1

Principles of Centrifugal Process Pumps

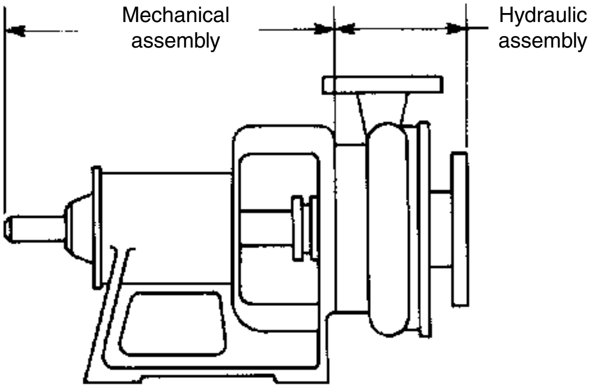
Pumps, of course, are simple machines that lift, transfer, or otherwise move fluid from one place to another. They are usually configured to use the rotational (kinetic) energy from an impeller to impart motion to a fluid. The impeller is located on a shaft; together, shaft and impeller(s) make up the rotor. This rotor is surrounded by a casing; located in this casing (or pump case) are one or more stationary passageways that direct the fluid to a discharge nozzle. Impeller and casing are the main components of the *hydraulic assembly*; the region or envelope containing bearings and seals is called the *mechanical assembly* or power end (Figure 1.1).

Many process pumps are designed and constructed to facilitate field repair. On these so-called "back pull-out" pumps, shop maintenance can be performed, while the casing and its associated suction and discharge piping (Figure 1.2) are left undisturbed. Although *operating* in the hydraulic end, the impeller remains with the power end during removal from the field.

The rotating impeller (Figure 1.3) is usually constructed with swept-back vanes, and the fluid is accelerated from the rotating impeller to the stationary passages into the surrounding casing.

In this manner, kinetic energy is added to the fluid stream (also called pumpage) as it enters the impeller's suction eye (A on Figure 1.3), travels through the impeller, and is then flung outward toward the impeller's periphery. After the fluid exits the impeller, it gradually decelerates to a much lower velocity in the stationary casing, called a volute casing, where the fluid stream's kinetic energy is converted into pressure energy (also called pressure head). The combination of the pump suction (inlet) pressure and the additional pressure head generated by the impeller creates a final pump discharge pressure that is higher than the suction pressure [3].

Pump Wisdom: Essential Centrifugal Pump Knowledge for Operators and Specialists,
Second Edition. Robert X. Perez and Heinz P. Bloch.



Hydraulic assembly

Impeller/propeller
Suction inlet
Volute
Seal rings

Mechanical assembly

Shaft seal

- Labyrinth
- Mechanical
- Packing

Shaft
Bearings
Housing/frame
Drive coupling/sheave

Figure 1.1 Principal components of an elementary process pump. *Source:* SKF USA, Inc. [1].

Pump Performance: Head and Flow

Pump performance is always described in terms of head H produced at a given flow capability Q, and hydraulic efficiency η attained at any particular intersection of H and Q. Head is customarily plotted on the vertical scale or vertical axis (the left of the two y-axes) of Figure 1.4; it is expressed in feet (or meters). Hydraulic efficiency is often plotted on another vertical scale, the right of the two vertical scales, i.e. the y-axis in this generalized plot.

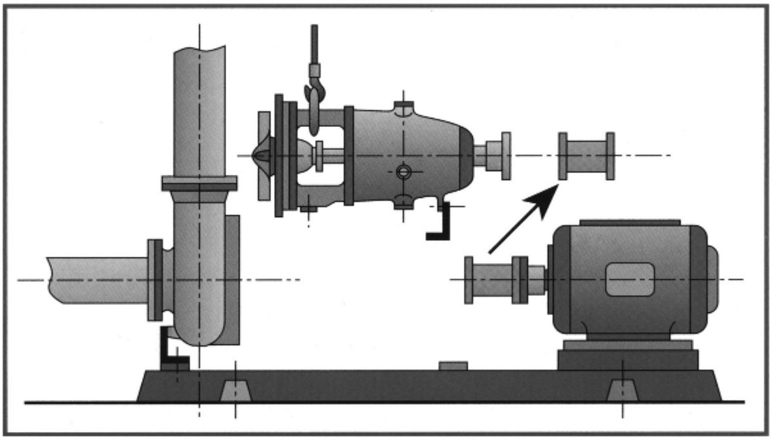


Figure 1.2 Typical process pump with suction flow entering horizontally and vertically oriented discharge pipe leaving the casing tangentially. *Source:* Emile Egger & Cie. [2].

