INFLUENCE OF GAS SEALS ON PUMP PERFORMANCE AT LOW SUCTION HEAD CONDITIONS

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ABSTRACT

Pressurized gas seal technology has been applied successfully to many types of equipment, in various processes, and under diverse operating conditions. Using a readily available gas supply, typically nitrogen, the seal is capable of providing zero process emissions, no process contamination, and high reliability, even at off-design pump operation. Recently, a discrepancy was observed during standard net positive suction head (NPSH) testing at the pump manufacturer. Under certain operating conditions, normal gas seal leakage into the pump may influence pump performance.

A standard pump NPSH test objective is to vaporize liquid within the impeller and to generate a measurable head loss as a direct result. When a gas seal is used in a pump NPSH test, the total differential head (TDH) is reduced for reasons other than cavitation. Before the liquid's vapor pressure can be reached, the gas leakage across the seal faces expands and reduces the pump's TDH by three percent or more. Although the pump test indicates a high $NPSH_R$ (the R is for required), in reality, the pump never reaches the point of cavitation.

Based on standard NPSH testing, a test program documented the effect of gas leakage into the pump under varying operating conditions. While operating at both low suction pressure and low pump flowrate conditions, the effects of gas leakage are the most evident. Because these conditions are inherent in standard NPSH tests but much less common in field installations, the relevance of pump NPSH performance data should be evaluated at true conditions. The results of these tests may help pump users and pump suppliers understand the effects of gas leakage into the pump. Pump and seal operating guidelines are outlined to minimize the influence of gas and to avoid field problems.

INTRODUCTION

The use of pressurized gas lubricated seals in process pumps is becoming increasingly popular to control pump emissions, reduce maintenance requirements, and improve seal reliability. Gas seals can be very tolerant of off-design operating conditions where conventional liquid lubricated seals fail. Although the inert gas used as a barrier fluid does not contaminate the product, gas leakage into the pump can cause other problems.

The catalyst that prompted this research was the failure of a routine net positive suction head (NPSH) test performed by the pump manufacturer for a specific end user. A major new project in Europe included the installation of nearly one hundred pumps with pressurized gas seals installed. Prior to shipping the pumps to the end user, the pump manufacturer performed standard NPSH tests on each pump. It was during these tests that an apparent anomaly was discovered on a limited number of the pumps.

In an effort to identify the root cause and to provide a working solution, a test program was initiated at several facilities. The inaugural facility in Etten-Leur, the Netherlands, first identified the problem on their test stand. In Dayton, Ohio, testing focused on the two most popular sizes of ANSI pumps. In Roosendaal, the Netherlands, experiments focused on vertical pumps with gas seals. At a German pump manufacturer, testing was performed with multiple gas seal manufacturers. Even with such a diverse mixture of pump and seal types, the common problem was at low suction head and low pump flow conditions.

It is important to understand the similarities and differences between head reduction caused by cavitation and head reduction that has been demonstrated with gas seal usage, specifically at low suction head and low flow conditions. Starting with a discussion of NPSH and the role of cavitation, a review of gas seal technology and the influence of entrained gas will help formulate the problem. Laboratory test results and field reports are examined relative to both OEM and end user issues. Gas ingestion is discussed for horizontal and vertical centrifugal pumps and recommendations are presented for current and future installations.

NPSH DISCUSSION

A discussion of net positive suction head (NPSH) is needed to understand pump cavitation. Because every liquid will vaporize under the right combination of temperature and pressure, it is important to operate the pumping system outside any vaporization opportunity. One of the outcomes of an NPSH test is a map of the vaporization limits relative to the pump's operating envelope.

$NPSH_A$ describes the net positive suction head available at the suction of the pump. It is the difference between the total suction head at the pump suction flange and the vapor pressure of the liquid. $NPSH_A$ can be determined by calculating the suction static elevation (h_{ss}), adding the absolute pressure on the liquid surface (h_{psa}), and subtracting both the total friction losses to the pump suction flange (h_{fs}) and the fluid vapor pressure (h_{vpa}).

$$NPSH_A = h_{ss} + h_{psa} - h_{fs} - h_{vpa} \quad (1)$$

$NPSH_A$ may also be determined by measuring static gauge pressure (h_{sg}) at the suction flange, adding the velocity head (h_{vs}) and the atmospheric pressure (h_a), and subtracting the fluid's absolute vapor pressure (h_{vpa}).

