REDUCING ANSI PUMP MAINTENANCE COSTS THROUGH COMPONENT UPGRADES

by

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ABSTRACT

Standard ANSI pumps are adequate for most services. In installations where they do not give adequate service, however, upgrades are available to improve reliability. These should be selectively applied to obtain the maximum reliability at the lowest cost. Traditional component modifications, as well as more recently available upgrades, are reviewed for the two highest failure pump components: mechanical seals and bearings. Analytical and test data are presented to validate the newer component upgrades from both a reliability and cost effectiveness viewpoint.

INTRODUCTION

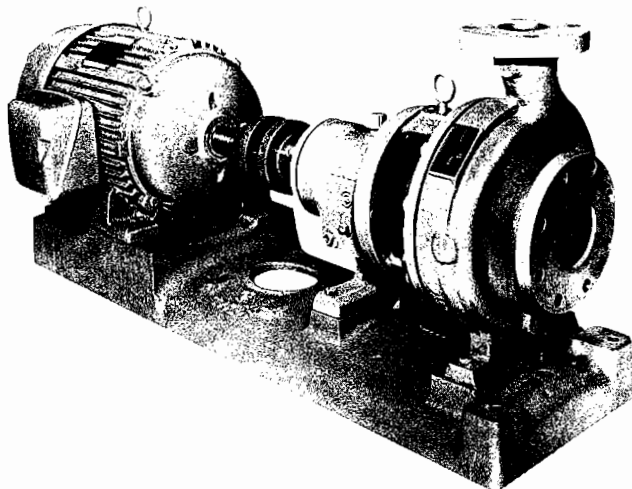Standard ANSI pumps, meeting specification B73.1M-1984 [1] (Figure 1), have proven to be very reliable for the majority of services encountered. A small number of installations, however, are plagued by a high incidence of failures, primarily of the mechanical seal and bearings. For these problem applications, components must be upgraded in the most cost effective manner possible. The solution chosen for a particular installation should be that which yields the maximum return in the form of avoided maintenance costs for the investment in upgraded components or features.

An investigation of the effect of improved component lives on overall pump life will be reported first. Discussions will focus next on upgrading components with the greatest potential for reducing pump maintenance costs since to obtain the best payback, it is the change in overall pump life and not the improvement in the life of a single component which is the significant factor. For example, it might be possible to increase the service life of a component by a factor of ten (i.e., the pump to bedplate attaching bolt) by improved metallurgy, controlled installation torques and periodic inspections. But if the life of this component is already long relative to that of the pump as a whole, the pump life would not be measurably affected with this upgrade, and time would be better spent on more critical components.

COST ANALYSIS

For the majority of ANSI pumps there is no economic justification to upgrade all pump components. Using published information on the typical service life of ANSI pumps and making reasonable estimates about component upgrade costs and effectiveness, an analysis can be made to determine what component modifications are economically beneficial. To illustrate this strategy, analysis will be carried out on the life of standard build ANSI pumps for both typical and problem installations. The problem pump is assumed to have a bearing life of only one-third that of the typical pump. The analysis will then be repeated after upgrading components to yield the largest improvement on overall pump life at a minimum cost.

Published data on the mean time between failure (MTBF) of ANSI pump components are shown in Table 1. This data was obtained from a large population of ANSI pumps operating in a southwestern chemical plant [2]. The MTBF of the overall pump is less than the shortest life component and is given by Equation (1) where "L" is the life of individual components.

$$MTBF = \frac{1}{\sqrt{\left(\frac{1}{L_1}\right)^2 + \left(\frac{1}{L_2}\right)^2 + \left(\frac{1}{L_3}\right)^2 + ... \left(\frac{1}{L_n}\right)^2}} \quad (1)$$

As can be seen from Equation (1), the effect of a single component life on the pump mean time between failures (MTBF) falls off dramatically as the life of this component

Figure 1. Standard ANSI Pump.

Table 1. Standard ANSI Component Lives [2].

Component	MTBF (Yrs.)
Mechanical Seal	1.2
Ball Bearings	3.0
Coupling	4.0
Shaft	15.0
Pump*	1.07

*Estimated Using Equation (1).

increases. For example, if the effect on pump life of the coupling and shaft in Table 1 are both ignored, the overall pump life increases by only 0.041 years or 3.8 percent, an insignificant difference, given the accuracy of the base data.

Standard ANSI Pump Cost Analysis

Applying Equation (1) to the component lives shown in Table 1 gives a MTBF of a typical ANSI pump of 1.07 years. Repeating these calculations for a problem pump which has a bearing life only one-third as long as that shown in Table 1 yields a pump MTBF of 0.75 yrs. These results are shown on Table 2.

Table 2. Component Upgrade Effect on Pump Reliability and Cost.

Pump Configuration	Seal MTBF (Yrs)	Bearing MTBF (Yrs)	Pump MTBF (Yrs)	Maintenance Costs ($/Yr)	Savings ($/Yr)	Upgrade Cost ($)	Payback (Months)
Typical Application							
Std. ANSI	1.2	3.0	1.07	2330	—	—	—
Full Upgrade	2.4	6.0	1.95	1284	1046	650	7
Seal Upgrade	2.4	3.0	1.70	1473	857	250	4
Bearing Upgrade	1.2	6.0	1.13	2216	114	400	42
Problem Application							
Std. ANSI	1.2	1.0	0.75	3314	—	—	—
Bearing Upgrade	1.2	2.0	1.00	2509	805	400	6

Pump rebuild costs vary widely with the extent of standard rebuilds, labor costs, parts costs, and overhead, as well as the accounting method used. For this economic analysis an average rebuild cost of $2500 will be used, although some companies report costs as high as $5000 per rebuild with full overhead included. The objective is to reduce the yearly maintenance cost to a minimum. Per annum costs are calculated by Equation (2).

$$\text{Yearly Rebuild Costs} = \frac{\text{Average Rebuild Cost}}{\text{Pump MTBF}} \qquad (2)$$

The yearly rebuild cost averages $2330 per year for the typical ANSI pump, and $3314 per year for the problem pump.