Head is related to the difference between discharge pressure and suction pressure at the respective pump nozzles. Head is a simple concept, but this is where consideration of the impeller tip speed is important. The higher the shaft rpm and the larger the impeller diameter, the higher will be the impeller tip speed – actually its peripheral velocity.

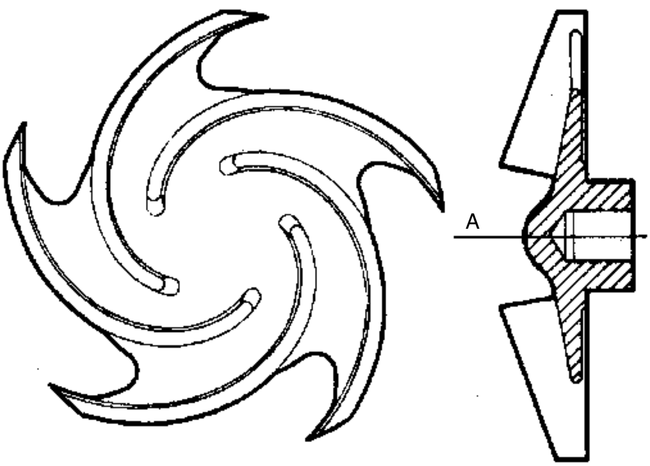


Figure 1.3 A semiopen impeller with five vanes. As shown, the impeller is configured for counterclockwise rotation about a centerline "A."

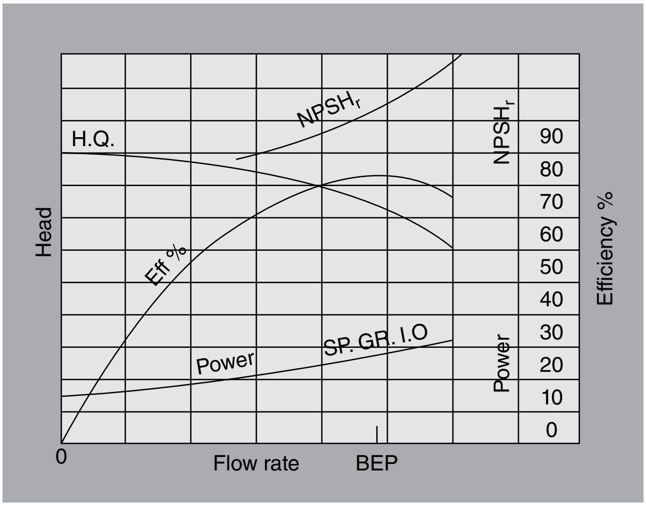


Figure 1.4 Typical "*H*–*Q*" performance curves are sloped as shown here. The best efficiency point (BEP) is marked with a small triangle; power and other parameters are often displayed on the same plot.

The concept of head can be visualized by thinking of a vertical pipe bolted to the outlet (the discharge nozzle) of a pump. In this imaginary pipe, a column of fluid would rise to a height "H". If the vertical pipe would be attached to the discharge nozzle of a pump with *higher* impeller tip speed, the fluid would rise to a greater height "$H+$". It is important to note that the height of a column of liquid, H or $H+$, is a function only of the impeller tip speed. The specific gravity of the liquid affects power demand but does not influence either H or $H+$. However, the resulting discharge pressure *does* depend on the liquid density (specific gravity or Sp.G.). For water (with an Sp.G. of 1.0), an H of 2.31 ft equals 1 psi (pound-per-square-inch), while for alcohol, which might have a Sp.G. of 0.5, a column height or head H of 4.62 ft equals 1 psi. So if a certain fluid had an Sp.G. of 1.28, a column height (head H) of $2.31/1.28 = 1.8$ ft would equal a pressure of 1 psi.

