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Variable Volume Pumping Fundamentals

HVAC engineers have been using primary-secondary pumping to design the most cost-effective pumping system while meeting their clients' needs.

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It is the objective of the HVAC engineer to design the most cost-effective pumping system while meeting the clients' needs. Primary-secondary pumping is a tool HVAC engineers have been using for nearly 50 years to accomplish this objective.'

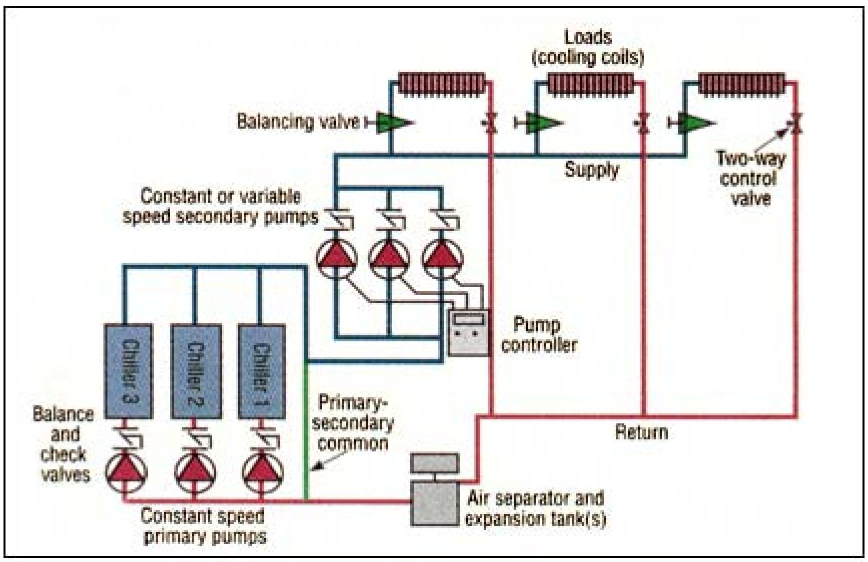
ASHRAE/IES Standard 90.1- 1989 - Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings[2] requires "all pumping systems with modulating or step open and closed valves which have total pump system horsepower greater than 10 must be capable to flow at 50 percent of design value or less." The standard 90.1 - 1989 User's Manual[3] "highly recommends" primary-secondary pumping for "systems with large, high pressure drop distribution systems such as those serving campuses and airports."

Primary-secondary pumping **(Fig. 1)** provides the means for the constant volume pumping of the low horsepower primary pumps through the chiller. These pumps are lower horsepower than the secondary pumps because they only haves to overcome the friction loss associated with the chiller, pipes, and valves in the primary loop. The chiller pumps are balanced to the design flow rate. Pump impellers should be trimmed discharge of the pump. This is an important energy saving measure.

The secondary pumps are higher horsepower because they must overcome the friction loss associated with the secondary loop-the distribution piping, fittings, valves, coils, etc. These pumps operate in the energy saving variable volume mode. Depending on the return on investment, the secondary pumps either remain constant speed and ride their characteristic head-capacity curve, or the designer can incorporate adjustable frequency drivers to saves additional pumping energy **(Fig. 2)**

There are three critical design areas that must be considered for any variable volume pumping system:

- The common pipe
- Chiller sequencing
- Control valves and actuators



1. Primary-secondary pumping system.

Common pipe design

The design of the common pipe **(Fig. 3)** is critical to the performance of a primary-secondary system. The function of the common pipe is to decouple hydraulically the primary and secondary pumps while still providing thermal interaction. To ensure proper system performance, the common pipe design criteria are:

• **The maximum pressure drop in the common pipe shall not exceed 1.5 ft -** Establish the pressure droop in the common pipe by assuming the flow of the largest resultant pressure drop should not exceed 1.5ft. This is the basis of primary-secondary pumping. Higher friction loss in the common tends to make the primary and secondary pumps act in series, resulting in an induced flow in the system. (This incorporates a safety factor.) Typically, a chiller is usually sequenced on or off y the time one half of the flow of the largest pump is achieved. For simplicity of design and installation, the common pipe is often the same diameter as the distribution piping.

