FEATURE

High Speed Design Enhances Pump Performance

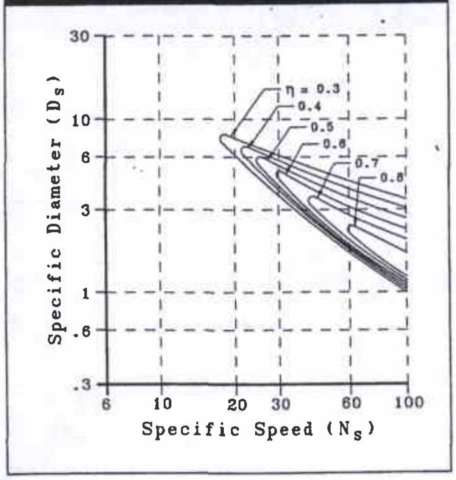
These designs can improve performance and provide mechanical advantages over conventional pump designs.

Many pumping applications could benefit from speeds higher than the usual 3600 rpm limit of centrifugal pumps. For example, a high speed pump can produce a high head at a low flow rate more efficiently than a conventional pump can. Furthermore, a high speed pump can be smaller, lighter and in some cases less expensive than conventional pumps.

Centrifugal pumps have historically been limited to 3600 rpm for two reasons. First, the most common pump drive is a two-pole, 60 hz motor which runs at about that speed. Higher speeds can be obtained with a gearbox but this adds to the cost and complexity of the pump. The second and more fundamental limitation is pump reliability and maintenance under high speed operation. Rotating shaft seals are of special concern.

Economical high speed drives (e.g, turbines and high speed motors powered by variable frequency drives) and sealless pump designs address these problems and are thus bringing high speed pumps into more extensive use. Another interesting design twist, the partial emission pump, is often advantageous in high speed applications. This unconventional design can achieve high efficiency for high head, low flow operations. In some applications where the performance of a conventional, full emission centrifugal pump would be unacceptably poor, a partial emission pump can be used in lieu of a fixed displacement pump.

FIGURE 1



Efficiency contours for a full emission pump

ADVANTAGES OF HIGH SPEED PUMPS

One advantage of high speed pumps is superior efficiency. The relationship between pump efficiency, speed, head and flow is best discussed in terms of non-dimensional groupings of pump variables called similarity parameters.

The Buckingham Pi Theorem describes four similarity parameters — specific speed, specific diameter, Reynolds number, and suction specific speed — useful in characterizing the performance of a pump. Since suction specific speed is only important in cavitating operations and Reynolds number has only a secondary effect, the performance of a pump can be fairly well correlated to specific speed and specific diameter.

Specific Speed $N_s = N * Q\wedge.5 / H\wedge.75$
Specific Diameter $D_s = D * H\wedge.25 / Q\wedge.5$

where: N = Impeller Speed (rpm)
D = Impeller Diameter (ft)
Q = Volume Flow Rate (ft^3/s)
H = Isentropic Head (ft)

These parameters can be made non-dimensional by selecting appropriate units. However, traditionally the units shown above are used and the parameters are not truly non-dimensional.

Using the relationship of efficiency to specific diameter and specific speed, one can determine the best possible efficiency for a pump (see Figure 1 for efficiency

BY MICHAEL D. FORSHA AND CHRISTINE A. BEALE, BARBER-NICHOLS INC.

[redacted]

contours for single-stage pumps with an optimum efficiency design). Here are three examples read from the figure:

	CASE 1	CASE 2	CASE 3
Pump Head (ft)	40	130	130
Flow Rate (gpm)	30	48	48
Flow Rate (ft³/s)	.067	.107	.107
Pump Speed (rpm)	3600	3600	7600
Impeller Diameter (in)	2.75	5.25	2.65
Impeller Diameter (ft)	.23	.44	.22
Specific Speed	58.6	30.6	64.6
Specific Diameter	2.23	4.54	2.27
Best Possible Efficiency (%)	80	60	81

The flow requirements for case 1 were selected so that a 3600 rpm pump could still achieve high efficiency. With the specified head, flow rate and pump speed, the specific speed is 58.6. According to Figure 1, the best possible efficiency is 80% for this specific speed and for a specific diameter of about 2.23 (equivalent to an impeller diameter of 2.75 inches).

