Bearing Life Extension and Reliability Features of Modern ANSI Pumps

Presented at
The 2nd International Conference
on Improving Reliability in
Petroleum Refineries and
Chemical and Natural Gas Plants

By:

Dr. Lev Nelik
Goulds Pumps, Inc.
240 Fall Street
Seneca Falls, New York 13148 USA

TABLE OF CONTENTS

1. OVERVIEW

ANSI pumps, similar to the one shown in Fig. 1. are used extensively in the chemical process industries. The pump is a single stage, overhung, end suction design built to ANSI specifications (Ref.[1]).

Introduced in 1961, this pump was redesigned in 1991 by enhancing power end features. The redesigned new version was named the "X-Series". In the "wet" end, the casing and impeller did not change but new seal chamber designs were added. The new design constitutes improvements as shown in Fig. 1 and 2 and Table 1.

These changes were made not only to improve the design, but also to maintain complete interchangeability with the old design for field retrofits.

This improved design was targeted to add value to a product, by increasing Mean Time Between Scheduled Maintenance (MTBSM), and Mean Time Between Failures (MTBF).

To quantify these improvements, a reliability testing program to confirm analytical methods used in this paper, was conducted.

This technical paper presents the results of a study conducted to compare pump/components reliability, in terms of extended bearing life and reduced maintenance costs, for a typical ANSI pump, equipped with a new design frame (MTX) versus the old design (MT).

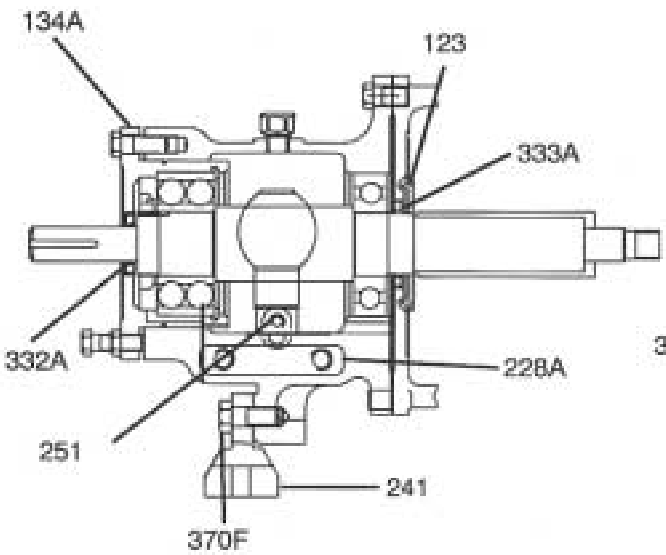
Rather than analyze all of the features mentioned in Table 1, it was decided to study the effects of just one of the features, namely the larger bearing oil sump. The effects of other features are available for presentation in separate studies, in order to keep each subject focused and within reasonable length.

The following discussions are, therefore, concentrated on the effect of the larger oil sump on bearing life, increased cycle life and reduced maintenance costs for a typical installation. Analytical formulations, assumptions and conclusions are presented first, followed by the test data as outlined in the table of contents.

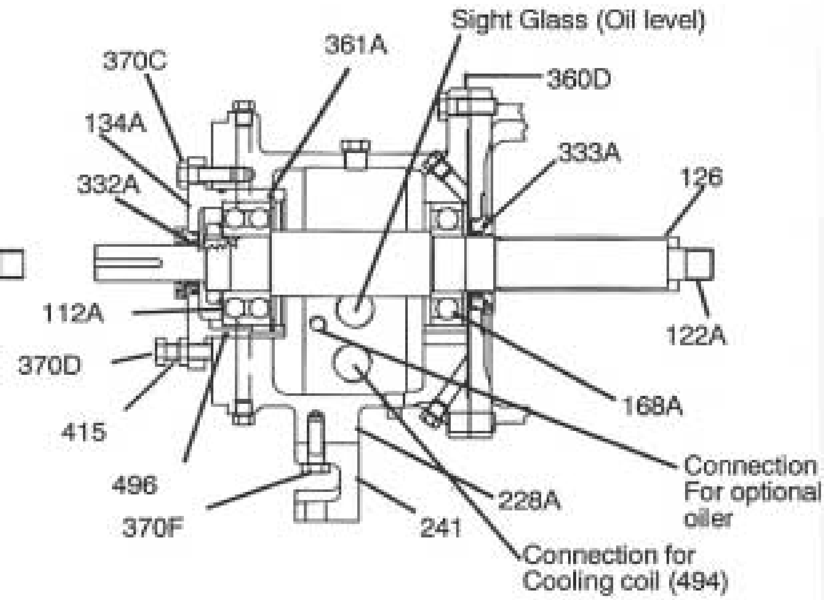
Table 1
X-Series features

1.	Larger oil sump volume.
2.	Sight glass for easier monitoring of actual oil level.
3.	Rigid frame (more rigid, solid) foot for better resistance to pipe load effects.
4.	Standard carbon-filled Teflon® labyrinth oil seals. Magnetic seals are optional.
5.	Optional C-face adapter for simpler pump-to-motor alignment and, in turn, reduced vibration.
6.	Large seal cavity for better seal life via improved seal heat conditions and larger liquid volume for cooling.

Power End Comparison



MT (older design)



MTX (enhanced design)

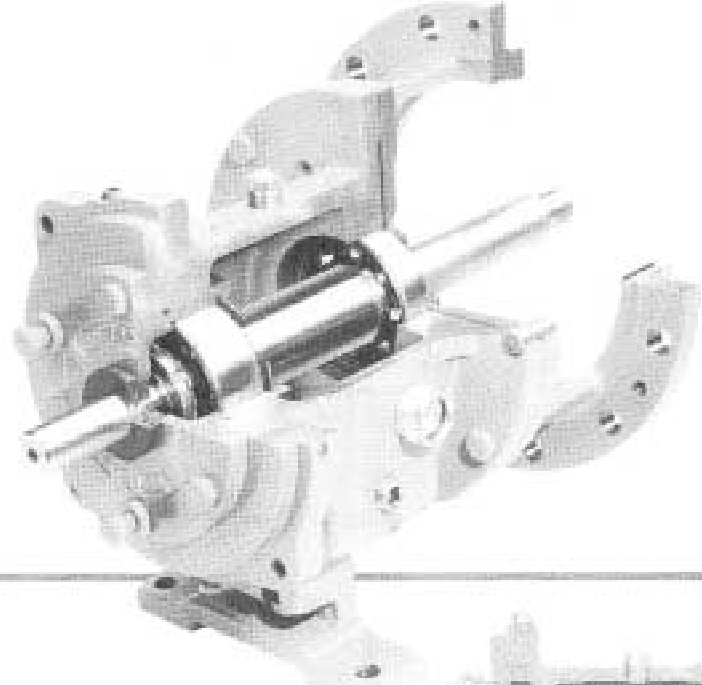
Maintenance Parts		Is it interchangeable?		
Item No.	Description	YES	NO	Comments
112A	Ball Bearing - Outboard	X		
122A	Shaft with Locknut & Lockwasher	X		
123	Deflector		X	Not used on STX
126	Shaft Sleeve	X		
134A	Bearing Housing	X		
168A	Ball Bearing - Inboard	X		
228A	Bearing Frame (Sub-Assembly)	X		
241	Frame Foot (8.25" or 10 CL)		X	
251	Constant Level Oiler	X		Optional on MTX
332A	Oil Seal - Outboard	X		Labyrinth or Lip
333A	Oil Seal - Inboard	X		Labyrinth or Lip
360D	Gasket—Frame to Adapter	X		
361A	Retaining Ring - Bearing Housing	X		
370 C, D	Bolts/Nuts - Impeller Adjustment	X		
370F	Capscrews—Foot to Frame		X	
415	Jam Nuts—Impeller Adjustment	X		
496	O-ring - Bearing Housing	X		
494	Bearing Cooling Coil (if required)		X	Not used on MT

Figure 1: Power End Comparison

Enhanced Power End

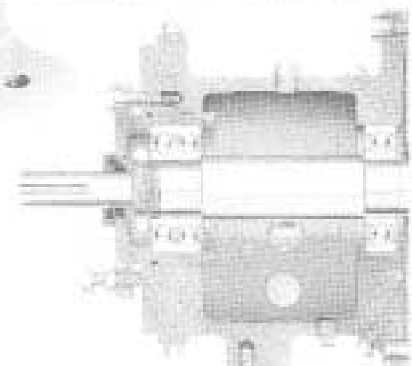
Designed for Reliability, Extended Pump Life

This power end is the result of customers' requirement for longer pump life. Significant improvements have been made to increase bearing life, decrease maintenance costs.



CARBON-FILLED TEFLON* LABYRINTH OIL SEALS
Prevent contamination of lubricant, the primary cause of premature bearing failure.

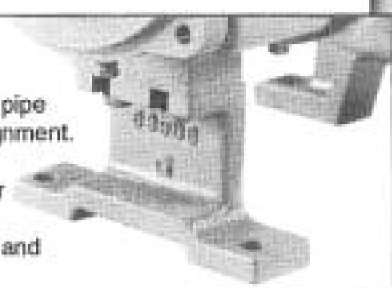
EXTRA LARGE OIL SUMP
Large oil capacity provides optimum heat transfer for cooler running bearings.