Upgraded ANSI Cost Analysis

For the purpose of cost analysis, the assumption was made that upgrading mechanical seal and bearing components will result in a doubling of the life of these parts. Details of seal and bearing upgrades which might accomplish this will be presented later. It will also be assumed that the costs to upgrade the seal and bearing are $250 and $400, respectively. These upgraded components, when used on a pump in a typical application, increase the pump MTBF from 1.07 to 1.95 years. These results are shown in Table 2 as "FULL UPGRADE." The improved life reduces maintenance costs $1046 per year which gives a pay back of seven months. While this appears to be a good investment, separating the

effect of seal and bearing upgrades shows that the seal change alone produces most of the benefits at only a fraction of the cost. For this application, the best solution is probably to upgrade only the mechanical seal.

These same calculations were repeated for the problem application with reduced bearing life. As can be seen in Table 2, the pay back is only 6 months, so, for this application upgrading of the bearings is a good economic decision.

Cost Analysis Summary

As shown, the effect of various upgrades on pump life varies widely depending primarily on the component life before the improvement. Curves of pump life versus component life for both mechanical seals and bearings where all other component lives are held constant are illustrated in Figure 2. Increasing the mechanical seal life from the typical MTBF point "A" to point "B" raises the pump life by a significant degree. Further increasing the seal life to "C" has only a small effect on overall pump life. As seen from the bearing curve, increasing the bearing life beyond the typical MTBF, point "D," has an insignificant effect on overall pump life.
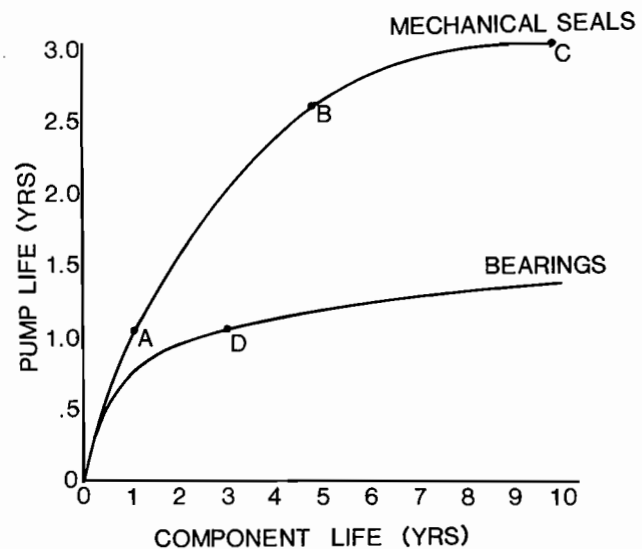

Figure 2. Pump Life vs Component Life.

From the calculations in the previous sections, it was shown that pump life is very sensitive to mechanical seal life and bearing life for a problem installation. Thus, upgrading of components should be considered for mechanical seals in typical applications or for bearings operating under extreme conditions. Extensive upgrading of other components cannot normally be economically justified.

Various changes that should greatly extend the lives of the components that have been identified as needing improvements will now be discussed in detail.

MECHANICAL SEAL

Mechanical seals have the shortest MTBF of any pump component and therefore have the best economic justification for upgrading. Before beginning any mechanical seal related upgrade, the specific installation should be reviewed with the seal supplier to determine if the optimum seal has already been chosen. Potential seal problems will be discussed, followed by a review of possible solutions.

Seal Problems

Excessive temperature of the seal environment can severely reduce seal reliability. Temperature will directly affect wear and corrosion rates of the primary sealing elements. The dramatic increase in wear that elevated temperatures can induce is shown in Figure 3. Corrosion rates double for every 18° temperature rise. This means a ten fold increase in corrosion for a 60° temperature rise. The problem is further exacerbated because most materials are not equally wear and corrosion resistant, and compromises in material selection must sometimes be made.
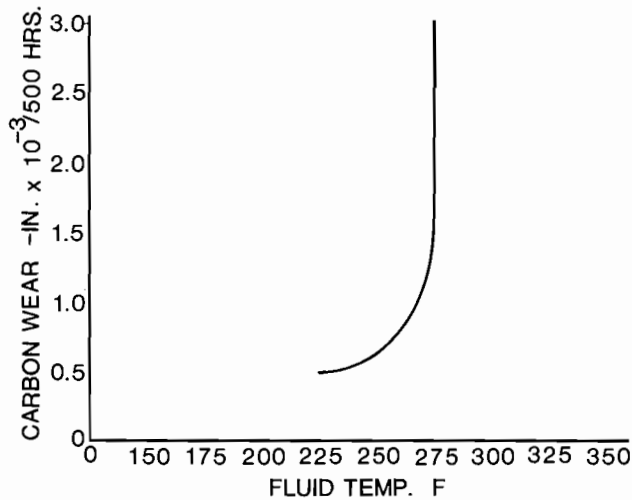

Figure 3. Carbon Wear Rate vs Temperature.

High temperatures can also indirectly reduce seal life by affecting the sealed fluid. Unless the seals are properly selected, heat being generated across the seal faces can vaporize a boiling liquid, thus losing the fluid film between the faces. If abrasives are present in the liquid then vaporization will also cause precipitation of these components, further accelerating face wear.

In addition to temperature, seal life can also be affected by other environmental, design, and human factors. Suspend-

Table 3. Sealing Problem Component Upgrades—Conventional Solutions.