$$NPSH_A = h_{sg} + h_{vs} + h_a - h_{vpa} \quad (2)$$

Net positive suction head required ($NPSH_R$) is defined by the Hydraulic Institute as "the amount of suction head, over vapor pressure, required to prevent more than three percent loss in total head from the first stage of the pump at a specific capacity (Hydraulic Institute, 1994)." As liquid enters the pump illustrated in Figure 1, it experiences head losses as described in Figure 2. Just after entering the impeller, where the liquid filters into the vanes at location D, the head loss is greatest. This is where vaporization begins.

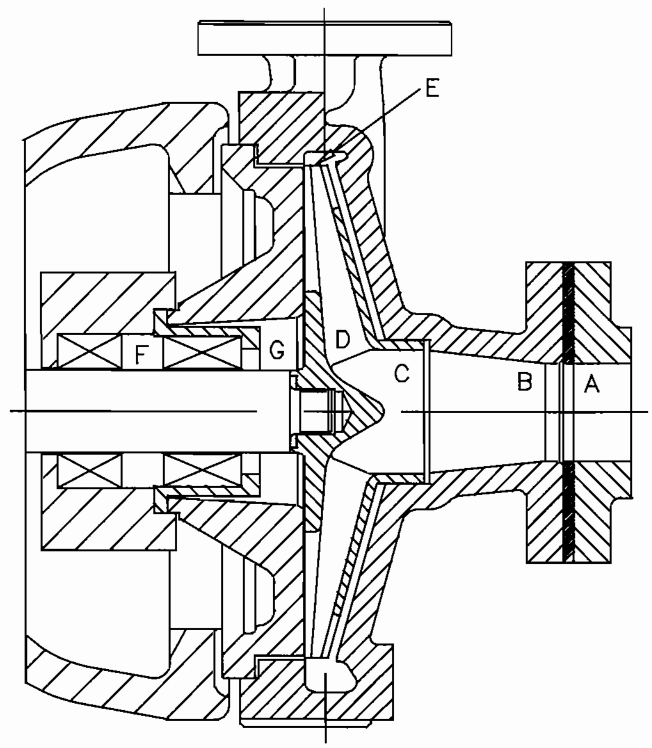

Figure 1. Centrifugal Pump Cross-Section.

If $NPSH_A$ is less than $NPSH_R$, cavitation occurs. A cavitation vapor bubble takes up space within the impeller and reduces the impeller's ability to develop pressure. In addition to head loss, cavitation causes noise, vibration, and possible damage to the pump. Vapor bubbles are formed in low pressure regions of the pump and collapse after traveling to higher pressure regions in the impeller. Vibration is caused by uneven hydraulic loading of the impeller induced by two-phase flow. Pitting damage on impeller surfaces may occur where the vapor bubbles violently collapse.

Pump manufacturers publish $NPSH_R$ curves for each individual pump based on performance test data. The pump is tested by establishing stable operation at a point on the head/flow curve for a given speed or impeller trim with deaerated water. While carefully measuring flow and head, the $NPSH_A$ is slowly reduced

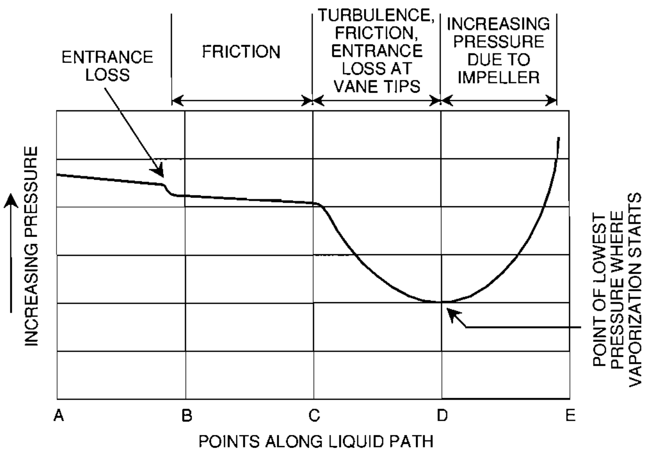

Figure 2. Relative Pressures in the Entrance of a Pump. (Courtesy Flowserve Corporation, 1980)

until vaporization (cavitation) starts. A three percent drop in discharge head has been established by the Hydraulic Institute to indicate cavitation and therefore represents the minimum $NPSH_R$ for the pump at a specific point on the performance curve. At this point, suction pressure, flowrate, liquid temperature, and rpm are recorded, and the information is combined to form the basis for the pump manufacturer's published curve.

