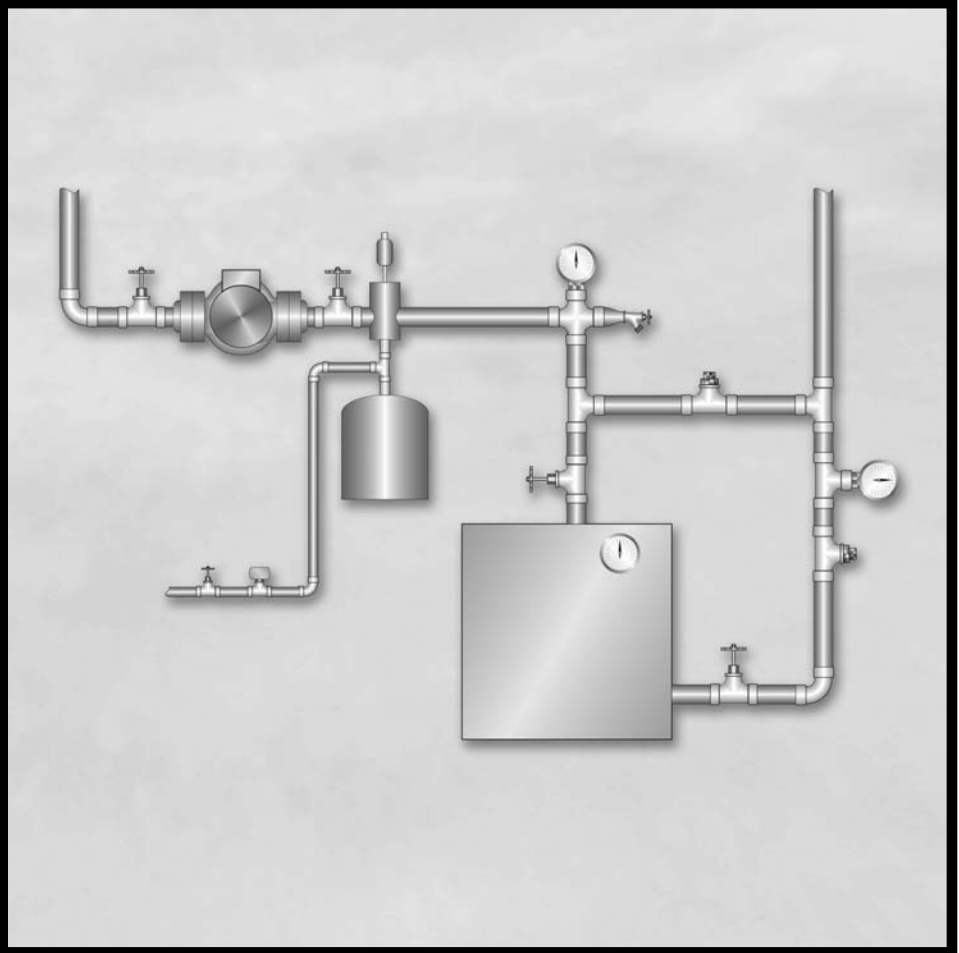


# **Water Survey**

## ***Hydronic Systems***



### **Pre-Installation Survey and Checklist for Water Systems**



PeerlessBoilers.com

# INSTALLATION SURVEY – WATER

## A. INTRODUCTION

1. Use this survey to review the **system** for operating behavior and history. The survey should give you some tips on making the installation successful. It might also give you some recommendations to pass on to the owner for corrections needed in the system to avoid problems later.
2. The main points of this survey are efficiency definitions; performance and selection of centrifugal pumps; piping alternatives; air control in hydronic systems; and prevention of condensation in the boiler caused by low operating temperature.

## B. DEFINITIONS AND RATINGS

<b>Boiler Horsepower</b>	33,475 Btu per Hour Output
<b>BHP (Brake Horsepower)</b>	33,000 foot-pounds/hour 746 watts
<b>MBH</b>	1000 Btu per Hour
<b>SCFH</b>	Standard Cubic Foot per Hour (at 60°F)
<b>Sensible Heat</b>	Heat required to raise the temperature (about 1 Btu per Pound per °F for water and 0.24 Btu per Pound per °F for air or flue gases)
<b>Latent Heat</b>	Heat required to vaporize (970 Btu per Pound for water at 212° F)

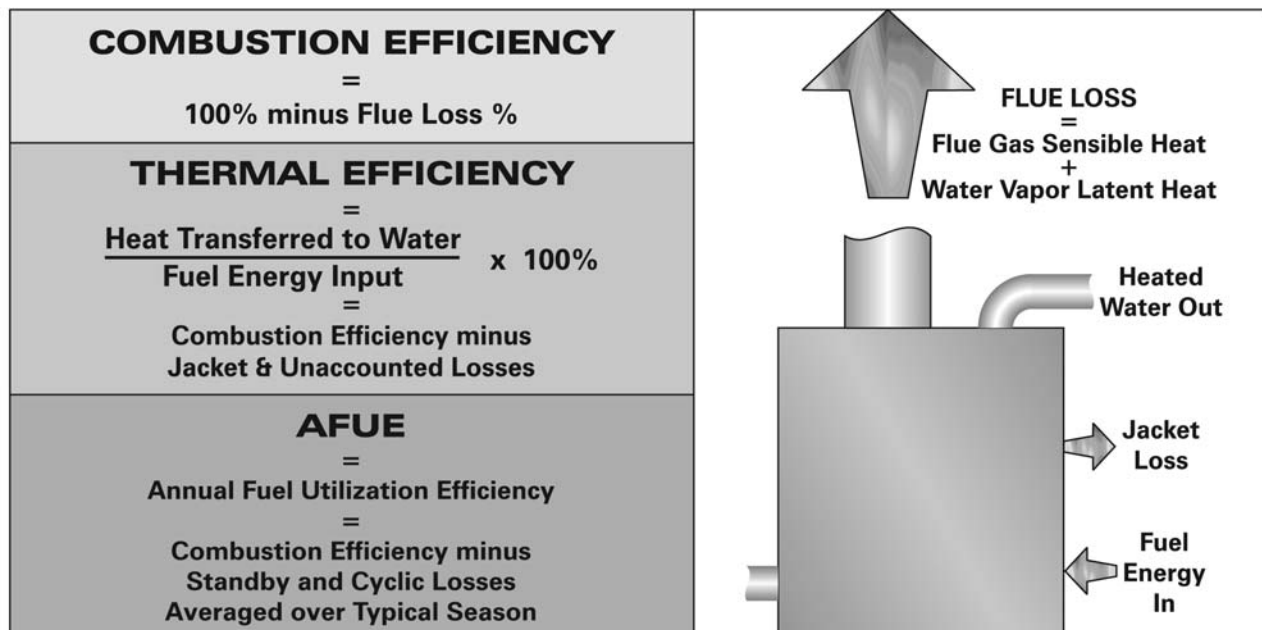
## C. EFFICIENCY AND BOILER OUTPUT

### 1. Combustion

- a) Fuels are made up mostly of **Carbon** and **Hydrogen**, the main elements in combustion. **Sulfur** content in the fuel is also important.
- b) In combustion, carbon and hydrogen combine with oxygen in the air to form **CO<sub>2</sub>** (carbon dioxide) and **H<sub>2</sub>O** (water). Small amounts of CO (carbon monoxide) will also form even in good combustion. The water is formed as vapor and contains a large amount of heat because of its heat of vaporization (latent heat).
- c) The sulfur in the fuel also burns to form SO<sub>2</sub> (sulfur dioxide) or SO<sub>3</sub> (sulfur trioxide).
- d) If the flame is chilled (such as by impingement on the boiler surfaces) or if there isn't enough air, **incomplete combustion** occurs. In incomplete combustion, some of the carbon may go to CO instead of CO<sub>2</sub> or may pass through as free carbon (soot).
- e) The amount of water vapor formed depends on the amount of hydrogen in the fuel. Natural gas has more hydrogen than oil, so the vapor content in the flue gases is higher (about 50% more).

### 2. Dewpoint

- a) When flue gases are cooled, they can reach a temperature too low to keep the water in vapor form. The water vapor will begin to condense.
- b) This temperature where condensation starts is called the dewpoint temperature, or just **dewpoint**. If the flue gas temperature always stays above the dewpoint, no condensation will occur. But if flue gases contact a surface colder than the dewpoint, water vapor condenses on the surface, just like water droplets on a cold glass.



- c) The water droplets formed from condensation alone are only mildly corrosive. But combustion air usually contains airborne contaminants which can cause acid formation with condensate. So don't let condensation occur.
- d) The water vapor dewpoint for natural gas flue products is about 130° F.
- e) **Prevent condensation in natural gas boilers by providing piping and controls to assure return water to the boiler at no lower than 130° F under all conditions.**
- f) Oil flue products contain less water vapor, so the water vapor dewpoint is lower than for natural gas. But the oil flue products also contain SO<sub>3</sub>. When water combines with SO<sub>3</sub> the result is sulfuric acid.  
  
The sulfuric acid has its own dewpoint. The higher the SO<sub>3</sub> content, the higher the dewpoint. For most boiler applications using #2 fuel oil, the acid dewpoint will be 150° F or slightly higher.
- g) Prevent acid condensation in oil fired boilers. The acid not only damages the boiler surfaces, it forms a sticky layer which attracts soot and accelerates flue blockage.
- h) **Prevent condensation in oil boilers by providing piping and controls to assure return water to the boiler at no lower than 150° F under all conditions.**

### 3. Vent Categories (Gas Boilers)

- a) Condensation occurs in the venting system if the venting system surfaces are colder than the vent gas dewpoint.
- b) A boiler rated “**condensing**” doesn't mean the boiler condenses. **It means the venting system will be wet at the completion of a typical boiler operating cycle.**
- c) Boilers are placed in four categories in accordance with ANSI Z21.13 test procedures:

#### ● CATEGORY I

- Category I boilers are **non-condensing** and **natural draft vented**.
- Standard B vent or masonry chimney can be used. Venting system sizing is typically determined using the National Fuel Gas Code Venting Tables.

#### ● CATEGORY II

- Category II boilers are **condensing** and **gravity vented**.
- Use only **special gas vent** designed for condensing operation and specified by the boiler manufacturer.

#### ● CATEGORY III

- Vent is pressurized.
- Category III boilers are **non-condensing** and require a **sealed vent system** as specified by the boiler manufacturer.

#### ● CATEGORY IV

- Vent is pressurized.

	Natural Draft Venting	Forced Draft Venting
Non-Condensing	<b>I</b> Suitable for use with standard venting, Type B or masonry.	<b>III</b> Use only Special Gas Vent specified in manufacturer's installation instructions.
Condensing	<b>II</b> Use only Special Gas Vent specified in manufacturer's installation instructions.	<b>IV</b> Use only Special Gas Vent specified in manufacturer's installation instructions.
<b>VENT CATEGORIES</b>		

–Category IV boilers are **condensing** and require a **sealed vent system** suitable for condensate handling as specified by the boiler manufacturer.

- **NOTE:** Use only special gas vent specified by the boiler manufacturer for Category II, III or IV boilers.

### 4. Combustion Air

- a) A lot of air is needed for combustion and draft regulators, such as draft hoods or barometric draft regulators.
  - For Category I and II gas boilers, allow 30 SCFH of air for each 1000 Btu/hr input. This includes allowance for draft hood operation.
  - For Category III and IV gas or oil boilers, allow 12.5 SCFH of air for each 1000 Btu/hr input (based on 9% CO<sub>2</sub> gas and 12.5% CO<sub>2</sub> oil).
  - For oil boilers equipped with a barometric draft regulators, add an additional 25% air for operation of the barometric draft regulators.