Problem	Solution	Advantages	Limitations
High Temperatures & Contaminants	External Cool Lubricating Flush	Highly Effective for Reducing Seal Chamber Temp.	Dilution of pumpage may not be acceptable. Flush may not be available.
High Temperatures	External Recirculation	Low Cost	Possibility of leakage. Min. temp. equal to pumpage temp.
	Internal Recirculation	Low Cost	Min. temp. equal to pumpage temp.
	Jacketed Box	No Pumpage Dilution	External cooling system required. Will cool pumpage. Limited effectiveness.
	Recirculation with Heat Exchanger	Can Produce Very Low Temperatures	External cooling system required.
Suspended Solids	Recirculation with Cyclone Separator	Lower Maintenance than Filter	Not effective w/small or low density solids. Requires 20-30 psi min. differential. Higher cost than w/filter.
	Recirculation with Filter	Low Cost	Increased maintenance costs. Plugged filter will restrict flush.
Excessive Shaft Deflection	Increased Diameter	Reduced Deflection	Increased cost.

ed solids can cause accelerated sealing face wear or product side hang up. Seal failures can also be the result of improper installation or excessive shaft deflection.

Seal Upgrades

The specific component upgrade must be chosen based on an understanding of the cost analysis for that change. By doubling the seal life of a typical pump, maintenance costs can be reduced by $857 per year (Table 2). For a problem application where seal life is only half that normally attained, a doubling of seal life produces savings of $1748. Upgrading of these installations must cost less than $1714 and $3496, respectively, if a two year payback is required. The cost to upgrade the pump itself will generally be within the $1714 limit, but associated supply lines, heat exchangers and added maintenance may add significantly to the cost. On the benefit side, the savings due to reduced downtime, process interruption, etc., must also be considered.

Conventional upgrades to extend seal life will be briefly reviewed along with their advantages and disadvantages. More recently available component improvements will then be presented in more detail. All upgrades are listed in Tables 3 and 4.

Table 4. Sealing Problem Component Upgrades—Improved Solutions.

Problem	Solution	Advantages	Limitations
High Temperatures & Suspended Solids	Enlarged Seal Chamber w/External Flush	Very Effective for Reducing Seal Chamber Temp. & Contaminants	Min. temp. equal to pumpage temp.
	Enlarged Tapered Chamber	Low Cost Effective for High Temp. & Contaminants	Not as effective as an external cool lubricating flush.

For maximum life, a mechanical seal should be provided with a cool, clean environment. Conventionally, this is done by adding flushes from the discharge or outside sources and adding coolers, filters, cyclone separators, etc., as needed. These solutions are listed and compared in Table 3 along with their pertinent limitations.

The conventional upgrades listed generally increase both complexity and cost and reduce efficiency due to recirculation losses. Two cost effective solutions (Table 4) for temperature or solids problems which have given good laboratory and field test results are the enlarged tapered bore seal chamber, (Figure 4) and the enlarged bore seal chamber shown in Figure 5.

It has been recognized for a number of years that one of the factors limiting the performance of seals used in ANSI pumps has been the restrictive radial space available. These seal chamber dimensions were chosen to provide an optimum packed box design with little regard for mechanical seals. Increasing the radial clearance as shown in Table 5 should allow for the design of improved performance seals utilizing this space. The ANSI committee has recognized the usefulness of the enlarged seal chamber and has included this option in the most recent draft of ANSI B73.1.

Both the tapered (Figure 4) and enlarged (Figure 5) seal chambers have large radial clearances to improve circulation and reduce seal temperatures. Tests were run in conjunction with a seal manufacturer to determine seal temperatures and wear rates for seals installed in a standard ANSI seal chamber and both enlarged seal chambers. All configurations were run with 200°F water, no flush, and 50 psi seal chamber pressure and utilized balanced seals. As shown in Table 6, the seal face temperature showed substantial reduc-
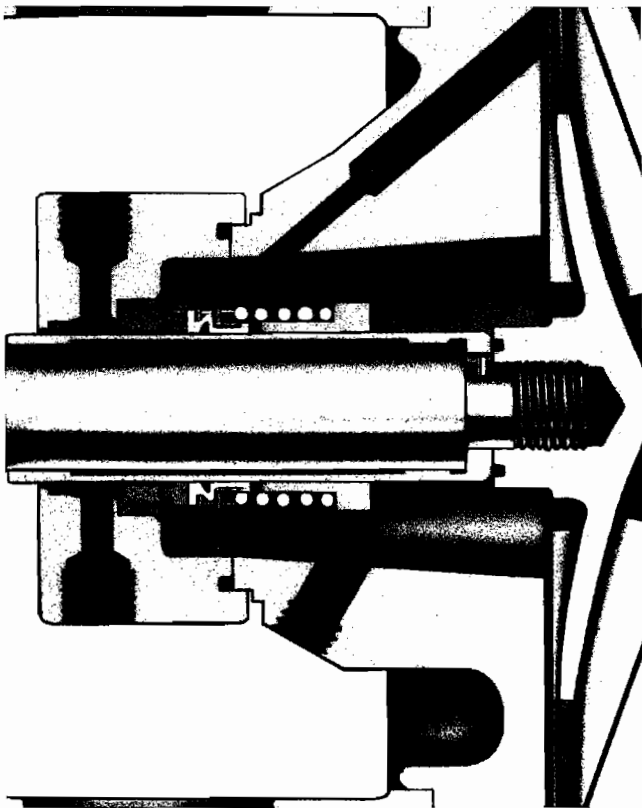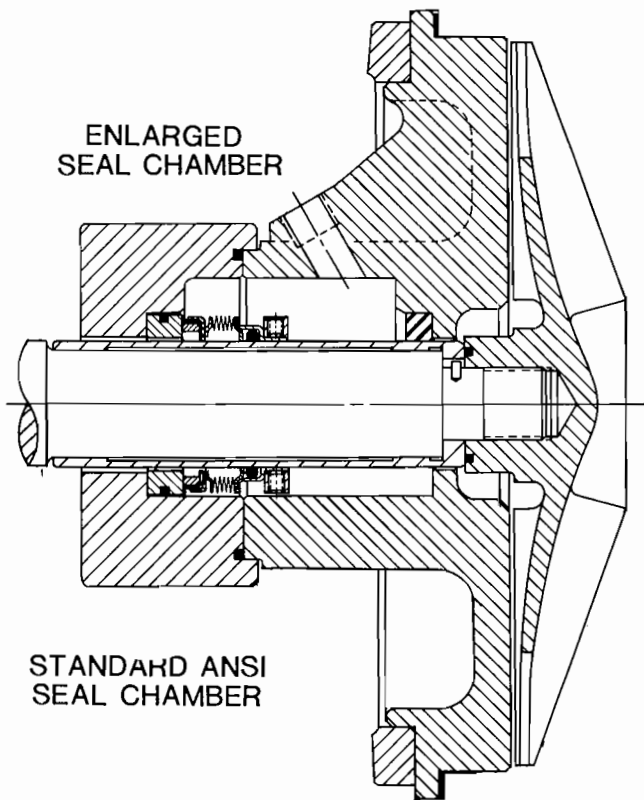
Figure 4. Enlarged Tapered Seal Chamber.

