CHEMICAL PROCESSING

Flow eHandbook

HOW TO FIGHT
FLOW CHALLENGES

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Flowmeter Suits Low-Flow Applications

Device uses surface acoustic wave technology to measure volume flow, temperature and density

THE FLOWAVE flowmeter uses surface acoustic waves (SAW) technology, which relies on the propagation of waves for measurements, similar to those in seismic activities.

FLOWave indicates its device status based on Namur NE107 definitions, and requires minimal maintenance. Both shorten downtime, resulting in lifecycle cost reductions. Transmitter, sensor and measuring tube stand up to strict hygienic requirements.

In addition to volume flow, temperature and density can be measured, too. Based on this data, the mass flow rate can be calculated. The flowmeter also works with stagnant liquids so results are available even with the smallest flow volumes. It recognizes quick flow rate changes reliably, thus making it suitable for very fast filling processes. The high excitation frequency of 1.5 MHz avoids disturbances due to inherent vibration in the plant. Magnetic and electrical effects have no influence on measurements. It's reliable even in the presence of gas bubbles or solid particles, and can distinguish between laminar and turbulent flows, says the company.

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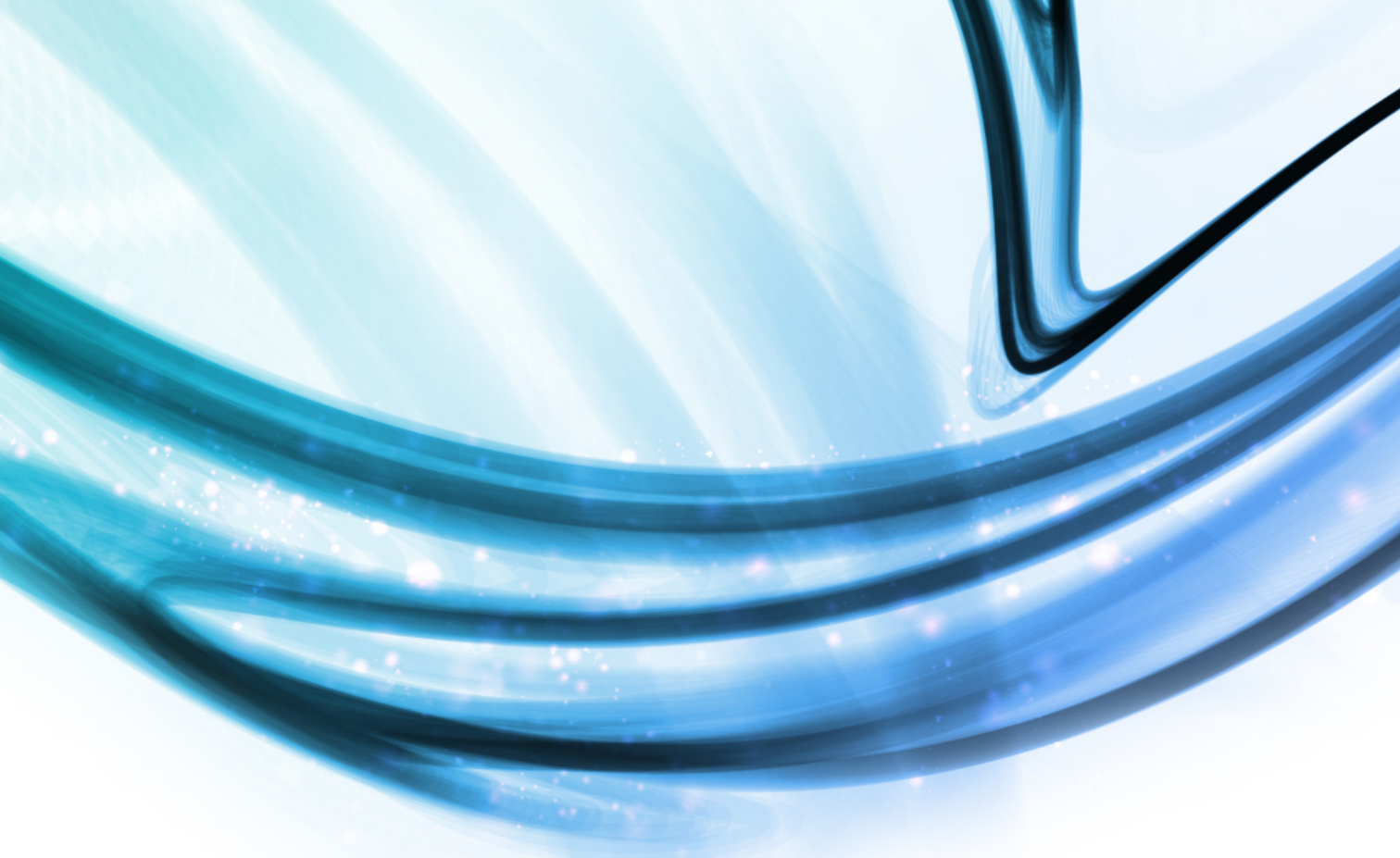
Carefully Commission Hydrogen Pipe