For reasons of material strength and reasonably priced metallurgy, one usually limits the head per stage to about 700 ft. This is a fairly important rule-of-thumb limit to remember. When too many similar rule-of-thumb *limits* combine, one cannot expect pump reliability to be at its highest. As an example, say a particular impeller-to-shaft fit is to have 0.0002–0.0015 in. clearance on average size impeller hubs. With a clearance fit of 0.0015 in., one might anticipate a somewhat greater failure risk if this upper limit were found on an impeller operating with maximum allowable diameter.

On Figure 1.4, the point of zero flow (where the curve intersects the *y*-axis) is called the shut-off point. The point at which operating efficiency is at a peak is called the best efficiency point, or BEP. Head rise from BEP to shut-off is often chosen around 10–15% of differential head. This choice makes it easy to modulate pump flow by adjusting control valve open area based on monitoring pressure. Pumps “operate on their curves” and knowledge of what pressure relates to what flow allows technicians to program control loops.

The generalized depictions in Figure 1.4 also contain a curve labeled $NPSH_r$, which stands for Net Positive Suction Head required. This is the head of liquid that must exist at the edge of the inlet vanes of an impeller to allow liquid transport without causing undue vaporization. It is a function of impeller geometry and size and is determined by factory testing. $NPSH_r$ can range from a few feet to a three-digit number. At all times, the head of liquid *available* at the impeller inlet ($NPSH_a$) must exceed the required $NPSH_r$.

Operation at Zero Flow

The rate of flow through a pump is labeled Q (gpm) and is plotted on the horizontal axis (the *x*-scale). Note that for a given speed and for every value of head H we read off on the *y*-axis, there is a corresponding value of Q on the *x*-axis. This plotted relationship is expressed as “the pump is running on its curve.” Pump H–Q curves are plotted to commence at zero flow and highest head. Process pumps need a continually rising curve inclination and a curve with a hump somewhere along its inclined line will not serve the reliability-focused user. Operation at zero flow is not allowed and, if over perhaps a minute’s duration, could cause temperature rise and internal recirculation effects that might destroy most pumps.

But remember that this curve is valid only for this particular impeller pattern, geometry, size, and operation at the speed indicated by the manufacturer or entity that produced the curve. Curve steepness or inclination has to do with the number of vanes in that impeller; curve steepness is also affected by the angle each vane makes relative to the impeller hub. In general, curve shape is verified by physical testing at the manufacturer’s facility. Once the entire pump is installed in the field, it can be re-tested periodically by the owner–purchaser for degradation and wear progression. Power draw may have been affected by seals and couplings that differ from the ones used on the manufacturer’s test stand. Occasionally, high efficiencies are alluded to in a manufacturer’s literature when bearing, seal, and coupling losses are not included in the vendor’s test reports.

Impellers and Rotors

Regardless of flow classification centrifugal pumps range in size from tiny pumps to very big pumps. The tiny ones might be used in medical or laboratory applications; the extremely large pumps may move many thousands of liters or even gallons per second from flooded lowlands to the open sea.

All six of the impellers in Figure 1.5 are shown with a hub fastening the impeller to the shaft, and each of the first five impellers is shown as a hub-and-disc version with an impeller cover. The cover (or "shroud") identifies the first five as "closed" impellers; recall that Figure 1.3 had depicted a semiopen impeller. Semiopen impellers are designed and fabricated without the cover. Finally, open impellers come with free-standing vanes welded to or integrally cast into the hub. Since the latter incorporate neither disc nor cover, they are often used in viscous or fibrous paper stock applications.

To properly function, a semiopen impeller must operate in close proximity to a casing internal surface, which is why axial adjustment features are needed with these impellers. Axial location is a bit less critical with closed impellers. Except on axial flow pumps, fluid exits the impeller in the radial direction. Radial and mixed flow pumps are either single or double suction designs; both will be shown later. Once the impellers are fastened to a shaft, the resulting assembly is called a rotor.

