

## INSIDE THE SEALLESS PUMP

by

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### ABSTRACT

Sealless pumps are now specified as the preferred type for handling of toxic, hazardous, and environment threatening chemicals. Magnetically driven sealless pumps have emerged as the first choice, due to their simplicity of design and ease of maintenance.

Much is written of the importance of pump/system interaction in ensuring maximum pump life. Inside of the sealless magnet drive is a complex internal flow system which is difficult to model, but is crucial to the successful operation of this type of machine, due to the need to effectively cool the magnetic drive, lubricate the bearings, and safely transport solids. With low specific heat, volatile liquids, a clear understanding of the internal flow system is essential if dangerous flashing resulting in premature pump failure is to be prevented.

The results of tests undertaken on a number of different internal feed systems are presented. Plexiglas models, flow visualization, and pressure profile analysis were all carried out to provide a solution to give optimum performance under widely varying pump conditions. The interaction of the main internal flow path with that to the bearings is also reviewed.

It is hoped that a clearer understanding of the inner workings of sealless machines will allow closer cooperation between the supplier and the customer in applying these types of pump correctly to fluids of various characteristics with maximum success.

## INTRODUCTION

Environmental concerns and the need for reliable, low maintenance equipment has fueled the move to sealless pumps as the preferred liquid mover in the chemical, petrochemical, and process industries.

Simplicity and ease of maintenance, coupled with the ability to use standardized prime movers has resulted in the magnetically driven sealless pump being that most universally installed.

As early as 1933, patents were in existence covering the design of magnet drive pumps, but it was not until the early 1950s that these became commercially available.

In these early years, the competitive battle was between canned motor and magnet drive sealless pumps with canned motor pumps having the majority of the market due to the limited power transfer of magnetic drives. The development of the canned motor pump reached a ceiling in the early 1970s with little innovation after that time.

The availability of increased power magnets and improved bearing materials allowed continued development of magnet drive pumps, which has resulted in a phenomenal growth in the installed base of this type of machine and a proliferation of designs and suppliers.

The increased supplier base has correctly prompted a healthy dialogue between user and manufacturer to allow users to compare different types of designs rationally.

When applied to hazardous fluids, users have understandable concerns when moving from mechanical seal technology with its fully understood failure modes to sealless pumps that seem to offer a panacea for all evils. Clearly, sealless pumps are not the answer for every pumping problem. To expand the range of applications to which they may be applied, manufacturers must be prepared to explain the design procedure behind the products offered, so that the user may contribute his process knowledge to the successful application of the pump to a particular service.

In 1989, the authors' company and a U.S. firm began the development of a new range of magnet drive pumps for the North American market in recognition of the need for pumps to ANSI B73.1 performance and dimensions.

A precondition of the project was that the authors should thoroughly investigate areas of design which have been proven to be crucial to the successful operation of pumps in service. The internal bearing and internal feed of the pump would receive critical attention. Bearings, because of their fundamental importance, and internal feed, because of its function in providing lubrication to the bearings and cooling of the magnetic drive would be investigated. The wide variety of potential liquid which would be used in this cooling circuit dictates a clear understanding of internal flow distribution and pressure.

Bearings were tested both inside and outside of the pump with various media.

Internal flow was routed around the coupling in a number of ways. Pressure, temperature, and flow measurements were taken, but due to the complex nature of the flow channels, complete understanding was only possible by the use of plexiglas models to allow visual observation of flow distribution.

An outline is provided of the design procedures used in the development of sealless machines and of the major areas of customer concern are discussed.

## THE DESIGN PROCESS

When designing a sealless pump there are five main elements to be considered: 1) the liquid end, comprising pump casing and impeller; 2) the magnetic drive consisting of an inner and outer magnet assembly and the containment shroud; 3) the internal support bearings; 4) the internal feed system which circulates

between 1), 2), and 3), which is required to cool the magnetic drive, lubricate the bearings, and transport any solids in suspension; and, 5) the power frame which comprises the external bearings supporting the outer magnet and the interface to the prime mover.

A successful design to give reliable service will require careful consideration of all of these elements individually and as interactive assemblies. Each must be designed with a clear understanding of its effect on the whole.

Apart from the Pressure Vessel Code aspects of the containment shroud, any discussion on magnetic drives will be largely proprietary.

Ninety percent of process applications are for single stage pumps for relatively low flow and pressure rise and, as such, numerous papers exist on the design of such machines. When designing sealless pumps, greater knowledge of the dynamic loads both axially and radially produced by the impeller is essential. Also, design for the lowest NPSH possible is a prerequisite in modern process plants.

Power frame bearing design has not been given serious enough consideration in the past, due to the error of considering the outer magnet to be a simple overhung load requiring only simplified support bearing and lubrication. This has resulted in many sealless pumps not achieving their ultimate service life, due to the failure of rolling element bearings and elastomeric lip seals costing only a few dollars. Adequate engineering of this design element and experience of site conditions has now resulted in designs with at least 50,000 hours B10 predicted life.

Two areas of sealless pump design are considered herein little has yet been published or discussed but these areas are critical to successful operation of the finished product. These elements are:

- the internal feed system.
- the internal support bearings.

## INTERNAL FLOW AND BEARING DESIGN

The key to the longevity of a sealless pump, be it canned or magnet drive, is the design of the internal feed system and product lubricated bearings. A clear understanding of the theoretical and practical product limitations is fundamental, along with a firm experimental foundation to ensure correct pump application.

### INTERNAL FEED

The internal feed system of a sealless pump is required to remove heat generated in the drive assembly by eddy current and viscous friction losses, and to lubricate the process lubricated bearings that support the loads exerted upon the rotor assembly. To understand this flow system, it is essential to review the design of the drive and the causes of these losses.

