Design of Cost Effective Seal to Protect Bearings Used in Conveyor Roller Housing in Mines

Amit S. Ghade, Sushil R. Lanjewar

Abstract— The primary sources of bearing failure are lack of lubrication and contaminant ingress. Industrial sealing devices are the primary protection against bearing failure. When the sealing device fails, bearing failure is imminent. Therefore, extending the life of sealing devices extends bearing life and in turn improves equipment uptime. Whether the equipment in question is a pulverizer, a turbine, conveyance equipment or something else altogether, there is usually a bearing system either driving or being driven by the equipment. In any application where power is transmitted from one point to the next, a bearing system is used to support rotating elements (usually a shaft) and to support the related loads, while at the same time reducing power losses due to friction. The most common types of bearings are ball and roller bearings. This paper investigates about various areas and factors that are important for designing a cost effective and a versatile bearing seal for roller conveyor typically used in dusty and environment found in mines and excavation sites.

Index Terms— Cost effective seal, Multiple Labyrinth seal, Radial Lip Seal.

I. INTRODUCTION

Industry sources have reported that over 90 percent of all rolling element bearings will not reach their projected design life. Millions will fail prematurely each year from such causes as dirt and moisture contamination.Bearings are among the most important components in the vast majority of machines and exacting demands are made upon their carrying capacity and reliability. Therefore it is quite natural that rolling bearings should have come to play such a prominent part and that over the years they have been the subject of extensive research. Indeed rolling bearing technology has developed into a particular branch of science. Bearings are one of the most critical components in many machines and various mechanical setups.(Refer Fig1)

Most bearing systems fail to meet their predicted life due to issues other than fatigue failure. Says one expert on bearing failure: “Only 1% [of bearings] actually fail due to pure fatigue. The majority of bearing failures are from a lubrication-related issue.” This means that approximately “95% of bearing failures can be either prevented or have their service life extended. The primary system to protect and extend the life of bearings is the sealing system. When compared to the costs of repairing or replacing the bearing system, the sealing system is much more economical to address. Typically, the sealing system protects the bearing in two ways: it reduces excessive bearing temperatures by retaining lubricant, and it prevents damage from foreign material by excluding external debris. Common sealing devices for rotating equipment include: compression packing’s, labyrinth seals, mechanical face seals, radial lip seals and hybrid combinations of these seals. For decades, radial lip seals have been the most common form of industrial bearing protection. In recent years, labyrinth seals (or bearing isolators) have increased in popularity due to their non-contact features.

Figure 1: Typical idler roller used in conveyer system in mining and similar industries

Manuscript received on April 26, 2012.

Amit S. Ghade, Research Scholar, Mechanical Department, G.H.Raisoni College of Engineering, Nagpur, India, Mobile No.09970286435, (e-mail: amitghade@yahoo.com).

Sushil R. Lanjewar, Mechanical Department, G.H.Raisoni College of Engineering, Nagpur, India, Mobile No.08956018815, (e-mail: sushil.lanjewar@gmail.com).

II. MAIN BODY OF PAPER

A. *Typical Sealing methods*

RADIAL LIP SEALS

A common misconception about radial lip seals is that the lip portion of the seal is intended to be in direct contact with the sealing surface at all times. While this was the case with early lip seal designs, modern lip seals include specialized geometries to create a hydrodynamic sealing element. These designs may include "raised helical or parabolic ribs, triangular pads, or sinuous wavy lip elements". The hydrodynamic effect causes lubricant to recirculate under the sealing lip and back into the bearing system, causing the seal to ride on a thin meniscus of oil, which significantly reduces friction and seal element abrasion. The meniscus film is typically 0.00018" (0.0046 mm) thick. It is also necessary to understand that single-lip seals are unidirectional—they can either act to retain lubricant or exclude debris, but cannot necessarily do both. For the seal orientation shown in Figure 2, the seal will only retain oil. It will not act to exclude foreign debris from the bearing system. To exclude debris in a light-duty environment, a seal with a dust or scraper lip may be used. For heavily contaminated environments, a positive excluder lip design is required.