Figure 5. Enlarged Seal Chamber vs Standard ANSI.

Table 5. Radial Clearance Standard ANSI vs Enlarged Box.

| Maximum ANSI Designation | Shaft Diameter (In.) | Radial Clearance | | % Increase |
		Standard ANSI (In.)	Enlarged Bore (In.)	
AA	1-3/8	5/16	3/4	140
A80	1-3/4	3/8	7/8	133
A80	2-1/8	3/8	7/8	133
A120	2-1/2	7/16	1	128

Table 6. Seal Temperature Rise.

Box	Seal Face Temperature Rise (°F)	Box Liquid Temperature Rise (°F)
Std. ANSI	40	27
Enlarged Tapered (see Figure 5)	3	0
Enlarged Cavity (see Figure 6)	16	7

Note: Temperature rise above pumpage temperature based on tests using 200°F pumpage.

tions for both enlarged seal chambers relative to the standard ANSI seal chamber.

The enlarged tapered seal chamber (Figure 4) showed the most improvement over the standard ANSI seal chamber, reducing the seal face temperature to within 3°F of the pumpage. The tapered seal chamber is also effective in reducing the concentration of solids. Heavy particles are centrifuged to the outside wall and then moved forward by the taper to the impeller pumpout vanes. When the seal temperature must be reduced to below that of the pumpage, or the solids separating effect of the tapered bore are not sufficient, the tapered box should be replaced by the enlarged seal chamber (Figure 5) fitted with a cool external flush and a throat bushing. The tapered seal chamber principal disadvantage is its inability to accommodate a throat bushing, without which any external flush is largely ineffective.

Seal Mounting Tolerances

For a seal to reach its design life the relative locations of the primary, stationary and rotating faces must be held within close tolerances. ANSI B73.1M 1984 specifies that radial runout of the shaft and axial runout of the stuffing box face must each be held to within 0.002 in total indicator reading, while shaft deflections under dynamic conditions must not exceed 0.002 in at the face of the stuffing box. Static runout tolerances can be met by adhering to good basic maintenance, design, and control of manufacturing processes.

Dynamic deflection is affected by a number of variables including impeller and volute design, pump speed, geometry and material of the shaft, as well as the flow point on the hydraulic curve. To better illustrate the variables affecting deflection, calculations were made for a standard sleeved shaft used in both a moderately loaded ANSI pump and a heavily loaded pump. The pump chosen for the moderately loaded calculation was a 2 × 3-10 operating at 3500 cpm, while the heavily loaded calculation was based on the same power end with a 2 × 3-13 wet end at 3500 cpm. The analysis was then repeated using a larger diameter upgraded shaft.

Deflection calculations were made (Appendix A) assuming maximum speed, and maximum impeller diameter.

The results of the shaft deflection analysis are shown in Table 7. The standard shaft when used in a moderately loaded pump has a maximum deflection that is well within the ANSI specification of 0.002 in at shut off. Increasing the flow to the best efficiency point (BEP) reduces the deflection to less than one tenth the shut off value. Upgrading the shaft diameter reduces deflections further but for this pump only increases the cost of bearings, seals and shaft and has an insignificant effect on extending seal life. The standard shaft is the proper choice for this application.

For the heavily loaded example, the deflection does not meet the ANSI requirement when the standard shaft is used. For this application, the optimum solution is the upgraded shaft with increased diameter.

Table 7. Shaft Deflection.

| Pump | Speed (RPM) | Shaft | Deflection | |
			@ Shut Off (In.)	@ BEP Flow (In.)
2x3-10 (moderate load)	3500	Std.	0.0011	0.00010
		Upgraded	0.0005	0.00004
2x3-13 (heavily loaded)	3500	Std.	0.0022	0.00018
		Upgraded	0.0010	0.00008

When seal installation errors become a factor in reduced seal life, then a cartridge seal as shown in Figure 6 may be economically justified. The cartridge seal guarantees a properly set seal and minimizes the opportunity for assembly errors in the field. When a cartridge seal is specified, however, its design should be carefully reviewed. Standard ANSI pumps meet the 0.002 in runout specifications on the outside diameter of the shaft sleeve. If a cartridge seal goes over the manufacturer's sleeve then this will add to the total runout and the seal will not be supported within the 0.002 in optimum runout tolerance. A properly designed cartridge seal will have a sleeve which replaces the standard shaft sleeve so as not to add to the tolerance stack up.
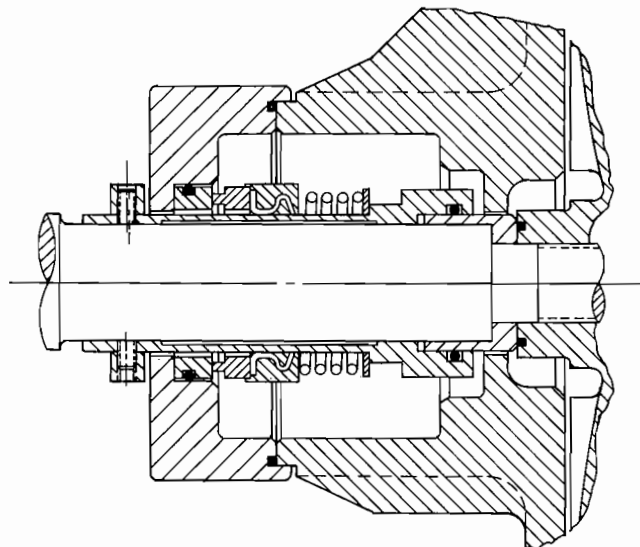

Figure 6. Cartridge Seal.