In radial and mixed flow pumps, the number of impellers following each other, typically called "stages," can range from one to as many as will make such multistage pumps practical and economical to manufacture. Horizontal shaft pumps with up to 12 stages are not uncommon; using more than 12 stages on a horizontal shaft risks causing the rotor to resonate or vibrate at a so-called "critical speed." Vertical shaft pumps have been designed with 48 or more stages. In vertical

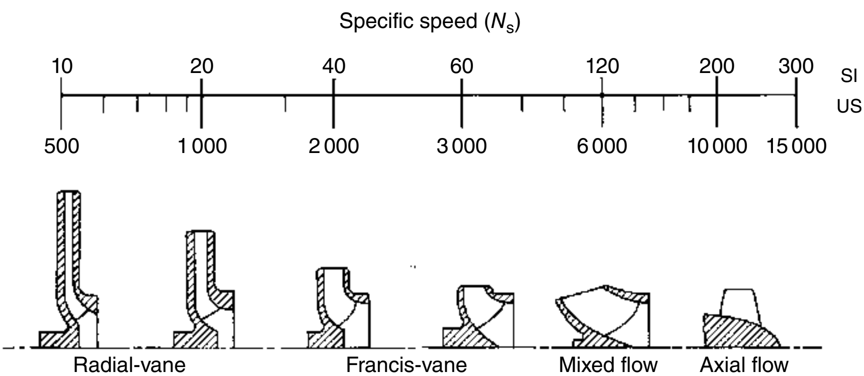


Figure 1.5 General flow classifications of process pump impellers.

pumps, shaft support bushings are relatively lightly loaded; they are spaced so as to minimize vibration risk.

The Meaning of Specific Speed

Pump impeller flow classifications and the general meaning of *specific speed* deserve further discussion. Moving from left to right in Figure 1.5, the various impeller geometries reflect selections that start with high differential pressure capabilities and end with progressively lower differential pressure capabilities. Differential pressure is simply discharge pressure minus suction pressure.

Specific speed calculations are a function of several impeller parameters; the mathematical expression includes exponents and is found later, in Figure 1.6. Staying with Figure 1.5 and again moving from left to right, we can reason that larger throughputs (flows) are more likely achieved by the configurations at right, whereas larger pressure ratios (discharge pressure divided by suction pressure) are usually achieved by the impeller geometries closer to the left of the illustration.

Impellers toward the right are more efficient than those near the left, and pump designers use the parameter *specific speed* (N_s) to bracket pump hydraulic efficiency attainment and other expected attributes of a particular impeller configurations and size. Please be sure not to confuse a very similar sounding parameter, *pump suction specific speed* (N_{ss} or N_{sss}), with the *specific speed* (N_s). For now, we are strictly addressing *specific speed* (N_s).

As an example, observe the customary use, whereby with N and Q – the typical given parameters that define centrifugal process pumps – one determines a pivot point. Next, with pivot point and head H, one can easily determine N_s. In Figure 1.5, Ns is somewhere between 500 and 15 000 on the US scale. Whenever we find ourselves in that range, we know such a pump exists, and we can even observe the general impeller shape. Keep in mind that thousands of impeller combinations and geometries exist. Impellers with covers are the most prominent in hydrocarbon processes, and an uneven number of impeller vanes is favored over even numbers of vanes for reasons of vibration suppression.

Pump specific speed, Figure 1.6, might be of primary interest to pump designers, but average users will also find it useful. On the lower right, the illustration gives the equation for N_s; it will be easy to see how N_s is related to the shaft speed N (rpm), flow Q (gpm or gallons/minute), and head H (expressed in feet). This mathematical expression also has two strange-looking exponents in it, but the N_s nomogram conveys more than meets the eye and can be quite helpful.

If now, we had the same or some other Q and would want to see what happens at some other speed, we would again draw a straight line to establish the pivot point. Drawing a line from whatever H is specified through the pivot

