Understand the Fundamentals of Centrifugal Pumps

J. Howard Kelly, P.E.
The Shaw Group, Inc.

Efficient and reliable pumps can significantly improve a plant's bottom line. This article reviews important concepts that a chemical engineer must understand and the factors that need to be considered when selecting radial-flow centrifugal pumps.

Pumps provide a wide range of services in a typical chemical process industries (CPI) plant. They are available with a variety of head/flow combinations and are expected to operate under a variety of process conditions, including many different temperatures, toxicities, and viscosities, and the fluids they handle may be harmless, corrosive, or vapor-forming. Thus, selecting pumps for CPI applications can be challenging. This article will help the novice engineer develop an understanding of pumping issues that can lead to more effective dialog during the early stages of a project, and ultimately improve the efficiency of the requisitioning process and the long-term reliability of the equipment.

Deciphering the pump performance curve

The first step in determining whether a pump will meet your process requirements is to review the pump performance curves (Figure 1). Created by the manufacturer based on actual tests, this diagram depicts the relationships between volumetric flowrate and total dynamic head, efficiency, required net positive suction head (NPSHR), and required power (*i.e.,* brake horsepower, BHP).

The crux of the figure is the plot of volumetric flowrate vs. head, which rises steadily from the rated flowrate (at the right) back to shutoff (or zero flow, at the left).

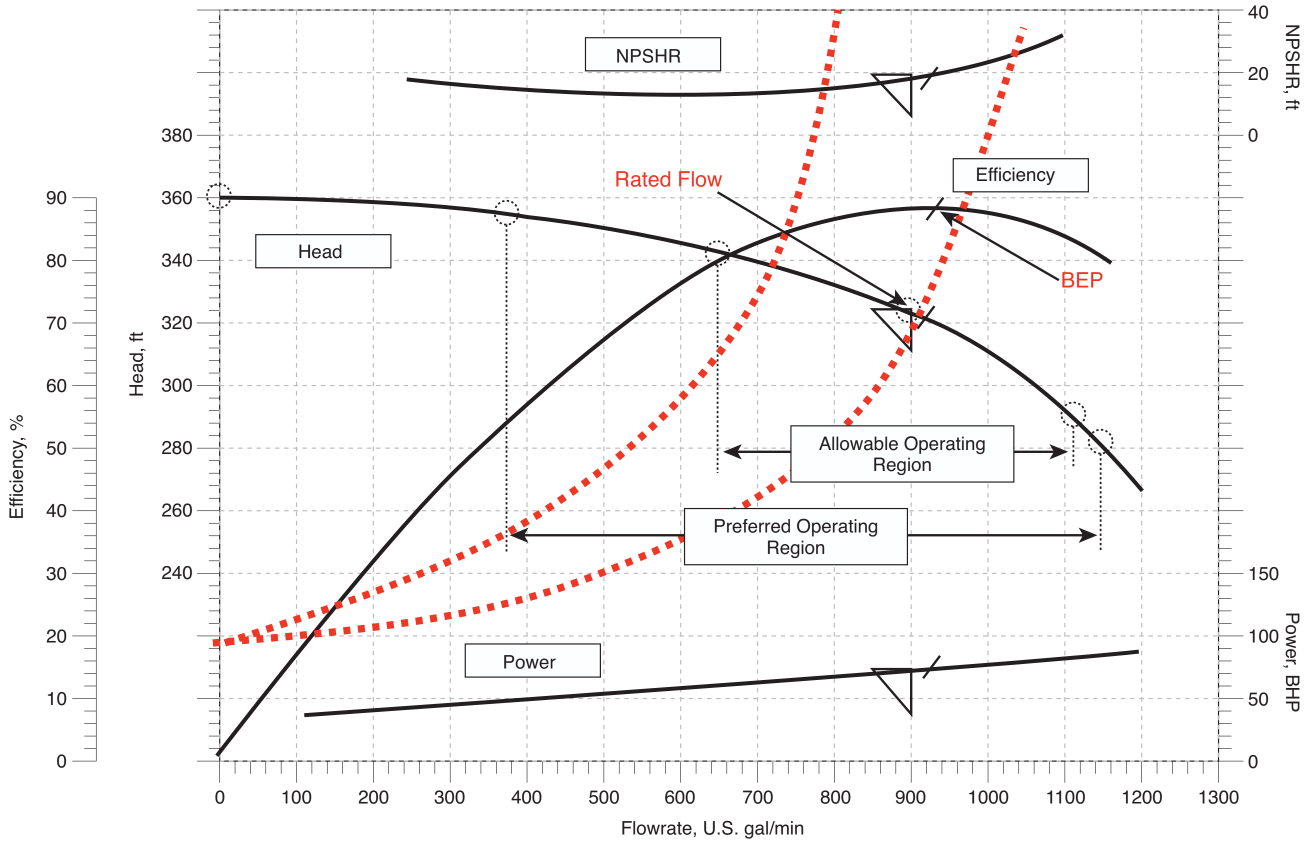
Every pump has a best efficiency point (BEP), which is the flow/head combination corresponding to the highest efficiency. The preferred operating region is between 70% and 120% of the BEP flowrate *(1)*, although most users require the rated flow to fall between 80% and 110% of the BEP. The allowable operating region varies from pump to pump, and is defined as the flow range within which vibrations do not exceed the limits established by the American Petroleum Institute (API) *(1)*. For example, in pumps running at less than 3,600 rpm and absorbing up to 400 hp per stage, vibration on the bearing housing is limited to no more than 0.12 in./s.