Mechanical Seal Summary

Methods to upgrade severe mechanical seal installations plagued by a high incidence of failures are widely available and utilized. But pumps with typical seal lives also have excellent upgrade potential and are often overlooked due to the perceived cost or complexity of upgrading. One low cost solution which would benefit a significant range of pump applications is the enlarged seal chamber.

POWER END

In a typical application, bearing life does not have a strong influence on overall pump reliability. Increased bearing life for these applications is worthwhile only if the upgrades can be done at a minimal cost. Pump installations with bearing lives significantly below the MTBF of 3.0 years, however, justify a substantial investment to upgrade. In this section, the factors affecting bearing life and methods of overcoming these limitations will be discussed. Particular emphasis will be placed on defining the limitations of the ANSI pump before proposing upgrades for each problem.

ANSI B73.1M-1984 requires a minimum fatigue life (L'_{10}) of two years, which is equal to a mean life of approximately 10 years. Most power ends are used with a variety of liquid ends, and are designed for a two year minimum bearing fatigue life when used with the liquid end which creates the highest load. Most liquid ends will impose significantly lower loads, and therefore have longer calculated fatigue lives. For the average pump used with a particular power end the calculated mean bearing fatigue life is therefore approximately 100 years.

In actual practice, however, the mean life attained is only three years, considerably less than the calculated value. This is because actual bearing failures are normally not from fatigue due to excessive load, but result from excessive temperatures, contaminants or poor installation. The calculated life is thus reached only a small percentage of the time. In the next two sections, the environmental factors which actually cause failure will be reviewed. The final section will review ANSI bearing design to determine when bearing upgrades are warranted.

Oil Contaminants

Clean lubrication is essential for optimum performance of antifriction bearings. Contaminants can be abrasive or corrosive causing excessive race and ball wear, or they can cause degradation of the oil lubricating properties. To eliminate contaminants, frames can be sealed or fresh oil can be continually supplied to the bearings.

Most ANSI pump bearing frames are sealed around the shaft with a lip seal. This method of sealing is inexpensive, simple to install and is very effective when new. Lip seals, however, even under normal operating conditions are effective for only three to four months. By then, the elastomer lip is worn beyond the limit where it can effectively seal. Upgrades can be used to more effectively exclude contaminants from the bearing frame. Two devices in particular, labyrinth seals (Figure 7) and magnetic seals (Figure 8), are currently available. The choice of method used is a function of the form of the contaminant present, plant restrictions, environmental considerations and cost.

The labyrinth oil seal is an effective means of protecting the bearing frame from most types of contaminants. The labyrinth seal is non-contacting so its performance will not deteriorate with time as lip seals do. Labyrinth seals are particularly well suited for use with oil mist systems. In such systems, the labyrinth seal can replace the normal vent

openings to the environment. The labyrinth seal, however, does not keep atmospheric moisture or corrosive vapors from entering the bearing frame.
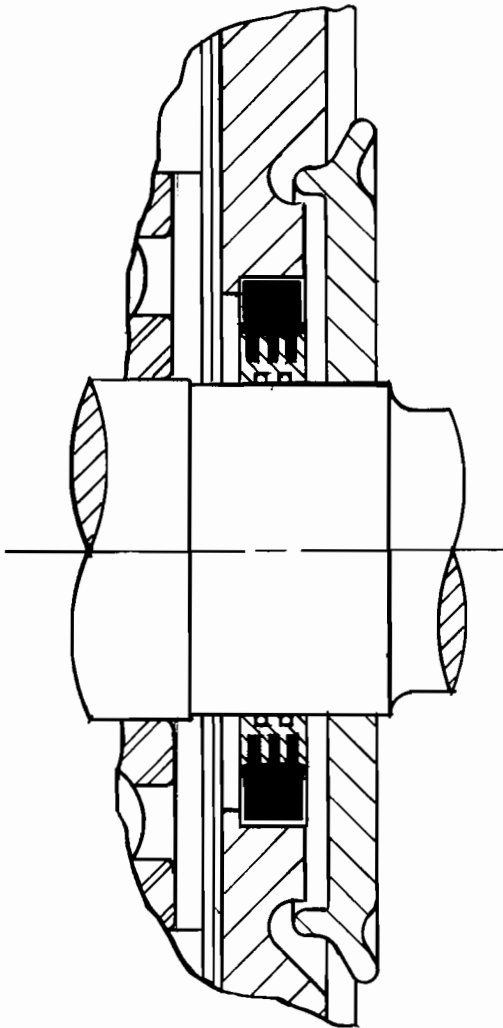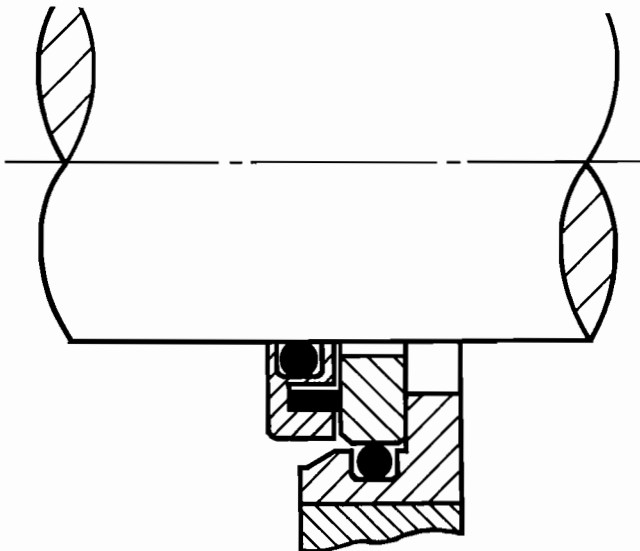

Figure 7. Labyrinth Frame Oil Seal.