### 5. Excess air

- a) Excess air is the amount of combustion air used over the theoretical amount needed for complete combustion (zero excess air). All boilers operate with excess air because perfect combustion with no excess air is not achievable.
- b) To calculate the amount of excess air, measure the CO<sub>2</sub> percentage in the flue gases and apply the following formulas:
  - Natural gas (usually 8% to 10%):  
–Excess Air % = [9.194 + 1096/(%CO<sub>2</sub>)] – 100
  - #2 Fuel oil (usually 11% to 13%):  
–Excess Air % = [6.668 + 1434/(%CO<sub>2</sub>)] – 100
  - Liquefied Petroleum (LP/Propane) (usually 8% to 10%):  
–Excess Air % = [8.41 + 1260/(%CO<sub>2</sub>)] – 100

## 6. Combustion Efficiency (or Flue Loss Efficiency)

- Combustion Efficiency equals **100% minus the % Heat lost in the flue gases.**
- Flue gas heat energy includes the heat of vaporization (latent heat) of the water vapor formed from combustion plus the sensible heat in the flue gases compared to room temperature.

## 7. Thermal Efficiency

- Same as Fuel to Water Efficiency
- Equals the **measured heat gain of the water** divided by the **fuel energy input**. Thermal efficiency indicates the heat energy actually delivered to the water.
- The thermal efficiency is equal to the combustion efficiency minus the heat lost from the jacket and any unaccounted losses.

## 8. Steady-State Efficiency

- Equals the thermal efficiency of the boiler after running long enough for steady state conditions to develop.

## 9. Annual Fuel Utilization Efficiency – AFUE

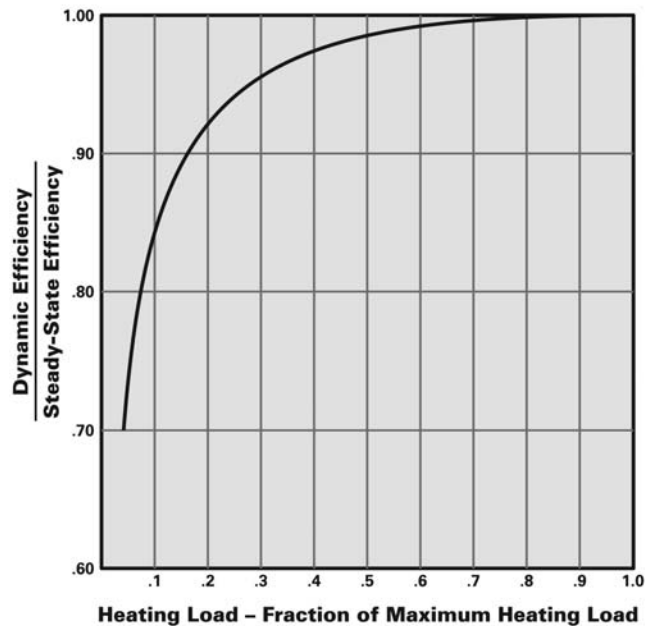
- Applicable only to boilers under 300 MBH input. The AFUE (or Seasonal Efficiency) is determined using the test procedure developed by the Department of Energy and is used to assure compliance with the National Appliance Energy Conservation Act of 1992.
- The AFUE takes into account the stand-by and cyclic losses of the boiler and averages the results over a typical heating season.
- There is only one definition of AFUE. There are no recognized alternative measures, such as “Adjusted AFUE”.

## 10. Part Load Efficiency

- Boiler part load efficiency (or Dynamic Efficiency) is the effective efficiency of the boiler when operating under cyclic conditions. There is no established test procedure to rate the part load efficiency of a boiler. But part load efficiency can be estimated based on the normal jacket and off-cycle flue losses. Part load efficiency will be higher as the percentage of on-time of the boiler increases.
- Figure 1 shows a typical part load (Dynamic) efficiency curve for a forced draft ON/OFF fired boiler. This curve is based on test data from Brookhaven National Laboratories on a residential oil-fired boiler.

The curve demonstrates that the effective efficiency for the boiler drops sharply as the load on the boiler reduces. This is due to the continuous loss of heat from the jacket and flue while the boiler is in stand-by between cycles. The lower the load—the longer the percentage of stand-by time—the lower the part load efficiency.

- A boiler with stage firing (HI/LO or Modulating) is capable of matching output closer to demand. So its part load curve is better than for an ON/OFF fired boiler. It behaves like two smaller boilers if the low fire rate is near 50%.



**Figure 1: Part Load Efficiency Curve, Typical**

### d) Multiple Boiler Usage and Staged Firing for Improved Efficiency

- Consider part load efficiency when selecting the boiler(s) for the installation. Use of staged firing and/or more than one boiler will provide both improved efficiency and back-up in case a boiler is shut down for service.
- Always use staged firing when available. The key to improved part load efficiency is keeping the boiler on for the largest possible amount of time.
- When using multiple boilers, **always isolate the idle units to prevent stand-by losses.** Even units equipped with vent dampers lose heat from the jacket and out the draft diverter to the room if heated water is allowed to flow through them.
- The best method of isolation of idle units is to use primary/secondary piping, with each boiler on a separate secondary loop.
- How many boilers? — From Figure 1, you can see that the part load efficiency is best if the boiler is loaded at 50% or higher of its rated output. Below this loading the efficiency drops off sharply. So select enough units to keep the smallest unit loaded at a minimum of least 50% under mild conditions.
- Two to four ON/OFF fired boilers or two stage fired boilers are usually enough to provide an efficient system. With more boilers, the likely increase in service and downtime due to the number of installed units could offset performance and energy savings.

- Consider also that smaller, residential-type boilers have smaller waterways and can encounter limiting problems from the large volume of water on commercial systems.
- Use multiple commercial boilers on commercial systems to be sure they have the water content and heating surface inside suitable for commercial applications.

11. **Boiler Output Ratings.** For boilers with inputs less than 300 MBH, output is defined as Heating Capacity.

a) **Output** (Gas Boilers)

$$\text{Output} = \text{Input} \times \text{Combustion Efficiency}$$

b) **Output** (Oil boiler with input less than 300 MBH)

$$\text{Output} = \text{Input} \times \text{Combustion Efficiency}$$

c) **Output** (Oil boiler with input 300 MBH or greater)

$$\text{Output} = \text{Input} \times \text{Thermal Efficiency}$$

d) **Net I=B=R Ratings**

$$\text{Net Rating} = \frac{\text{Output}}{1.15}$$

- The 1.15 factor is for a 15% piping and pickup allowance.

e) **Net Square Feet Water**

$$\text{Net Square Feet Water} = \frac{\text{Net Rating}}{150 \text{ Btu per Ft}^2}$$

- The water rating per square foot is based on 170° F average water temperature.

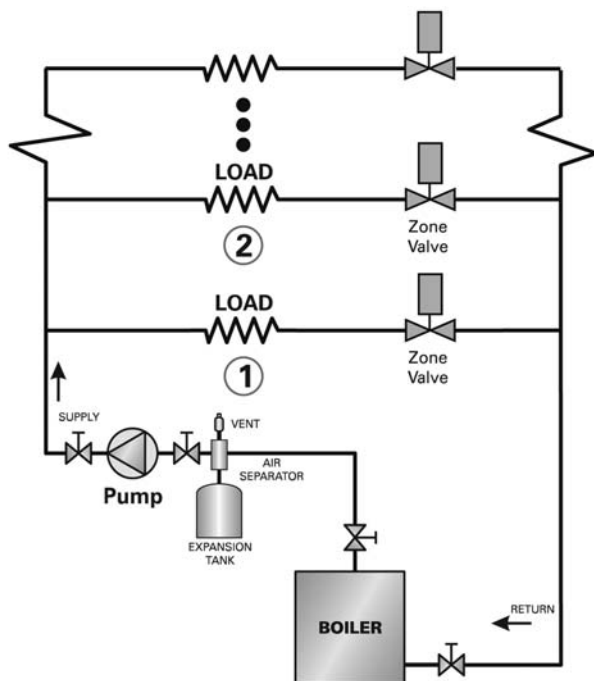
## D. PIPING HYDRONIC SYSTEMS

### 1. Residential System Types

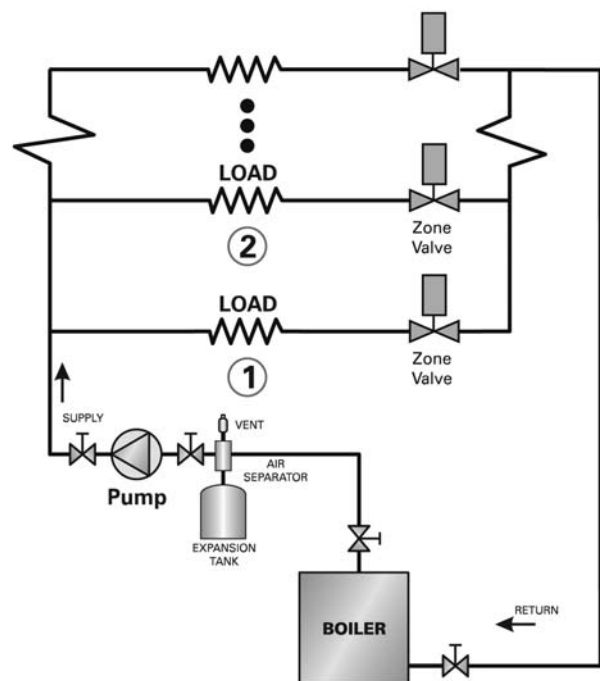
- Residential hydronic systems are most often piped as one-pipe series loop or two-pipe parallel loop (Figures 2 through 5). You will find zoning done with either zone valves or separate circulators for each loop as shown in the figures.
- On some larger residential systems, you may find primary/secondary piping (discussed below).
- Because residential systems are smaller, balancing is not as difficult as for commercial systems.
- The best heat distribution in a residential system can be accomplished with a parallel loop reverse return system.

### 2. Commercial System Types

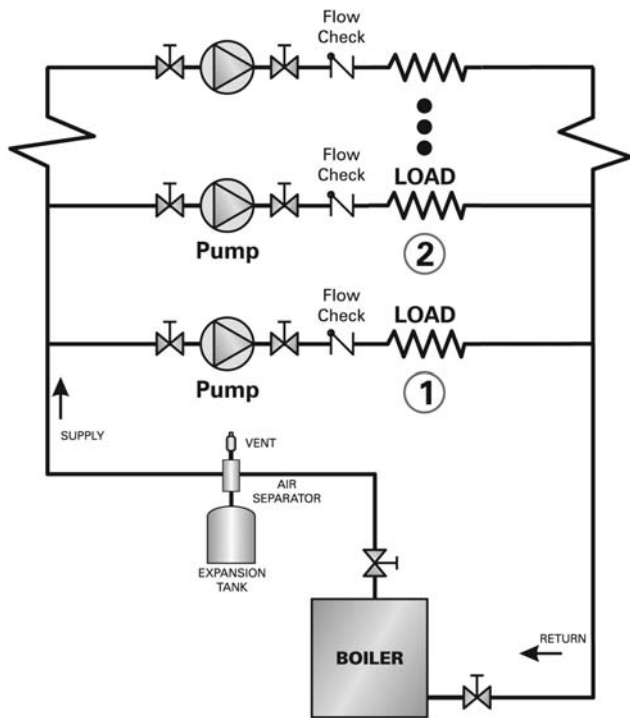
- Most commercial hydronic systems are piped with two-pipe direct return (Figure 6) or reverse return loops (Figure 7).
- Note Figure 8, which shows the correct method for isolating the boiler and chiller on chilled water systems.
- Reverse return systems are easier to balance since the loss in the system piping to each loop is approximately the same. Even with reverse return, the system will be difficult to balance if the individual loop pressure drops are not similar.
- The pressure drop on the risers (common piping) of a direct return system should be less than 5% of the circuit pressure drop.
- With either direct return or reverse return systems the pressure loss through the system piping depends on the amount of flow through each circuit. As control valves modulate closed, the pressure loss through the system changes.