Factory NPSH tests are generally conducted using cool water. The vapor pressure of water at 70°F (21°C) is 0.8 ft (0.24 m) of water absolute, or −33.1 ft (−10.1 m) of water gauge. Therefore, very low pressures are needed in the pump to vaporize the liquid. For example, in a pump with an $NPSH_R$ of 2.5 ft (0.76 m) of water absolute, a suction gauge pressure of −30.6 ft (−9.3 m, 0.1 bara) is required to cavitate water in the pump. The significance of low suction pressure relative to a gas seal will be discussed in the next section.

Vapor bubbles formed during cavitation have a similar effect on pump performance as entrained gas or dissolved gas evolved from the liquid. Details of effective cavitation created by dissolved gas and operating guidelines for pumping liquids with dissolved gas content were discussed by Wood, et al. (1998). For this paper, entrained gas is considered the same as vaporized liquid even though some of their physical properties are quite different. Entrained gas can be present at pressures lower than the vapor pressure and may not dissolve into the liquid at higher pressures. The life of a vapor bubble is very short, completely collapsing within a fraction of a second after being formed. Both gas forms degrade pump performance through the displacement of liquid by gas.

DUAL GAS SEALS

Dual gas seal technology has been used effectively to allow zero product emissions, eliminate product contamination, simplify barrier system maintenance, and improve reliability during off-design pump operation. The secret is in using a pressurized inert gas, typically nitrogen, as the barrier fluid. With the gas pressure set higher than the seal chamber pressure, inert gas enters the process instead of process entering the atmosphere.

In today's pump gas seal market, a variety of seal face technologies are used to provide many options and features for particular performance philosophies. While a main categorization is between contacting and noncontacting seal faces, surface features such as spiral grooves, waves, T-slots, and hydropads produce greater product differentiation. Additional gas seal terminology and discussion were compiled by Adams and Parker (1994). The mutual similarity of designs relative to this paper is the use of pressurized gas as the barrier fluid. Testing was performed with the seal illustrated in Figure 3, a noncontacting pusher seal with an advanced pattern spiral groove seal face feature.

