room temperature (RT) and use 0.67 power instead of 0.5)
Now adjust the water flow with a balancing valve to obtain:

$$WTD = \Delta T = \frac{Q}{500 \times USGPM}$$

$$= \frac{3000}{500 \times USGPM} = \frac{3000}{500 \times 0.5} = 12\text{ F}$$

Another method to estimate the ΔT is to plot the water temperature conditions versus outside air as shown and approximate the ΔT at the actual outside conditions.

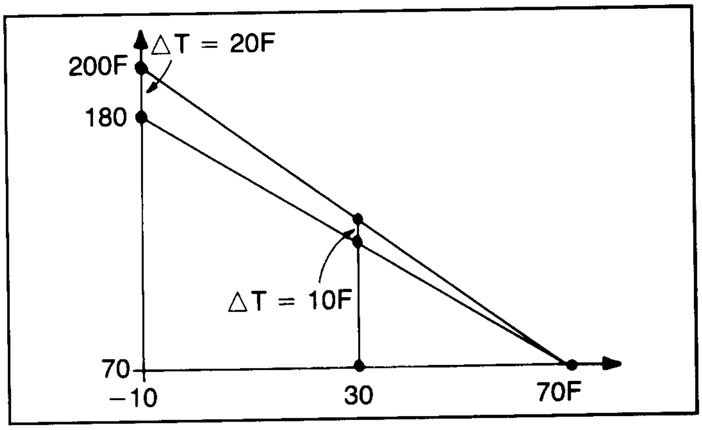


Fig. 31. Approximation of ΔT

(Note: At 70°F OA no heating should occur and therefore water supply should be 70°F or less.)

Design temperature:
Outside = 10°F
Room = 70°F
Temp. Drop ($\Delta T1$) = 20°F
Design Temp. Rise = -10°to 70° = 80°F
Actual Temp. Rise = 30°to 70°= 40°F

$$T2 = \frac{T1 \times \text{Actual Temp. Rise}}{\text{Design Temp. Rise}}$$

Unfortunately, when the flow is changed the emission changes. Normally the new emission must be calculated and the new $\Delta T2$ adjusted, but this second operation is generally missed.

If all other coils are working under the same conditions, the flow of their coils must be adjusted to obtain a $\Delta T = 12°F$.

The advantage of the temperature method is that the water flows are not measured, but are adjusted by controlling the water temperature drop (WTD).

(This method is *very rough* but is better than no balancing at all.)

The quality of this method depends on the precision of the temperature measurement, especially when using surface mounted thermometers. In cooling, differences in water temperature become so low that they converge on the normal limits of accuracy of measurement.

Another difficulty in cooling is the estimation of the coil consumption which depends on the latent load.

The temperature method is only recommended for small heating systems.

In large systems, if flow corrections are substantial, it is highly likely that after adjusting the last coil, the differential pressure applied to the first coil will increase considerably. The procedure must then be applied several times to all the terminals unless the differential pressures are stabilized. It is suggested that the proportional method be used in place of the temperature drop method.

6.4 Preset method

The preset valve method is a calculated method to determine the balancing valve setting before the system is installed, based upon the design flow USGPM and an estimate of the pressure drop.

The difficulty is to carefully calculate the balancing valve pressure drop ΔHb on each branch of the system.

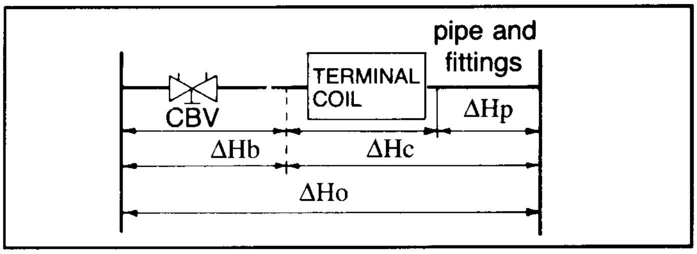


Fig. 32. Terminal preset condition

For its nominal flow USGPM a coil has a pressure drop ΔHc. The riser differential pressure is Ho. The difference $\Delta Hb = \Delta Ho - \Delta Hc - \Delta Hp$ has to be taken in the balancing valve. The Cv value is given by:

$$Cv = \frac{USGPM}{0.66 \text{ x } \sqrt{\Delta Hb}}$$

Example:
$\Delta Ho = 30$ ft, $\Delta Hc = 14$ Ft, USGPM = 2,
therefore valve is adjusted for a Cv value of 0.76.

$$\text{Then } Cv = \frac{2}{0.66 \text{ x } \sqrt{30 - 14}} = 0.76$$

In theory, it seems very easy, but in practice we do not know the value of ΔHo.

