CENTRIFUGAL PUMP SELECTION, SIZING, AND INTERPRETATION OF PERFORMANCE CURVES 4

4.0 PUMP CLASSES

Pumps may be classified in two general types, dynamic and positive displacement. Positive displacement pumps are those in which energy is imparted to the liquid in a fixed displacement volume, such as a casing or a cylinder, by the rotary motion of gears, screws, or vanes, or by reciprocating pistons or plungers. Centrifugal pumps are dynamic pumps. Energy is imparted to the liquid by means of a disk with curved vanes rotating on a shaft called the impeller. The impeller imparts kinetic energy to the fluid by means of its shape and high rotational velocity. This energy is transformed to pressure energy when the fluid reaches the pump casing (see Figure 1-12). The pressure head difference between the inlet and the outlet, or Total Head produced by the pump, is proportional to the impeller speed and diameter. Therefore, to obtain a higher head, the rotational speed or the impeller diameter can be increased. To learn more about how a centrifugal pump increases a fluid's pressure, see reference 15.

How a pump produces pressure is beyond the scope of this book, but an interesting experiment you can try at home will illustrate a similar process. A small plastic bottle is required to which a string is attached. Twist a rubber band around the bottle's neck a few times and attach two 3-foot long strings, one on each side of the glass. Tie the other ends of the string together, fill the glass half full with water and hold it suspended from the strings. Start spinning. As you may have guessed, the fluid inside the glass will become pressurized. How do you know that the fluid is pressurized? To prove it to yourself, make a very small hole in the glass bottom. Make the hole just large enough for water to dribble through. Now spin the glass again. The water will spray out of the glass bottom no matter what its position, up or down.

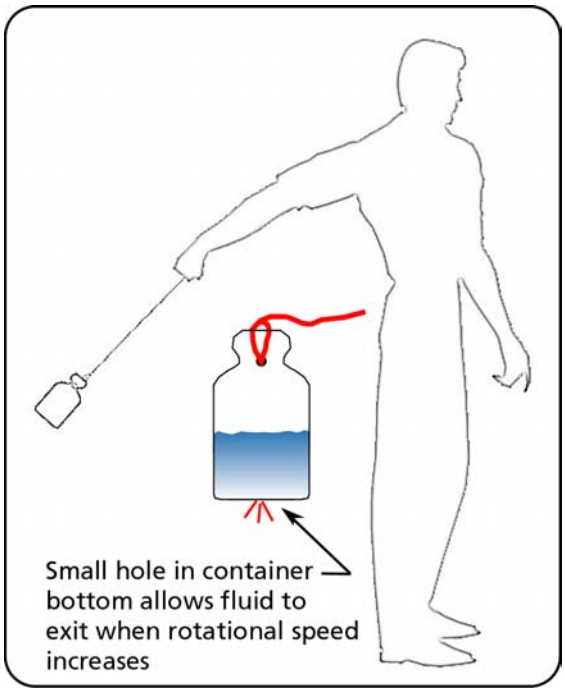

Figure 4-1 Using a spinning bottle to demonstrate centrifugal force.

4.1 COVERAGE CHART FOR CENTRIFUGAL PUMPS

A coverage chart (see Figure 4-2) makes it possible to do a preliminary pump selection by looking at a wide range of pump casing sizes for a specific impeller speed. This chart helps narrow down the choice of pumps that will satisfy the system requirements.

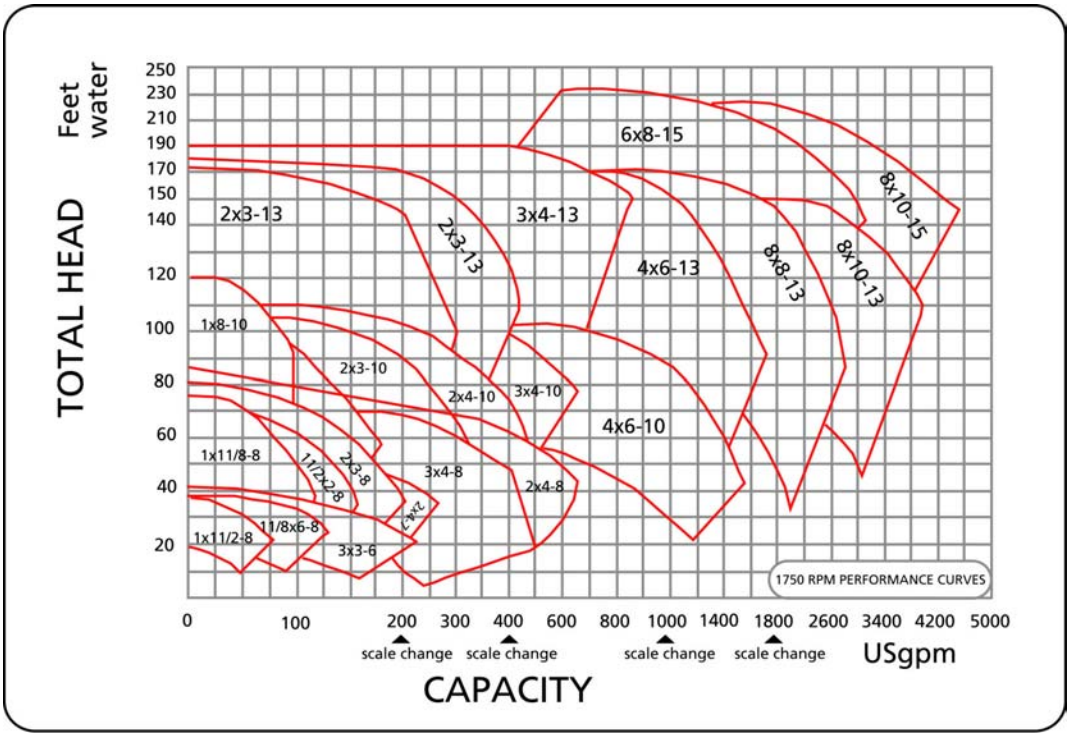

Figure 4-2 Typical pump capacity coverage chart.

4.2 PERFORMANCE CURVE CHART

The following figure shows a typical pump performance chart for a given model, casing size, and impeller rotational speed. A great deal of information is crammed into one chart and this can be confusing at first. The performance chart covers a range of impeller sizes, which are shown in even increments of 1/2" from 7 1/2" to 9 1/2". Impellers are manufactured to the largest size for a given pump casing and machined or "trimmed" to the required diameter when the pump is sold.