#### *Drive Construction*

The containment shroud is the barrier that separates the product (Figure 1) from the atmosphere and is designed to meet international pressure vessel codes (ASME VIII or BS5500). The accepted standards for materials for this shroud that are generally used in the chemical and petrochemical industry are 316 Stainless Steel and Alloy C. As the inner and outer components rotate, eddy currents are created by the rotating magnetic field cutting through the stationary shroud, resulting in losses and reducing the overall drive efficiency. These eddy currents are a function of:

- Speed <sup>2</sup>
- length of magnet in rotor
- diameter of shroud <sup>2</sup>
- shroud thickness
- field strength <sup>2</sup>

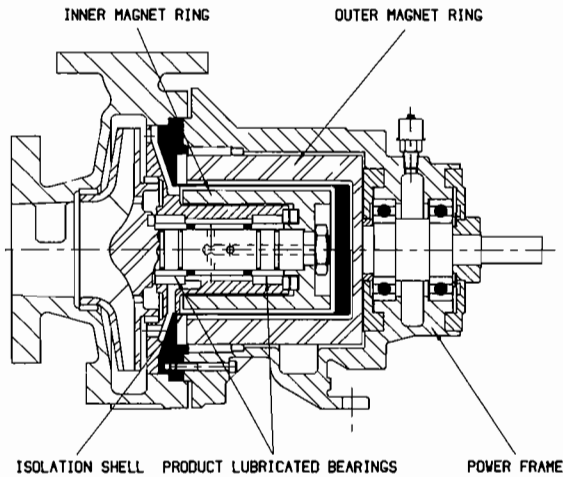


Figure 1. Sectional Drawing of a Typical ANSI B73.1 Sealless Pump.

- resistivity of shroud material<sup>-1</sup>
- number of magnets

The resistivity of the shroud materials commonly used is  $80 \times 10^{-3}$  ohm cm for 316 Stainless Steel and  $130 \times 10^{-3}$  ohm cm for Alloy C. This variation in resistivity between materials will result in a 62.5 percent increase in losses in the 316L assembly over the Alloy C design and the temperature losses will increase pro rata.

Recent advances in the manufacture of ceramic and plastic materials for pressure vessel containment have been introduced into the magnetic drive pump designs. The use of these materials effectively eliminates the eddy current loss. However, the thickness required for the shroud to contain the design pressure, coupled with manufacturing tolerances, result in a significant increase in drive size (typically 50 percent). The corresponding increase in viscous frictional losses together with manufacturing and performance limitations have, until now, limited the application of these materials.

To remove the heat generated in the drive assembly liquid is taken from the discharge of the pump and recirculated through the drive and returned at a lower pressure point within the pump. There are two types of systems that are available to achieve this flow. Each manufacturer will use internal flow configurations based upon one of the two types, although exact details will vary from supplier to supplier.

#### Discharge to Suction Internal Flow System

The simplest feed system available, and at the lowest manufacturing cost, is the internal flow system that takes process liquid from the discharge of the pump and returns the process liquid back to the suction of the pump. Typically, process liquid leaves the exit of the impeller and enters the magnetic drive through a small hole in the rear casing plate. The liquid exiting the impeller has increased in temperature, due to the inefficiency of the impeller. This liquid is then routed through channels and allowed to pass over the drive components, through the annulus, between the inner rotor and shroud. Heat generated in the shroud is removed by this liquid and results in a further temperature rise. At this point, the liquid has undergone a head drop and a corresponding reduction in the net pressure. From the rear of the pump, process liquid is returned to the suction eye of the pump through drillings in the shaft. Process lubricated bearings are fed using ancillary drillings within the pump, designed to ensure adequate lubrication and to ensure that no stagnation points exist within the pump. In some cases, the waste flow is returned into the void behind the back neck

ring and, subsequently, pass to suction through the impeller thrust balance holes (Figure 2).

With this type of feed system, the temperature of the liquid is always returned to the suction of the pump at a higher temperature

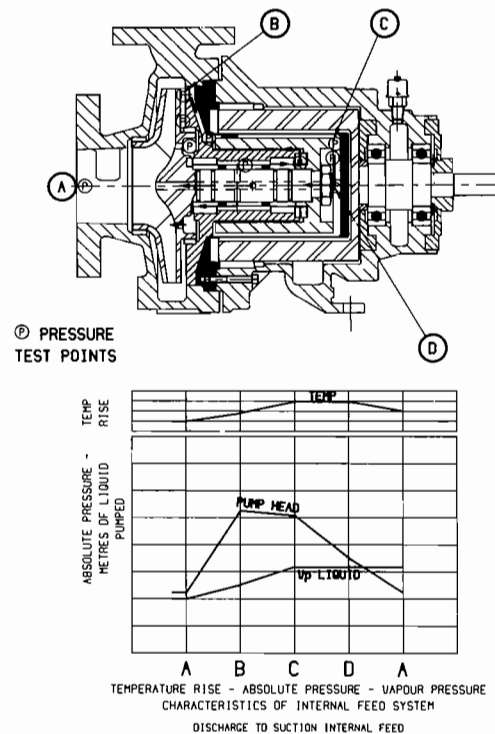


Figure 2. Flow Regime of Pump with Discharge to Suction Flow.

than the bulk suction temperature of the process liquid. Under these circumstances, it is important to ensure that this higher temperature liquid does adversely affect the NPSHR of the pump and vaporize as it returns to the impeller eye. This can easily be overlooked, since most suppliers will test only on water, a liquid with a high specific heat and gradual vapor pressure curve at ambient temperatures, thus masking this problem.

#### Discharge to Discharge Internal Flow System

An alternative system is one where the liquid is taken from the discharge of the pump and returned back at a point of pressure higher than suction pressure. The basic internal feed system is similar to that described for the discharge to suction feed system, with the exception that the liquid is channelled to a high pressure area in the casing, typically behind the back shroud of the impeller (Figure 3). It should be noted that in both cases the total flow through the pump is the sum of a series of smaller flows with heat input at various stages throughout the drive.

The advantage of this construction is twofold. The NPSHR problem is eliminated and the pressure distribution within the drive is related to a high pressure area, not suction, thus reducing the chance of vaporization within the drive. The disadvantage is that the supplier must conduct expensive tests to ensure that there is, under all conditions, sufficient pressure difference to force sufficient flow through the drive. Low flow and excessive temperature rise will have the same effect as high flow and excessive pressure drop, i.e., vaporization.