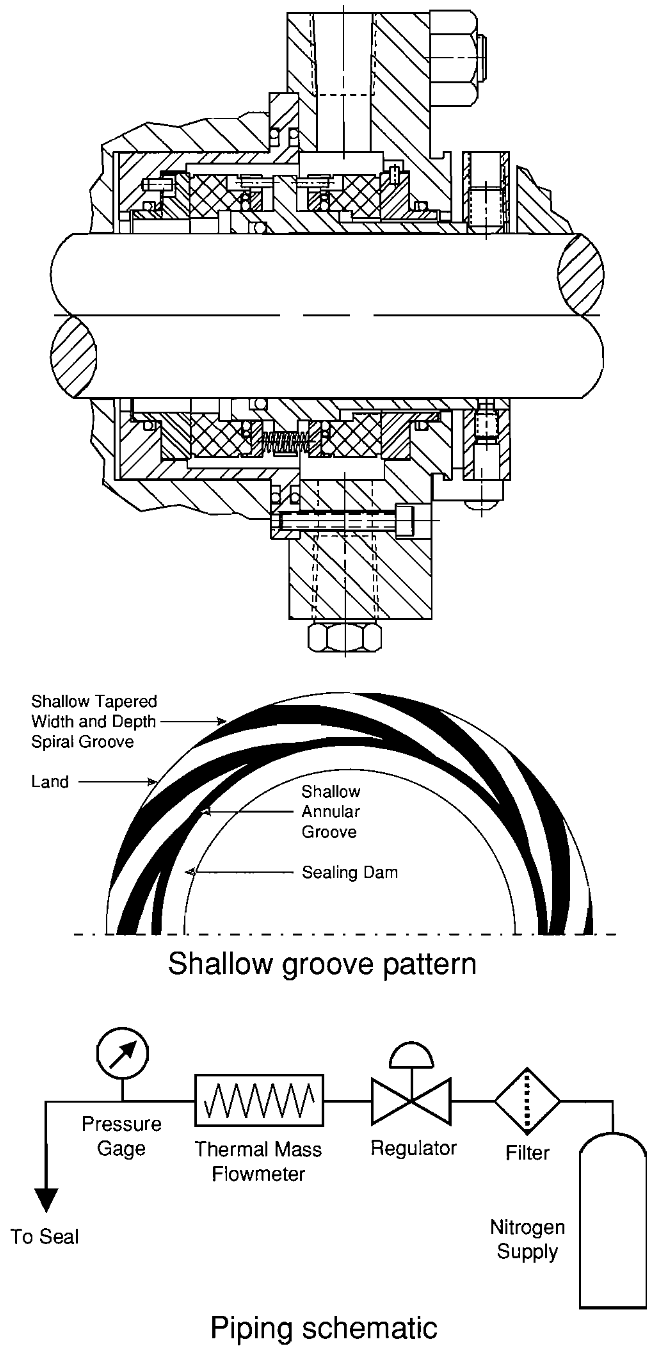

Figure 3. Noncontacting Gas Seal for Pumps Used in Laboratory Testing.

Every dual seal, whether gas or liquid lubricated, leaks a small amount of barrier fluid if the seal is pressurized greater than the seal chamber pressure. In the case of liquid lubricated seals, leakage both into the pump and to atmosphere is typically on the order of a few milliliters per day. With gas lubricated seals, the leakage is a function of many specific operating and seal design parameters and is measured in terms of volume per time, e.g., cubic feet per hour (cfh) or liters per hour (L/h). Gas leakage is measured as it passes into the seal barrier from the pressure source, therefore, the measured flow is for two sets of seal faces. Determining the flow into the pump depends on the relative pressure of the barrier to the seal chamber.

A significant difference between the effects of liquid and gas leakage into the process is the compressibility of the fluid.

Liquid is incompressible and occupies the same amount of volume at any pressure. Gas is compressible and follows an inverse relationship between volume and pressure. As identified by the ideal gas law ($p_1V_1 = p_2V_2$), a decrease in pressure causes an inversely proportional increase in volume for any unit of gas. A sample of gas that has leaked across the seal faces from the high pressure barrier occupies more space at the low pressure suction.

The problem with gas leakage into the pump is not necessarily the amount of flow passing through the gas seal, but the actual volume the gas occupies elsewhere in the system. This is illustrated in Figure 4 with a constant gas flowrate introduced into various suction pressures. After expanding the gas flow from standard pressure conditions to low suction pressure, the gas fraction increases significantly as the pump flow decreases.

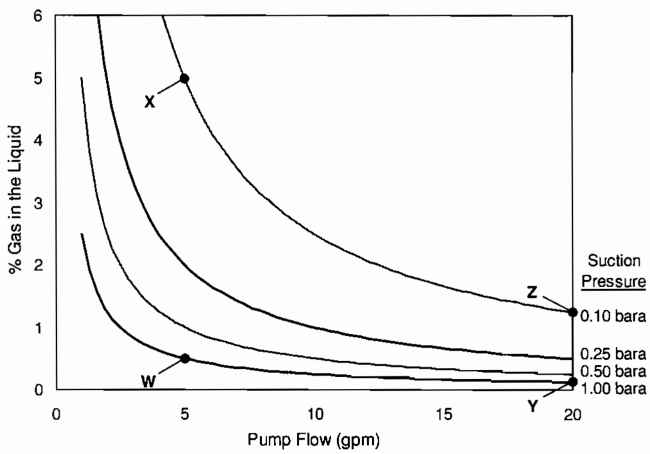

Figure 4. Gas Percentage in a Pump at Constant 0.2 SCFH (5.7 NL/h) Gas Flow.

