Isomag Corporation

A Key Variable in Predicting and Extending Bearing Life

Executive Summary

Bearings are at the heart of all rotating equipment, yet they are often an afterthought. Neglected until the "heart" stops beating and the equipment is forced to shut down wasting time and money. This paper puts bearings first. More specifically, it goes into detail about the history of bearing life calculation and the best practices by today's standards with primary reference to ISO 281:2007. Further, it provides methods for more accurate life estimations by examining current bearing protection devices. The history and evolution of these devices is described in detail along with the role they play in extending bearing life.

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Introduction

There are many variables involved in rotating equipment bearing life including, but not limited to: load conditions, lubrication selection and storage, shaft alignment and vibration, lubrication level, and bearing type/material. Through proper specification and best maintenance practices, these parameters are within a Reliability Engineer's control.

So, what is missing? This white paper is going to focus on another variable and how it can not only provide a more accurate means for bearing life prediction but will also provide the means of improving bearing life. It will cover common bearing life prediction practices and one key component that is commonly left out.

Basic Rating Life

In 1947, a couple of years after the second world war, when bearings were just chunks of steel compared to today's standards, two Swedish engineers, Gustaf Lundberg and Alvrid Palmgren wrote a book called "Dynamic Capacity of Rolling Bearings." This book remains at the heart of today's bearings standards [1]. Lundberg and Palmgren realized the amount of dynamic loading applied to the bearing surfaces had a direct correlation to how quickly these surfaces would fatigue and/or crack. About a decade later, the International Organization of Standardization (ISO), would adopt the Lundberg-Palmgren equation as the Basic Rating Life or L_{10} equation:

$$\boldsymbol{L_{10}} = \left(\frac{\boldsymbol{C}}{\boldsymbol{P}}\right)^{\boldsymbol{p}} \qquad (\mathbf{1})$$

Where,

C	=	Bearing dynamic load rating (written C_r for radial bearings and C_a for thrust bearings) [N or lb]
P	=	Bearing load or equivalent load [N or lb]
p	=	A constant power: 3 for ball bearings or 10/3 for typical roller

Bearing life, in the broad since of the term, is defined as the length of time a bearing can endure a specific loading until the surfaces inside the bearing succumb to fatigue and a spall develops. The critical spall size when failure occurs, regardless of bearing size, is defined as an area of 0.01 in^2 [2]. L_{10} life, as written in equation (1), is the point at which 10% of bearings in a control group will fail. So, if a group of identical bearings is tested under the same conditions, 90% of them will meet or exceed the L_{10} rated life. This rating is commonly broken down into hours as follows:

$$\boldsymbol{L_{10h}} = \frac{\mathbf{10^6}}{\mathbf{60}\boldsymbol{n}} * \boldsymbol{L_{10}} \qquad (\mathbf{2})$$

Where, n is the rotational shaft speed in revolutions per minute (RPM).

As a baseline example consider a midsized ANSI pump's thrust bearing. This bearing is a 3309 A double row, angular contact ball bearing with the following stats:

Calculation Data	Symbol	Value	Unit
Basic Dynamic Load Rating	C	16861	lb
Fatigue Load Limit	C_u	504	lb
Calculation Factor	X	0.63	-
Calculation Factor	Y	1.24	-

Table 1: 3309 A Angular Contact Ball Bearings, Double Row [3]

First, an equivalent dynamic loading is determined using the following equation:

$$\boldsymbol{P = XF_{Radial} + YF_{Axial}} \quad \textbf{(3)}$$

Maximum radial and axial design loads will be assumed as follows [4]:

$$\boldsymbol{F_{Radial} = 400\ lbs}$$

$$\boldsymbol{F_{Axial} = 900\ lbs}$$

Equation (3) becomes:

$$\boldsymbol{P = 1368\ lb}$$

If a shaft speed of 3600 RPM is assumed, equation (2) can be used as follows:

$$\boldsymbol{L_{10h} = \left(\frac{16861}{1368}\right)^3 \left(\frac{10^6}{60(3600)}\right) = 8668.38\ hours}$$

So, the L_{10h} rated life is approximately 8668 hours. Does this seem to be an over or under estimation? It depends on certain circumstances.