Case 2 represents flow requirements for a liquid helium circulating pump for a superconducting magnet string. According to Figure 1, the best possible efficiency is 60% if this pump is limited to 3600 rpm. (The limitation is the lower specific speed which is a result of the higher head requirement.)

In case 3 the pump has the same head and flow requirements as for case 2, but the speed has been increased to 7600 rpm. This example shows that by increasing the speed, the specific speed can be increased and an efficiency of 81% can be achieved. High efficiency is critical in this application since refrigeration at liquid helium temperatures is expensive and the energy transferred into the liquid helium must be minimized.

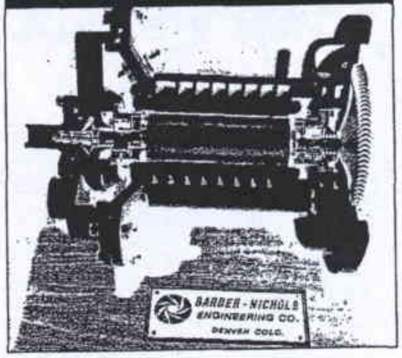
High speed pumps offer superior efficiency.

In addition to improved efficiency, high speed pumps can also offer reduced size, weight and cost. For example, notice that the impeller diameter in case 3 is half that for case 2. This size reduction carries through to other parts of the pump assembly as well. For example, a 7600 rpm motor can be rated for nearly twice the power as the same size 3600 rpm motor. The reduced size of the pump components results in size and weight savings for the pump assembly.

A more dramatic example of the size benefits of a high speed pump is a toluene pump functioning as the feed pump for a solar driven Rankine cycle power system. (The pump is shown on the left side of the turbine-alternator-pump assembly in Figure 2.) In this application, minimal size and weight of the entire power system is critical. The system is part of a lightweight structure mounted on the solar collector dish and moves as the dish tracks the sun.

The toluene pump is driven by the turbine at 60,000 rpm. At this speed the required 2000 feet of head and 5 gpm flow rate can be obtained with a single-stage impeller 1.4 inches in diameter. These flow conditions could not be achieved with a single-stage 3600 rpm impeller because of the low specific speed. A 20 stage centrifugal pump with 6 inch diameter impellers would be required if speeds were limited to 3600 rpm. The size, weight and cost savings of high speed pumping are obvious.

FIGURE 2



A toluene pumping system for a solar electric generating system

HIGH SPEED PUMP DRIVES

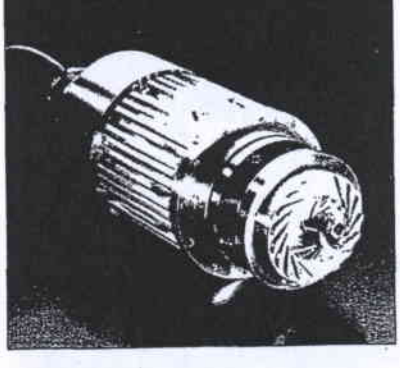
Until recently, most pumps that operated above 3600 rpm were driven with a motor and gearbox. While many pumps still use a gearbox drive, this approach compromises size, weight and cost of the pump assembly. Maintenance and reliability questions are introduced with the additional bearings, seals and gear sets.

To eliminate these problems, designers are turning to direct high speed motor drives. With a standard 400 hz. Variable Frequency Drive (VFD), a two-pole motor can produce synchronous speeds up to 24,000 rpm.

High costs have kept VFDs from extensive use in the past. However, recent advances in solid state devices have reduced costs significantly. Current representative costs range from $600 to $1400 for VFDs that can drive .75 hp to 15 hp motors. VFDs still aren't inexpensive but they are cost effective in certain applications.

A synchronous motor speed of 7600 rpm was achieved with a two-pole motor driven at 126 hz with a VFD in a liquid helium pump (Figure 3).

