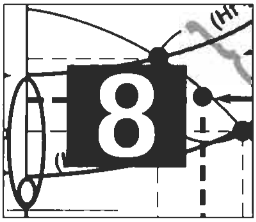

The System Curve

Demand may increase or decrease with time

The system controls the pump

All pumps must be designed to comply with or meet the needs of the system. The needs of the system are recognized using the term 'Total Dynamic Head', TDH. The pump reacts to a change in the system. For example, in a small system, this could be the changes in tank levels, pressures, or resistances in the piping. In a large system, an example would be potable water pumps designed for an urban area consisting of 200 homes. If after 5 years the same urban area has 1,000 homes, then the characteristics of the system have changed. New added piping adds friction head (Hf). There could be new variations in the levels in holding tanks, affecting the static head (Hs). The increase in flow will affect the pressure head (Hp), and the increased flow in old, scaled piping will change the velocity head (Hv). New demands in the system will move the pumps on their curves. Because of this, we say that the system controls the pump. And if the system makes the pump do what it cannot do, then the pump becomes problematic, and will spend too much time in the shop with failed bearings and seals.

The elements of the Total Dynamic head (TDH)

The Total Dynamic Head (TDH) of each and every pumping system is composed of up to four heads or pressures. Not all systems contain all four heads. Some contain less than four. They are:

1. **Hs** – the static head, or the change in elevation of the liquid across the system. It is the difference in the liquid surface level at the suction source or vessel, subtracted from the liquid surface level where the pump deposits the liquid. The Hs is measured in feet of elevation change. Some systems do not have Hs or elevation

change. An example of this would be closed systems like water in the radiator of your car. Another example would be a swimming pool re-circulating filter pump. The vessel being drained (the pool) is the same level as the vessel being filled (the pool). If there is a difference in elevation across the system, this difference is recorded in feet and called Hs.

2. **Hp** – the pressure head, or the change in pressure across the system. It is expressed in feet of head. The Hp also may, or may not exist in every system. If there is no pressure change across the system, then forget about it. An example of this would be a recirculated closed loop. Another example would be if both the suction and discharge vessels have the same pressure. Think of a pump draining a vented atmospheric tank, and filling a vented atmospheric vessel. The atmospheric pressure would be the same on both vessels, thus no Hp. If Hp is present, then note the pressure change and employ it in the following formula. Sometimes, it is necessary to use a pump to drain a tank at one pressure (like atmospheric pressure), while filling a tank that might be closed and pressurized. Think of a boiler feed water pump where the pump takes boiler water from the deaerator (DA) tank at one pressure, and pumps into the boiler at a different pressure. This is a classic example of Hp. The formula is:

$$\mathbf{Hp} = \frac{\Delta \mathbf{psi} \times \mathbf{2.31}}{\mathbf{sp.\ gr.}}$$

where: **Δpsi** = boiler pressure – DA tank pressure

3. **Hv** – the velocity head, or the energy lost into the system due to the velocity of the liquid moving through the pipes. The formula is:

$$\mathbf{Hp} = \frac{\mathbf{V^2}}{\mathbf{2g}}$$

where: V = velocity of the fluid moving through the pipe measured in feet per second, and **g** = the acceleration of gravity, 32.16 ft/sec^2

AUTHOR'S NOTE

Hv is normally an insignificant figure, like a fraction of a foot of head or fraction of a psi, which can't be seen on a standard pressure gauge. But you can't forget about it because it is needed to calculate the friction head. If the Hv converts to a pressure that can be observed on a standard pressure gauge, like 6 or 10 psi, the problem is the inadequate pipe diameter.

4. **Hf** – friction head is the friction losses in the system expressed in feet of head. The Hf is the measure of the friction between the pumped liquid and the internal walls of the pipe, valves, connections and accessories in the suction and discharge piping. Because the Hv and the Hf are energies lost in the system, this energy would never reach the final point where it is needed. Therefore these heads must be calculated and added to the pump at the moment of design and specification. Also it's necessary to know these values, especially when they're significant, at the moment of analyzing a problem in the pump. The Hf and the Hv can be measured with pressure gauges in an existing system (see the Bachus & Custodio formula in this chapter). If the system is in planning and design stage and does not physically exist, the Hf and Hv can be estimated with pipe friction tables (ahead in this chapter). The Hf formula for pipe is:

$$\mathbf{Hf} = \frac{\mathbf{Kf \times L}}{\mathbf{100}}$$

where: **Kf** = friction constant for every 100 ft of pipe derived from tables **L** = actual length of pipe in the system measured in feet.

The Hf formula for valves and fittings is:

$$\mathbf{Hf} = \mathbf{K \times Hv}$$

where: **K** = friction constant derived from tables, and **Hv** = $\frac{\mathbf{V^2}}{\mathbf{2g}}$

The sum of these four heads is called the total dynamic head, TDH.

$$\mathbf{TDH = Hs + Hp + Hf + Hv}$$

The reason that we use the term 'dynamic' is because when the system and the pump is running, the elevations, pressures, velocities, and friction losses begin to change. In other words, they're dynamic.

AUTHOR'S NOTE

When the system is designed, the engineer tries to find a pump that's BEP is equal to or close to the system's TDH (the system's TDH ≅ BEP of the pump). But once the pump is started, the system becomes very dynamic, leaving the poor pump with a static BEP.

The purpose of the system curve is to graphically show the elements of the TDH imposed on the pump curve. The system curve shows the complete picture of the dynamic system. This permits the purchase, installation, and maintenance of the best pump for the system. The system curve is most useful when mated with the pump family curve. This is why the family curves are the most useful to the design engineer, the maintenance engineer, and purchasing personnel.

At the beginning of this chapter, we stated that the system governs the pump. This being the case, the pump always operates at the intersection of the system curve and the pump curve. And the goal of the engineer is to do everything possible to assure that this point of intersection coincides with the pump's BEP. Consider the following graph (Figure 8–1).

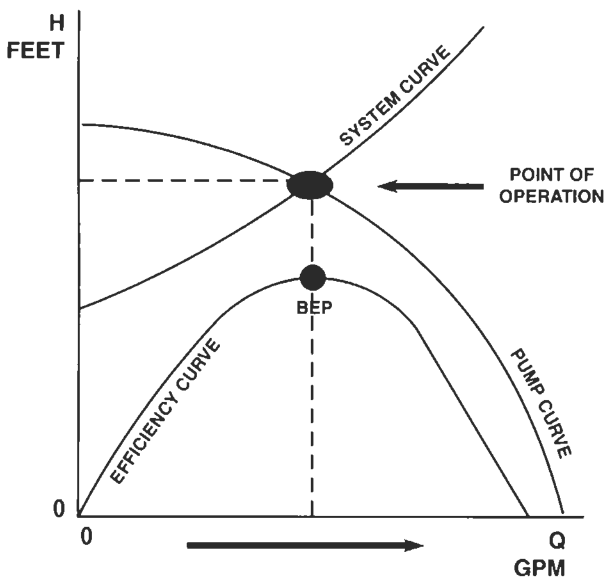

Figure 8-1

It is necessary to understand the TDH and it's components in order to make correct decisions when parts of the system are changed, replaced, or modified (valves, heat exchangers, elbows, pipe diameter, probes, filters, strainers, etc.) It's necessary to know these TDH values at the moment of specifying the new pump, or to analyze a problem with an existing pump. In order to have proper pump operation with low maintenance over the long haul, the BEP of the pump must be approximately equal to the TDH of the system.

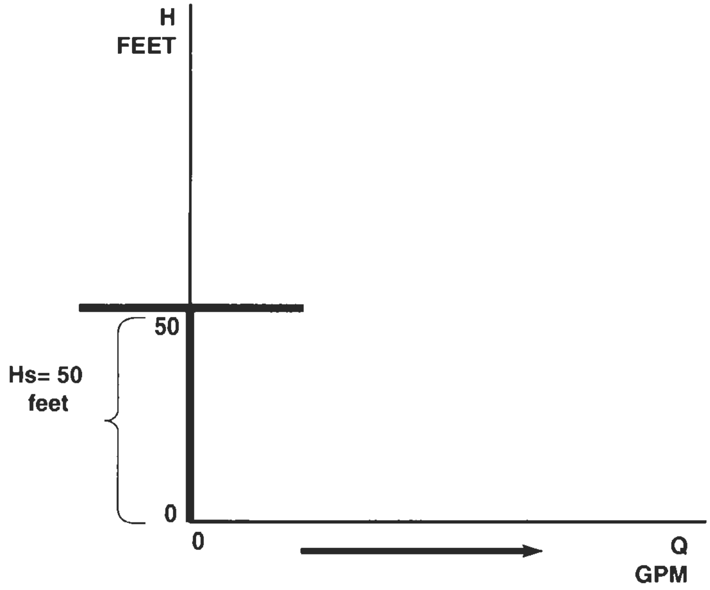

Figure 8–2

Determining the Hs

Of the four elements of the TDH, the Hs and the Hp (elevation and pressure) exist whether the pump is running or not. The Hf and the Hv (friction and velocity losses) can only exist when the pump is running. This being the case, we can show the Hs and the Hp on the vertical line of the system curve at 0 gpm flow. The Hs is represented as a T on the graph below. For example, if the pump has to elevate the liquid 50 feet, the Hs is seen in Figure 8–2.

Determining the Hp

The Hp also can exist with the pump running or off. We can represent this value with an O or oval on the vertical line of the below graph. The Hp is added to and stacked on top of the Hs. Let's say that our system is pumping cold water and requires 50 ft of elevation change and 10 psi of pressure change across the system. Now, our pump not only has to lift the liquid 50 ft, but it must also conquer 23 ft of Hp. Remember that 10 psi is 23.1 ft of Hp:

$$Hp = \frac{10 \text{ psi} \times 2.31}{\text{sp. gr.}}$$

Here is the system curve showing Hs and Hp (Figure 8–3).

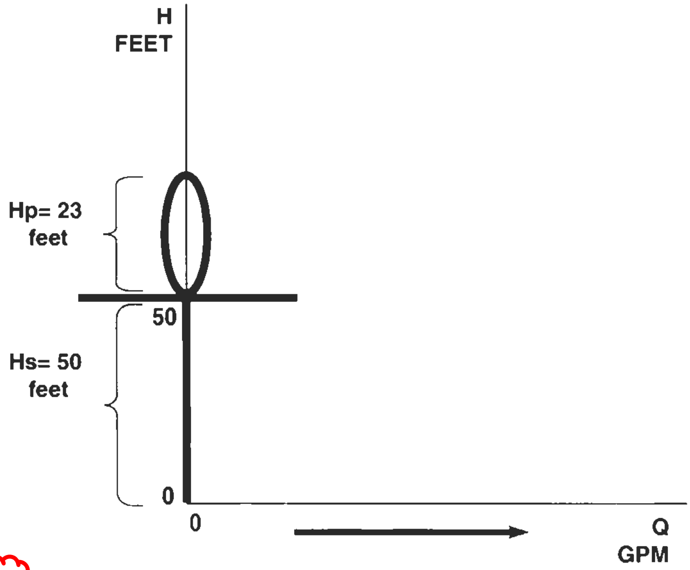

Figure 8-3

Calculating the Hf and Hv

Continuing with our example, before starting the system we already know that the pump must comply with 73 ft of static and pressure head. At the moment of starting the pump, the elements of Hf and Hv come into play as flow increases. Remember that Hf and Hv work in concert because the Hv is used to calculate the Hf. These values can be calculated using a variation on the Affinity Laws. The Affinity Laws state that the flow change is proportional to the speed change ($Q \alpha N$), and that the head change is proportional to the square of the speed change ($H \alpha N^2$). Therefore algebraically, the head change is proportional to the square of the flow change ($\Delta H \ \alpha \Delta Q^2$). Also, the friction head change and velocity head change are proportional to the square of the change in flow (ΔHf and $\Delta Hv \ \alpha \Delta Q^2$). On the system curve, the Hf and Hv begin at 0 gpm at the sum of Hs and Hp, and rise exponentially with the square in the change in flow. On the graph, it is seen as in Figure 8–4.