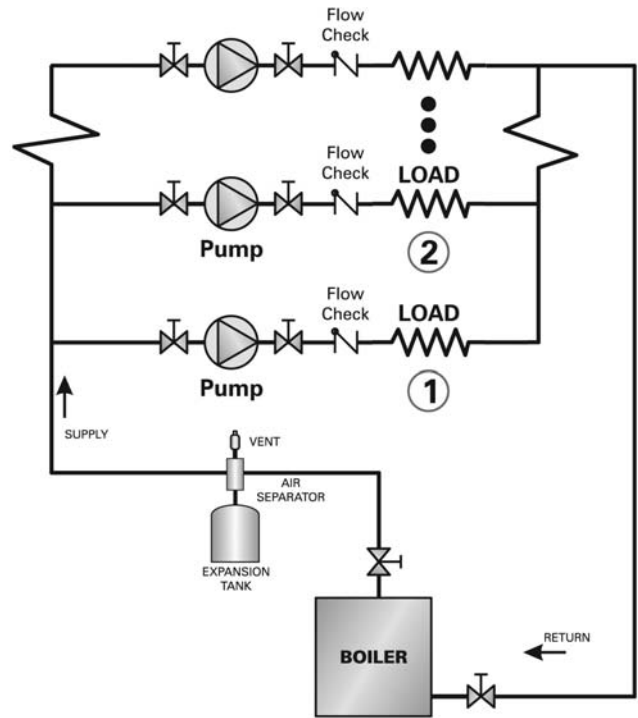
**Figure 2: Typical Residential Two-Pipe Direct Return with Zone Valves**



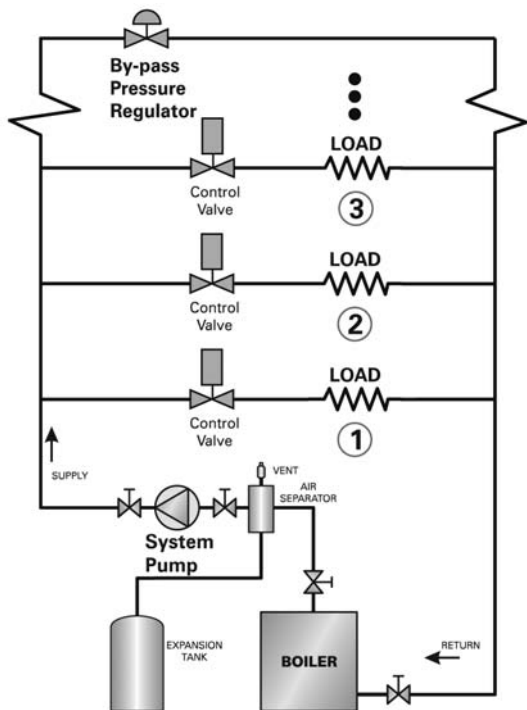
**Figure 3: Typical Residential Two-Pipe Reverse Return with Zone Valves**



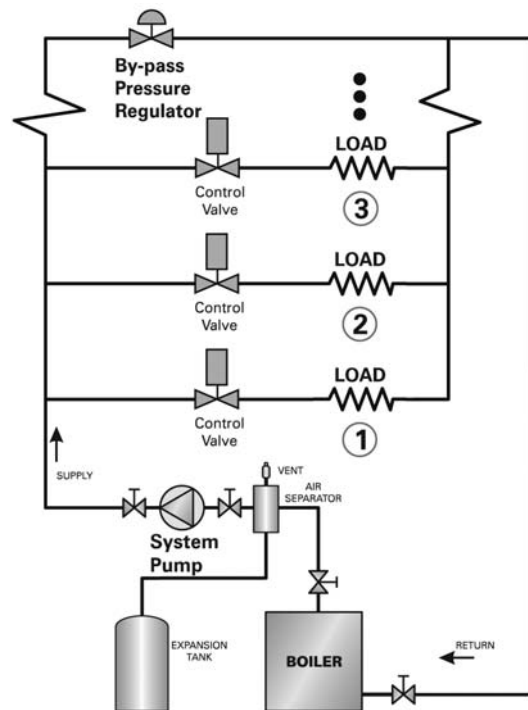
**Figure 4: Typical Residential Two-Pipe Direct Return with Circulators**



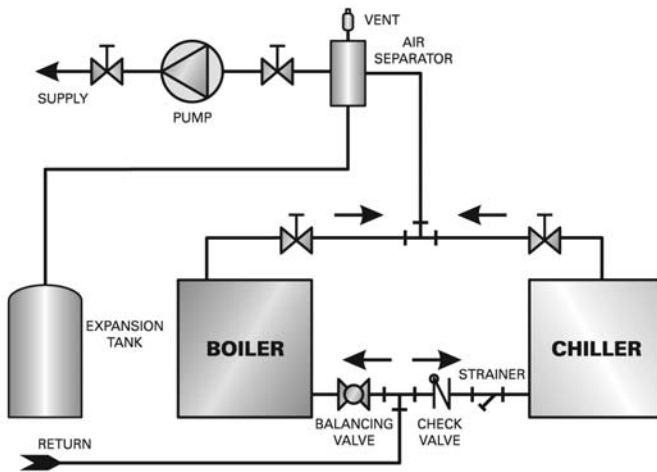
**Figure 5: Typical Residential Two-Pipe Reverse Return with Circulators**



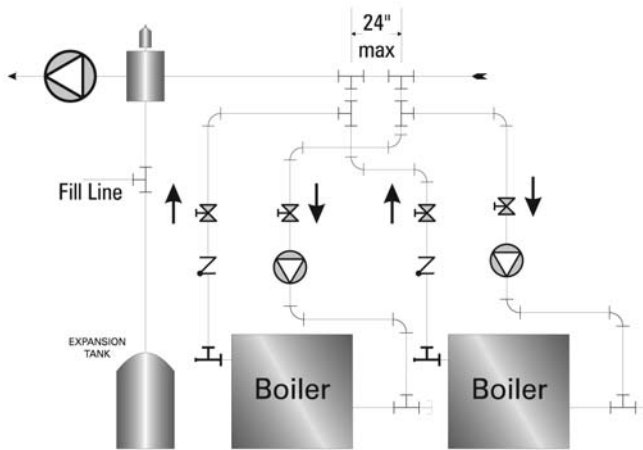
**Figure 6: Schematic, Typical Commercial Two-Pipe Direct Return System**



**Figure 7: Schematic, Typical Commercial Two-Pipe Reverse Return System**



**Figure 8: Piping to Isolate Boiler from Chilled Water on Chiller Systems**



**Figure 10: Piping Multiple Boilers - Commercial Systems**

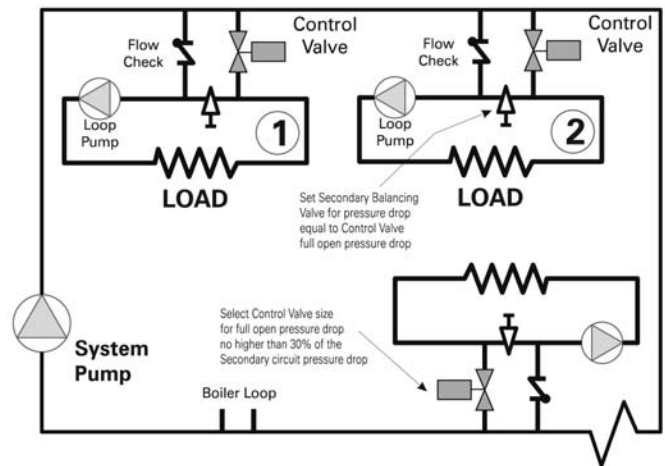
- f) Many systems are piped with a by-pass pressure regulator (differential pressure valve) as shown in the figures. This regulator prevents the pump head from exceeding a preset maximum. If the pump were allowed to reach maximum head or run in a near shut-off head condition for too long, the heat generated in the pump would cause cavitation and result in damage to the pump impeller.

Because the pump head increases with decreased flow, some circuits may encounter too much flow as control valves in other circuits close off. The regulator limits the head, preventing excess flow and lifting of spring-loaded valve seats due to too much pressure.

- g) Another, more versatile, piping design for commercial systems is the primary/secondary system (Figures 9 and 11). The primary loop is the loop containing the system pump. The boilers can either be directly in the primary loop or piped in secondary circuits (preferred method).

## E. PRIMARY/SECONDARY PIPING

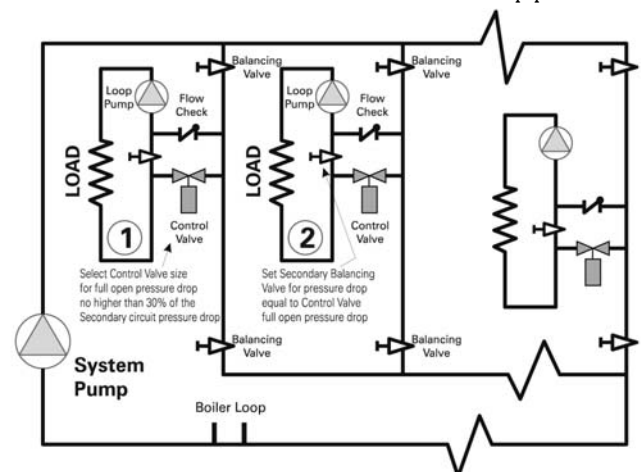
1. Primary/secondary systems are easily balanced since the pressure drop for each secondary loop is handled by its own pump and the primary pump always sees the same



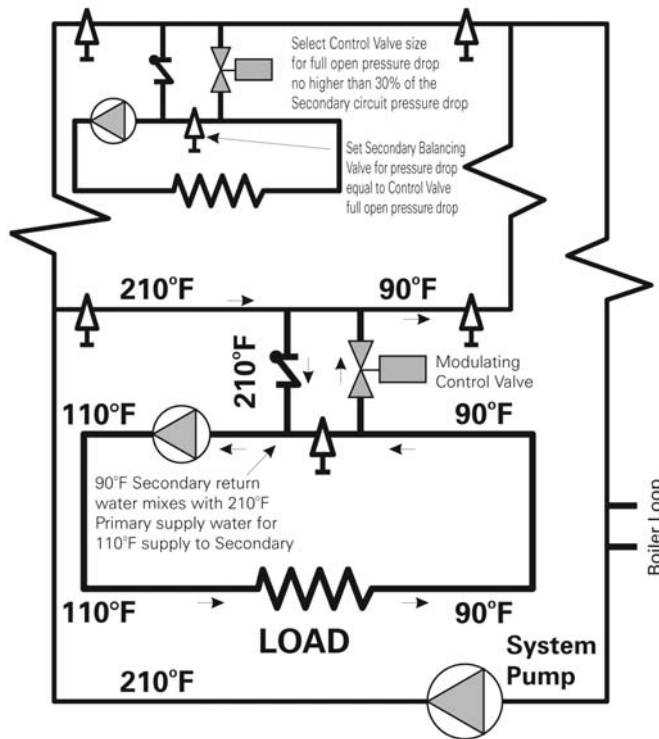
**Figure 9: Schematic, Typical Commercial Series Loop Primary/Secondary**

flow conditions. The primary pump only pumps water around the primary loop. And the primary loop pressure drop is constant regardless of control valve movement in the secondary loops.

2. The secondary loops can either be piped off the main loop in series as in Figure 9 or off of parallel crossover bridges as in Figure 11. The crossover bridge design is much more versatile since all of the secondary loops see the same supply water temperature. In the series system the supply temperature drops as the water proceeds around the main loop since hot water is being taken by the loops and cooler water is returned into the main loop flow.
  - a) The greatest advantage of the crossover bridge design is that the primary loop temperature drop can be greater than any of the secondary loops if desired. This allows lower flow rates and smaller piping in the primary loop.
3. Always pipe the secondary loop connections to the crossover bridge or main loop **between 6 and 24 inches apart**, as close as practical for the pipe size used. This is necessary to make the pressure drop through the common piping as small as possible, avoiding forced flow and loss of control in the secondary circuit.
4. **Use two flow control valves** if a control valve is not used in the secondary loop. These prevent gravity circulation in the circuit or in the individual pipes.



