APPLICATION GUIDELINES FOR PUMPING LIQUIDS THAT HAVE A LARGE DISSOLVED GAS CONTENT

by

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ABSTRACT

A case history is presented pertaining to five (four operating and one spare), 150 hp sealless pumps operating in parallel in a closed loop $-65^{\circ}F$ chilled liquid system. The pumping system is designed to use a liquid refrigerant as a heat transfer media, due to its desirable pumping characteristics at low temperatures.

All five pumps experienced severe damage and repeated failures of the process lubricated bearings during startup of the system. Investigation and analysis of the system revealed two major problem areas in pumping liquids with dissolved gases.

- The standard textbook calculation for $NPSH_A$ may not accurately predict the effective $NPSH_A$ for a liquid with high dissolved gas content.

- Two phase flow (gas and liquid) through sealless pumps may lead to product lubricated bearing failures.

The authors review a method for calculating the effective $NPSH_A$ when dissolved gases are present. The authors also present

appropriate system design considerations intended to assist in minimizing the amount of gas that enters, or is released within, a pump to prevent major equipment damage.

Also discussed are limitations imposed on the chilling capacity of the case history pumping system due to heat rejected to the chilled liquid as it passed through the magnetic drive pumps.

INTRODUCTION

Description of Pumping System

The pumping system consists of four operating sealless pumps operating in parallel in a closed loop −65°F chilled liquid system, as shown in Figure 1. The purpose of the system is to transfer chilled liquid to the condenser for use in cooling the process liquid.

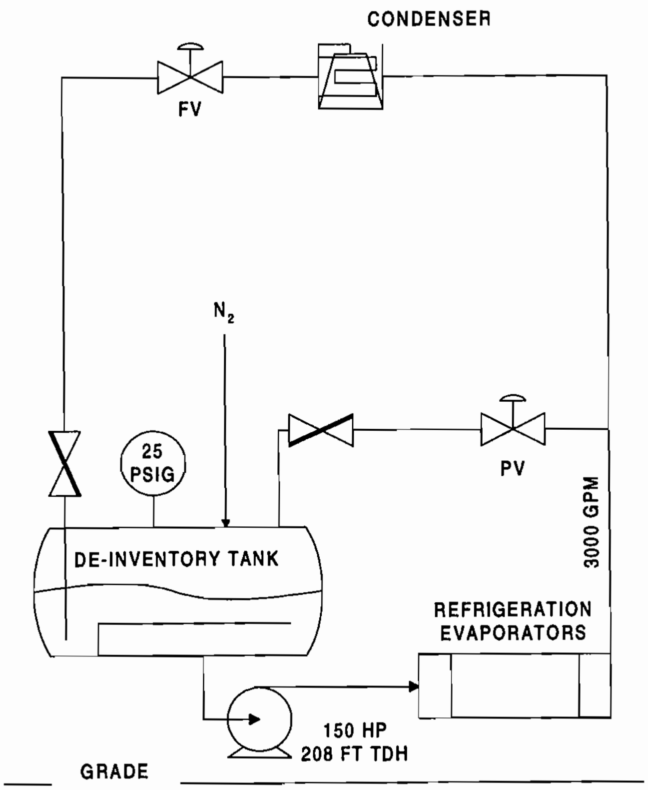

Figure 1. Case History Chilled Liquid Pumping System.

Although no refrigeration cycle is taking place in the confines of the pumping system, a liquid refrigerant is used as the pumping media. Liquid refrigerants are often used as a heat transfer media down to −110°F for process cooling, due to their desirable low temperature pumping characteristics such as a water-like viscosity.

In the pumping system, the liquid is fed into the pumps from a deinventory tank at grade level. A 25 psig nitrogen gas pad is maintained in the deinventory tank vapor space. In the past, suction vessels in these pumping systems were elevated to provide adequate $NPSH_A$ for proper operation of the pump. However, due to increased economic pressures, new suction vessels are now mounted at grade and pressurized with gas to achieve $NPSH_A$ for the pump. After the liquid exits the suction vessel, it flows through the pump suction piping, which follows good engineering design practice to provide equal flow to all the pumps (note that the pumps are operating in parallel). The elbows and tee branches in the suction line are also kept to a minimum to avoid liquid prerotation and excessive turbulence. After passing through the pumps, the liquid flows into two refrigeration evaporators. Some liquid is bypassed back to the deinventory tank by the use of a pressure control valve, while the main flow continues on to 11 process condensers. The flow through the condensers is controlled by a flow control valve. The flow then returns to the deinventory tank and enters below the liquid level via a dip tube to avoid gas entrainment. A baffle system in the deinventory tank also acts as an additional measure to avoid gas entrainment.

Description of Pump

The pump applied to this service is a magnetic drive pump similar to Figure 2. The reasons magnetic drive pumps were chosen for this service were environmental regulations and liquid cost. At the time of pump selection, it was felt that magnetic drive pumps would provide the best means for maintaining a leak-free environment.