SHAFT/BEARING ASSEMBLY
Shaft designed for minimum deflection for long seal and bearing life. Bearings sized for optimum life. Duplex thrust bearings optional.

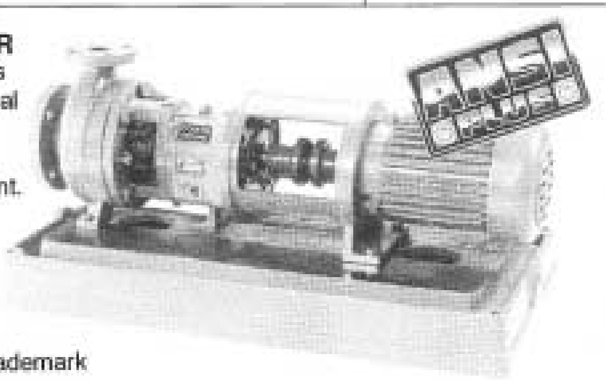
LARGE OIL SIGHT GLASS
Allows for viewing condition and level of oil - critical for bearing life. Frame pre-drilled for optional bottle oiler.

RIGID FRAME FOOT
Reduces effect of pipe loads on shaft alignment. Pump/driver alignment is better maintained for extended bearing and seal life.



CONDITION MONITORING SITES
Allow easy and consistent monitoring of temperatures and vibration for preventive maintenance. Optional installation of sensors.

C-FACE ADAPTER
X-Series Power Ends accommodate optional C-Face motor adapter - simplifies pump/motor alignment.



FRAME ADAPTER
Ductile iron standard for safety and strength.

*E.I. DuPont Reg. trademark

Figure 2: Highlights of X-Series features

2. FEATURE ANALYSIS

The ANSI X-Series pump has a modular design, and has three main power end sizes: STX, MTX and XLT-X. The "X" designation indicates the new power end design.

The MTX bearing frame size covers the majority of applications, and can be reasonably called a "typical" bearing frame. The analysis performed in this work used MTX versus MT power ends, although basic conclusions and findings of this work apply similarly to other frame sizes.

A key to a trouble-free operation of the pump power end is life extension of its ball bearings. Bearings and mechanical seals are known to contribute to the majority of pump failures.

The life duration of ball bearings depends on several factors. The most important of these factors is lubrication and the condition of the oil in the sump. The MTX frame (new design) has an oil sump with approximately three times the capacity, of the older design MT frame. An MTX frame contains 2.6 pints (42 oz) of oil as compared to 0.8 pint (13 oz) for the MT (Ref. [2]).

There are several benefits to this: cooler oil, a cleaner lubrication environment and reduced oil oxidation. Specifically, a wider sump makes it less sensitive to oil level fluctuations by turbulence, while running with a constant level oiler (option to a sight glass, which is standard). A greater sump depth results in particulate settling at the bottom and cleaner oil reaching the bearings, as well as an increase in net heat transfer area of the housing walls.

2.1 Effect of free surface area increase

The X-Series is provided with a sight glass, which gives an accurate indication by visual observation of the oil level in the bearing frame. A constant level oiler is available as an option, with the tap provided for quick installation, if desired. In this section we will analyze operation of the bearings and oil condition with the oiler installed, as the case may be. The phenomenon called "oiler burping" is well known in the industry, and can be described as follows. Consider an oiler attached to a bearing frame as shown in Fig. 3A, when the pump is not running, and the oiler has just been filled to a proper level, as specified in the oiler setting instructions, dimension "A" ($^{19}/_{32}$in. for MT/MTX frame). (Note: It may be helpful to obtain a copy of Technology Note #2 (a VCR tape) for better visualization of the oiler operation and proper oil filling method, Ref. [8].)

Initially, the level of oil in the frame $L_{H(A)}$ is equal to the level in the oiler $L_{oiler(A)}$, which actually controls the level in the frame by cutting off the supply of air into the oiler glass chamber as oil rises to $L_{oiler(A)}$ during oil filling (Ref [8]). At this point the oil level inside the oiler chamber is at $L_{chamber(A)}$ level.

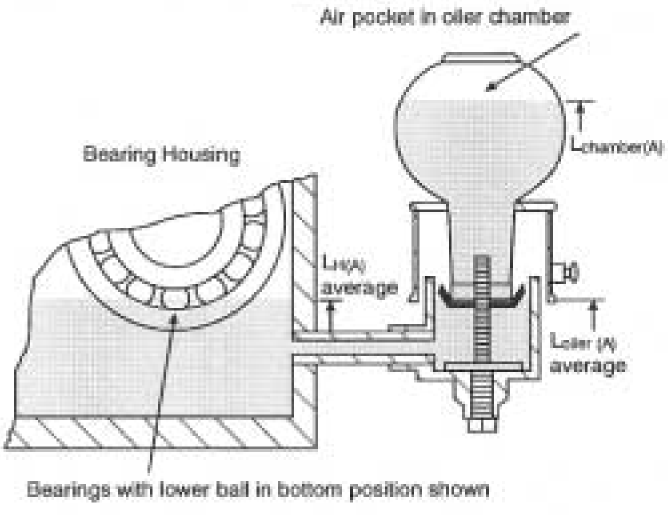


Figure 3A: Oil level as initially set after fill. Pump not running ($L_{H(A)} = L_{oiler(A)}$).

Consider now what happens when the pump starts. The disk friction action of rotating bearings picks up some oil from the housing and forms a rotating liquid disk. The level of oil in the oiler will drop to level $L_{oiler(B)}$ to compensate for the "disappeared" oil in the sump $L_{H(B)}$, (Fig. 3B).

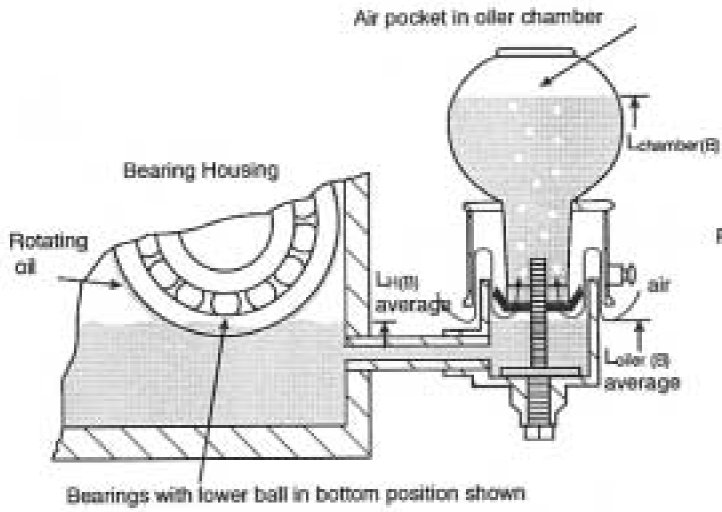


Figure 3B: Oil level dropped as a result of some oil taken into rotation by bearing, as pump starts up.

Clearly, air can now get into the oiler chamber (as shown) where it will displace some more oil out of the glass chamber into the oiler and housing. The oil level will rise until it will touch the lower end, i.e., the opening to the glass chamber, cutting off any further air supply (Fig. 3C).

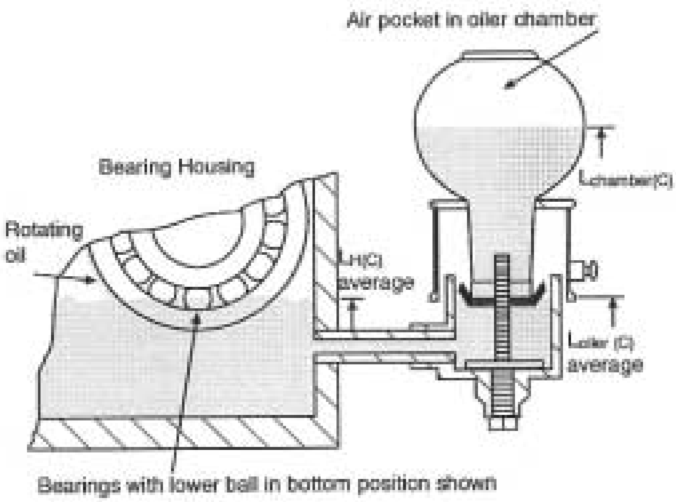


Figure 3C: Oil level in chamber $L_{chamber}$ (C) dropped to compensate for rotating oil in bearing frame.

The process, by itself, is not detrimental to pump operation, since the oil level will still be at the desired level (center of the lower ball).

The problem is, however, that the levels L_H and L_{oiler} are not perfect, but exhibit waviness due to turbulence and churning of oil in the bearing housing.