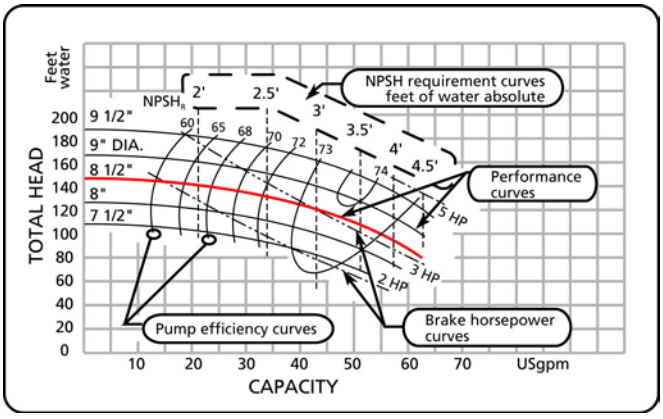

Figure 4-3 Typical pump performance curve.

Performance Curve

At some point in the pump selection process, the impeller diameter is selected. For an existing pump, the diameter of the impeller is known. For a new pump, our calculations of Total Head for a given flow rate will have determined the impeller diameter to select according to the performance curve. Figure 4-4 shows only the information relevant to the 8 1/2" impeller performance curve.

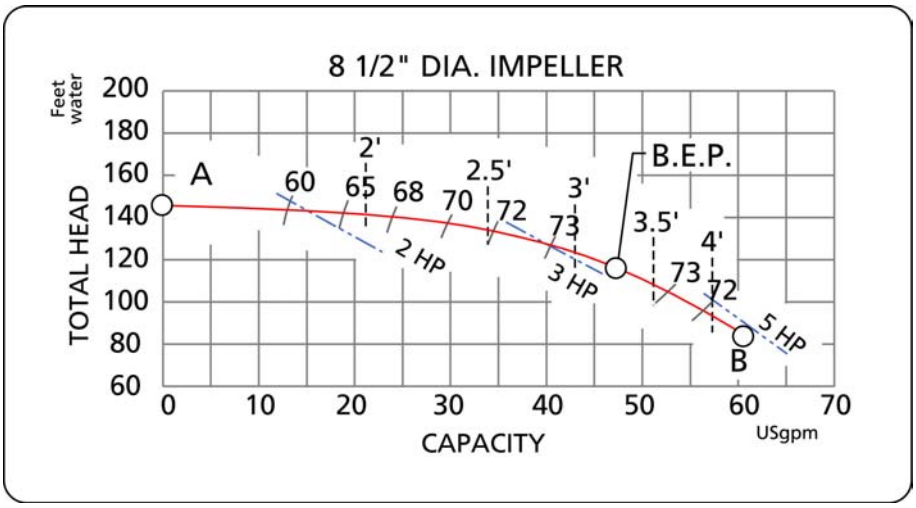

Figure 4-4 Typical performance curve for a specific impeller diameter.

A performance curve is a plot of Total Head vs. flow rate for a specific impeller diameter and speed. The plot starts at zero flow. The head at this point corresponds to the shut-off head of the pump, point A in Figure 4-4 (more about this later). Starting at this point, the head decreases until it reaches its minimum at point B. This point is sometimes called the run-out point and represents the maximum flow of the pump. Beyond this, the pump cannot operate. The pump's range of operation is from point A to B.

Efficiency Curves

The pump's efficiency varies throughout its operating range. This information is essential for calculating the motor power (see section 4.9).

The B.E.P. (best efficiency point) is the point of highest efficiency of the pump. All points to the right or left of the B.E.P have a lower efficiency (see Figure 4-4). The impeller is subject to axial and radial forces, which get greater the further away the operating point is from the B.E.P. These forces manifest themselves as vibration depending on the speed and construction of the pump. The point where the forces and vibration levels are minimal is at the B.E.P.

In selecting a pump, one of the concerns is to optimize pumping efficiency. It is good practice to examine several performance charts at different speeds to see if one model satisfies the requirements more efficiently than another. Whenever possible the lowest pump speed should be selected, as this will save wear and tear on the rotating parts.

Note: The pump performance curves are based on data generated in a test rig using water as the fluid. These curves are sometimes referred to as water performance curves. The use of these curves for fluids with a different viscosity than water can lead to error if the proper correction factors are not applied. These correction factors are applied to the Total Head, the flow and the efficiency of the pump, and are published in form of curves by the Hydraulic Institute (see Standards book published by the Hydraulic Institute, web site: http://www.pumps.org and for the correction factor charts see the web site www.fluidedesign.com).

Horsepower Curves

The horsepower curves are shown on the chart and give the power required to operate the pump within a certain range. For example (see Figure 4-5), all points on the performance curve to the left of the 2 hp curve will be attainable with a 2 hp motor. All points to the left of 3 hp curve and to the right of the 2 hp curve will be attainable with a 3 hp motor. The horsepower can be calculated with the Total Head, flow and efficiency at the operating point. (More on horsepower calculations and operating point later). *Note that the horsepower curves shown on the performance curves are valid for water only.*

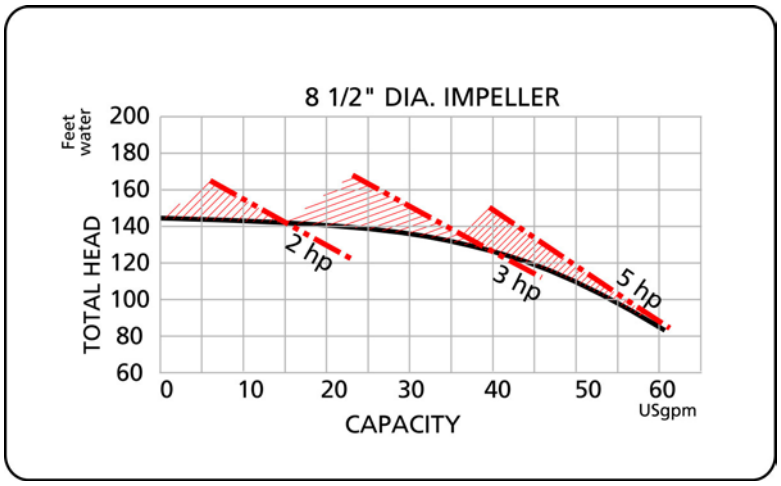

Figure 4-5 Coverage area of horsepower curves.

