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Vanquish Vertical Pump Vexations

Address common problems including undersized motor drivers

By Amin Almasi, rotating equipment consultant

Vertical pumps suit a diverse range of industrial applications for services ranging from low to high pressure, say, from 0.3 barg to 700 barg, and cryogenic to high temperature, e.g., -190°C to +500°C. Moreover, they cover a wide range of flow rates, roughly from 6 to 60,000 m³/h. The pumps contain from 1 to 50 stages and run at speeds from 180 to 20,000 rpm. They come in numerous varieties.

A vertical wet-pit pump can handle applications involving clean or lightly contaminated liquids. The drive motor is mounted directly above the pumping head, which is immersed in the liquid. Such a pump has a common discharge and support column. It has a vertical line shaft with the slide bearings typically lubricated by the pumped liquid.

A vertical sump pump is used primarily in services with lightly contaminated liquids, slurries and liquids containing solids. It also has a motor mounted directly on top of the pumping head, which is suspended in the pumped liquid. Such a pump has a separate discharge pipe and support pipe column. It has a vertical line shaft with slide bearings or cantilever design without slide bearings.

Submerged pumps are the other major category of vertical pumps but there are other types and categories as well.

Bearings and the lubrication system significantly impact the pump's reliability, maintenance and long-term operation. Indeed, bearings and their lubrication usually account for most problems. So, specify all bearings according to API-610 whether

the pump is an API type or not. Likewise, insist upon a sophisticated lubrication oil system as per API-610 or via a robust specification (preferably with two lubrication pumps, dual filters, etc.) for any large vertical pump.

Thrust bearings are critical components. They should be designed to carry the weight of all rotating parts plus the hydraulic thrust (both steady state and dynamic forces) and should be readily accessible for inspection. The hydraulic design of the

pump should ensure that residual loads on the thrust bearings are always downwards.

Because vertical pumps commonly are direct driven, couplings should have provisions to secure the hub to the shaft with minimal dismantling.

VERTICAL PUMP DYNAMICS

A vertical pump, particularly a suspended unit, usually is a flexible system that runs at a speed located between or around natural frequencies. This makes it susceptible to

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resonant vibration if its separation margins and dynamic situation aren't verified during design and installation. Preferably maintain a 10–15% margin of separation between any natural frequency of the support structure and operating speed (and its first few harmonics).

In case of high vibration, it may be possible to rebalance the rotor assembly on site. However, in the case of resonance, this usually won't markedly reduce the vibration level. Furthermore, the balancing planes probably aren't easily accessible on site and rebalancing has some limits. The unbalance condition somewhat depends upon temperature and other operating conditions. So, expect increased vibration levels during transient conditions, due to fouling of a pump impeller or for other reasons. Also, any changes to the pump, such as maintenance work on the electric motor, coupling, pump rotor, etc., can affect vibration level.

Basic structural elements typically include the foundation, pump structure and electric motor frame. Strive to use accurate foundation data in a dynamic analysis. However, the deflection of the foundation typically represents less than 5–8% of the total deflection of the structural elements. Therefore, if accurate foundation data aren't available, use estimated or rough data.

Because of its flexible structure, a large vertical pump can get excited by random

broadband turbulences. However, vibration levels due to turbulent excitation usually pose no major concerns.

Large vertical pumps located side by side can be excited by the foundation structure to some extent. Some cross-couplings of vibrations between pumps can occur via the foundation or soil; measurements can detect these. If manufacturing and construction tolerances are controlled, identical pumps will exhibit only minor difference in vibration characteristics. In other words, natural frequencies usually differ by approximately 0.5–1.5 Hz. Slightly different speeds and natural frequencies of identical pumps occasionally result in moderate beat phenomena.

Operation may change the natural frequency of a vertical pump system. For instance, liquid level can alter the natural frequency slightly. However, such a variation usually is small. As a very rough indication, you can expect 0.05–0.4-Hz higher bending frequency at low liquid level relative to high liquid level. Also, the natural frequency often increases when a vertical pump is stopped (because, for example, the pump column isn't full of liquid). Such a change in the bending natural frequency might be on the order of 0.1–0.6 Hz.