Note how efficiency increases with flowrate up to the BEP, then decreases at higher flowrates. The NPSHR curve often has a bowl shape, sloping upward at both low flow and high flow.

The dashed lines represent the process systems and show how process head typically drops as flow drops (which is the opposite of the pump's head curve). In applications where the flow changes over time (such as in a filtration system), a throttling valve or other control device is needed to compensate for the pump's response.

Head

A centrifugal pump increases the absolute pressure of a fluid by adding velocity energy to the fluid ($0.5\ mv^2$) and then converting that to head energy (mgH) in the volute, as shown in Figure 2. The fluid is drawn into the impeller eye (Point 1) at a velocity V_1, which is approximately equal to the volumetric flowrate divided by the cross-sectional area of the impeller eye. The rotation of the impeller increases the velocity and pressure of the fluid (Point 2). When the fluid reaches Point 3, it is slowed down by the increasing area of the volute, and the



▲ **Figure 1.** The characteristics of a centrifugal pump are described by the pump performance curves. Source: *(1)*

velocity is converted into pressure head.

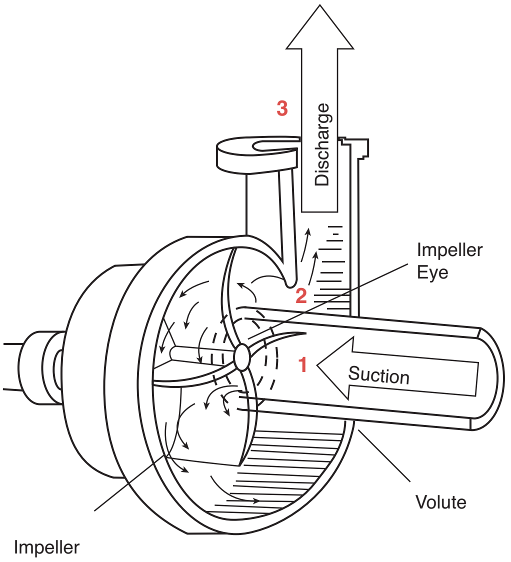
Euler's pump equation relates the head to the change in velocity of the fluid through the impeller shroud as:

$$\Delta H = \eta_{HY}\Delta(UV_u)/g \quad (1)$$

where: η_{HY} is hydraulic efficiency (excluding mechanical losses); U is the centrifugal velocity component of the fluid, and is equal to the angular velocity (Ω) multiplied by impeller radius (r); V_u is the circumferential velocity component of the absolute velocity vector **V**; and g is acceleration due to gravity. (The absolute velocity vector, **V**, at any point along the fluid's route through the vane is calculated by adding the vectors **U** and **v**, where **v** is the velocity of the fluid moving along the impeller viewed from the rotating frame.)