In a perfect and static world, we could apply the Affinity Laws to calculate the Hf and Hv, and calculate how the Hf and Hv change by the square of the change in flow. Well, the world is neither perfect, nor is it static. And, pipe is not uniform in its construction.

Some engineers (who normally are precise and specific) are charged with the task of approximating the friction losses (the Hf and Hv) in

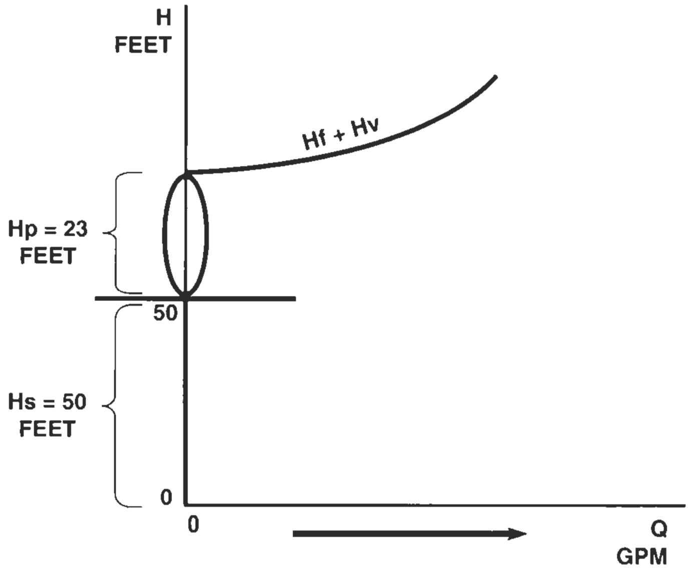

Figure 8-4

piping before the system exists. In the design stage, when the system exists only in drawings and plans, the civil engineer knows the proposed heads and elevations. And, he knows the proposed pressures in the system under construction. But he does not know, nor can he calculate the friction and velocity losses with the variations in pipe construction. Over the years, civil engineers have found refuge in the 'Hazen and Williams' Formula, and also the 'Darcy/Weisbach' Formulas for estimating the friction (Hf), and velocity (Hv) losses in proposed piping arrangements.

The Hazen and Williams formula

Mr. Hazen and Mr. Williams were two American civil engineers from New England in the early 1900s. In those days, piping used to carry municipal drinking water was ductile iron, coated on the inside diameter with tar and asphalt. The tar coating gave improved flow characteristics to the water compared to the flow characteristics of the ductile iron piping without the coating. The engineers Hazen and Williams derived their formula, a variation on the Affinity laws, and introduced a correction factor for friction losses of about 15%. Simply put, their formula is: $\Delta Hf \propto \Delta Q^{1.85}$. The H & W method is the most popular among civil and design engineers. The formula is empirical, simple, and easy to apply. It is the method to calculate friction losses that is required by most of the municipal water agencies. The H & W formula assumes a turbulent flow of water at ambient temperature. As an approximation, it is most precise with velocities between 3 and 9 feet per second in pipes with diameters between 8 and 60 inches.

The Darcy/Weisbach Formula

This formula is another variation on the Affinity Laws. Monsieur's Darcy and Weisbach were hydraulic civil engineers in France in the mid 1850s (some 50 years before Mr. H & W). They based their formulas on friction losses of water moving in open canals. They applied other friction coefficients from some private experimentation, and developed their formulas for friction losses in closed aqueduct tubes. Through the years, their coefficients have evolved to incorporate the concepts of laminar and turbulent flow, variations in viscosity, temperature, and even piping with non uniform (rough) internal surface finishes. With so many variables and coefficients, the D/W formula only became practical and popular after the invention of the electronic calculator. The D/W formula is extensive and complicated, compared to the empirical estimations of Mr. H & W.

AUTHOR'S NOTE

The merits of the Hazen and Williams's formula versus the Darcy/Weisbach formula are discussed and argued interminably among civil engineers. It is our opinion that if a student learned one method from his university professor, normally that student will prefer to continue using that method. The two formulas are variations on the Affinity Laws, which are probably equally adequate to 'guestimate' the friction losses in non-uniform piping. Both the H & W and D/W formulas try to approximate the friction losses (Hf and Hv) in a piping system that physically does not exist. It doesn't exist because these calculations occur during the design phase of a new installation. But in this phase, it is necessary to begin specifying the pumps, although based on incomplete information. It's somewhat like a blind man throwing an invisible dart at a moving dartboard.

It really doesn't matter which formula (the H & W or the D/W) one prefers to use in calculating friction losses (Hf and Hv) in a pipe. Both formulas have deficiencies. Both formulas assume that all valves in the system are completely and totally open (and this is almost never the case). Both formulas assume that all instructions on construction and assembly (the pipes, supports, connections, valves, elbows, flanges and accessories) are followed to the letter (practically never). Both formulas assume that there are no substitutions during construction and assembly due to back orders and delivery shortages (Yeah, right!). Neither formula considers that scale forms inside the piping and that the interior diameters, thus Hf and Hv, will change over time. Neither formula considers that control valves are constantly manipulated, nor that filters clog. One formula doesn't consider that viscosity, thus stress and friction, can change with temperature or agitation. And both formulas are based on municipal water with piping adequate for that service only.

In recent years new equipment has been invented, chemical processes, piping materials, valve designs, and new technologies not considered when these formulas were developed with cold water in the 19th century. There is a need to measure the actual losses once an industrial plant is commissioned and operations begin. The authors of this book have developed a formula that permits the measurements of these losses in a live functioning system. Here it is:

The Bachus & Custodio Formula

Also called the DUH!! Factor. You'll need to gather information from pressure gauges mounted to the existing system. With the previously mentioned formulas, the Hf and the Hv are estimated in the initial phase when everything is new. The Bachus & Custodio method measures the exact Hf and Hv in any existing system. It doesn't matter when it was built.

The **Bachus & Custodio** Formula is the following:

$$\boxed{\textbf{System Hf and Hv} = \left(\frac{(\Delta PDr - \Delta PDo) + (\Delta PSr - \Delta PSo) \times 2.31}{\text{sp. gr.}}\right)}$$

where: **ΔP** = pressure differential from an upstream to a downstream gauge in a section of pipe **Dr** = Discharge Running, the discharge piping with the pump running. **Do** = Discharge Off, the discharge piping with the pump not running. **Sr** = Suction Running, the suction piping with the pump while running. **So** = Suction Off, the suction piping with the pump not running. **2.31** = conversion factor between psi and feet of elevation **sp.gr.** = Specific Gravity

In general, the Hf and Hv are observed while considering the system. We'll see this further ahead. The Hf and the Hv are the reasons that companies contract civil engineers to design their new plants. Years later, those design parameters have changed due to erosion and other factors. Let's look at the following situation, pumping a liquid from one tank into another.

System example

This system, although simple, with only one pump, is more or less representative of all systems. This system is composed of 180 ft of pipe; 40 ft of 6 inch suction pipe, and 140 ft of 4 inch discharge pipe.

This system piping uses fittings with bolted flanges, see Figure 8–5. The 6 inch elbows have a constant (K value) of 0.280. The 4 inch elbows have a K value of 0.310. The 6 inch gate valves have a K value of 0.09. The 4 inch gate valves have a K value of 0.15. The 4 inch globe valve

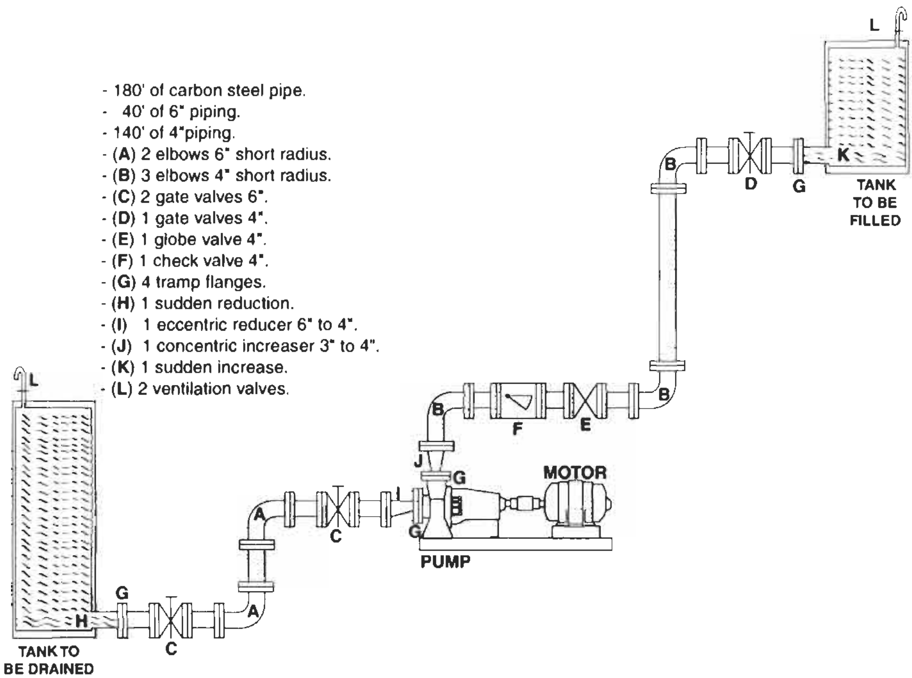

Figure 8–5

has a K value of 6.4. The 4 inch check valve has a K value of 2.0. The 4 inch tramp flanges have a K value of 0.033. The 3 inch tramp flange has a K value of 0.04. The sudden reduction has a K value of 0.5. The 6 to 4 inch eccentric reducer has a K value of 0.28. The 3 to 4 inch concentric increaser has a K value of 0.192. The sudden increase has a K value of 1.0. The required flow is 300 gallons per minute. The constants mentioned (K) are given values provided by manufacturers and can be found on charts provided by different organizations.

The goal is to apply the formulas, the K values, and the pipe and connections friction values to determine the Hf and Hv, plus the Hs and Hp, and then the TDH, total dynamic head in the system. Then we can specify a pump for this application.

One component of the TDH is the Hs, the static head. In this example the surface level in the discharge tank is 115.5 ft above the pump centerline. The surface level in the suction tank is 35.5 ft above the pump centerline. The ΔHs, by observation is 80 ft. See Figure 8–6.

Another component of the TDH is the Hp, pressure head. We can see in Figure 8–7 that both tanks have vent valves. These two vessels are exposed to atmospheric pressure, which is the same in both tanks. So by simple observation, pressure head doesn't exist. ΔHp = 0.