A vertical pump, particularly one with a long shaft, has relatively large inertia in the driver and pump stages and is susceptible to

torsional excitations. Therefore, it requires a complete torsional study.

Any critical vertical pump package demands careful condition monitoring. So, specify electric motor winding temperature sensors, an X-Y vibration probe for each bearing as well as an axial probe and key phasor for each shaft. In addition, insist upon an accelerometer at the pump casing to monitor the casing vibration — particularly for a large vertical pump with flexible structure. Ensure the number and location of vibration sensors will suffice for machinery diagnostics.

PARALLEL OPERATION

Plants often operate vertical pumps in parallel. For such operation, the “head versus flow” curves of the pumps ideally should match within 2–3% between the minimum continuous flow and the end of curve flow. Always check that all combinations of parallel running are efficient for long-term operation without any performance or reliability issues.

Parallel operation of pumps can be risky; indeed, it often causes problems. Use of dissimilar pumps poses a particularly challenging situation. In such cases, you should match the head at the rated point of each pump (not at the same volumetric-flow-rate points). The surrounding area of rated points should agree within certain limits (say, 3–4%). Check with the pump manufacturers about whether the expected

combinations of parallel operation raise any issues. In addition, plot the operating points for all scenarios and compare them between the minimum continuous flow and the end of curve.

When two pumps are running in parallel and providing a total flow greater than either pump could produce individually, if one pump trips off for any reason it's extremely important that the other pump doesn't trip off on overload. Indeed, such a requirement is common and well established for many critical applications. This mandates the sizing of the electric motor drivers for the end of pump curve. If the electric motors weren't sized for the end of curve, then the remaining pump would overload and trip off, ending all liquid flow. In the case of critical pump services, this would result in a disastrous and costly shutdown.

There's also another economic driver for sizing the electric motors for the end of the curve — maintenance. When a pump requires service, it's desirable to maximize the flow from the remaining pump even if this puts that pump at a flow rate outside its preferred operating range. The plant may need to run the pump at that rate for a relatively short time to capture a high unit incentive or get through an emergency situation.

Not all services require sizing the electric motor driver for the end of curve. So, assess

this case by case. Many process pumping services and some critical utility applications do demand sizing the driver for end-of-curve operation (with ample margins), though. A design team may decide to deviate slightly from this requirement based on the results of thorough studies and simulations of many different operating cases. However, I strongly recommend sizing the electric motor based on the end of curve. Because such sizing usually must be done in the bidding stage, you either should follow that guideline or conduct a full study then. Finally, it's better to slightly oversize the pump driver than slightly undersize it.

Some pump manufacturers now don't show the pump performance curve beyond 120% or 125% of the rated flow; I've recently seen such truncated curves from four vendors in different services. The main purpose is to avoid sizing drivers based on the above rule. In such cases, a major question is "where is the end of curve?" The curve usually should end at the point where the efficiency drops. The end of curve generally isn't a point where the pump can operate continuously; therefore, for many pumps, the point of "maximum continuous flow" can't be the real end of the curve.

Usually, a pump's head-versus-capacity curve ends on the far-right side of its best efficiency point (BEP), say, roughly at 135%, 140% or 150% of BEP flow, where the pump shows relatively low efficiency or starts

to cavitate or net positive suction head becomes a limiting factor. Obviously, points with efficiencies close to that at BEP or slightly lower can't be by any measure the end of the curve.

Unfortunately, the consequence is that many vertical pumps have undersized electric motor drivers. Some vendors resort to any trick possible — cutting the curves, misusing and misinterpreting API guidelines or project specifications, etc. — to employ the smallest possible (and definitely undersized) driver.

Based on my experience, you can address this issue and get proper electric motor sizing in one of two ways:

1. Forcefully ask the vendor to extend the curve to show the real end and then insist upon sizing the driver based on the end of curve.
2. If a vendor won't agree to option 1, define the point of 120% (or 125%) BEP flow as one of rated points. This automatically results in motor sizing for the power required in this newly defined rated point using all project and code margins (for example, 10% API margin).

