

Estimating Maximum Head in Single – and Multi-Stage Pump Systems

BY JAMES NETZEL

The maximum head or discharge pressure of a centrifugal pump can be easily estimated if the impeller diameter, number of impellers used, and rpm of the driver are known.

Q: How can you estimate the maximum (shutoff) head that a centrifugal pump can deliver?

A: The maximum pressure a centrifugal pump delivers should be known in order to ensure that a piping system is adequately designed. Any pump that operates at a high flow rate could deliver significantly more pressure at zero (0) gpm flow, such as when the discharge valve is closed, than it delivers at operating flow.

The maximum head or discharge pressure of a centrifugal pump, which usually occurs at shutoff con-

ditions (0 gpm), can be easily estimated if the impeller diameter, number of impellers used, and rpm of the driver (electric motor, gas engine, turbine, etc.) are known.

Let's say we have a single-stage pump with a 10-in. diameter impeller and an 1,800 rpm driver. To determine the head in feet, simply take the impeller diameter in inches and square it. Our 10-in. impeller at 1,800 rpm would yield 10^2 , or 100 ft of head. An 8-in. impeller would yield 8^2 , or 64 ft of head, while a 12-in. impeller would yield 12^2 , or 144 ft of head.

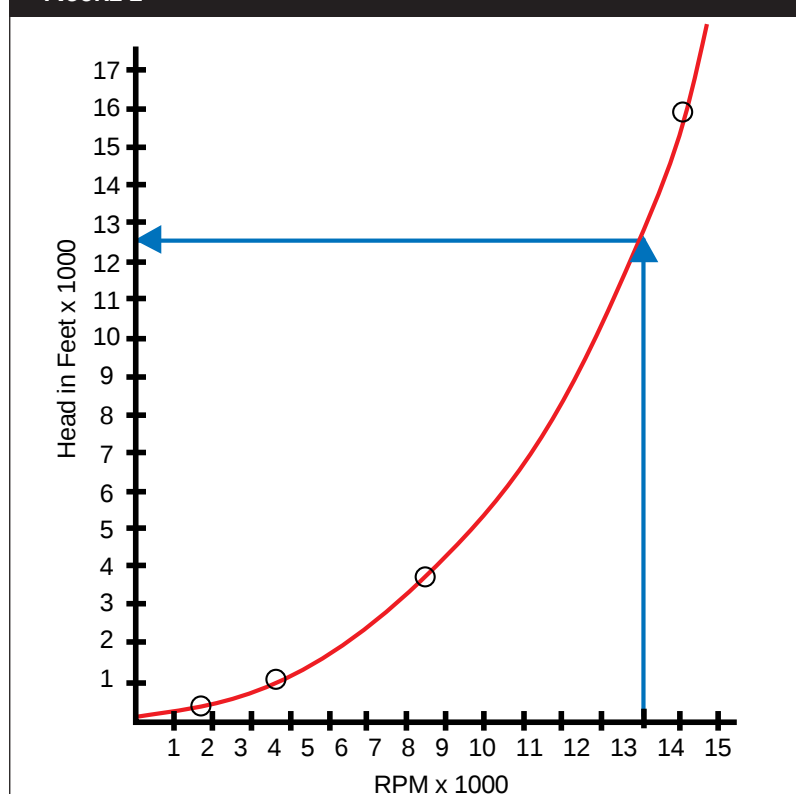
Now let's assume that our 10-in. diameter impeller is driven by a 3,600 rpm motor. We first determine the head at 1,800 rpm, but then multiply this value by a factor of four. The basic rule is that every time the rpm changes by a factor of two, the head changes by a factor of four. The head at 3,600 rpm for our 10-in. impeller is therefore $10^2 \times 4$, or 400 ft of head. Our 8-in. impeller at 3,600 rpm would give us $8^2 \times 4$, or 256 ft of head, and our 12-in. impeller would give us $12^2 \times 4$, or 576 ft of head.

For multiple stages (more than one impeller), simply multiply the final head for one impeller by the total number of impellers in the pump. For a pump with three 10-in. impellers and a speed of 3,600 rpm, we get $(10^2 \times 4) \times 3 = 400 \times 3 = 1,200$ ft. of head.