For many years, the use of this basic rating life as a criterion for bearing performance has proved satisfactory. However, while it was a milestone in the bearing industry at the time, a lot has changed since Lundberg and Palmgren defined this L_{10} equation. Engineers started to recognize that many factors besides the loading and speed affect bearing life and more specifically the amount of stress in the contact area of a bearing. These factors are considered interdependent and under favorable operating conditions, end users can see very long bearing lives when compared to the traditional L_{10}. On the other hand, bearing lives can be shorter than L_{10} in unfavorable conditions. Either way, for a more accurate life estimation on payload equipment, new variables must be considered. So, what are they?

Lubrication Regime

For starters, there's the lubrication regime. A variable defined as kappa, κ, is the ratio of the actual lubricant viscosity to a reference kinematic viscosity defined in the ISO 281 standards [5]:

$$\boldsymbol{\kappa} = \frac{\boldsymbol{v}}{\boldsymbol{v_1}} \qquad (4)$$

This ratio defines the effectiveness of a lubricant or the degree of separation between the rolling contact surfaces. For an adequate film thickness to form, the lubricant must retain a minimum viscosity at the operating temperature. Therefore, bearing life may be extended by increasing the operating viscosity, v. Generally, the lubrication improves as κ increases and any lube regime where $\kappa \geq 1$ is a good one. While this is a critical variable, for the sake of this paper we are going to assume an ideal lube regime where $\boldsymbol{\kappa = 4}$.

Fatigue Load Limit

What else? How about the fatigue stress limit? Sure, fatigue to the point of fracture has already been considered but this is different. It is common knowledge that if you bend a paper clip back and forth far enough many times... it breaks. This is fatigue. However, if you only bend it a little, it will not break. Technically, it could last forever. The same goes for bearings, if you minimize stress below a certain point, with a load well within the bearing's dynamic capacity, it will never reach it's breaking (or spalling) point. The factors influencing this load limit are: type, size and internal geometry of the bearing; the profile of the rolling elements and raceways; the manufacturing quality; and the fatigue limit of the raceway material. Once again this is a critical variable, but not the focus of this paper. In fact, this variable is supplied to end users by the bearing manufacturer. To continue this bearing life calculation, the fatigue load limit used will be defined by well-known bearing manufacturer SKF as $\boldsymbol{C_u = 504\ lb}$ (see Table 1).

Assumptions

Fatigue load limit, lube viscosity, what's left?   To narrow it down further, variables within the control of an apt reliability engineer along with assumptions made so far can be summarized as follows:

1. An ideal lube selection and regime: $\kappa = 4$
2. Pump is running at but not above maximum design loading (see F_{radial} and F_{axial})
3. Ideal shaft alignment, no vibration
4. Sump oil remains at a constant and correct level
5. Constant shaft speed (3600 RPM)
6. Ideal bearing spec, design, and material properties ($C_u = 504lb$)
7. Operating temperature within material specifications
8. Corrosion is negligible

Contamination Factor

There is a variable that is out of the realm of a reliability engineer's control… One that cannot be summarized in a simple assumption like the ones listed above. It's the *environment* and more specifically, what's in it. Environments can contain just about anything: water, dust, mud, humidity, chemical powders, salt, pulp, you name it. Even seemingly clean environments can contain aerosols or tiny particles that are so small and light they can float in air. All these environmental occurrences are only seen as one thing from a bearings point of view: *contamination*.

According to a study conducted by SKF, contamination leads to 47% of bearing failures, by far the leading cause when compared with other common failure modes as shown in Figure 1 [6].

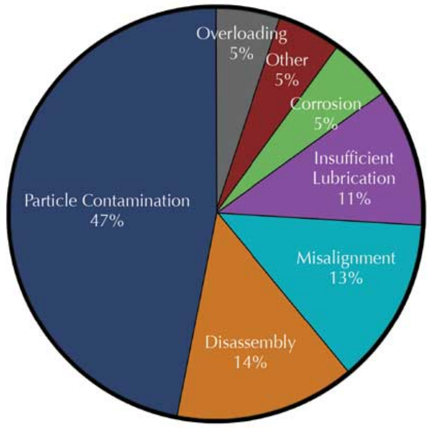

Figure 1: Modes of bearing failure [5]

Why is contamination so critical? Consider a single solid particulate, a grain of silt, which is approximately 25 micrometers (μm) in diameter ($\approx$.00098 inches). To put this size in perspective, the smallest size visible to the human eye is around 40μm (see Figure 2). In this case, 25 μm is slightly larger than the approximate radial clearance in a 3309 thrust bearing [7]. This grain of silt enters the housing. Now while floating around in the lubricant the particle finds its way into the thrust bearing and is forced between the ball and one of the races (see Figure 3). What will happen to the stress inside the bearing? Now the ball is riding on a solid particle instead of the intended lubricant film and all the stress will be concentrated at this one point. The material will inevitably reach a fatigue beyond its limit, spalling and flaking occur, etc... For obvious reasons this is not good. Now, maybe it's a long shot that this one microscopic particle just happens to wedge itself between the rolling element and the bearing race, but what if hundreds of these tiny particles find their way into the lubricant? What about thousands?