Although in principle radial lip seals ride on a meniscus of oil, in practice this is not always the case. There will be periods, particularly at start-up and shut-down, when the seal lip is in direct contact with the shaft, resulting in power losses. As hydrodynamic sealing

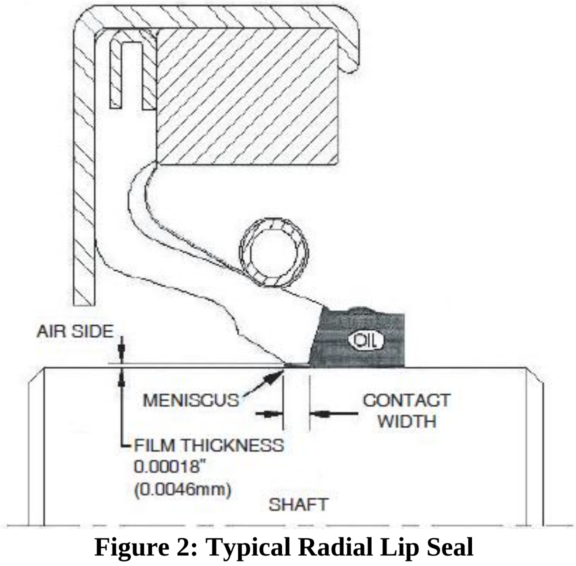

Figure 2: Typical Radial Lip Seal

is achieved, this power loss is reduced. Further, the direct contact of the sealing lip against the shaft leads to seal abrasion and eventual failure. The friction and abrasion properties of the sealing material, therefore, play an important role in seal performance. Other factors that will affect seal performance include, but are not limited to, operating temperature, pressure, misalignment and runout and bore condition. (Fig 2)

LABYRINTH SEALS

As the need for energy conservation has increased, noncontact seals have become more common place in industry. T e most common type of non-contact seal is the labyrinth seal. Traditional labyrinth seals use a tortuous pathway to block both the escape of fluids and the ingress of contaminants. They include a static portion that is mated to the application housing and has one or more inside diameter grooves. A dynamic portion of the seal is mated to the shaft and has one or more protrusions
(Sometimes referred to as teeth or knives) that run inside the grooves of the static portion of the seal. For this reason, the static portion of the seal is referred to as the stator, while the dynamic portion of the seal is referred to as the rotor.(Fig 3)

The principle of operation for a basic labyrinth seal is based in statistical motion of a particle on either side of the labyrinth. The more complex the pathway, the less likely that the particle can penetrate from one side of the labyrinth to the other. Early labyrinth seals were considered an option only in applications where some degree of leakage was allowable. Today, labyrinth seals have evolved into bearing isolators (hybrid labyrinth designs), which utilize basic labyrinth technology along with other methods of retention/exclusion including centrifugal force, pressure differential and drain back design. Today, bearing isolators can provide a much higher-performing sealing solution than traditional labyrinth seals [1].

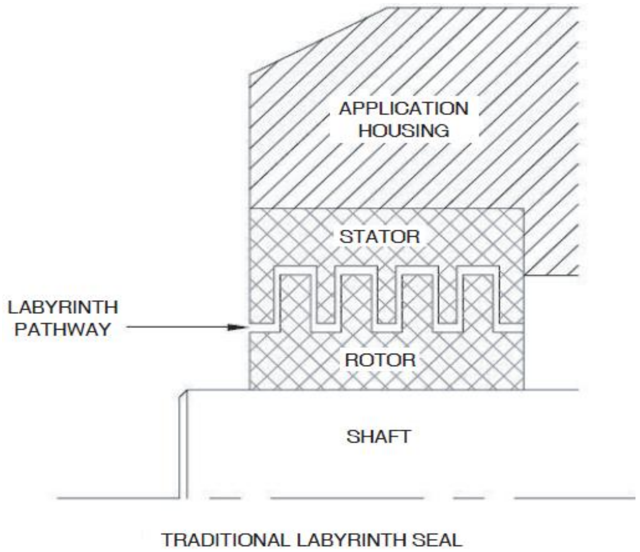

Figure 3: Typical Labyrinth seal

MAGNETIC AND SPRING-LOADED FACE SEALS

Where conventional seals use either springs or elastomeric lips to apply sealing force. Magnetic seals use magnetism. Each magnetic seal consists of two elements; one of these is a magnetized ring with an optically flat surface that is fixed in housing and sealed with a secondary O-ring. The rol11er element is a rotating ring, or seal case, which
couples the shaft and is sealed with an a-ring.(Fig 4)