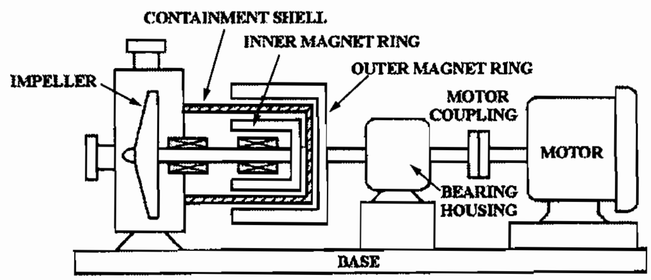

Figure 2. Construction of a Magnetic-Driven Sealless Pump.

In the magnetic drive pump, the pump shaft/impeller assembly is driven by permanent magnets external to the containment shell. Within the containment shell, another set of permanent magnets is attached to the pump shaft. The pump shaft bearings (Figure 3) are lubricated by the liquid being pumped. These bearings are typically constructed of hard materials such as silicon carbide. About three to five percent of the fluid being pumped is directed through a recirculation path starting at a high pressure point in the pump casing, entering the bearing area, and returning to the casing behind the impeller. Because the fluid moves from a high pressure point in the casing and returns to a low pressure point, the liquid is driven by the pressure/velocity differential. The liquid moving through this recirculation path both lubricates the bearings and carries away the heat generated by hydraulic friction, bearing friction, and eddy-currents in the containment shell.

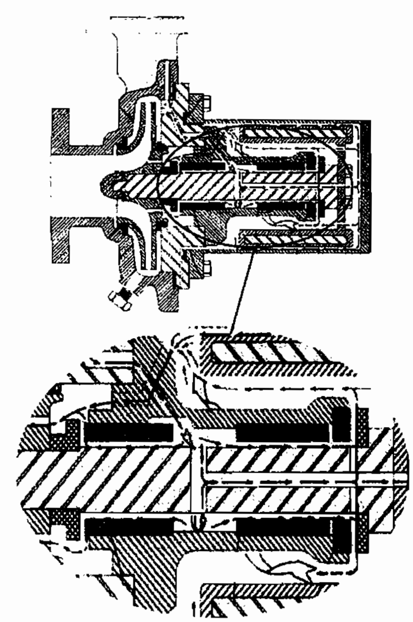

Figure 3. Magnetic Drive Pump Shaft Bearings with Lubrication Path Shown.

Description of Problem

During startup and commissioning of these pumps, severe damage and repeated failures of the process lubricated bearings occurred. These process lubricated journal and thrust bearing failures occurred due to the lack of lubrication caused by the two phase flow in the bearings. Obviously, this resulted in lost time and production. No additional damage was done to the pump.

An additional problem was encountered with marginal refrigeration capacity due to inefficiency of the magnetic drive pumps. The original design predicted a larger margin of cooling capability design for these evaporators that did not reflect in the actual operation.

FAILURE ANALYSIS

Bearing Failures

Dry running of product lubricated bearings in sealless pumps is the most common cause of failure for these types of pumps. A typical cause of dry run failures can be as obvious as the attempted startup of a pump prior to filling it with liquid. Other causes of dry running relate to misapplication, such as insufficient NPSH, pressure-temperature transients, entrained air and gases, or blocked lubrication and circulation paths.

After verifying that the pump was filled with liquid and the pump's lubrication paths were open, the authors narrowed down the possible problems to insufficient NPSH and entrained gas.

Previous experience with pumping the same liquid revealed that entrained gas could be a serious problem. A similar application at another plant site utilized an elevated deinventory tank (which was not pressurized with gas), as shown in Figure 4, with magnetic drive pumps similar to the case history pumps. These pumps experienced similar process lubricated bearing failures. To alleviate this problem, a baffle and dip tube were installed in the deinventory tank, so that the gas could liberate itself from the liquid before entering the suction pipe. The installation of the dip tube and baffle system eliminated the pump failures. Knowledge of the failures due to the entrained gas on this application was known prior to the design of the case history application. Therefore, prior to the first startup, a dip tube and baffle system was installed on the case history deinventory tank to avoid entrainment of nitrogen into the liquid refrigerant.

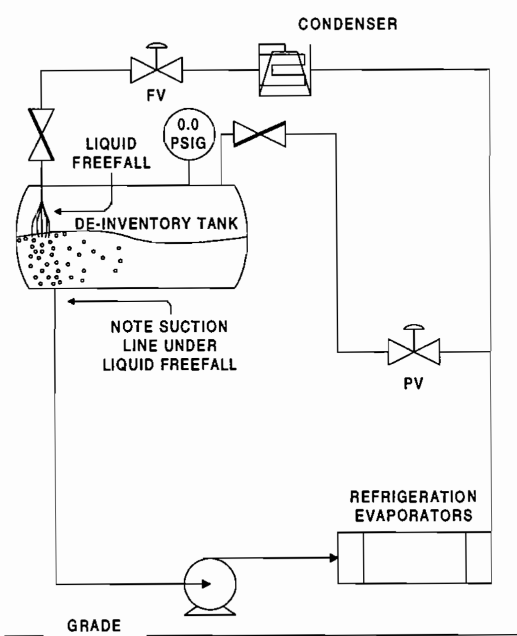

Figure 4. Application Showing Previous Experience with Gas Entrainment Pumping Same Liquid as Case History Application. (No nitrogen blanket on vessel).