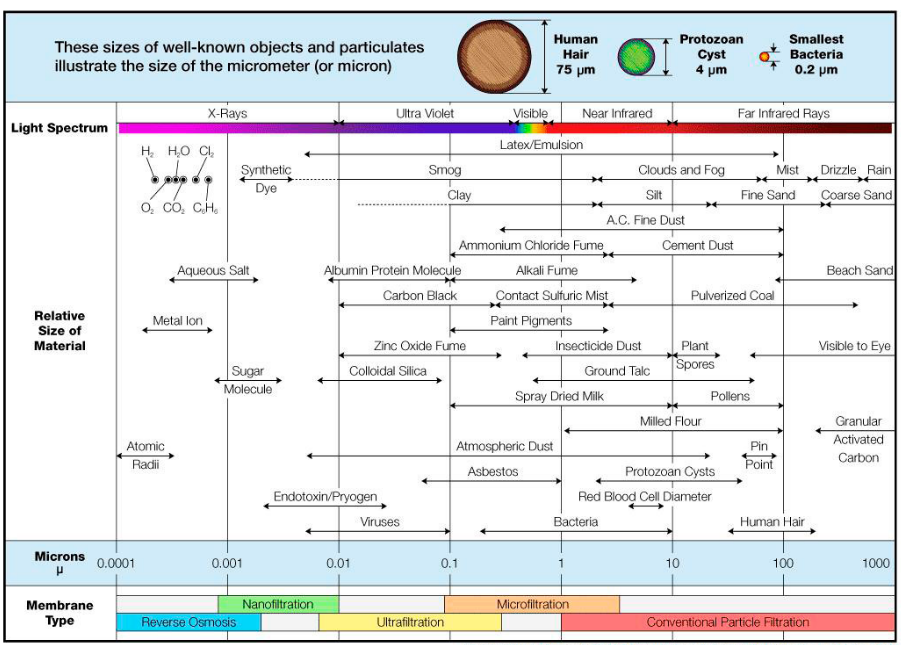

Figure 2: Particle Size Chart [8]

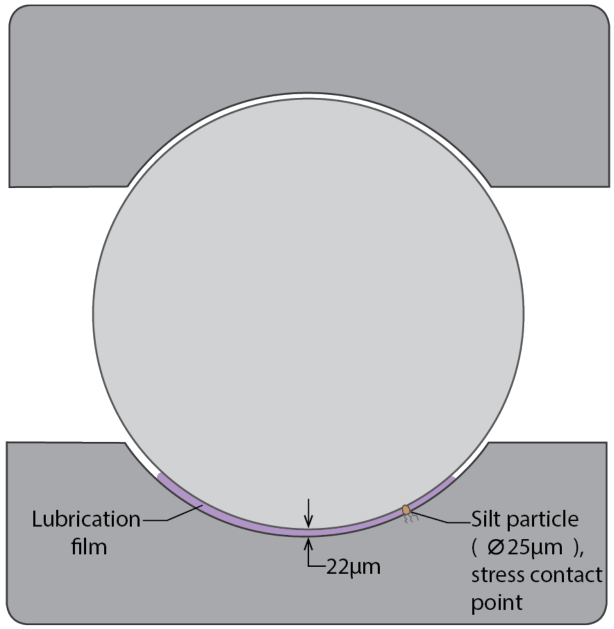

Figure 3: Bearing with silt particle contamination

It should be obvious now why particle contamination is a leading cause of bearing failure. So, the question becomes how can we take a variable that is seemingly out of our control, like the environment, and bring it within our control?

In 1978 the ISO group launched its ISO 281 standard which has since evolved to the 281:2007, and includes "Life Adjustment Factors." These factors include the lubrication regime (κ), fatigue load limit (C_u), and last but certainly not least the *contamination factor*, e_c. These factors are then summarized into a single life modification constant, a_{ISO} , which then multiplied by the basic life rating to create the new "Modified Life Equation":

$$L_{10m} = a_{ISO} L_{10} \qquad (5)$$

How exactly a_{ISO} is determined will be discussed further into the paper.