Safety depends upon scrupulously performing a variety of inspections and tests

By Dirk Willard, Contributing Editor

A LOW-PRESSURE alarm sounded on a refinery hydrotreater-compressor suction line during startup. Two operators were dispatched. Minutes later, hydrogen collecting under the roof of the poorly ventilated compressor building exploded, killing them. Investigators later determined the cause to be a leaky, poorly sized gasket that passed inspection during construction.

This accident highlights the need for greater care in commissioning hydrogen systems. Unfortunately, advice is sparse. Background yourself on such systems by reviewing: ASME STP-PT-006, "Design Guidelines for Hydrogen Piping and Pipelines;" NACE Standard TM 0284-2003, "Evaluation of Pipeline and Pressure Vessel Steels for Resistance to Hydrogen-Induced Cracking;" and API-RP 941, "Steels for Hydrogen Service at Elevated Temperatures and Pressures in Petroleum Refineries and Petrochemical Plants."

Begin the commissioning process by focusing on the spools. Examine every component before assembly. Require vendors to provide complete records on materials and testing data for gaskets and pipe. Welds and parent metal both should have a maximum Rockwell hardness of 22 and a Brinell hardness of 250. Don't treat instruments as whole devices: inspect the parts. After reviewing the data files and

correcting component issues, weld the piping; follow with commercial-grade cleaning. Assemble the spools completely: do bench assembly tests. Pay attention to weight — badly balanced, heavy spools are difficult to assemble, so add lifting lugs as needed. Redesign poorly placed vents and drains to ensure proper slope. Polish smooth all rough welds and perform surface treatment to reduce thermal expansion and cold working. If pigs will be used in cleaning, confirm a clear path and the means of pigging. Take special precautions with relief vents: sharp angles and a sudden discharge promote mixing that can lead to spontaneous H_2 fires. Inspection failures take you back to square one.