N.P.S.H. Requirement Curves

The pump manufacturer specifies a minimum requirement on the N.P.S.H. in order for the pump to operate at its design capacity. These are the vertical dashed lines in Figure 4-6. The N.P.S.H. required becomes higher as flow increases, and lower as flow decreases. This essentially means that more pressure head is required at the pump suction for high flows than low flows. Keep in mind that N.P.S.H. is a head term and therefore independent of the fluid density and is in absolute fluid column height. The N.P.S.H. required for the maximum flow in Figure 4-4 is approximately 4 feet absolute. This is not very restrictive; most industrial pumping systems will have much more N.P.S.H. available.

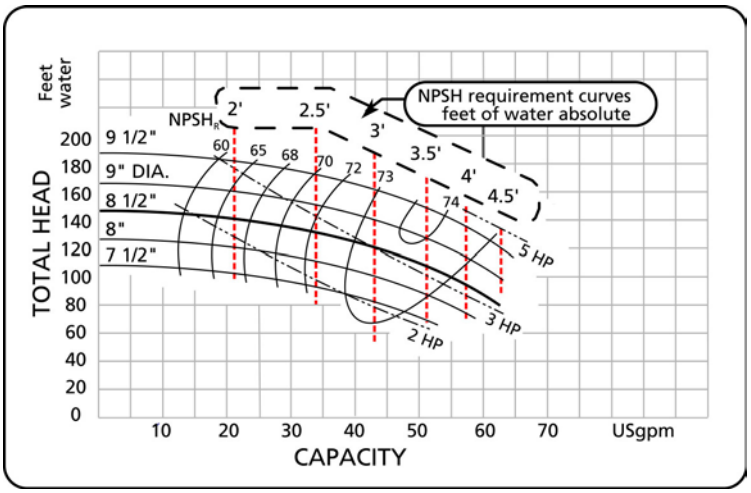

Figure 4-6 N.P.S.H required curves.

4.3 IMPELLER DIAMETER SELECTION

Quite often, the operating point is located between two curves on the performance chart. We can calculate the impeller size required by linear interpolation. For example, if the operating point falls between the 9" and 9 1/2" impeller curve (see Figure 4-7), the following equation will give the correct size:

$$D_{OP} = 9 + \left(\frac{9.5 - 9}{\Delta H_{9\,1/2} - \Delta H_9}\right)(\Delta H_{OP} - \Delta H_9)$$

where D_{OP} : impeller diameter required;

ΔH_{OP} : pump total head at the operating point;

ΔH_9 : pump total head at the intersection of the 9" impeller curve and flow rate;

$\Delta H_{9\,1/2}$: pump total head at the intersection of the 9 1/2" impeller curve and the flow rate.

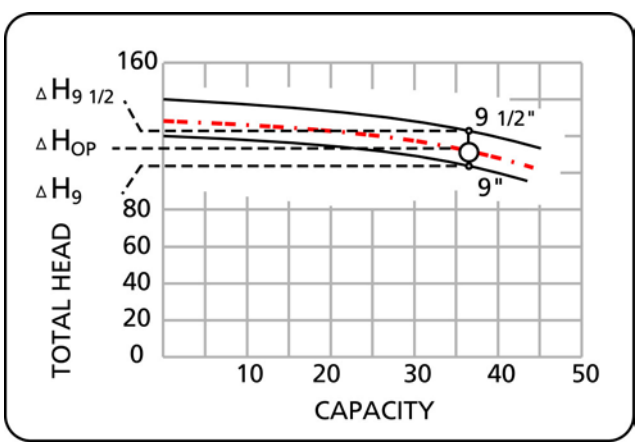

Figure 4-7 Interpolating impeller diameter.

It is good practice to select (when possible) a pump with an impeller that can be increased in size, permitting a future increase in head and capacity. Or alternatively, an impeller which can be reduced in size. As a guide, select a pump with an impeller size no greater than between *1/3* and *2/3* of the impeller range for that casing with an operating point in the high efficiency area (see Figure 4-8). It is also important not to go too far right or left from the B.E.P.. A guideline is to locate the operating point between 110% and 80% (see reference 16) of the B.E.P. flow rate with an operating point in the desirable impeller selection area (see Figure 4-8).

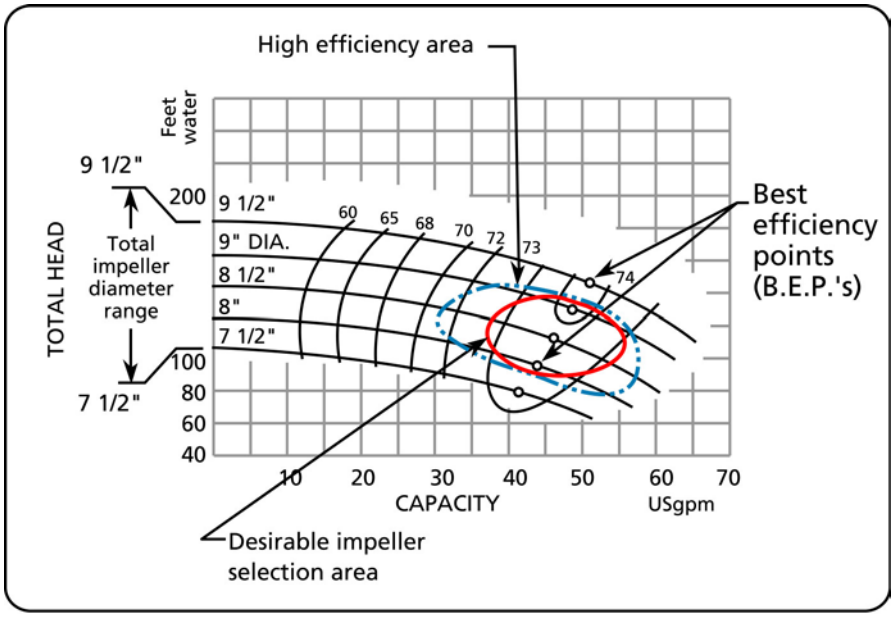

Figure 4-8 Desirable pump selection area.