Example:

As a rule of thumb ΔHo can be calculated as follows: assuming the pressure drop for the pipe and fittings is 3 Ft per 100 Ft length of pipe (L).

$$\Delta Ho = \text{Pump head} - 2L \cdot \frac{3}{100}$$

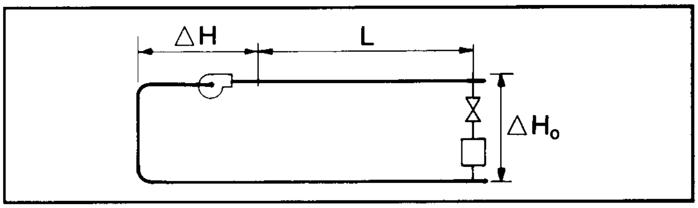


Fig. 33.

The pressure drop through the control valve, the coil, fittings and connecting pipe must be subtracted from the ΔHo to arrive at the ΔHb to determine the valve presetting.

A second way to determine the balancing valve setting is, once the pressure drop ΔHb of the valve is known and using the manufacturer's slide rule, set the USGPM and ΔHb together. Then for the valve size used, read the new setting under the indicator line.

If Flow/Pressure Drop curves are being used, draw a horizontal line across the chart at the calculated value of ΔHb, then draw a second but vertical line up from the designed USGPM. The intersection of these two lines will be at the valve setting required.

In theory it seems very easy, but in practice we do not know the value of ΔHo.

It is necessary to calculate the friction loss through each section of pipe, including fittings, and the ΔP for each component used for their respective design flows. Then for each unit the applicable pressure drops are added together and total ΔP or Ho for each unit is known.

The unit with the highest ΔP is the most resistant circuit in the system and becomes the reference unit to which all other units are compared. For all other circuits, if we take the ΔP of the reference unit and subtract the ΔP of a second circuit we would have the ΔP required to be taken by that second circuit balancing valve or its Hb.

By doing this for all circuits, we could then preset all balancing valves in the system.

In reality, the preset valve method is a preparatory step to balancing, as it does not take into consideration the system's "as-built" conditions.

6.5 Reverse return method

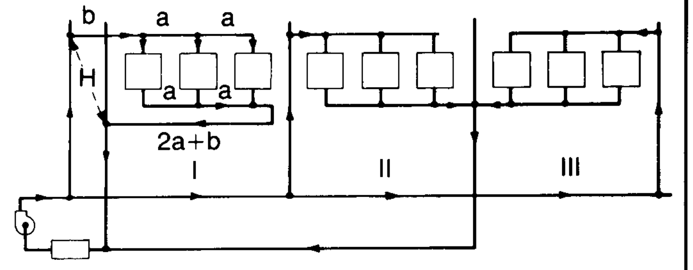


Fig. 34. Reverse return circuit

Figure 34 shows circuit I with three coils mounted with reverse return piping. If a and b represent the pressure drops in the pipes, we can easily see that each coil has the same differential pressure which is equal to H-(4a + 2b). This result is valid if all flows correspond to values calculated.

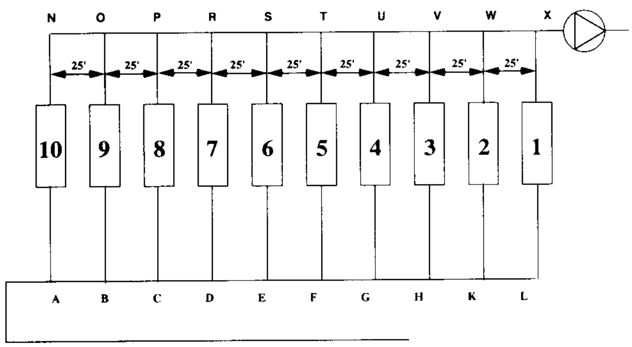
This solution is interesting if those coils need the same differential pressure. If this is not the case, the coils will not receive the calculated flows and balancing valves are needed with or without a reverse return circuit.

If the coils are identical, we have seen that the balancing problem is reduced in practice if the nominal pressure drop of each coil is large enough to reduce the influence of the pressure drops in the pipes.

Moreover, the reverse circuit does not solve the question of overflowing of "favored circuits," which is the most serious problem.

When balancing a section or a system using the equal distance flow for reverse return piping, the pipe sizes must be reduced as flow is diverted in order to maintain a constant friction loss or flow though the units will change.

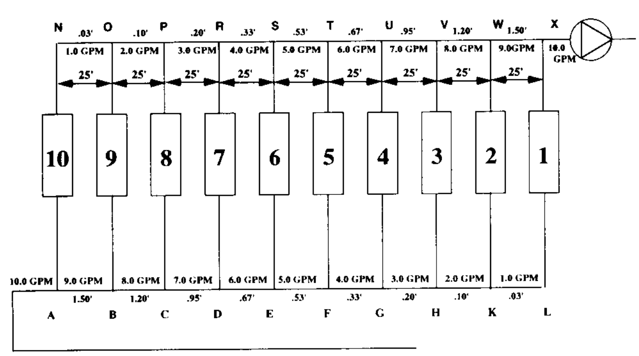
Example:
We take 10 equal terminal units (1 USGPM each) and make all piping connections the same using 1" copper pipe. Maintaining an equal distance between the supply and the return and an equal distance (25 Ft) between each unit (Figure 35) we will observe the following.



10 Equal Units 1.0 GPM Each
Total System 10 GPM 1' Copper Pipe

Fig. 35.

In figure 36 the USGPM flow is calculated and from this the pressure drop through each segment of supply and return pipe is given. In section A-B there is 9.0 USGPM and 1.50 ΔP, while in section H-K there is 2.0 USGPM and 0.10 ΔP.



10 Equal Units 1.0 GPM Each
Total System 10 GPM 1' Copper Pipe

Fig. 36.

UNITS	PATHWAY	ΔP
10	N,O,P,R,S,T,U,V,W & X	5.51'
9	A,B,/O,P,R,S,T,U,V,W & X	6.98'
8	A,B,C,/ P,R,S,T,U,V,W & X	8.08'
7	A.B.C.D,/ R,S,T,U,V,W & X	8.83'
6	A,B,C,D,E,/ S,T,U,V,W & X	9.17'
5	A,B,C,D,E,F,/ T,U,V,W & X	9.17'
4	A,B,C,D,E,F,G,/ U,V,W & X	8.83'
3	A,B,C,D,E,F,G,H,/ V,W & X	8.08'
2	A,B,C,D,E,F,G,H,K,/ W & X	6.98'
1	A,B,C,D,E,F,G,H,K &L	5.51'

Fig. 37.

In figure 37 the pressure drops affecting each terminal unit have been added together and we can observe that units 5 & 6 in the center have the largest ΔP, 9.17 Ft while units 1 & 10 have the lowest ΔP of 5.51. It can also be seen that the ΔP is equal for pairs of units starting from the center and working to the ends 5 & 6, 4 & 7, etc.

Since we know that in actual use this system would always have an equal pressure drop from point A to point X, regardless of the path the water chooses to go, the difference would be in the amount of flow through each unit. Units 1 and 10 would have the most flow while units 5 and 6 would have the least flow.

It is important to understand the limitations of the reverse return pumping systems. If the terminal units are close together and piping is sized for equal friction loss to flow, adequate balancing may be achieved for a section or branch of a system. In most cases, however, it is easier, less costly and more accurate to use a direct return piping system and balancing valves to obtain the correct flow.

It is a costly solution to install a third pipe to create pressure drops, not only because of the installation costs but also because we create useless pressure drops. The supplementary pipe increases the piping heat losses.

However, we can obtain a reverse return circuit without supplementary pipe.

6.6 The proportional balancing method

6.6.1 Introduction

When several terminals of a branch are piped in parallel, there is a distribution of the total water flow which depends on the hydraulic resistance of each circuit.

The following fig. 38 shows this:

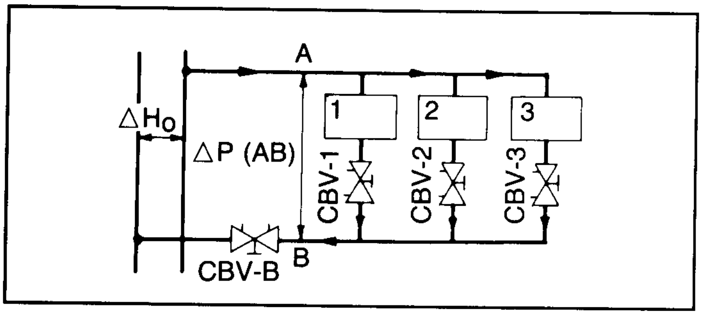


Fig. 38. Branch

If the setting of valve CBV-B is changed, the differential pressure ΔP (AB) changes for all the terminals. If ΔP is increased 4 times, all the flows in the circuits are increased by 2 since:

$$\frac{H1}{H2} = \left(\frac{USGPM1}{USGPM2}\right)^2$$

$$USGPM2 = \sqrt{\frac{H2}{H2}} \times USGPM1$$

$$= \sqrt{4} \times USGPM1 = 2 \times USGPM1$$

This means that all external influences change the terminal water flows *IN THE SAME PROPORTION*, as long as the setting of the valves CBV-1, CBV-2 and CBV-3 is not changed.

This principle can be used to balance all terminals in a branch and is fundamental to the proportional method.

In reality, the typical branch may consist of a supply and return riser with unequal supply and return lengths as shown in Figure 39.

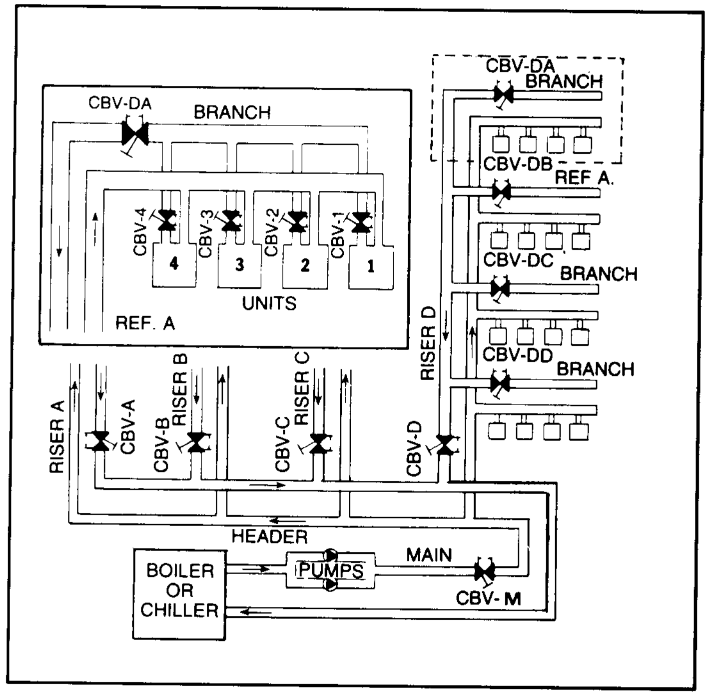


Fig. 39. Typical hydronic system

The procedure will use a reference balancing valve CBV-1 with balancing valves CBV-2, CBV-3 and CBV-4 to be adjusted respectively. This means that each valve that is adjusted is an external force to all the valves that have previously been adjusted proportionately. For example, when CBV-3 is decreased, the proportional relationship between CBV-1 and CBV-2 remain identical

6.6.2 Balancing procedure

Preparation (see paragraph 6.1.2)

1. Start by setting all risers or zones at approximate USGPM flow per plans and specification using a differential meter or Armstrong's COMPUFLO meter for direct calculations.

2. Select any riser or zone. Open that riser or zone CBV balancing valve to the wide open position, thereby obtaining overflow.

Branch selection:

3. With the COMPUFLO meter, check the flow of each branch CBV balancing valve (see Figure 39) and calculate the proportion Ø (or quotient) or the actual (measured) flow USGPM (L/S) divided by the design (specified) flow USGPM(L/S). This can be done readily:

$$\text{Branch } \varnothing = \frac{\text{Flow Actual}}{\text{Flow Design}}$$

4. From this survey select the branch that has the highest flow proportion Ø. Balancing of this branch first, then the next highest, etc. will make water flow available for the other branches. (No need to adjust branch balancing valves at this time.)

5. Data organization is important, and a report form should be used to record each CBV and to keep this data organized.

Balancing of the terminals of one branch:

6. Measure the flow of each terminal CBV balancing valve in a branch; for example units 1, 2, 3, and 4. (See Figure 40)

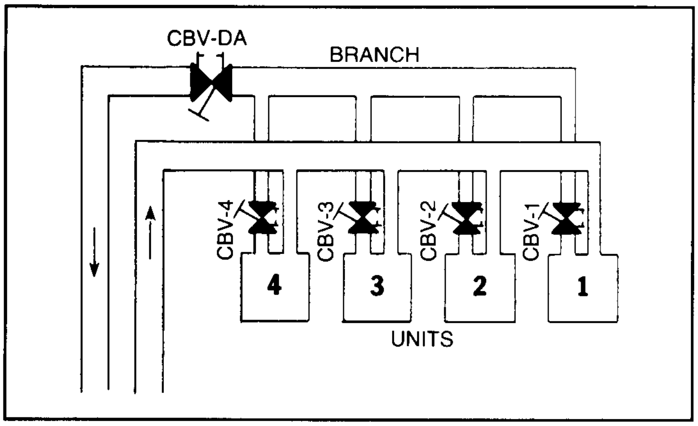


Fig. 40. Branch and terminals example

7. Calculate flow proportion "Ø " for each unit.

$$\text{Unit } \varnothing = \frac{\text{Flow Actual}}{\text{Flow Design}}$$

8. Start by selecting the unit with the lowest proportion (Ø) value. Adjust the handwheel of the last unit CBV-1 to approx. 95% of this proportion. Lock the handwheel memory. The last unit (CBV-1) will now become the "reference unit" for this branch. The setting of this valve must not be changed. If two COMPUFLO meters are available, we suggest one be left connected to the reference unit in order to continually monitor its flow proportion.

9. Now adjust CBV-2 (unit 2) to the same proportion as the reference unit CBV-1. The proportion in unit CBV-1 might increase slightly. Adjust CBV-2 until the same proportion is on both units. Lock the handwheel memory on CBV-2.

10. Move upstream to adjust CBV-3 and then CBV-4 in the same way, using CBV-1 as the reference unit, until the branch or zone is completed.

11. The flow proportion of CBV-1 will continue to increase while adjusting the upstream units CBV 2, 3 and 4. This is normal and each upstream unit has to be balanced to the actual value of CBV-1. Keep in mind that once a valve has been proportioned to the actual Ø value of CBV-1 their values will remain identical even though the adjusting of a third valve may cause that value to change.

12. Continue the same procedure by adjusting the terminals on other branches on the same riser or zone, taking the next highest proportion (from the initial survey) until all the terminal CBV valves are adjusted and locked.

Balancing the branches of one riser(Figure 39):

13. The next step is to balance each branch on this riser using the same method. First, based on information from the survey completed in step 3 Branch Selection, select the branch with the lowest flow proportion Ø.

$$\text{Branch } \varnothing = \frac{\text{Flow Actual}}{\text{Flow Design}}$$

14. Make a record of this.

15. Adjust the last branch CBV-DA balancing valve (Figure 39, Riser D) to approximately 95% of the lowest flow proportion (Ø) and use it as a reference unit . Lock its handwheel memory. Connect the COMPUFLO meter to CBV-DA to continually measure its flow proportion.

16. Now adjust the next upstream branch CBV-DB to the same flow proportion Ø as the reference unit CBV-DA and lock its handwheel memory.

17. Gradually balance all the rest of the branches on this Riser D, working upstream in sequence. Adjust CBV-DC and CBV-DD to the actual reference flow proportion Ø of CBV-DA and lock their handwheels. The flow proportion will be increasing.

18. With the COMPUFLO meter still on CBV-DA, adjust riser CBV-D to a proportion Ø of 1.0 at CBV-DA.

Balancing the branches on the other zones or risers (Fig. 39):

19. Continue to balance each riser or zone upstream according to the same procedure just outlined.

20. After all the branch CBV valves are set and locked, move on to the riser balancing valves.

Balancing of the risers (Fig. 39):

$$\text{Riser } \varnothing = \frac{\text{Flow Actual}}{\text{Flow Design}}$$

21. Measure each riser balancing valve CBV with COMPUFLO meter to determine the riser with the lowest flow proportion Ø:
22. Make a record of this on the form.

23. Adjust the last riser CBV-A to the lowest flow proportion(Ø) and use it as the reference unit. Lock its handwheel memory. Connect the COMPUFLO meter to this CBV-A in order to continually measure its flow proportion.

24. Now adjust the next upstream riser CBV-B to the same flow proportion (Ø). Lock its handwheel memory.

25. Gradually balance all the upstream riser CBV balancing valves in sequence to the actual flow proportion Ø of CBV-A and lock their handwheel memory. The riser balancing valves are now set.

26. Now adjust the supply main CBV-M balancing valve to bring the riser-zone reference CBV-A valve to flow proportion (Ø) = 1.0. This means that the flow USGPM (L/S) in each riser-zone is a design condition. Since the riser CBV valves were set in proportion, the design condition is now flowing in each loop.

27. Move the COMPUFLO meter to the main CBV-M balancing valve to measure the total flow. Check this against the pump curve and system's design specification. If the main CBV-M balancing valve setting has been reduced substantially, then there may be an opportunity to reduce operating costs by reducing impeller size. (See section 6.8)

6.7 Optimum pump evaluation

An example of optimizing a hydronic pumping system is shown in Figure 41. The original design called for a theoretical flow of 1200 USGPM at 110 Ft Hd. This is represented by point A on the pump head curve. When installed, it was found that the pump was actually operating at a flow of 1400 USGPM at 99 Ft Hd, represented by point B.

Since the as-built system has less resistance than the actual design, 99 Ft versus 110 Ft, the operating point B moved out on the pump curve at an increase of brake HP from 40.7 HP to 43.8 HP.

Balancing of the system was then undertaken to provide the design flow (USGPM) in each circuit. This was completed and the operating point moved from point B to A with a design flow of 1200 USGPM and 110 Ft Hd.

Consideration was given to reducing the size of the pump impeller, since the pump could operate at a lower horsepower and still provide the design flow (USGPM) at a lower resulting head.

Point C is the point on the system curve with a reduction of head to 73 Ft and a reduction of brake horsepower to 30.3 HP.

Point C was found by first determining the actual flow through the main supply CBV-M (Figure 41) at point B along with pump differential head of 99 Ft, and then calculating the new impeller size with the help of the affinity laws for pumps.

At constant speed:

$$\frac{\text{Diameter 1}}{\text{Diameter 2}} = \frac{\text{USGPM 1}}{\text{USGPM 2}} = \frac{\sqrt{H1}}{\sqrt{H2}} = \frac{\sqrt[3]{HP1}}{\sqrt[3]{HP2}}$$

$$\text{Diam 2} = (\text{Diam 1})\frac{\text{USGPM 2}}{\text{USGPM 1}} = (11.0) \times \frac{(1200)}{(1400)}$$

$= (11.0) \times (0.857) = 9.427$ in. new impeller diameter

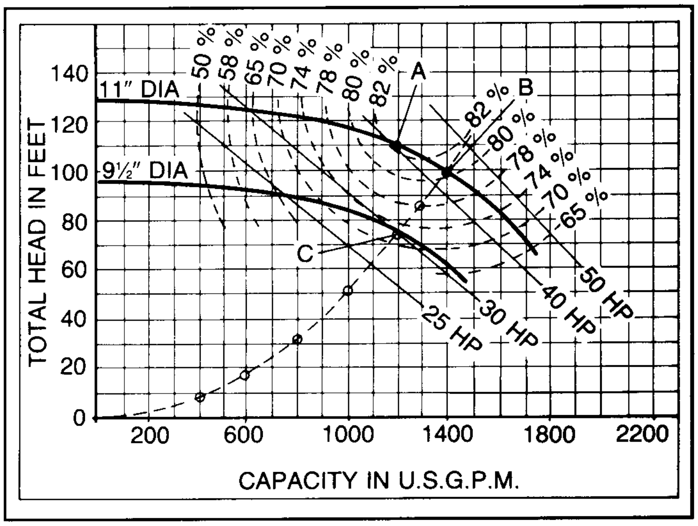


Fig. 41. Pump head and system curve

The diameter is rounded off to 9.5". As a result of the balancing, we are at point A on the original impeller curve and a new system curve through this point.