Effective cleaning of the spools eliminates future field-assembly problems. Some components will require disassembly and separate cleaning and drying. Pigs usually are used for pipe: first, soft shell pigs to look for obstructions; then rubber shell pigs for moving cleaning agents and dewatering; and finally wire brush pigs to remove pipe scale from carbon steel (CS), followed by a descaler chaser. Wire brush pigs generally aren't used with stainless steel (SS) pipe. If you intend to send a wire brush pig through SS pipe, confirm the pig previously hasn't been used for CS. Do a ferroxyl test of SS to detect rust contamination. Cleaning
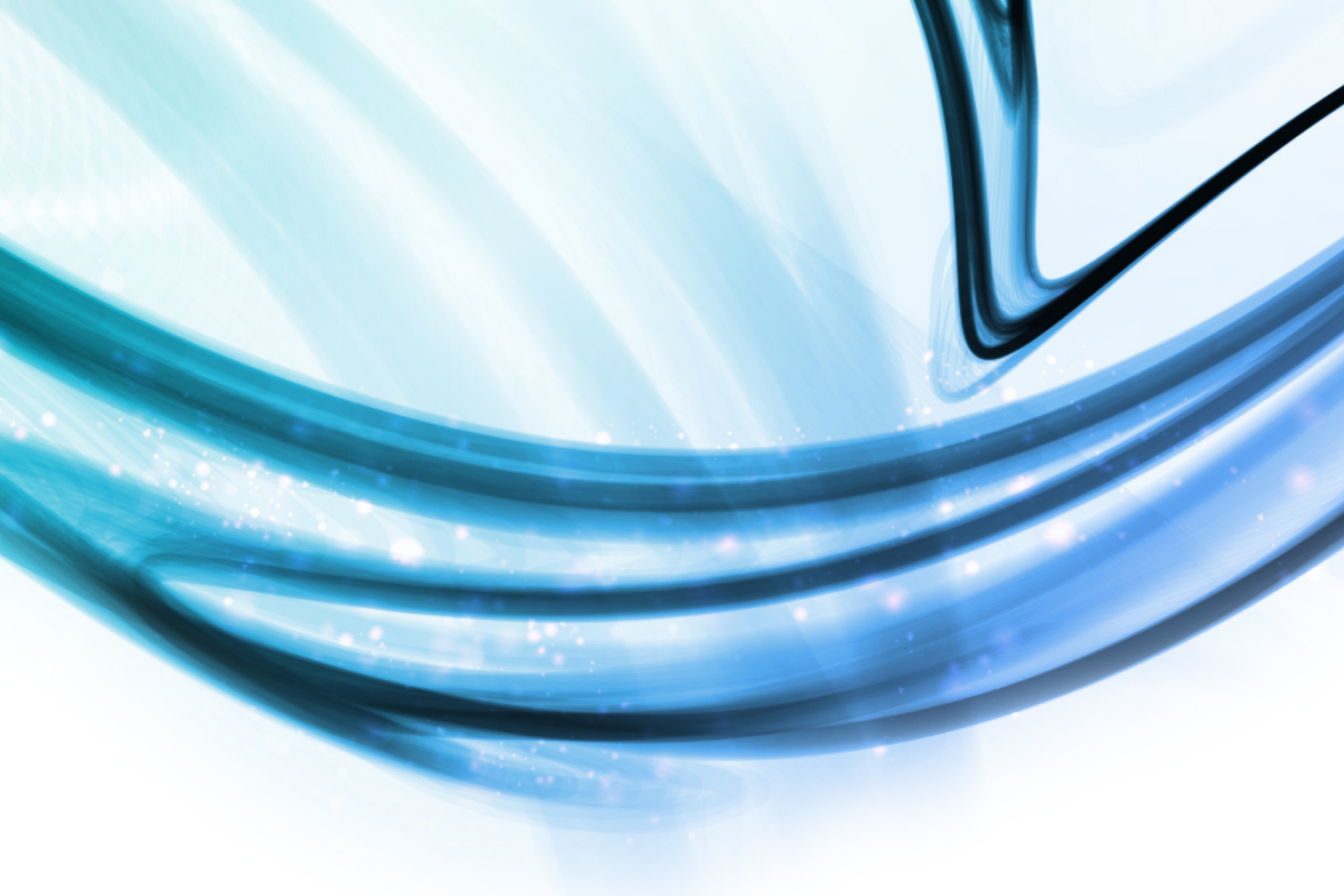
generally involves a slug of detergent or mild caustic, followed by water and a citric acid solution to passivate the SS, and finally buffered water and dewatering. Muriatic acid is a descaler often used for CS after detergent. Verify that gasket materials are compatible with cleaning solvents.