**Figure 11: Schematic, Typical Commercial Two-Pipe Primary/Secondary**



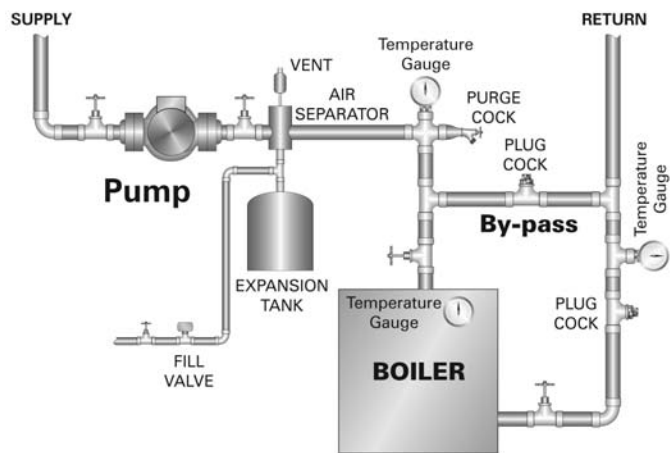
**Figure 12: Temperature Mixing in Secondary Piping Circuits**

## 5. Supply and Return Temperatures

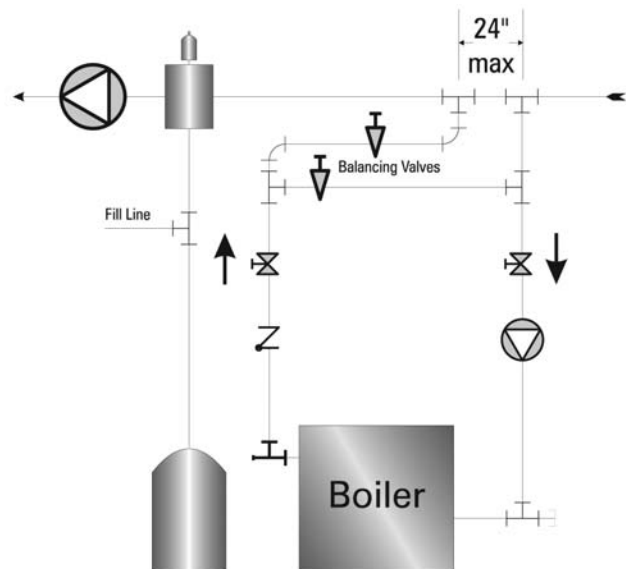
- a) With a crossover bridge two-pipe design, the temperature entering the bridge is the main loop supply water temperature. The supply temperature to the secondary loop, though, depends on the secondary circuit flow rate compared to the crossover bridge flow rate.
- b) The secondary circuit pump and control valve will mix bridge supply water temperature with the circuit return temperature to control the water temperature as desired. The return temperature to the system will depend on the return temperature from the secondary loop and the flow rates.
- c) If the primary supply temperature is 210° F, for example, the secondary circuit can mix this temperature down to any desired temperature by blending return water.
  - Example (See Figure 12):  
 Primary supply temperature: 210° F  
 Secondary temperature drop: 20° F  
 Secondary supply temperature: 110° F  
 Secondary return temperature: 90° F
  - The secondary circuit will blend the 210° F supply water with its 90° F return water to supply the circuit with 110° F water. The return temperature on the crossover bridge will equal 90° F. So the primary temperature drop will be  $210 - 90 = 120^\circ \text{F}$ . This means the crossover bridge flow rate will only be  $(20/120)$  of the secondary circuit flow rate, or only 1/6 as much flow.
  - Applying this approach for heating systems can greatly reduce the pipe size and pump requirements for the primary circuit.

## F. PREVENTING CONDENSATION

1. When system return water temperature will be below 130° F on gas boilers or 150° F on oil boilers, pipe the boiler with a bypass arrangement to blend the system return water with hot boiler water to obtain at least 130° F entering the boiler on gas or 150° F on oil.
2. **Residential Systems – Large Volume**
  - a) On large volume residential systems, such as those equipped with radiators or on converted gravity flow systems, install a fixed flow bypass arrangement as shown in Figure 13. This will cause a higher temperature rise in the boiler, allowing the average boiler temperature to be kept above the dewpoint.



**Figure 13: Residential Boiler Bypass Piping for Large Volume Systems**



**Figure 14: Commercial System Bypass Piping for Constant Low Temperature Operation**



### 3. Constant Low Temperature Commercial Systems

- a) On systems with a constant or nearly constant operating temperature (such as heat pump systems), the return water temperature to the boiler can be controlled by piping with a fixed-flow bypass loop. See Figure 14 for single boiler installations and Figure 16 for multiple boiler installations. Set the balancing valves such that the return water to the boiler is at least 130° F on gas or 150° F on oil.

### 4. Variable Low Temperature Systems

- a) Systems operated with an outdoor reset control, systems with a large temperature drop, snow melt systems, etc. will have return temperatures that can be very low. Outdoor reset, for instance, should be allowed to lower the system supply temperature to as low as 70° F to 90° F in mild weather, with return temperatures even lower.

- b) A fixed bypass arrangement at the boiler will not work with these systems. When a fixed flow bypass is set, it causes a fixed temperature rise through the boiler loop. The temperature rise must be high when the system temperature is low. But, if this same high temperature rise through the boiler loop were maintained when the system temperature is hotter, the boiler would cycle on the limit control frequently. To allow the boiler loop temperature rise to adjust to system temperature, **use a temperature control valve in the boiler loop to provide a variable flow bypass** as in Figure 15 for single boiler installations and Figure 17 for multiple boiler installations.
- c) This variable flow bypass arrangement provides automatic control of boiler return water while allowing the system return temperature to range as low or as high as needed for best operation of the heating system.

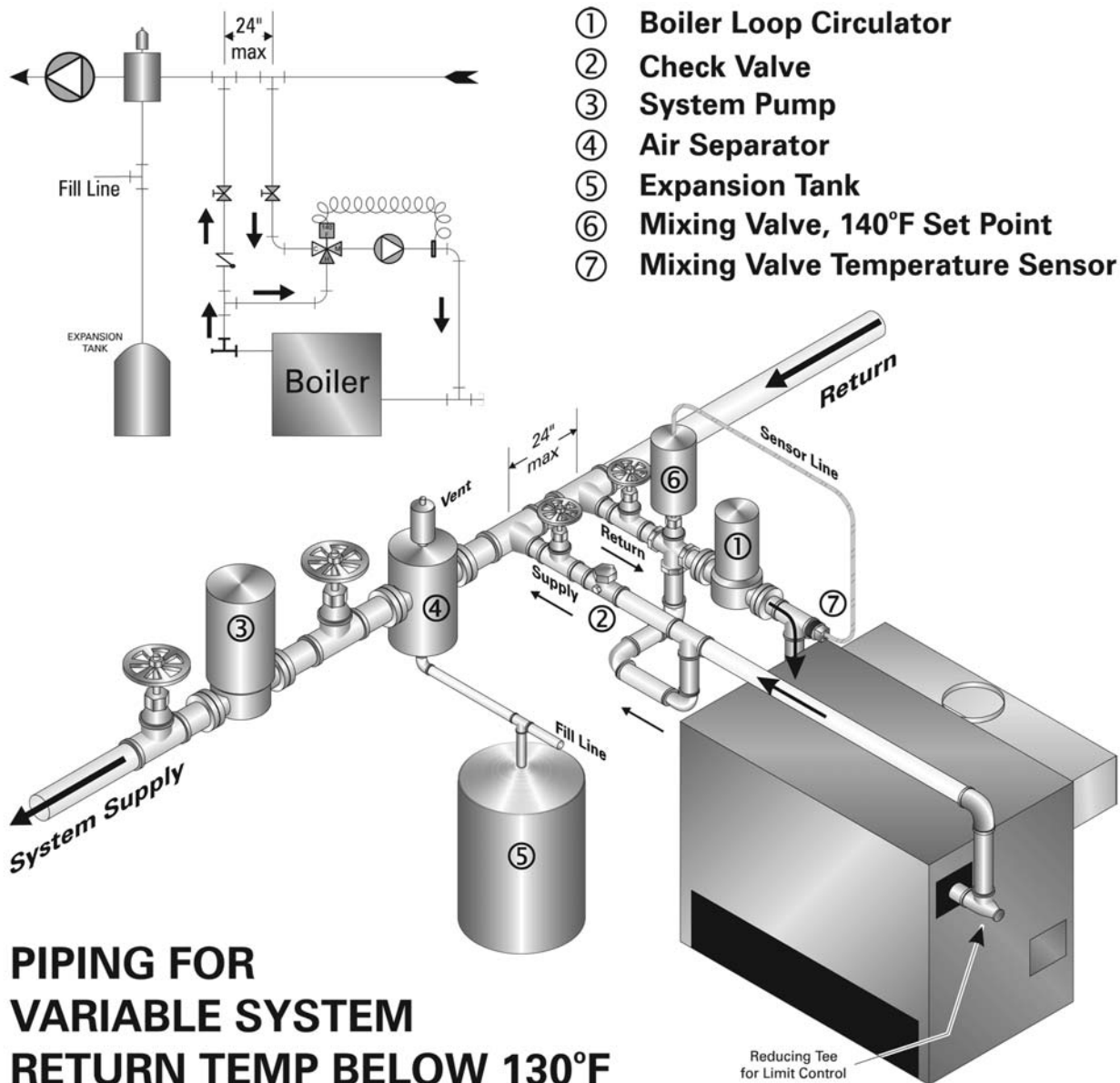
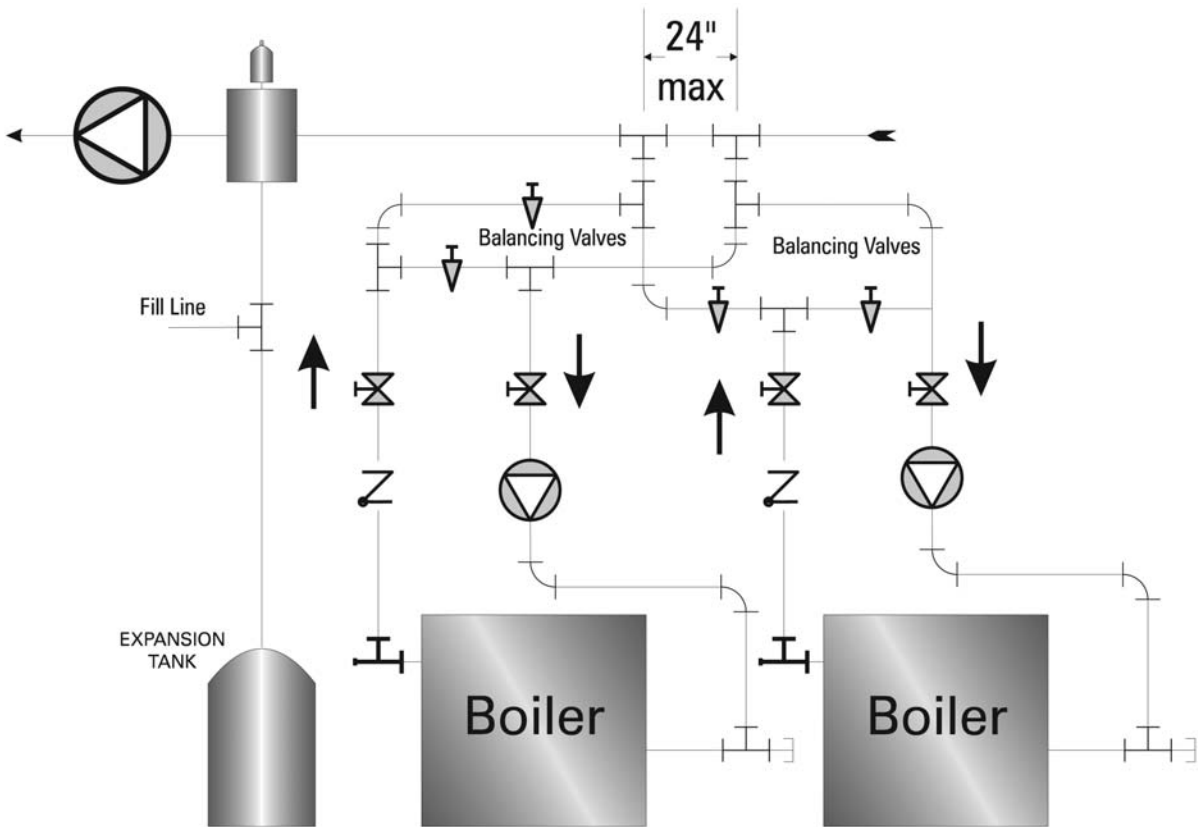
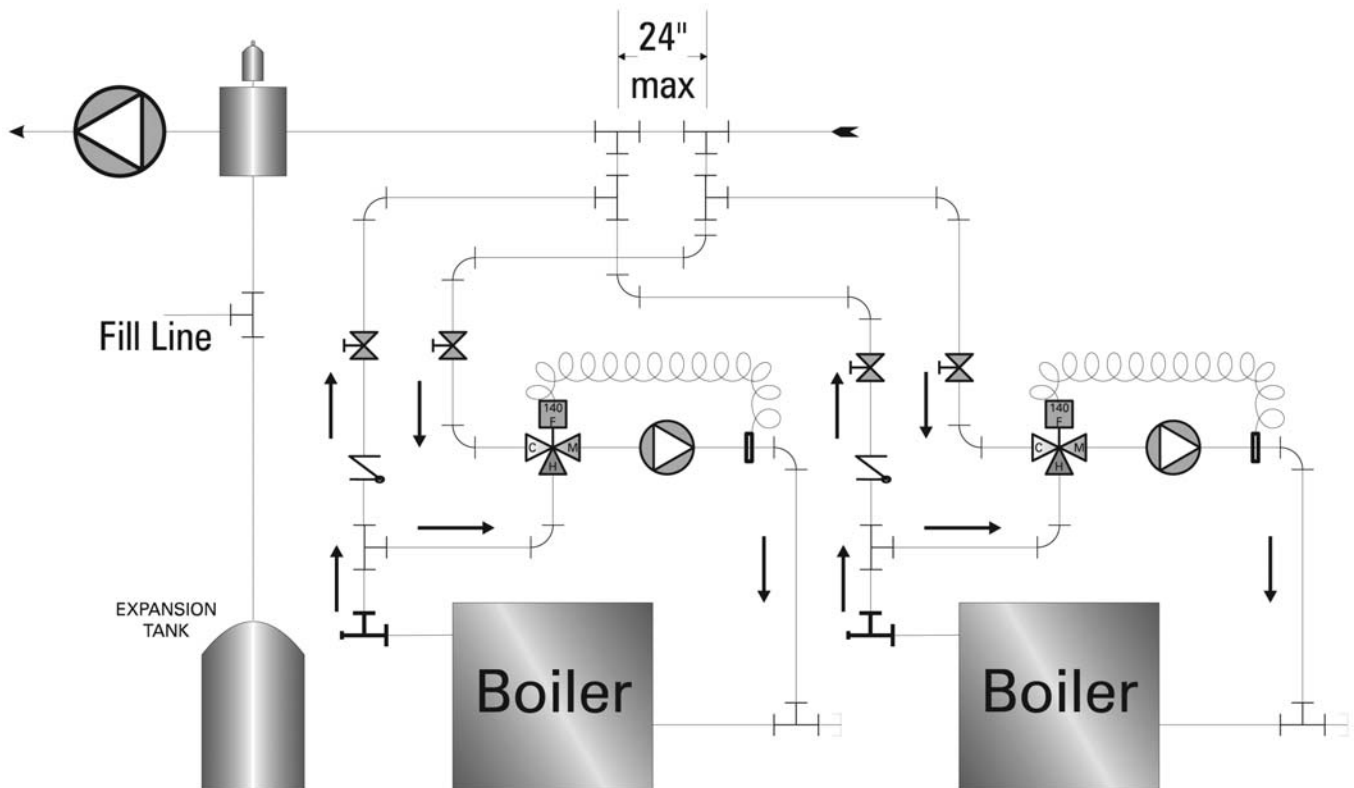