4.4 SYSTEM CURVE

The system curve is a plot of the Total Head vs. the flow for a given system. The higher the flow, the greater the head required (see Figure 4-9). The shape of the system curve depends on the type of system being considered. The system curve equation for a typical single outlet system such as in Figure 2-10 is:

$$\Delta H_P(q) = \Delta H_F(q) + \Delta H_{EQ}(q) + \Delta H_v(q) + \Delta H_{TS}$$

The system curve is superimposed on the pump performance chart. The Total Static head is constant and the friction head, equipment head and velocity head are flow dependent. The calculation of Total Head at different flow rates produces a plot of Total Head vs. flow that is called the **system curve**.

The operating point is the point on the system curve corresponding to the flow and head required. It is also the point where the system curve intersects the performance curve. The design system curve is usually calculated with extra flow capacity in mind. It is good

practice to plot the system curve for higher flow rates than the design flow rate, since flow demand may change and extra capacity may be required.

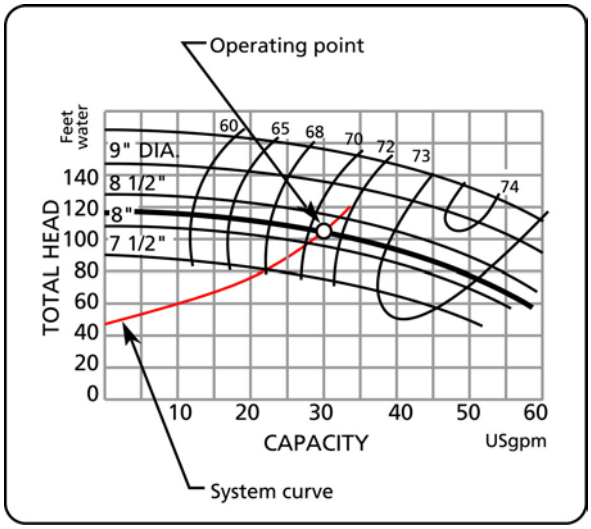

Figure 4-9 Superposition of system curve and pump performance curve.

4.5 OPERATING POINT

The question in example 2.1 has remained unanswered until now. That is: How can we guarantee that the pump will deliver 500 USGPM at 70 feet of head?

First, determine the right pump size by using the coverage chart (see Figure 4-2). Locate the head and flow coordinate (operating point) on the chart. This will identify the casing size and pump speed. Locate the correct performance curve chart for this pump casing size and again identify the operating point.

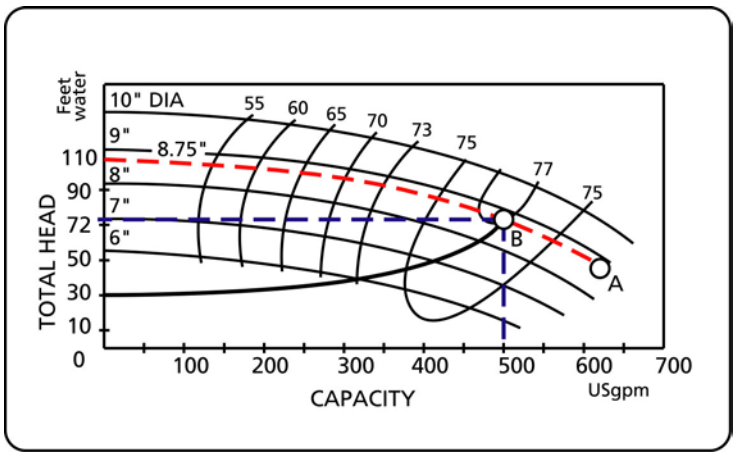

Figure 4-10 Location of operating point in example 2.1.

Determine the impeller size. The impeller size will be approximately 8.75". The dashed line represents the performance curve of the pump. The pump can only operate on its performance curve. The system can only be run somewhere on the system curve. The intersection of these two defines the operating point which is the only point that the pump and system can operate.

As is often the case, one question leads to another**:** How does the pump get to the operating point from the moment it is switched on?

There are two methods which can be used to start a pump. With method 1 the discharge valve is closed, whereas with method 2 the discharge valve is open. The term discharge valve refers to the manual valve located close to the pump outlet flange.

Method 1 (discharge valve closed)

Pumps are often started with the discharge valve closed. Immediately after the pump is started, the head rises to point D (see Figure 4-11). The system curve at this moment is vertical. By gradually opening the discharge valve to its full open position, point D will move down the performance curve to point C, where the discharge valve should be fully open. The system curve's shape is progressively modified as the operating point is moved towards point C. This is a typical way of starting large pumps and should be the preferred method for any pump operating at more than 500 USGPM.

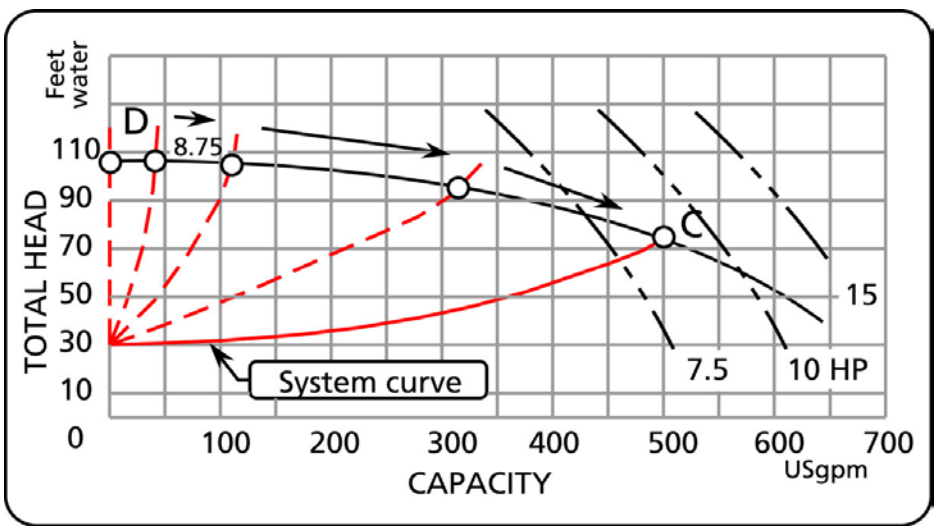

Figure 4-11 Starting the pump with discharge valve closed.