The contamination factor, e_c, is defined in the ISO standards as the adjustment factor for the bearing life reduction due to contamination in the lubricant film. As mentioned before, when solid particles contaminate the lubricant, permanent indentations in the bearing raceway can be generated. Local stress risers are generated at these indentations which leads to a reduction in rolling element life. The following table provided in the ISO standards provides e_c values for different levels of contamination.

Level of contamination	e_C	
	$D_{pw} < 100$ mm	$D_{pw} \geqslant 100$ mm
Extreme cleanliness Particle size of the order of lubricant film thickness; laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter; conditions typical of bearing greased for life and sealed	0,8 to 0,6	0,9 to 0,8
Normal cleanliness Oil filtered through fine filter; conditions typical of bearings greased for life and shielded	0,6 to 0,5	0,8 to 0,6
Slight contamination Slight contamination in lubricant	0,5 to 0,3	0,6 to 0,4
Typical contamination Conditions typical of bearings without integral seals; course filtering; wear particles and ingress from surroundings	0,3 to 0,1	0,4 to 0,2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing	0,1 to 0	0,1 to 0
Very severe contamination	0	0

Table 2: ISO 281:2007 Contamination Factor chart, e_c [5]

Dynamic Filtered Particle Size

This is where it gets interesting. The effect of contamination on bearing life should be common knowledge for any adequate reliability engineer. Surely *contamination* cannot be the key variable to predicting and extending bearing life? Correct. It is not contamination nor the contamination factor itself, but rather is a variable that aids in determining the contamination factor. This variable is the size or diameter of the largest particle that will be filtered out of a bearing-lubrication system, x.

The ISO standards define x in terms of a "filtration ratio" $\beta_{x(c)}$. This ratio determines how many of a certain size particle are upstream of the filter compared to downstream:

$$\beta_{x(C)} = \frac{number\ of\ particles\ of\ a\ certain\ upstream}{number\ of\ particles\ of\ a\ certain\ size\ downstream}$$

For example $\beta_{6(c)} = 200$, basically means for every $6\mu m$ particle allowed into the system, 200 $6\mu m$ particels are kept out of the system or kept upstream. e_c is then determined using several charts provided by the ISO group, each chart representing different particle sizes. For the sake of simplicity and since the results are comparable, this paper will adopt the clear method from global bearing manufacturer, NSK, for determining filter size, x, as defined in the following chart (to be used in conjunction with table 1) [9].

	Very clean	Clean	Normal	Contaminated	Heavily contaminated
a_c factor	1	0.8	0.5	0.4–0.1	0.05
Application guide	10 µm filtration	10–30 µm filtration	30–100 µm filtration	Greater than 100 µm filtration or no filtration (oil bath, circulating lubrication, etc.)	No filtration, presence of many fine particles

Table 3: NSK Contamination Factor Chart [6]

Any connection, plug, joint, vent etc. has a filter size. One more safe assumption that will be added to our assumptions list is that all static filters fall within the "Extremely Clean" category where all particles larger than $10\mu m$ are filtered out. This assumption is safe because it is much easier to "tighten" a static filter than a dynamic one. It also allows us to narrow down our key variable even further to the **filtered particle size between the dynamic surfaces of the bearing system, $\boldsymbol{x_{dyn}}$**. At some point (or surface), the rotation of the shaft must be coupled with a static surface connected to the housing. This juncture is where x_{dyn} is measured. It typically happens within the bearing protection device (BPD) that is being utilized. So, the key to determining x_{dyn} is diving deep into the inner workings of various BPD designs.

Before analyzing bearing protection devices, it is important to establish a time of when x_{dyn} should be measured. The thing about dynamic surfaces is they tend to change over time. This change is due to wear. Luckily the ISO standards have already provided us with the perfect baseline, L_{10h}. Going back to our ANSI pump example $L_{10h} = 8668\ hrs$. For each BPD it should be asked "will x_{dyn} at the time the pump is commissioned be the same after 8668 hours?" If not, how much will it change?