#### FLOW REGIME DEVELOPMENT

For any new product, or enhancement to existing product, it is essential that the sealless pump supplier fully considers and

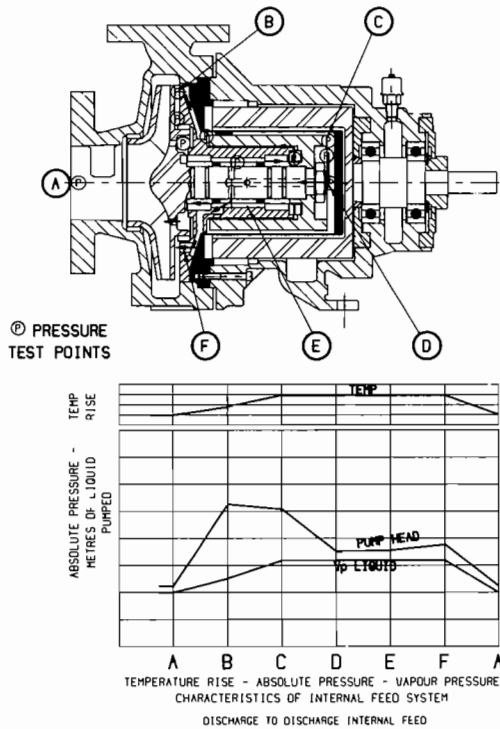


Figure 3. Flow Regime of Pump with Discharge to Discharge Flow.

evaluates the effect of changes on the drive hydraulic performance. This test program must, as a minimum evaluate:

- The flow through every passage within the drive and the corresponding pressure drop.
- The rules governing the flow regimes, i.e., is the flow down the end of the inner magnet ring predominately radial or circumferential? If the flow is circumferential, what are the factors affecting the angular velocity of the fluid?
- What is the effect of varying the coupling type? The drive shown in Figure 3 has an option of three lengths of inner magnet ring.
- What will the effect be of poor assembly, tolerances, flow passage blockage, etc. be?
- What are the rules governing the internal flow of the drive with respect to impeller diameter and impeller flowrate? Does the need to ensure adequate drive flow compromise the operating envelope of the pump?
- When air locked, running shut suction valve (snore), cavitation or loss of suction, how long will the drive survive, and will the gas clear when liquid is reintroduced?

For each of the above conditions, it is necessary to measure the pressure at ten positions within the drive. Much of the testing also requires the use of plexiglas containment shroud assemblies that enable flow visualization tests to be carried out by injecting dye into the drive through the various pressure tappings.

Answering these questions on a range of five, Frame 1 ANSI pumps with three inner magnet rings, has absorbed approximately seven hundred and fifty man hours in testing and analysis, exclusive of manufacture of parts and nonproductive testing.

## TEST RESULTS

A return to discharge flow system with a typical pressure distribution at salient points (A to F) is shown in Figure 3. These

points are the key points of interest within the pump. The pressure is shown in Figure 4 at three of these points, relative to the differential head of the pump. Given a pump head value, it is possible to read across to obtain the head above suction pressure for the point. One point to note is that there is a head rise between point D and E, due to the pumping action of the cross drilling in the shaft. The second key element in estimating the flow and pressure distribution within the pump is shown in Figure 5. It has been found that the flowrate through the coupling is a function of differential head and also impeller diameter. This is due to the pumping action of the vortex between the flow inlet and outlet holes (position B and F). This can be seen in Figure 5 where a series of flows can be obtained for the same discharge head for different impeller diameters. This result can be related to impeller designs where head is a function of outlet diameter and widely differing impellers will give similar closed valve head if machined to the same diameter. Note that this graph is also a function of pump speed.

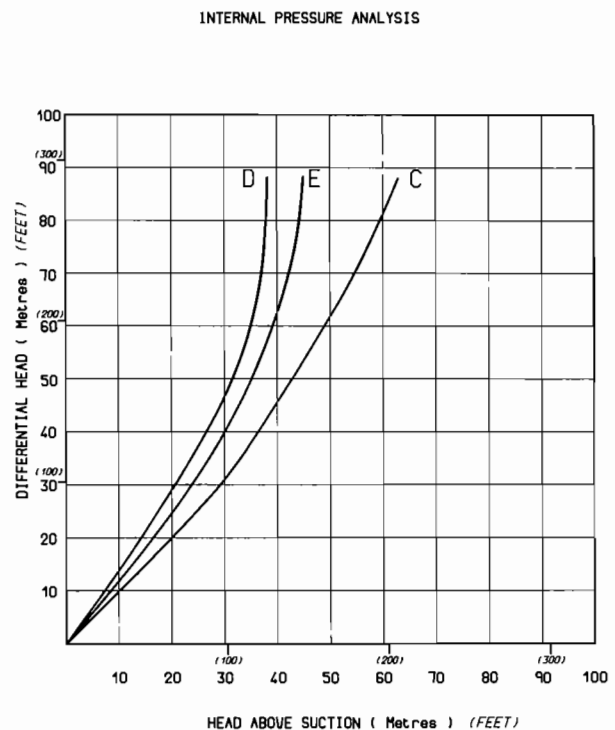


Figure 4. Graph Showing Differential Pressure vs Internal Pressure at Three Positions.

In a similar way, the flow through the journal bearings can be plotted (Figure 6) and through the thrust bearing. From this information, it is possible to calculate the temperature rise of the pump, shroud and bearings and compare the pressure at key positions with the vapor pressure curve for the process fluid.