As a result, the following regime may exist, (Fig. 3D):

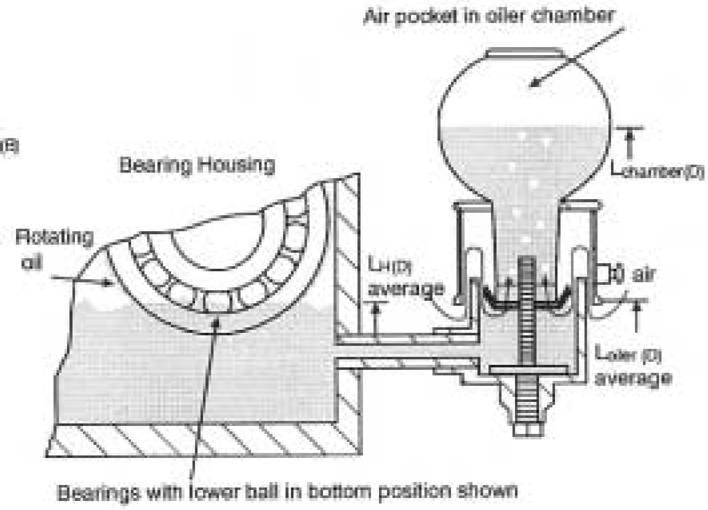


Figure 3D: Oil level waviness caused by turbulence.

Now, the oil can find ways among the turbulent peaks and valleys to flow into the frame, lowering the level in the chamber to $L_{chamber(D)}$, and the level in the frame will be *higher* than required, covering the lower ball above its center. This oiler "burping" will continue until the oil rises high enough to prevent air from entering the glass chamber even at the lowest "valley" of the turbulent level (Fig. 3E).

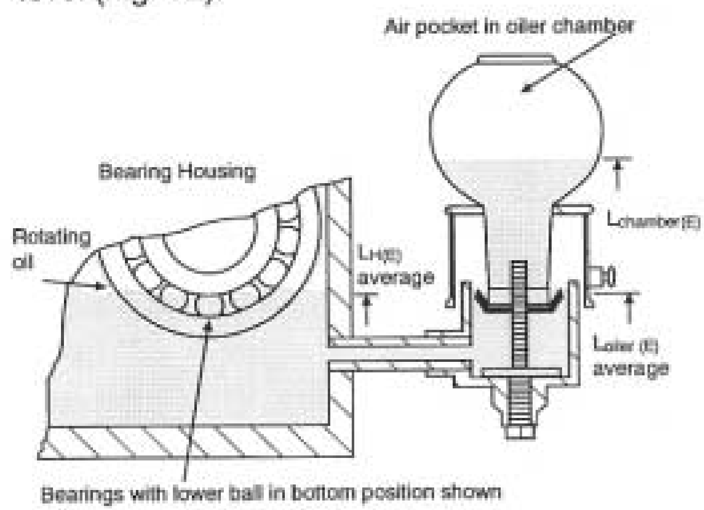


Figure 3E: Stabilized operation. Turbulent oil, with effective average level higher than designed.

This resulting higher oil level will cause increased heating due to the churning action of the oil-bearing contact and, in turn, will negatively affect bearing life (as will be shown later).

Let's designate the difference between this artificially high oil level and design level, as ΔL, illustrated in (Fig. 4).

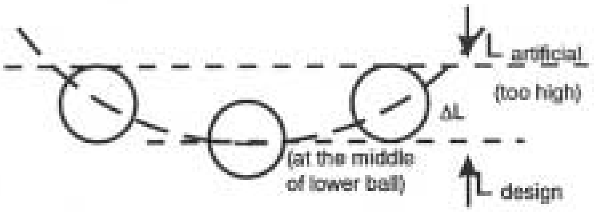


Figure 4: Overflooded bearings due to oiler "burping" effect.

Note that in cases when the oiler option is chosen (versus sight glass) the setting of the oil level is "blind". A maintenance mechanic sees the oil in the oiler, and therefore is unaware of the oil level being too high at the housing. Obviously, the sight glass option would solve this problem, since no bottle oiler is present i.e., no "burping", and the oil level can be properly adjusted when the pump is started.

Furthermore, even if the oiler option is chosen for the MTX frame, the turbulence of oil in the frame would be less due to increased free surface (as well as greater depth). This would have a dampening effect on turbulence.

In order to quantify the effect of increased free surface on bearing life, the following testing was conducted. An MT frame was used for the tests, equipped both with the oiler and special oil level gauge, for visual verification of the level.

First, the "ΔL-testing" (Fig. 4) was done. The frame was filled with oil via bottle oiler, and the level in the oiler chamber was recorded. The pump was started and ran until all oiler "burping" stopped. The level was checked again, and was found to be increased by approximately $\frac{1}{16}$in.

The question became: How to *quantifiably* relate this data to bearing life?

To answer this, another set of tests were conducted for each frame, to determine the effect of ball submergence on oil and bearing temperature, and, correspondingly, on bearing life.

Five oil levels were studied, (Fig. 5).

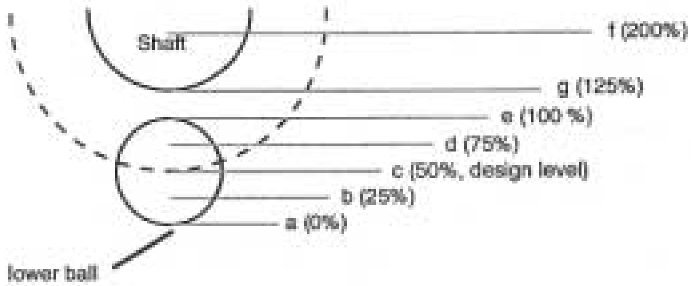


Figure 5: Bearing oil submergence test, for effect on oil temperature.

a = oil barely touches the lower ball (designated as 0%)

b = oil level between bottom and center of the lower ball (25%)

c = middle of the ball (50%), design setting

d = 75%

e = 100% (ball just completely covered)

f = 125% (oversubmergence)

g = 200% (almost half of the shaft covered)

Tests were run for 4 hours each, to ensure a steady state temperature was reached(Fig. 6).

As expected, the increased level of oil covering the lower ball caused an increased oil temperature. The bearing temperature was also directly measured and found to be very close to the oil temperature.

Note also that the temperature decreases significantly for the oil level below design, 50%. This is due to reduced churning of oil in the sump, and the existence of an initial oil film within the bearing itself. However, continuous operation in this mode is not allowed. Eventual oxidation and deterioration of oil film in the bearing without a sufficient supply of fresh oil, will quickly cause bearing failure.

Before we derive the relationship between bearing life and the oil level (as shown to vary due to oiler "burping"), it is important to briefly review the basics of bearing life calculations.

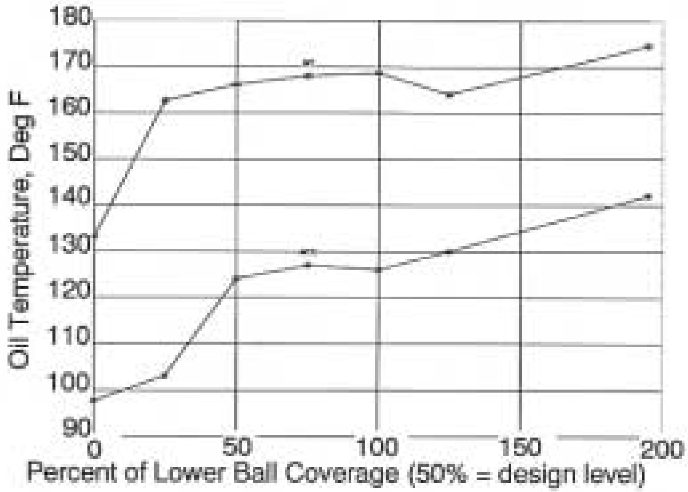


Figure 6: Oil temperature as a function of oil level at the lower ball.

2.2 Bearing life (as designed)

In the following analysis we will use data from the SKF bearings manual (Ref. [3]).

Both MT/MTX frames use 6309 radial bearing, and 5309 A-C_3 thrust bearing. The thrust bearing operates under higher load, and therefore determines bearing life minimum requirements.

The maximum design loads imposed on this bearing are the radial load, F_R=400 lbs, and the axial load, F_a=900 lbs. These are caused by pump hydraulic loads (worst case), transmitted to the bearings due to the internal pump pressure distribution.