After feeling confident that the entrained gas problem did not exist on the case history application due to the extensive design measures taken, attention turned toward the NPSH margin. Calculations revealed that $NPSH_A$ = 61 ft using the standard handbook calculation as follows:

$$h_{sv} = h_{sa} - h_{vpa} \tag{1}$$

where:

h_{sv} = available net positive suction head, in ft of liquid
h_{sa} = total suction head, in ft of liquid, absolute
h_{vpa} = vapor pressure of liquid at suction nozzle, in ft of liquid, absolute

The manufacturer's published $NPSH_R$ was 10 ft, making the overall NPSH margin 51 ft. Note that the magnetic drive pumps were not tested for $NPSH_R$ as it was determined that the NPSH margin was more than adequate.

After numerous failures, the cause of the process bearing failures could not be determined without losing more production time. It was decided to remove the magnetic drive pumps from service and install standard double mechanical sealed centrifugal pumps. The replacement pumps would still provide a leakfree environment, but would not be subjected to process lubricated bearing failures. The replacement pumps had an $NPSH_R$ of 11 ft, which was confirmed by manufacturer testing.

The double mechanical sealed centrifugal pumps operated for three days without any noticeable problems. However, audible crackling noises in and around the pump suction started to occur after this initial period. When this cavitation began, it was noted that a differential pressure gauge reading the pressure drop through a suction strainer had increased to 1.75 psi from its initial reading three days prior of 0.75 psi. It did not make sense at the time that cavitation would appear with such a large NPSH margin, which was now 50 ft with the replacement pumps. The strainer was cleaned and replaced with a subsequent audible decrease in pump cavitation. At this point, this was a clue that some parameter must be marginal to cause pump cavitation with such a small increase in the suction side frictional drop. This was strong enough evidence to convince the authors that the pump was operating in the breakdown area of the NPSH curve (Figure 5).

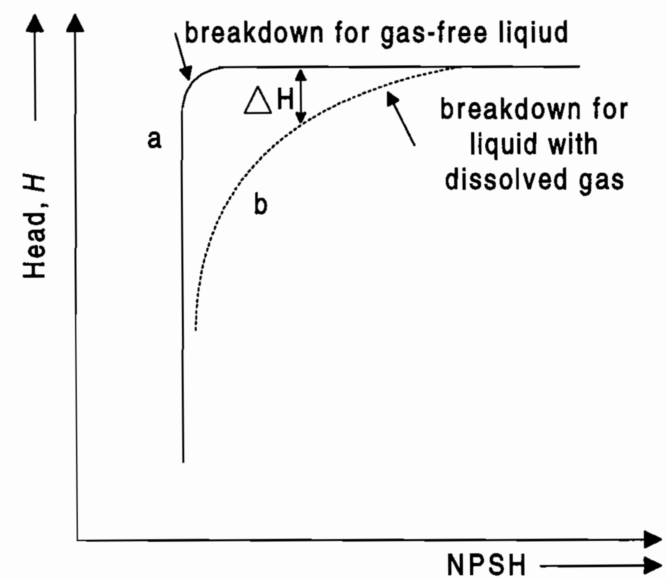

Figure 5. NPSH Breakdown Curve.

It can be noted in Figure 5 that the gas-free liquid in *curve a* has a much sharper breakdown than does the liquid with dissolved gas in *curve b*. The solubility of any gas in a liquid decreases with reduced pressure. Low available NPSH, therefore, liberates a major part of the gases dissolved in the liquid into the low pressure zones of the suction line and impeller. This gas reduces the effective suction area of the pump that, in turn, decreases the head developed by the pump. Also, when dissolved gas is coming out of solution, a more gradual head drop-off, such as in *curve b* in Figure 5, may be seen than when pumping a gas-free liquid. It should be stated that the gradual drop-off seen in *curve b* can also be observed without gas present when the pump operating capacity is less than the best efficiency point capacity.

It can also be noted from Figure 5 that the breakdown of *curve b* begins to happen at a higher NPSH than the gas-free liquid of *curve a*. This phenomenon causes the NPSH needed to maintain reliable pump operation to be higher when dissolved gas is present.

In the case history application, the authors felt that a phenomenon that causes the effective $NPSH_A$ to be lower than anticipated must be occurring. Therefore, the next step was to understand this phenomenon and investigate the possibility of dissolved gas, specifically dissolved nitrogen, coming out of solution and reducing the effective $NPSH_A$.

Gas Solubility in Liquid

It is quite well known that gases are soluble in liquids such as water. The ability of fish to breathe is dependent upon this fact. All carbonated beverages have their characteristic properties because of the solubility of carbon dioxide in water. It is known that a soda left open in a warm room will get flat quickly. All of these observations are dependent upon the solubility of gases in liquid.

Henry's Law

Solids and liquids have some well-defined solubility in liquid solvents. The solubility of gases, on the other hand, depends on the pressure, according to Henry's Law. This says that as we increase the pressure of the gas P_b (or more correctly, the partial pressure) the amount of gas dissolved (the solubility S_b) increases, depending on the Henry's law constant k_H. Some typical constants are shown in Table 1.