Lip Seals

First off, radial lip seals. When bearing protection devices are thought of, lip seals are likely the first ones that come to mind. Lip seals are both the original BPD and the most widely used in industry today. Born around the 1920's, radial lip seals started as just an oil resistant leather strap assembled into a metal case pressed onto a shaft [10]. They have since evolved to the design shown below which first came to market around the 1970's.

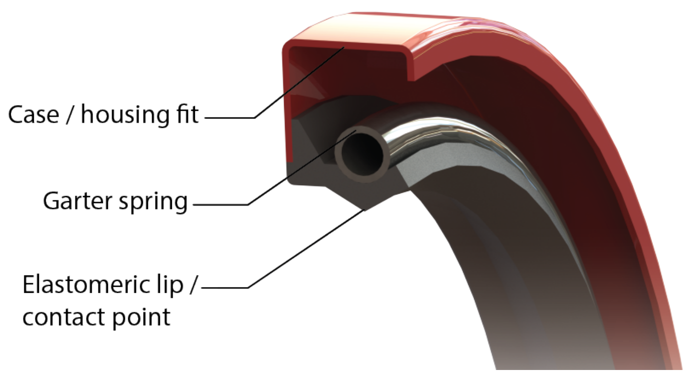

Figure 4: Radial lip seal

Radial lip seals use the shaft as the primary sealing surface, in this case, the point where x_{dyn} will be measured. A significant amount of pressure is produced from the garter spring to apply closing force at this point. Normally, new lip seals have an average radial contact pressure of $1\ N/mm^2$ (approximately 145 psi) [11]. This significant amount of pressure leads to wear between the shaft and the lip's elastomeric surface. As the surfaces wear a gap will form, and the x_{dyn} increases. To put an exact number on this after 8668 hours is extremely difficult. A lot of variables are involved: the lip material, shaft material, smoothness and hardness, and of course the particles present in the environment. If many tiny hard particles are present, like sand for instance, they can wedge between the lip and shaft creating an abrasive grinding effect. In this case, wear tracks hundreds of micrometers deep can form in even a hardened and smoothed (64rms) shaft (x_{dyn}>$500\mu m$). In the case of extremely clean environments, wear tracks may only be 100 or so micrometers deep after 8000 hours. According to Chicago Rawhide, a leading lip seal manufacturer, a lip can be completely worn out after approximately 3000 hours of continuous run time.

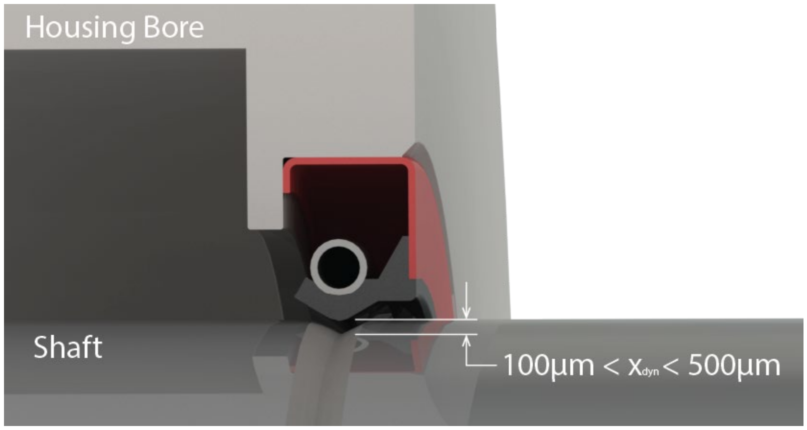

Figure 5: Radial lip seal estimated wear after 8668 hours

For this study we will assume the environment severity is somewhere in between the two cleanliness extremes and $100\mu m < x_{dyn} < 500\mu m$ after 8668 hours. So, using x_{dyn} with tables 1 and 2, $e_C \approx 0.05$ where the contamination level is considered severe, as would be the case when running a lip seal on an ANSI pump for this period of time.

With e_c determined it is now time to calculate a_{ISO}. This constant is found for thrust ball bearings using the graph shown in Figure 6. Similar graphs are provided in the ISO standards for radial ball bearings, radial roller bearings, and thrust roller bearings.