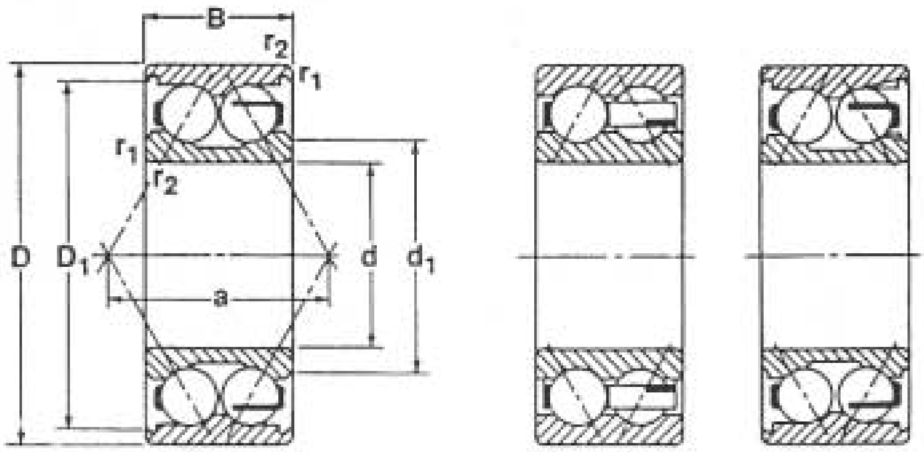


Table 2									
Principal dimensions			**Basic load ratings**		**Fatigue load limit**	**Speed ratings lubrication**		**Mass**	**Designation**
			dynamic	**static**					
d	D	B	C	C_0	Pu	grease	oil		
mm in			N lbf		N lbf	r/min		kg lb	
45 1.7717	100 3.9370	39.69 1$\frac{9}{16}$	72,800 16,400	53,000 11,900	2,240 504	4,500	6,000	1.15 2.55	5309A

Table 2: Data for the 5309 A thrust bearing, used for MT/MTX frames (Ref. [3], courtesy of SKF)

Dimensions					**Abutment and fillet dimensions**			**Calculation factors**					
								dynamic					**static**
									Fa/Fr=e		Fa/Fr>e		
d	d_1	D_1	$r_{1,2}$ min	a	d_a min	D_a max	r_a max	e	X	Y	X	Y	Y_0
					mm in	mm in	mm in						
45 1.7717	56.0 2.205	91.1 3.587	1.5 0.059	58 2.283	54 2.2126	91 3.583	1.5 0.059	0.80	1	0.78	0.63	1.24	0.66

Table 2 p.10 contains basic parameters for the 5309 A bearing. The calculations for the bearing life usually done for basic rated life, for 90% reliability, L_{10} and adjusted rating life L_{na} (Ref. [3], p.40):

$$L_{10} = \left(\frac{C}{P}\right)^3 \frac{10^6}{60xRPM} \text{ (hrs), where}$$

C is basic dynamic load,
P is an equivalent dynamic rating.
(C=16,400 lbs., for 5309 A bearing):

$L_{na} = a_1\, a_2\, a_{23}$ or $L_{na} = a_1\, a_{23}\, L_{10},$
where
a_1 = 1 for 90% reliability calculations (Ref [3], p.31), and
a_{23} is a material and lubrication factor, obtained from charts.

The equivalent dynamic load can be calculated as follows (Ref. [3], p. 42):

$P = XF_r + YF_a,$
and for 5309 A bearing (Table 2) it is equal to:

$P = 0.63 \times 400 + 1.24 \times 900 = 1368$ lbs.

The required oil viscosity for adequate lubrication (Fig. 7) can be determined from Diagram 1 of (Ref [3], p. 32), and is equal to υ_1 = 8.2 cSt. The oil used for MT/MTX frames is ISO VG68 high quality turbine oil and recommended temperatures range from 120°F to 180°F. (Fig. 8) shows Diagram 2 from the same reference [3], showing viscosity versus temperature. Using 120° F, we will get υ= 45 cSt actual viscosity, which would give us a K (kappa) factor of

$$K = \frac{\upsilon}{\upsilon_1} = {}^{45}\!/_{8.2} = 5.5$$

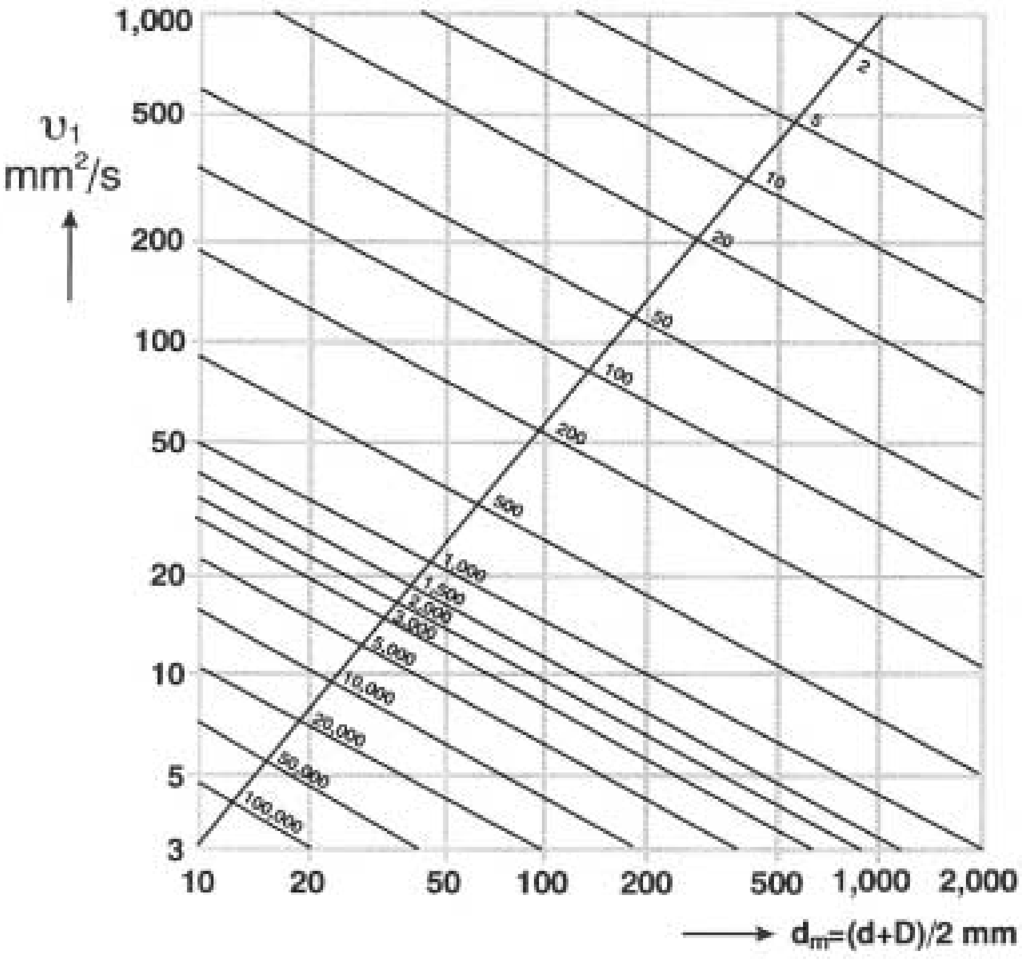


Figure 7: Minimum required oil viscosity for a given bearing d_m. (Ref. [3], p.32)

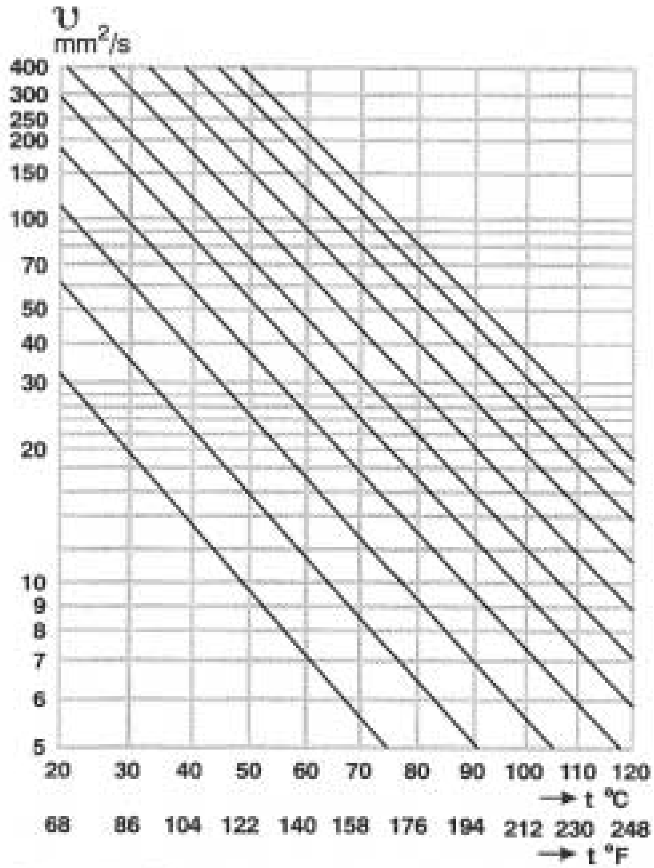


Figure 8: Oil viscosity vs. temperature. (ref. [3], p.33)

The coefficient a_{23} can now be read from the chart in Table 4 (Ref. [3], p. 34, Diagram 3):

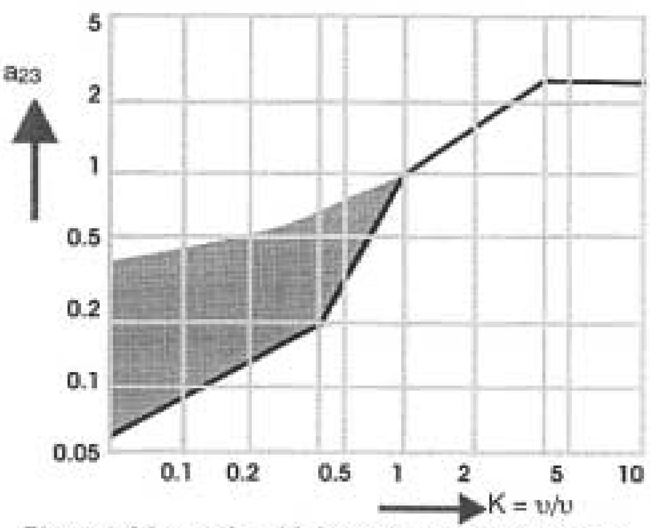


Figure 9: Material and lubrication coefficient versus viscosity factor. (Ref. [3]. p.34)

For our K = 5.5, a_{23} = 2.5

We can now calculate the bearing life,

$$L_{10} = \left(\frac{16{,}400}{1368}\right)^3 \frac{10^6}{60 \times 3580} = 8021 \text{ hrs.}$$

and

L_{na} = 2.5 x 8021 = 20,052 hrs. ≈ 2.3 years.