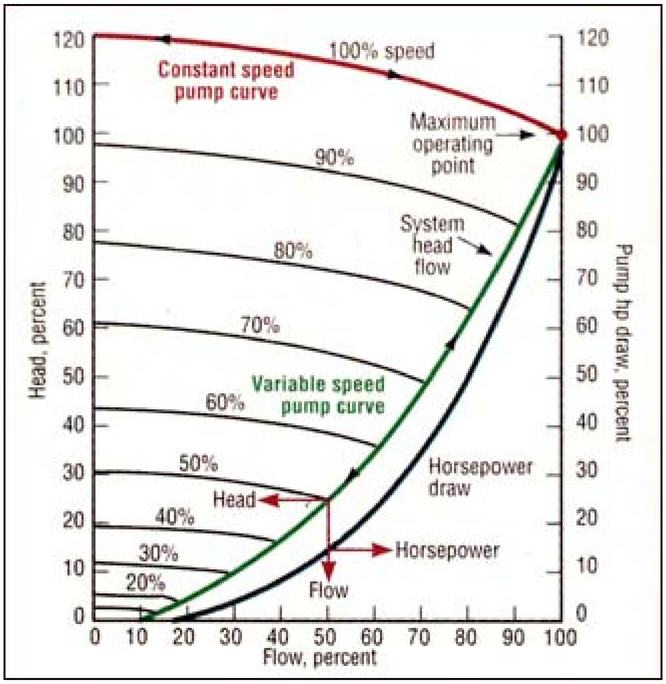
• **Maximum three pipe diameters of separation between the secondary supply tee and the secondary return tee -** A common pipe of this length is more than adequate to eliminate mixing due to excessive return velocity in the secondary return piping. Loner length common pipes may result in an excessive pressure drop greater than 1.5 ft with results as described above.

Under no circumstances should a check valve be located in a common pipe. The addition of a check valve will result in the primary and secondary pumps acting in series when secondary (distribution) flow exceeds primary (production) flow. The increase in demand results in an increase in flow through the chiller that fan lead to higher chiller discharge temperatures and chiller tube erosion. Further details on this subject are discussed below in the chiller sequencing section.

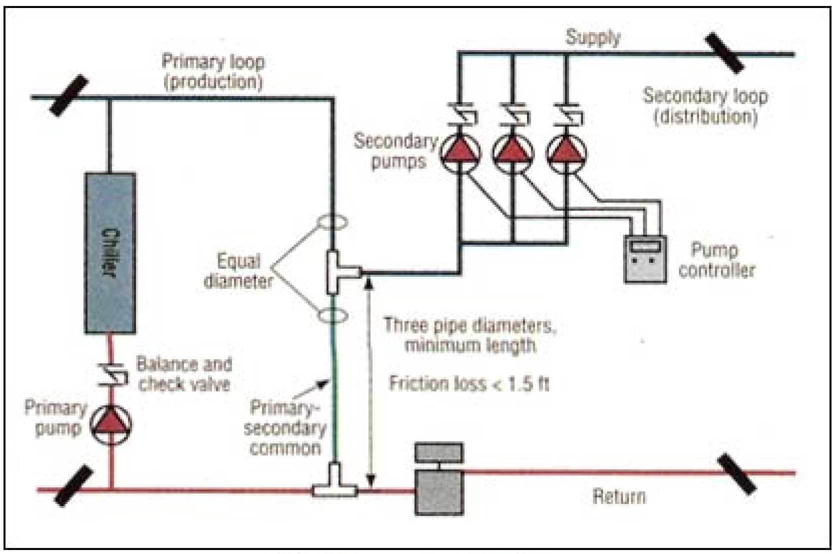
Other terms are often used interchangeably when discussing primary-secondary pumping. The primary system is also known as the production system - the place where the chilled or hot water is produced.