Drying is next, usually with nitrogen or air, although argon has been used because it sweeps slightly better than nitrogen. Dry by intermittent purging until the difference between input and output dew points is <5°C; vacuum drying works best — if you can avoid leaks. Venting is important because H_2 can form when acids react with metals. If air is used, the requirements are a dew point ≤-40°C and a hydrocarbon content <140 mg/1,000 std ft^3. Perform a helium leak test after drying to detect flange surface irregularities; helium is mixed as a trace gas in nitrogen and detected by mass spectrometer. Once purged to <1 mole % O_2, seal and maintain the spool with 10 psig pressure. By the way, going the other way, to remove H_2, aim for <1 mole % H_2. (Keep in mind the lower explosive limits: 5.7% for O_2 and 5% for H_2.) These values are at 1 atm and 25°C: values decrease with increasing pressure or temperature. You may need to mix vent gas with a secondary fuel for the flare once the H_2 is depleted.

Field assembly is next. Record the flange torques after inspecting the final gaskets; tighten again during initial operation. Fires have resulted from leaks caused by poor monitoring of flange torques during hot-bolting, especially where SS bolts are used with CS flanges (because SS undergoes greater thermal expansion). Take care to ensure that instruments are included. A standard pressure test is next on the checklist. This is followed by cleaning, drying and, perhaps, another helium leak test, and then finally by a hydrogen sweep to eliminate trace impurities. Follow a broad sampling plan during purging and leak-testing.

For further information, see:
http://goo.gl/K3QNVQ
http://goo.gl/laaJGs
http://goo.gl/ydZ9Sk

For tips on material selection and piping design for hydrogen systems, see: "Head off Hydrogen Hazards," http://goo.gl/PmPyuB. ●

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Get Up to Speed on Axial Compressors

These complex machines provide benefits but also pose concerns

By Amin Almasi, rotating equipment consultant

AN AXIAL compressor is a compact turbo-compressor that suits applications with a very large flow and a relatively small pressure difference (head). It probably is one of the most crucial and complex turbo-machines at many process plants. Achieving and maintaining desired performance depends upon properly addressing some complicated design and operational issues. These include fragile blades, manufacturing problems, surge, stall, noise-related concerns and many more.

An axial compressor (Figure 1) offers higher efficiency, speed capability and capacity for a given size than a centrifugal compressor. However, it has a narrower recommended application range (Figure 2) and delicate components. Some compressors contain both axial and centrifugal stages (Figure 3).

Some operating companies will use whenever possible rugged, versatile and reliable centrifugal compressors instead of dedicated, efficient but fragile axial machines. Large horizontally split centrifugal compressors now are available in capacities up to ≈450,000 m³/h or even more. However, for very large capacities (say, >1,000,000 m³/h), an axial compressor may be the only option.

It's difficult to give a general rule for selecting between a very large, sturdy centrifugal compressor and a compact, relatively more efficient, properly optimized more-economical axial machine.

DESIGN ISSUES

Many chemical plants require axial compressors to operate within a relatively wide operating envelope (capacity/pressure range), and sometimes relatively far from nominal conditions. Considering the steep nature of an axial compressor's curve, this is a great challenge. Variable-speed and the variable inlet-guide-vanes (IGV) systems can provide additional flexibility in operation. A number of issues, including structural, vibration, weight, cost, manufacturability, accessibility and reliability, need evaluation for any axial compressor.
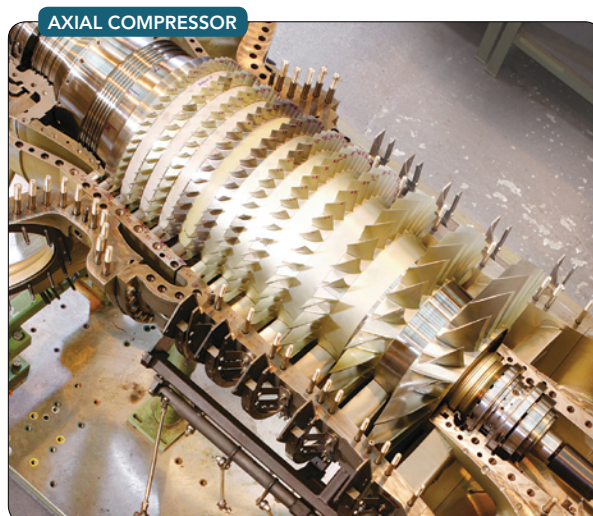