Referring to the NPSH test example at the end of the preceding section, assume that a gas seal is leaking 0.2 scfh (5.7 NL/h) into the pump so that the data in Figure 4 can be used. At a pump flowrate of 5 gpm (1.1 m^3/h) of an arbitrary liquid, the gas flow represents 0.5 percent of the total volume through the pump without compensating for gas expansion (point W). Now considering the suction pressure of -30.6 ft of water gauge (0.1 bara), the same gas flow becomes five percent of the volume through the pump (point X). Likewise at 20 gpm (4.5 m^3/hour), the 0.2 scfh (5.7 NL/h) gas flow is 0.12 percent of the total pump flow at standard conditions (point Y) and 1.25 percent at -30.6 ft gauge (0.1 bara) suction head (point Z). Remember that the -30.6 ft water gauge (0.1 bara) suction pressure was the point where cavitation begins. If this example had been part of an NPSH test, the five percent gas volume at 5 gpm (1.1 m^3/h) pump flow would have forced the target three percent head loss long before any cavitation. Likewise, at 20 gpm (4.5 m^3/h) pump flow, 1.25 percent gas volume would induce head loss before cavitation. Keep in mind that the lowest pressure in the pump occurs at the vane inlet (Figure 1, location D). This pressure is very difficult to measure directly so the gas expansion example was based on suction pressure. There will be additional expansion as gas passes by the vane inlet.

NPSH TESTING

A series of NPSH tests was run on standard ANSI pumps operating in a closed loop system with ambient temperature water. A schematic of the test loop is shown in Figure 5. Tests were conducted in accordance to procedures sanctioned by the Hydraulic Institute (1994). Each pump was first tested using conventional contacting liquid seals to establish a baseline for comparison. Subsequent tests utilized noncontacting gas seals and contacting liquid seals with an external metered gas flush. During the gas seal and gas flush testing, instrumentation and control were provided by a thermal mass flowmeter and precision regulators.

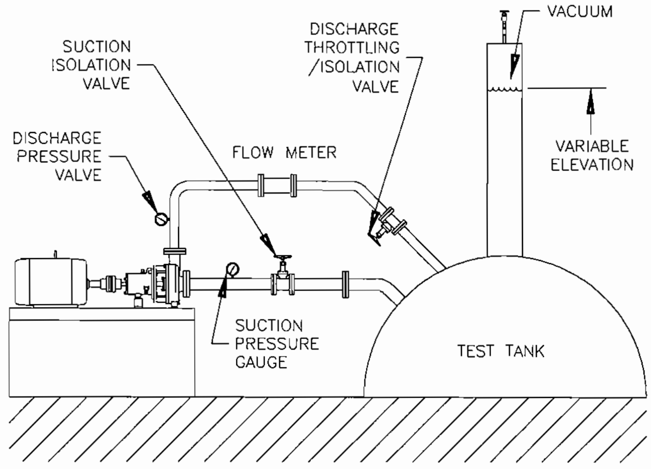

Figure 5. Pump Test Loop Configuration.

Standard tests are conducted maintaining a constant capacity and speed while the suction conditions are varied. Suction head at the pump was changed by drawing a vacuum in the tank and changing the elevation of liquid above the pump. A TDH drop of three percent is the classical indicator that cavitation is present. A three percent head loss was often found to occur at higher suction gauge pressures when gas seals were used.

The NPSH test results on a 1.5×1×8 ANSI pump at 1750 rpm are shown in Figures 6, 7, and 8. Head capacity and the best efficiency point (BEP) are overlaid on Figure 6. The baseline data represent the performance characteristics under normal NPSH test conditions with a liquid seal. The NPSH test goal is to vaporize liquid within the impeller and generate a measurable head loss as a direct result. Any head loss reported in the baseline test data is attributed to vaporization of the liquid or cavitation. With the gas seal, gas leakage across the seal faces expands in the pump during the low pressure phase and causes head loss before the water's vapor pressure can be reached. This head loss is attributed to gas bubbles displacing liquid in the pump as opposed to vapor bubbles in the baseline test. Although the pump test indicates a high $NPSH_R$, in reality, the pump never reaches the point of cavitation.