SOME EXAMPLES

Let's now look at two cases — both involving multiple wet-pit vertically suspended diffuser-type pumps.

Seawater pumps. In our first case, two vertical pumps were operated in parallel to

provide the 11,000-m³/h seawater required by a chemical complex. Each had a pump capacity of 5,500 m³/h and a discharge pressure of 3.5 barg, and used a direct-drive vertical electrical motor to provide a speed of 750 rpm. Each is a single-casing diffuser pump with discharge through the column. The discharge nozzle of each pump is 36 in. with an ASME B16.5 150-lb flange. The overall vertical pump train exceeds 12 m.

The BEP flow of each pump is around 6,300 m³/h; so, the rated-point flow is about 87% of the BEP flow. The end of the curve occurs at a flow of about 8,200 m³/h and a discharge pressure of 2.4 barg; these values represent around 130% of BEP flow (or 149% of rated flow) and 69% of the generated pressure, respectively.

The estimate for head rise to shutoff is around 28%, which is suitable for this service. The rated power for the rated point is around 760 kW. The electric motor is sized for the end of curve; therefore, it's rated for 920 kW. The pump efficiency at the rated point is estimated to be around 84% which is relatively high for a vertical pump.

The selected impeller diameter is 720 mm; the maximum impeller diameter possible for the selected pump model is around 760 mm, enabling a 5.5% increase in diameter if required in the future (e.g., for a system

revamp, plant expansion, malfunction, seawater consumption rise, etc.).

Three parallel pumps. Here, the design team didn't perform an accurate dynamic analysis; later it was found that the pumps have bending natural frequencies just below the rotating speed. Each pump is expected to have two distinct natural modes at each natural frequency because of slight asymmetry. The pair of bending natural modes for these vertical pumps occurred at about 88% and 98% of the normal speed — the 2% separation margin was far too small. Therefore, the unavoidable unbalance of the pump rotor combined with the nearby natural frequency resulted in increased vibration levels. The sad reality is this can happen to many vertical pumps.

When the pumped liquid level was low, natural frequencies increased somewhat and shifted closer to the pump speed (making the margin less than 2%), leading to a slightly higher vibration response. The pumps were rebalanced on site to reduce the excitation. However, this didn't improve the separation margin of the natural frequencies and, so, vibration levels remained unsatisfactory. Solving the issue required some pump modifications to increase the separation margin. ●

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Improve Slurry Handling

Consider a variety of techniques when working with wet cakes, pastes and slurries

Tom Blackwood, Contributing Editor

Particulate solids can be troublesome to handle at times — but pastes, cakes and slurries can cause your hair to turn grey. While some are pumpable, others require special treatment without changing the overall characteristics of the product. The biggest problems often occur in storage and conveying processes. You should explore two categories of techniques to solve these problems.

First, consider altering the bulk properties to improve handling. Methods involve paste pre-formation, agglomeration, shear strength reduction, stickiness reduction and manipulation of the particle characteristics. However, altering the product's bulk properties can pose complications. Often the desired product is the paste or cake, so any modification might reduce its value or usability.

Second, look at equipment selection, design and operation. Most dryers don't handle lumps very well. Typically, cake from compression filters requires reduction in size before it can be conveyed or fed to a dryer. You must address not only the feeding problems but also the removal of these solids from any intermediate operation or storage. Equipment selection is a much broader topic than altering the bulk properties, so always investigate the latter first.

When altering the bulk properties doesn't pose insurmountable product quality or performance issues, you have a number of options. Let's focus on the major changes available:

- Pre-forming pastes using roller or pressure extruders with or without a cutter can create a more-uniform feed or even

Pastes, cakes and slurries can cause your hair to turn grey.

pellets that dry quicker or don't stick to the feed zone of calciners or kilns. Pre-formed or extruded material with high strength may allow use of screw or vibratory conveyors for transport.