Figure 15: Commercial System Bypass Piping for Variable Low Temperature Operation



**Figure 16: Piping Schematic, Multiple Commercial Boilers, Constant Low System Return Temp Below 130° F on Gas or 150° F on Oil**



**Figure 17: Piping Schematic, Multiple Commercial Boilers, Variable Low System Return Temp Below 130° F on Gas or 150° F on Oil**

**G. DETERMINING PRESSURE DROP**

1. Determine the piping head loss in feet by first determining the Total Equivalent Length (TEL) for each circuit. The TEL is the number of feet of piping plus the allowance in equivalent feet for each fitting.
2. The equivalent length in feet for a fitting depends on the pipe size. Begin by estimating the TEL as 50% longer than the actual length of piping. **For small systems this estimate is generally adequate without doing an actual calculation of equivalent lengths for the fittings.**
3. Select a trial pipe size from Table 1 based on the design flow rate in gpm.
  - The maximum flow rates (based on a maximum of 4 feet per hundred feet loss) are intended to prevent flow generated noise.
  - The minimum flow rates are set for adequate movement of air through the system (based on a minimum 0.75 feet per hundred feet head loss).
4. Table 1 provides an easy means of calculating the loss in the pipe in feet per hundred feet of pipe. This data closely (within 3%) matches the ASHRAE Fundamentals, 1993, pressure loss charts developed from the Moody friction factor curves in the range up to 10 feet per hundred feet.

5. To calculate a pressure drop for any given flow rate, gpm (U. S. Gallons per minute), find “a” and “b” from the table and solve as:

$$\Delta H = a \times \text{gpm}^b \text{ feet per hundred feet}$$

$$\text{Total Head Loss} = \left( \frac{\text{TEL}}{100} \right) \times \Delta H \text{ feet}$$

6. You may find this calculation easier than using the ASHRAE log/log pressure drop curves, and the formulas can be easily programmed into a spreadsheet or calculator.
7. If the pressure drop for the trial pipe size and TEL is acceptable, determine the actual TEL for the system by totaling the fitting and valve equivalent lengths given in Table 2. Then recalculate total head loss as:

$$\text{Total Head Loss} = \left( \frac{\text{TEL}}{100} \right) \times a \times \text{gpm}^b \text{ feet}$$

8. For systems which will be filled with 50% glycol, calculate the pressure drops and flows for a water only system. Then apply the correction factors in the section following on glycol applications.

**Table 1: Head Loss in Piping – Factors “a” and “b” for Calculations**

Schedule 40 Steel Pipe Pressure Drop Information										
Head Loss, Feet per 100 Feet of Pipe, = a x gpm <sup>b</sup>										
Pipe Size Inches	Pipe OD Inches	Sch 40 Wall	Pipe ID Inches	Velocity = gpm x	Minimum Recom. gpm	Minimum Velocity ft per sec	Maximum Recom. gpm	Maximum Velocity ft per sec	a	b
1/2	0.840	0.109	0.622	1.05595	0.7	0.7	1.9	2.0	1.41393	1.775
3/4	1.050	0.113	0.824	0.60169	1.5	0.9	4.1	2.5	0.35528	1.795
1	1.315	0.133	1.049	0.37126	2.9	1.1	7.9	2.9	0.10682	1.811
1 1/4	1.660	0.140	1.380	0.21452	6.2	1.3	16	3.5	0.02685	1.828
1 1/2	1.900	0.145	1.610	0.15761	9.4	1.5	25	3.9	0.01228	1.837
2	2.375	0.154	2.067	0.09562	18.4	1.8	48	4.6	0.003421	1.851
2 1/2	2.875	0.203	2.469	0.06702	30	2.0	78	5.2	0.001371	1.860
3	3.500	0.216	3.068	0.04340	53	2.3	138	6.0	0.0004452	1.871
4	4.500	0.237	4.026	0.02520	110	2.8	284	7.2	0.0001081	1.883
5	5.563	0.259	5.045	0.01605	200	3.2	515	8.3	3.318E-05	1.893
6	6.625	0.280	6.065	0.01111	326	3.6	836	9.3	1.260E-05	1.900
8	8.625	0.322	7.981	0.006414	674	4.3	1721	11.0	2.962E-06	1.910
10	10.750	0.365	10.020	0.004069	1229	5.0	3126	12.7	8.886E-07	1.918
12	12.750	0.376	12.000	0.002837	1977	5.6	5013	14.2	3.415E-07	1.924
Type L Copper Pipe Pressure Drop Information										
Head Loss, Feet per 100 Feet of Pipe, = a x gpm <sup>b</sup>										
Pipe Size Inches	Pipe OD Inches	Type L Wall	Pipe ID Inches	Velocity = gpm x	Minimum Recom. gpm	Minimum Velocity ft per sec	Maximum Recom. gpm	Maximum Velocity ft per sec	a	b
1/2	0.625	0.040	0.545	1.3754	0.5	0.7	1.4	2.0	2.4135	1.709
3/4	0.875	0.045	0.785	0.6630	1.4	0.9	3.9	2.6	0.4275	1.731
1	1.125	0.050	1.025	0.3888	2.9	1.1	8.0	3.1	0.1186	1.746
1 1/4	1.375	0.055	1.265	0.2553	5.1	1.3	14.2	3.6	0.04280	1.756
1 1/2	1.625	0.060	1.505	0.1804	8.2	1.5	22.6	4.1	0.01836	1.764
2	2.125	0.070	1.985	0.1037	17.3	1.8	47.6	4.9	0.004727	1.776
2 1/2	2.625	0.080	2.465	0.06723	31.1	2.1	84.9	5.7	0.001626	1.784
3	3.125	0.090	2.945	0.04710	50.2	2.4	137	6.4	6.742E-04	1.791
4	4.125	0.110	3.905	0.02679	107	2.9	290	7.8	1.660E-04	1.801

Note: Some “a” numbers above are in engineering notation. 3.415E-07 = .0000003415, for example

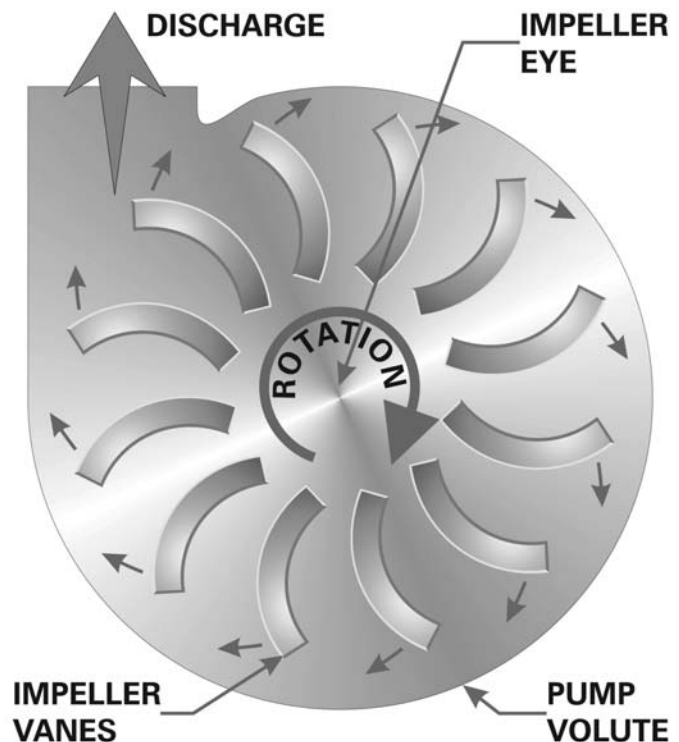
**Table 2: Equivalent Feet of Pipe for Common Pipe Fittings and Valves**