A good example to understand the use of Henry's law is to examine a phenomenon called the bends, which affects deep-sea divers. Air is 78 percent nitrogen, which means that at ambient (atmospheric) pressure the pressure of nitrogen in the atmosphere is 0.78 × 760 mm Hg = 590 mm Hg. With the Henry's law constant in Table 1, we can calculate that at ambient pressure, the concentration of N_2 in water is:

$$S_b = 590 \text{ mm Hg} \times 8.42 \times 10^{-7} \text{ mol/(L mm Hg)} = 4.97 \times 10^{-4} \text{ mol/L} \quad (2)$$

Table 1. Gas Solubility at Ambient Pressure in Water.

Gas	k_H, mol/(L·mm Hg)
N_2	8.42×10^{-7}
O_2	1.66×10^{-6}
CO_2	4.4×10^{-5}

Since one mole of nitrogen at 37°C occupies about 25.4 L, if there are 4.97×10^{-4} mol of N_2, that corresponds to 1.26×10^{-2} L = 12.6 mL of nitrogen dissolved per liter of water. Blood is mostly water, so this is the amount of nitrogen dissolved in blood under normal conditions.

Under the ocean, the pressure increases about one atmosphere every 10 m, or 33 ft. So if we dive to a depth of 155 ft, the pressure is now:

$$P_b = 1 \text{ atm} + 155 \text{ ft/(33 ft/atm)} = 6 \text{ atm} \quad (3)$$

If the diver is breathing air, then his or her blood will dissolve 6 × 12.6 mL = 76 mL N_2/L water, six times more than at atmospheric pressure.

Effect of Temperature on Gas Solubility

As temperature increases, the solubility of gases decreases, as shown in Figure 6. This is the simple explanation of why an open soda kept cold will keep its fizz longer than one allowed to warm up.

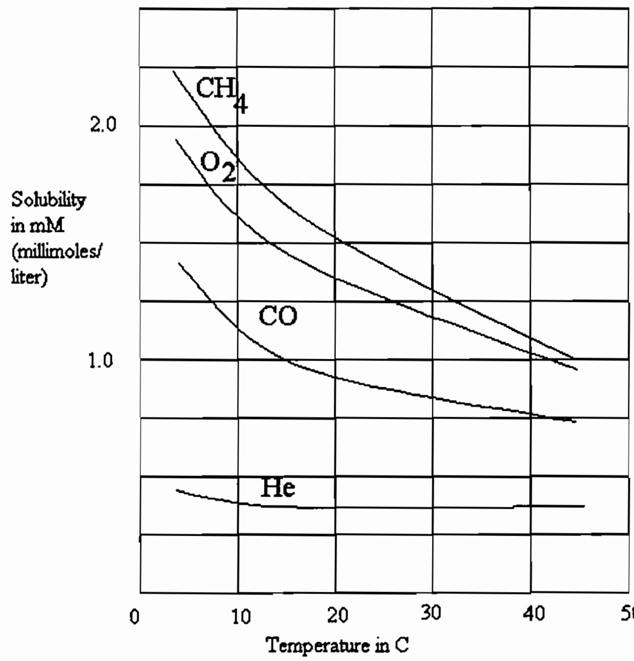

Figure 6. Gas Solubilities in Water as a Function of Temperature.

Dissolved Gas and Its Effect on NPSH

In order to prevent cavitation in the pump, the pump suction pressure must be higher than the fluid's vapor pressure at the pumping temperature. As stated earlier by Equation (1), this is the definition of $NPSH_A$. For cases in which the liquid contains no dissolved gases, the vapor-pressure determination is determined by Equation (1).

Stepanoff (1993) indicates that the presence of gases in the liquid does not affect the validity of Equation (1) except that, according to Dalton's law of partial pressures, the vapor will behave as if it occupied the voids alone, and vaporization will begin at an absolute pressure higher than its normal boiling point and corresponding to the existing temperature. However, with dissolved gases, the situation is more complicated, because vapor pressure data are usually not available.

Consequently, when the liquid enters the eye of the impeller, some of the dissolved gas is liberated due to the reduced pressure. The pump inlet pressure drop in gas saturated liquid pumping systems may cause gas to be released, reducing the calculated NPSH margin to a value such that heavy cavitation is experienced (Figures 7, 8, and 9). Relatively small amounts of gas experienced during two phase flow can dramatically affect hydraulic performance and reliability.

In sealless pumps, a gas saturated liquid may release gas from solution as the liquid passes through the process lubricated bearings and the static pressure drops. The gas release in the process lubricated bearings causes the bearings to run dry and eventually leads to failure. The authors feel that this was the cause for failure in the magnetic drive pumps that were initially installed.