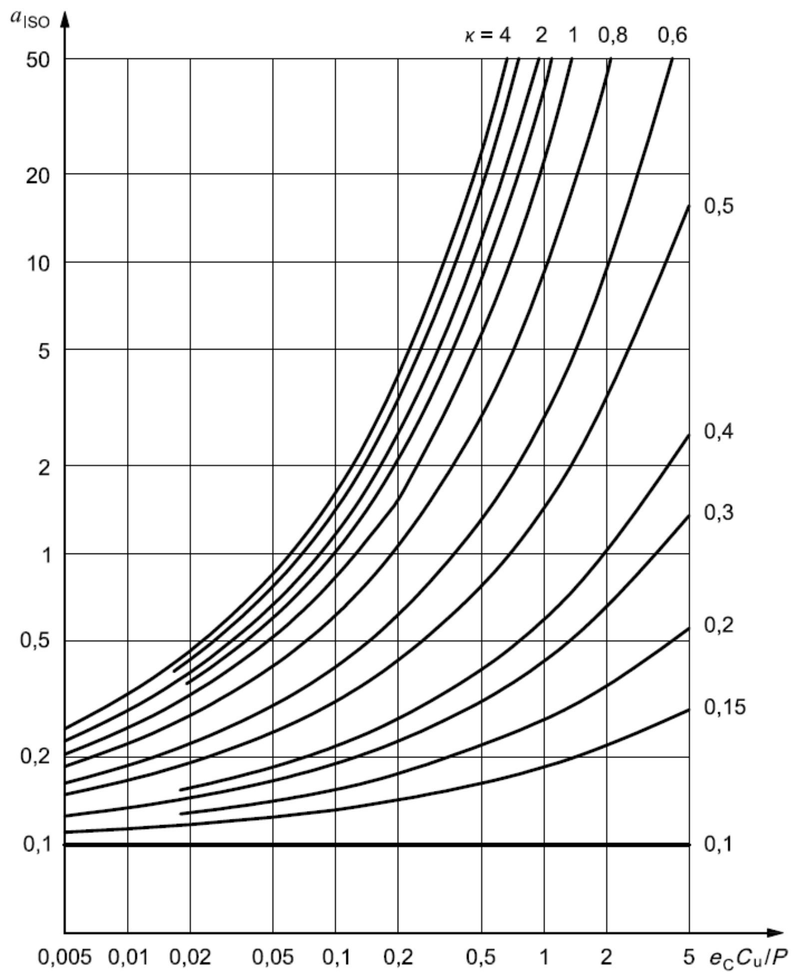

Figure 6: Life modification factor, a_{ISO}, for thrust ball bearings

The kappa (κ) variable is already known to be 4. We can easily determine the x-axis value as follows:

$$\frac{e_c C_u}{P} = \frac{.05 * 504}{1368} = 0.02$$

Therefore, by using the chart:

$$a_{ISO} \approx 0.45$$

Finally, substituting into the life modification equation (5):

$$L_{10m} = 0.45(8668\ hrs) = 3901\ hours$$

3,900 hours is in line with Chicago Rawhide's estimate of approximately 3,000 hours of lip seal wear life. How does this compare to other BPD technologies?

Compound Labyrinth Isolators

Next on the list is the compound labyrinth bearing isolator, also known as a dual-labyrinth. First patented in 1977, the labyrinth isolator was born as an upgrade to lip seal technology. “We avoided the name ‘seal’ and called it a ‘Bearing Isolator,’” said Inpro founder and dual-labyrinth inventor David Orlowski [12]. The compound labyrinth or “bearing isolator” was a big leap in BPD technology. It intuitively combined single stationary labyrinths, or close-clearance labyrinths, with deflector rings into a compact compound design as shown below in Figure 7. The idea, as the name implies, was to “isolate” the bearings inside the bearing housing.

Close-clearance Labyrinth   Deflector Ring   Compound Labyrinth Isolator

Figure 2: The Evolution of the Compound Labyrinth Isolator

Compound labyrinths work by establishing a restrictive path (or maze/labyrinth) which makes it difficult for oil to flow through or leak out. Dual labyrinths typically contain two main components: a stationary and rotor. The stationary fits the housing typically on a metal-to-metal press fit, while the rotor fits the shaft and rotates with it driven by O-ring compression. This eliminates the dreaded shaft wear created from lip seals. This rotor also acts as a flinger ring, using centrifugal force from the rotating shaft to disperse contaminants away from the housing. Of course, for proper function, this requires dynamic conditions where the shaft is spinning. Without centrifugal force (i.e. during shut downs) contamination can be an issue.