This value exceeds the minimum required bearing life of 17,500 hrs, per ANSI specification (Ref. [2]). It is possible to construct a relationship between the oil temperature and bearing life, for any bearing. Fig. 10 shows such a relationship using Figs. 8 and 9.

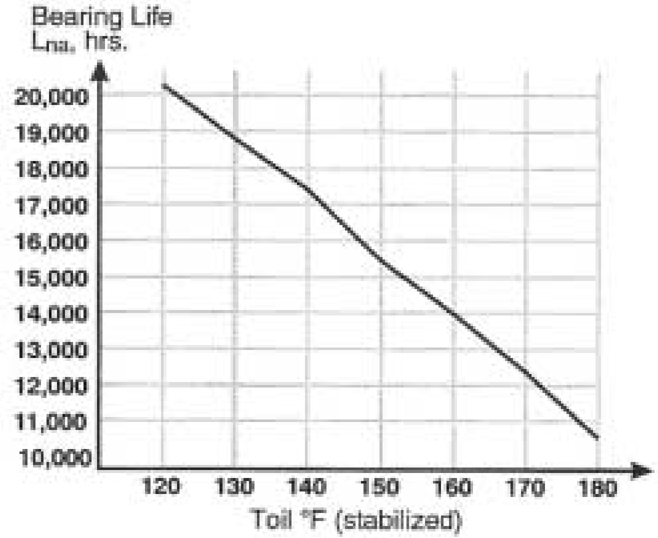


Figure 10: Bearing life, L_{na}, as a function of oil temperature.

The value of 17,500 hours (as stated in the ANSI specification), corresponds to approximately 140°F oil temperature. In this range (Fig. 10) the increase in each 10°F represents approximately 2000 hours decrease in bearing life, or

$$\frac{2000}{17{,}500} \times 100\% = 11.4\%.$$

If we combine Fig. 6 with Fig. 10, a relationship between the ball submergence in oil and bearing life becomes evident, (Fig. 11):

Using 50% oil coverage as a design value, with ±10% as tolerance margin, (Fig. 11) provides a useful relationship for determining the optimum lubrication regime. It indicates too low a level (film deterioration), or too high a level (possibility of frame seal leakage).

The difference in values for bearing life between MTX and MT frames due to differences between the actual oil operating levels in these two frame designs was determined from Fig. 11. As was shown in section 2.1, the oiler "burping" effects were analyzed and tested. The ΔL level for a MT frame was 1/16". (Fig. 4).

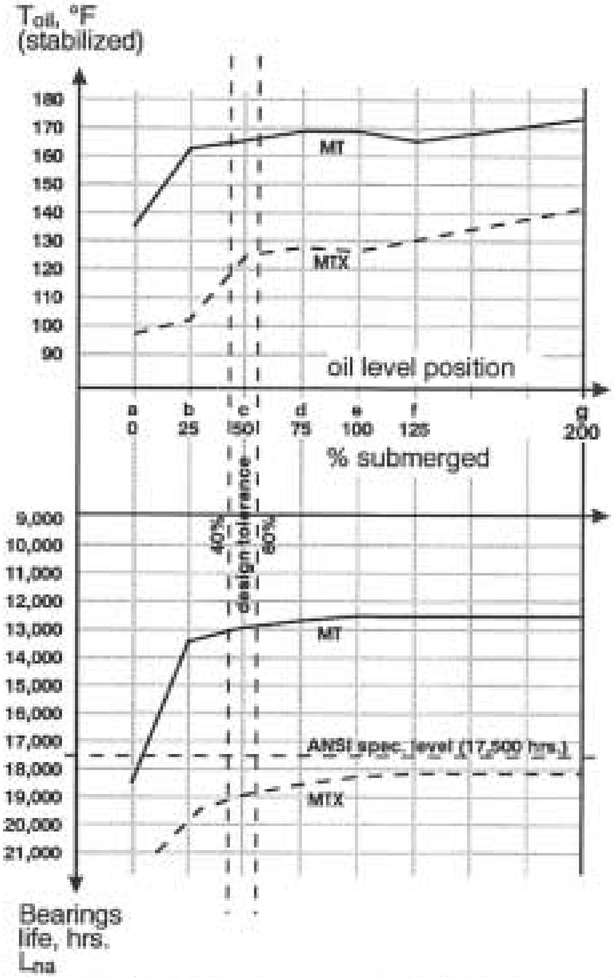


Fig. 11: Relationship between ball submergence and bearing life. (Note: chart given for composition purposes. Actual design values are higher for both MT and MTX).

Obviously, since the MTX frame comes with the sight glass, no variation in ΔL level need be accounted for, since the sight glass allows visual control over the oil level.

$\Delta L = \frac{1}{16}$" corresponds to approximately 59% level (or 9% over design level) for the ball size of 5309 bearing being approximately 0.69" diameter, (Fig. 12). This corresponds to approximately 400 hours decrease in bearing life, per (Fig. 11).

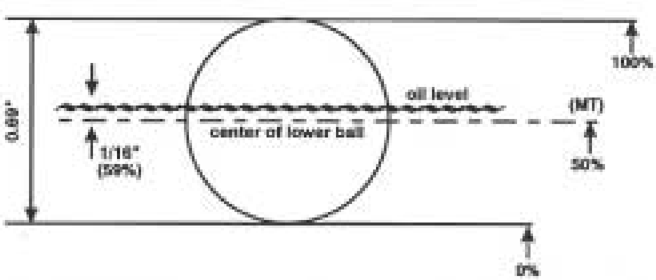


Fig. 12: Oil submergence change due to oiler "burping".

Again, using 17,500 hours as a design bearing life, this decrease corresponds to

$$\frac{400}{17500} \times 100\% = 2.3\% .$$

We can now say that the referenced "burping" effect due to oiler means 2.3% life decrease as compared to MTX. In actual installations, even higher levels were reported, so that 2.3% is possibly conservatively low.

2.3 Effect of deeper sump on oil oxidation rate

The new design (MTX) sump is considerably larger in volume (nearly tripled); the dimensional envelopes of oil in MTX and MT frames are shown in Table 3:

Greater sump volume (more oil) has deterring effects on oxidation of oil. Since oxidation takes place mostly at the layer of oil near the free surface and propagates into the oil with time, oxidized oils are present in lower amounts in a larger sump case. The oxidation of oil chemically transforms its most unstable constituents into acids, resins and other residues (Ref. [4], p.514). The accepted rule of thumb is that an increase in oil temperature of 10°C (18° F) doubles the oxidation rate. A prominent "Installation and Operation Manual" (Ref. [5]), recommends an oil change at least every 6 months. As was shown in section 2.2, the MTX design results in approximately 40°F cooler oil, (Fig. 11), at design oil level. To achieve the same rate of oxidation for the MT frame as for MTX, the oil would need to be changed more frequently; its oxidation rate would be

$1 + \frac{40}{18} = 3.2$ times greater.

The frequency of oil change would, correspondingly be $\frac{6}{3.2} = 1.9$ months, or the larger 6.4 times a year, versus 2 times a year for the largest MTX frame.

Typically, an oil change takes 15 minutes, which, at a rate of $60/HR, is $15, plus the cost of oil, say, of $5, to a total of $20 per change. The net savings would then be:

$20 x (6.4 - 2) = $88 per pump per year.

Table 3: Oil Sump Comparison			
	MT	MTX	Ratio = $^{MTX}/_{MT}$
Sump width, in	5.42	5.94	1.13
axial length, in.	4.75	4.75	1.0
sump depth, in.	1.18	2.75	2.33
(approx.) volume, in.^{3} (cm^3)	29.4 (482)	77.6 (1269)	2.63
(actual) volume, in.^{3} (cm^3)	23.4 (384)	76.3 (1250)	3.26
(Note: dimensions are approximate, due to simplification of internal geometry to a "block" shape. Actual volumes were measured by filling with oil).			

2.4 Effect of sump depth on contamination concentration

The deeper sump allows contaminants to settle at the bottom, allowing bearings to operate in a cleaner oil environment. As was shown in Table 3, the new sump design is 2.3 times deeper. To quantify the effect of contamination reduction, we will again refer to the SKF catalog (Ref. [3]):

Assume that an MTX frame contains oil in average concentrations allowing clean operation (η_c = 0.8), and an MT operates in a somewhat contaminated regime (η_c=0.5).

Using the values for bearing fatigue limits from Table 2, we can calculate the quantities:

$$\eta c \frac{Pu}{P} = 0.8 \quad \frac{504}{1368} = 0.29 \text{ (MTX)}$$

and

$$\eta c \frac{Pu}{P} = 0.5 \quad \frac{504}{1368} = 0.18 \text{ (MT)}$$

The life correction factors a_{SKF} can be read from Ref [3]. pg. 36, which is reproduced in Fig. 13.