The pressure head (H) can be thought of as the increase in height of a column of fluid that the pump would create if the velocity head were converted, without loss, into elevation head. The actual change in pressure resulting from this head can be calculated using:

$$\Delta P = \rho g H \quad (2)$$



Note that the pumps illustrated in this article have open-vane impellers, which are not allowed by API 610. They are used here for illustrative purposes.

▲ **Figure 2.** Centrifugal pumps increase process head by adding energy to a fluid.

Nomenclature

A_1 = inlet area to the pump, m^2
d = diameter of the impeller, m
f_L = suction pipe friction head loss, m
g = acceleration of gravity, m/s^2
H = head, m
H_{stg} = head per stage, m/stage
L_h = static suction head or lift measured from the impeller centerline, m
m = mass, kg
N = pump speed, rpm
n_q = specific speed based on Q in m^3/s and ΔH in m, dimensionless
N_s = specific speed based on Q in gal/min and ΔH in ft, dimensionless
NPSHA = available net positive suction head, m
NPSHR = required net positive suction head, m
P = pressure, kPa
P_{in} = inlet or suction pressure, kPa
P_v = vapor pressure of a fluid, kPa
Q = volumetric flowrate of a fluid, m^3/s
Q_s = flow coefficient, dimensionless
r = radius of the impeller, m
S = suction specific speed
U = centrifugal velocity component of the fluid, m/s
V = velocity of any point in the fluid, m/s
V_u = circumferential velocity component of the absolute velocity vector $\boldsymbol{V}$, m/s
V_1 = velocity of the fluid at the inlet to the pump, m/s
$\mathbf{U}$ = centrifugal velocity component of the fluid as a vector, m/s
$\mathbf{v}$ = velocity vector component of the fluid moving along the impeller viewed from the rotating plane, m/s
$\mathbf{V}$ = absolute velocity vector of any point in the fluid, m/s

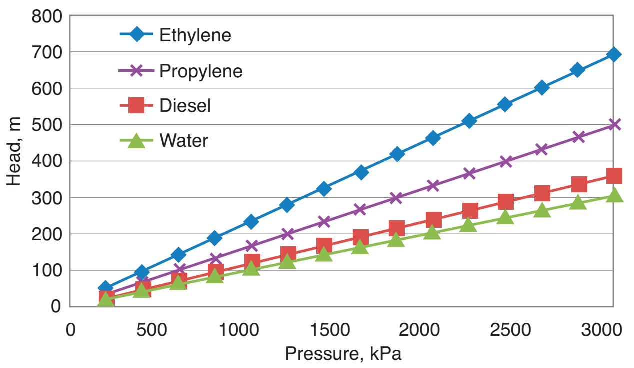
Greek Letters
η_{HY} = hydraulic efficiency (does not include mechanical losses)
ρ = fluid density, kg/m^3
Ψ = head coefficient, dimensionless
Ω = angular velocity, rad/s
Ω_s = specific speed based on the constants Ψ and Q_s

where ΔP is the change in pressure and ρ is the density of the fluid.

Moreover, a low-density fluid (*e.g.*, ethylene) requires more head to produce the same differential pressure as a higher-density fluid (*e.g.*, water). Thus, pumps for low-density fluids must incorporate more stages and/or larger impellers to achieve the same results as a pump producing the same differential pressure with a higher-density fluid. Table 1 and Figure 3 illustrate how fluid density affects the head required to produce the same change in pressure.