FIGURE 3



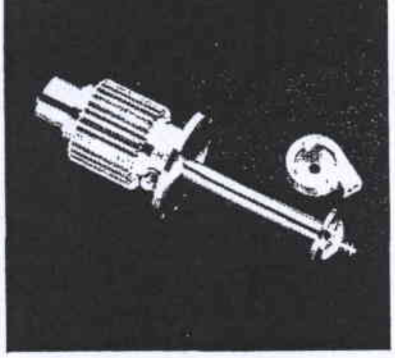
A liquid helium pump

The turbine drive is another high speed drive with several advantages. Turbine drives have no inherent speed limits and are reliable components well suited for either intermittent or continuous operation. Since the turbine is driven with thermal rather than electrical power, the operating costs can be significantly less than those for a motor.

If facility steam is available, a steam turbine can be used. Inexpensive back-pressure steam turbines are readily available and require a minimum of peripheral equipment.

Turbine drives can also be used in more sophisticated arrangements. If a pressure drop exists across a throttle valve in the process stream, it may be possible to replace the throttle valve with a turbine used to drive the pump. The turbine can be either a Pelton wheel design (if the pressure drop is in a liquid stream) or an expansion design (if the stream is a vapor).

FIGURE 4



A liquid nitrogen pump

This approach can provide two main benefits. It reduces operating costs since the drive power is obtained regeneratively from the process stream. Also, in many applications it is desirable to drop the pressure in an isentropic process. In these situations an expansion across a turbine is better than an isenthalpic expansion across a throttle valve.

The toluene pump used in the solar system is driven by the toluene vapor expansion turbine (on the right side of the assembly in Figure 2). The turbine drive was the only reasonable means of achieving the 60,000 rpm speed which permitted the use of a small, single-stage feed pump.

OVERCOMING MAINTENANCE AND RELIABILITY PROBLEMS

High speed pumps will not gain widespread acceptance until they can overcome the perception that high speed decreases reliability and increases maintenance requirements. This perception is not accurate. With proper designs, high speed pumps can be highly reliable and require only low maintenance.

In most centrifugal pumps, the only major components that experience wear are bearings and seals. Bearings are not a significant problem in high speed pumps. High speed bearing technology is well developed, and either a hydrodynamic bearing or rolling element bearing can generally be selected. Both of these choices will provide long life and high reliability.

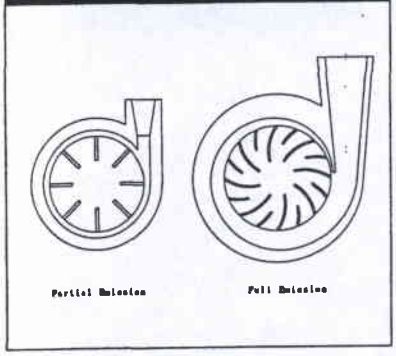
Shaft seals are more problematic. The rotating shaft seal is probably the most significant limitation to achieving high reliability and low maintenance in high speed pumps. Most pumps use either a packing gland or a mechanical seal between the pump and the motor. These seals have rubbing surfaces and their life and reliability decrease as the pump speed increases. Since no seal design can guarantee high reliability and long life at high speed, it is best to use a sealless pump design which completely eliminates the shaft seal.

There are two major challenges in designing a sealless pump. First, since the motor and all interior surfaces of the housing are exposed to the pumped fluid, materials of construction must be compatible with the fluid. Second, the bearing system must operate in the pumped fluid. Thus, conventional oil lubricated bearings can not be used. Yet sealless designs have been successfully incorporated in high speed pump applications.

A liquid nitrogen pump (Figure 4) is one example. The pump impeller is mounted on a shaft that extends from the motor and the entire rotating assembly is contained in a common housing. There are no seals between the impeller and the motor. Since nitrogen is compatible with the stainless steel pump parts and the iron and epoxy motor parts, these pumps require no shielding from the pumped fluid.

An unusual bearing configuration is necessary because of the large range of temperatures to which the assembly is exposed. The long shaft between the motor and impeller minimizes heat transfer from the motor (which operates at about room temperature) to the liquid nitrogen at the impeller end of the assembly (which is at -320° F). The nitrogen in the motor end of the assembly is a vapor, and since nitrogen vapor provides no lubricity or cooling for bearings, grease-packed ball bearings are used at the motor. The impeller end of the assembly operates in liquid nitrogen, which provides reasonable lubricity and excellent cooling. Thus, a liquid nitrogen lubricated ball bearing is used at the impeller.