- Size enlargement improves the handling properties of wet solids because the internal shear strength and wall friction generally decrease as material size increases. Enlargement can reduce caking and lump formation, lessen hazards associated with obnoxious materials and create a better blend that doesn't separate. Techniques include agglomeration, briquetting and granulation. You can grind the material, once dried, to the desired size.
- Shear strength reduction of a paste or cake can allow pumping at low pressure over long distances of a material that otherwise would require a high-pressure pump, which is limited to only short distances. A high-shear mixer or agitator, either on its own or as part of a feeder, can do the job. This technique suits thixotropic materials as well as material from compression filters because the moisture in the cake can be redistributed to decrease stickiness.
- Partial drying of the surface of a cake can reduce the tendency of the material to stick to the equipment surfaces. In addition to treating the solid, cooling of equipment surfaces sometimes will lessen adhesion. Companies in the food industry often rely on this approach to improve hygiene along with flowability. Another method is to blend part of the dry material with wet material. For wet cake, a double paddle mixer is common, while for slurry, a high-speed mixer is preferred.
- You can improve the flow curve (shear stress versus shear rate) of pumpable pastes through use of flocculants or dispersants to pare power requirements. Wet grinding of particles to below 5 micron can form a flocculated product that behaves like a coarse material without adding another chemical. Size distribution, shape and concentration all change the viscosity and pumpability. A wide-size-distribution material of rounded, low-aspect ratio particles has a lower viscosity at the same solids' concentration. Diluting the concentration generally will reduce viscosity — but at an increased cost. However, at very high solids' concentrations, a small change in concentration can be very

effective. Some other methods of creating stabilized slurry involve the use of fine particles to create a denser non-Newtonian suspended medium, and relying on fibers or oils to agglomerate coarse solids.

Slurry, paste and cake processing is both a chemical and physical challenge. Some items required to design these systems include: particle size distribution; fluid viscosity; solids loading; particle shape; stickiness; chemical composition (both of

particles and fluid); density (particle and fluid); and extrinsic parameters such as temperature and pressure. Phase-change parameters such as enantiomers and polymorphs along with particle shear properties can complicate a technology's application. Begin by looking at what bulk properties you can manipulate in your process. ●

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Head Off Cavitation

Air injection into centrifugal pumps sometimes can prevent the problem

Andrew Sloley, Contributing Editor

Occasionally, you may spot something in an article or an equipment installation that you consider really clever and worth remembering for future use. One example is injecting gas into a centrifugal pump suction to reduce cavitation damage and improve pump performance. I came upon this years ago. It involved injecting air into a booster pump moving cooling water to the top of the cooling tower. This seemed odd. Engineers spend a lot of effort preventing air getting into pump suctions. What was going on?

To understand this, some basic knowledge of pump cavitation and damage mechanisms helps. Cavitation is the physical response of a system to meet mass balance requirements. It involves vapor formation

in the inlet line to the pump to reduce discharge head of the pump. This vapor formation results in decreased pump discharge capacity against the downstream pressure requirements. (This definition is focused on how process engineers understand systems rather than on the more conventional definitions that are focused on how machinery engineers understand pumps.)

As any process engineer knows, a mass balance requires that flow out equals flow in. Cavitating systems have restrictions on flow in. The restriction may be in the feed piping or in a limit of liquid available. If the system downstream of the pump can accept more flow out than can flow into the pump, something must change to make flow out equal flow in.

When discharge flow exceeds suction flow, the upstream accumulation on top of the pump suction changes (out – in = Δ accumulation). The accumulation continues to drop until the pump capacity drops to out – in = 0. Cavitation accomplishes this. The out flow rate drops until it reaches the in flow rate.

We express the required accumulation to avoid cavitation as a liquid height over the pump suction, or a net positive suction head required (NPSH_R). The NPSH_R is

determined by testing a pump and measuring the discharge pressure. The Hydraulic Institute defines NPSH_R as the pump suction head that reduces the discharge head by 3%. NPSH_R isn't the NPSH that equals

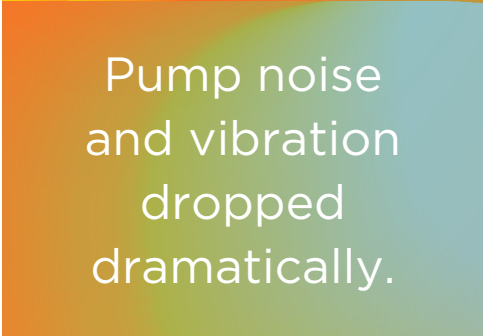
no cavitation. Rather, NPSH_R is the suction head where a specific amount of cavitation already is occurring.

Cavitation damage comes from vapor bubbles in the pump collapsing and creating a damaging shockwave. The maximum damage happens at a suction head just above the NPSH_R value.

I recently reviewed an installation having a cooling tower booster pump with an NPSH_R of 12 ft and an NPSH_A (available) of 14.5 ft. Maintenance history showed extensive and frequent repairs on the pump. The

operation with the 14.5-ft NPSH_A was very close to the worst operating point possible. Total collapse of vapor pockets was causing severe damage.

The solution was to use air injection. Air bubbles in the cooling water don't completely collapse. This reduces the damage to the pump. You must take care to avoid too high an air rate, which can cause the pump to lose prime and stop working.



Pump noise
and vibration
dropped
dramatically.

We hooked up instrument air to the cooling water pump suction. We installed two check valves in the instrument air injection line followed by a 3/8-in. needle valve and then a pressure regulator. This is more

than strictly needed but dealt with some concerns from the plant operators. The pressure drop across the system was high enough for the orifice valve to operate as a critical flow device. The air flow rate didn't vary with cooling water system pressure. We tried various settings to find a stable operating point and found that at a roughly 11-psig regulator pressure and a specific position on the needle valve, the system was very stable.

With the air injection, pump noise and vibration dropped dramatically. Over an extended period, pump maintenance

needs fell and reliability improved. The best solution, of course, would be to have a system correctly constructed for reliable operation without this type of patch. Nevertheless, air injection did help substantially here.

Air injection is acceptable in this system because the water goes straight to the top of the cooling tower. The air requires no downstream handling. Process systems

often are more difficult to address because of problems with injecting inert gas.

For more on cavitation in centrifugal pumps, particularly how to use pump curves for troubleshooting, see “Kayo Cavitation,” <http://goo.gl/4kHmSC>. ●

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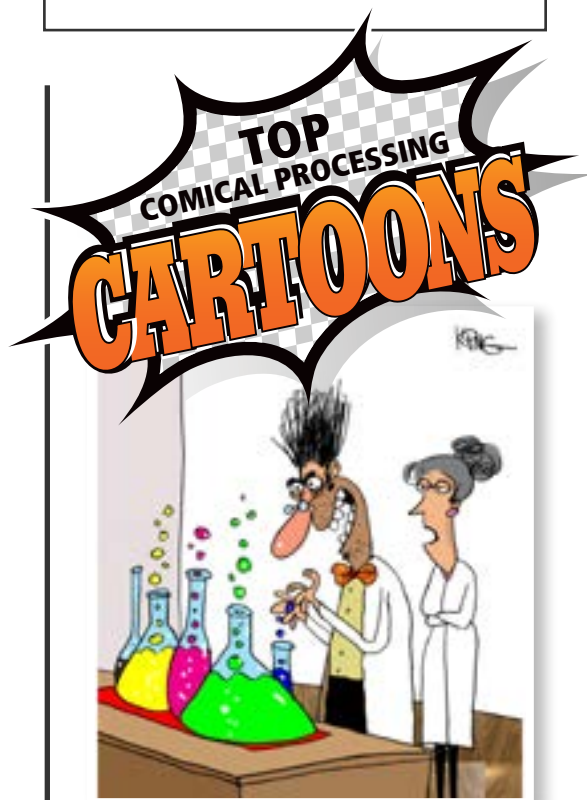
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