So the question becomes—how does one approximate $NPSH_A$ if dissolved gas is present. Accordingly, if the engineer plays it safe

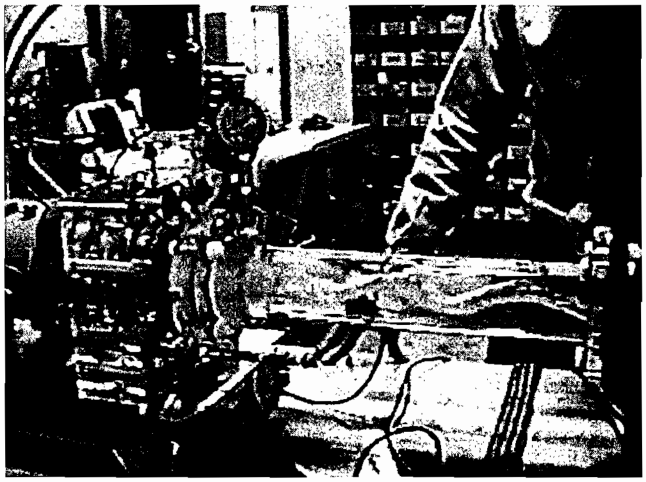

Figure 7. Test Setup with Clear Plastic Suction Pipe to Show Gas Coming Out of Solution—Standing Vortex Shown in Figure.

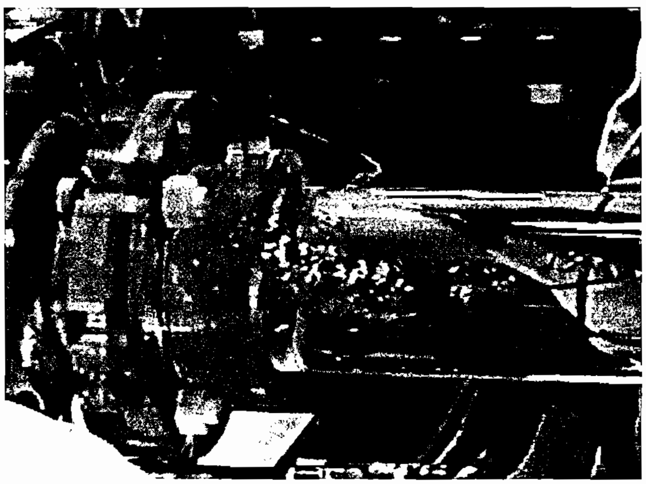

Figure 8. Dissolved Gas Coming Out of Solution at Pump Suction with Pump Shown on Left.

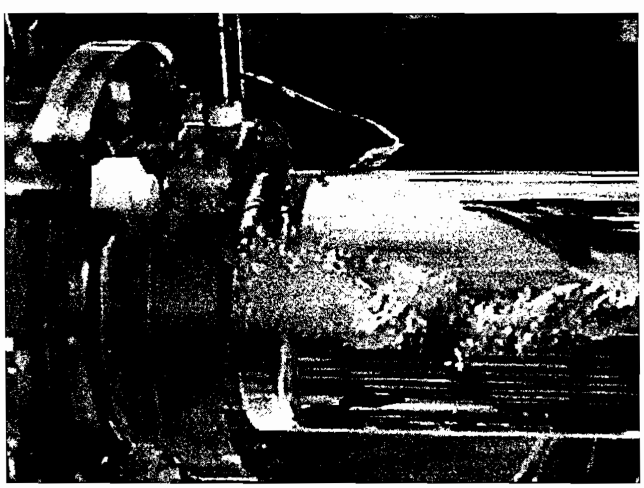

Figure 9. Dissolved Gas Coming Out of Solution.

and assumes (for lack of data) that the process pressure of the liquid-gas solution in the suction vessel is its vapor pressure, and the engineer consequently designs enough vessel elevation to fully compensate for friction losses between the vessel and pump, this elevation may be greater than necessary. Because of the elevation requirement, this in turn would most likely be an uneconomical solution. On the other hand, if the engineer ignores the presence of the gas and bases the vessel elevation on the vapor pressure of the vapor-free liquid, the vessel may not be high enough.

For an economical suction vessel design, an effective vapor pressure that lies between the process pressure and the liquid vapor pressure should be used in calculating $NPSH_A$. Chen (1993) offers an analytic way to calculate the effective vapor pressure of a gas-saturated liquid that can then be used for calculating the effective $NPSH_A$ as follows:

1. Determine the liquid density ρ_L and the gas density ρ_{Go} at the process conditions in the pump suction vessel. The liquid density is assumed to be constant (because only a small amount of gas is flashing). The gas density is at the temperature and pressure (total pressure, not partial pressure) in the suction vessel.

2. Determine W_o, the weight fraction of dissolved gas at the pump suction vessel conditions, from simulation results, solubility data or Henry's law.

3. Use the results of Steps 1 and 2 to calculate the solubility factor S by Equation (4).

$$S = W_o \rho_L / \rho_{Go} \tag{4}$$

4. Select a design value for the tolerable volume fraction f of vapor at the pump eye: 0.025 (2.5 percent), 0.030 (3 percent), or some other value suggested by the pump vendor. Centrifugal pumps can tolerate a small amount of vapor, generally around three percent by volume. Note that dissolved gas coming out of solution has essentially the same effect on the pump as vapor for the pump. Therefore, it can also be said that a centrifugal pump can generally tolerate around three percent by volume of dissolved gas coming out of solution.