Figure 8. Magnetic Frame Oil Seal.

If water or chemical vapors are present in high concentrations or are condensing in the bearing frame, and are suspected of contributing to reduced bearing life, then a more positive sealing method such as a mechanical seal must be used. The magnetic seal is a practical alternative for these applications. It is basically a mechanical seal with the spring replaced by a magnetic seat or rotor to reduce overall length. This results in very light face loading which extends seal life to seven to ten years. To completely protect the bearings, the breather should also be replaced with an expansion chamber. This will compensate for pressure rise within the frame induced by temperature swings, which might otherwise open the seal faces and allow contaminants to leak into the bearing frame cavity.

Laboratory tests were run to determine the effectiveness of the various shaft sealing devices relative to the lip seal. The tests, run in conjunction with another seal manufacaturer used a spray wash down for each device in both the static and dynamic conditions. The bearing oil was then drained and analyzed for water content. The results for the devices tested—new lip seals, lip seals used three months, lip seals used four months, magnetic seals and labyrinth seals are shown in Figure 9. After being subjected to a high pressure water spray for 15 minutes at each end of the bearing frame, the new lip seals allowed only enough water to pass through to dilute the frame oil less than 0.1 percent. The three month old lip seals let through ten times the water of the new lip seals. With four months wear the lip seal lets in twenty times the volume as when new. The magnetic seals and the labyrinth seals both gave results near that of a new lip seal with the labyrinth slightly lower but within experimental error. Either of these improved seals are substantially more effective than the worn lip seal. While the wash down used in this test was more severe than that used in a typical field installation, it is useful to show the general degradation of the lip seal with time.

Another approach to eliminating contaminants is to provide the bearings with a continual supply of clean lubricant. A good system for this is a pure or purge oil mist system as shown in Figure 10. This is not normally practical for a single problem unit, but is a good system if it is available in the plant.
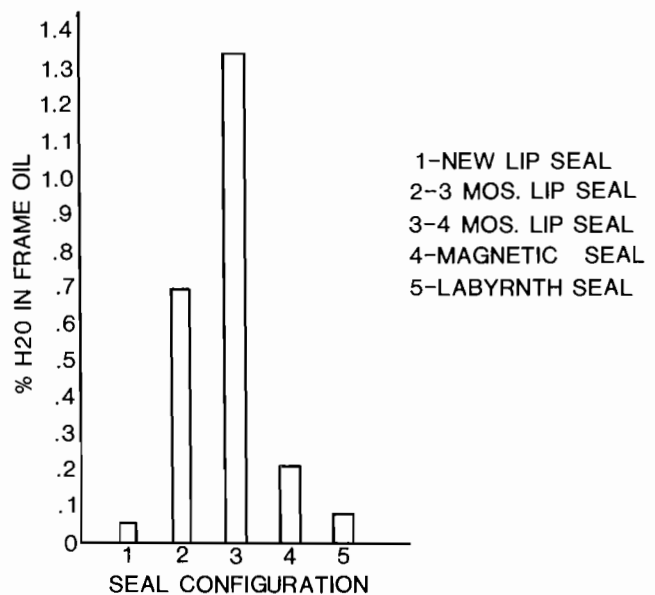

1-NEW LIP SEAL
2-3 MOS. LIP SEAL
3-4 MOS. LIP SEAL
4-MAGNETIC   SEAL
5-LABYRNTH SEAL

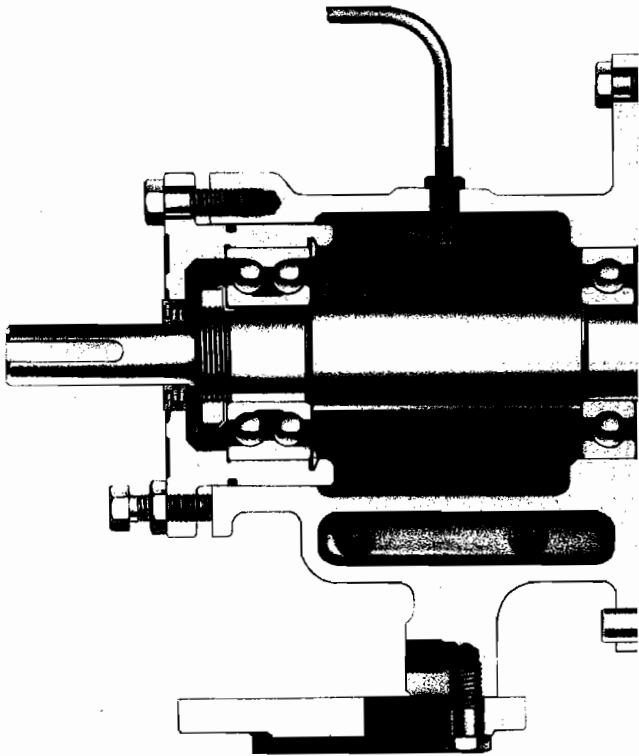*Figure 9. Results of Water Spray Tests.*

Figure 10. Oil Mist Lubrication of ANSI Bearing Frame.