Equivalent Length, Feet, for Threaded Pipe Fittings and Valves										
Pipe Size Inches	Elbow, 90 std	Elbow, 90 LR	Elbow, 45	Return Bend	Tee Through	Tee Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check
0.5	3		0.6	3	1.3	3.5	20	0.5	0	7.8
0.75	3.6	2	0.8	3.6	2	4.5	21	0.6	13	7.9
1	4.5	2.4	1.1	4.5	2.7	5.4	27	0.8	13.7	8.9
1.25	5.7	2.9	1.5	5.7	3.9	7.4	37	1	15.6	11.7
1.5	6.5	2.9	1.8	6.5	4.9	8.6	43	1.1	15.5	13.4
2	7.5	3.2	2.4	7.5	6.8	10.5	52	1.3	15.7	17.2
2.5	8.1	3.4	2.9	8.1	8.5	12.3	62	1.6	15.1	21
3	10.1	3.9	3.7	10.1	11.3	15.1	75	1.8	16.4	27
4	12.6	4.3	5.1	12.6	16.1	19.7	102	2.2	17.9	36

Equivalent Length, Feet, for Flanged Pipe Fittings and Valves											
Pipe Size Inches	Elbow, 90 std	Elbow, 90 LR	Elbow, 45	Return Bend, Reg	Return Bend, LR	Tee Through	Tee Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check
1	1.3	1.3	0.7	1.3	1.3	0.8	3	39		14.3	6
1.25	1.8	1.7	1	1.8	1.7	1.1	4.2	52		16.1	8.7
1.5	2.2	1.9	1.2	2.2	1.9	1.3	4.9	54		16.1	10.7
2	2.9	2.3	1.5	2.9	2.3	1.5	6.3	67	2.6	18.7	15
2.5	3.4	2.7	1.8	3.4	2.6	1.7	7.5	76	2.6	21.7	18.9
3	4.3	3.2	2.3	4.3	3.2	2.2	9.6	88	2.8	28	25
4	5.6	4	3.3	5.6	4	2.7	12.6	116	2.9	38	36
5	7.2	4.8	4.1	7.2	4.8	3.2	14.9	143	2.4	50	48
6	8.8	5.5	5.2	8.8	5.5	3.7	18.2	172	2.8	64	61
8	11.7	7	7.4	11.7	6.5	4.4	25	247	3.5	91	87
10	14.6	8.2	9.3	14.6	8.2	5.3	31	331	3.5	122	116
12	17.3	9.4	11.5	15.1	9.4	5.8	36	409	3.6	151	144

## H. CENTRIFUGAL PUMPS

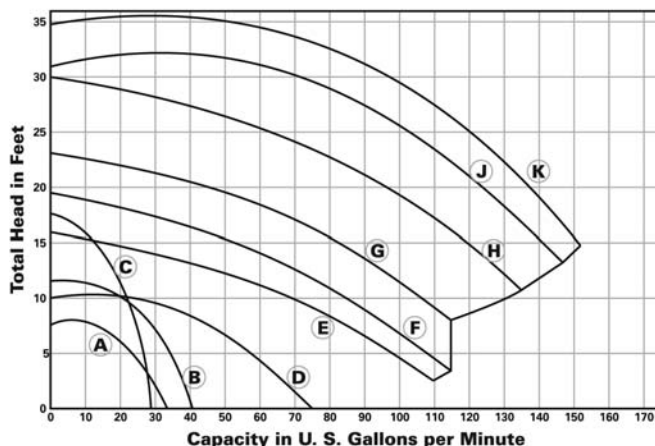
- Figure 18 is a typical centrifugal pump. The impeller vanes are curved and designed to push the water to the outside of the volute rather than to scoop it. Rotation is as shown in the drawing. A centrifugal pump causes water to flow this way by raising the pressure of the liquid, not by pulling it through.
- Figure 19 is a typical curve for a family of in-line pumps (labeled A through K) and Figure 20 is typical for a base-mounted end suction pump (the individual curves are for impeller diameters).
- Reading Pump Curves**
  - Pump curves are usually plotted as Feet of Head vs Flow, gpm. Feet of head is used because it is independent of the density of the liquid. The density will affect the pump horsepower required, but not the pump curve. (Multiply pump BHP times the specific gravity of the fluid for the power required). Even viscosity of the liquid will not affect the pump curve in the range from 40° F to 400° F, easily covering the range of hydronic heating.
  - When pump curves are rated in psig, a correction to the curve is needed if the actual conditions are different from those of the curve; i.e., the curve only holds for a given temperature and fluid density.
  - The pressure gauge reading (in psig) across a pump will equal the pump head times the density of the fluid (minus the velocity increase effect if the pump


**Figure 18: Centrifugal Pump Components**

discharge is smaller than the pump suction, such as on end suction pumps).

## 4. Pump Motor Horsepower Selection

- The in-line pumps in Figure 19 show no horsepower curves like the ones in Figure 20. These pumps are called “non-overloading” because the motor is sized to handle the load at any point on the pump curve. You will find the motor horsepower in separate tables in the pump manufacturer’s literature.
- When selecting an end-suction pump, such as in Figure 20, **select a motor which will handle the pump load throughout the expected range of operation.**
- You can consider the **motor service factor** when selecting the motor to handle the pump load. The service factor is cushion in motor capability. It means the motor could actually perform with a load up to its service factor above the rated capacity. Motors 3 HP and larger usually have a service factor of 15%, meaning they could handle loads up to 115% of their rating. The maximum load for a 3HP motor would be 3.4 HP, then. Smaller motors usually have a higher service factor, typically up to 35% or 40% for 1/3 HP and smaller.
  - Look at curve D in Figure 20, the curve for a 6 – 1/2” impeller diameter. The pump horsepower requirement is just under 3 HP at the design point of 250 gpm, 30 feet of head. A 3 HP motor would work well under these conditions. And the motor would also allow operation up to 15% more flow than anticipated, making this a good selection.
  - Selecting a 5 HP motor for this pump would make the pump non-overloading.
- Where possible, select the motor for non-overloading operation, i.e., a motor horsepower high enough to cover the entire pump curve regardless of the design point. Even if the motor is oversized it will not use excess energy. The load is determined by the pump.



**Figure 19: Typical Pump Curves for In-Line Centrifugal Pumps**

## 5. Pump Power Requirements

- End-suction pump curves will also show lines for pump efficiency. The pump load will be, in brake horsepower, BHP:

$$\text{BHP} = \frac{\text{gpm} \times \text{Head, ft}}{3960 \times \text{Pump Efficiency}}$$

- You can calculate pump efficiency approximately by using the horsepower (BHP) lines on the pump curves.

$$\text{Pump Efficiency} = \frac{\text{gpm} \times \text{Head, ft}}{3960 \times \text{BHP}}$$

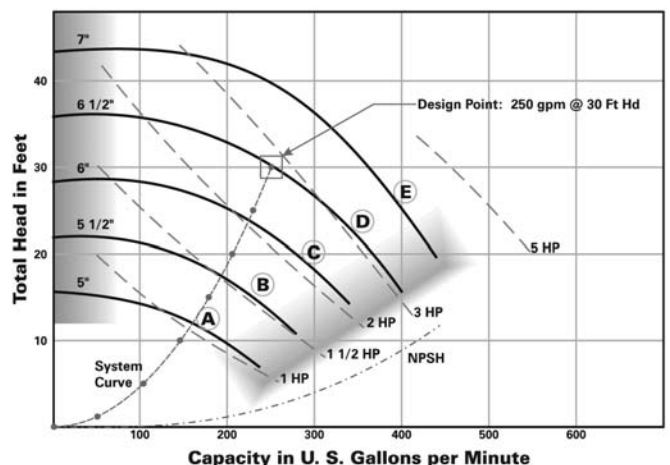
## 6. Electrical Power Usage

- The electrical power usage of the pump depends on the motor efficiency and the horsepower required by the pump:

$$\text{Electrical Power, BHP} = \frac{\text{Pump BHP}}{\text{Motor Efficiency}}$$

- For an 85% motor, the operating costs per horsepower are given in the Operating Cost table following. For a typical heating season in which the pump operates from 3500 to 5000 hours, the cost would be between \$307.00 to \$439.00 per horsepower. Controlling pump horsepower requirements by minimizing system pressure loss and using variable speed pumping or series/parallel pumps can significantly impact operating costs.

Running Time	OPERATING COST, DOLLARS PER BHP for 85% Efficient Motor				
	\$0.04	\$0.08	\$0.12	\$0.16	\$0.20
1 Hour	0.04	0.07	0.11	0.14	0.18
24 Hours	0.84	1.69	2.53	3.37	4.21
30 Days	25.28	50.55	75.83	101.1	126.38
6 Months	153.76	307.53	461.29	615.06	768.82
9 Months	230.65	461.29	691.94	922.58	1153.23
1 Year	307.53	615.06	922.58	1230.11	1537.64



**Figure 20: Typical Pump Curves for an End-Suction Pump**



## 7. NPSH

- Net positive suction head, or NPSH, is a measure of the pressure available to keep the water from vaporizing. The pressure at the eye of the impeller is lower than at the pump suction connection. The higher the flow through the pump, the greater the pressure difference. This is why NPSH curves increase with increasing flow.
- The NPSH curve is the NPSH required by the pump to prevent cavitation (formation of vapor in the pump impeller). For most hydronic heating systems, with cold fill pressure at least 12 psig and operating temperature at or below 240° F, NPSH requirements are not likely to present a problem.
- Never install a strainer on the suction side of a pump. The pressure loss that develops across the strainer will cause a lower pressure in the pump suction and will cause cavitation. Always install strainers on the pump discharge side.
- In high altitude applications, NPSH may be more of a factor. The available NPSH reduces 1/2 psig per 1000 feet of elevation, so fill pressures in psig will often need to be higher at altitude.
- If NPSH is a concern, consider a larger pump to operate more to the left on the pump curve where NPSH required is reduced.

## 8. Pump Selection

- You can select a pump by using only the system design point (maximum flow and pressure drop). But always select a pump which can operate at plus or minus 25% from the selection point. This allows for “drift” to account for system pressure drop actually lower than anticipated (moving to right on the pump curve) or higher (as control valves close off).
- Do not select the pump either near the shut-off head or on the flat portion of the curve near the left side. Also avoid the right side of the pump curve. At the end of the pump curve on end-suction pumps, the pump can easily cavitate, causing noise and severe damage to the impeller. These areas are shown in gray in Figure 20.
- If the design point doesn't fall directly on one of the impeller diameter curves for an end-suction pump, the pump manufacturer can trim the impeller to a diameter which will match the actual load. Pump curves for diameters between those shown can be plotted by splitting the distance between the curves shown and drawing a parallel curve for the new diameter.

## 9. Draw a System Curve

- Your pump selection will be improved by drawing a system curve for your system on the pump curves. Do this by plotting the **design point** on the curve. Then either use a heating slide rule (like the B & G System Syzer), apply the formula from Table 1 or apply the square law approximation:

$$\text{Head}_2 = \text{Head}_1 \times (\text{gpm}_2 / \text{gpm}_1)^2$$

- Plot enough points to draw in a system curve above and below the design point. The design point may not fall directly on a pump curve. By extending the system curve, you

can find the actual operating point where the system curve intersects the pump curve.

- Avoid oversizing pumps. Oversizing will cause higher pressure and flow through the system. Noise and control valve problems will occur and electrical power usage will be unnecessarily high.
- Where possible, select a pump with a relatively flat curve to prevent excessive pressure on closed or modulating control valves. Most control valves should operate with no higher than a 20 foot pressure drop. Use a differential pressure valve (by-pass pressure regulator) to trim off excess pump pressure in the system and prevent dead-heading the pump in low flow conditions.