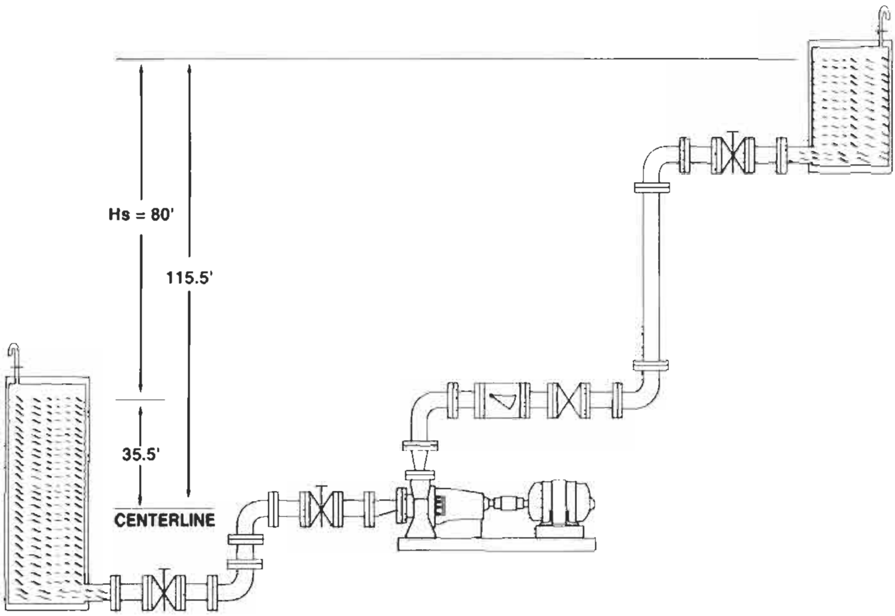

Figure 8-6

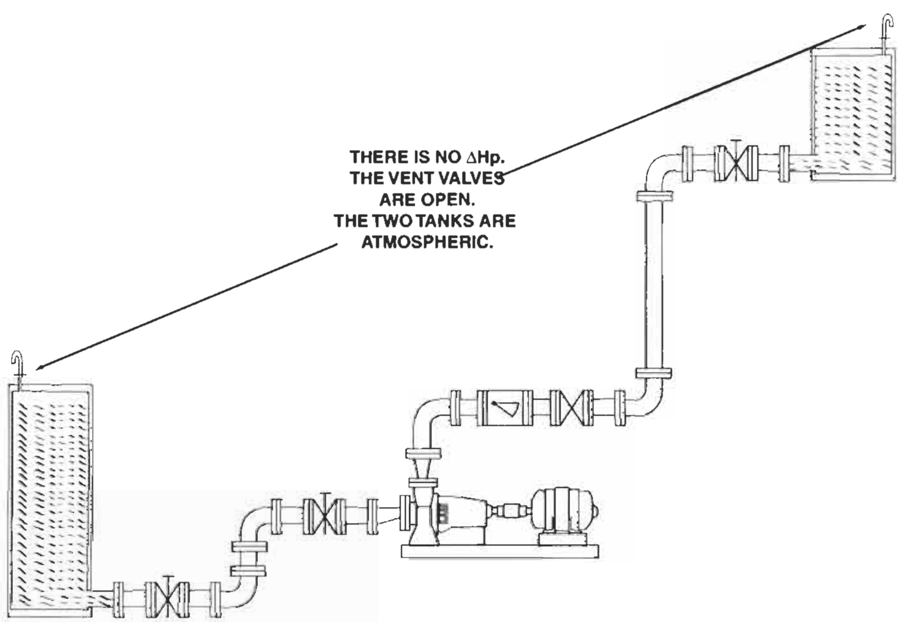

Figure 8-7

AUTHOR'S NOTE

The following is not very entertaining to read. The authors have included this section so that the readers can gain an appreciation for the detailed work of the design engineer on calculating the frictions and velocities in a piping system. Admittedly, there are computer programs today that will perform these calculations in a flash. But 30 years ago and before, these calculations were done with mental software (the engineer's brain), a mechanical computer (a slide rule), and a manual printer (pencil and paper).

In most cases, the design engineer and architect begin with an open field of rabbits and weeds. Two years later there is a hotel, gasoline station, or paint factory built on the site of the open field. And the day that the new owners open their hotel, or start mixing paints, most of the new pumps are running within 5% of their best efficiency points. The pumps were mostly designed correctly into their new systems, and run for various years without problems, an amazing feat of engineering, math, and art ... before computers.

Remember that we're calculating the TDH. Two elements of the TDH, the Hs and the Hp, were determined mostly by observation of the system and drawings. The remaining two elements, the Hf and the Hv, are the most illusive and difficult to calculate. Yet, they determine how and where the pump will operate on its curve. Continue reading.

Using the formulas, the K values, and the pipe schedule tables found in the Hydraulic Institute Manual, ($V_{suction}$ = 3.33 ft/sec for 6 inch pipe @ 300 GPM and $V_{discharge}$ = 7.56 ft/sec for 4 inch pipe @ 300 GPM) or other source, we can estimate or calculate the friction and velocity heads in the system. Because the Hv is used to calculate the Hf, we'll begin with the Hv. The formula is:

System Hv = Hv suction + Hv discharge

= $V^2 \div 2g$ suction + $V^2 \div 2g$ discharge

= $3.33^2 \div 64.32 + 7.56^2 \div 64.32$

= 0.172 ft. suction + 0.888 ft. discharge

System Hv = 1.06 feet

The **Hv suction** and **discharge** values will be used in the **Hf** formula.

System Hf = Hf pipe + Hf elbows + Hf valves + Hf tramp flanges + Hf other

Taking this formula in groups, we begin with the Hf pipes.

Hf system piping = Hf suction piping+ Hf discharge piping.

$= (\text{K suction} \times \text{L}) \div 100 + (\text{K discharge} \times \text{L}) \div 100$

$= (4.89 \times 40) \div 100 + (.637 \times 140) \div 100$

$= 1.956 + 0.891$

Hf system piping = 2.848 feet

Now we calculate the Hf in the elbows

The formula is:

Hf elbows = Hf suction elbows + Hf discharge elbows

$= 2 \times 0.280 \times 0.172 + 3 \times 0.310 \times 0.888$

$= 0.096 + 0.82$

Hf elbows = 0.916 feet

Next, we calculate the Hf for the valves

There are 5 valves in all. There are two 6 inch gate valves in the suction pipe. There is a 4 inch gate valve, a 4 inch globe valve, and a 4 inch check valve in the discharge pipe. The formula is:

Hf system valves = Hf suction valves + Hf discharge valves

$= K_{6"\,gate} \times Hv_{suction} + K_{4"\,gate} \times Hv_{disch.} + K_{4"\,check} \times Hv_{disch.} + K_{4"\,globe} \times Hv_{disch.}$

$= (2 \times .09 \times 0.172) + (1 \times 0.16 \times 0.888) + (1 \times 2 \times 0.888) + (1 \times 6.4 \times 0.888)$

$= 0.031 + 0.142 + 1.776 + 5.683$

Hf system valves = 7.632 feet

Next we calculate the Hf in the tramp flanges in the system

A tramp flange is an unassociated flange or union. In the friction tables, valves, elbows, and other fittings are categorized as to whether they are flanged or screwed. This means they connect to the piping either by a bolted flange, or screwed into the pipe with male and female threading. For example, the friction losses through a 2 inch flanged elbow, or a 4 inch check valve, already takes into account the losses at the entrance and exit port fittings. Then there are unassociated 'tramp' flanges and unions. Examples would be unions between two lengths of pipe, or between a pipe and a tank, or between a pipe and a pump. They must be calculated because there is friction (and energy lost) as the fluid passes through a union. In our simple system, there is a 6 inch tramp flange on the suction pipe with the tank, and a 4 inch tramp with the pump. There's a 3 inch tramp flange at the pump discharge and another 4 inch tramp at the discharge tank. The formula is:

Hf system tramp flanges = Hf suction tramps + Hf discharge tramps

$$= K_{6''} \times Hv_{suct.} + K_{4''} \times Hv_{suct.} + K_{4''} \times Hv_{disch.} + K_{3''} \times Hv_{disch.}$$

$$= (0) + (0.033 \times 0.172) + (0.033 \times 0.888) + (0.04 \times 0.888)$$

$$= 0 + 0.005 + 0.029 + 0.035$$

Hf system tramp flanges = 0.007 feet

Admittedly, Hf of 0.007 foot is an insignificant number. Think of it this way. With only one pump and less than 200 ft of pipe in our simple system, there are four tramp unions. Imagine an oil refinery with 20,000 pumps and thousands of miles of pipe and equipment on site. Imagine the number of tramp flanges in the fire water system in a skyscraper building. In a real set of circumstances the Hf values through tramp flnages unions could be significant, and they would have to be calculated to specify the correct pumps.

Last, we need to calculate the Hf losses through other connections in the piping

There is a sudden reduction in the suction between the tank and the piping. There is an eccentric 6-to-4 reducer between the suction pipe and the pump. There is a concentric 3-to-4 increaser from the pump back into the piping, and a sudden enlargement going into the discharge tank. The formula is:

$$\text{Other Hf} = Hf_{\text{sudden reduction}} + Hf_{\text{eccentric reducer}} + Hf_{\text{concentric increaser}} + Hf_{\text{sudden enlargement}}$$

$$= (0.05 \times 0.172) + (0.28 \times 0.172) + (0.192 \times 0.888) + (1 \times 0.888)$$

$$= 0.086 + 0.048 + 0.170 + 0.888$$

Other Hf = 1.192 feet

Now we have all the information to calculate the Hf in the system and then the TDH of the system. Once again:

System Hf = Hf pipe + Hf elbows + Hf valves + Hf flanges + Hf other

$$= 2.848 \text{ ft} + 0.916 \text{ ft} + 7.632 \text{ ft} + 0.007 \text{ ft} + 1.192 \text{ ft}$$

$$= 12.595 \text{ ft}$$

Consider all the mathematical gyrations required just to determine the Hv and Hf. This is a lot of math for one pump. Imagine the work to specify pumps for a paper mill or beer brewery or municipal water system. Now you can see why governments and pharmaceutical companies contract consulting engineering companies to do this work and specify the pumps. Finally, we can calculate the **TDH** of the system:

$$\begin{aligned}\mathbf{TDH} &= \mathbf{Hs + Hp + Hf + Hv} \\ &= \mathbf{80\ ft + 0\ ft + 12.595\ ft + 1.06\ ft} \\ &= \mathbf{93.655\ ft}\end{aligned}$$

This system requires a pump with a best efficiency point (BEP) of 94 feet at 300 gallons per minute. If this is a conventional industrial centrifugal pump with a BEP of 94 feet, the shut-off head should be approximately 110 feet. And if the motor is a standard NEMA four-pole motor spinning at about 1800 rpm, the diameter of the impeller should be approximately 10.5 inches. If this pump were bought off the shelf from local distributor stock, it would probably be a 3 × 4 × 12 model end-suction centrifugal back pullout pump with the impeller machined to about 10.5 inches before installing the pump into the system. And that's the way it is done.