For clean oil, $a_{SKF} \approx 50$, and for contaminated $a_{SKF} \approx 24$, with a ratio of $\frac{50}{24} \approx 2.1$.

This illustrates that cleaner oil can potentially more then double bearing life.

Obviously, more frequent oil changes as discussed in the previous section would reduce the effects of contamination. Similarly, a larger sump is a good way to directly reduce the effects of oil contamination on bearings life.

Table 4: Effects of oil contamination (Ref. [3], p.40, courtesy of SKF)	
Values of adjustment factor ηc for different degrees of contamination (SKF catalog)	
Condition	ηc *
Very Clean Debris size of the order of the lubricant film thickness	1
Clean Conditions typical of bearings, greased-for-life and sealed	0.8
Normal Conditions typical of bearings greased-for-life and shielded	0.5
Contaminated Conditions typical of bearings without integral seals, coarse lubricant filters and/or particle ingress from surroundings	0.5 .. 0.1
Heavily contaminated **	

* The scale for η_c refers only to typical solid contaminants. Contamination by water or other fluids detrimental to bearing life is not included

** Under extreme contamination, values of η_c can be outside the scale, resulting in a more severe reduction of life than predicted by the equation for L_{naa}

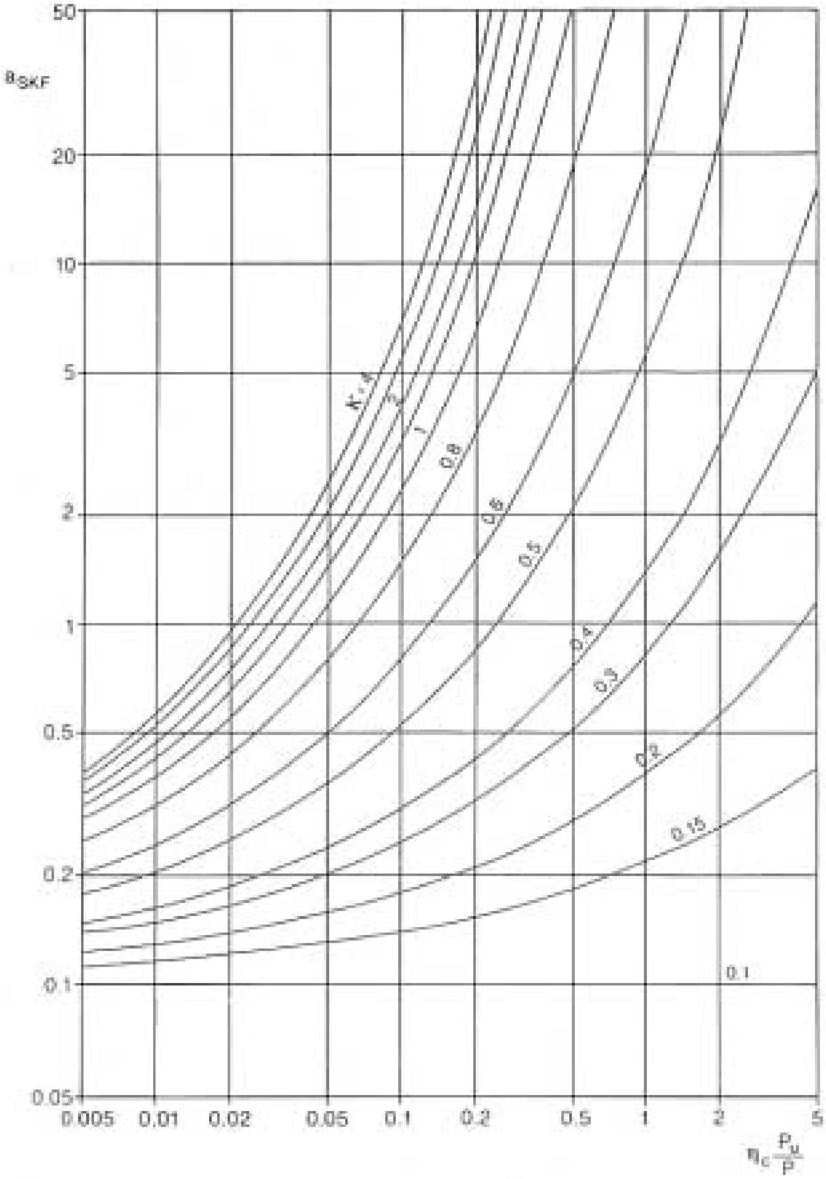


Figure 13: a_{SKF} correction factors due to oil contamination. (ref [3], p. 36)

2.5 Effect of increased heat transfer surface

Friction from the bearing motion results in heat generation. This heat is rejected to the oil, and must be carried out to the outside air via conduction through the frame walls, and convection to the outside air. Some of the heat also being transported to the pump casing via shaft and bearing spacer piece. For the cases of pumpage temperatures in the range of 120 - 180°F, this axial conduction is relatively small, due to the small temperature gradient between the bearing frame (heated by oil) and the casing (heated by pumpage).

In this study, we will assume that all heat is transported to the outside air via bearing frame outside surfaces.

First, let's calculate the amount of heat generated by the bearing, using (Ref. [3], p. 46).

NR = $1.05 \times 10^{-4} \times M \times RPM$ (heat),

where
M = 0.5 MFd,
and
M=0.0015, for deep groove ball bearings,

F = 6080N (1368 lbs.), bearing load,
d= 45 mm,
so that
$N_R = 1.05 \times 10^{-4} \times (0.5 \times 0.0015 \times 6080 \times 45)$

$$\times\ 3580 = 120\text{ W} = 432\,\frac{BTU}{HR}$$

The radial bearing also generates heat. Usually it is approximately 25% of the thrust bearing, i.e.

$$432 \times 0.25 = 108\,\frac{BTU}{HR},\ \text{and total heat is}$$

$$432 + 108 = 540\,\frac{BTU}{HR}$$

The heat transfer surfaces of the bearing housing can be approximated as a series of cylinders, with areas as shown on (Fig. 14).

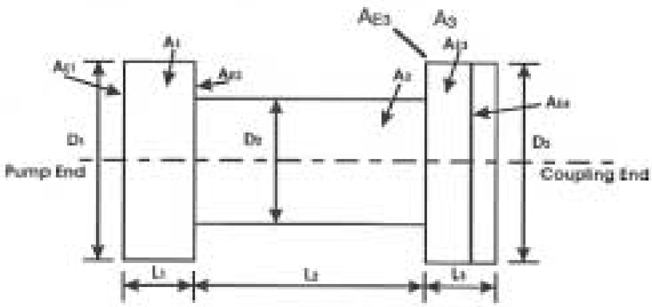


Figure 14: Bearing frame geometry approximations for heat rejection analysis

The surfaces (cylindrical and ends) (Fig. 13) can be calculated as follows:

$$A_1 = \pi_1 D_1 L_1 \qquad A_{E2} = \frac{\pi_1}{4}(D^2_1 - D^2_2)$$

$$A_2 = \pi_1 D_2 L_2 \qquad A_{E3} = \frac{\pi_1}{4}(D^2_3 - D^2_2)$$

$$A_3 = \pi_1 D_3 L_3 \qquad A_{E4} = \frac{\pi_1}{4} D^2_3$$

$$A_{E1} = \frac{\pi_1}{4} D^2_1$$

Table 5 lists measured approximate dimensions for the surfaces in (Fig. 13) for MT and MTX frames:

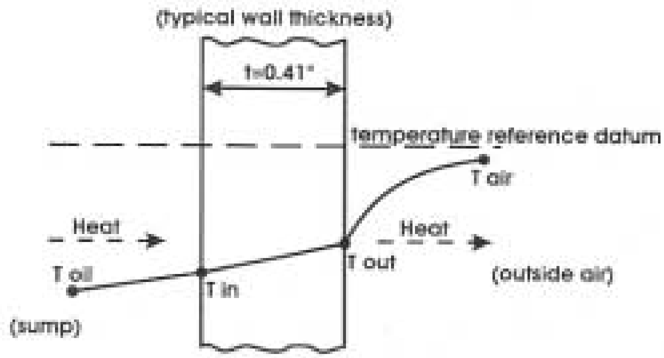
Table 5: Data for bearing frame surface

	MT	*MTX*
D_1, in.	11.0	11.0
D_2	7.5	8.5
D_3	7.0	10.0
L_1	.5	.5
L_2	7.0	6.0
L_3	.5	1.5
A_1, in^2	17.3	17.3
A_2	164.9	160.2
A_3	11.0	47.1
A_{E1}	95.0	95.0
A_{E2}	50.9	38.3
A_{E3}	5.7	27.8
A_{E4}	38.5	78.5
A_{TOTAL}, in^2	383.3	464.2

The MTX frame has $\frac{464.2}{383.3} = 1.21\ (21\%)$ greater total heat transfer area of the bearing frame outer surface. This results in cooler surface and cooler oil, as can be seen from the following calculations.