Table 1. Effects of fluid density on head.

Property	Ethylene	Propylene	Diesel Fuel	Water
ρ (kg/m^3)	440	614	850	1,000
ΔP (kPa)	2,000	2,000	2,000	2,000
H (m)	463	332	240	204



▲ **Figure 3.** Head required to produce similar pressures is higher for lower density fluids.

The differential pressure required to produce flow is created by adding energy to the fluid through the spinning impeller. The amount of pressure increase is directly related to the density of the fluid (ρ) and the product of the impeller radius and the shaft rotation speed squared $(r\Omega)^2$.

The radius of the impeller determines the pump's size as well as its initial cost. Although fewer problems arise and less NPSH is required at slow speeds, a slow-speed pump requires a larger impeller — and thus will be larger and more expensive — than a higher-speed pump producing the same head. Depending on the application, the increased reliability may offset the higher initial cost of the larger pump.

Specific speed

Specific speed is a convenient variable used to optimize the geometry of a pump. It is based on the scaling and similitude characteristics of centrifugal pumps and the principle of dimensional homogeneity, which states that the head coefficient, Ψ, and the flow coefficient, Q_s, are constant for similar pump geometries:

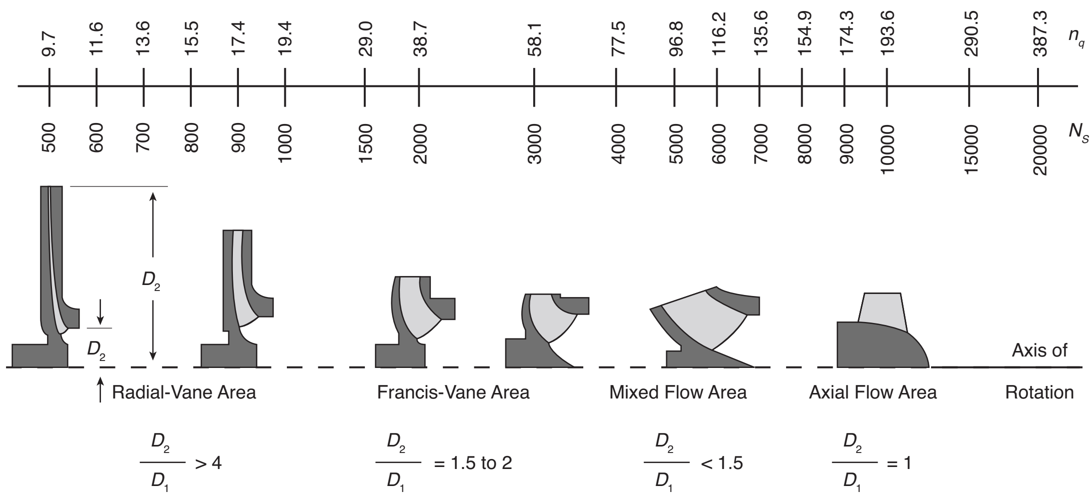
$$\Psi = (g\Delta H)/(r\Omega)^2 = \text{constant} \quad (3)$$
$$Q_s = Q/(r^3\Omega) = \text{constant} \quad (4)$$

where Q is the volumetric flowrate of the fluid.