If the system already exists and the equipment is running, we can recover the **Hf** and **Hv** from gauges using the Bachus & Custodio Method, and forget about all those calculations. See Figure 8–8 opposite, with the corresponding elevations and placement of pressure gauges installed into the piping numbered 1 through 5.

In this system drawing, pressure gauges 1, 2, and 3 are in the suction piping. Gauges 4 and 5 are in the discharge piping. With the system and pump turned off, we would open the vent valves on both the suction and discharge tanks, this assures that both sides of the system are atmospheric and cancels the **Hp**. The discharge tank and all piping should be full with water for the test, or if required, the pumped liquid. Remember that gauge readings will be adjusted by the specific gravity. Expel all air bubbles in the piping. Some pumps have a little petcock valve to allow expelling any trapped air in the volute. On the pump, conventional stuffing boxes can also trap air. This must be expelled too. Vertical valve stems in the piping can trap air. Loosen the packing to expel this trapped air. This is done so that there is a complete column of liquid from the top to the bottom of the system. Air pockets and bubbles might cause inaccurate pressure gauge readings. All valves in the column (including the check valve) should be opened, except for the gate valve between gauges 1 and 2. It should be closed to hold the column of liquid and prevent draining the line.

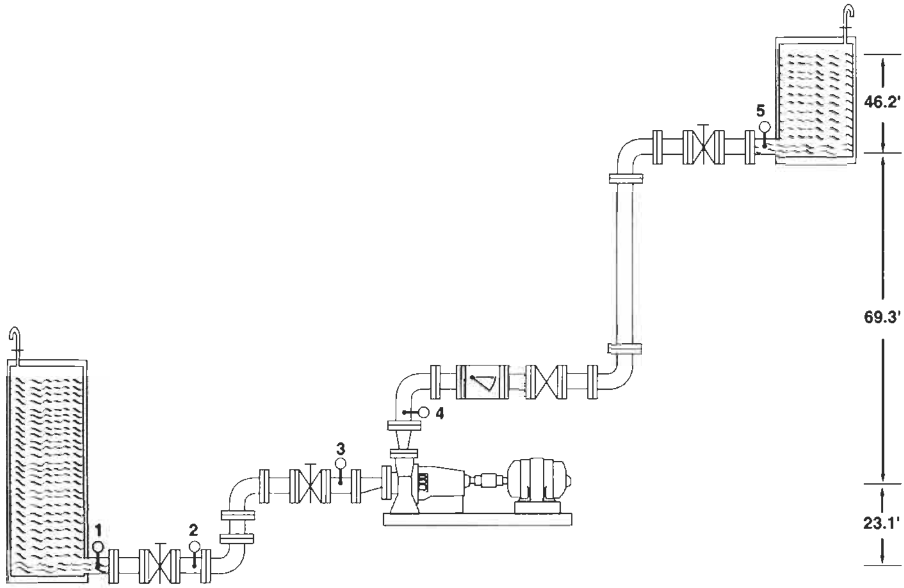

Figure 8-8

Here's a quick review of the Bachus & Custodio Formula:

System Hf and Hv = [(ΔPDr – ΔPDo) + (ΔPSr – ΔPSo) × 2.31] ÷ sp. gr.

Let's take our readings with water as the test liquid just to keep the conversions simple. With the system and pump off, note that gauge 5 should be reading 20 psi. This is because it is 46.2 feet below the surface level in the discharge tank. Confirm that gauge 4 is reading 50 psi. It is 115.5 feet deep into the column. The difference between gauges 4 and 5 is 30 psi. The ΔPDo = 30 psi.

In the suction line, note that gauge 3 is also reading 50 psi. It also has 115.5 feet of liquid elevation on it. Pressure gauge 2 should read 60 psi because it is 138.6 feet deep into the column. This indicates that the ΔPSo = 10 psi.

Gauge 1, on the other side of the closed valve, is reading the elevation in the suction tank. This gauge should be reading 25 psi because it is 58.6 feet deep into its column.

Now, open the gate valve between gauges 1 and 2. Start the pump motor, and relieve the check valve if it is being mechanically held open. Permit the pump to run a few minutes to stabilize, relieving any surging. We'll continue to note pressure gauge readings with the system functioning.

Because all valves are now open, gauge 1 becomes our upstream gauge on the suction line. With the pump running all activity on the suction side of the pump is separated from the activity on the discharge side of the pump. Gauge 1 continues to read 25 psi. Gauge 2 should also record 25 psi. Gauge 3 should now be reading 15 psi, because this gauge is 23.1 feet above gauges 1 and 2.

However, gauge 3 is recording 13 psi (it should be reading 15 psi) with the system running. The ΔPSr is 12 psi.

AUTHOR'S NOTE

Gauges 1 and 2 should be reading the same pressure with the system running, as gauge 1 was reading with the system off. If you're using precision digital pressure instrumentation gauges, gauge 2 might possibly record a fraction psi less. This is because gauge 2 is now recording minute losses between the tank and the gauge including losses through the opening into the pipe and the losses through the gate valve.

If there should be a divergence in the readings of the two gauges, something is out of control. There might be an obstruction at the tank drain line or maybe the gate valve is not totally open. Maybe the level has dropped in the tank. Maybe the vent valve on the tank top is not open. Maybe the gauges need calibration. Send them to a calibration shop a couple of times per year. But, isn't it interesting how much more you know about your system after learning to interpret the pressure gauges. Who is responsible for specifying, selling, and buying pumps without adequate instrumentation?

Now we consider the pump. We've already discussed in this book that the pump takes the energy that the suction gives it, the pump adds more energy, jacking the energy up to discharge pressure. In this case the pump is designed with a BEP of 94 feet, which also is the TDH of the system. The 94 feet indicate that the pump can generate about 40 psi at 300 gpm ($94 \div 2.31 = 40.6$ psi if the liquid is water). This is confirmed with a flow meter and a pump curve. The suction pressure is 13 psi. The discharge pressure gauge (4 gauge) should be reading 53 psi ($40 + 13$).

AUTHOR'S NOTE

The pump's discharge pressure is a function of the suction pressure. Regrettably, most pumps in the world don't have a gauge reading suction pressure. In our example here, if our pump is generating less than 40 psi, the pump is operating to the right of its BEP, and is losing efficiency. Was the pump assembled correctly? Was it repaired correctly, with all parts machined to their correct tolerances? Is the motor's velocity correct? Is there a flow meter installed? The pump is always on its curve. If this pump were generating more than 40 or 41 psi, it would be operating to the left of its BEP. Verify the other factors.

With gauge 4 on the pump discharge reading 53 psi, the 5 gauge should be reading 30 psi less, or 23 psi. This is because the 5 gauge is 69.3 feet above gauge 4. However gauge 5, by observation, is only reading 18 psi. Therefore ΔPDr = 35psi (53 – 18). We have all the information we need to insert into the Bachus & Custodio Formula:

$$
\begin{aligned}
\textbf{Hf \& Hv} &= (\Delta PDr - \Delta PDo) + (\Delta PSr - \Delta PSo) \times 2.31 \div \text{sp. gr.} \\
&= (35 - 30) + (12 - 10) \times 2.31 \div \text{sp. gr.} \\
&= 5 + 2 \times 2.31 \div \text{sp.gr.} \\
&= 7 \times 2.31 \div 1.0 \\
\textbf{Hf \& Hv} &= 16.17 \text{ feet}
\end{aligned}
$$

The Bachus & Custodio Formula does not make mistakes. It is not based on models, or experiments developed 150 years ago. It doesn't depend on valves being completely open. It doesn't depend on the specific instructions regarding equipment assembly. It doesn't depend on new piping. It is not based on municipal water. It depends on the actual piping and other system fittings, as they are now, and the next shift, and tomorrow, and next month. If a resistance load changes, it will be registered on the gauges. If the pipe diameter changes, it is recorded on the gauges. If new equipment is added, it is visible on the gauges. The pressure gauges and other instrumentation are the pump's control panel. You wouldn't drive a car without a dashboard. Who is responsible for specifying, selling and buying pumps without adequate instrumentation? Regarding pump failure, problematic seals and bearings that need emergency maintenance: in about 80% of all cases, the pump is telling the operators what the problem is, hours, days and weeks before the failure event occurs. What's really happening is that no one is interpreting the information on the gauges.

Regarding the TDH, isn't it interesting that the Hs and the Hp are determined by simple observation? This detailed discussion on the Hf and Hv probably has the reader ready to throw this book into the garbage. With the Bachus & Custodio Formula, the differential pressure gauge readings on the system with the pump turned off, will cancel any elevation changes (Hs) existing in the system. Exposing both sides of the system to atmospheric pressure cancels the pressure changes (Hp). And then with the system operating and the pump turned on, the further differential gauge readings will record the Hf and Hv that are being lost in the system. Remember too, that the other mentioned resistance approximations, Hazen & Williams, and Darcy/Weisbach, are only valid in the first few hours or days of service. The system begins to change once the pump is turned on and production begins. Operators open and close valves to meter the flow through the pipes. Filters and strainers begin to clog. Inside pipe diameters form scale.

New equipment is installed. Other changes occur with maintenance. The equipment loses its efficiency. Install gauges on your pumps and teach the operators and maintenance personnel to interpret the information.

The dynamic system

Let's continue with system curves. Up to this point, all elevations, temperatures, pressures and resistances in the drawings and graphs of systems and tanks have been static. This is not reality. Let's now consider the dynamic system curve and how it coordinates with the pump curve.

Variable elevations

In the next graph we observe that at the beginning of the operation, the lower tank is full, and the work of the pump is to complete the distance between the surface level in the lower tank and the discharge elevation above at the upper tank. At the end of the operation, the lower tank is empty and the work of the pump is to complete the new distance between the two elevations. Consider the next graphic (Figure 8–9).

At the beginning of the operation, the work of the pump is to complete elevation Hs_1. This elevation becomes Hs_2 at the end of the operation.