The mechanism of this heat transfer is shown in (Fig. 15):

Figure 15: Heat transfer mechanism



Assume that for practical purposes
Tin = Toil, and write the conduction equation through the housing wall:

$$Q = k\,A_{in}\,\frac{T_{out} - T_{in}}{t}, \text{ where } k \approx 34.9$$

$$\frac{BTU}{HR \times FT \times {}^{o}F}$$ for iron (ref [6], p.635)

For convection:

$$Q = hA_{out}\,(T_{out} - T_{air}),$$

where

$$h \approx 2.7\,\frac{BTU}{HRFT^{2\,o}F}$$ (Ref [7], p. 385)

Let's assume for simplicity

$A_{in} = A_{out} = A_{TOTAL}$ (from Table 4), and outside air temperature

$T_{AIR} = 100°F$ (summer).

Calculations for temperature distribution are now straightforward:

MT frame:

$$T_{OUT} = T_{AIR} + \frac{Q}{hA} = 100 + \frac{540}{2.7\ x\,\frac{383.3}{144}} = 175.1°F$$

$$T_{IN} = T_{OUT} + \frac{Qt}{kA} = 175.1 + \frac{540\ x\,(^{41}/_{12})}{34.9\ x\,\frac{383.3}{144}} = 175.3°F$$

MTX frame:

$$T_{OUT} = 100 + \frac{540}{2.7\ x\,\frac{464.2}{144}} = 162.0°F$$

$$T_{IN} = 162.0 + \frac{540\ x\,(^{.41}/_{12})}{34.9\ x\,\frac{464.2}{144}} = 162.2°F$$

It is clear from the above calculations that the oil temperature is lower for the larger sump design (MTX) by
175.3 - 162.2 = 13.1°F,
just due to the effect of the increased outside heat transfer surface of the housing.

In (Fig. 10), this translates to approximately 2200 hrs difference in bearing life, or, using 17,500 hrs as nominal design life,

$$\frac{2200}{17{,}500} \times 100\% = 12.6\% \text{ improvement.}$$

2.6 Summary of above effects

In the above sections (2.1 through 2.5) the following was demonstrated:

a) Free sump surface extension (section 2.1, 2.2): increases bearing life by 2.3%

b) Deeper sump decreases oil oxidation concentration and results in savings of approximately $88 per pump per year. (section 2.3)

c) Deeper sump also reduces oil contamination at the bearings and improves life 2.1 times (section 2.4)

d) The increased heat transfer surface keeps the oil cooler and extends bearings life by 12.6%.

The next section quantifies the above data in terms of economic benefits.

3. ECONOMIC BENEFITS

In section 2.6 it was shown that the new X-Series design improves bearing life by a total of approximately 124.9%, plus an additional $88 per pump per year savings due to improved oil conditions.

These savings can indeed be materialized since they apply to **real** pump installations in a realistic field operating environment.

However, lab testing comes in handy when a particular feature of a pump such as sump volume is compared with another pump. Such testing can examine oiler setting effect on temperature, bearing submergence effect, and other readily identifiable and quantifiable data.

Additionally, it is important to account for benefits due to decreased pump downtime (as pumps get more reliable). This obviously improves plant production output. Let's examine this briefly.

ANSI specification stipulates a 2-year (17,500 hrs) bearing design life. This means that the number of bearing changes due to scheduled and unscheduled maintenance is about 0.5 per year. With the X-Series frame improving bearing life by 124.9%, this constitutes almost 4.5 years, or 0.22 bearing changes per year. Assuming that typical maintenance on a pump takes about 4 hours at $60 per hour, it is easy to estimate yearly savings:

(60 x 4 + 20) x (0.5 - 0.22) = $72.80

(an average of $20 was assumed as the cost of bearings). Adding to this $88 savings due to cleaner oil, the total is:

$72.80 + $88 = $160.80, per pump per year.

A typical plant may have several thousands pumps installed. Assuming there are 3000 pumps at the average size plant, the total maintenance savings would amount to

$160.80 x 3000 = $482,400 per year.

Plant maintenance departments can best estimate additional savings due to improved production as related to increased pump reliability. For example, the following logic can be applied:

For a typical plant, each hour of forced downtime can cost, say, $500 in lost production. The MTX frame, with 0.22 outages a year for 4 hours each, accrues 0.88 hours of lost production per pump per year. The MT frame has 0.5 outages, or 0.5 x 4 = 2 hours downtime.

The savings are:

500 x (2 - 0.88) = $560 per pump per year,

and again, using 3000 pumps per plant,

560 x 3000 = $1,680,000 for the plant!

Obviously, as was stated earlier, a more accurate assessment can be done locally by each individual plant using their own data. The general trend in savings nevertheless is clear.

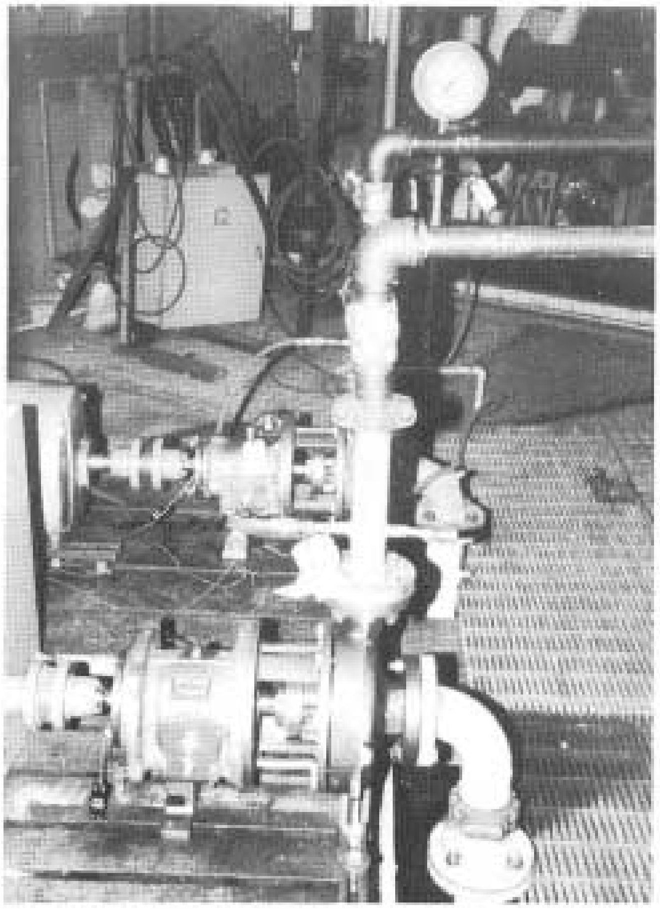
4. TESTING

A testing program was conducted at the Technology Center to verify the theoretical calculations and to support with data the assumptions made in this paper.

Two ANSI 3196 pumps were tested, having two different power ends, but with identical hydraulics. The first was an older design, MT frame, and the second was an MTX frame (X-Series). (Fig. 16) is a side-by-side comparison of these two designs.

The following tests were conducted:

1. Effect of oil level on bearings and oil temperature.
2. Oiler "burping" effect, increasing oil level in the sump.



4.1 Effect of oil level on oil temperature

As discussed in the previous sections, the level of oil in the bearing housing, covering the lower ball to different submergences, causes increased churning effect, and results in increased oil and bearing temperature, leading to the bearing life reduction (Fig. 11).

The oil level was controlled by visual observation via sight glass with calibration marking for 0%, 25%, 50% (design level), 75%, 100%, 125% and 200%, (Fig. 17 and 18).

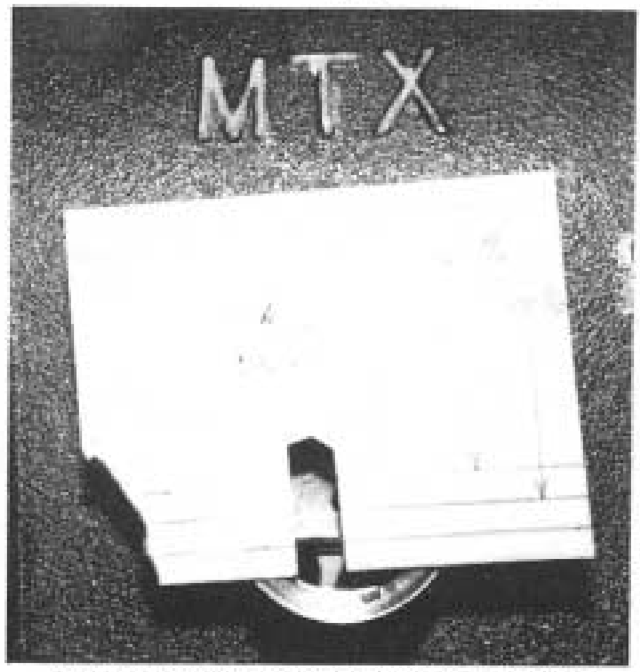


Figure 17: MTX oil level calibration template.

Figure 16: Comparison of new (MTX) (foreground) and old (MT) designs.