Temperature

Pumps meeting ANSI B73.1M-1984 are typically supplied with bearings having a maximum service temperature of 240°F while the hydrocarbon lubricating oil used is normally limited to 180°F. Some users operate at temperatures above 180° by increasing the oil change frequency. When temperatures exceed these limits, a high temperature oil and bearing must be used or a cooling option must be added to the frame. As a point of reference, test data, Cappellino and Osborne [4], have shown that oil temperatures at the bearing race are at least 10°F higher than those in the oil sump.

A high temperature lubricant could be either a synthetic oil or a hydrocarbon oil with additives to provide satisfactory properties at the higher temperature. High temperature bearings use a heat treated stabilized steel, which can significantly raise the maximum operating temperature. Although
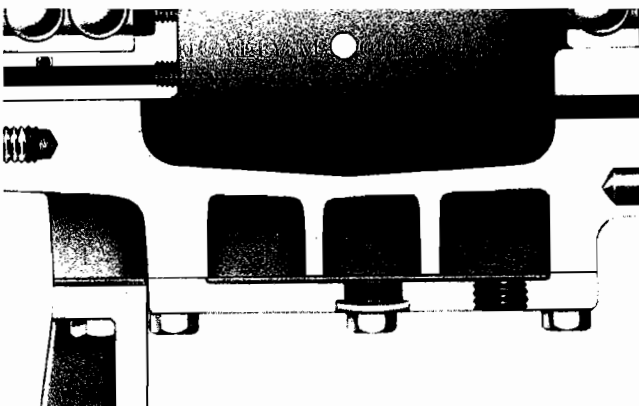

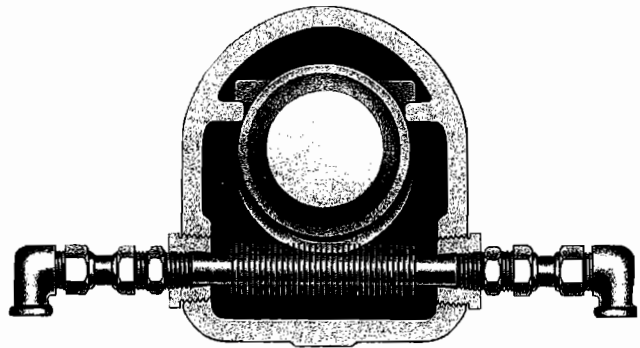Figure 11. Bearing Frame Cooling Jacket.

Figure 12. High Temperature Bearing Frame Cooler.

both the oil and bearings are available, they are both expensive. The use of synthetic oil may also be a maintenance problem if it is to be used as a special in only one or two pumps.

Cooling is normally accomplished by circulating a cool liquid through a jacket in the frame (Figure 11). For higher temperature applications, an internal heat exchanger, as shown in Figure 12, can be used to cool the oil directly. Either of these systems are effective in reducing temperature but the installed cost can be considerable when the coolant source and disposition are taken into consideration.

Where a liquid coolant is not available or the costs are excessive, an air cooled frame can be an effective solution. This method uses a fan mounted on the pump shaft to air cool the frame by forced convection.

Bearing Load Factors

Most bearing failures are due to a poor operating environment rather than excessive loads and will not, therefore, be reduced by increasing the bearing capacity. The discussion on bearing upgrades and their effect on bearing reliability which follows will consider only the thrust bearing since it has higher combined loads and failure rates than the radial bearing.

ANSI pumps are normally supplied with a double row angular contact ball thrust bearing as shown in Figure 13. A commonly used method to improve bearing life is to substitute a pair of angular contact duplex bearings with high contact angles, as shown in Figure 13. The effects of this
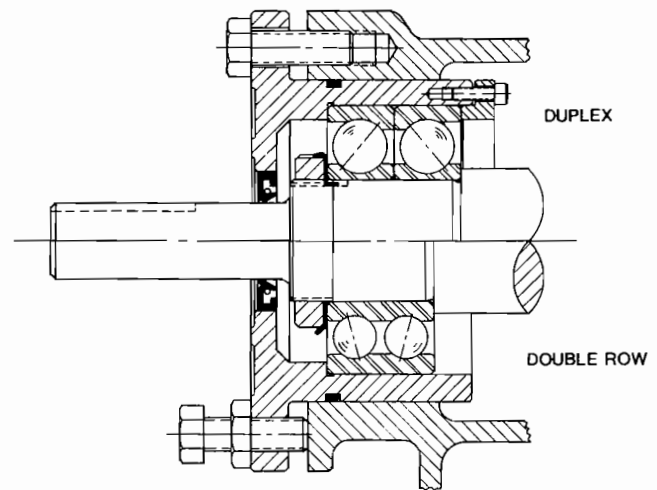

Figure 13. Thrust Bearings: Double Row vs Duplex.

change on bearing life and operating temperatures for both a typical and a highly loaded ANSI pump were analyzed.

Pump sizes and speeds were chosen to provide representative nominal and maximum bearing loads. The fatigue life was then calculated for shutoff conditions using a predetermined method [5]. Bearing operating temperatures were also calculated for both bearings at the same load typical of a $1\frac{1}{2} \times 3$-10 operating at 3500 cpm and 68° ambient temperature using the equations shown in Appendix B.

As seen in Table 8, the fatigue life increases almost by a factor of three for the duplex bearing relative to the double row bearing. The duplex bearing also runs about 20°F hotter due to the larger contact angles, higher preloads.

A double row bearing will typically reach only three years MTBF. The theoretical mean fatigue life of 111 years is not reached due to oil contaminants, mishandling or poor maintenance. The high angle duplex bearing has a longer fatigue life, but it is equally susceptible to contaminants, mishandling, and poor maintenance, so no increase in actual life (MTBF) should be expected. In fact, the duplex bearing may have a shorter actual life, due to the higher operating temperature.

For the double row angular contact bearing, the race temperature is calculated to be 173°F (Table 8). With the duplex bearing the race temperature rises to 194°F which is beyond the normally acceptable safe operating temperature for hydrocarbon oil. For this special heavily loaded case the high angle duplex bearing will have a shorter life than the double row bearing unless special measures are taken to cool the oil. This is despite the fact that the duplex bearing fatigue life is substantially larger than for the double row.