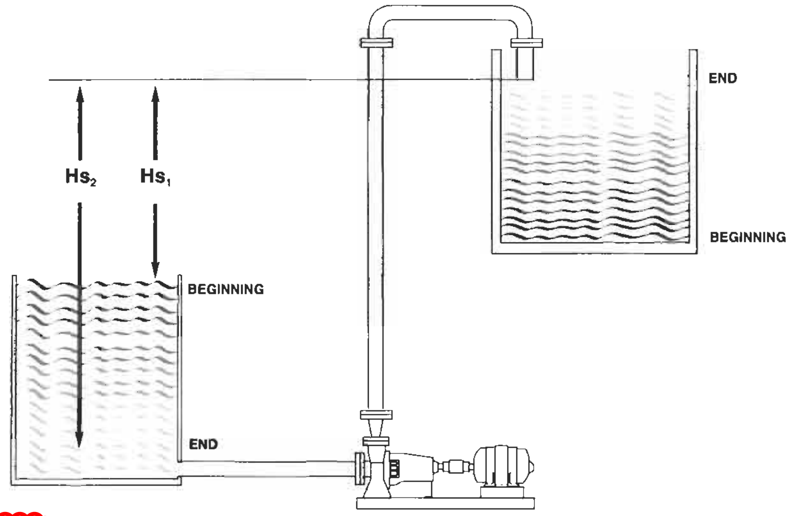

Figure 8–9

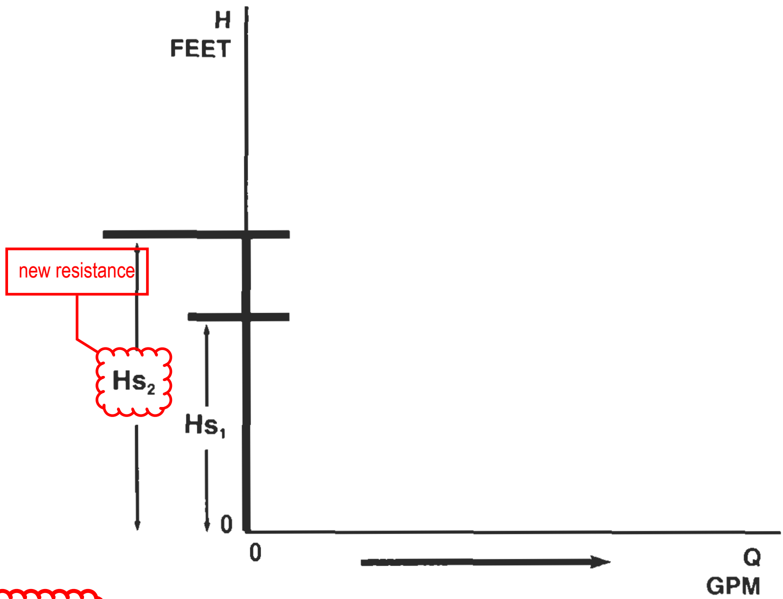

Figure 8-10

On starting the pump, and initiating flow we add the resistances in the pipes and they are shown above (Figure 8–10).

Now, due to the fact that both tanks are exposed to atmospheric pressure, there is no ΔHp to consider. Upon initiating flow, the Hf and Hv come into play and we have (Figure 8–11):

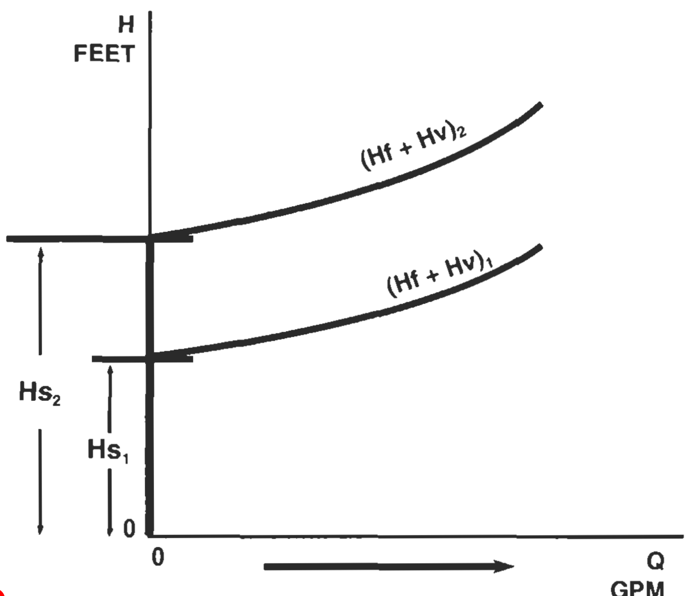

Figure 8-11

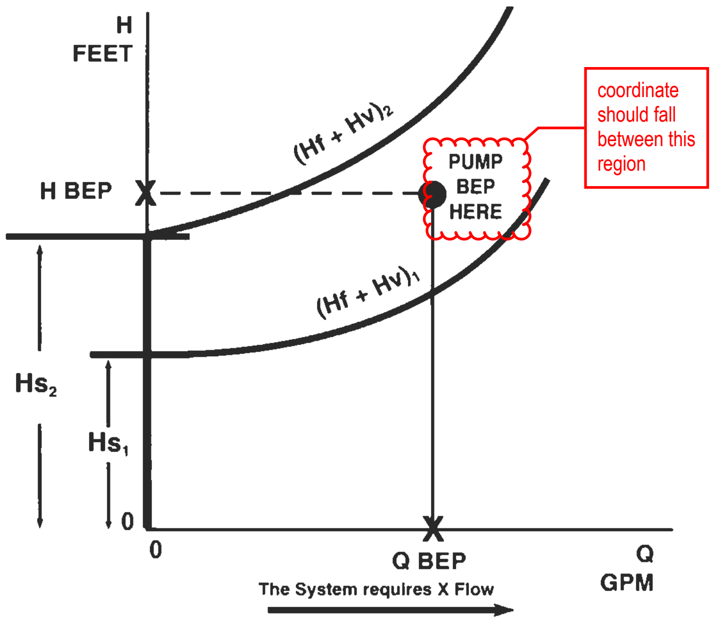

Figure 8–12

Next, we should find a pump who's BEP coordinates fall right between the Hs_1 and Hs_2 at flow X, as seen on the graph above (Figure 8–12).

With this information, the pump curve, coordinating with this system's demands according to the two tank levels, is seen this way (Figure 8–13).

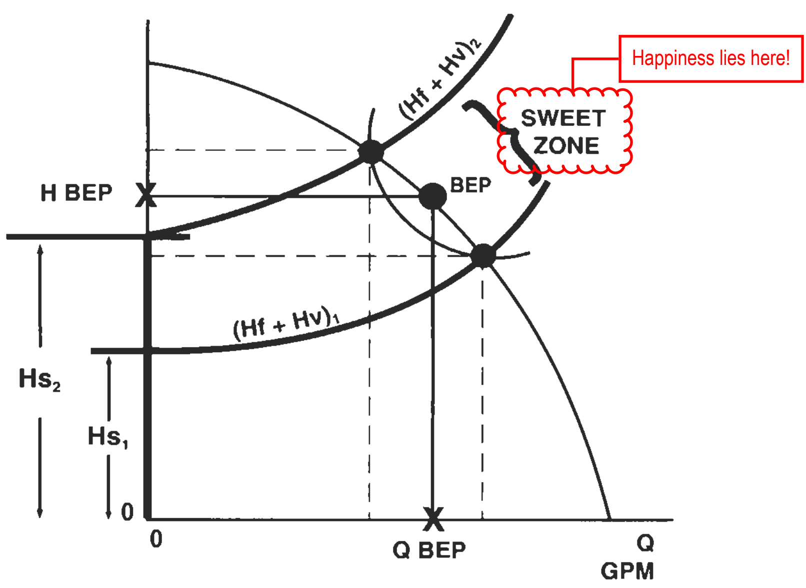

Figure 8–13

The happy zone

Now we can see the importance of the concentric ellipses of efficiency on the pump family curve. As much as possible we should find a pump whose primary efficiency arc covers the needs of the system. Certainly the needs of the system should fall within the second or third efficiency arcs around the pump's BEP. If the system's needs require the pump to consistently run too far to the left or right extremes on its curve, it may be best to consider pumps in parallel, or series, or a combination of the two, or some other arrangement, possibly a PD pump. We'll see this later.

As elevations change in the process of draining one tank and filling another, the pump moves on its curve from one elevation extreme to the other. If we've selected the right pump for the system, it will move from one extreme of its happy zone, through the BEP to the other extreme.

AUTHOR'S NOTE

This is the beginning of many problems with pumps. A pump is specified with the BEP at one set of system coordinates. Then the system (the TDH) goes dynamic, changing, and the pump moves on its curve away from its BEP out to one or the other extreme. It is necessary to determine the maximum and minimum elevations in the system and design the pump within these elevations. If the system continues to change on the pump, you'll either have to modify the system or modify or change the pump, unless you really like to change bearings and seals.

Dynamic pressures

Let's consider now a system with dynamic pressures and a constant elevation. A classic example of this would be where a pump feeds a sealed reactor vessel, or boiler. The fluid level in the reactor would be more or less static in relation to the pump. The resistances in the piping, the Hf and Hv, would be mostly static although they would go up with flow. The Hp, pressure head would change with temperature. Consider Figure 8–14.

The system curve, once again, is the visual graph of the four elements of the TDH. The Hp is stacked on top of the Hs. The Hp changes with a change in temperature in the reactor. If the reactor were cold, the Hp would be minimum or zero. We'll call this Hp_1. When the tank and fluid are heated, the Hp rises to its maximum. This is represented as Hp_2 on the graph (Figure 8–15).

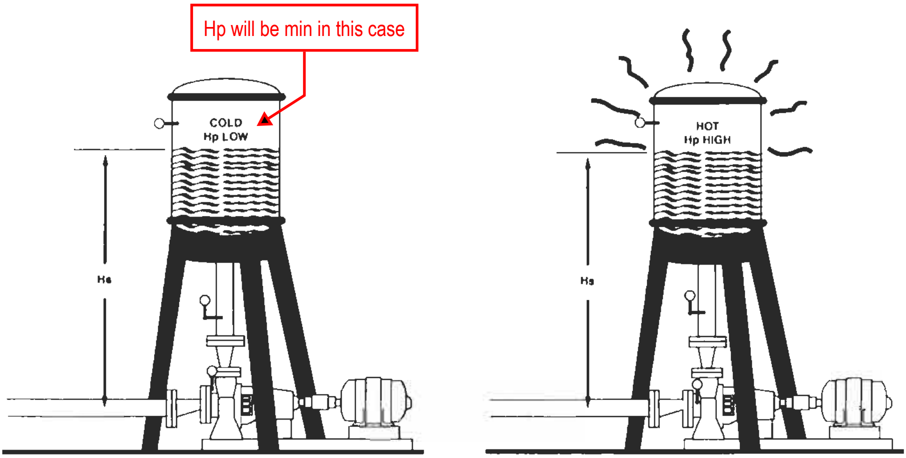

Figure 8–14

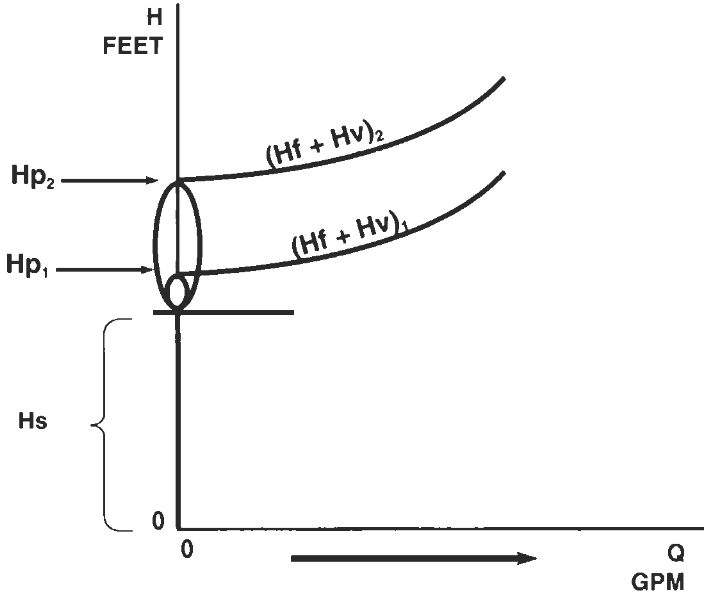

Figure 8–15

Let's say that the needs of the system require X flow. Now we search for a pump with a BEP at X gpm, at a head falling right between Hp_1 and Hp_2 on the system curve. See the next graph (Figure 8–16).