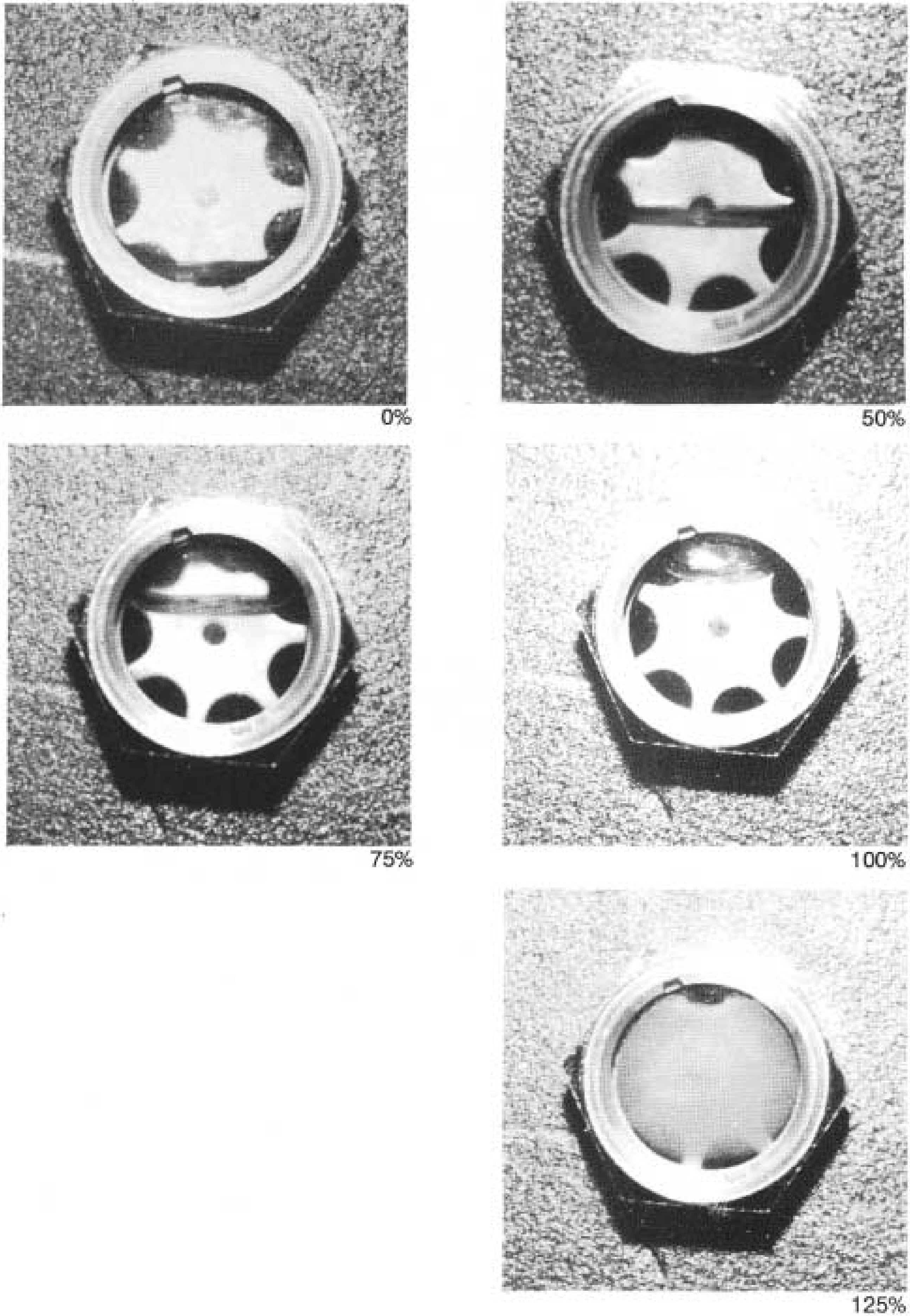


Figure 18: Oil level (MTX frame) as seen through the sight glass.

The old design does not have a sight glass provision. A special L-shape oil level gauge was used to indirectly measure the oil level for this case, (Fig. 19).

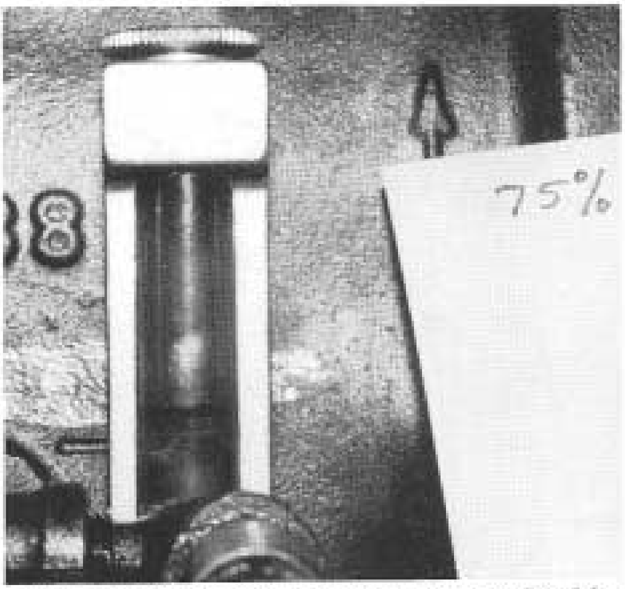


Figure 19: Oil level gauge used to measure level for the MT frame.

In this testing, constant level oilers were not installed. This eliminated the effect of oiler "burping", and allowed concentrating on the oil level effect on temperature.

Pumps were run for 4 hours at each oil level setting, until steady state equilibrium temperatures were reached. Thermocouples were imbedded in the oil and at the bearing outer race. The steady state temperature of oil and bearing were found to be very close to each other (Fig. 11).

4.2 Oiler "burping" effect

An MT frame was equipped with an oiler and a special oil level gauge, instrumented (Fig. 20.)

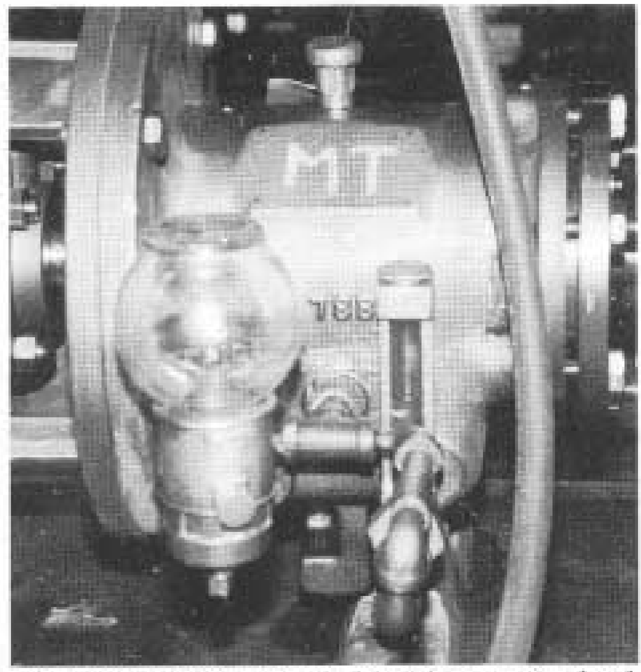


Figure 20: MT frame equipped with a constant level oiler and a special level gauge, for "burping" testing.

The pump was started and ran until oiler "burping" stopped, which took approximately 10 minutes. The pump was then stopped and restarted again, causing additional "burping." After 3 such cycles, no more burping was observed, and the final oil level in the housing increased by approximately 1/16in., as discussed in Section 2.2 and (Fig. 12). This data is believed to be conservative, since the additional piping which was required to install level gauge (with elbows, nipples, etc.) causes a restrictive dampening effect on the oil turbulence, or "waviness", (Fig. 3). The above instrumentation method was necessary simply because the MT frame provides no other method to **directly** control oil level. This is another example of the MTX (with sight glass as standard) design advantage.

Therefore, a 2.3% life decrease due to the oiler burping is believed to be conservative and may probably reach 6 - 7% for the actual particular installation in the field.

5. SUMMARY

This paper presented some of the results of work done at the Technical Center of a pump manufacturer, on the subject of improving pump reliability, and its impact on MTBSM and MTBF. Quantifiable data were obtained, comparing two designs - the old design and a new X-Series power end.

Studies continue on other elements of pumping systems, to similarly quantify their effect on both components life improvements and the economic benefits, associated with such improvements.*

REFERENCES

1. "Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process", American National Standard, ASME B73.1M-1991 (revision of ANSI/ASME B73.1M-1984)
2. "Installation, Operation and Maintenance Instructions", Model 3196, Goulds Pumps, Inc., 1992.
3. "SKF General Catalog" (bearings), 1991
4. "CRC Handbook of Lubrication", Vol. I, CRC Press, R. Booser, 1983
5. "Installation and Operation Manual", Supplement, Goulds Pumps, Inc., 1992
6. Hollman J.P., "Heat Transfer", 6 Ed. McGraw-Hill, 1986
7. Shigley, T.E., "Mechanical Engineering Design", 3rd Edition, McGraw-Hill, 1977
8. "Proper oiler operation and common mistakes", Goulds Technology note, Issue 2, (VHS tape, 17 min.) May 1993.

*Note:These data are periodically reported in "Goulds Technology Review" issues: publications and video illustrations, (such as Ref. 8).

GOULDS PUMPS, INC.