Specific speed (a dimensionless quantity) is defined as:

$$\Omega_s = Q_s^{0.5}/\Psi^{0.75} \quad (5)$$

▶ **Figure 4.** Impeller designs vary with specific speed constraints imposed by process requirements and impeller speed. Source: *(3)*



For convenience, specific speed is generally represented in terms of pump speed (*N*), volumetric flowrate (*Q*), and head (ΔH):

$$n_q = NQ^{0.5}/\Delta H^{0.75} \tag{6}$$

In Eq. 6, specific speed is represented by n_q, and it is calculated using values of *Q* in m^3/s and ΔH in m. When English units are used for flowrate and head (*Q* in gal/min and ΔH in ft), the equation has the same form but specific speed is represented by N_s:

$$N_s = NQ^{0.5}/\Delta H^{0.75} \tag{7}$$

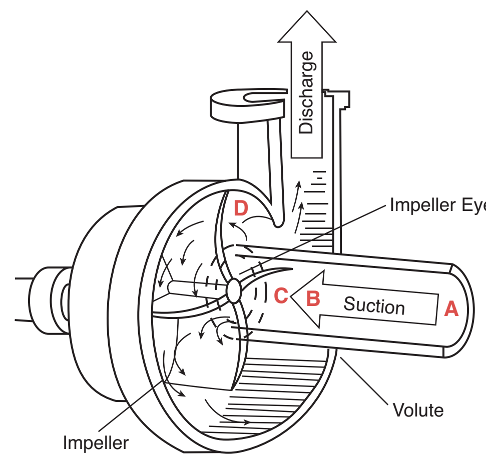
The relationship between n_q and N_s is:

$$n_q = N_s / 51.64 \tag{8}$$

Figure 4 shows the specific speed ranges of various types of impellers. High-flow, low-head pumps (at the right end of the spectrum) require axial-flow impellers. Radial-flow pumps (on the left) have lower flow-to-head ratios, which require larger-diameter impellers to create the higher head and narrower passages to accommodate the lower flow. The smaller passages increase friction, decreasing efficiency and generating heat — which is why low-flow pumps tend to be less efficient than high-flow pumps. At lower specific speeds (to the left of those shown in Figure 4), centrifugal-style impellers are not practical, and positive-displacement pumps that can handle very-low-flow, very-high-head applications are required.

Net positive suction head

NPSH is the difference between the suction pressure and the vapor pressure of the fluid:



▲ **Figure 5.** Static pressure losses occur as the fluid travels into the pump suction and moves in and out of the impeller.

$$NPSH = (P_{in} - P_v)/(\rho g) \tag{9}$$

where P_{in} is the inlet or suction pressure and P_v is the vapor pressure of the fluid.

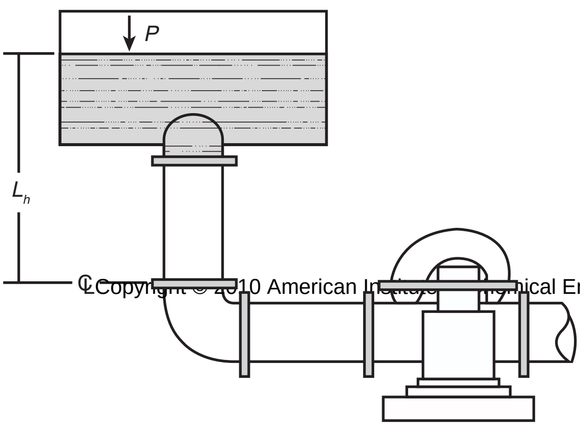
Net positive suction head available (NPSHA) is the amount of NPSH available to the pump, based on the configuration of the upstream piping and anticipated fluid levels of upstream vessels. NSPHR is the amount of NPSH required by the pump to operate properly.

As the fluid moves through the pump, pressure losses occur, as shown in Figure 5, in the inlet passage (Point A to Point B), due to internal frictional (Point B to Point C), and at the blade and within the impeller (Point C to Point D).

If the static pressure drops below the fluid's vapor pressure, the fluid begins to boil, creating vapor bubbles and reducing the density of the fluid. When this occurs, the differential pressure created by the dynamic head of

◀ **Figure 6.** Cavitation damage has occurred on an impeller. Source: The Hydraulic Institute, Inc.