The system's ΔHp should fall within the pump's primary or secondary sweet zone. At the beginning of the operation, with the cold reactor vessel, the pump operates to the right of the BEP but within the sweet zone, and as the reactor vessel is heated, the pump migrates on its curve toward the left, crossing the BEP, to the other extreme of its sweet zone. When the reaction is completed and the tank cools, the pump

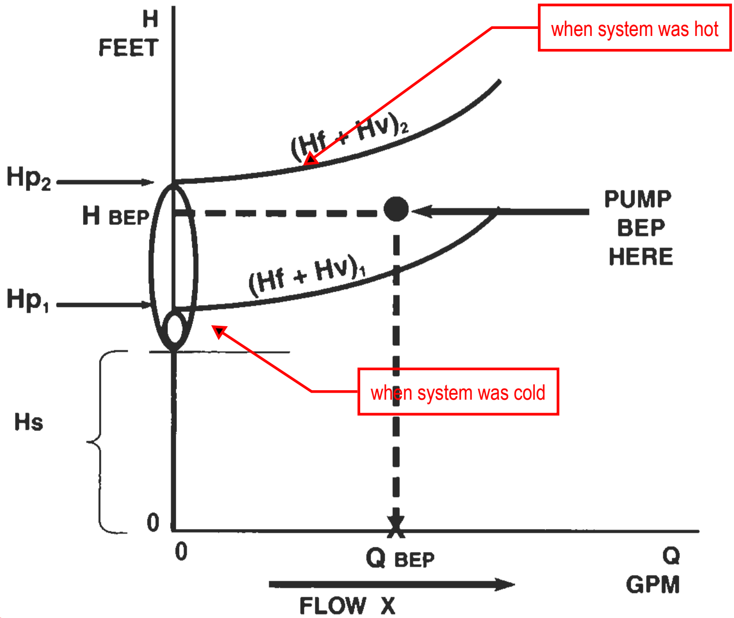

Figure 8-16

migrates again on its curve, this time toward the right, crossing the BEP and comes to rest on the right end of its sweet zone. See the next graph (Figure 8–17).

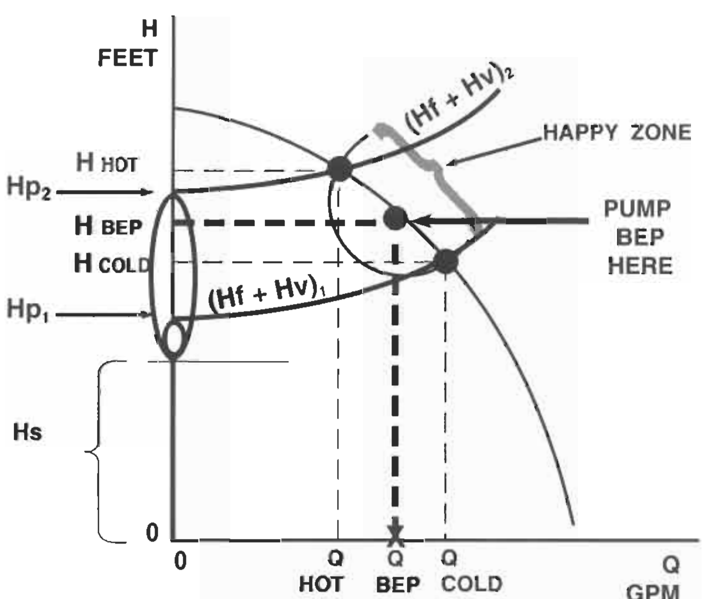

Figure 8-17

Again, we see the importance of the pump family curve, with its concentric ellipses of efficiency. It shows that in the beginning of the operation, the pump is operating to the right of its BEP. As the pressures rise in the system, the pump moves toward the left of the BEP. When the temperatures and pressures are reduced at the end of the process, the pump migrates again on its curve to the right of the BEP. During the entire operation, the pump is inside its primary or secondary sweet zone.

Variable resistances

The Hf and Hv represent the resistance losses in the system. Specifically, the Hf represents the energy consumed or lost due to friction in the system, and the Hv represents the energy consumed or lost due to the velocity of the fluid moving through the system. The Hf and the Hv are linked or connected because the Hv is used to calculate the Hf. If there is no velocity in the fluid, then the fluid is not moving through the pipes, and if the fluid is not moving through the pipes there can be no friction between the fluid and the internal pipe walls. If the resistance rises in the system, as in the case of a filter whose function is to clog over time, the flow is reduced through the filter, and the pressure or resistance rises. This means that the pump is moving toward the left on its curve. The resistances can change in the short term, or in the long term, with operations, with maintenance, or with a design change. Let's see how:

Short term resistance changes

The resistance in the pipes and system can change suddenly or in the short term due to a design change, operation, or maintenance. For example, many systems are designed incorporating variable speed motors, VFDs, to control production in a plant. The resistance is multiplied 4 times simply by doubling the velocity of the fluid in a pipe. Sometimes, in an existing system, the engineer orders to install a new control. For example, installing a flow meter into a pipe increases the resistance and the pump moves on its curve. In an effort to improve the final product, a production engineer orders a change to the screen meshes in the filters. This changes the Hf and the Hv in the system and the pump migrates on its curve.

In some plants, the operators have free reins to govern the flow in the pipes by opening and closing flow control valves. Strangling a valve reduces the flow and increases the resistance and pressure in the system in front of the valve. The pump moves away from the sweet zone of efficiency.

In a maintenance function, working against the production clock, someone changes a globe valve for a gate valve. A globe valve has between 20 and 40 times more resistance than a gate valve. Someone orders to exchange a bolted flanged long radius pipe elbow, with a welded mitered 90° elbow. This affects the resistance in the system and the pump on its curve.

AUTHOR'S NOTE

The authors recommend that all plant personnel including the engineers, operators, and mechanics receive training to recognize these rapid, unexpected, quick changes in a system. Some of these changes can be controlled within certain limits. Others must be avoided as standard plant procedure. Almost no one in maintenance or reliability relates today's failed mechanical seal with the inoffensive change in a pipe diameter six weeks ago. Engineers should train their personnel to understand the result of these inadvertent changes. These rapid changes in a system are the source of pump maintenance.

Long term resistance changes

In the long term, filters and strainers become clogged: this is their purpose. Minerals and scale start forming on the internal pipe walls and this reduces the interior diameters on the pipe. A 4 inch pipe will eventually become a 3.5 inch pipe. This moves the pump on its curve because as the pipe diameter reduces, the velocity must increase to maintain flow through a smaller orifice. The Hf and Hv increase by the square of the velocity increase.

Also in the long term, the equipment loses its efficiency, and replacement parts are substituted in a maintenance function. Also, the plant goes through production expansions and contractions: new equipment is added into the pipes. In short, the system and its elevations and pressures, its resistances and velocities, are very dynamic. The BEP of the pump is static.

What must be done is establish the maximum flow, and the minimum flow, and implement controls. Regarding filters, you've got to establish the flow and pressure (resistance) that corresponds to the new, clean filter, and determine the flow and resistance that represents the dirty filter and its moment for replacement. These points must be predetermined. The visual graph of the system curve with its dynamic resistances are seen in this example filtering and recirculating a liquid in a tank. Consider the following graphs (Figures 8–18 and 8–19).

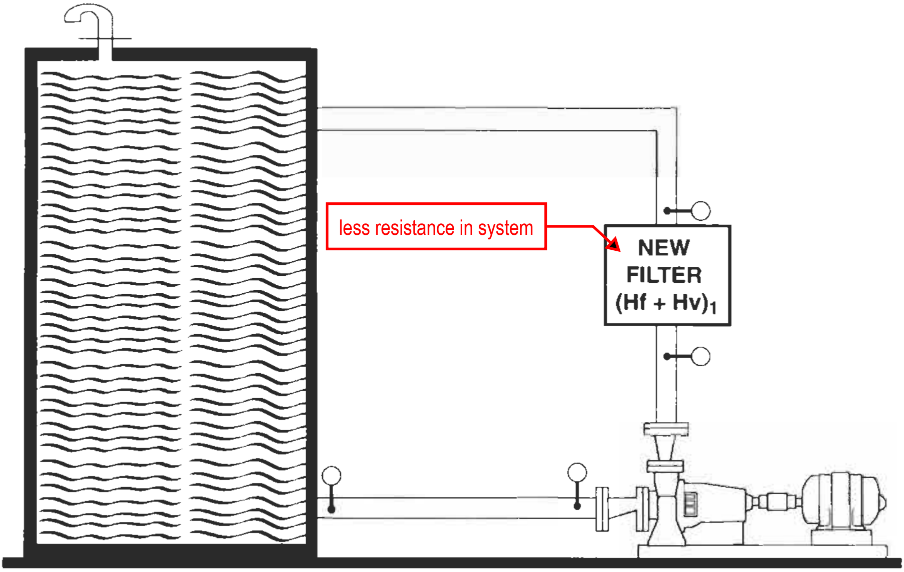

Figure 8–18

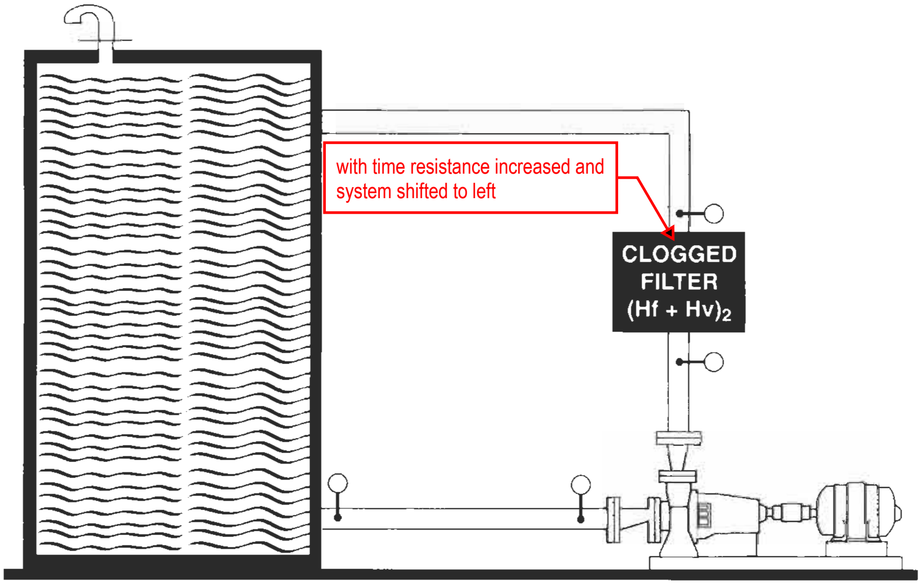

Figure 8–19

As mentioned earlier, the system curve with the clean and dirty filters should coincide within the sweet zone of the pump on its curve. (Figure 8–20 and Figure 8–21).