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

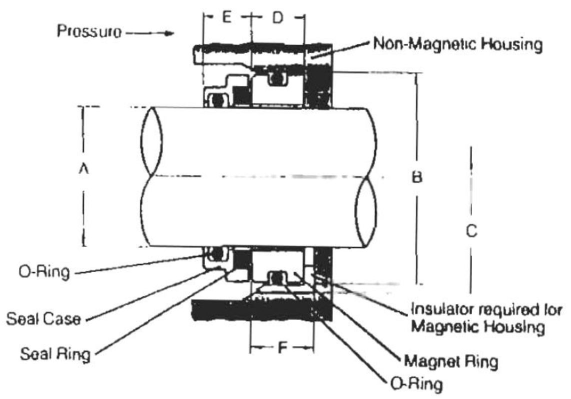

Figure 4: Magnetic Seal

When no fluid pressure exists, magnetic force holds the two sealing surfaces tightly together. This force minimizes friction between sealing faces while ensuring proper alignment of surfaces through equal distribution of pressure. Other magnetic seal ring designs employ small rare earth rod magnets instead of fully magnetized rings to provide the force of attraction. The addition of magnets. or changing their distance from the opposing rotating ring element, or seal case. will allow changes in desired magnetic attraction force[2].

B. Material Selection for Seal

The factors that contribute to seal failure are directly related to the properties of the materials used to manufacture sealing products. Common sealing materials include acrylonitrile butadiene (Buna-N, NBR); hydrogenated nitrile rubber (HNBR), fluoroelastomer (FKM—Viton), silicones, and polytetrafluoroethylene (PTFE — Teflon). Abrasion resistance and ability of the material to retain its chemical and physical properties over the period of time are most important during selection of seal. Now a day's various grades of plastic are also used in developing seals as these materials are of low cost and durable.

The material selected for this seal is Nylon-66 as this material is easily available is much cheaper than Teflon. The key potential attributes that can be obtained from plastics are summarized below. However, some of these benefits are only available from a limited number of plastic types:

- Low friction and dry running potential: Many plastics will permit dry running which is an advantage for dry or marginally lubricated seals. They can also have good boundary lubrication properties which benefit low speed or poorly lubricated applications. Many plastics offer this benefit, within the temperature and fluid limitations of the material.
- Good wear resistance possible: When compounded with appropriate filler many plastics can provide good wear resistance.
- High strength possible: Some plastics offer comparatively high strength and when assessed on strength to weight basis can often compete with metals, which can make them attractive for structural components. This high strength provides valuable attributes for sealing such as good extrusion resistance.
- Wider temperature range possible than when using elastomers: Plastics are available that can be used at both higher and more particularly lower temperatures than elastomers.
- Wide chemical resistance possible: Some plastics widely used for seals, such as PTFE, have a very wide chemical resistance. As with elastomers this is very dependent on the individual material and chemical compatibility must be checked carefully [2].

III. DEVELOPMENT OF PARTS OF SEAL

The steps involved in development of seal are

- Prepare part drawings in Pro-E according to the diameter of roller shaft and housing dimensions
- Prepare mold cavities and other mold assembly parts like core, stripper plate etc.
- Generate tool path to generate mold cavities in Power Shape software and feed the program to CNC machine
- Mount the mold on injection molding machine and test the sample product

Customer Input:
Shaft Diameter = 20mm
Housing Inner Diameter = 70mm
The seal is to be made in five parts (Fig 5). They are