The secondary systems are also called the distribution because its purpose is to convey the chilled and hot water to and from the load(s).

The common pipe is also known as a decoupler. This pipe - common to both the primary and secondary systems - provides the hydraulic separation between the two.



2. Constant speed versus variable speed pump operation



3. Common pipe design

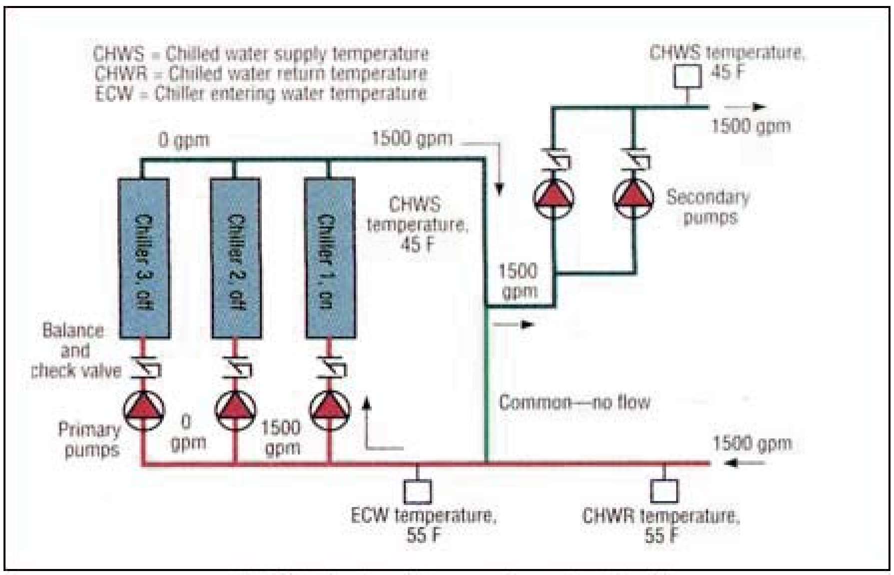
Chiller sequencing

Proper chiller sequencing plays an important role in the overall performance of a primary-secondary pumping systems. The operator of the system must recognize the three flow conditions that can occur. These are:

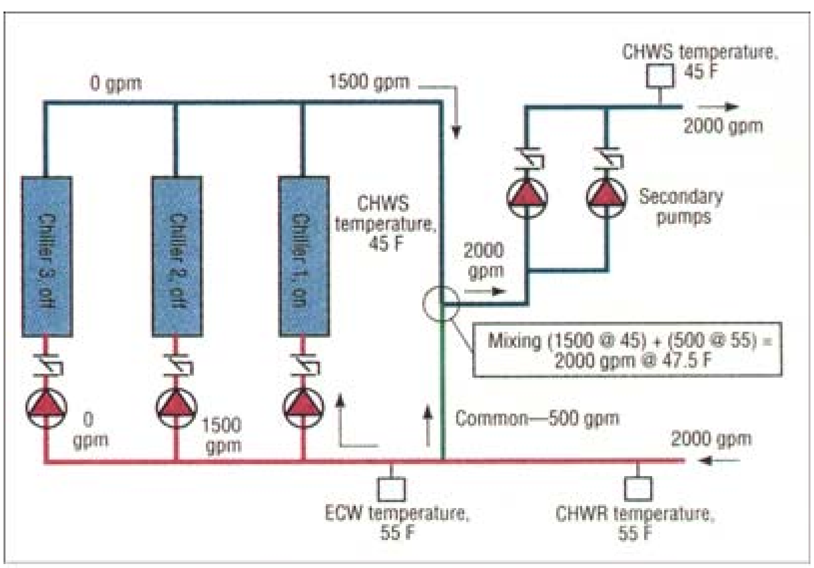
- Distribution (secondary) flow equals production (primary) flow.
- Distribution flow is greater than production flow. ·
- Production flow is greater than distribution flow.