▼ **Figure 7.** NPSHA is illustrated by the height of fluid above the impeller centerline (minus pipe friction losses). Source: Adapted from *(2).*



the impeller decreases (Eq. 2). API 610 defines NPSHR as the amount of suction head needed to limit head loss at the first stage of the pump to 3% (using water as the test fluid). Although hydrocarbons generally require less NPSH than water *(4),* reduction factors for hydrocarbons are not allowed by API.

Insufficient NPSH can also result in pump cavitation. Cavitation occurs when vapor bubbles that have formed in low-static-pressure areas move along the impeller vanes into higher-pressure areas and rapidly collapse. The forces produced by these bubbles as they implode erode the impeller vane surfaces, causing progressive pitting damage such as that shown in Figure 6. Cavitation is associated with a distinct crackling noise that resembles the sound of a fluid starting to boil. Note that at a 3% head loss, cavitation has already begun.

An acceptable margin between NPSHR and NPSHA is therefore required to ensure pump reliability, particularly for high-suction-energy pumps. A minimum NPSH margin of 1 m is an industry rule of thumb. However, since NPSHR increases with flow, it is important to consider the maximum expected flow as well as the rated and normal flow when specifying the NPSH margin.

NPSHA, the suction head available to the pump, can be easily calculated based on the piping configuration and process fluid levels in upstream vessels. NPSHA for the equipment shown in Figure 7 is calculated as:

$$NPSHA = (P - P_v)/\rho g + L_h - f_L \qquad (10)$$

where P is the absolute pressure on the surface of the fluid, L_h is the static suction, and f_L is suction pipe friction loss.

If the fluid is in a tank open to the atmosphere, P is the absolute barometric pressure. The sign of L_h depends on whether the fluid is located above or below the centerline of the impeller: Fluid above the centerline produces a positive head, while fluid below the centerline imposes a negative lift. As stated earlier, there should be a margin of at least 1 m between NPSHA and NPSHR to account for variability in process conditions as well as increased losses due to equipment aging.

Suction specific speed

NPSHR is equal to the inlet velocity head:

$$NPSHR = V_1^2/2g \qquad (11)$$

At the pump inlet, the velocity is a function of the volumetric flowrate and inlet area:

$$V_1 = Q/A_1 \qquad (12)$$

Suction specific speed quantifies the suction capability of the pump as it relates to NPSHR:

$$S = NQ^{0.5}/NPSHR^{0.75} \qquad (13)$$

A high value of S corresponds to a low NPSHR. It would therefore seem reasonable to assume that a higher suction specific speed is better. However, Eqs. 12 and 13 show that the easiest way to reduce the NPSHR for a particular flow is to increase the inlet area. This is not always a good practice, though, because it increases the minimum flowrate for the pump (which will be discussed later). Common practice is to keep S between 7,000 and 12,000, depending on the fluid. When a pump cannot be found that meets the suction specific speed criteria for a particular project, a recirculation line can be added to meet the minimum flow requirements.

Minimum flow

As the flowrate drops below that of the BEP, the NPSH required by the pump initially decreases until it reaches a minimum before beginning a steady increase (Figure 1). As flow decreases, the tendency of the fluid to fall back on itself, or recirculate, increases. At the BEP flowrate, the fluid moves smoothly through the impeller passage. As the flow decreases, there is not enough fluid to fill in the volume between the impeller vanes, the fluid accumulates against the pressure side of the vane, and recirculation occurs in the suction side (Figure 8). The horseshoe-shaped recirculation vortices erode the impeller at both the discharge and inlet. Pumps with high suction specific speed will have correspondingly high minimum flow requirements that will likely require a recirculation line to provide satisfactory operation.

There is an important difference between the minimum stable flow just described and the minimum thermal flow. API 610 *(1)* defines them as:

- *minimum continuous stable flow (MCSF)* — the lowest flowrate at which the pump can operate without exceeding the vibration limits imposed by API 610
- *minimum continuous thermal flow (MCTF)* — the lowest flowrate at which the pump can operate without being impaired by the temperature increase of the pumped liquid.