- Inner race cover
- Outer race cover
- Dust cover
- Bush
- Back Seal

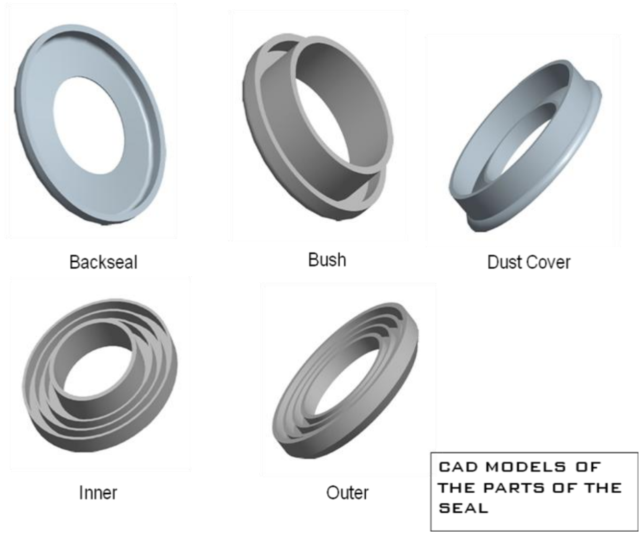

Figure 5: CAD models of Seal Parts

The assembly of the parts is shown in following fig 6

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

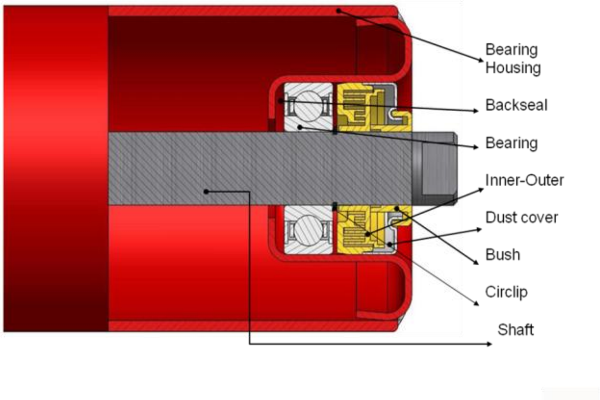

Figure 6: Assembly of seal parts

The backseal provides effective sealing to the contaminants coming from the inner side of roller housing. The outer is in contact with the seal housing and is stationary while the inner rotates inside it. The circlip provides stability of the position of all the parts of the seal. The dust cover and bush provides the final seal to the bearing. The assembly of all the parts is done manually by the workers. This type of seal can also be called as multiple labyrinth seals. This type of design reduces the chances of any contact of contaminants directly with the bearing.

The next step was to generate mold parts in Pro-E software

The mold is to be made in 4 major parts, they are

- Core
- Cavity
- Punch Plate
- Stripper plate

The other location and alignment components include

- Guide pillars and bushes
- Liner bushes
- Ejector plate guidance systems
- Register rings
- Side core slides

Ejector Components

- Ejector pins
- Blade ejectors
- Return pins
- Ejector plate early return systems
- Angled lift pins
- Blank form pins
- Spring ejectors

The figure 7 shows the mold parts generated

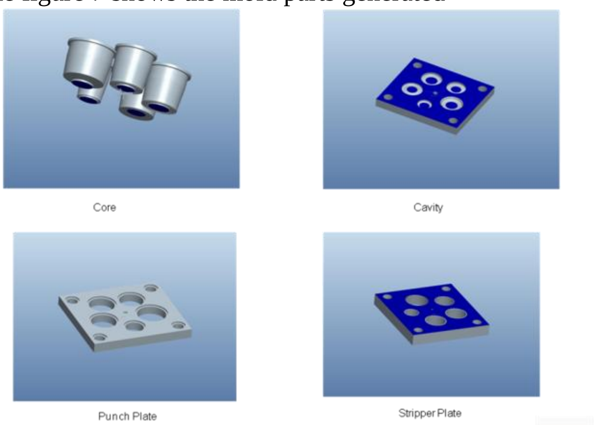

Figure 7:The mold parts generated

The assembly of mold is done as shown in fig 8

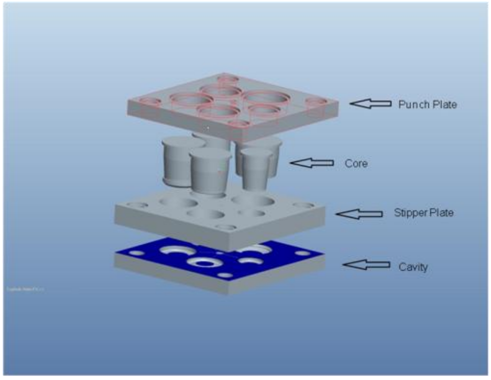

Figure 8: Assembly of mold

IV. MOLD DESIGN CALCULATIONS

The important things which are to be taken care of in any mold design are

- Production Rates
- Cooling Channel Diameters
- Runner Length
- Gate Design

1. *PRODUCTION RATES*

$$Production\ Sets = \frac{3600}{Cycle\,(s)}$$

$$Production\ Sets = \frac{3600}{8}$$

= 450 sets (approx)