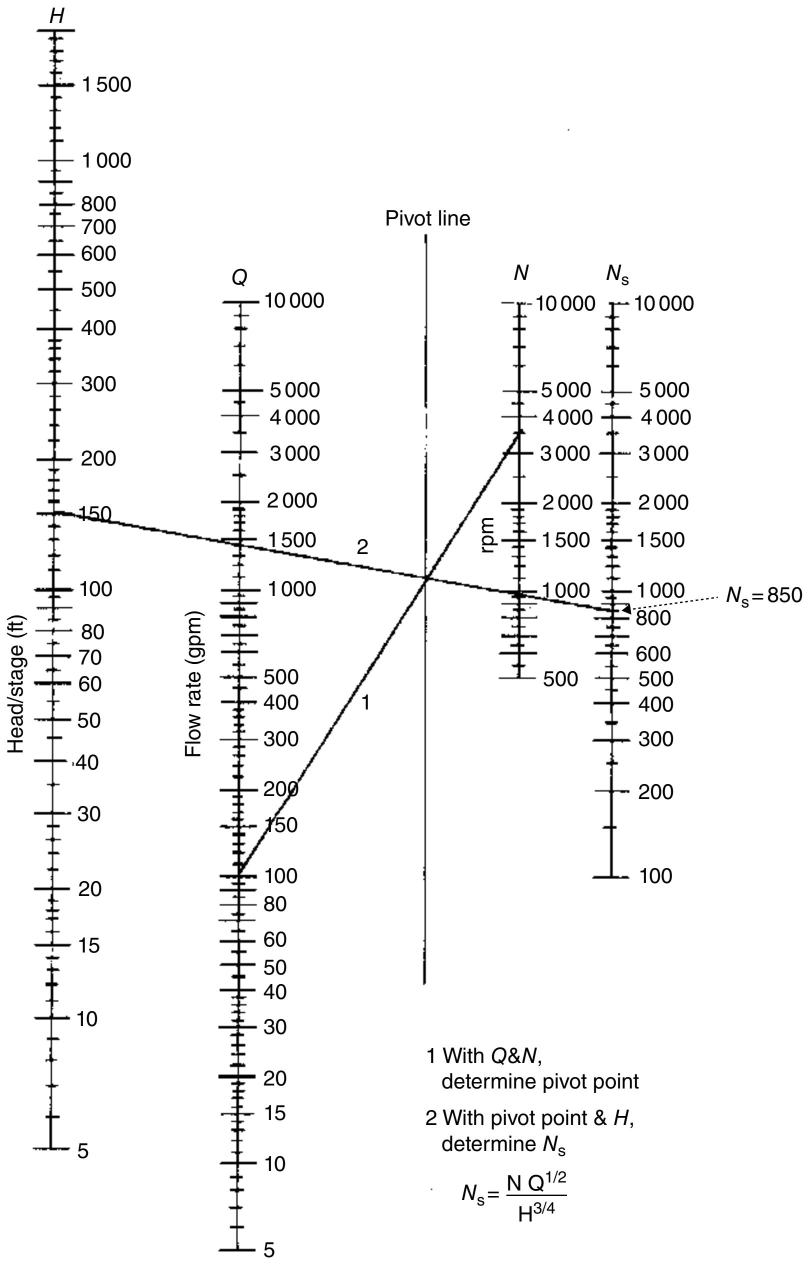


Figure 1.6 Pump specific speed nomogram allowing quick estimations. Shown in this illustration is a hypothetical pump application with a flow of 100 gpm operating at 3600 rpm (line 1). To develop 150 ft of head with a single stage, an impeller with a specific speed of 850 (line 2) would be required.

point and to N_s we would not like to select pumps with an N_s outside the rule-of-thumb range from 500 to 15 000. In another example, we might, after establishing the pivot point, wish to determine what happens if we select an impeller with the maximum head capability of 700 ft and draw a line through the pivot point. If the resulting N_s is too low, we would try a higher speed N and see what happens.

While there are always fringe applications in terms of size and flow rate, this book deals with centrifugal pumps in process plants. These pumps are related to the generic illustrations of Figures 1.1 and 1.2 and others in this chapter. All would somewhat typically – but by no means exclusively – range from 3 to perhaps 300 hp (2–225 kW).

Process Pump Types

The elementary process pumps illustrated in Figures 1.1 and 1.2 probably incorporate one of the radial vane impellers shown in Figure 1.5. If a certain differential pressure is to be achieved together with higher flows, such a pump is often designed with a double-flow impeller (Figure 1.7). One of the side benefits of double-flow impellers is very good axial thrust equalization (axial balance). A small thrust bearing will often suffice; it is shown here in the left bearing housing.