Dual labyrinths are commonly referred to as “non-contacting”, in that there is a small clearance between the rotor and stationary. This clearance is typically around .010” or 254μm. However, most current labyrinth designs contain an internal component wedged between the rotor and stator, which in fact is a contacting surface. This can be an O-ring, a composite ring, or even a miniature lip seal. Figure 8 shows a variety of common labyrinth cross-sections.

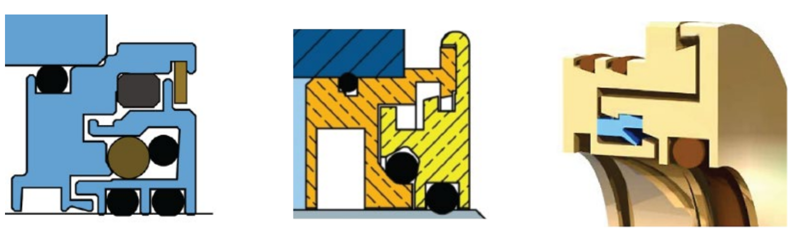

Figure 3: A variety of current compound labyrinth technology [13] [14] [15]

These internal components ensure x_{dyn} remains smaller than 254 μm, which for obvious reasons would be unacceptably large. However, like lip seals, these components have wearing surfaces, where x_{dyn} will increase over time. The good news is that the shaft is not used as the wearing surface, rather this is kept internal to the compound labyrinth design. Again, exactly how much x_{dyn} increases in 8668 hours is extremely difficult to determine and will vary depending on the application, it will be easier to define a range as was done for lip seals. It is safe to assume the wear gap will be smaller than the gap seen in lip seals due to the compact nature of the dual labyrinth design, and likely falls within the “Normal Cleanliness” range of $30\mu m < x_{dyn} < 100\mu m$ (see Figure 9). Therefore, choosing a contamination constant in the middle of this range, $e_c \approx 0.55$.

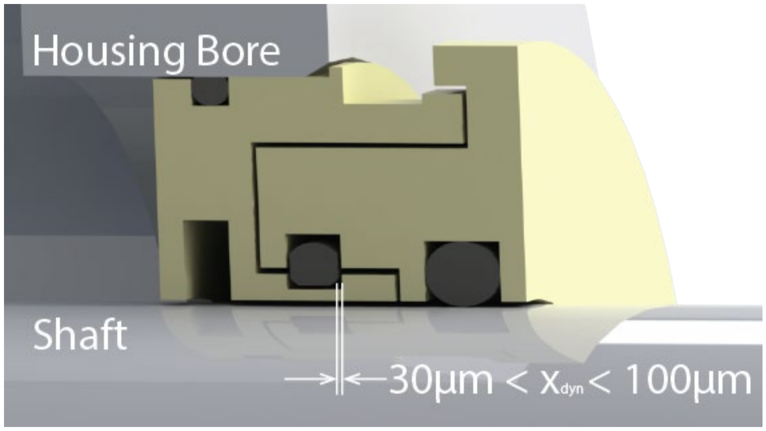

Figure 4: Compound labyrinth estimated wear after 8668 hours

Calculating the x-axis of the graph in Figure 6:

$$\frac{e_c C_u}{P} = \frac{.55 * 504}{1368} = 0.20$$

It’s then determined, $a_{ISO} \approx 4$. Therefore, the modified life estimation becomes:

$$L_{10m} = 4(8{,}668\ hrs) = 34672\ hours$$

Obviously, a clear improvement over lip seals, but there is one more BPD technology to discuss.

Magnetic Face Seals

In the mid-1990s, a couple of decades after bearing isolators were first conceived BPD technology continued to evolve, adapting to the need for complete protection and longer bearing life. This evolution led to the magnetic face seal. Similar to the concepts driving mechanical seals used on the wet end of pumps, magnetic face seals utilize lapped flat faces held together by magnetic force rather than mechanical energy to create a positive, liquid-tight seal. Magnetic face seals contain a stationary assembly and a rotor, also driven by O-ring compression, to ensure no wear to the shaft surface. The rotating face is manufactured out of a ferrous stainless steel making it the target for the magnets held in the stationary case.