## WORKED EXAMPLE

To demonstrate and highlight the preceding comments, the following examples have been calculated. The necessary pressures and flowrates can be obtained from Figures 2, 3, 4, and 5. The design examples are based on a range of flowrates for two liquids, water and Arcton 134a (tetrafluoroethane  $\text{CH}_2\text{FCF}_3$ , ICI chlorine free alternative to CFC12 in refrigeration applications). The following fluid properties have been used:

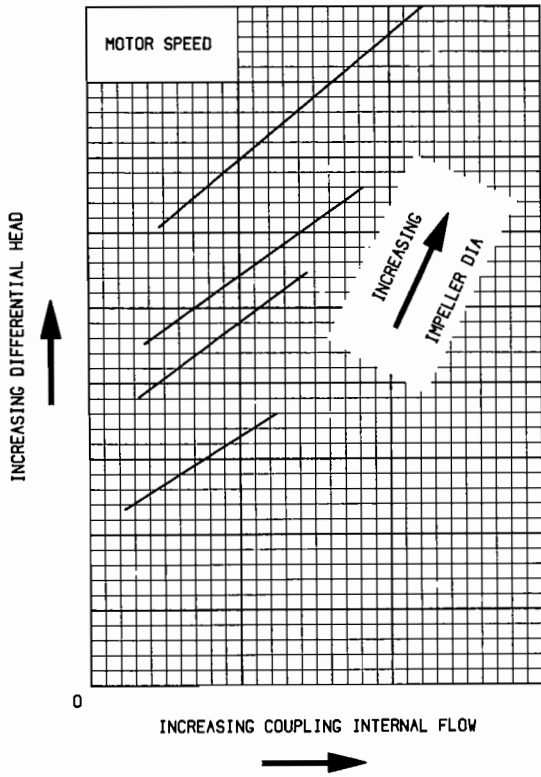


Figure 5. Graph Showing Differential Pressure vs Internal Flow for a Range of Impeller Diameters.

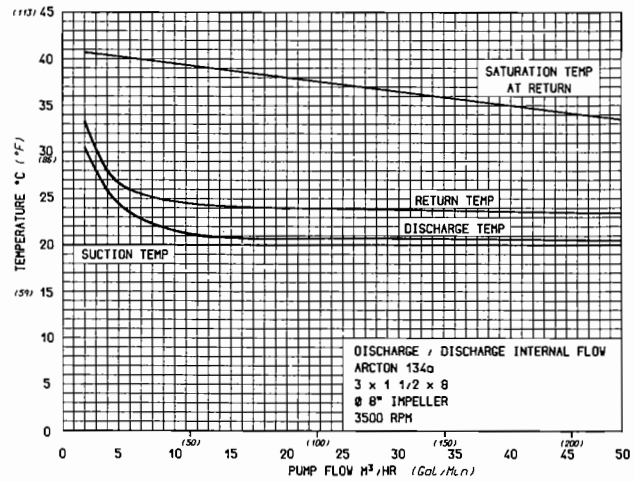


Figure 7. Vapor Pressure vs Temperature Curve for Water and Arcton 134a.

The results of this analysis are shown in Figures 8 and 9 for the two pumps shown in Figures 2 and 3 on Arcton 134a. The following points can be noted:

- With 4 M NPSHA the water temperature at suction would have to rise by 9°C. This, at a coupling flowrate of 3m<sup>3</sup>/hr, would require a power loss of 31 kW, far in excess of the total absorbed power. Both flow regimes are, therefore, suitable for use on water. The temperature rise with water is insufficient to cause vaporization at any point in the pump.

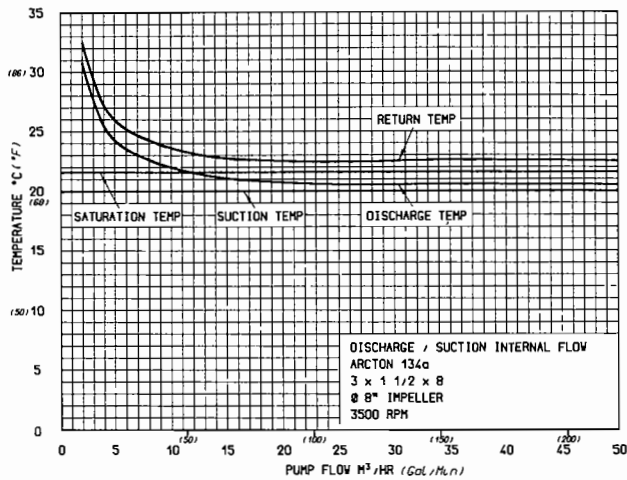


Figure 6. Pressure Drop Against Flow Rate for a Typical Journal

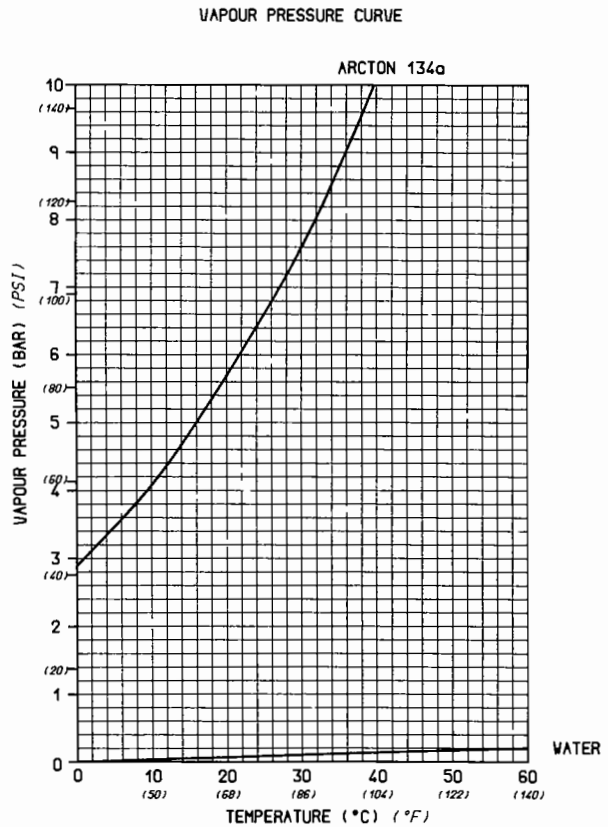
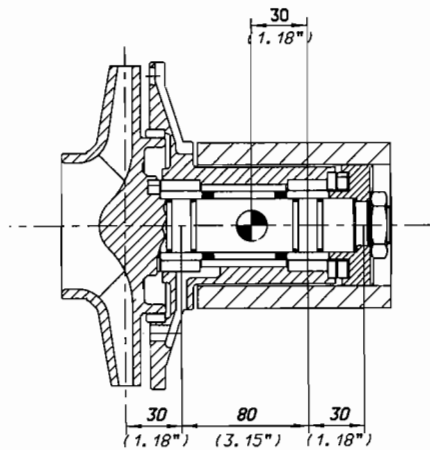


Figure 8. Temperature Rise of Product Against Flow for the Discharge to Suction Flow Regime on Arcton 134a.

