PUMPS THAT EXPERTS SELECT.

System Analysis

The System Curve

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" or 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 portable 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 the 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 pumps. 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 level surface at the suction source or vessel, subtracted from the liquid level surface 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 a closed system 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 re-circulated closed loop. Another example would be if both the suction and the 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, and 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 $\mathbf{7 \times 2.31 \div 1.0}$ the boiler at a different pressure. This is a classic example of Hp. The formula is:

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

Where change in psi= boiler pressure- DA tank pressure

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

$$Hv = \frac{V^2}{2g}$$

Where V= velocity of the fluid moving through the pipe measured in feet per second; and g= the acceleration of gravity, $32.16\, ft/\sec^2$

Note: Hv is normally an insignificant figure, like a fraction of a foot 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 of 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 are 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 and Custodio formula later in this article). 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 article). The Hf formula for pipe is:

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

Where Kf= friction constant for every 100 feet of pipe derived from tables; and L= actual length if pipe in the system measured in feet.

The Hf formula for valves and fittings is:

Hf= K x Hv

Where K= the friction constant derived from tables; and $Hv = \frac{V^2}{2g}$

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

A) TDH= Hs + Hp + Hf + Hv

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

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 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 family curves are the most useful to the design engineer, the maintenance engineer, and the purchasing personnel.

Being that the system governs the pump, 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):

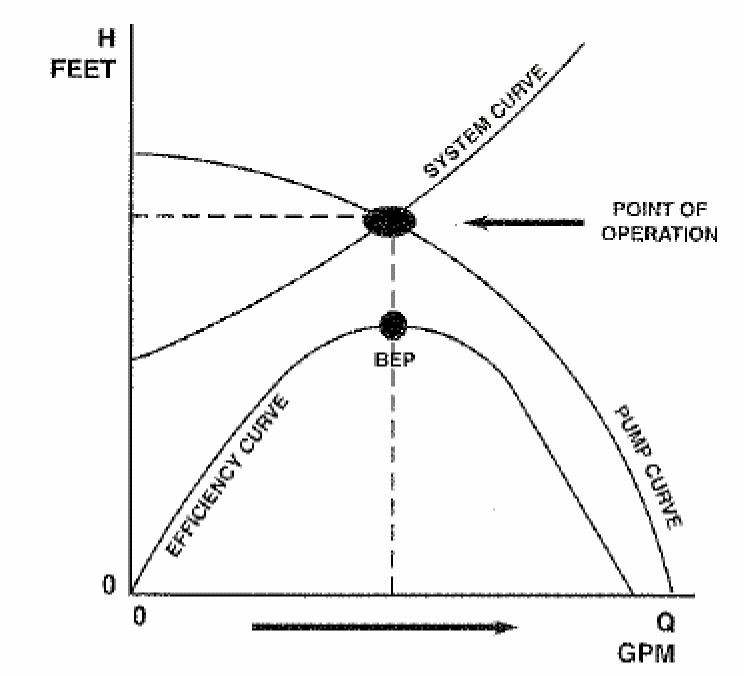


Figure 8-1

It is necessary to understand the TDH and its components in order to make correct decisions when parts of the system are changed, replaced, or modified. It's necessary to 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.

B) 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 0gpm flow. The Hs is represented as the T in Figure 8-2, wherein the pump has to elevate the liquid 50 feet (Hs= 50 ft).