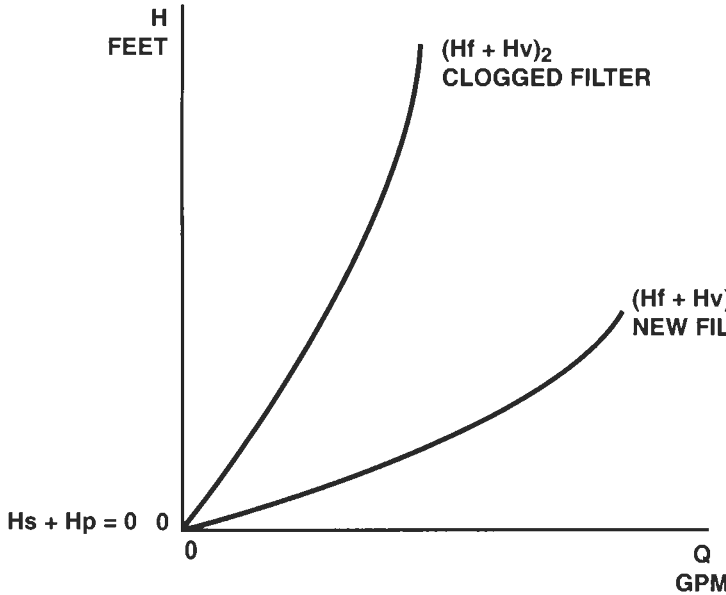

Figure 8-20

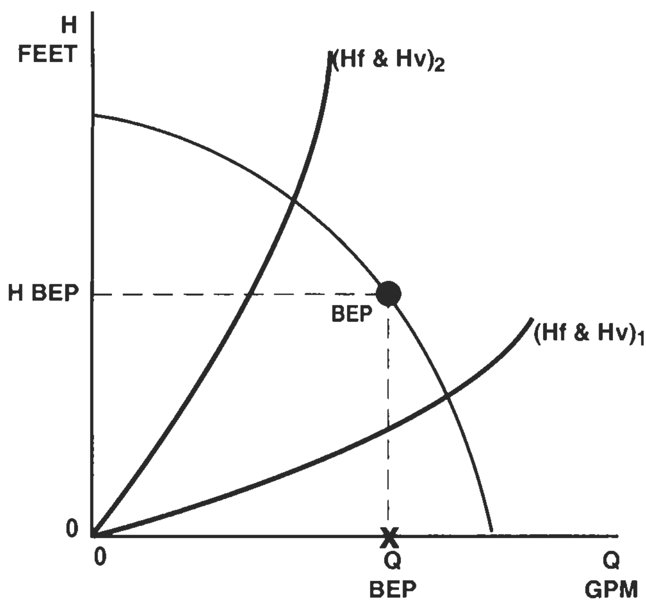

Figure 8-21

The pump will run to the right of its BEP within its sweet zone with the new filter, and slowly over time, move toward the left crossing the BEP as the filter screen clogs (Figure 8–21 and Figure 8–22).

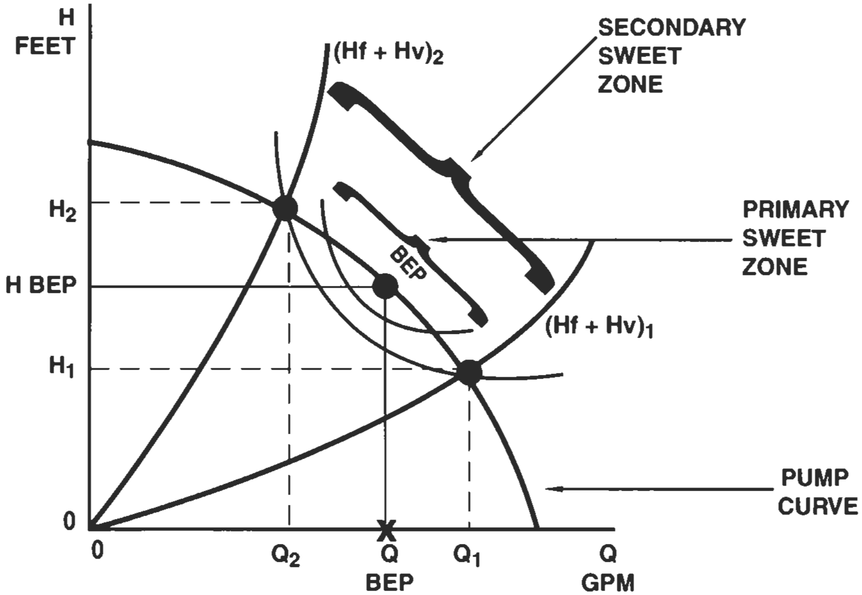

Figure 8–22

On superimposing the curve of a single pump over this system curve, we see that the system extremes are too wide for the pump to cover on its curve (Figure 8–22).

You should install pressure sensors that transmit a message to shut-off the pump, sound an alarm, or indicate to the operator that the moment to change the filter has arrived. With a new filter installed, the pump begins operating again to the right of the BEP within the sweet zone and slowly over time proceeds moving toward the other end of the sweet zone.

Pumps in parallel and pumps in series

Up to this point we've considered dynamic elements in the system with other elements static. There are times, and systems where everything is moving in concert together, with elevations rising and falling, variable pressures, clogging filters, and control valves opening and closing. When the entire system is dynamic, you've got to determine the elevation extremes, the pressure extremes, and the resistance extremes. The totally and completely dynamic system appears as Figure 8–23 and Figure 8–24.

When this happens, you need to consider an arrangement of pumps running in parallel, or in series, or in a combination of the two. Pumps in parallel are two or more pumps working side by side, taking the

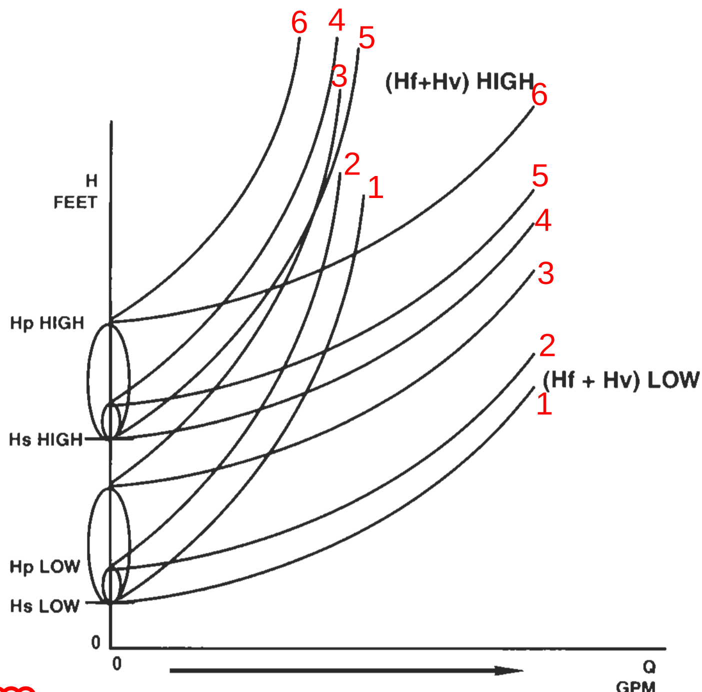

Figure 8-23

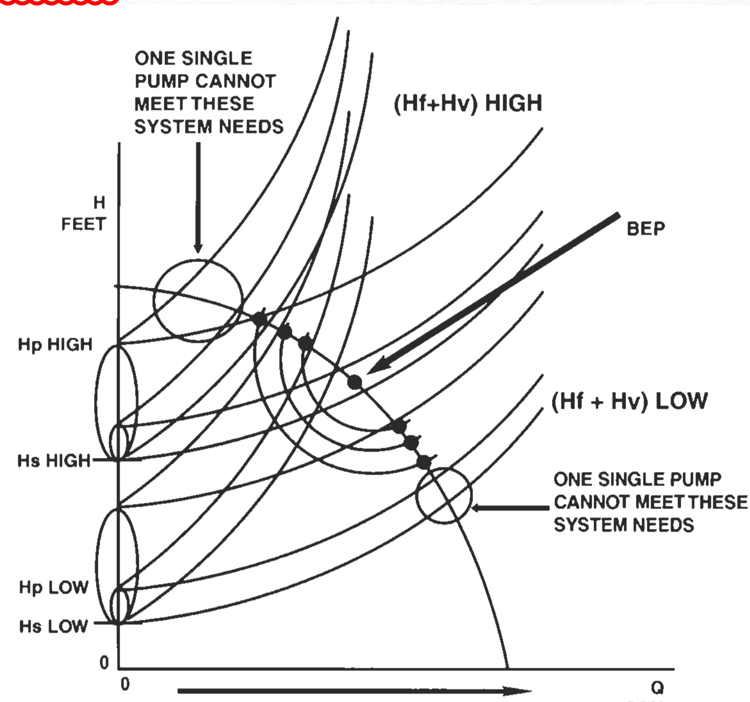

Figure 8-24

liquid from a common system, and discharging the liquid into the same common system. Two pumps running in parallel offer twice the flow at the same head. Pumps in series are two or more pumps where the discharge of one pump feeds the suction of the next pump in series. Two pumps running in series offer twice the head with the same flow. And the combination of the two arrangements offers up to multiples of both factors. First let's consider the arrangement of pumps in parallel.

Pumps in parallel

The system is designed so that two equal pumps are operating together side by side. The system can support the production of both pumps. If the needs of production are reduced, this system can operate with only one pump, simply by removing one pump from service (Figure 8–25).

The curves of pumps 'A' and 'B' individually, and 'A and B' in parallel are seen in Figure 8–26.

Because the system is designed for both pumps running together in parallel, the system curve appears as shown in Figure 8–27.

Here we see something interesting. Because the system is designed for both pumps running in parallel, when only one pump is operating, this pump will run to the right of its BEP. This situation brings it's own peculiar set of implications, not often understood in industry.