AXIAL COMPRESSOR

Figure 1. Units such as this can handle large flows with relatively small head. *Source: Siemens.*
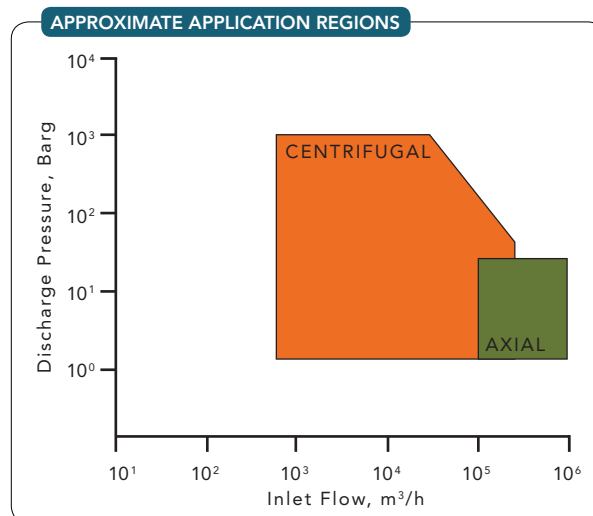

APPROXIMATE APPLICATION REGIONS

Figure 2. Axial compressors have a narrower operating range than centrifugal machines.

The operating Mach number usually is less than 0.8 for a subsonic cascade but can go up to 2 or more at the tip of a transonic blade assembly. Some subsonic axial stages can develop pressure ratios on the order of 1.5–1.8. The transonic stages operate with pressure ratios of ≈2 and greater while maintaining an acceptable efficiency and aerodynamic design. A well-designed subsonic axial stage can achieve a polytropic efficiency of ≈0.9. The polytropic efficiency for transonic blades is a bit lower, say, ≈0.82–0.89. High peripheral-mean-stage rotor velocities can reach ≈300–340 m/s for subsonic rotors and up to ≈580 m/s for transonic ones. Designers set the annulus radius (or hub-to-tip) ratio, R_{hub}/R_{tip}, after a careful optimization that considers aerodynamic, technical, mechanical and economic constraints. For inlet stages, assigned R_{hub}/R_{tip} values usually range between 0.45 and 0.65 while outlet stages often get a higher value, typically from 0.75 to 0.9, to achieve a relatively high Mach number.

A proper axial compressor design should avoid a flow separation inside the machine. Analysis of axial Mach number distribution along the different blade stages is essential. This distribution should follow an acceptable
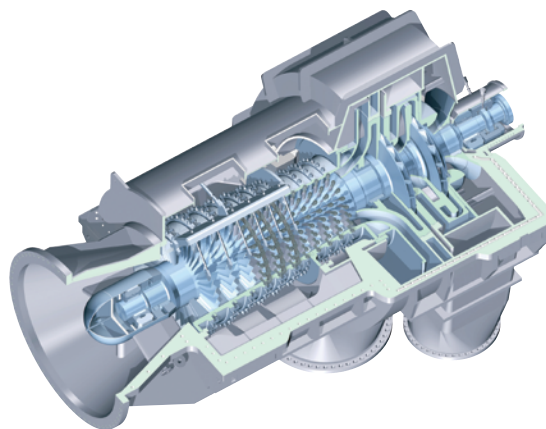

DIFFERENT TYPES OF STAGES

Figure 3. The initial (high flow) stages of this compressor are axial while the final two stages (low flow) are centrifugal. *Source: Siemens.*

pattern; variation shouldn't exceed a specified level.