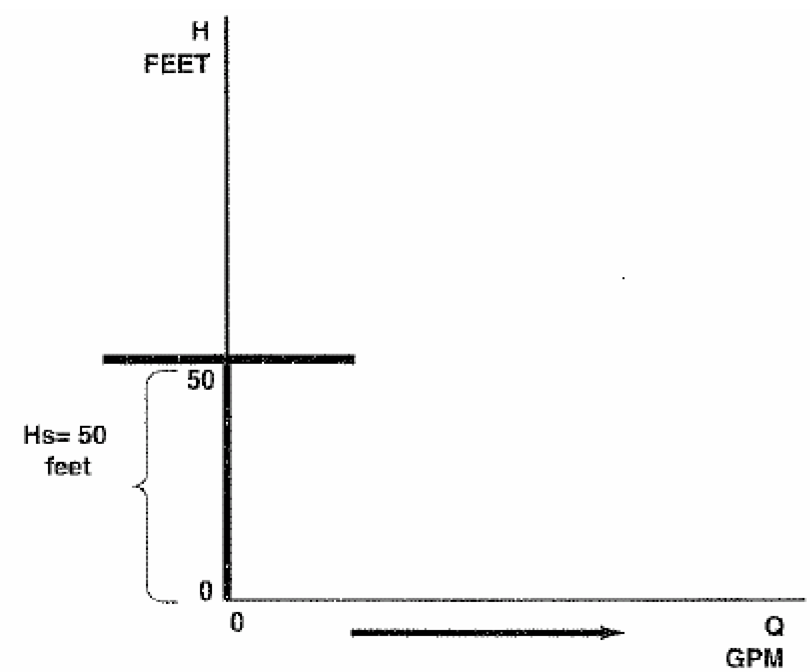


Figure 8-2

C) Determining the Hp

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

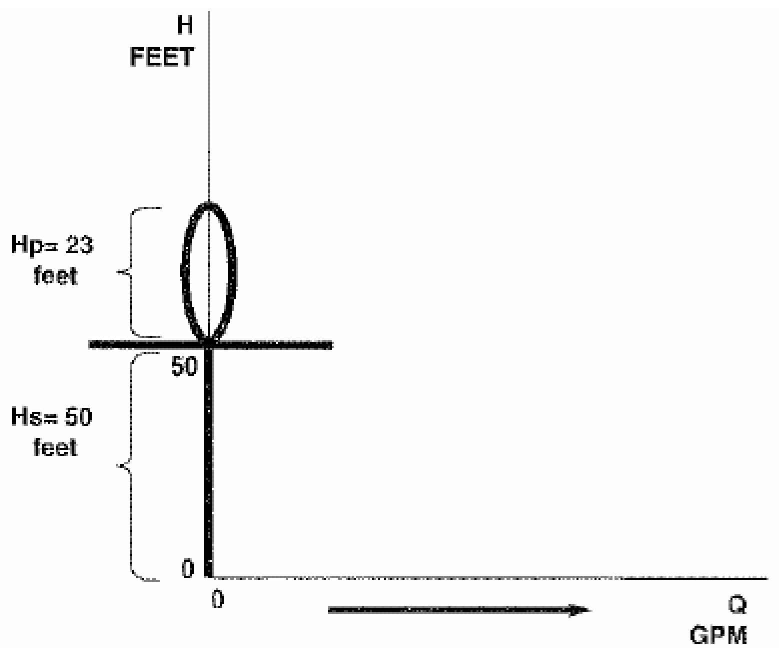


Figure 8-3

D) Calculating the Hf and Hv

Continuing with our example, before staring 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 of flow. On the graph, it is seen as in Figure 8-4.

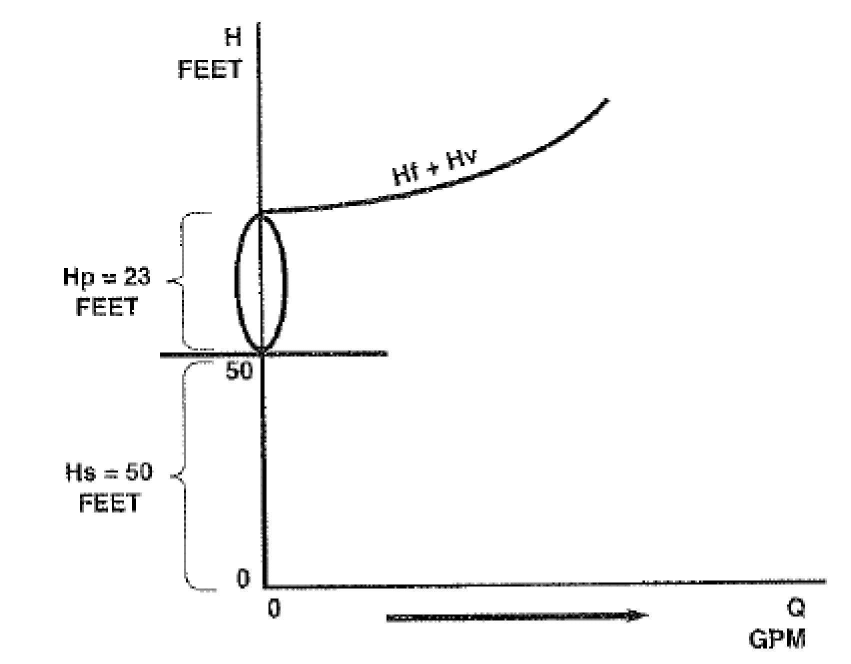


Figure 8-4

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 static. And, pipe is not uniform in its construction.