Table 8. Comparison of Duplex and Double Row Bearings.

	Pump Size	Pump Speed (RPM)	L'_{10} (Years)	Median Life (Years)	Bearing Race Temp. (°F)	Oil Sump Temp (°F)
Nominal Load, Double Row	4×8-10	1750	22.1	111	—	—
Nominal Load, Duplex Brg.	4×8-10	1750	53.8	269	—	—
Maximum Load, Double Row	$1\text{-}\frac{1}{2} \times 3$-10	3500	2.0	10	173	163
Maximum Load, Duplex Brg.	$1\text{-}\frac{1}{2} \times 3$-10	3500	5.6	28	194	184

In summary, bearing thrust capacity can be increased by using a larger bearing or steeper contact angles. Either method will result in higher bearing race and oil sump temperatures and should, therefore, only be considered when the minimum fatigue life is below two years as for an installation, with very high suction pressures or pumping a high specific gravity liquid. Some API type users have found that their thrust loads are high enough to justify duplex bearings for their ANSI pumps. Most pump manufacturers offer this option.

Power End Conclusion

For typical services, pump MTBF is reasonably insensitive to changes in bearing MTBF, but the situation changes dramatically for pumps on tougher services where actual bearing lives are short. For these installations various methods of reducing oil contaminants and temperature are economically justifiable.

In most applications double row angular contact bearings are the proper choice for ANSI pumps. Increasing the bearing capacity through larger contact angles or bearing size increases heat generation. Increasing the minimum fatigue life beyond two years will not result in longer bearing life attained in the field but will add to the cost of the pump. In some applications the result will be a reduced bearing life. Proper bearing selection requires the user to avoid the simplistic approach that bigger is necessarily better and instead must analyze all the factors that contribute to bearing reliability.

PUMP INSTALLATION AND MAINTENANCE

Good installation and maintenance practices can be as important in increasing pump life as any component upgrade. A properly mounted pump will have both flange loads and driver to pump alignment within specifications. Lack of conformance in either area can cause reductions in bearing, coupling and mechanical seal life. Piping loads should be within limits as specified by the pump supplier. If thermal loads are present which can generate higher loads, then either the piping will need to be modified or a spring mounted bedplate should be used. Static loads should always be supported by pipe brackets. Misalignment between the driver and pump causes an added radial load on the shaft resulting in reduced bearing, mechanical seal and coupling life. Parallel and angular alignment of the pump and driver shafts must be held within 0.002 total indicator reading (TIR). Coupling manufacturers sometimes endorse larger amounts of misalignment and their couplings will usually hold up, but the load transmitted to the pump would cause early failures. It is extremely important that a final alignment be done after the pump and piping have reached operating temperature. A good maintenance program with periodic inspections and oil change at the recommended intervals is necessary for satisfactory pump operation and will probably pay dividends in terms of increased pump MTBF.

CONCLUSIONS

For most applications, the standard ANSI pump performs well, and broad upgrading will not improve overall pump life in a cost effective manner. However, component upgrades should be considered whenever the payback for that change is attractive. Misapplied upgrades can actually reduce overall pump life. This was illustrated for the high angle duplex bearing, but is also true for a filter used in a recirculation line that is susceptible to plugging or any change that has more potential problems than are being solved.

Therefore, time spent by the pump user in understanding his applications and their problems, and then translating this knowledge into proper component selections, will be amply repaid by reduced equipment cost due to appropriate specifications and reduced maintenance costsdue to increased MTBF.

APPENDICES

Appendix A
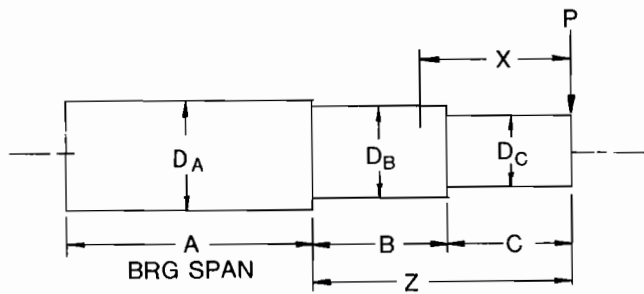
Calculations for Shaft Deflection

$$\text{Delta}_X = \left(\frac{P}{3E} \left[\frac{ZA\,(Z\text{-}X)}{I_A} + C^3 \left(\frac{1}{I_C} - \frac{1}{I_B} \right) + \frac{Z^3}{I_B} + \frac{X^3}{2I_C} - \frac{3X}{2} \left[C^2 \left(\frac{1}{I_C} - \frac{1}{I_B} \right) + \frac{Z^2}{I_B} \right] \right] \right) \quad (A\text{-}1)$$

Delta_X = deflection at X'

P = load

E = elastic modulus

$I_A, I_B I_C$ = moments of inertia of each section

BRG SPAN

Appendix B

$$t_m = \frac{N (M_0 + M_1) (0.1357)}{K (0.00254 \ d_m)^{1.65}} + t_o$$

t_m = bearing temperature, °F

N = bearing speed, cpm

M_0 = viscous frictional moment, ft.-lbs.

M_1 = load frictional moment, ft.-lbs.

K = cooling factor

d_m = bearing mean diameter, in.

t_o = ambient temperature, °F [6]

where

$$M_0 = 1.183 \times 10^{-6} \ (4.0) \ (VN)^{2/3} \ d_m^3$$

V = lubricant kinematic viscosity, centistokes [7]

and

$$M_1 = .083 \ Z \left(\frac{P_o}{C_o}\right)^y$$

z = constant

y = constant

P_o = static equivalent load, lbs.

C_o = basic static load rating, lbs.

P_B = thrust load, lbs. [7]

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