The ultimate goal of an axial compressor design is to create an axial blade arrangement with the maximum pressure rise and the minimum total pressure loss, i.e., a relatively high efficiency, along with an acceptable operating range.

The different blade and component profiles play an important role because these can affect the nature of the boundary layers and, therefore, the amount of losses (and the operating margins). The stage arrangement is critical; the stage stacking procedure intrinsically is iterative.

Maximizing the adiabatic efficiency can significantly impact the choice of stage geometrical and functional variables. In addition, it's important to optimize the surge/stall margins.

An optimum axial compressor design combines minimum weight with compactness. This calls for decreasing the number of stages and increasing individual stage loading, which can affect the choice of blade shape and cascade parameters.

The availability of advanced materials for blade/component construction and high-quality production methods makes it possible to reach levels of aerodynamic loading never before possible in axial compressors while preserving high levels of efficiency for normal and alternative operation cases. This is true both for high-speed subsonic as well as ultra-high-speed transonic blades.

SURGE CONTROL

A surge event can damage or even separate the fragile blades of an axial compressor. However, the machine's surge line maps are complex. The surge line could change with a slightly different gas condition or composition. So, an axial compressor requires a dedicated anti-surge system. This usually includes five protection arrangements:

- an anti-surge valve;
- a hot-gas-bypass valve;
- an inter-stage bleed valve (IBV);
- an IGV system; and
- speed variation.

The speed reduction and IGV characterization should be used to map the "surge area" in a two-dimensional plot. In addition to an anti-surge valve, an axial compressor most often is protected by a hot-gas-bypass (HGBP) recycle loop, usually with a hot-gas-bypass valve. This is mandatory if the anti-surge valve isn't installed immediately after the compressor discharge. The IBV also should be opened to provide sufficient flow to the compressor suction to avoid surge at initial stage(s).

Dynamic simulation is crucial. The model requires accurate, actual dynamic performance data such as the IGV stroke speed, the control and actuator delay, and the valve stroke time. These data play important roles in dynamic simulation results, anti-surge system design, reliability and overall safety. Proper validation of the model is essential, considering the criticality of avoiding surge and the disastrous consequences of a surge event.

The IGV stroke time usually is in the range of 2–6 s. Conceptually, fast response of the IGV system might seem desirable as it could help unload the compressor quickly. However, the IGV stroke could affect the performance curve — for example, the distance between the operating point and surge. Results from some dynamic simulations indicate that fast closing of the IGV mechanism sometimes (depending upon the compressor's operating map) could drive the machine toward surge. This suggests that a moderate IGV stroke time, say, 3–5 s, rather than the fastest time, might be better for surge prevention. The stroke time of an IBV possibly could be as short as 1.5–2 s. However, for a machine in which surge could initiate at the final stages, fast IBV opening could pose problems because it can significantly reduce the gas flow at the final stages. So, it's important to determine an optimum window for the opening time to avoid surge in either section of axial stages. Accurate dynamic simulations are essential for identifying all these optimum values. ●

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Fight Flow Recirculation

But first understand why the phenomenon occurs in centrifugal pumps

By Andrew Sloley, Contributing Editor

SUCTION RECIRCULATION can undermine operation of a centrifugal pump at low flows. The pump's suction specific speed and suction energy can provide insights about potential difficulties (see: "Cut Pump Speed to Cut Problems," http://goo.gl/smNPqX). Variable frequency drives (VFDs) frequently can provide a solution ("Consider VFDs for Centrifugal Pumps," http://goo.gl/cXFbPk).

Fixing suction recirculation problems often costs money. Typically, getting the funding requires convincing the holder of the purse strings of what's the real cause of a problem.