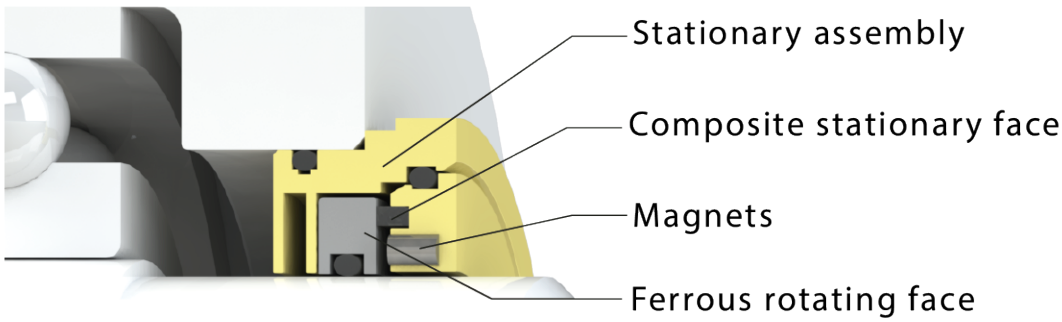

Figure 5: Magnetic face seal

Both lapped flat faces labelled in Figure 10, are each optically measured around 1-2 light bands. If looked at under a microscope, even surfaces at a precision flatness of 1 light band have slight undulation which when pushed together form a tiny air gap. This air gap is between 0.5μm and 1μm, making it easy to determine the dynamic filtered particle size, $x_{dyn} = 1\mu m$. The unique aspect of magnetic energy is that it is constant and always pulling. Meaning that, after 8,668 hours, the faces may wear slightly but x_{dyn} will remain at 1μm because the magnets continue pulling the rotating ferrous face into the composite stationary face. Both faces are optimized through material specification and magnetic loading to have an extremely long wear life. So, after 8,668 hours $x_{dyn} = 1\mu m$, and therefore $e_c = 1$ (see Figure 11).

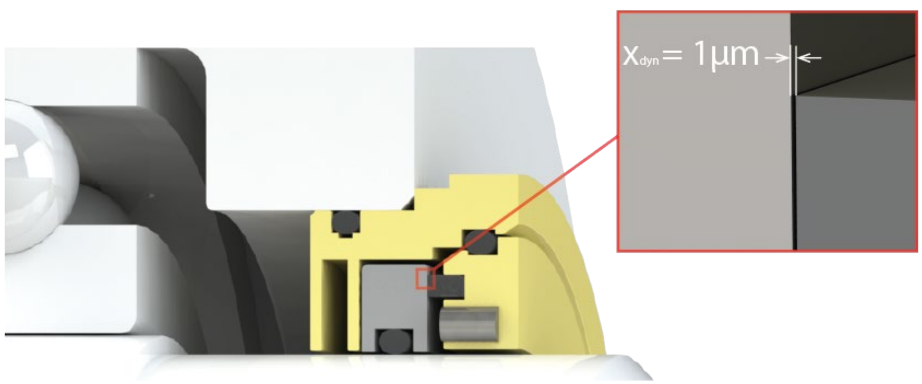

Figure 6: Magnetic face seal's dynamic filtered particle size

Our x-axis becomes:

$$\frac{e_c C_u}{P} = \frac{1 * 504}{1368} = 0.37$$

Using the graph in Figure 6 again, $a_{ISO} \approx 10$; and the modified life for magnetic face seals becomes:

$$L_{10m} = 10(8668\ hrs) = 86680\ hours$$

Conclusion

Table 4 below summarizes the L_{10m} results.

Bearing Protection Device	x_{dyn} (μm)	a_{ISO}	L_{10m} (hours)
Radial Lip Seal	100 - 500	.45	3901
Compound Labyrinth Isolator	30 - 100	4	34672
Magnetic Face Seal	1	10	86680

Table 4: Modified Life Comparison Chart

It should be obvious from Table 4, how much of an impact x_{dyn} has on bearing life. To put it simply, the smaller the particle that is filtered out of the bearings, the longer the bearings will last. In the case of magnetic face seals, all particles greater than $1\mu m$ in diameter will be filtered out. Referring to Figure 2, this means the tiniest grains of sand and even water particles will be filtered out. As the filter grows, the bearing life decreases. In the case of compound labyrinths, filtering out particles from $30\mu m$ to 100 μm means a bearing life less than half that of the magnetic seal. As the filter grows even higher, with lip seals for instance, the modified bearing life (L_{10m}) can be significantly lower than the basic life rating (L_{10h}). Based on empirical field studies, these life estimation numbers are in line with recorded values for ANSI pumps running these various bearing protection devices. As shown, it is important not to neglect the dynamic filtered particle size when estimating bearing life.

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