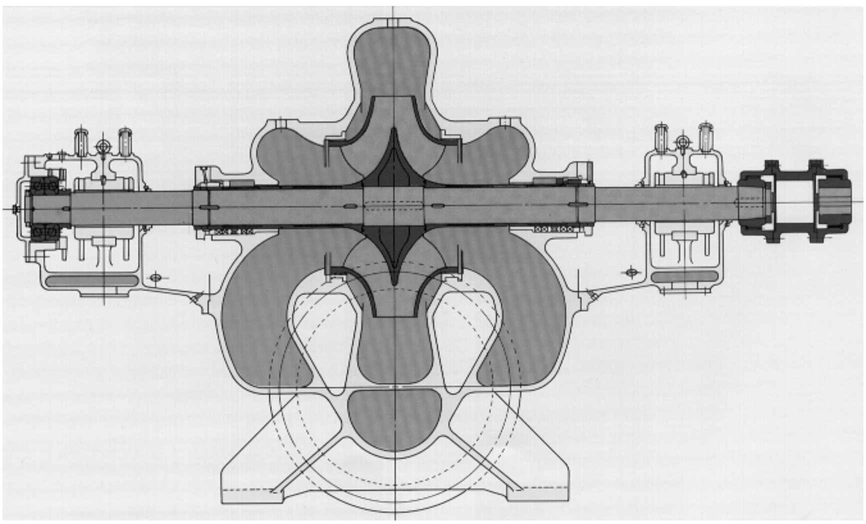


Figure 1.7 Double-flow impellers are used for higher flows and relatively equalized (balanced) axial thrust. *Source:* Mitsubishi Heavy Industries, Ltd. [4].

Note that the two radial bearings are plain, or sleeve-type. Certain sleeve bearings have relatively high speed capability.

If elevated pressures are needed, several impellers are lined up in series on the same pump rotor. Of course, this would then turn the pump into a multistage model Figure 1.8.

Process Pump Mechanical Response to Flow Changes

After the pumped fluid (also labeled pumpage, or flow) leaves at the impeller tip, it must be channeled into a stationary passageway that merges into the discharge nozzle. Many different types of passageway designs (single or multiple volutes, vaned diffusers, etc.) are available. Their respective geometry interacts with the flow and creates radial force action of different magnitude around the periphery of an impeller (Figure 1.9). These forces tend to deflect the pump shaft; they are greater at part-flow than at full flow.

Recirculation and Cavitation

Recirculation is a flow reversal near either the inlet or discharge of a centrifugal pump. This flow reversal produces cavitation-erosion damage that starts on the high-pressure side of an impeller vane and proceeds through the metal to the low-pressure side [5].

Pump-internal recirculation can cause surging and cavitation even when the available net positive suction head ($NPSH_a$) exceeds the manufacture's published

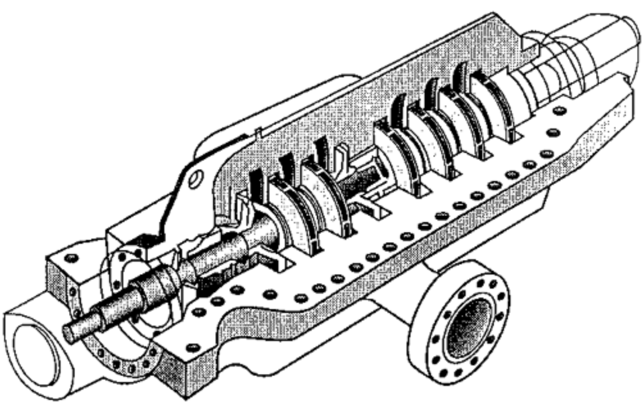


Figure 1.8 A multistage centrifugal process pump.