Liquid	Water	Arcton
Suction Temperature	20.0	20.0
Specific Heat KJ/KgK	4.183	1.413
Density Kg/M <sup>3</sup>	1000	1223
Speed rpm	3500	3500
NPSHA M	4.0	4.0
Drive Power Loss Kw	3.2	3.2
Vapor Pressure Curve	Figure 7	Figure 7



<b>PUMP TYPE</b>	<b>3 x 1 1/2 x 8 SEAL/LESS ANSI PUMP</b>
<b>SPEED</b>	<b>3500 RPM</b>
<b>PUMP BEP FLOW RATE</b>	<b>40 M<sup>3</sup>/Hr (176 Gal./Min)</b>
<b>PUMP BEP HEAD</b>	<b>75 M (246 Feet)</b>
<b>DESIGN FLOW</b>	<b>20 M<sup>3</sup>/Hr (88 Gal./Min)</b>
	<b>40 M<sup>3</sup>/Hr (176 Gal./Min)</b>
<b>IMPELLER DIAMETER</b>	<b>200 mm (7.87")</b>
<b>IMPELLER WIDTH</b>	<b>18 mm (0.708")</b>

Figure 9. Temperature Rise of Product Against Flow for the Discharge to Discharge Flow Regime on Arcton 134a.

- For the discharge to discharge flow system, there is a 10 to 20°C difference between the maximum temperature of the product and the saturation temperature. There is, therefore, no chance of vaporization of the product in the drive.

- For the discharge to suction flow system, it can be seen that the product will be vaporizing at suction for the complete performance envelope. This is due to the lower saturation temperature, which is due to the low suction pressure and the return of the hot fluid to suction.

## DISCUSSION

The design testing and selection of flow regimes for use on sealless pumps involves a major commitment in time and money to ensure a thorough understanding of the parameters under all conditions. The following "traps for the unwary" should always be considered.

- It should not be assumed that the flow across rotating components is in any particular direction. The strength of the vortices within the drive are highly variable and surprisingly susceptible to small changes.

- Care must be exercised in eliminating restrictions prior to the areas of heat input. Journal bearings should not be used as throttle bushes, because the pressure drop will cause local vaporization.

- The use of auxiliary impellers should be avoided by careful design of the flow system. Loop or auxiliary impellers will add to the power loss of the drive, due to their inefficiency, and create localized high velocity areas that encourage erosion due to solids. Their use further complicates an already complex flow regime, making necessary a full performance curve, i.e., flow, head, power, and NPSH that must be used for each application involving heat sensitive liquids.

- The design should not rely on orifices or small passages. They are prone to vaporization, variable pressure drop due to the large

range of Reynolds Numbers involved, erosion, cavitation, and blockage. Designs should be insensitive and tolerant to these factors.

- The design should not rely on throttle bushes or restrictors to ensure the supply of fluid to the drive. The restrictors will be subject to wear in the same way as neck or wear rings. Furthermore, it is essential that there is a flow of liquid over the inner magnet ring through the gap with the shroud. A significant heat loss takes place in the shroud material and this heat can only be cooled by means of the passage of liquid. Once vapor forms, the design must be such that no air locks form.

- No positions of stagnant flow must be present, because these will be areas where solids will collect or product degrade.

Assuming that thought is given to the points just discussed, a full and complete test and evaluation program is carried out and training is given to pump selection personnel, then there should be no cause for drive failure.

## BEARING DESIGN

While the study of bearings generally is covered in great depth and referenced in numerous texts, the study of product lubricated bearings is not so widely understood. It falls to the sealless pump manufacturer to understand the lubrication regimes involved and the design rules required to successfully design bearings for these arduous applications. It is true that almost any design of plane journal or thrust bearing will support a load in specific products. The key to good design is the understanding of the limitations of the bearings and the design for low viscosity products, volatile liquids, off duty loads and system abuse. Furthermore, a good understanding is required for accurate and speedy fault finding after bearing failure.

There are three basic modes of lubrication that can exist in product lubricated bearings, each with design rules. These modes can be defined as full film, mixed, and boundary lubrication. It is generally assumed that the mode of operation experienced in sealless pumps is full film, the fallacy of this assumption is best shown by means of a simple example. This example will outline the basic, fundamental steps in determining the mode of operation of pump journal bearings.

## BEARING DESIGN EXAMPLE

The most important step in bearing design is the determination of the operating regime of the bearing.

Full film lubrication is the term given to the condition whereby the sliding surfaces are separated by a full film of lubricant. As a minimum, these films must be in the order of several times the surface roughness of the surfaces to prevent contact. These films can be maintained by externally pressurized lubricant supply as found in hydrostatic bearings. Also, as is more common in sealless pumps, the films can be self generating by means of favorable surface geometry and the relative surface velocity. The term for this condition is hydrodynamic lubrication.

It is generally assumed that this is the most common mode of lubrication experienced in pump bearings, particularly journal bearings. Irrespective of profile, there are several basic rules for this type of lubrication:

- The load carried is proportional to the fluid viscosity.
- The load carried is proportional to relative sliding velocity.
- The frictional force on the bearing is proportional to both speed and viscosity. This, therefore, means that the bearing loss will be small for low viscosity lubricants.
- The load carried and friction force are both a function of the pad area and geometry.