Method 2 (discharge valve open)

The alternate method is to start with the discharge valve open. When the pump is just starting the initial RPM is low; the pump produces little head and flow. As the pump accelerates, it will intersect the system curve at point A at 200 RPM, point B at 500 RPM and finally point C at the normal rotational speed of the pump and motor. This happens very quickly since the motor will reach its full operating speed in a few seconds. The disadvantage of starting with this method is that high initial velocity can produce a severe water hammer, shaking pipes and equipment.

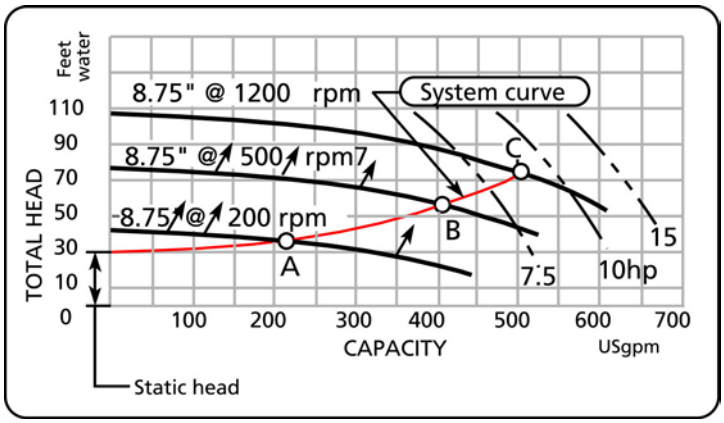

Figure 4-12 Starting the pump with discharge valve open.

Method 1 is the preferred method of starting a pump. To minimize personnel requirements, it is the practice in many plants to equip their pumps with remote operated on-off discharge valves for large pumps (larger than 500 USGPM). The valves are opened automatically by a control system, thereby providing a smooth start-up.

4.6 SAFETY FACTOR ON TOTAL HEAD OR CAPACITY

Before finalizing the choice of impeller size, consider whether you should apply an extra capacity factor or not (capacity = flow). There are no rules; however, since we normally size pumps for someone else, it is a good idea to agree on a safety margin with your client. Point 1 on impeller curve A (see Figure 4-13) is the operating point (on system curve a) before a capacity factor is applied.

Safety margin on total head

If we apply a head factor of 10%, we will have to select impeller C. The operating point will now be at point 2. This means that there must be a different system curve than curve a, which is curve c. The only explanation for this is that we have underestimated the pressure drops throughout the system. If it turns out that the pressure drop calculations were right after all, and that we really are operating with curve a, then the operating point will shift from point 2 to 3 on impeller curve C (assuming there is sufficient horsepower to operate at point 3). If we need to get back to the flow corresponding to point 1 for process reasons, then throttling a valve at the pump discharge will change the operating curve to match curve c and bring the operating point back to point 2.

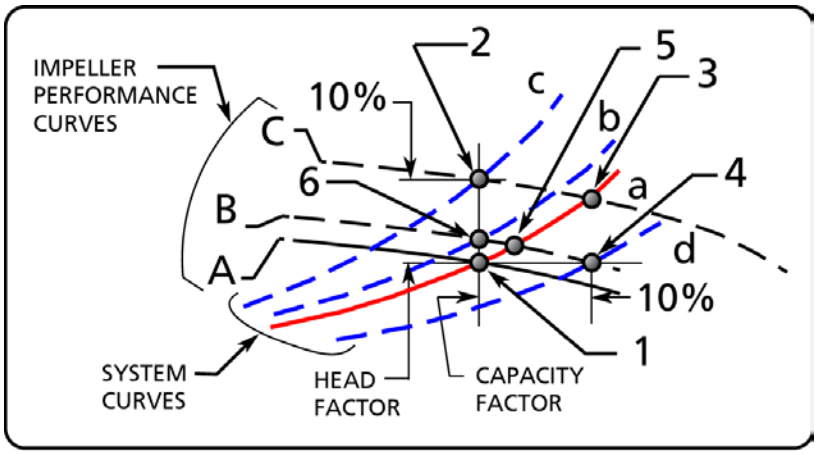

Figure 4-13 Location of capacity factor.

Safety margin on flow

If we apply a capacity factor of say 10%, we will have to select impeller curve B. This means that the system curve is in reality curve d. If in actuality our original flow estimate was correct, then the operating point will shift from point 4 to 5. To get back to the original flow, we have to throttle back so that we shift to point 6, and we will operate on a new system curve b.

Safety margin on total head and flow

If we decide to put on suspenders and a belt (i.e. a head and capacity factor), and assuming that we are operating on system curve a, then we would select impeller curve C and operate at point 3. If we needed to get back to our original flow at point 1, then we would throttle back until we reached point 2.

This is a similar situation to applying solely a head factor because of the shape of the performance and system curves. Therefore, applying a head factor only also provides reserve capacity on flow.

4.7 PUMP OPERATION TO THE RIGHT OR LEFT OF THE BEST EFFICIENCY POINT (B.E.P.)

The impeller is subjected to axial and radial forces. The level of the radial force depends on the pressure within the casing and is taken up by the drive shaft bearings.

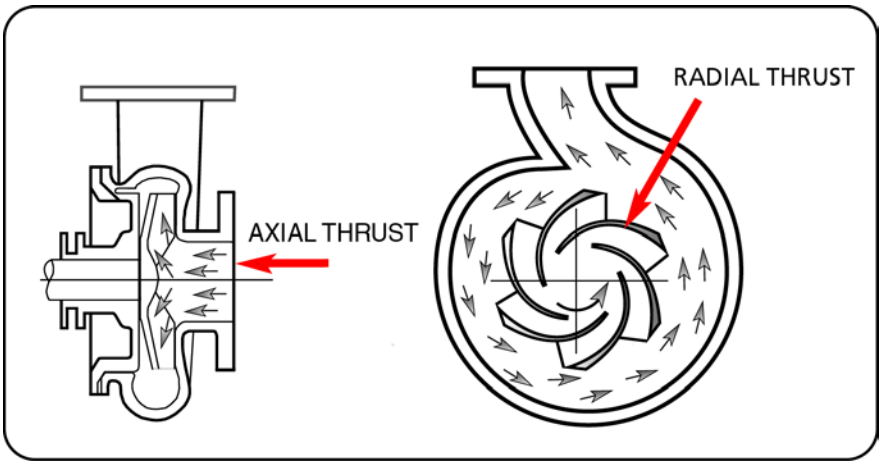

Figure 4-14 The forces on the impeller.