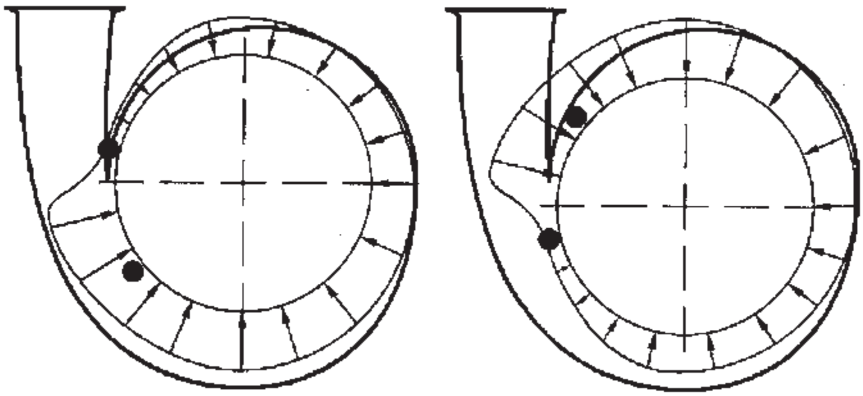


Figure 1.9 Direction and magnitude of fluid forces change at different flows.
Source: World Pumps, February 2010, pp. 19.

NPSH required (NPSHr) by considerable margins. Also, extensive damage to the pressure side of impeller vanes has been observed in pumps operating at reduced flow rates. These are the obvious results of recirculation; however, more subtle symptoms and operational difficulties have been identified in pumps operating in the recirculation zone.

Symptoms of discharge recirculation are the following:

- Cavitation damage to the pressure side of the vane at the discharge
- Axial movement of the shaft, sometimes accompanied by damage to the thrust bearing
- Cracking or failure of the impeller shrouds at the discharge
- Shaft failure on the outboard end of double-suction and multistage pumps
- Cavitation damage to the casing tongue (see Chapter 11) or diffuser vanes

Symptoms of suction recirculation are the following:

- Cavitation damage to the pressure side of the vanes at the inlet
- Cavitation damage to the stationary vanes in the suction
- Random crackling noise in the suction; this contrasts with the steady crackling noise caused by inadequate net positive suction head
- Surging of the suction flow

A quick-reference illustration was provided by Warren Fraser Figure 1.10. We should note that recirculation and the attendant failure risks are low in pumps delivering 2500 gpm or less at heads up to 150 ft. In those pumps, energy levels may not be high enough even if the pump operates in the recirculation zone. As a general rule for such pumps, minimum flow can be set at 50% of recirculation

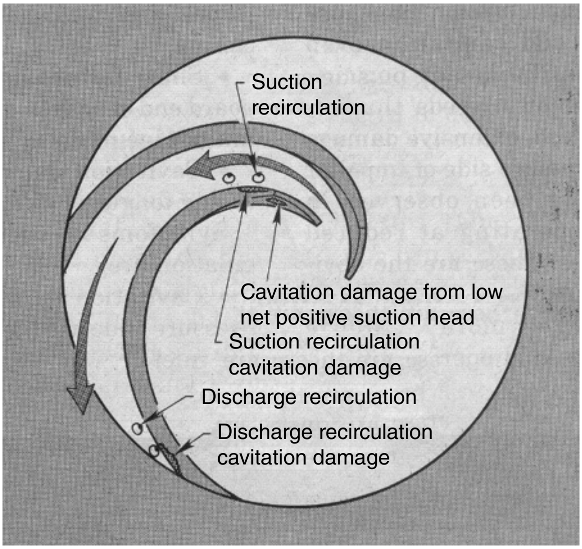


Figure 1.10 Where and why impeller vanes get damaged. *Source:* Fraser [5].

flow for continuous operation and 25% of recirculation flow for intermittent operation [6].

The Importance of Suction Specific Speed

Note that pump suction specific speed (N_{ss} or N_{sss}) differs from the pump specific speed N_s discussed earlier. For installations delivering over 2500 gpm and with suction specific speeds over 9000, greater care is needed. Suction specific speed (N_{sss} or N_{ss}) is calculated by the straightforward mathematical expression:

$$N_{ss}\frac{(\mathrm{r/min})\left[(\mathrm{gal/min})/\mathrm{eye}\right]^{1/2}}{(\mathrm{NPSH_r})^{3/4}}$$

wherein both the flow rate and $\mathrm{NPSH_r}$ pertain to conditions published by the manufacturer. In each case, these conditions (flow in gpm) and $\mathrm{NPSH_r}$ are observed on the maximum available impeller diameter for that particular pump.