Now what happens if we reduce the speed below 1,800 rpm? The same rule still applies: a change in speed by a factor of two changes the head by a factor of four. Therefore, a 10-in. diameter impeller spinning at 900 rpm delivers only one fourth the head it would at 1,800 rpm: $10^2/4 = 25$ ft.

Plotting several head-versus-rpm points on a curve will allow the user to estimate the maximum

FIGURE 1



Rotations per minute (rpm) vs. head in feet to estimate maximum head

head at any given speed. Let's say we have a turbine-driven pump that injects water into the ground to raise the subterranean oil reserves to the surface for processing. The vendor tells you that the maximum head is classified, but you have been requested to resolve system problems that you believe are pressure related. The vendor tells you that the pump has four 8-in. diameter impellers and is driven by the turbine at 13,000 rpm. You would estimate the maximum head as follows:

Step 1 Determine the head at 1,800 rpm:

$$8^2 \times 4 \text{ stages} = 256 \text{ ft}$$

Step 2 Multiply the head at 1,800 rpm by four to get the head at 3,600 rpm:

$$256 \times 4 = 1,024 \text{ ft}$$

Step 3 Multiply the head at 3,600 rpm by 4 to get the head at 7,200 rpm:

$$1,024 \times 4 = 4,096 \text{ ft}$$

Step 4 Multiply the head at 7,200 rpm by 4 to get the head at 14,400 rpm:

$$4,096 \times 4 = 16,384 \text{ ft}$$

Step 5 Plot the rpm-versus-head points to obtain the curve shown in Figure 1.

As you can see, the estimated head at 13,000 rpm is 12,500 ft. To convert head in feet to psi, simply divide the head by 2.31 to get 5,411 psi.

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Q: What different types of seal lubrication exist?

A: A mechanical seal is designed to operate in many types of fluids. The product sealed becomes the lubricant for the seal faces. Many times the fluid being sealed is a poor lubricant or contains abrasives that must be taken into account in the seal design. The design of the seal faces, materials of construction, and seal lubrication play an important role in successful operation. Achieving a high level of reliability and service life is a classic problem in the field of tribology, the study of friction, wear, and lubrication.

The lubrication system for two sliding seal faces can be classified as follows: 1) hydrodynamic, 2) elastohydrodynamic, 3) boundary, and 4) mixed film.

Hydrodynamic conditions exist when the fluid film completely separates the seal faces. Direct surface contact between seal faces does not take place, so there is no wear, and heat generation from friction is zero. The only heat generation occurs from shearing of the fluid film, which is extremely small. A hydrodynamic seal may rely on design features such as balance factors, surface waviness, or spiral grooves to separate the seal faces. The Society of Tribologists and Lubrication Engineers (STLE) guideline in "Meeting Emissions Regulations with Mechanical Seals" lists hydrodynamic seals as a technology to control emissions.

Elastohydrodynamic lubrication (EHD) is also found in sliding surfaces, but more often this involves rolling surfaces separated by an oil film. Here the moving surfaces form

an interface region that deforms elastically under contact pressure. This deformation creates larger film areas and very thin films. Such lubrication systems are normally used to control wear in rolling element bearings. In seals where the viscosity of the fluid sealed increases with increasing pressure, elastohydrodynamic lubrication occurs.

Boundary lubrication is important for seal faces that are moving very slowly under heavy load. Here, hydrodynamic and elastohydrodynamic lubricant pressures are insufficient to separate the seal faces. The sliding surfaces are protected by the tribological properties of the materials of construction. An example of a seal operating within this lubrication system is a dry-running agitator seal.

Mixed-film lubrication, a combination of all the previous systems discussed, occurs in all contact seals. Here the fluid film becomes very thin and is a combination of both the liquid and the gas phases of the fluid sealed. Asperities from one surface may penetrate the lubricating film and contact the opposite surface. The seal face load is then supported partially by the fluid film and partially by solid contact. If the generated head at the seal faces is not removed, surface wear and damage can occur. For applications where the seal face load is too high or the fluid viscosity is too low, designs of seal faces can be changed through balance and face geometry to improve seal performance. ■

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