2. COOLING CHANNEL DIAMETERS

Q=Mass x (Enthalpy at melt temperature - Enthalpy at ejection temperature)

= *M* x (*H*m - *He*)

Q = *0.068* x (280-20)

= 17.68 KJ

Where:

M = the shot mass in kg

*H*m = enthalpy at the material melt temperature in kJ/kg

*H*e = enthalpy at the molding ejection temperature in kJ/kg

The cooling capacity required is then this value divided by the cycle. i.e.,

$$Cooling\ Capacity\ Q' = \frac{M \times (H_m - H_e)}{C}$$

$$Cooling\ Capacity\ Q' = \frac{17.68}{8}$$

$$= 2.21\ kJ/s$$

Where *Q*′ = the cooling capacity in kJ/s.

For maximum cooling efficiency there should be a difference of 5°C between the cooling inlet and outlet temperatures. The specific heat of water is 4.19. Therefore, it takes 4.19 kJ of energy to increase the temperature of 1 kg of water by 1°C. Hence to raise it by 5 °C we would need

5 x 4.19 = 20.95 kJ.

The volumetric flow of water required to remove the heat in the mould is given by

$$V_f = \frac{Q}{20.95}\,\text{kg/s}$$

$$= \frac{Q}{20.95}\,\text{litres/s (since 1 litre of water weighs 1 kg)}$$

We also need a linear flow rate of 2.5 m/s to promote turbulent flow, hence the volumetric flow can also be expressed as V_f = 2.5 x cross-sectional area of channel. If the channel is circular we can write this as:

$$2.5 \times \frac{\pi d^2}{4} = \frac{Q}{20.95} \quad \text{(Here d is in metres)}$$

Transposing for d gives:

$$d = \sqrt{\frac{76.37 \text{ x } Q}{\pi}}\ \text{mm}$$

$$d = \sqrt{\frac{76.37 \text{ x } 2.21}{\pi}}\ \text{mm}$$

$$= 8 \text{ mm}$$

3. RUNNER LENGTH FORMULAE

Given: Mold temperature = 280°C ……….(All data provided by company)

Flow Rate of Polymer(Nylon66) = 1.8 cm^3/s

Runner Length = 100 mm

Runner Diameter = 3 mm

Viscosity = 1000 Pa-s

$$\dot{\gamma} = \frac{4Q}{\pi r^3}$$

$$\dot{\gamma} = \frac{4 \times 1.8 \times 10^{-6}}{\pi \times 1.5^3}$$

$= 0.679 \times 10^{-3} = 679 sec^{-1}$

$$\tau = \eta\dot{\gamma}$$

$$\tau = 1000 \times 0.679 \times 10^{-3}$$

= 0.679 MPa

$$P = \frac{2\tau L}{r}$$

$$P = \frac{2 \times 0.679 \times 100 \times 10^{-3}}{1.5 \times 10^{-3}}$$

= 90.53 MPa

As this figure is well below the maximum allowed figure of 120 MPa, the runner length is satisfactory.

Where,

γ = shear rate (s^{-1})

Q = flow rate (m^3/s)

r = runner radius (m)

η = viscosity of material at melt temperature (Pa-s) P = pressure drop (MPa)

τ = shear stress (MPa)

L = runner length (m)

4. <u>GATE DESIGN</u> (Page 370 Section 17.4.2)

$$d = NC\sqrt[4]{A} \quad (where\ d = gate\ diameter)$$

$$d = 0.8 \times 0.206\ \sqrt[4]{1355}\ (where\ d = gate\ diameter)$$

= 1mm

Parameters	**Values**
Production Sets/ Hour	450 sets
Cooling Channel Diameters	8mm (Circular Channel)
Runner Radius (r)	3mm
Gate Design (d)	1 mm

These are the summarized results for design of mold made for bearing seal[3].