There is also a net radial force whose level depends on the pressure level within the casing, and also on the position of the operating point with respect to the B.E.P.. This force increases rapidly the further away the operating point (see Figure 4-15) gets from the B.E.P. of the pump (for more information see reference 16).

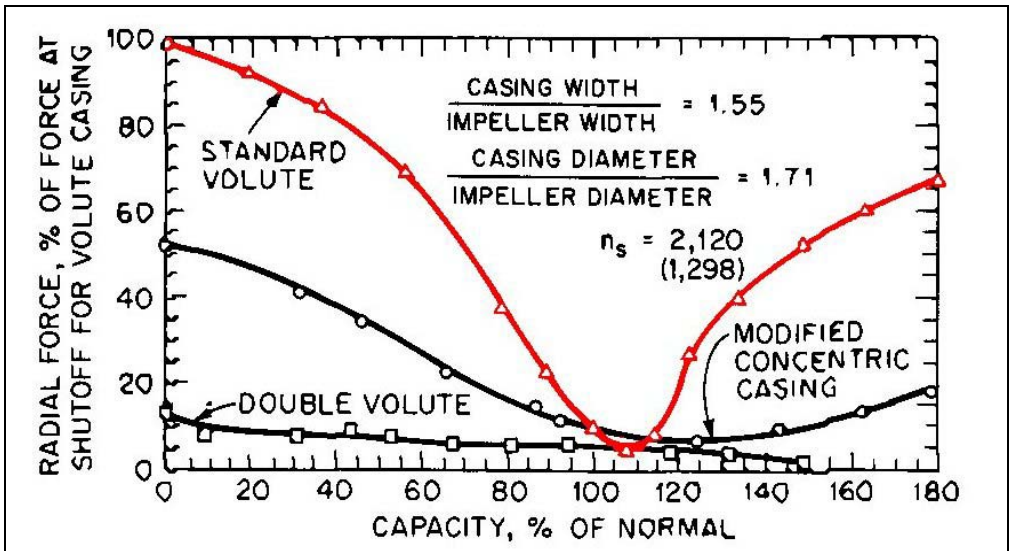

FIG. 61 Comparison of the effect of three casing designs on radial forces for n_s = 2120 (1298). (Ref. 55)

Figure 4-15 Variation on the magnitude of the radial force on the impeller according to the position of the operating point with respect to the B.E.P.(reprinted with permission of McGrawHill).

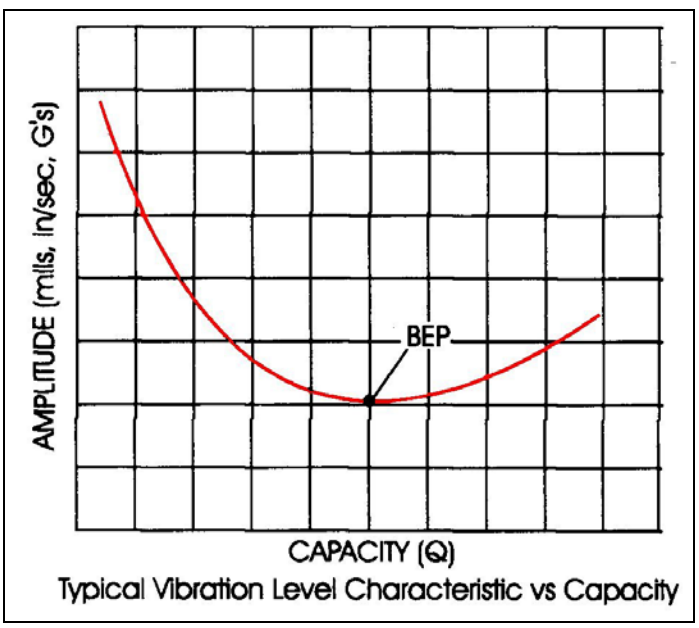

Figure 4-16 Variation in the level of vibration at the operating point vs. the position of the B.E.P.(reprinted with permission of the Goulds pump company).

What are the consequences of operating to the right or left of the B.E.P.?

To the right of the B.E.P., or at high flows

Operating at the far right of the curve, near run-out point of the pump (point B, Figure 4-4) should be avoided. As flow increases, so does the N.P.S.H. required, and therefore cavitation is more likely to occur.

To the left of the B.E.P. or at low flows

Operation of centrifugal pumps at reduced capacity leads to a number of unfavorable results that may take place separately or simultaneously, and should be anticipated and circumvented. Some of these are:

- *Operating at less than best efficiency*
  On occasion, reduced flows may be required by the process. This can be accommodated by a variable speed drive, or by using several pumps. One or more pumps can then be shut down to provide the reduced flow.

- *Higher bearing load*
  If a pump is of a single volute design, it will be subjected to higher radial thrust, which will increase the load on the radial bearing. Therefore, the bearing life would be expected to diminish.

- *Temperature rise*
  As capacity is reduced, the temperature of the pumped liquid increases. To avoid exceeding permissible limits, a minimum flow by-pass is required.

- *Internal re-circulation*
  At certain flows below the best efficiency, all centrifugal pumps are subjected to internal re-circulation, in both the suction and the discharge area of the impeller. This can cause hydraulic surging and damage to the impeller metal, similar to that caused by classic cavitation, but taking place in a different area of the impeller.

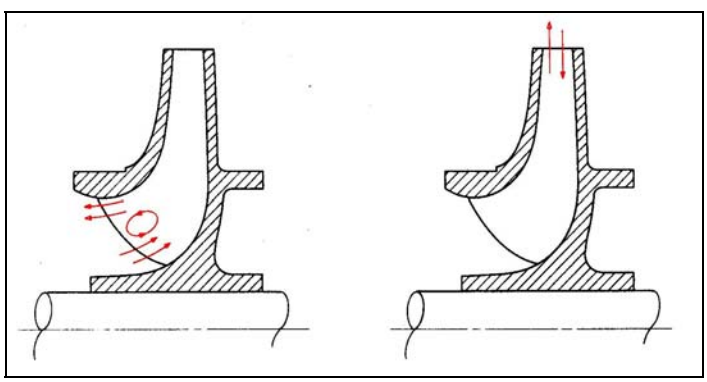

Figure 4-17 Internal recirculation at low flow (reprinted with permission of McGraw-Hill).