- The load carried is an inverse relationship with film thickness squared. For example, a decrease in film thickness will lead to a large increase in load carried. The frictional forces are inversely proportional to film thickness, therefore, the net effect is that friction coefficient is decreased with a decreasing film thickness.

The basic starting point for any hydrodynamic bearing design is the Reynolds equation, the standard form of which can be found in any quality bearing design text book. By means of a series of assumptions, the Reynolds equation can be solved to obtain expressions for pressure and subsequently load carrying capacity.

For journal bearings, however, the solution of these equations is highly complex, and the solution of Reynolds equation leads to several approximations that can lead to significant error. Several authors, notably Ockvir and Du Bois, and Sommerfeld, have both produced solutions to these complex equation each, however, with their own limitations. As a result of their limitations there has been a trend to solve the equations numerically using finite element (FE) type solutions. Reason and Narang [1] presented a solution to the bearing integrals that combined the accuracy of FE solutions to the simplicity of a closed analytical technique. The basis for their approach was to take the reciprocal of the sum of the reciprocals of both the Ockvir and Sommerfeld solutions. The accuracy of the final solution is high and has been used for the basis of the bearing evaluation being discussed herein. For full details of the mathematics and method, the reader is referred to the original paper, but the application of the basic method can be seen in the following design example.

The following example outlines the basic design calculations required for the analysis of the bearing fitted to the tested pump.

The first stage is to estimate the loads applied to the journal bearings. It is a simple procedure to calculate the resultant loads on the bearing caused by component weight, radial hydraulic load and unbalanced magnetic pull.

The shaft arrangement is shown in Figure 10. Using the work of Agnostinelli [2] and of Stepanoff [3], it is possible to obtain the following radial loads from the impeller.

FLOWRATE	RADIAL LOAD (N)		PUMP END	DRIVE END
	Agnostinelli	Stepanoff	LOAD (N)	LOAD (N)
20	106 N	716 N	238.0 N	-35.30 N
40	Assume Zero		92.1 N	4.47 N

From the work of Biheller [4] it is possible to show that there is close agreement between Biheller and Agnostinelli. For this discussion, the values given by Agnostinelli have been used. There is, however, a significant potential error in the initial values used in the calculation that the designer should be aware of.

Unbalance magnetic pull can be calculated using the techniques outlined by Binns, et al. [5]. This load is the net radial force between the inner and outer rotor caused by the eccentric running of the components leading the uneven flux densities on opposite sides of the rotor. For low eccentricities, Binns showed how theory could be applied to electric motors. This theory can be adapted to magnet drive couplings and used to give an estimate of the UMP for the selected drive. For this simple analysis, the calculated value is considered negligible for the selected drive and bearings in the "as new" condition. This figure can, however, be significant for larger drives and for canned motor pumps.

Using the method outlined by Reason [6], it has been possible to calculate, for a given bearing, the pressure at any given point (assuming a total of 1296 points of integration). This pressure distribution is subsequently integrated using numerical techniques to give the load carrying capacity, flowrates and temperature rises within the bearing. The temperature rise calculated can subse-

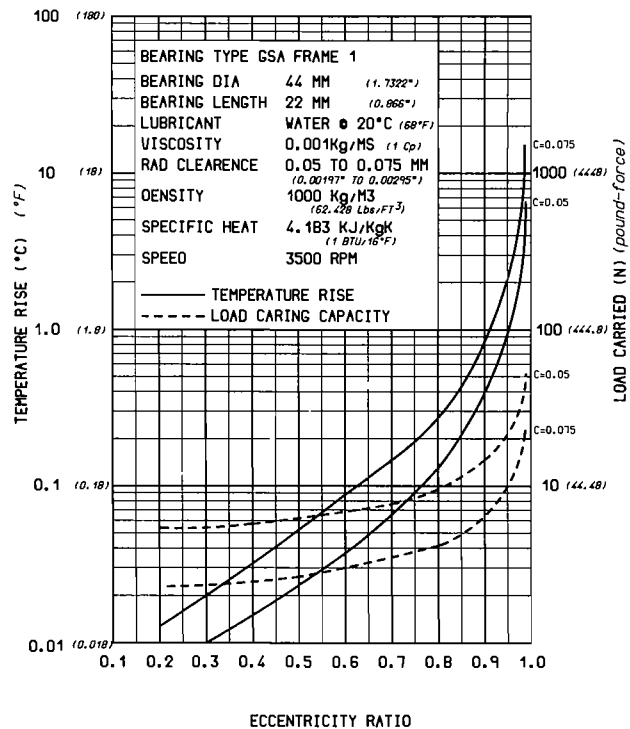


Figure 10. Shaft Layout for Radial Load Analysis.

quently be used to recalculate the viscosity of the product using Vogel's viscosity correction law [7]. While this law was initially devised for use with oils, it has been found to give a very close approximation to published viscosity figures for typical products. Using a series of iterations for successive eccentricity ratios, it is possible to calculate the actual running conditions of the bearing for a given, fixed, applied load. Typical results can be seen in Figure 11. Several points can be noted:

- The load carrying capacity is exponential at high eccentricities and at these eccentricities the variation in maximum and minimum running clearance becomes negligible.
- The temperature rise only becomes significant at high eccentricities but, for this example, is negligible.
- The difference in eccentricity ratio between the two journal bearings is sufficient to cause significant misalignment of the shaft. The following table shows the actual running clearance from the relationship:

$$h = C(1 - e)$$

Where  
 h = film thickness (mm)  
 C = radial clearance (mm)  
 e = eccentricity ratio

BEP Flowrate	Clearance	e	Pump End		Drive End	
			100%	50%	100%	50% flow
0.05mm	h	0.91	0.95	0.45	0.825	
		0.0045	0.0025	0.275	-0.00875	
0.075mm	h	0.95	0.975	0.625	0.895	
		0.00375	0.001875	0.028	-0.007875	