When the new impeller is installed and the main balancing valve CBV-M is readjusted to a wide open position, the operating point A shifts to point C which is 1200 GPM and 73 Ft Hd.

Figure 42 shows the result of pump optimization. At 10¢ per KWH, balancing alone reduced the original design operating cost from $28,645 per year to $26,630, a reduction of $2,015 per year (7%). Balancing and pump impeller trimming reduced original design operating cost from $28,847 per year to $19,798, or a reduction of $8,847 per year (30.9%).

The CBV balancing valve is not just a tool for accurately setting the flow to design conditions, but accurately measures the flow so that the pump size can be evaluated based on actual installed running conditions.

6.8 Variable speed drives

An electronic variable speed drive accepts the electrical supply from the utility. It first converts the supply to direct current, then inverts it to an alternating current of the required frequency and voltage to produce the desired speed of the pump motor. The frequency can either be adjusted manually at the drive, or by an external signal from a pressure or flow controller.

If the speed of a pump is varied, we will find that the flow and the head will vary. The flow increases in proportion to the speed, but the head increases by the power of two. This means that the power consumption of the motor increases very significantly, by the cube or by the power of three. If the speed is doubled, the flow will double, the head will have a four fold increase and the power consumption will increase eight times. Conversely, a reduction in speed (and flow) to 50%, will lower the energy consumption to 0.5 (cubed) x 100% =12.5%. It is obvious that a variable speed drive has the potential for remarkable energy savings, but it is best to analyze the situation carefully.

It is possible to realize large savings in the pump energy if a variable speed drive is applied to a hydronic system that is not yet balanced. Unfortunately, this is not a true saving, because the savings are counted against a flow that, due to the a lack of balancing, is artificially high. If the system is first balanced, and the excessive flows in the favored circuits are brought into control, the total flow is reduced. Now we can realize significant savings simply by trimming the impeller, and allow the pump to run at constant speed.

It is possible that the savings are already so large that the installation of a variable speed drive no longer is justified. When analyzing the two alternatives we should consider the following.

Alternating pumps are usually used for the main circulation; therefore we have to consider the expense of trimming two impellers, instead of only

COMPARISON OF ELECTRICAL SAVINGS THROUGH SYSTEM BALANCING AND TRIMMING OF IMPELLER

	Designed and System Balanced A	Actual Unbalanced B	Designed, System Balanced and Impeller Trimmed C
GPM	1200	1400	1200
FT. HD.	110	99	73
PUMP EFF.	82 %	80 %	73 %
IMP. OD	11"	11"	9½
BHP	40.7	43.8	30.3
KW	30.4	32.7	22.6
COST/KWH	10c	10c	10c
OPERATING COST/DAY	$72.96	$78.48	$54.24
YEAR	$26,630.	$28,645.	$19,798.
SAVINGS:			
Balanced vs. Unbalanced:			$2,015.00 per year
Impeller Trimmed and System Balanced:			$8,847.00 per year

Fig. 42. Balancing and impeller reduction optimization technique results

one. On the other hand, the same variable speed drive can be used for the two alternating pumps. To further reduce the costs, instead of sizing the variable speed drive after the power rating of the pumps, size the drive after the actual power requirement, based upon the total flow when the system is balanced.

The simplest solution is to set the speed at a constant value, so the desired flow is accomplished. A more advanced solution is to install a differential pressure transmitter between the supply and return at the end of the system. The signal from the transmitter is connected to the controller, which varies the in-signal to the variable speed drive, so a constant differential pressure is accomplished at the furthest point in the system. This will improve the working conditions for the control valves, so it will be easier to accomplish a stable control.

Additional energy savings in the operation of the pumps will certainly be realized, but it is very hard to predict how great these savings will be. The reason is that if the supply water temperature is reset the flow will depend upon how well the reset curve corresponds to the load. The expected flow reduction will not take place, because the control valves at the air handlers and other points will have to open more, if the supply water temperature is reduced as the outdoor temperature increases.

Before making a decision to use variable speed valves, several steps should be taken. Examine the hydronic system, and balance it properly, using only good quality balancing valves.

Size the control valves correctly, and reset the supply water temperature. Once these steps are completed and the result is evaluated, variable speed drives should be considered if further enhancement is necessary.