5. Determine vapor pressure of the pure liquid P_v at the process temperature and calculate R by Equation (5).

$$R = P_v / P_o \tag{5}$$

6. Determine the saturation factor a (between zero and one; use one if the factor is not known) and calculate the saturation coefficient b by Equation (6).

$$b = a + (1 - a)\, P_v / Po \tag{6}$$

7. Use the values of S and f obtained above to calculate the parameter N by Equation (7).

$$N = [f\,(1 - f)] / S \tag{7}$$

8. Use the results of Steps 5, 6, and 7 to calculate coefficients A, B, and C by Equations (8), (9), and (10).

$$A = N\,(1 - R) + 1 \tag{8}$$

$$B = 2NR\,(1 - R) + b \tag{9}$$

$$C = NR^2\,(1 - R) \tag{10}$$

9. Calculate y by Equation (11), and then the effective vapor pressure P_e by Equation (12).

$$y = \frac{B + \sqrt{B^2 - 4AC}}{2A} \qquad (11)$$

$$P_e = yP_o \qquad (12)$$

Calculated results for $f = 0.030$ appear in Figure 10. It can be seen from the curves that the difference between the process pressure and the effective vapor pressure becomes significant only when the gas solubility is low.

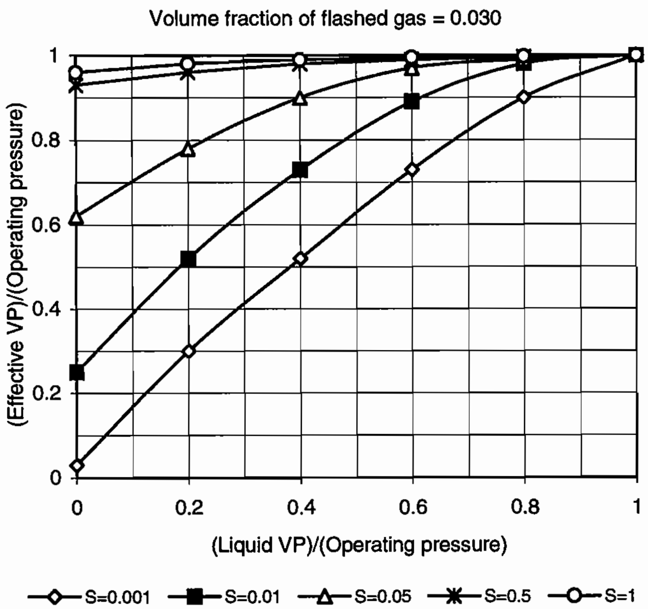

Figure 10. Calculation of Effective Vapor Pressure for f = 0.030 and Various Listed Solubility Factors.

Case History $NPSH_A$ Calculation

When a blanket of inert gas is employed over a liquid to maintain a desired process pressure above the liquid vapor pressure, some of the gas will dissolve in the liquid. As the liquid enters the pump, part of the dissolved gas will flash if the liquid static head is insufficient to overcome the pressure drop between the tank and the pump. This may cause severe cavitation if the amount of flashed gas is substantial. In the case history application, it was desired to not exceed a maximum level of 2.5 percent dissolved gas coming out of solution at the pump suction. The given input for the case history is shown in Table 2.

Table 2. Inputs for Case History Effective $NPSH_A$ Calculation.

P_o	40.0 psia
T_o	−54.0°C
Suction Piping Pressure Loss	0.50 psi
Suction Strainer Pressure Loss	1.75 psi
ρ_{Go}	0.2597 lb/ft^3
ρ_L	100.0 lb/ft^3
Sp. Gravity	1.60
f	0.025
W_o	9.52E-04
P_v	0.25 psia
$NPSH_R$	10 ft
h	7.0 ft

Using Chen's method and assuming a saturated solution, both *a* and *b* have a value of one. The calculations for the case history effective vapor pressure are shown in Table 3.

Table 3. Calculations for Effective Vapor Pressure for Case History Application.

S	0.3667
R	0.01
a	1.0
b	1.0
N	0.0699
A	1.0695
B	1.0009
C	0.0000
y	0.9358
P_e	37.43 psia

The equation for $NPSH_A$ is:

$$NPSH_A = (P_o - P_v - \text{friction loss}), \text{ in feet of liquid} + h \qquad (13)$$

Let *h* stand for the suction-side static head, in ft. In this case, *h* equals 7 ft. Computing the $NPSH_A$:

• Ignoring the dissolved gas:

$$NPSH_A = (40 - 0.25 - 2.25) \text{ psi} \times \frac{2.31}{1.60} + h \qquad (14)$$

$$NPSH_A = (54.0 + h) \text{ ft} = 61.0 \text{ ft}$$

• Taking dissolved gas into account (i.e., using the effective vapor pressure):

$$NPSH_A = (40 - 37.43 - 2.25) \text{ psi} \times \frac{2.31}{1.60} + h \qquad (15)$$

$$NPSH_A = (0.5 + h) \text{ ft} = 7.5 \text{ ft}$$

• NPSH based on operating pressure of pump suction vessel:

$$NPSH_A = (40 - 40 - 2.25) \text{ psi} \times \frac{2.31}{1.60} + h \qquad (16)$$

$$NPSH_A = (-3.2 + h) \text{ ft} = 3.8 \text{ ft}$$

If the required NPSH is 10 ft, the required static heads above pump centerline are as follows:

• Based on ignoring the dissolved gas: no additional positive static head is required.

• Based on taking dissolved gas into account: an additional static head of 2.5 ft is required, plus 5 ft for safety margin, for a total of 7.5 ft additional required static head.