The higher the design suction specific speed, the closer the point of *suction* recirculation to what is commonly described as rated capacity. Similarly, the closer the *discharge* recirculation capacity is to rated capacity, the higher the efficiency. Pump system designers are tempted to aim for highest possible efficiency and suction specific speed. However, such designs might result in systems with either limited pump operating range or, if operated inside the recirculation range, disappointing reliability and frequent failures.

Although more precise calculations are available, trend curves of probable NPSH**r** for minimum recirculation and zero cavitation-erosion in water, Figure 1.11, are sufficiently accurate to warrant our attention [7]. The NPSH**r** needed for *zero* damage to impellers and other pump components may be many times that published in the manufacturer's literature. The manufacturers' NPSH**r** plot (lowermost curve in Figure 1.11) is based on observing a 3% drop in discharge head or pressure; at Q = 100%, we note NPSH**r** = 100% of the manufacturer's claim. Unfortunately, whenever this 3% fluctuation occurs, a measure of damage may already be in progress. Assume the true NPSH$\underline{r}$ is as shown in Figure 1.11 and aim to provide an NPSH$_a$ in excess of this true NPSH**r**.

In Ref. [7], Irving Taylor compiled his general observations and alerted us to this fact. He cautioned against considering his curves totally accurate and mentioned the demarcation line between low and high suction specific speeds somewhere between 8000 and 12 000. Many data points taken after 1980 point to 8500 or 9000 as numbers of concern. If pumps with N_{ss} numbers higher than 9000 are being operated at flows much higher or lower than BEP, their life expectancy or repair-free operating time will be reduced.

In the decades after Taylor's presentation, controlled testing has been done in many industrialized countries. The various findings have been reduced to relatively accurate calculations that were later published by HI, the Hydraulic

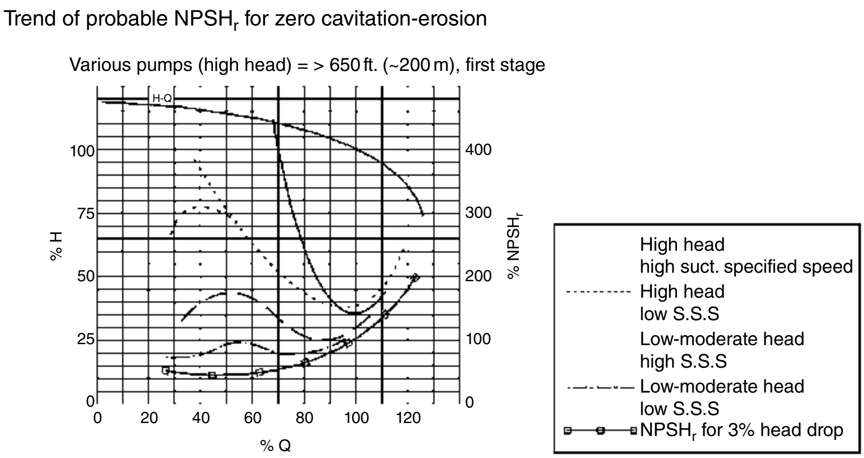


Figure 1.11 Pump manufacturers usually plot only the NPSHr trend associated with the lowermost curve. At that time a head drop or pressure fluctuation of 3% exists and cavitation damage is often experienced. *Source:* Taylor [7].

Institute [8]. Relevant summaries can also be found in Ref. [9]. Calculations based on Refs. [8, 9] determine minimum allowable flow as a percentage of BEP.

Note, again, that recirculation differs from cavitation, a term which essentially describes vapor bubbles that collapse. Cavitation damage is often caused by low net positive suction head available ($NPSH_a$). Such cavitation-related damage starts on the low-pressure side and proceeds to the high-pressure side. An impeller requires a certain net positive suction head; this $NPSH_r$ is simply the pressure needed at the impeller inlet (or eye) for relatively vapor-free flow.

What We Have Learned

Understanding the concepts of N_s and N_{ss} will assist in specifying better pumps. In addition to fluid properties pump life is influenced by throughput. Just as an automobile transmission is designed to work best at particular speeds or optimum gear ratios, pumps have desirable flow ranges. Deviating from optimum flow will influence failure risk and life expectancy.

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