MCSF relates to recirculation of the fluid that can result in cavitation and vibration, whereas MCTF is concerned with temperature rise. A pump efficiency curve (Figure 1) shows that efficiency drops as flow decreases below the BEP. This decrease in efficiency is characterized by an increase in temperature. MCTF is the point at which this rate of temperature increase hinders the operation of the pump. The minimum operating flow is the higher of MCTF and MCSF. Generally, MCSF occurs at a higher flowrate than MCTF and becomes the defining variable. However, for very-low-flow pumps, MCTF may dominate.

Example 1: Ethylene product pump

To illustrate how process conditions affect pump selection, consider an ethylene product pump under the following process conditions:

- rated flow = 400 m^3/h
- rated suction pressure = 1,600 kPag
- discharge pressure = 5,700 kPag
- differential pressure = 4,100 kPa
- specific gravity = 0.46
- head = 4,100 kPa ÷ 460 kg/m^3 ÷ 9.81 m/s^2 × 1,000 = 909 m
- NPSHA = 7.2 m

First, determine the number of stages required, assuming that each stage produces the same head and that a specific speed of $N_s \approx 1{,}100$ ($n_q \approx 21.3$) is desired. Assume a shaft speed of 3,600 rpm and calculate the head per stage (H_{stg}) from Eqs. 6–8:

$$H_{stg} = \{3{,}600 \text{ rpm} \times (400/3{,}600 \text{ m}^3/\text{s})^{0.5} \div (1{,}100/51.64)\}^{4/3} = 216 \text{ m/stage}$$

Divide the total head required, 909 m, by 216 m/stage and round up to the next-highest integer to get 5 stages

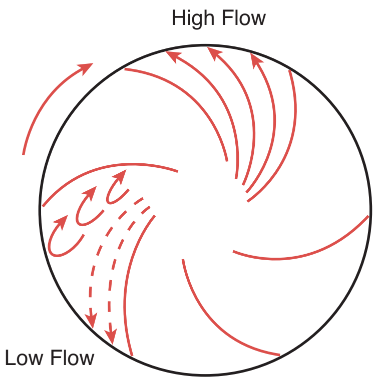


Figure 8. Flows moving too slowly tend to fall back on themselves, while higher flows generally exhibit a smoother pattern and increased efficiency.

(each producing 181.8 m/stage). The specific speed is then:

$$N_s = 51.64 \times 3{,}600 \text{ rpm} \times (400/3{,}600 \text{ m}^3/\text{s})^{0.5} \div (181.8 \text{ m})^{0.75} = 1{,}252$$

which is sufficiently close to the desired value of 1,100.

The size of the impellers can be approximated based on the impeller tip speed:

$$H \approx (r\Omega)^2/2g$$
$$d \approx 2(H \times 2g)^{0.5}/\Omega$$
$$= 2(181.8 \text{ m} \times 2 \times 9.81 \text{ m/s}^2)^{0.5} \div (3{,}600 \times 2\pi \div 60)\text{rad/s}$$
$$= 317 \text{ mm}$$

Since an NPSH margin of at least 1 m is desired, assume NPSHR = 6.2 m. Use Eq. 13 to calculate the suction specific speed (recall that the suction specific speed concerns conditions at the inlet while the specific speed pertains to the overall performance of the pump):

$$S = 51.64 \times 3{,}600 \text{ rpm} \times (400/3{,}600 \text{ m}^3/\text{s})^{0.5} \div (6.2 \text{ m})^{0.75} = 15{,}772$$

This is too high.

The suction specific speed for a double-suction style pump is calculated based on one-half of the total flow:

$$S = 51.64 \times 3{,}600 \text{ rpm} \times (200/3{,}600 \text{ m}^3/\text{s})^{0.5} \div (6.2 \text{ m})^{0.75} = 11{,}152$$

This is acceptable.