FIGURE 5



Partial emission and full emission pumps

By eliminating the shaft seal and incorporating this novel bearing configuration, the pump can provide continuous service with a scheduled maintenance interval of more than one year. Scheduled maintenance requires less than an hour and consists of regreasing the motor bearings and replacing the impeller bearing.

For most noncryogenic sealless pumps, grease-packed bearings can not be used since the pumped fluid will degrease the bearing. The toluene turbine-alternator-pump (Figure 2) uses a different design approach. Like the liquid nitrogen pump, this unit uses no shaft seals. However, instead of rolling element bearings, the rotating assembly is supported with fluid film bearings that use the pumped fluid, toluene, for lubrication.

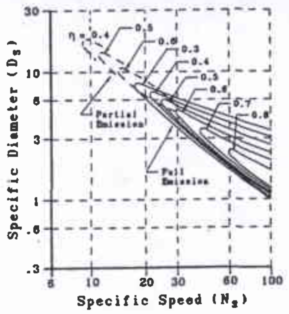
Except for start-up and shut-down the bearings do not rub, so there is no wear. By eliminating the shaft seal and using fluid film bearings, the unit achieves very high reliability and a nearly unlimited life.

A similar design can also be used for a motor driven, sealless, high speed pump. The motor driven pump would be similar to the turbine-alternator-pump assembly if the turbine was removed and the alternator was replaced with an electric motor. If the pumped fluid is compatible with motor materials, no other design modifications are required. If the pumped fluid is not compatible with the motor, the motor must be shielded. Either a canned motor or a magnetic coupling can be used. Both designs eliminate the shaft seal and can use bearings lubricated with the pumped fluid. The result: high reliability and low maintenance in a high speed pump.

HIGH SPEED, PARTIAL EMISSION PUMPS

A design that extends the operating envelope of centrifugal pumps to handle even higher heads and lower flow rates is the partial emission design. This unconventional design was developed during World War II to handle the high head, low flow rate requirements of the German ram jet fuel pumps. The differences between a partial emission pump and a conventional full emission pump are in the impeller and diffuser flow passages (Figure 5). The partial emission design uses straight radial blades on the impeller and has a diffuser which only allows flow from a small sector of the impeller channels to pass to the pump discharge at any time. A full emission pump uses backward curved blades on the impeller and has a collector scroll which allows flow from all of the impeller channels to continually pass to the pump discharge. The toluene pump (Figure 2) and the liquid nitrogen pump (Figure 4) used a partial emission design; the liquid helium pump (Figure 3) used a full emission design.

FIGURE 6



Efficiency contours for partial emission and full emission pumps

Efficiency contours for a partial emission pump and a conventional full emission pump (Figure 6) show that for specific speeds below about 30, a partial emission pump is more efficient than a full emission pump. This is significant because in some applications it is not possible to operate at pump speeds that are high enough to achieve a high efficiency specific speed for a full emission pump. In these cases it may be possible to use a partial emission design that can still achieve high efficiency at a lower specific speed.

Partial emission pumps not only offer improved efficiency at low specific speed, but also can be used at specific speeds where a full emission centrifugal pumps would not make sense (e.g., applications with specific speeds less than about 20). For low specific speeds associated with high head/low flow rates, a constant displacement pump (such as a reciprocating piston pump or a rotating gear pump) would usually be specified. But the life and reliability of these constant displacement pumps are inherently limited by their close fitting components and rubbing surfaces. A partial emission centrifugal pump, which operates well at specific speeds as low as eight, brings the advantages of centrifugal pumps (namely generous clearances and no rubbing parts) to such applications.

High speed, sealless, and partial emission pump designs offer advantages to many applications — not just the demanding situations described in this article. The informed pump user can often achieve greater efficiency and reduced size, weight and cost by choosing high speed pumps. ■

Michael Forsha is engineering manager at Barber-Nichols Inc. He has more than 10 years experience in turbo-machinery design.

Christine Beale is an applications engineer at Barber-Nichols Inc. She has more than 6 years experience in project engineering and sales/marketing.