• **Distribution flow equals production flow -** Although this flow condition rarely occurs, we will observe the flow pattern of this scenario **(Fig. 4)**. The chiller is supplying

1500 gpm of 45F water to the load systems. The two way valves in the secondary are responding to conditions equal to 1500 gpm x 10F ΔT x 500 = 7.5 MBtuh or 625 tons. The two-way valves, coils, and associated piping represent a pressure drop of 100ft. The secondary pump is riding its curve, producing the required 1500 gpm at 100ft of head. Because the load is equal to 625 tons, the return water temperature to the chiller is 55F at a flow rate of 1500 gpm. The thermal balance is complete. There is no flow in the common pipe. The remainder of the time the secondary or load flow will be greater or less increases in the space, the two-way valves begin modulating to a more open position and deliver additional chilled water to satisfy the additional load. When this occurs, the pressure and flow relationship changes such that the distribution flow is greater than the production flow **(Fig. 5)**



4. Distribution flow equals production flow



5. Distribution flow greater than production flow

• **Distribution flow greater than production flow -** With this condition, the flow rate through the chiller is 1500 gpm at 45F. Because the flow rate is fixed at 1500 gpm, the pressure drop in the primary loop is constant. As the load increases in the secondary and the two-way valves modulate open in response, the secondary pumps run out on their curves corresponding to the reduced pressure, flow increases to the required 2000 gpm. If 2000 gpm is flowing into the secondary, 2000 gpm must return from the secondary. The balanced chillers will only accept 1500 gpm.

To balance the mass flow, the excess 500 gpm must run through the common pipe. The temperature of the 500 gpm in the common pipe is 55F. This blends with the 1500 gpm of 45F supply water, resulting in 2000 gpm of 47.5F blended supply water.

Whenever the flow is greater in the distribution loop than in the production loop, the excess flow in the common pipe is in the direction towards the secondary pumps. The result will always be a blending of the return water with the supply water at a temperature higher than what the chiller produces. What will happen to the temperature control? A higher supply water temperature could mean a loss of humidity control in the Zones. Higher supply water temperatures must be considered during the coil selection process.

Other options can be considered. For example, chiller temperature reset can be employed. Within the limits of the type of machine, chiller temperatures can be reset to a lower temperature to compensate fort he increased load and secondary flows. In essence, more capacity is provided at a lower operating efficiency. The increase in cost of chiller operation due to the lowering of the chiller supply temperature can range from 1 to 3 percent per degree of reset. This is a very desirable alterative, especially when large chillers are in use. The longer the start of a lag chiller can be in use. The longer the start of a lag chiller can be delayed, the better it will perform when it is finally brought on line.

If a small portion of the load requires a fixed temperature, a small chiller in series with the load may also be considered.

• **Production flow greater than distribution -** When incremental chiller capacity is added, the third flow condition occurs **(Fig 6)**. The flow rates through the chillers are again fixed, this time at 3000 gpm. The new load is 875 tons. As we are again seeing a 45°F supply water temperature and a 55°F return water temperature, the flow rate for the secondary pump is 2100 gpm delivered to the secondary and 3000 gpm being pumped in the primary loop, there is an excess flow of 900gpm. The 900 gpm excess must flow through the common pipe at the chilled water supply temperature of 45°F. The 900 gpm common flow blends with the 2100 gpm secondary chilled water return to produce 300gpm at a reduced entering chilled water temperature of 52°F.