• Based on the pump suction vessel pressure: an additional static head of 6.2 ft is required, plus 5 ft for safety margin, for a total of 11.2 ft additional required static head.

The result obtained by using liquid vapor pressure in the $NPSH_A$ calculation, namely that no positive static head is required, is too optimistic. However, the calculation taking the dissolved gas into account concludes that inadequate $NPSH_A$ existed in the system. It also shows that the $NPSH_A$ was close to the $NPSH_R$. As previously noted, audible cavitation was much more evident as the suction strainer grew dirty, signifying that the pumps were on the breakdown area of the NPSH curve.

The above strongly suggests that the Chen method was fairly accurate in predicting the $NPSH_A$ for this application.

Also, the result based on using the process pressure is conservative, though not far off. The vessel support height thus calculated is unnecessarily high by about 3.7 ft. However, the authors do not feel an adequate design exists when $NPSH_R = NPSH_A$. In this application, an NPSH margin of 5 ft should be employed.

Heat Generation

Another problem encountered in the case history application, when the magnetic drive pumps were installed, was excessive heat load on the chillers. This excessive heat load came from the inefficiency of the magnetic drive pumps and caused the chillers to run close to their maximum capacity.

The excessive heat load would not have been seen as a large problem if the system were a heating unit. However, since the objective was to cool the liquid, the problem was highly evident, as most of the heat from the inefficiency of the pumps went directly into the liquid.

In the application, the overall efficiency of the magnetic drive pumps was 43 percent. The efficiency of the standard centrifugal double mechanical sealed pumps that replaced these pumps was 69 percent. This efficiency difference equated to a delta of 47 hp per pump and, since there were four operating pumps, the total pump hp that the refrigeration evaporator had as an additional heat load was 188 hp, assuming 100 percent of the inefficiency went into heating the process fluid (typically, this might be closer to 85 to 90 percent). This 188 hp additional head load equated to 40 tons of chiller capacity. The chiller needed 4 hp for every ton of refrigeration it produced and, thus, the chiller required 160 hp to make up for the 40 ton heat load. Taking the 188 hp of pump drive inefficiency and adding it to the 160 hp refrigeration evaporator power requirement, equates to 348 hp of power losses. At five cents per kWh and 5870 hours per year utility, the 348 hp energy wasted by the inefficiency of the magnetic drive pumps was approximated at costing $115,000 per year.

Since the solution to the bearing failure problems was to use a standard mechanical sealed centrifugal pump, the heat generation problem disappeared, as did the power requirement for the pumps.

In this particular application, there was a lesson learned about the inefficiency of sealless pumps. The objective of this pumping system was to chill the process fluid, making the performance of the originally installed magnetic drives contradictory due to the heat the pumps added to the system. The magnetic drive pumps installed were better intended for a high temperature heat transfer application, as they would have kept the process fluid hot and reduced the load on a heat source such as a boiler. However, certain magnetic pump designs exist that have composite containment shells that have no eddy currents. The overall efficiency of these magnetic drive pumps is two to three points less than the equivalent pump with a mechanical seal, when pumping low viscosity liquids.

The lesson is that the use of a sealless pump should be weighed versus the use of a standard sealed pump, due to the heat added to the system in cryogenic applications.

The published performance curves of sealless pumps may not readily identify the portion of the power that enters the hydraulic system as heat energy. Most users are familiar with the performance curves provided on the more standard shaft sealed pumps and may not have adequate information to properly account for the difference caused by the drive inefficiency (i.e., magnetic coupling losses). Such an oversight might cause the pumping system designer to underdesign a component in the system (such as a refrigeration evaporator). In fact, in the case history application, if it were not for asking extra questions about the pumps, the efficiency issue would not have been uncovered, and the refrigeration evaporators might well have been mistakenly underdesigned. The ASME B73.3 Sealless Pump Standard will address this issue. The identification of the true overall pump efficiency is especially critical for closed loop systems, circulating systems, and systems with fluids with low specific heat and high vapor pressure.

Heat Generation Effect on Gas Released in the Pump

In sealless pumps, a gas saturated liquid may release gas from solution as the liquid passes through the process lubricated bearings and rotor to stator gap as the temperature increases. A second problem that showed up in the magnetic drive pumps also had to do with heat generation. A temperature monitor that measured the containment shell temperature was displaying much higher temperatures than expected. The liquid temperature at the pump suction was −65°F, whereas the reading on the containment shell was 46°F, giving a temperature rise of 111°F. The pump manufacturer had estimated a temperature rise of approximately 48°F, which meant that the shell temperature probe reading should have been about −17°F. The authors hypothesized that the explanation for the higher than predicted temperature rise could rest within at least three possibilities.

The first hypothesis is that gas came out of solution in the containment shell due to the localized heating of the process liquid. In sealless pumps, a gas saturated liquid may release gas from solution as the liquid passes through the rotor to stator gap, and the temperature increases. The released gas reduced the heat transfer capability from the containment shell to the liquid-gas mixture and, thus, the higher than expected temperature reading was experienced.

