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# Pump control strategies benefit from compressor know-how

## Similar techniques can be used to avoid problems with parallel operation

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Operating pumps in parallel is not without risk. Parallel operation takes on added importance whenever high-pressure pumps are involved. In fact, some users have experienced serious pump failures on their first installation with perhaps only two pumps operating in parallel. Indeed, *some* users actually *expect* problems controlling parallel operation of multiple pumps.

This is where centrifugal compressor control strategies and related know-how should come into play. For many years, competent manufacturers and providers of surge abatement devices have implemented these strategies with great success in compressors. The pump projects recently analyzed by at least one company lead to the conclusion that the related know-how can easily be applied to pumps and will resolve design challenges.

The application explained here refers to high-pressure, six-stage centrifugal pumps used to inject seawater into oil wells (Fig. 1). It involves a set of three lower-pressure motor-driven pumps operating in parallel. This set of pumps takes seawater from the deaerator at nearly atmospheric pressure and discharges it to a common manifold. In turn, the manifold connects to the inlet of the three gas turbine-driven high-pressure pumps. The three pumps operate in parallel at a design head (pressure) of 2,500 m (approximately 8,200 ft or 250 bar). The rated power per pump at design flow is approximately 2,500 kW.

The design calls for a control system to modulate speeds of the three gas turbine-driven pumps to achieve the desired flow. On the pump performance map (Fig. 2), mass flow and pump head are shown on the x and y coordinates.

Within the low-flow or internal recirculation region, operation for prolonged periods is not allowed. Location of the low-flow boundary is similar to the surge line found on the typical centrifugal compressor map; it originates at the x-y intersect and is represented by a positive, slightly parabolic sloping line as flow and pressure (pump head) increase. Depending on the pump geometry and such parameters as speed, temperature rise and *NPSHR*, the pump should not be allowed to operate to the left of this line for more than 30 to 60 seconds. Again, depending on the pump design and associated parameters, the low-flow boundary is tied in with specific flow/head intersects and, in each case, a corresponding pump efficiency.

As internal recirculation leads to partial vaporization and bubble implosion, cavitation-erosion intensifies and internal heating increases; pumping efficiency will then drop. The situation tends to become worse at progressively lower flows. Eventually this leads to serious pump damage. The degree of damage depends on how far the operating point has penetrated into the undesirable region of diminishing efficiency, and how much energy was dissipated in bubble collapse and temperature excursions. Moreover, damage to pump internals, bearings and mechanical seals often accelerates and progresses to the point of risking catastrophic failure. In certain instances and where the fluid density is thus falling, the resulting load reduction may even lead to turbine overspeed events.

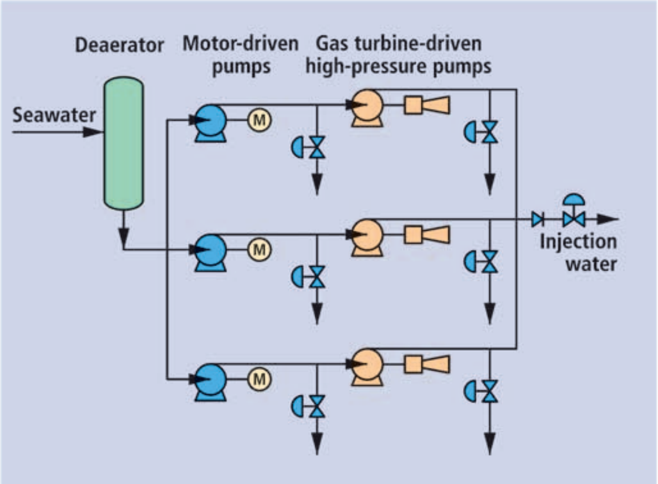


FIG. 1 The application involves a set of high-pressure, six-stage gas turbine-driven centrifugal pumps used to inject seawater into oil wells.

Parameters associated with risk-prone or life-shortening parallel operation of centrifugal pumps are known and can be measured. The situation is analogous to that of surge in centrifugal and/or axial (dynamic) compressors. There is a similar quick rise in temperature, loss of pump performance at the moment when the event occurs and a step change in axial vibration severity. All of these can result in damage ranging from immediate and catastrophic to gradual and long-term.