Many people have trouble grasping the idea that flow can go backward within a centrifugal pump. Over the years, I've often struggled to find a fundamentally correct but simple explanation.

First, we must realize that what's simple to a mechanical engineer may appear complicated to a chemical engineer. So, let's explain the phenomenon in ways both groups can understand.

For mechanical engineers, a centrifugal pump increases the pressure of a liquid stream. The natural direction of fluid flow is from high pressure to low pressure. Fluid flow through the pump occurs because the rotating impeller provides velocity to generate pressure. The velocity gradient creates the discharge pressure.

In a centrifugal pump, liquid enters the suction eye at the center of the impeller disc. The liquid changes velocity before exiting the pump at the impeller periphery. To deal with the geometry of flow from the center to the edge of the impeller and changes in velocity, the flow passage shape changes. Figure 1 shows an end-on view of an impeller that's rotating counter-clockwise and Figure 2 shows a side view.

Inlet flow contacts the impeller's leading edge at a specific incidence angle (Figure 1). Vector analysis of liquid flow directions shows the incidence angle affects eddy
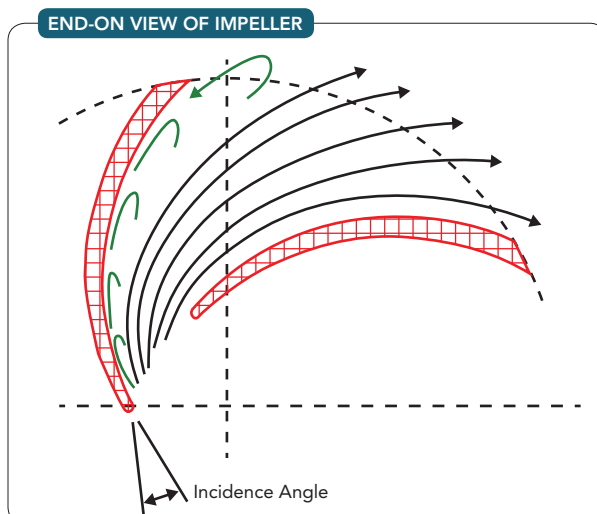

Figure 1. The incidence angle at which inlet flow contacts the impeller's leading edge affects eddy flow formation.

formation. As flow rates drop, the eddies formed become larger. Eventually, eddies can create partial flow from the pump discharge to the pump suction. The local flow pattern follows the outline shown in Figure 1.

For chemical engineers, it's better to consider how pump flow patterns interact with material balance boundaries. A series of material balance boundaries through the pump always will have the same net flow as the pump suction flow. If pump suction flow drops, the net flow through any cross section of the pump drops. Because typical fluids are incompressible at most conditions, this creates an absolute requirement that average velocity in any cross section varies linearly with flow rate.

However, close to the pressure edge (the leading surface) of the impeller, impeller speed sets liquid velocity. If average liquid velocity must drop but velocity in that area of the flow passage is close to constant, velocity in