All chilled water retuning to the chilled waters plant is blended prior to reaching the chillers. All t of the chillers on line will therefore be receiving the same temperature water at their return. When the system is piped in this manner, the chillers will always be equally loaded. Furthermore, the chillers will always be subjected to their design flow at an equal temperature. Because the chillers are receiving heir design flow rate at 52°F rather than the design temperature of 55 F, the chilers will be "unloaded" at the ratio of:

$$100\left[1 - \frac{DT_R - AT_R}{DT_W - DT_S}\right] = \text{Percent chiller loading}$$

DT_R = Design return temperature

AT_R = Actual return temperature

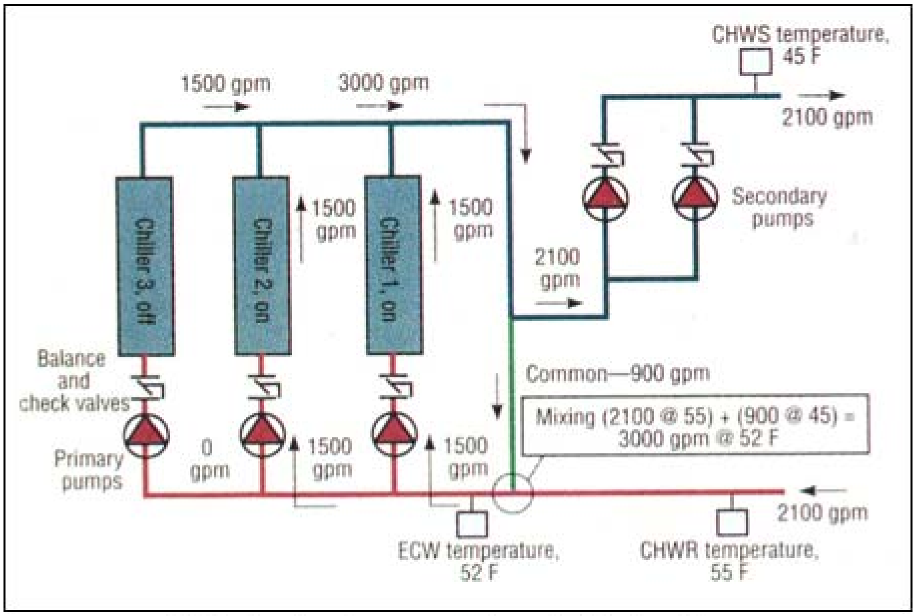
DT_W = Design water temperature

DT_S = Design supply temperature

$$100\left[1 - \frac{(55F - 52F)}{(55F - 45F)}\right] = 70 \text{ Percent}$$

When piped with a common, the flow through the chillers is a constant. In our example, the two steps are either 50 percent flow (1500 gpm) or 100 percent flow (3000 gpm). The secondary flow is very close to a "linear function." As the two-way valves modulate in response to a varying load, the flow follows directly. The more chillers in the plant, the smaller the steps. If we incorporate chillers in a variety of sizes, the incremental steps are smaller. The additional chillers and variety of chiller sizes can produce a curve that is nearly linear. When the chiller plant is designed to produce a near-linear flow function, the supply temperature rises, the secondary is minimized, and the increased supply waters temperature lasts for a shorter duration. This is a very critical factor in design regions where humidity control is a concern.

Chiller sequencing can be as simple as manually turning on and off a second chiller. It can also become complex when multiple chillers of different sizes, types, and efficiency are installed. The chiller manufacturer is usually best suited to provide the information and instrumentation to effectively stage and destage its product.



6. Production flow greater than distribution flow

Conceptually, the designer needs to determine the best combination of chillers that will meet the flow demand. This is often done by calculating the Btuh consumed **(Fig 7)**. Instrumentation must be provided to determine:

Secondary supply water temperature, $T_{S.S}$.

Secondary return water temperature, $T_{S.R}$

Secondary flow, F_S

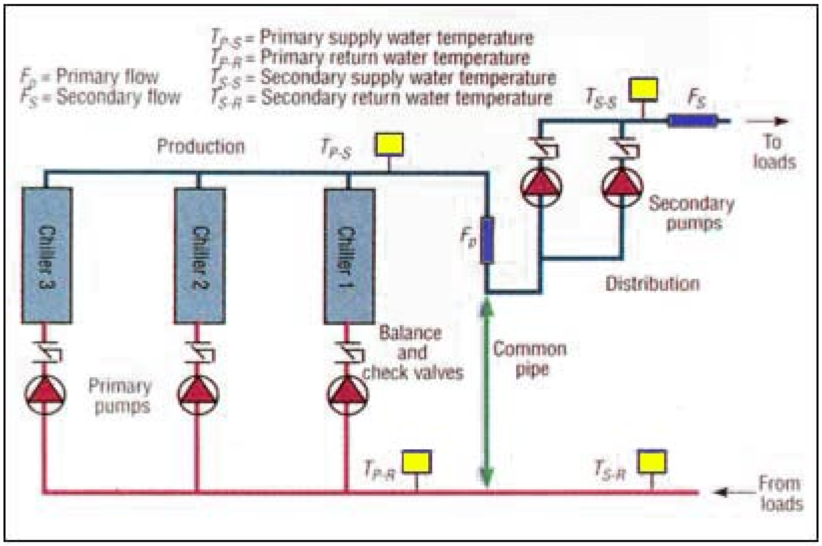
The amount of cooling Btuh produced is also valuable information. The necessary instrumentation is:

Primary supply water temperature (chiller leaving), $T_{P.S}$

Primary return water temperature (chiller entering), $T_{P.R}$

Primary flow (chiller flow), F_P

An algorithm is then prepared, which utilizes the best combination of chillers for the actual load. The measurement of flow and direction in the common pipe does not provide the operator with enough information to determine the time of sequencing. The same can be said for the measurement of flow in the secondary system.



7. Chiller sequencing

Control valves and actuators

Control valves are a critical part of the variable volume hydronic system. The control valves functions are to vary flow properly through the water coil in response to a variety of building load conditions. Because of their critical nature, great care must be taken when selecting control valves so they perform properly. Undersized valves may provide insufficient capacity while oversized valves provide poor control. Equal percentage type valves are typically applied to cooling and heating coils because of their favorable flow characteristic. They provide a high degree of control accuracy with wide variations in pressures, flow the heat transfer characteristic of a cooling coil, the change in stem position almost provides a nearly linear change in heat transfer if an equal percentage control valve is used.

Knowing the maximum differential pressure across the valve at any flow is an important criterion for valve selection for two reasons **(Fig 7)**:

- Choosing the appropriate actuator
- Avoiding valve noise and cavitation in valves and piping

In variable volume systems, the selected valve actuator must be capable of closing the valve against the maximum pump head pressure **(Fig 8)**. In variable volume closed loop systems, all loads could be reduced causing the pump to ride back on the pump curve. As the pump rides back on the curve, the head being produced by the pump increases. The control valve actuator must be strong enough to continue modulating the valve closed as the pressure increases. In the worst case, this could be the "shutoff pressure" being produced by the pump. Improper actuator sizing may result in some water passing through the valve and coil, which add to low return water temperatures.

Excessive valve noise and valve cavitation are caused by high liquid velocities. As water passes through a valve, it is accelerated. With the increase in velocity pressure, the static pressure falls. If it droops below the vapor pressure, bubbles form. Then immediately downstream of the valve, velocity decreases and the static pressure increases, causing the bubbles to collapse. The result is noises coupled with excessive wear in the valve body, throttling mechanisms, and sometimes in the downstream piping where this takes place.

When variable speed secondary pumping is employed, valves are normally subjected to less differential pressure. In this mode the pump follows the characteristic control curve **(Fig. 2)** This results in reduced differential pressure when conditions are at less than maximum design. Care must still be taken if an across-the-line (ATL) adjustable frequency drive bypass (electrical bypass) is used. When the system function in the ATL mode, the pump acts like a constant-speed pump. It is now following the pump flow-head curve at a much higher pressure. The designer may overlook this condition. Under low flow conditions, the valves may lift up as described earlier. If the selected actuators have in adequate holding power at the higher head shut-off condition, the valves may lift up as previously described.

Three-way valves have little use invariable volume pumping systems. Three-way valves bypass unused chilled water past the load; this results in multiple problems. Bypassed water lowers the return water temperature and increases flow rates. As a result, additional chillers and pumps are brought on-line to provide flow, rather than to control the cooling load.

A circumstance where three-way valves can be used (carefully) in the system is to maintain minimum flow as a means of pump protection. As a rule of thumb, the Hydraulic Institute[4] and ITT Fluid Handling recommend a minimum flow rate of 20 percent of the best efficiency point flow rate. To limit the radial and axial loads and shaft deflection, our recommendation is to maintain a minimum flow rate of 25 percent of the best efficiency point flow rate.

The Hydraulic Institute and ITT Fluid Handling also recommend limiting the temperature rise of the fluid flowing through the pump to 15F. (This is not to be confused with the system temperature rise.) When a pump is operating in a low flow condition, the temperature rise occurs in a small volume of water. If an insufficient volume of water flows through a pump, the inefficiency of the pump is converted to heat. In time, the heat build-up can cause the fluid to vaporize and eventually result in cavitation.

To calculate the flow rate for a 15F temperature rise use the following equation :

$$Q = \frac{P}{2.95 \times C_p \times S}$$

Where:

Q = Minimum flow rate, gpm

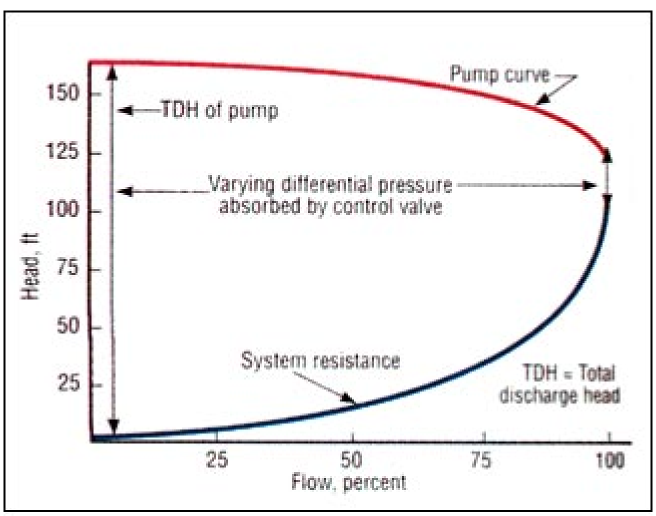
P = Input power at the minimum flow rate, hp. Assume that the shutoff hp is approximately equal to the minimum flow rate hp

2.95 = Constant, (hp - lb - min - F) /(Btu.gal)

C_P = Specific heat, Btu/(lb.F)

S = Specific gravity

When careful attention is given to these fundamentals, energy savings will be achieved and design system ΔT will be achieved. As system characteristics change due to load changes, the aging of the piping and coils, or even as coils get dirty, system ΔT may suffer. There is no substitute for knowledgeable operators and proper thermostats and the lowering of supply air temperatures will only add to the inefficiency. Unless problems are cured only patches can be applied to treat symptoms. "Fixes" consume additional energy. The only way to keep a properly designed HVAC system operating efficiently is to maintain it in the appropriate manner.



8. Control valve differential pressure

Conclusion

To optimize system performance and design flexibility, the designer must select the type of distribution piping and pumping configuration that will best meet the needs of the client.

The following are the four basic types of pump0ing systems in use today.

- Primary-secondary
- Primary-secondary-tertiary
- Primary-secondary zone
- Primary variable speed

These pumping methods will be discussed in a second article in this series.

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