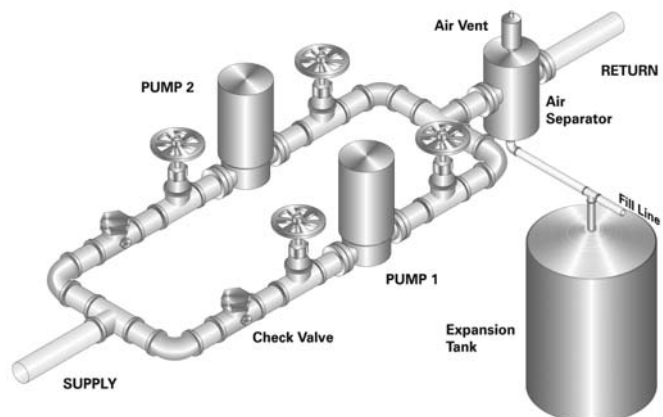
## 10. Parallel or Series Pumping

- The heating system only operates at or near the design point for a small fraction of the year (normally only about 2% of the time). There is an opportunity to improve the effectiveness of the system by reducing flow rate in milder weather. This can be done with variable speed pumping or, more simply, with parallel pumps or series pumps.

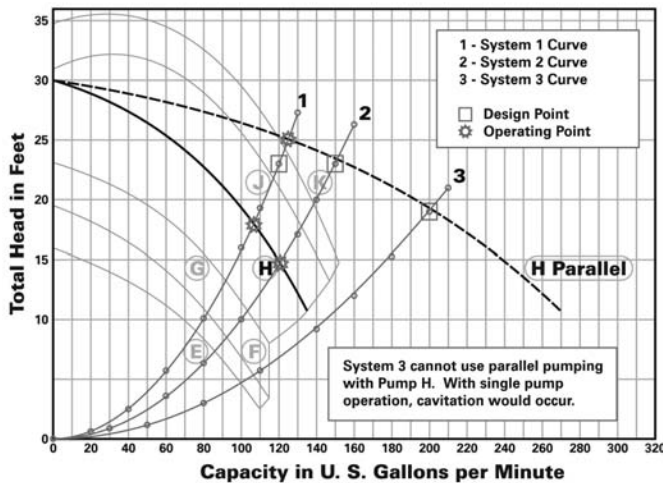
● **Two centrifugal pumps in parallel will deliver twice the flow of a single pump at the same head.**

● **Two centrifugal pumps in series will deliver the same gpm at twice the head of a single pump.**

- 21 and Figure 22 show a typical in-line pump applied in parallel. Pump H in Figure 22 is used in parallel with another pump H. The new combined operation curve is the “**H Parallel**” curve. Each flow rate point on the curve is twice that of a single pump H.
- We have shown three system curves considered for use with this parallel pump application.
- System 1* will work well. With both pumps running, the pump curve slightly exceeds the design point of 120 gpm at 23 feet head. When a single pump runs (see intersection of *System 1* curve with Pump H curve) the pump will deliver about 107 gpm at 18 feet head.
- Figure 23, from Bell & Gossett Bulletin No. TEH-1165, “Basic System Control and Valve Sizing Procedures”, shows that reduced flow has a minor



**Figure 21: Typical Piping, Parallel Pumps**



**Figure 22: Typical Pump Curves for an End-Suction Pump**

impact on heat transfer in a heating system designed for a 20° F temperature drop even to flow reductions up to 50%.

- f) The parallel pump application to *System 1* in Figure 22 will still deliver over 95% of the design heating requirement with only one pump running since 107/120 is 89% of the design flow. This means the parallel pump application:
  - Provides stand-by of over 95% capacity even with one pump down.
  - Allows the use of in-line pumps instead of a base-mounted end-suction pump.
  - Reduces the pump energy usage if the pumps are staged with heating load.
- g) Note that *System 2* in Figure 22 may be marginal since the operating point on the single pump curve doesn't allow plus or minus 25% of the flow for drift.
- h) *System 3* in Figure 22 is not acceptable for use with this parallel pump application. With both pumps running, the application is acceptable. But when a single pump operates, the pump would go into cavitation because the operating point is to the right of the pump curve.
- i) You can apply pumps in series to obtain higher head. Or use four pumps, two in series, two in parallel to achieve both increased flow and head. Parallel or series pumping is an effective alternative to a single pump and provides turndown, standby and efficient operation when staged.

#### 11. Pump Operating Equations

- a) The Affinity Laws for pumps provide a way of estimating the pump performance when the impeller diameter is changed or pump speed (RPM) is reduced. These laws are represented by the following equations:

- b) **Flow Rate, gpm:**

$$\text{gpm}_2 = \text{gpm}_1 \times \left( \frac{\text{Diameter}_2}{\text{Diameter}_1} \right)$$

$$\text{gpm}_2 = \text{gpm}_1 \times \left( \frac{\text{Speed}_2}{\text{Speed}_1} \right)$$

- c) **Feet Head:**

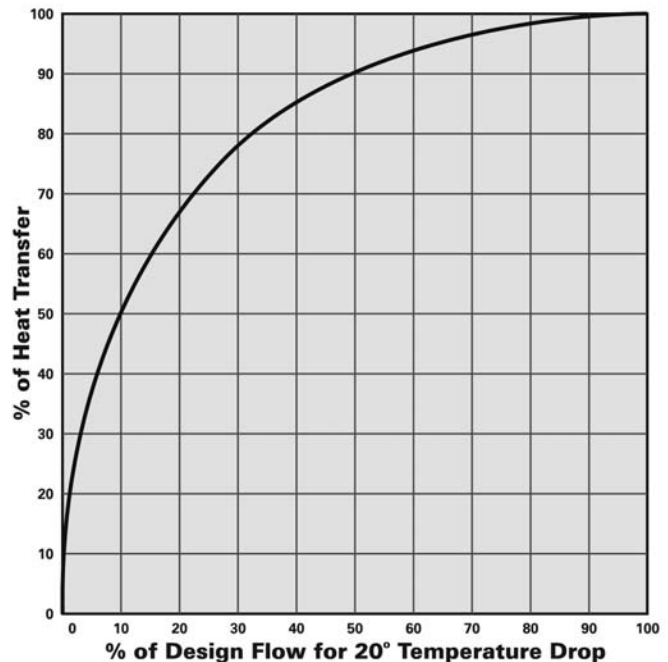
$$\text{Head}_2 = \text{Head}_1 \times \left( \frac{\text{Diameter}_2}{\text{Diameter}_1} \right)^2$$

$$\text{Head}_2 = \text{Head}_1 \times \left( \frac{\text{Speed}_2}{\text{Speed}_1} \right)^2$$

- d) **Power, BHP:**

$$\text{BHP}_2 = \text{BHP}_1 \times \left( \frac{\text{Diameter}_2}{\text{Diameter}_1} \right)^3$$

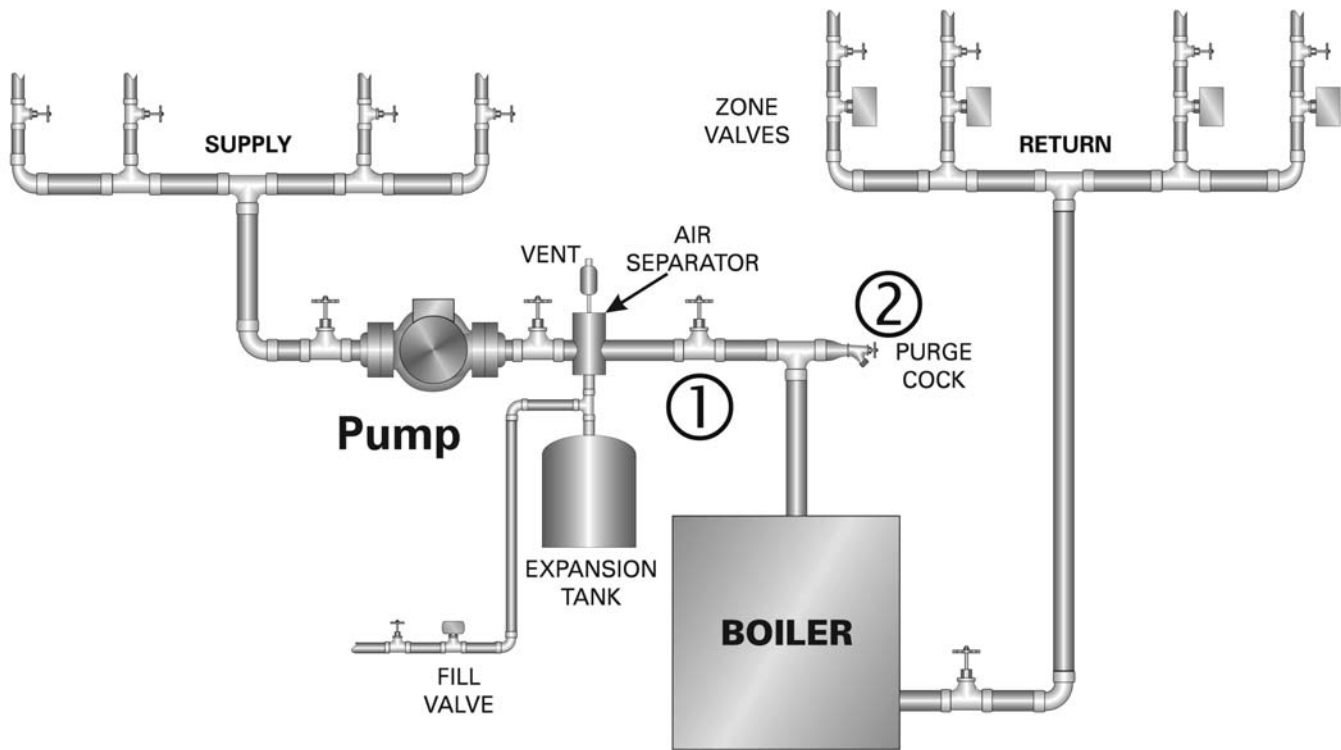
$$\text{BHP}_2 = \text{BHP}_1 \times \left( \frac{\text{Speed}_2}{\text{Speed}_1} \right)^3$$



**Figure 23: Heat Transfer vs. Flow**

#### 12. Pump Selection Summary

- a) Select a pump using the following alternatives:
  - Avoid using too much throttling with valves to control system pressure. Throttling uses energy. Consider natural balancing with alternative system piping and pump selections.
  - Consider Primary/Secondary piping. The primary pump sees a constant head. And, for two-pipe primary/secondary systems, the pump flow rate can be reduced for heating applications by allowing a higher primary system temperature drop.
  - Use Parallel or Series smaller pumps instead of one large pump. The pumps can be smaller, in-line type, and can be staged to improve performance and reduce energy consumption.
  - When using end-suction pumps, consider having the impeller trimmed to the diameter needed for



**Figure 24: Residential Boiler and Small System Piping for Zone Purging**

the application. This will reduce power consumption and prevent excessive flow or pressure drop.

## I. AIR CONTROL IN HYDRONIC SYSTEMS

### 1. Pump and Expansion/Compression Tank Location

- Always locate the system pump with its suction side as near as possible to the compression tank or expansion tank connection. The pressure at the expansion tank only changes if the volume of the water changes (due to water expansion or contraction, leakage or addition of water to the system) or if the tank charge pressure is changed manually.
- By locating the pump suction at the expansion tank, the pump discharge pressure adds to the system pressure. The solubility of air is greatest at highest pressure and lowest temperature. So pressurizing the system with the pump keeps the air in solution better, allowing the air to be pushed through the system back to the low pressure area at the air separator and expansion tank.
- If the pump discharge were at the expansion tank connection, the pump pressure would subtract from the system pressure. It can't increase the pressure at the expansion tank. This would lower the system pressure, causing the air to come out of solution at the top of the system instead of returning to the air separator and expansion tank.
- Locate the fill valve connection at the same point. **Never pipe the fill valve to the pump suction side if the pump discharge side is at the**

**expansion tank connection.** This would cause the fill valve to overfill the system because the pump would lower the pressure when it comes on.

### 2. Install an Air Separator

- Use an in-line air separator at the expansion tank connection whenever possible. The air separator separates more air from the water, reducing the possibility of system air problems.
- On residential systems, you can usually use a "purger" to accomplish the air separation.