By use of similar triangles and the proportions shown in Figure 10, it is possible to show that for off duty flowrates (50 percent in

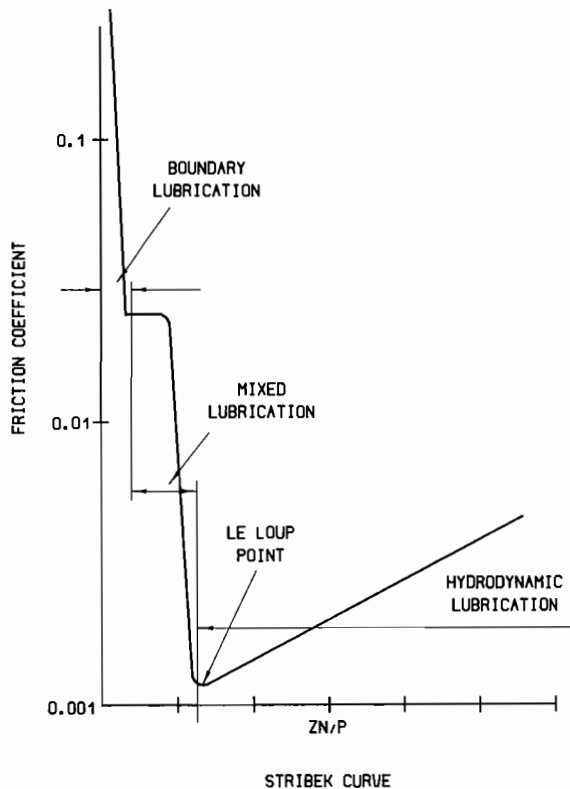


Figure 11. Eccentricity vs Temperature and Load Carrying Capacity for a Product Lubricated Journal Bearing.

this example) edge contact will occur, primarily due to the negative reaction on the drive end bearing.

## DISCUSSION

Any analysis of bearings must be carried out with caution, taking full account of past experiences, theory and common sense. The result above would suggest that edge contact will cause rapid wear of the bearings. Practical experience of tens of thousands of bearings purchased shows that the bearings in this example work well under off duty conditions. What, then, is the reality of the bearing operating regime?

Assuming identical eccentricities, i.e., a parallel longitudinal bearing film, the only outstanding action is to determine if the calculated running clearance is realistic. At some stage, due to severe operating conditions, a complete load carrying film cannot, in itself, separate the surfaces. This is mixed film lubrication. There may be localized areas of fluid film separation with the balance of the surface area in contact with the disparities of the surface roughness. At best, these areas are separated by a layer a few molecules thick that will prevent serious wear and high friction. These areas can be considered as operating under boundary lubrication conditions.

With the total absence of a supporting lubricant film the entire surface of the bearing can be considered to be working under boundary lubrication conditions. This condition should be avoided, due to the high friction losses and excessive wear that will take place. Boundary lubrication is, therefore, the extreme of mixed lubrication, full film being the other. These factors can be seen graphically by reference to the Stribek curve for the bearing.

The Stribek curve is a simple but effective way of determining the operating regime of a bearing. This method uses the bearing parameter,  $ZN/p$ , where  $Z$  is the viscosity in centipoise,  $N$  the

speed in rpm and  $p$  the specific pressure in  $lb/in^2$ . This is the commonly accepted form of the equation and no attempt is made to present it in SI units. The diagram is shown in Figure. 12 with friction coefficient  $\mu$  plotted against  $ZN/p$ .

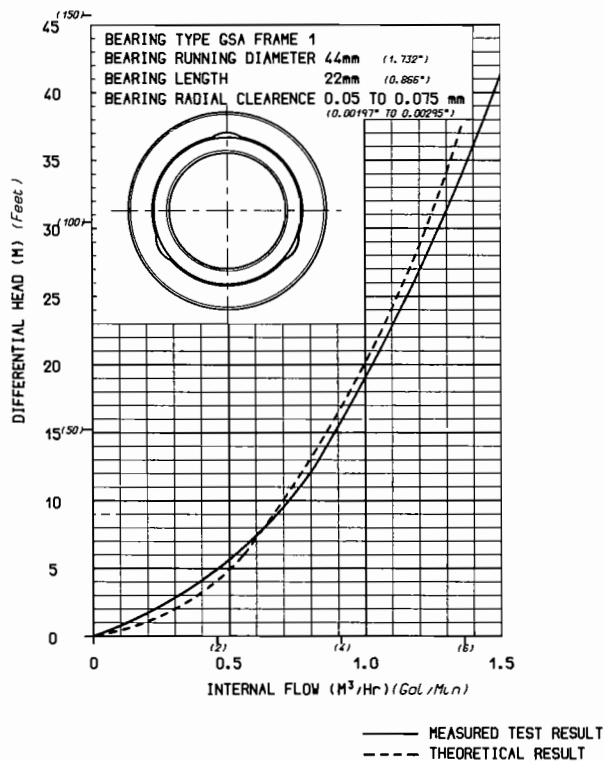


Figure 12. Typical Stribek Curve for Journal Bearing.

An increase in applied load, while operating hydrodynamically, causes a corresponding increase in pressure and increasing temperature with corresponding reduction in viscosity leading to a reduction in the value of  $ZN/p$ . At some point, the applied load is sufficient to cause rupture of the lubricant film, i.e., contact between the bearing surfaces. At this point, known as the Le Loup point, a small increase in load will cause a significant reduction in the hydrodynamic effects within the bearing. The friction coefficient will rapidly rise to the value quoted in manufacturers' text for friction coefficients of wetted materials. For carbon, this is between 0.01 and 0.015. The value of  $ZN/p$  at the Le Loup point is a function of several features notably bearing geometry, surface finish and bearing distortion.

With ever increasing load, there will be a point at which the frictional losses in the bearing give rise to temperatures that lead to vaporization of the product. At this point, there will be a second rapid rise in friction coefficient. This second point is not a function of  $ZN/p$ , but rather a function of load, lubricant properties, and cooling supply. It is, however, shown diagrammatically on the Stribek curve to demonstrate the range of friction coefficients experienced in a bearing set.