Finally, a pump supplier's online program identifies a pump with the following specifications as being suitable for this application:

- number of stages = 5
- double suction? Yes
- rated impeller diameter = 318 mm
- NPSHR = 5.2 m
- suction specific speed = 11,657

- efficiency = 79.2%
- minimum flowrate = 193.7 m^3/h
- shutoff head = 997.2 m

Example 2: Cooling water pump

Cooling water tends to be a high-flow and low/medium-head application. This example has the following conditions:

- rated flow = 2,000 m^3/h
- rated suction pressure = –10 kPag
- discharge pressure = 410 kPag
- differential pressure = 420 kPa
- specific gravity = 1.00
- head = 420 kPa ÷ 1,000 kg/m^3 ÷ 9.81 m/s^2 × 1,000 = 43 m
- NPSHA = 12 m

Here, a specific speed of $N_s \approx 1{,}500$ ($n_q \approx 29$) is desired. Assume a shaft speed of 1,200 rpm and calculate the head per stage from Eqs. 6–8:

$$H_{stg} = \{1{,}200 \text{ rpm} \times (2{,}000/3{,}600 \text{ m}^3/\text{s})^{0.5} \div (1{,}500/51.64)\}^{4/3} = 97 \text{ m/stage}$$

Since the required head is 43 m, only one stage is needed for this application. A speed of 600 rpm could be used if desired, but that would require a larger pump. With a 1,200-rpm pump, the specific speed is:

$$N_s = 51.64 \times 1{,}200 \text{ rpm} \times (2{,}000/3{,}600 \text{ m}^3/\text{s})^{0.5} \div (43 \text{ m})^{0.75} = 2{,}751$$

According to Figure 5, this requires a Francis vane impeller, which has characteristics between those of a radial-vane and a mixed-flow design.

The size of the impellers can be approximated by:

$$\text{H} \approx (r\Omega)^2/(2g)$$
$$\text{d} \approx 2(H \times 2g)^{0.5}/\Omega$$
$$= 2 \times (43 \text{ m} \times 2 \times 9.81 \text{ m/s}^2)^{0.5} \div (1{,}200 \times 2\pi \div 60) \text{ rad/s}$$
$$= 462 \text{ mm}$$

Assume NPSHR = 11 m and use Eq. 13 to calculate the suction specific speed:

$$S = 51.64 \times 1{,}200 \text{ rpm} \times (2{,}000/3{,}600 \text{ m}^3/\text{s})^{0.5} \div 11^{0.75} = 7{,}647$$

The pump manufacturer's online program selects a pump with the following specifications:

- number of stages = 1
- double suction? No
- rated impeller diameter = 496 mm
- NPSHR = 7.0 m
- suction specific speed = 7,960
- efficiency = 89.6%
- minimum flow = 779 m^3/h
- shutoff head = 59.7 m

Closing thoughts

Close coordination between the process engineer and the rotating-equipment engineer is essential throughout the design and specification phases of a project. By understanding the concepts discussed here — flowrate, head, suction pressure, and NPSH — and how these parameters affect the impeller selection (specific speed), minimum flow requirements and required NPSH, you will be well-equipped to take on the task of selecting a pump. CEP

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J. HOWARD KELLY, P.E., is a rotating-equipment engineer with The Shaw Group's energy and chemicals division (E-mail: howard.kelly@shawgrp.com; Website: www.shawgrp.com) and a registered professional engineer in Texas. For his thesis project, he wrote a program using transfer matrices to analyze the dynamic and static characteristics of multistage centrifugal pumps. He has served as a missionary in Thailand and worked as an engineer in the aerospace, semiconductor, and energy and chemicals industries over the past 20+ years. He is currently the onsite rotating-equipment engineer supporting the construction of a new ethylene plant in Singapore. He holds a BS in aerospace engineering and an MS in mechanical engineering, both from Texas A&M Univ.