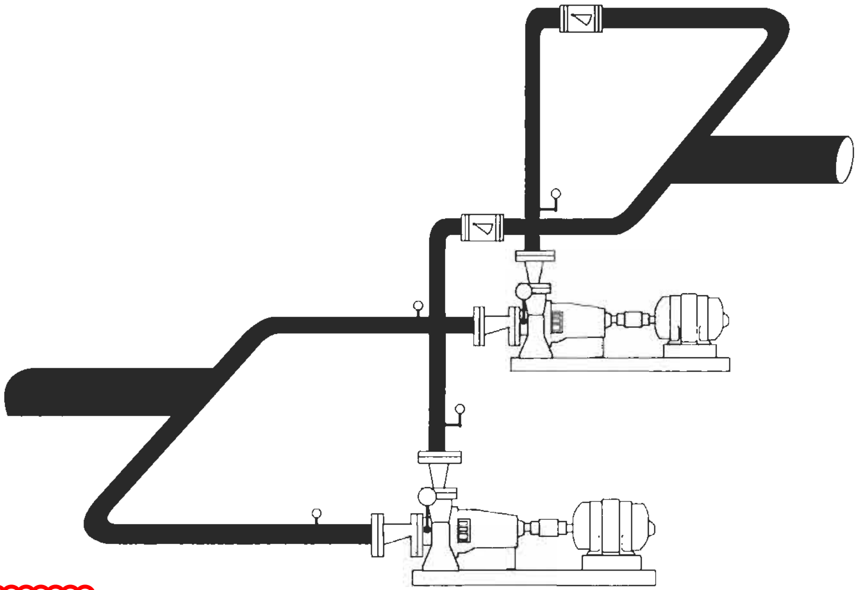

Figure 8–25

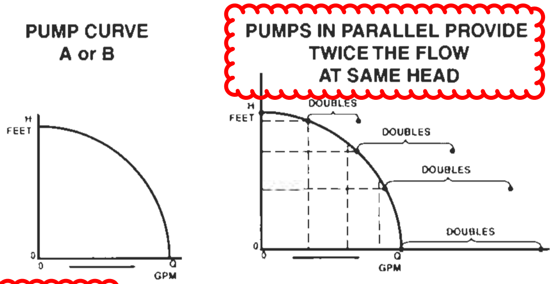

Figure 8-26

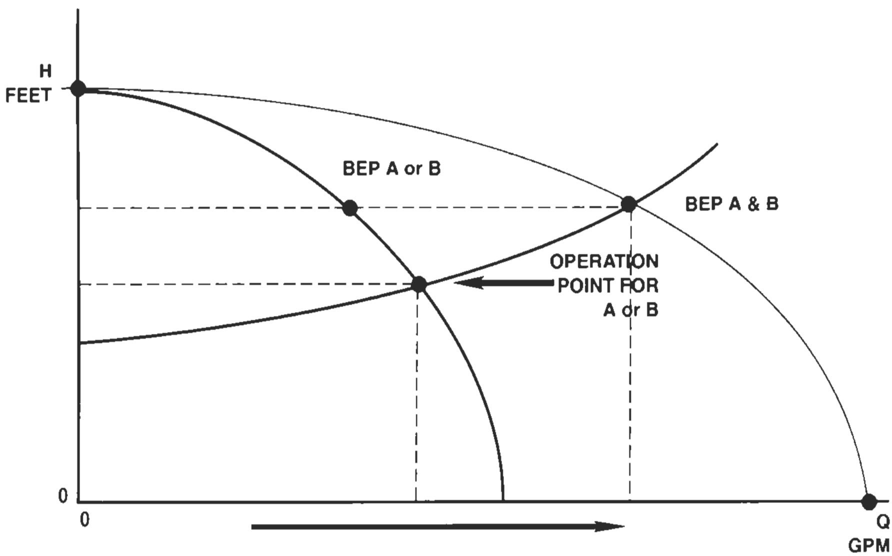

Figure 8-27

Three tips

First, one pump running in a parallel system tends to suffer from cavitation because operation to the right of the BEP indicates that the NPSHr of the pump rises drastically. To survive this condition, you should use dual mechanical seals on these pumps. Dual or double mechanical seals can withstand cavitation better than a single seal. There is a discussion on this in the mechanical seal chapter of this book. Many engineers perceive that parallel pumps are problematic because they appear to suffer a lot of premature seal failure. Parallel pumps deserve double seals even if it's only a cold water system.

The solution is: Parallel pumps should have dual or double seals installed to withstand cavitation when one pump is running solo.

Second, one single pump operating to the right of the BEP indicates that the pump will consume more energy and may require a more powerful motor. For example, if two parallel pumps running together consume 19 horsepower (BHp) of energy, it would seem natural to install a 10-Hp motor on each pump, where the individual consumption would be 9.5 horses each. But operating one pump to the right of its BEP, indicates that this pump might consume 11 or 12 horsepower. Therefore, it might require a 15 horsepower motor installed for running solo. Operating together, the two parallel pumps will only burn 9.5 horses each for a total of 19 BHp.

The solution is: Be prepared to step-up the horsepower on the motor of one solo pump in a parallel system.

Third, you would suppose that parallel pumps are identical, that they were manufactured and assembled together. But it is possible that one pump of the pair is the dominant pump and the other is the runt pump. If you start the dominant pump first in the parallel system, and then decide to add the runt pump of the pair, the weaker pump may not be able to open the check valve. The pump operator perceives that the flow meter on the second pump is stuck or broken. This is because the second pump might be 'dead heading' against a closed check valve, maintained that way by the dominant pump. If this situation exists, it may result in premature failure of bearings and seals, leading maintenance and operations personnel thinking that parallel pumps are problematic.

The solution is: Identify the dominant and weak pump should they exist. To do this, take pressure gauge readings with the pumps running at shut-off head. Verify that the impellers are the same diameter, and that the wear bands and motor speeds are equal. If you can identify one pump in the pair as dominant, always start the weak pump first and then add the dominant pump in parallel with the weaker. The dominant pump coming on stream will push open the check valve. It may be necessary to override a sequential starter.

Once these three points are understood regarding parallel pumps, these pumps give good service in systems that demand more than one single pump can deliver.

Pumps running in series

Let's begin by viewing an arrangement of pumps running in series, followed by the development of the series curve (Figure 8–28).

Series pumps theoretically offer twice the pressure at the same flow (Figure 8–29). The second pump takes the discharge head of the first

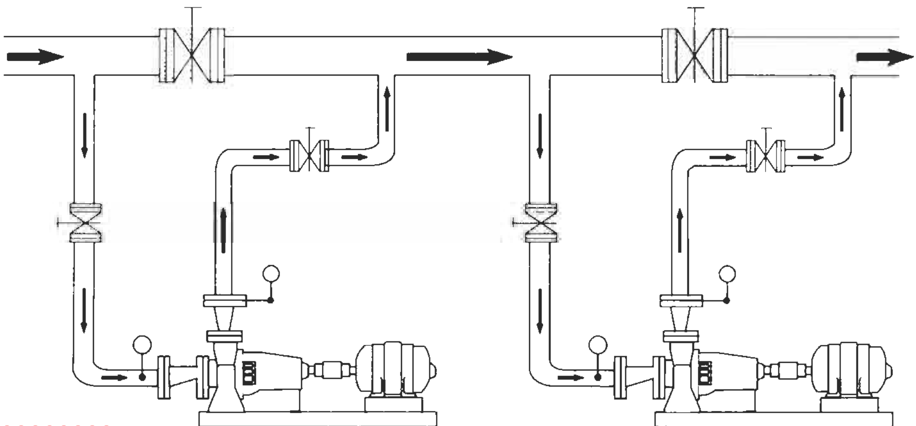

Figure 8-28

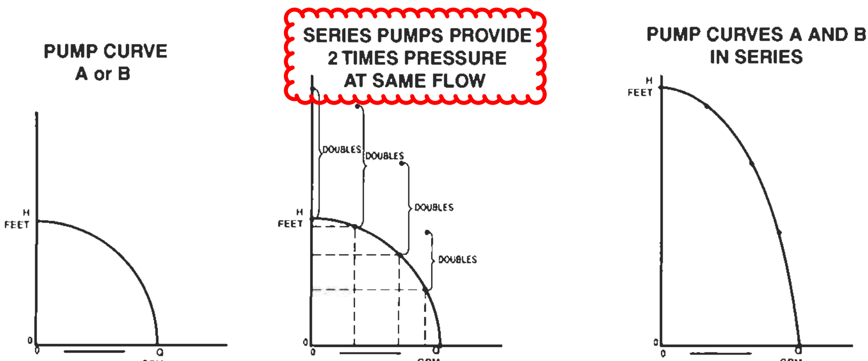

Figure 8-29

pump and jacks up the head again. However, because the system design includes 4 'T' connections, 6 valves, and at least 6 pipe elbows right at the pumps, the actual pressure is not quite doubled because the Hf is significant through the arrangement. The same tips that apply to pumps in parallel, also apply to pumps in series. Depending on the profile of the system curve, one solo pump running in a series arrangement may be running to the left of its BEP or even at shut-off head. If it is running at shut-off head, you don't really have the option of running one solo pump. Use double mechanical seals. It will be necessary to identify and trace the elements of the TDH, and match the TDH to the curve of the pumps running in series.

Combined parallel and series pump operation

Finally, we consider an arrangement of pumps running in combination

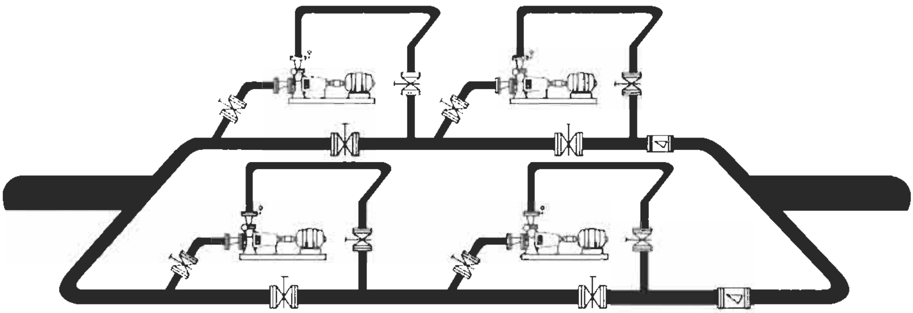

Figure 8–30

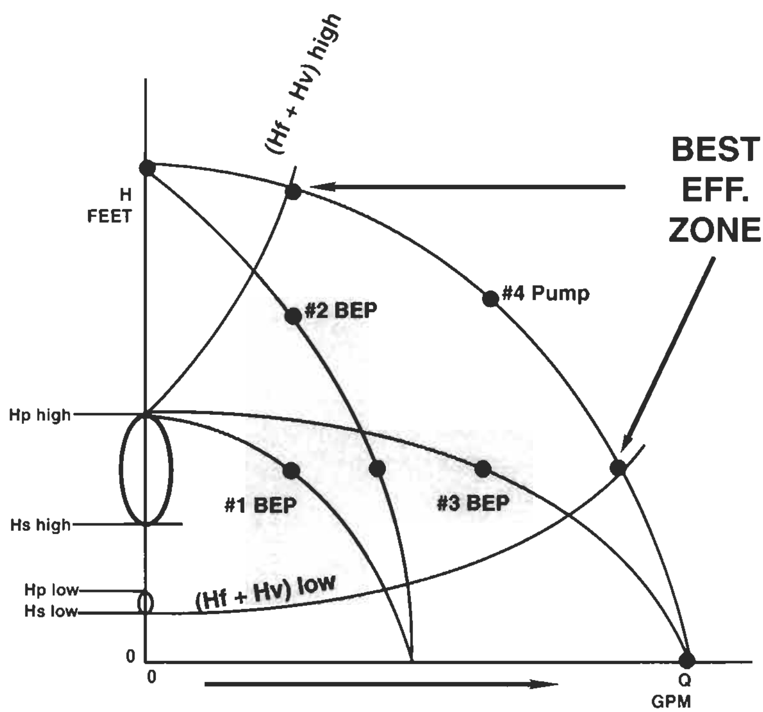

Figure 8–31

parallel and series. Notice that this system design requires 12 gate valves, 2 check valves, 10 'T' connections, and 20 elbows. Because of the high Hf in the area of the pumps, the actual head and flow characteristics may be less than the theoretical characteristics. It appears as in Figure 8–30.

The same previously mentioned critical tips apply, plus one more. Upon observing the system curve, with the pump curves, it appears that the operator can operate any one pump, or any two, or any three or four pumps. Actually there is no option to run three pumps in this arrangement. Any three pumps, by the system design, indicate that you'll be operating two pumps on one side of the system and one pump on the other side. The third pump will not be able to open the check valve with two pumps keeping it closed. So in practice, you can operate any one pump, or any two pumps (with the aforementioned hints from the parallel operation section), or four pumps, but not three pumps.

The curve, shown in Figure 8–31, is indicative of this operation.