The most general rule applied to this type bearing is to define the bearing rating in terms of the product of velocity and contact pressure (PV). From this, it is clear that as speed increases, the load carrying capacity of the bearing decreases. Frictional force is a direct function of the friction coefficient of the material combination, but will vary due to the degree of contact of the surfaces. For example, a carbon/steel combination will vary from a friction coefficient of 0.01 to 0.3, dependent upon mixed lubrication regime existing or full boundary lubrication, i.e., dry running.



In the example discussed herein, the bearing films are anything but parallel, and will lead to edge contact. At the point of contact, there will be high loads and mixed lubrication, but it would be wrong to assume that there is no hydrodynamic effect. Reason [6] and Dyer outlined the result of work to show the effect of misalignment in journal bearings. The authors showed herein how the pressure profile became distorted due to misalignment, and how the load capacity was reduced. Note that the angular misalignment is three dimensional, because the eccentricity ratio varies between bearings and the attitude angle. The authors did show, however, that there was still significant hydrodynamic effect due to the converging journal film, and suggest methods of adjusting the calculated values for ideal bearings for misalignment.

A further factor that affects journal bearings is the use of lubrication grooves. Circumferential grooves serve to split the bearing into two halves. Each half can be calculated as a short bearing, and it can be shown that the product of the capacity for each half is less than the product for the full length. Axial grooves, or spiral grooves, again significantly affect the bearing load carrying capacity and are primarily used as a path for solids that become trapped in the bearing. The use of these grooves is an essential feature on carbon bearings where the presence of solids will lead to bearing wear. The use of grooves in ceramic bearings is more debatable. It can be argued that if the bearing clearance is small, a groove will only serve to introduce solids to the surfaces.

For evaluating the performance of bearings that operate in the mixed lubrication field, the key considerations are bearing power loss and subsequent temperature rise and material PV value. The key consideration when evaluating PV values is the test methods used by the material supplier. Care must be exercised in extrapolating results based upon journal bearing tests, since these results are generally based upon projected bush area not contact area. The contact area used should be based upon the area of contact [8]. Assuming the loads on the journal bearing of 716 N and a friction coefficient of 0.015, the frictional power loss will be 86 W. It is, therefore, essential that the flow, and pressure drop across the journal bearings, are such that there is no vaporization of product due to this loss. The loss calculated above is a "worst case" calculation assuming zero hydrodynamic bearing film support. The pump supplier should, therefore, know the pressure against flow curve for his journal bearings.

It has been found that the Darcy equations for flow through pipes, coupled with the simple Moody diagram, can be used to give a good estimate of the flow through a bush with the diameter based upon the equivalent hydraulic diameter calculated from the wetted perimeter and flow area. These results can be seen plotted in Figure 6. It should be noted that entry and exit losses have been added to this figure as detailed in Massey [9]. Based upon this curve and the power loss from above it can be shown that a flowrate of 1 l/min is required for a bearing running on butane with a temperature rise of less than 1°C. Each supplier must, therefore, have definitive values for the flowrates and pressure drops across the journal and, similarly, thrust bearings.

## SUMMARY

The operating regime of the bearing is highly complex and is of the full film or mixed lubrication regime. Proof of this statement is that:

- Bearing material selection directly affects bearing performance. For the typical temperature losses shown for full film lubrication, together with a separating film, there would be no need for silicon carbide bearings to improve performance, yet ceramic bearings can beneficially affect some bearing solutions.
- Journal bearings do work hydrodynamically. The presence of half speed whirl (bearing instability) on vertical pumps [10] proves

that there are instances where lightly loaded bearings cause excessive vibration, vibration that can be solved by reducing then bearing load carrying capacity and increase the bearing eccentricity [11].

- Bearings work in the mixed lubrication mode as demonstrated by the polishing of silicon carbide while running, often witnessed by the diamond checker pattern that becomes visible after running, that is a function of the original machining process.

- Bearings do not operate in the boundary lubrication mode. Both carbon and silicon carbide will heat up rapidly while dry running. This temperature rise is often sufficient to cause surface distortion, glazing of surfaces, differential expansion problems, or seizure. The bearing in the above example will dissipate in excess of 1.5 kW when dry running. This heat will leave a witness, practical experience shows that there is only this witness in failure conditions or when there is a fundamental design flaw. The corresponding power loss figures for mixed and full film lubrication are typically 86 W and 8.6 W, respectively.

- Most designs of bearing have provision for lubrication grooves. Experience shows that they improve performance, this is contrary to full film design rules. This is best explained by reference to the mixed lubrication friction loss and the need to dissipate this waste heat.

## CONCLUSIONS

Successful installation and operation of sealless pump demands a partnership between the supplier and the user. The user must supply as much information as possible about the liquid pumped covering physical characteristic including:

- SG
- Viscosity
- Specific Heat
- Vapor Pressure/Temperature Rise
- Presence of Dissolved Gasses
- Solids percentage and nature

In addition, all process operating conditions must be fully evaluated. For his part, the supplier must provide adequate information on the changes to the process liquid as it passes through the pump, under design and off design operating conditions, justifying heat balance in the product cooling and lubrication channels to give confidence that the pump and system are an ideal match.

Not surprisingly, users of equipment have reacted to protect themselves from the confusion of suppliers by the rapid production of site engineering specifications and the preparation of national and international standards for sealless pumps.

These are essential for purchasing offices when comparing bids and are attractive to manufacturers wishing to enter the market with a standard mass produced product.

The standards that are currently being prepared and adopted, to a great extent document sealless technology as it stands today. Although a mature industry of over 40 years, increased competition will stimulate innovation and the necessity to educate users about the products available. The sealless pumps of tomorrow may be very different than those produced today, and will be the result of a continued dialogue between users and suppliers.

It will be essential that any standards produced do not limit the ongoing improvement to this important type of pump.

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