Some engineers (who normally are precise and specific) are charges with the task of approximating the friction losses (the Hf and Hv) in 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 evaluations. 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 the proposed arrangements.

E) The Hazen and Williams Formula

Mr. Hazen and Mr. Williams derived their formula, a variation of the Affinity Laws, and introduced a correction factor for friction losses of about 15%. Put simply, their formula is: $\Delta Hf \ \alpha \ \Delta Q^{1.85}$. 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 temperatures. As an approximate, it is most practical with velocities between 3 and 9 feet per second in the pipes with diameters between 8 and 60 inches.

F) The Darcy/Weisbach Formula

This formula is another variation in the Affinity Laws. Darcy and Weisbach based their formulas on friction losses of water moving in open canals. They applied other friction coefficients from some private experiments, 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, temperatures, 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 Hazen and Williams.

Note: The two formulas are variations on the Affinity Laws, which are probably equally adequate to "guestimate" the friction losses in non-uniform piping. Both 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. It is necessary to begin specifying the pumps during this stage, even though they are based on incomplete information.

It really doesn't matter which formula you prefer 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 are followed to the letter (and this happens practically never). Both formulas assume that there are no substitutions due to back orders or delivery shortages during construction and assembly. 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, chemical processes, piping material, valve designs, and new technologies have been invented that were not considered when these formulas were developed with cold water in the 19^{th} century. There is a need to measure the actual losses once operation begins.

G) <u>The Bachus and Custodio Formula</u>

You will need to gather information from the 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 and Custodio method is the following:

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

Where P = pressure differential from an upstream to a downstream gauge in a section of the 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, suction piping with the pump while running; So= suction off, 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.

The system, although simple, with only one pump, is more or less representative of all systems. The 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 has a K value of 0.6.4. 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 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 the different organizations.