### 3. Expansion/Compression Tank Sizing

- Size the compression tank or expansion tank based on the tank manufacturer's instructions. For air tanks, always use a tank fitting into the bottom of the tank. The tank fitting is a tube which fits up into the tank and prevents the tank water from gravity circulating back into the system. If gravity circulation from the tank occurs, air is pulled down into the system again and the tank can become waterlogged.

### 4. Guidelines

- Never use an automatic air vent on a system equipped with an air compression tank. The air will vent from the air vent and cause the tank to waterlog.
- Be careful if you drain a system with an air tank. You must open the air vent valve on the tank to allow air to flow in as the water leaves the system. Otherwise, a vacuum would develop. The vacuum could be high enough to collapse the tank from external pressure.
- Always charge diaphragm or bladder expansion tanks to the cold fill pressure with the tank disconnected from the system to set and check the charge pressure.



If the cold fill pressure is higher than the charge pressure, the usable volume in the tank is reduced and the system may over pressurize, causing weeping of the relief valve and excessive makeup.

- d) Size the piping based on the design flow rate. Design the piping for a minimum flow rate equivalent to 0.75 feet per hundred feet head loss in the piping and a maximum flow rate not to exceed 4 feet per hundred feet. See Table1.

#### 5. Air Venting and Purging

- a) Figure 24 shows a valving arrangement which works well on residential and small commercial systems. Vent the air from the system one zone at a time.
  - Close all zone shut-off valves
  - Close the boiler main shut-off valve (**valve 1**)
  - Open the purge valve (**valve 2**)
  - One at a time, open each zone shut-off valve and allow water to flow through and push the air out the purge valve. Close the zone shut-off valve when completed, then open the next zone shut-off valve.
- b) You will find manual air vents at the top of some residential systems and on most hot water systems with radiators. Use these manual vents to vent the initial air from the radiators.
- c) **Venting Commercial Systems**
  - Where possible, purge initial system air using individual circuit flow as for residential. Use manual venting at the top of the system as needed.
  - **Purge and vent the system air with the pump off.** If the pump is allowed to run, it will trap air in the impeller due to the low pressure zone.

6. To achieve generally the same heat transfer as water, increase the design flow for 50/50 glycol water as given in the table below.

APPROXIMATE FLOW INCREASE FOR SAME HEAT TRANSFER WITH 50/50 GLYCOL VS. WATER ONLY	
Average Water Temperature, °F	Multiply Water-Only Flow Rate by:
40	1.22
100	1.16
140	1.15
180	1.14
220	1.14

7. Centrifugal pump curves are not affected by the glycol/water solution. The pump HP is affected by the fluid density, though.
8. To determine pressure drop at the new flow rate, determine the pressure drop for water only at the new rate. Then multiply the pressure drop by the factor in the table below.

APPROXIMATE PRESSURE DROP OF 50/50 GLYCOL VS. WATER ONLY	
Average Water Temperature, °F	Multiply Water-Only Pressure Drop by:
40	1.48
100	1.10
140	1.00
180	0.94
220	0.90

### J. GLYCOL/WATER SYSTEMS

1. Use only inhibited propylene glycol solutions up to 50% by volume with water. Have the system mixture checked annually to make sure the inhibitor level is adequate. **Glycol without the inhibitor is very corrosive.**
2. Do not use ethylene glycol since it will attack gaskets and seals used in boilers and hydronic systems.
3. Compression/Expansion Tank Sizing – Consult tank manufacturer. General rule is to increase the tank size by 20%.
4. Glycol/water has a higher viscosity, lower heat content and lower heat transfer effects than water alone. So all system components need to be evaluated for use with this solution.
5. Heat exchangers need additional heating surface to do the same job with glycol/water. The approximate increase in shell length for a shell and tube heat exchanger ranges from a 10% increase at 240° F to a 53% increase at 120° F. Consult the manufacturer's literature for actual requirements.

9. This data is taken from ITT Bell & Gossett Bulletin No. TEH-176, "Hydronic Systems Anti-Freeze Design."

#### 10. Tips for glycol applications:

- a) Don't use galvanized pipe in the system. The coating reacts with glycol.
- b) Clean the system thoroughly with trisodium phosphate or other chemical cleaner before filling.
- c) You will probably want to use a manual fill system instead of an automatic fill valve. If the valve adds water to the system, the glycol will be diluted. With manual fill a drop in system pressure will indicate a leak problem. With automatic fill, you will not be warned of the problem.
- d) Do not use chromate water treatment. The chromate reacts with glycol.
- e) Use pumps with mechanical seals—not packing glands. Glycol leaks easier than water and may seep through the glands.

## K. INSTALLATION CHECKLIST

### ☐ Inspect the **Old Boiler (Replacement)**

#### Why was it removed?

- If it was limed or damaged from oxygen corrosion, the system leaked or had a lot of make-up water.
- If the heat exchanger was corroded the operating temperature may have been too low.
- If the heat exchanger was plugged, the burner may have been out of adjustment; the vent may have been blocked or damaged.
- If it was cracked and leaking, this may have been caused by liming or flame impingement.

If you don't fix what damaged the old boiler, the new boiler won't last very long.

### ☐ Inspect the **Vent System**

Masonry Chimneys – Check the liner: it must be intact with mortar in good condition with no visible holes; Check the cleanout door: it must be tightly sealed; Check the vent connection into the chimney: it must not protrude into the liner.

Metal Vents – Check the vent connector and metal venting: make sure there is no corrosion, blockage or leakage and all joints should be tight. Unheated areas: vents through unheated areas must be insulated to prevent condensation.

### ☐ Check the **Fuel Supply**

Make sure oil lines are unobstructed; check the oil tank for water or sediment; make sure the oil tank position and pipe sizes don't cause excessive lift for the burner oil pump.

Check the gas supply lines and regulators.

### ☐ Check the **System History**

If the system has a history of **no-heat problems** in some areas: the flows may not be balanced; some radiation or loops may be air bound; the piping may have been connected in series causing less heat at the ends of the loops.

If the system has a history of **flow noise**: this could be due to air pocketing and poor air removal in the system; the flow rates may be too high.

If the system has a history of **banging or noisy pipes**, check for expansion allowance in the piping. Make sure hangers allow the pipe to move without making noise. Make sure control valves don't close too quickly, causing water hammer.

If the system **leaked or the relief valve weeped**: check for pipe leaks; make sure the expansion tank is working and sized correctly; check diaphragm tank fill pressure; check the fill valve; does the compression tank or elevated tank have a tank fitting to prevent recirculation?

### ☐ Check the **Pump or Circulator**

Inspect the **Old Pump** – Check the impeller condition.

- If it is damaged from pitting, it probably cavitated due to too low pressure on the suction side or excessive air in the system. Another cause of cavitation is operation of the pump in no-flow

conditions, allowing the heat generated by the pump motor and friction to heat the water in the pump to the vapor temperature.

- Install a by-pass pressure regulator if the system piping would allow this to occur. Make sure the pressure setting of the regulator is higher than any of the loop pressure drops.
- If it is eroded, the system may have a lot of sediment and should be cleaned and a strainer installed on the pump discharge side.
- If it is corroded, check the water for possible acidity or possible glycol without enough inhibitor.

**Pump Noise** – Whirring or high-pitched running sounds are usually due to cavitation, caused by too low a pressure at the pump suction. If the pump suction is not connected to the same point (or near) as the expansion tank, repipe the pump. If there is a strainer on the pump suction line, move it to the discharge side.

**Pump Rotation** – Make sure the pump is rotating in the correct direction. It can't perform to specification if the rotation is wrong.

**Pump Orientation** – Make sure the oiling openings are pointed up and the pump motor is in an orientation approved by the pump manufacturer. Oil the pump if needed.

**Strainers** – Don't pipe a strainer on the pump suction side and relocate it if it is installed there. This will cause pump cavitation and impeller damage as the strainer becomes plugged. Make sure the building maintenance personnel or owner is aware of the need to check and service the strainer. Even on the pump discharge, a strainer can cause no-heat problems or pump cavitation (due to running under no-flow conditions).

### ☐ Check out **No-Heat Problems**

Heat distribution problems are usually due to poor system balance, series piping without bypasses or air trapping in zones.

- Check the system balance to see if the problem zones are getting enough flow.
- On series piped installations, add a by-pass line and balancing valves to provide some of the hot supply water to the problem zones.
- Get the air out using the methods described here.
- Also check thermostatic radiator valves and air vents. Check control valves.

### ☐ Check out Leaks or **Relief Valve Weeping**

Relief valve weeping is caused by undersized expansion or compression tanks, fill pressure higher than charge pressure on diaphragm tanks, waterlogged compression tanks or defective fill valve. The valve will begin to weep with pressure of about 90% of the valve pressure setting. So select the expansion tank to avoid the pressure getting this high.

- Check these components out as described in this survey and in the troubleshooting section of the manual.

## ☐ Check out **Flow Noise Problems**

Flow noise is due to air bubbles carried in the water or to excessive water velocity through pipes and valves.

- An air separator and location of the pump with its suction at the expansion tank connection will help operating air removal.
- Purge and manually vent the system thoroughly. If the system is equipped with a high head pump or small pipes, the flow could be too high under some conditions. Fix this by adjusting the system balance. Add a bypass pressure regulator if not already installed.

## ☐ Check for **Piping Leaks**

Repair or replace all leaking pipes or pipe joints.

- If you don't, the system and the boiler will soon develop problems.
- Fill the system, pressurize and make sure it holds pressure to prove there are no leaks.
- Recommend to the owner to install a water meter on the fill line. If any leaks occur the meter will tell the story.

## ☐ Check the **Air Removal System**

- Install an air separator. Locate the expansion tank on the supply line. Pipe the pump with its suction connection as near as possible to the expansion tank connection.
- Make sure compression tanks have a tank fitting in the bottom to prevent recirculation and water logging.
- Install purge piping if possible. Purge the system and vent manual vents.

## ☐ Check the **Expansion Tank**

- Make sure the expansion tank is sized correctly for the system.
- Is the diaphragm or bladder intact? Check by seeing if it will hold pressure.
- Disconnect the tank from the system to check the charge pressure. Use a tire gauge or pressure gauge to check diaphragm tank charge pressure. Charge it to the required cold fill pressure (usually 12 psig for sea level residential and small commercial systems).

## ☐ Check the **Electrical Wiring Connections**

A common problem on zone valve systems is switching of the leads from three-wire zone valves. Check to be sure they are correct by doing the following:

- Remove the valve end switch connection(s) at the boiler.
- Operate each zone and check across the leads with a voltmeter.

- The meter should never show a voltage reading under any combination of zone valve conditions. If it does, correct the wiring to the zone valve in question.
- If any valve is mis-wired it will cause burn up of transformers, gas valves or other boiler components.

Check all other control and power connections.

## ☐ Check the **Anti-Freeze Inhibitor Level and Concentration**

Glycol in hydronic applications must be specially formulated for the purpose. It includes inhibitors which prevent the glycol from attacking metallic system components. Have the system fluid checked for the correct glycol concentration and inhibitor level.

## ☐ Check the **Pipes for Freeze Protection**

Check pipes adjacent to cold walls or in unheated spaces. Insulate and trace them if necessary to be sure they can't freeze up. Keeping the water moving at all times will reduce the likelihood of freezing. If the location is very cold, tracing will be necessary in addition to insulation. Also consider a glycol/water fill, if necessary, to be sure the pipes won't freeze.

## ☐ Protect the **Boiler from Condensation**

Review the section in this survey on prevention of condensation. Review the recommended piping section in the manual. Recommend repiping to the owner if the system is likely to provide low temperature return water to the boiler. The boiler will be seriously damaged and the flue passages are likely to be corroded and/or blocked with soot and sediment if condensation occurs.

# **Water Survey**

## ***Hydronic Systems***

### **Pre-Installation Survey and Checklist for Water Systems**



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