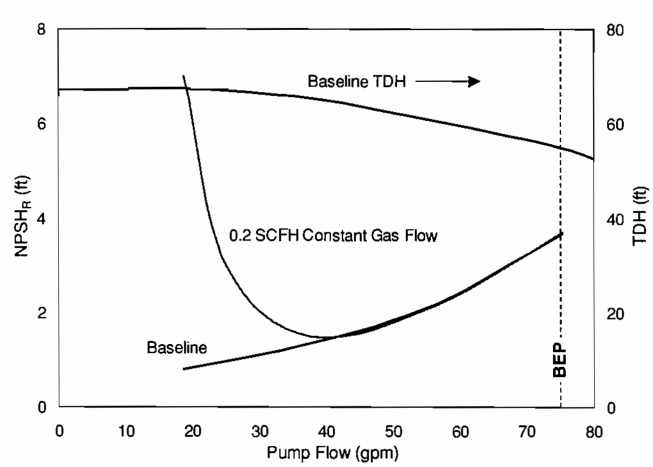

Figure 6. NPSH Requirements for a 1.5×1×8 ANSI Pump at 1750 RPM.

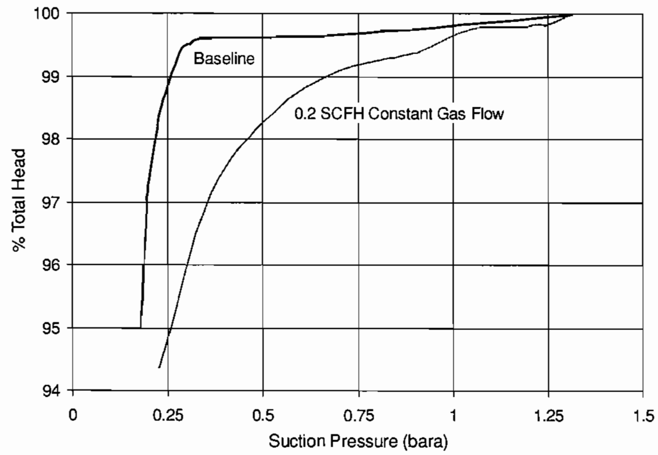

Figure 7. Suction Pressure Requirements for a 1.5×1×8 ANSI Pump at 1750 RPM.

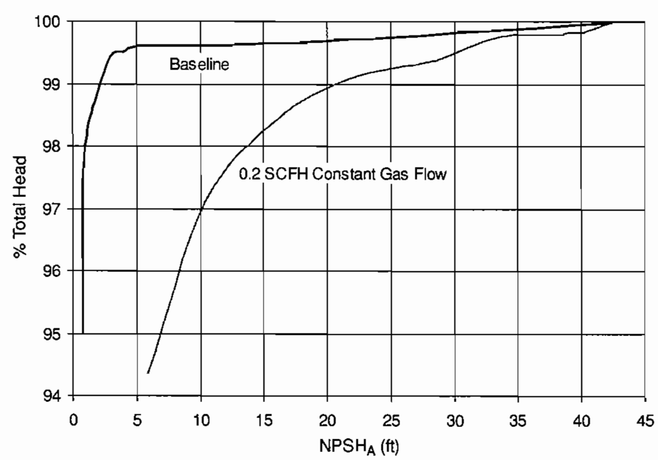

Figure 8. $NPSH_A$ Requirements for a 1.5×1×8 ANSI Pump at 1750 RPM.

When gas bubbles enter the flow streams in the impeller, they expand and contract depending on the local pressure. While this action may affect the pump's ability to develop pressure, none of the damaging effects of cavitation are present. Noise and vibration are not byproducts of TDH reduction caused by gas expansion and contraction. The primary cause of cavitation damage is shock wave energy from the vapor bubble collapsing. This violent collapse does not occur when TDH reductions are caused by gas entrainment from a gas seal.

It is important to note that because of the relatively small gas ingression rates, significant TDH reductions only occur at low suction pressures and at low pump flows. When the gas flow at the local pressure conditions approaches one percent of the pump flow, measurable TDH reductions can be expected. Large populations of conventional pumps successfully handle two-phase liquids with up to four percent gas content. Performance corrections are usually applied when pumping with two percent or more gas content. Increasing the impeller diameter to offset the reduced head is a solution commonly used to pump a two-phase liquid. TDH corrections because of a gas seal would only be recommended if the flow and operating suction head are very low. Figure 4 provides a guide to help define "low" conditions for a constant 0.2 scfh (5.7 NL/h) gas ingression rate. If the actual operating conditions (not NPSH test conditions) with a gas seal produce a gas content of two percent or more, consider increasing the impeller size to offset the TDH loss.