4.8 PUMP-SHUT-OFF HEAD

The shut-off head is the Total Head that the pump can deliver at zero flow (see Figure 4-19). The shut-off head is important for 2 reasons.

1. In certain systems (admittedly unusual), the pump discharge line may have to run at a much higher elevation than the final discharge point. The fluid must first reach the higher elevation in the system. If the shut-off head is smaller than the static head corresponding to the high point, then flow will not be established in the system.

2. During start-up and checkout of the pump, a quick way to determine if the pump has the potential capacity to deliver the head and flow required, is to measure the shut-off head. This value can be compared to the shut-off head predicted by the performance curve of the pump.

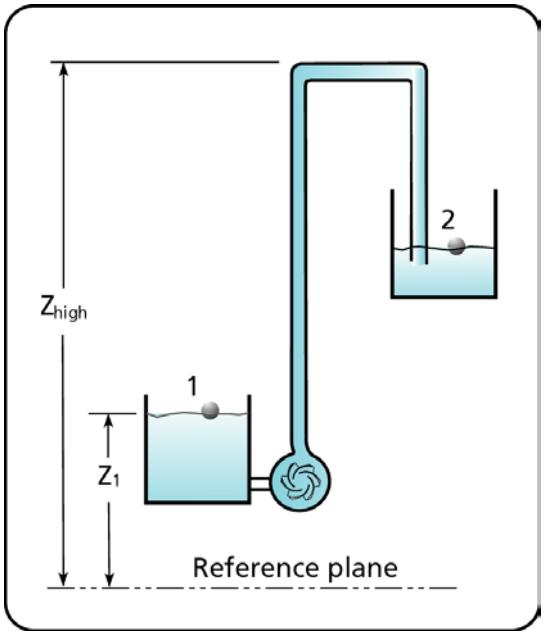

Figure 4-18 Discharge pipe coming from a higher elevation into the discharge tank.

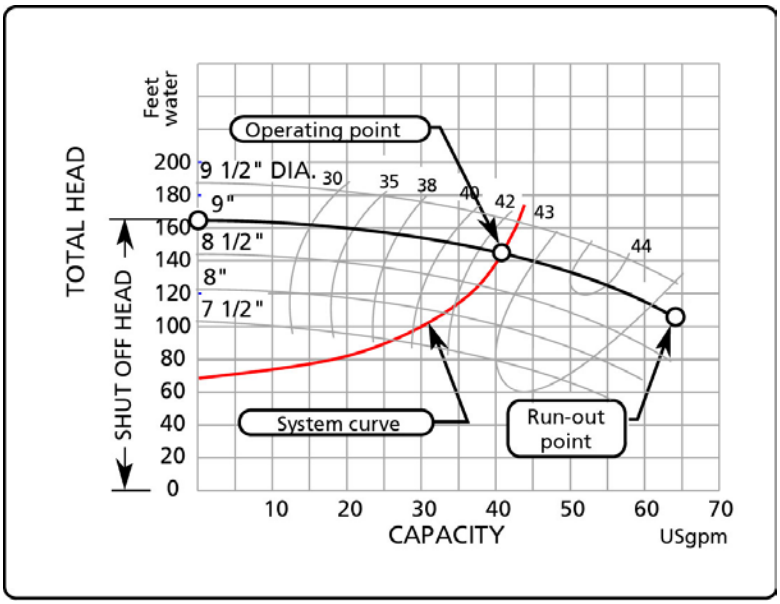

Figure 4-19 Location of pump shut-off head on the performance curve.

4.9 PUMP POWER

A pump's power demand is directly proportional to the difference between the inlet and outlet pressure and the flow rate.

$$P = \Delta p_P \, q$$

where Δp_P is the difference in pressure at the inlet and the outlet of the pump, and q the flow rate. The above equation expressed in imperial units is:

$$P(hp) = \frac{\Delta p_P(psi)\ q(USgal/\min)}{1714.2} \qquad [4\text{-}1]$$

Δp_P is converted to head. From equation [1-5]:

$$\Delta p_P(psi) = \frac{1}{2.31} SG\ \Delta H_P(ft\ of\ fluid) \qquad [4\text{-}2]$$

by substituting equation [4-2] into equation [4-1] we obtain:

$$P(hp) = \frac{SG \Delta H_P(ft\ of\ fluid)\ q(USgal/\min)}{3960} \qquad [4\text{-}3]$$

Equation [4-3] would be true if the pump were 100% efficient. Pump efficiency data are available from the pump manufacturer. All manufacturers test their pumps for various flows, heads and impeller sizes. The resulting efficiency data are mapped on the pump performance curve. The true power required at the pump shaft is:

$$P(hp) = \frac{SG \;\; \Delta H_P(ft\, of\; fluid)\, q(USgal/\min)}{3960 \;\; \eta} \qquad [4\text{-}4]$$

where η is the pump efficiency. If the pump efficiency is 60%, then the value of η is 0.6.

It is good practice to design the pump base in such a way that the selected motor will stand on spacer blocks, being high enough that the next largest motor frame can be installed when the blocks are removed. This will allow the installation of a larger motor without a major disturbance if it should be required.

4.9 AFFINITY LAWS

The affinity laws are derived from a dimensionless analysis of three important parameters that describe pump performance: flow, total head and power (ref: The Pump Handbook by McGraw-Hill, chapter 2). The analysis is based on the reduced impeller being geometrically similar and operated at dynamically similar conditions or equal specific speed.

The affinity laws were developed using the law of similitudes which provide 3 basic relationships.

Flow vs. diameter and speed

$$\frac{Q}{nD^3} = K \qquad \text{or} \qquad \frac{Q_1}{Q_2} = \frac{n_1}{n_2}\frac{D_1^{\,3}}{D_2^{\,3}}$$

Total Head vs. diameter and speed

$$\frac{gH}{n^2 D^2} = K \qquad \text{or} \qquad \frac{H_1}{H_2} = \frac{n_1^{\,2}}{n_2^{\,2}} \frac{D_1^{\,2}}{D_2^{\,2}}$$

Power vs. diameter and speed

$$\frac{P}{\frac{\gamma}{g} n^3 D^5} = K \qquad \text{or} \qquad \frac{P_1}{P_2} = \frac{n_1^{\,3}}{n_2^{\,3}} \frac{D_1^{\,5}}{D_2^{\,5}}$$

where subscripts 1 and 2 denote the value before and after the change. P is the power, n the speed, D the impeller diameter, H the total head.

If the speed is fixed the affinity laws become:

$$\frac{Q_1}{Q_2} = \frac{D_1^{\,3}}{D_2^{\,3}} \qquad \frac{H_1}{H_2} = \frac{D_1^{\,2}}{D_2^{\,2}} \qquad \frac{P_1}{P_2} = \frac{D_1^{\,5}}{D_2^{\,5}}$$

If the diameter is fixed the affinity laws become:

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \qquad \frac{H_1}{H_2} = \frac{n_1^{\,2}}{n_2^{\,2}} \qquad \frac{P_1}{P_2} = \frac{n_1^{\,3}}{n_2^{\,3}}$$

The process of arriving at the affinity laws assumes that the two operating points that are being compared are at the same efficiency. The relationship between two operating points, say 1 and 2, depends on the shape of the system curve (see Figure 4-20). The points that lie on system curve A will all be approximately at the same efficiency. Whereas the points that lie on system curve B are not. The affinity laws do not apply to points that belong to system curve B. System curve B describes a system with a relatively high static head vs. system curve A which has a low static head.

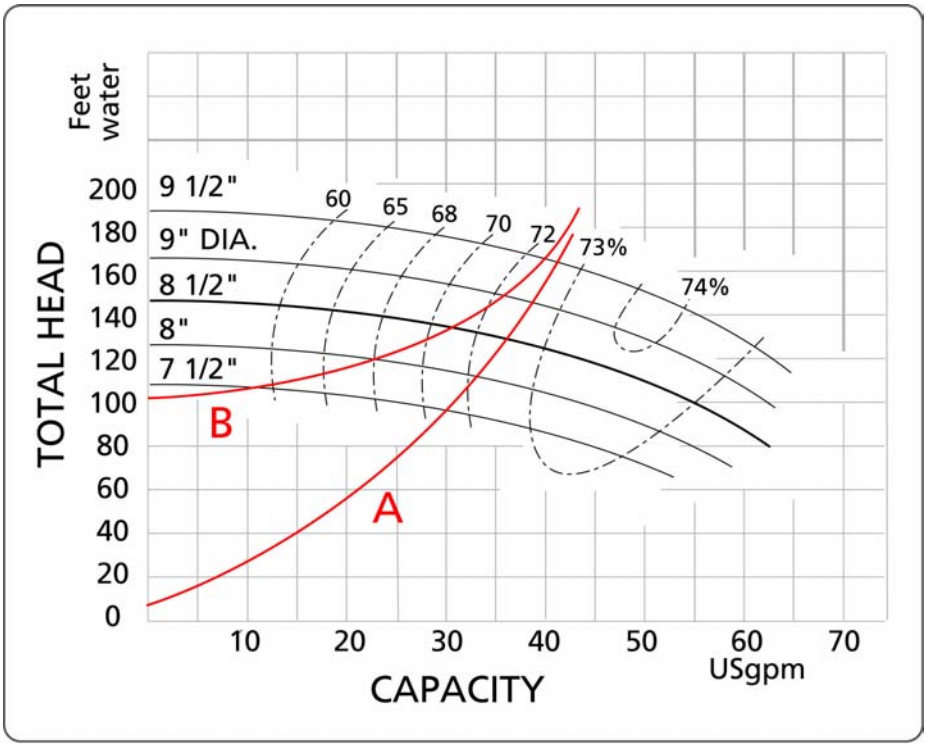

Figure 4-20 Limitation on the use of the affinity laws.

Diameter reduction To reduce costs pump casings are designed to accommodate several different impellers. Also, a variety of operating requirements can be met by changing the outside diameter of a given radial impeller. Euler's equation shows that the head should be proportional to $(nD)^2$ provided that the exit velocity triangles remain the same before and after cutting. This is the usual assumption and leads to:

$$\frac{Q_1}{Q_2} = \frac{n_1 D_1}{n_2 D_2} \qquad \frac{H_1}{H_2} = \frac{n_1^2 D_1^2}{n_2^2 D_2^2} \qquad \frac{P_1}{P_2} = \frac{n_1^3 D_1^3}{n_2^3 D_2^3}$$

Which apply only to a given impeller with altered D and constant efficiency but not a geometrically similar series of impellers.

If that is the case then the affinity laws can be used to predict the performance of the pump at different diameters for the same speed or different speed for the same diameter. Since in practice impellers of different diameters are not geometrically identical, the author's of the section called Performance Parameters in the Pump Handbook recommend to limit the use of this technique to a change of impeller diameter no greater than 10 to 20%. In order to avoid over cutting the impeller, it is recommended that the trimming be done in steps with careful measurement of the results. At each step compare your predicted performance with the measured one and adjust as necessary.

Sizing things up ….

1. Head is independent of fluid density.

2. For fluids other than water, it should be determined if the fluid is Newtonian. Many pure fluids are Newtonian (see the table on rheological properties of fluids in Appendix A as a starting point). If the fluid is non-Newtonian, depending on the severity of its departure from Newtonian behavior, a centrifugal pump may not be an appropriate pumping device. If the fluid is Newtonian but with a different viscosity than water, apply the correction factors to the performance curve suggested by the pump manufacturer (see reference 1 and 2).

3. Order the pump base with spacer blocks for the motor, allowing the next larger frame to be installed when the blocks are removed.

4. Select a pump size and speed in such a way that the impeller is not close to its maximum size (i.e. within 2/3's of it's total range) in order to allow for a future increase in capacity. Locate the operating point somewhere between 110% and 80% of the B.E.P. flow.

5. During the selection process, if the operating point falls between two performance curves, by interpolation calculate the exact impeller size required to intersect the operating point. Impeller size is easily machined down to the correct diameter.

6. Determine the total static head corresponding to the highest point of the system and make sure that it is less than the pump shut-off head.

7. Finally, beware using the affinity laws for calculating a new pump diameter or speed for systems that have a high static head, the affinity laws apply only between two points that are at the same efficiency.