IMPELLER SIDE VIEW

Average flow velocity vector at 1.0 × BEP

Example vane exit velocity vectors at 1.0 × BEP

Example inlet velocity vectors at 1.0 × BEP

Average flow velocity vector at 0.25 × BEP

Example vane exit velocity vectors at 0.25 × BEP

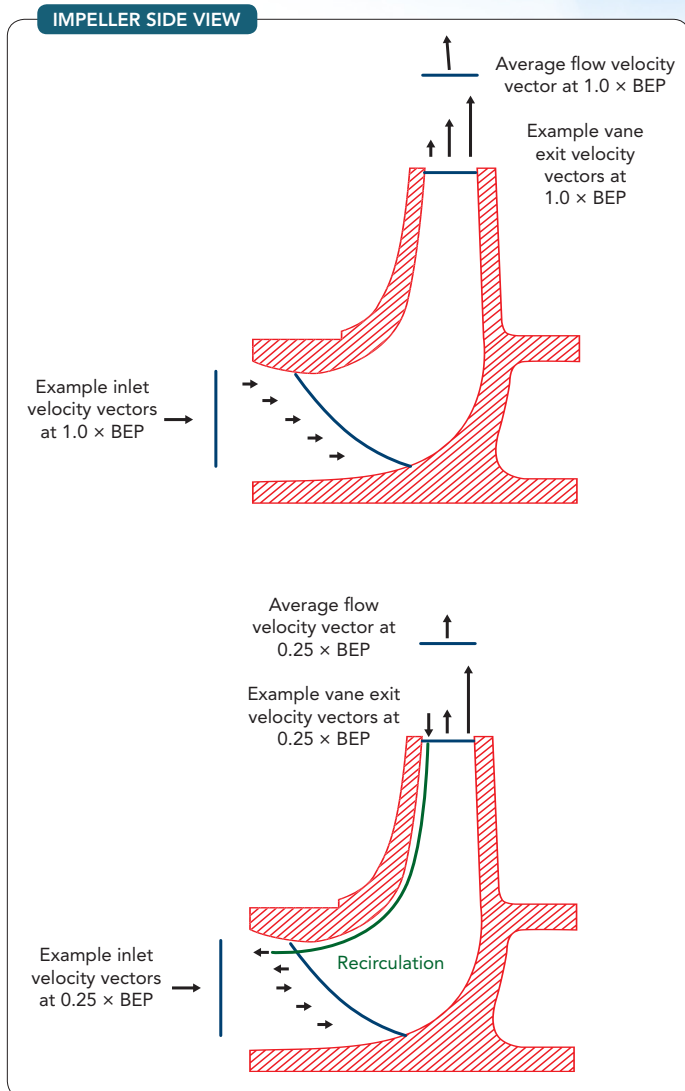Example inlet velocity vectors at 0.25 × BEP

Recirculation

Figure 2. Proper flow takes place at the BEP (top) but at 0.25 BEP (bottom), to satisfy the average velocity requirement, recirculation occurs.

other areas must fall even more than the average. At some point, when average velocity decreases enough, the flow direction in areas far from the impeller's leading edge must reverse to meet the average velocity requirement. Figure 2 shows a schematic of net flow in a pump at its best efficiency point (BEP) and at a low flow condition (0.25 of the BEP). At low rates, the flow passages in the pump are simply too big. Nevertheless, they must be filled with liquid. Flow recirculation results.

Flow recirculation can damage the impeller due to cavitation caused by vaporization in the low-pressure regions that recirculation creates. Flow recirculation also stresses pump components with unbalanced forces and vibration.

Both mechanical and process changes can reduce the consequences of flow recirculation.

Mechanical solutions focus on the pump. For instance, impeller-volute-geometry matching and vane-angle, leading-edge and inlet-eye modifications, as well as pump speed changes all can improve pump flexibility. However, each affects efficiency, discharge head and capacity differently.

One process modification adds a recirculation loop to keep the pump out of the low flow region. Recirculation systems require extra equipment (piping, restriction orifices, control valves, etc.). In my experience, many flow control loops are abandoned due to maintenance costs or ignorance of their importance.

Another process modification provides excess suction head to the pump. This helps prevent cavitation-like damage. Even the low-pressure regions in the pump have sufficient head to keep the fluid above its bubble point. However, extra suction head doesn't solve stress and vibration problems. At some point, pump vibration may exceed good practice values. Extra stress and vibration decrease mean time between repair and mean time between failure. Maintenance costs rise with high vibration and stress. Reduced operating speed lowers vibration and pump loads.

While it may not be a perfect solution, switching to a VFD can benefit nearly every centrifugal pump service suffering from inlet recirculation.

Picking the right option requires a thorough analysis of both mechanical and process constraints and costs. ●

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