How does one pump run into trouble before its *matching parallel* pump encounters difficulty? It is highly unlikely that two pumps are totally identical; neither their internal surface roughness and clearances nor their external piping configurations are the same. Because the flow vs. head curves are more flat at flow ranges to the left of the best efficiency point (Fig. 2), one of the two or more pumps operating in parallel tends to drive the other(s) off the curve and into the extreme low-flow region. Or, if too many

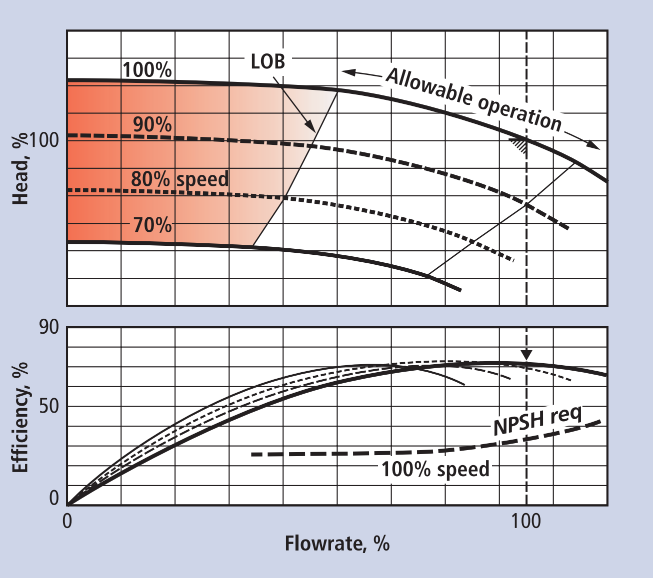


**FIG. 2** Pump performance map.

pumps are operating in parallel, none of them are likely to operate at optimum flow. Conceivably, then, each is operating with internal recirculation or cavitation.

**Comparison with compressor surge.** The entire problem is not too different from surge events in dynamic compression equipment. Here, too, low-flow avoidance, efficient control and appropriate load-sharing requirements must be addressed as part of the system's design. No two machines operate at precisely the same internal flow bypass conditions. There are always differences in piping arrangements, fluid boundary conditions, potentially different localized temperatures, etc.

Although pressure equalization takes place and causes multiple units to operate at the same head, they may each operate at a different flow. The pump operating at the lowest flow will hit the low efficiency boundary (LOB) first. It could be argued that this LOB is equivalent to a surge line in a dynamic gas compressor. As the remaining flow is further redistributed between pumps, the pump at disadvantage regresses further into the zone to the left of the LOB. It now encounters the flat portion of the performance curve, where minor changes in system resistance can cause major flow changes.

Pump controls might benefit from compressor technology where the familiar load-sharing systems have worked exceedingly well for years. Measuring the system variables on each pump and using flow and pump pressure differential as the primary inputs, the so-called *S*-variable reflects the distance to individual LOBs. Unlike compressor systems, there are generally no individual recycle valves. However, a rigorous system analysis will often lead to recommendations for recycle valves, even if only for starting up the pump. The measured *S*-variable will be used to load-share between units by sending the setpoint to the speed controllers, and a master controller will control overall flow to the flow setpoint (Fig. 3).

**What can we conclude?** Reliability professionals should take away from our brief overview at least four relevant conclusions:

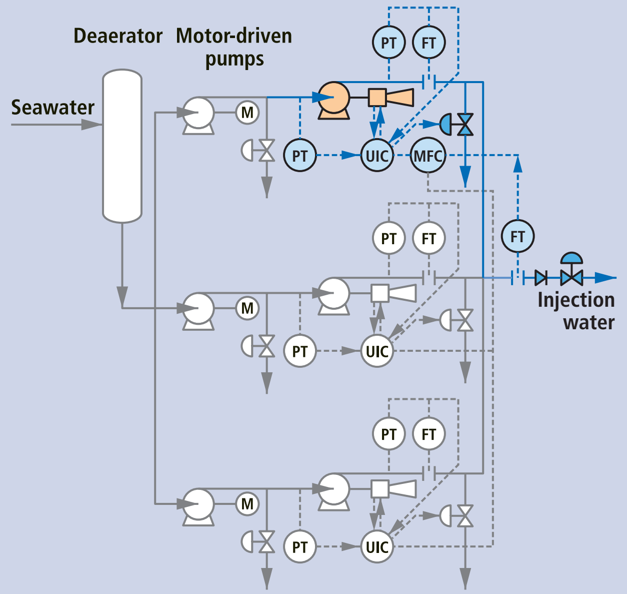


**FIG. 3** A master controller controls the overall flow to the flow setpoint.

1. High-pressure centrifugal pump systems suffer from problems similar to those experienced in systems and process loops utilizing centrifugal compressors. This should be a powerful incentive to apply compressor experience in design of functionally similar pump control systems.

2. Attention must be given to methods that accurately represent the pump performance map in the control equipment. As with compressor systems, accuracy and stability are of paramount importance and must be addressed competently.

3. Just as it would be an unquestionable requirement in compressor control system design, a pump control scheme must incorporate relevant decoupling features to suitably address issues of multiple interacting variables.

4. An optimized pump control system must, therefore, address the issues relating to load sharing between multiple units, as well as certain aspects of available or pertinent operating envelopes. **HP**

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