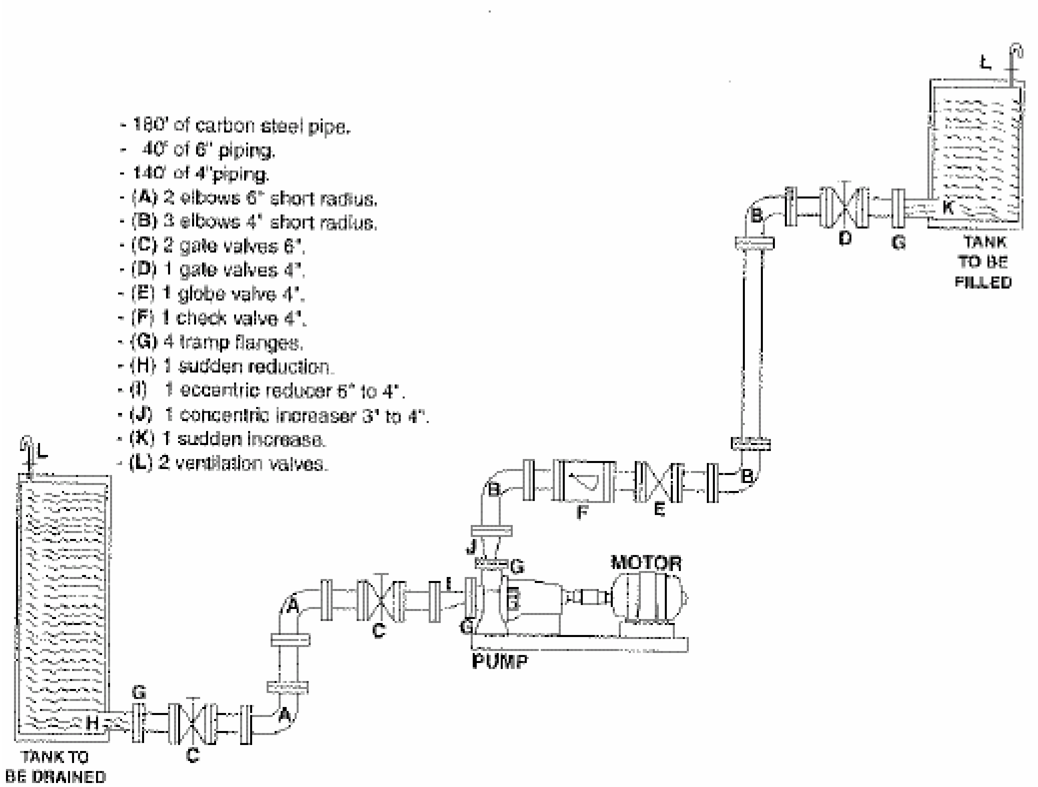


Figure 8-5

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.

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 change is H, by observation is 80 ft. See Figure 8-6.

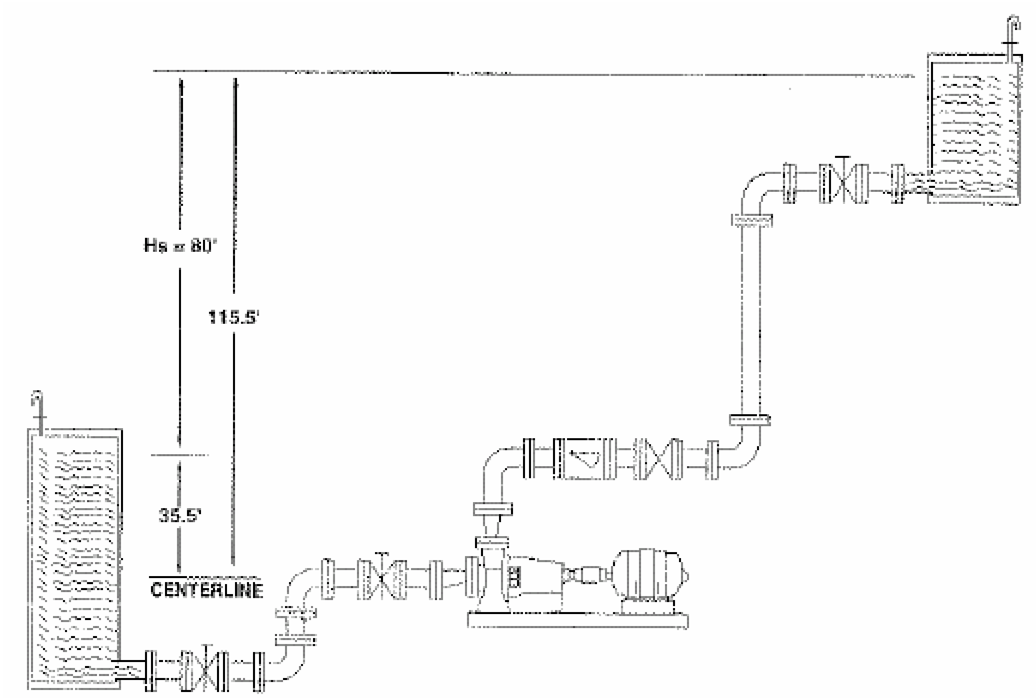
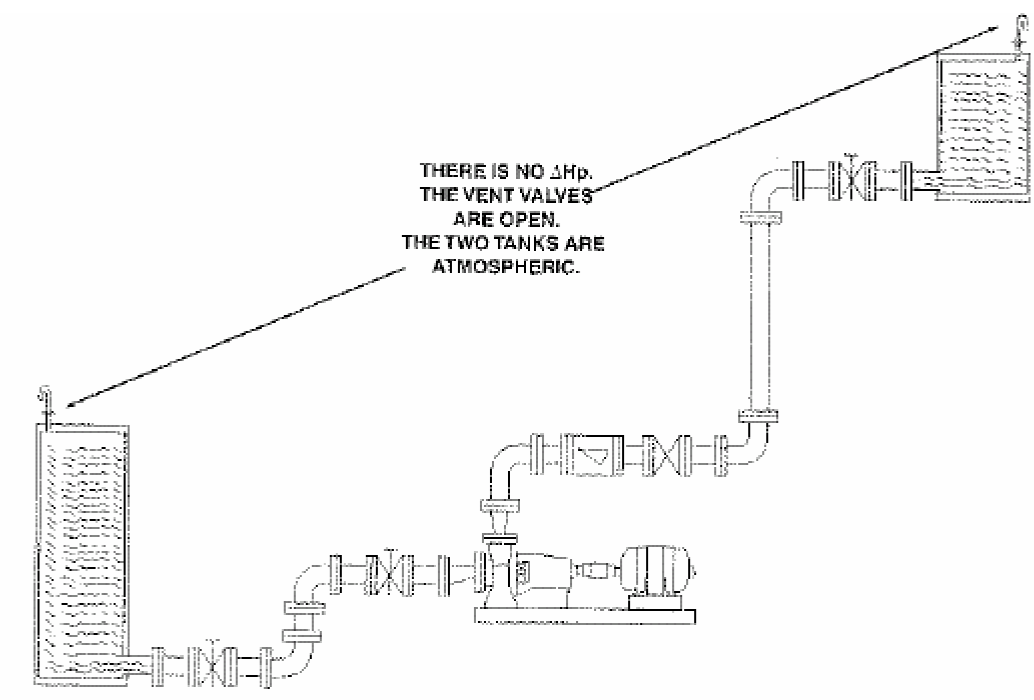


Figure 8-6

Another component of the TDH is the Hp, the 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 does not exist; the change in Hp is 0.



Note: There is a lot of detailed work that goes into calculating the frictions and velocities in a piping system. Admittedly, there are computer programs today that will perform these calculations in a flash. The remaining two elements needed to calculate the TDH, the Hf and the Hp, are the most illusive and difficult to calculate. Yet, they determine how and where the pump will operate on its curve.

Using the formulas, the K values, and the pipe schedule tables found in the Hydraulics 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 needed to calculate the Hf, we will begin with the Hv. The formula is:

System Hv =Hv suction + Hv discharge

$$= V^2 \div 2g \text{ suction} + V^2 \div 2g \text{ discharge}$$

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

=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

$$= (Ksuction \times L) \div 100 + (Kdischarge \times L) \div 100$$

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

= **1.956 + 0.891**

Hf system piping = 2.848 feet

Now we can calculate the Hf in the elbows. The formula is:

Hf elbows = Hf suction elbows + Hf discharge elbows

= 2 x 0.280 x 0.172 + 3 x .0310 x 0.888

= 0.096 + 0.82

Hf Elbows = 0.916 feet

Now we can calculate the Hf value for the valves.

There are 5 valves in all. There are two six inch gate valves at 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

$$= K_{6'' gate} \times Hv_{suction} + K_{4'' gate} \times Hv_{discharge} + K_{4'' check} \times Hv_{discharge} + K_{4'' globe} \times Hv_{discharge}$$

$$= (2 \times 0.09 \times 0.172) + (1 \times 0.16 \times 0.888) + (1 \times 2 \times 0.888) + (1 \times 6.4 \times 0.888)$$

= 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 are 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 take into account the losses at the entrance and exit port fittings. Then there are unassociated “tramp” flanges and unions. Examples would be unions between a pipe and a tank, or between a pipe and a pump. They must be calculated because there is friction (and energy loss) 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

$$= K_{6''} \times Hv_{suction} + K_{4''} \times Hv_{suction} + K_{4''} \times Hv_{discharge} + K_{3''} \times Hv_{discharge}$$

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

= 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 flanges 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:

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

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

= 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:

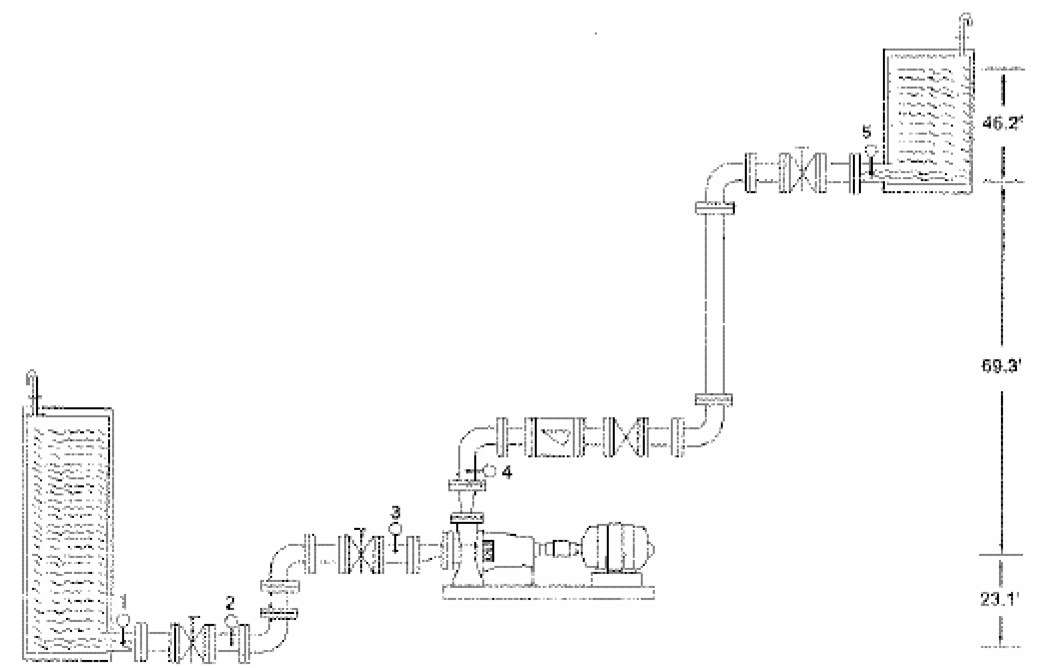
System Hf = Hf pipe + Hf elbows + Hf valves + Hf tramp flanges + Hf other
= 20848 ft + 0.916 ft + 7.632 ft + 0.007 ft + 1.192 ft
= 12.595 ft

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 companies contract this work out to consulting engineering companies. Finally, we can calculate the TDH of the system:

TDH = Hs + Hp + Hf + Hv
= 80 ft + 0 ft + 12.595 ft + 1.06 ft
= 93.655 ft

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 a local distributor stock, it would probably be a 3 x 4 x 12 model end-suction centrifugal back pullout pump with the impeller machined out ot 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 and Custudio Method, and forget about all those calculations. See Figure 8-8 with the corresponding elevations and placement of pressure gauges installed into the piping numbered 1 through 5.



In the 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, as this assures that both sides of the system are atmospheric and thus cancels out 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 for expelling any trapped air in the volute. On the pump, conventional stuffing boxes can also trap air. This must be expelled too. Vertical valve systems 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 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.

Here's a quick review of the Bachus and Custudio Formula:

System Hf and Hv = $\left[\left(\Delta PDr-\Delta PDo\right)+\left(\Delta PSr-\Delta PSo\right)\times 2.31\right]\div sp.gr.$

The Bachus and Custudio Formula does not make mistakes. It is not based on models or experiments developed 150 years ago.

Let's take our readings with the water as the test liquid just to keep the conversation simple. With the system and pump off, not that the 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 Δ PDo= 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 Δ Pso= 10 psi.

Gauge 1, on the other side of the closes 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 the 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 Δ PSr = 12 psi.

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 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 is not open. Maybe the gauges need calibration; they should be sent to a calibration shop a couple times a year. It is interesting to see how much more you know about your system after learning to interpret the pressure gauges.

Now we consider the pump. The pump takes the energy that the suction gives it, and 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 is also 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 discharge pressure gauge (4 gauge) should be reading 53 psi (40 + 13).

Note: The pump's discharge pressure is a function of the suction pressure. Regrettably, most of the pumps in the world don't have a gauge that reads suction pressure. In the example here, if the 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 the 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 50 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, Δ PDr= 35 psi (53-18). We now have all the information we need to insert into the Bachus and Custodio Formula:

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

The Bachus and Custodio Formula does not make mistakes. It is not based on models or experiments that 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 on the next shift, and tomorrow, and next month. If a resistance load changes, it will be registered in the new 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. Regarding pump failure due to problematic seals and bearings that need emergency maintenance, in about 80% of all cases, the pump is telling its 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 Hp are determined by simple observation? The detailed discussion as to the Hs and Hp here probably has the reader ready to throw this article into the garbage. With the Bachus and Custosio Formula, the differential pressure gauge readings on the system with the pump system 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 and 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.

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