Conventional factory NPSH tests are intended to find the operational point where liquid vaporization occurs within the impeller. It is important to recognize that NPSH tests with a gas seal may not accomplish this task. Instead, an NPSH test with a gas seal will determine the minimum suction head required to suppress gas expansion, thereby limiting the effect of two-phase flow and allowing the pump to operate without head losses. To determine the true NPSH of a pump, a separate test with a conventional contacting face seal may be required. Remember that factory tests with cold water often require very low suction pressures to reach the vapor pressure, whereas actual pump applications are often at much higher suction pressures. If the combination of flow and suction head is high enough, the pump will pass an NPSH test with a gas seal.

APPLICATION GUIDELINES

Field experience has found a low percentage of installations with both low suction head and low flow conditions. Gas entrainment problems attributed to the normal performance of gas seals on pumps have been observed only under these conditions. Negative suction pressure is not a common system design goal, it usually happens as a result of system modifications, process changes, or system upsets. In general, operating a pump at low flowrates is encouraged only if the flowrate is within the normal operating envelope for the pump.

Without entrained gas, common practice recommends as large an NPSH margin as possible, at least 20 percent or 5 ft (1.5 m), whichever is greater. With entrained gas, the same NPSH margin is recommended in addition to an adequate suction head, at least positive pressure. Even a substantial NPSH margin cannot offset gas entrainment losses caused by low suction pressure.

Running a standard NPSH test is not representative of the pump's performance in the actual installation if the expected suction pressure is positive. Instead, the pump manufacturer should provide suction head data for the pump to predict performance and outline an effective NPSH margin.

In existing installations where both the suction pressure and flowrate are low, operating the seal at low differential pressure has proven to help. Before this action is taken, be aware that not all gas seals can tolerate low barrier pressure. The seal design must be capable of generating sufficient film stiffness and overcome secondary seal drag without the positive benefit of high gas pressure. Simply lowering the barrier pressure also increases the risk of pressure reversals if a constant pressure source is used. A differential pressure regulator allows the user to set the barrier pressure at a constant differential over a reference pressure, the seal chamber. Keep in mind that if the seal chamber pressure is negative, the differential pressure must be set relative to atmospheric pressure to maintain positive barrier pressure for the outboard seal.

Gas seals work successfully in applications with heavy cavitation as long as either the suction pressure is positive or the flowrate is adequate. For example, a chemical manufacturer in Louisiana had an installation pumping acrylonitrile where the normal conditions at 50 psig (3.4 barg) suction pressure caused continuous cavitation. The previous liquid barrier dual seals failed every two months because of poor heat transfer conditions at the inboard seal faces. A noncontacting gas seal was installed and has been operating for over one year with no pumping or sealing problems.

FUTURE STUDY

The laboratory research presented in this paper was focused on a narrow set of operating conditions in order to thoroughly examine a particular pumping scenario. The influence of many other conditions could prove to either increase or decrease the effects of gas ingestion in pumps. A sample list of potential variables include:

- Pump size, orientation, type, and speed
- Impeller configuration, size, setting, balance holes, and clearances

- Gas seal configuration and vendor
- Fluid temperature, viscosity, specific gravity, and vapor pressure
- Suction pipe size, piping/flush plans

CONCLUSIONS

- Field experience has shown that in the vast majority of pump gas seal installations, normal gas seal leakage into the pump does not influence pump performance.
- Excessive gas ingestion into both horizontal and vertical centrifugal pumps lowers the total discharge head and increases suction head requirements.
- As pressurized barrier gas leaks into the pump during normal operation, it changes in volume inversely proportional to the local pressure.
- With any manufacturer's pressurized gas seal installed, routine low flowrate NPSH tests can fail if gas expansion at low suction pressure conditions drops the TDH more than three percent, artificially imitating cavitation TDH loss.
- Testing supports classic pump theory that performance degrades as gas ingestion approaches one percent to two percent of the volumetric pump flowrate at the lowest pressure location.
- To determine if gas seal leakage will affect pump performance, calculate the gas percentage in the pump stream at the known suction pressure, pump flow, and gas flow conditions.
- If the combination of pump flow and suction head is high enough to maintain a low gas percentage, the pump with a gas seal will pass an NPSH test.
- Larger diameter impellers can be used to compensate for head reduced by a gas seal.

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