The second hypothesis is similar to the first in that gas came out of solution in the containment shell. However, in this hypothesis, enough gas was released to completely stop the flow through the pump cooling path to vapor lock the path. On one occasion, one pump's containment shell appeared to have expanded and contacted the outer magnet, thus rupturing the shell. The expansion of the shell may have occurred due to a complete lack of cooling, which would support the theory of the cooling path being vapor bound.

The third hypothesis is that the pump supplier did not account for the fluid properties and their effect on temperature rise in the pump. One of the properties of the liquid refrigerant was a low specific heat, which would cause a high temperature rise (Figure 11).

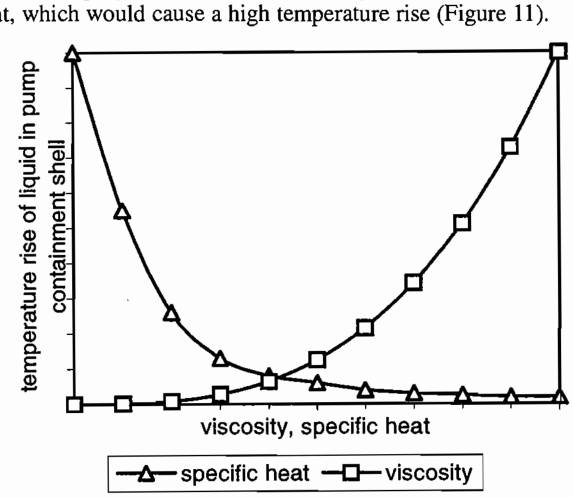

Figure 11. Relative Effects of Specific Heat and Viscosity on Containment Shell Temperature Rise.

CONCLUSIONS AND RECOMMENDATIONS

- For single component gas-liquid systems (i.e., where a single gas exists in a blanket system), Chen's (1993) method provided excellent results for calculating the effective $NPSH_A$ in the case history application.

- If test data are not available when using Chen's (1993) method for making effective $NPSH_A$ calculations, it is recommended to assume the liquid is fully saturated with blanket gas ($a = b = 1$).

- In a water-air system, Chen (1993) recommends that pumping water from a tank that is higher than the pump, usually offers no NPSH problem. However, if water is pumped from a level below the pump, the effect of dissolved air on pump lifting capability should be investigated.

- For a multicomponent system (i.e., liquid saturated with a gas mixture containing several major components), Chen (1993) notes that the effective $NPSH_A$ method is not reliable. Process simulation is the only trustworthy method for obtaining the effective vapor pressure for this type of system.

- To the authors' knowledge, none of the network flow analysis software packages on the market today take into account dissolved gases in the fluid. Make sure this is not overlooked when using these software packages.

- As a guideline, the $NPSH_A$ should exceed the $NPSH_R$ by a minimum of 5 ft, or be equal to 1.35 times the $NPSH_R$, whichever is the greater value. For example, for an $NPSH_R$ of 10 ft, the $NPSH_A$ should be a minimum of 15 ft.

- Avoid designs where the possibility of gas entrainment exists due to liquid freefall or vortexing. The gas entrained in these situations can be just as harmful to the pump as dissolved gas.

- For systems that have a high dissolved gas content, give careful consideration to the pump type to be used. In general, sealed pumps are much less susceptible to dissolved gas than are sealless pumps.

- For systems that have high dissolved gas content, use caution in evaluating readings from transducers such as temperature or pressure. Dissolved gas may come out of solution in close proximity to the transducer, causing erroneous readings or other phenomena to occur.

- In chilled liquid applications, careful consideration should be given to the evaluation of sealed versus sealless pumps. The additional heat generated by sealless pumps due to their inefficiency as compared with sealed pumps may require a much larger refrigeration design to be required, not to mention the increased energy requirements.

NOMENCLATURE

A	= Coefficient in Equation (11) as defined in Equation (8)
a	= Saturation factor [(dissolved gas at unsaturated condition)/(dissolved gas at saturated condition)]
B	= Coefficient in Equation (11) as defined in Equation (9)
b	= Saturation coefficient as defined in Equation (6)
C	= Coefficient in Equation (11) as defined in Equation (10)
f	= Volume fraction of vapor at pump eye [(volume occupied by flashed gas and vaporized liquid)/(total volume)]
h	= Suction static head, ft of liquid
k_H	= Henry's law constant
N	= Parameter defined in Equation (7)
NPSH	= Net positive suction head, ft of liquid
$NPSH_A$	= Net positive suction head available, ft of liquid
$NPSH_R$	= Net positive suction head required, ft of liquid
P	= Pressure, psia
P_b	= Partial pressure
P_e	= Effective vapor pressure corresponding to an allowable f, psia
P_o	= Operating pressure of pump suction vessel, psia
P_v	= Vapor pressure of liquid (without dissolved gas), psia
R	= Ratio of P_v and P_o
S	= Solubility factor defined in Equation (4)
S_b	= Solubility
T_o	= Temperature of suction vessel, °F
W_a	= Weight fraction of dissolved gas at total pressure of P_o
W_o	= Weight fraction of dissolved gas at total pressure of P_o
y	= Ratio of P_e and P_o
ρ_{Go}	= Density of gas at pump suction vessel pressure (P_o) and temperature, lb/ft^3
ρ_L	= Liquid density, lb/ft^3

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