V. BEARING LIFE CALCULATION

The bearing life for the idler rollers used in typical mining industry is found out to be 10712 days [7].

Idler Bearing Details			
Shaft Diameter	Bearing Type	Dynamic Bearing Rating (N)	Wear rating (N)
20	6204	12700	0
25	6205	14000	28000
25	6305	22500	0
25	420205	0	0
30	6206	19500	0
30	6306	28100	0
30	420306	0	0
35	6307	33200	0
40	6308	41000	0
50	6310	61800	0
60	6312	81900	0

Usage Factor	
U(f) Usage Factor	Description
1.0	24 hrs/day
0.7	12 hrs/day
0.5	6 hrs/day

Seal efficiency Factor	
(Sef)	Description
0.75	most efficient sealing
0.65	Hi-torque contact seal
0.6	Extended labyrinth seal
0.5	For shaft deflection >0.004 radians (13.75 min.)
0.4	Seals with grease churning action

Environmental Factor	
E (f)	Description
0.11	Conveyors subjected to occassional flooding and concentrated dusts-underground mines
0.95	Conveyors subjected to concentrated dust and rainfall - plant conveyors
0.77	Conveyors subjected to natural countryside and rainfall - overland conveyors

Input Values

Bearing Rating [N]	Load on Center Roll [kg]	Belt Mass [kg]	Belt Speed [m/s]	Roll Dia [mm]
12700	200	10	11	70
Idler Spacing [mm]	**Seal Eff. Factor [Sef]**	**Environmental Factor [Ef]**	**Usage Factor [Uf]**	**Bearing Wear Rating [N]**
800	0.6	0.11	1.0	0

Reset Calculate

Output Values

B_10 Life[Hrs]	Wear Life [Hrs]
10712	0

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

VI. CONCLUSION

Although much research related to bearing seals have being done but this type of seal provides a much simpler and cost effective solution to seal the bearings which are installed in many numbers. Its cost and simple design provides a distinct advantage in its operation and which can be adopted to gain maximum life of bearing. This type of sealing arrangement has wide areas of application. Improving equipment life is a continuous process. Demands for increased efficiency and decreased power consumption will continue. Thus the need to innovate will continue, building on existing technologies and developing new ones.

REFERENCES

1. David C. Roberts; *Improved Sealing Technology Extends Equipment Life* , Presented at Power-Gen International 2007 New Orleans.
2. Flitney R.; *Seals and sealing Handbook,* Fifth Edition 2007 Butterworth-Heinemann; pp. 394.
3. Peter Jones; The Mould Design Guide, Smithers Rapra Technology Limited 2008; pp. 447- 449.
4. Heinz P. Bloch; Better bearing housing seals prevent costly machinery failures, Fifteenth National Industrial Energy Technology Conference, Houston, Tx, March 24-25, 1993.
5. K.Yamamoto, D.Ozaki, T.Nakagawa; Influence of Surface Roughness on Sliding Characteristics of Rubber Seals; Koyo Engineering Journal English Edition No.166E (2005).
6. Charles A. Harper; Modern Plastics handbook; *McGraw-Hill Publication.*
7. http://www.ckit.co.za/secure/conveyor/troughed/idlers/idlers_calc_bearing_life.html.

AUTHORS PROFILE

Amit S. Ghade has completed his Bachelor of Engineering (B.E) in Mechanical Engineering from Yashwantrao Chavan College of Engineering, Nagpur under R.T.M. Nagpur University with 60.73% and is pursuing his Master in Technology (M. Tech.) in CAD/CAM at G.H. Raisoni College of Engineering (Autonomous), Nagpur,Maharashtra.

Sushil R. Lanjewar has completed his Bachelor of Engineering (B.E) in Production Engineering from K.B.P College of Engineering, CAMP, Satara, Affiliated to Shivaji University, Kolhapur with 65.76% in June 2006 and Master in Technology (M. Tech.) in CAD/CAM from Yeshwantrao Chavan College of Engineering, Affiliated to R.T.M. Nagpur University with 67.75 % in August 2008.

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication