DEVELOPMENT AND EXPERIENCE WITH A NEW THRUST BEARING SYSTEM FOR CENTRIFUGAL PUMPS

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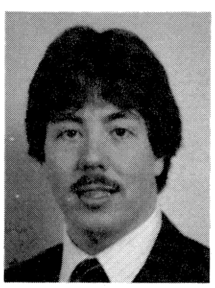

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ABSTRACT

The general use of rolling element bearings in centrifugal pumps comes as a result of their low power consumption, high stiffness and high load support capabilities. Although these bearings are popular in the light to medium duty pump applications, they are sometimes cause for concern in certain pump applications. Thrust load application of rolling element bearings has the attention of bearing specialists and machinery engineers, because less than ten percent of the bearing population ever reach their theoretical expected life. This has motivated improvements in manufacturing tolerances and processes, material selection and lubrication. The result of which has been an improved service factor for rolling element thrust bearings over the years. However, further development should concentrate on areas, other than rolling contact fatigue, that might get us closer to the theoretical life.

INTRODUCTION

Experience has shown that the majority of costly thrust bearing failures in centrifugal pumps are not related to end-of-life or fatigue phenomena. Nevertheless, these failure events have prompted the hydrocarbon processing industry to push for specifications (i.e., API 610) requiring forty degree angular contact ball bearings for thrust applications in centrifugal pumps. This requirement further specifies an L_{10} fatigue life of 25,000 hr and a light preload. These specification clauses can be traced back to the fact that higher angular contact ball bearings tend to predict larger theoretical thrust capacities.

Numerous failures with these theoretically stronger bearings prompted a fundamental reassessment of this philosophy. Fatigue is only one of many failure mechanisms. In centrifugal pumps, bearings rarely achieve their theoretical (i.e., L_{10}) life. Therefore, other contributing factors must be examined. These include internal dynamics of rolling elements, lubrication, installation, and machine operating parameters.

Internal dynamics examines the effects of thrust loads, contact angle orientation, ball mass, rotational speed and bearing geometry on actual bearing life.

Lubrication is a factor assumed adequate and therefore not addressed.

Poor installation techniques can cause premature failures, therefore, dimensional accuracy requirements with regards to shaft fits, housing fits, and axial locations along the shaft must be adhered to during installation. Oversized diameters invite overheating and preloading while undersize shafts contribute to fretting. For thrust bearing installation, a shaft interference fit of no more than 0.0005 in and ISO-H7 fit for bearing housing to outer race is recommended. Special attention should be paid to axial tolerance and perpendicularity of locating shoulders, since out-of-tolerence conditions can induce both misalignment and cross loading of the bearings.

Finally, *pump operating parameters* change with varying process conditions and, therefore, any thrust bearing system designed specifically for centrifugal pumps must permit operation for a wide range of applications.

Three cases involving persistent thrust bearing failures in centrifugal pumps are cited, the solutions implemented are elaborated, and a new thrust bearing which shows promise for centrifugal pump applications is discussed.

CASE HISTORY NUMBER 1

Pump Data

Case History Number 1

Pump Data

Suction Pressure	415 psig
Discharge Pressure	490 psig
Differential Head	361 feet
Rotational Speed	3550 cpm
Suction Specific Speed	18000
Service	Cryogenics/Light Hydrocarbon
Shaft Size Through Stuffing Box	3⅝ in
Original Thrust Bearing Type	7314 duplex, back-to-back
Type of Construction	API double suction overhung

Background. During the first two years of operation of these pumps, eight thrust bearing failures were recorded. Numerous seal failures also occurred, twenty-five percent of which were directly related to the bearing failures. The average life experienced by these thrust bearings was approximately three months. The pumps are classified as critical service pumps and, therefore, a fairly high reliability was expected. The thrust bearings specified for this service originally by the OEM were forty degree angular contact duplexed ball bearings mounted back-to-back. Bearing housing temperatures slightly in excess of 200°F would be experienced on startup immediately after new thrust bearings were installed. If the bearings survived the first forty-five minutes, the temperature would gradually fall with a final "settling out" temperature of approximately 170°F. Sometimes, two attempts were made before a set of thrust bearings would survive. During this period, several enhancements of the forty degree angular contact ball bearing were attempted with marginal success. The enhancements included light preloads and several different cage designs. The best results were achieved with a non-preloaded (i.e., flush-ground with no preload and no end-play) forty degree angular contact ball bearing with a machined bronze separator.

The mean-time between failures for the thrust bearing improved from three months to eight months with bearing housing temperatures operating at 150°F. The calculated L_{10} life for this application was four months. On inspection, the unloaded or inactive bearing would fail. The improvement gained in reliability was appreciated but still unacceptable for this service. A thrust bearing pair consisting of a split inner ring bearing mounted in tandem with a 7000 Series angular contact bearing was installed with immediate success [1]. Both bearings had contact angles of 29 degrees. The tandem pair is designed to have 63 percent more capacity in the active direction than a typical 29 degree back-to-back or face-to-face duplex pair, while maintaining equivalent capacity in the reverse thrust direction. Bearing operating temperature dropped to 130°F and mean-time between failures increased to 18 months. In an effort to further improve the reliability of these pumps, a split inner ring bearing having a 40 degree contact angle was mounted in tandem with a 40 degree 7000 series bearing, thus increasing the theoretical capacity. The bearing failed within the first sixty minutes of operation. Excessive heat was noted. Another attempt with this same type of bearing had similar results. The reason these theoretically better bearings failed was not understood at the time. However, the reason the forty degree back-to-back duplex and 40 degree tandem bearings failed is now understood and led to the development of a new thrust bearing discussed later.

CASE HISTORY NUMBER 2

Pump Data

Case History Number 2

Pump Data

Suction Pressure	1 psig
Discharge Pressure	205 psig
Differential Head	507 ft
Rotational Speed	3560 cpm
Suction Specific Speed	15000
Service	Hydrocarbon
Shaft Size Through Stuffing Box	2.5 in
Calculated Thrust Load	<20 lbf
Original Thrust Bearing Type	7313 40 degree back-to-back mounted
Pump Type	API-Single Suction Overhung

Background. The original thrust bearings installed in these pumps by the OEM were 7313 duplexed angular contact ball bearings with 40 degree contact angles, no internal preload-/no end-play, mounted back-to-back with a pressed brass separator. The calculated thrust load on these bearings allowed for a theoretical L_{10} life expectancy in excess of five years. The first thrust bearing failure occurred after less than 12 months of operation. The inactive bearing had a higher intensity of brinelling marks as well as discoloration from intense heat. Skidding of the balls in the inactive bearings created excessive heat which produced a phase transformation causing softening of the material. The softer material is more easily brinelled. Details of the skidding mechanism are discussed later. Extremely high thrust loads were caused by cavitation resulting in failure of the thrust bearings. Attempts to improve the operating conditions of the pumps to some acceptable level proved unsuccessful. A slight improvement in thrust bearing reliability was achieved by installing bearings with machined bronze separators. This improved the mean-time-between-failures just in excess of 12 months which was still unacceptable. The fact of the matter is that there was no way to improve the suction conditions without reconfiguration of the piping system and possible change out of the pumps. This costly alternative was more than enough motivation to concentrate on efforts to improve thrust bearing reliability.

Thrust bearing reliability improved from an mean time between failures (MTBF) of 12 months to 18 months by installing a 29 degree angular contact split inner ring bearing mounted in tandem with a 29 degree 7000 series bearing similar to that used in case Number 1. Bearing housing temperatures dropped by approximately 30°F down to 120°F. In an effort to further improve reliability of these pumps, a new bearing design was installed that would incorporate the design advantages of the split inner ring bearing, without the inherent disadvantage of bearing (i.e., rotor) end-play. This new design has proven to be successful having been tested in excess of 12 months under adverse operating conditions caused by continuous cavitation. The expectation, based on cooler operating temperatures (e.g., 110°F) and low vibration levels, is that these bearings should achieve an actual life that is closer to the theoretically predicted fatigue life.

CASE HISTORY NUMBER 3

Pump Data

Case History Number 3

Pump Data

Suction Pressure	12 psig
Discharge Pressure	250 psig
Differential Head	280 ft
Rotational Speed	3560 cpm
Suction Specific Speed	13000
Service	Hydrocarbon
Shaft Size Through Stuffing Box	2.75 in
Calculated Thrust Load	1000 lbf
Original Thrust Bearing Type	7313 40 degree Back-to-back
Type of Construction	API-Single Suction Overhung

Background. This pump experienced two bearing failures during the first three years of operation while operating at designed conditions. The reliability decreased when the service operating conditions changed. The effect of these changes was to promote low flow conditions resulting in internal recirculation type cavitation and high thrust loads. The MTBF decreased from 18 months to approximately eight months. The high thrust loads were verified by performing an actual field test to measure the generated loads. It was found that the thrust loads were much higher than calculated, as shown in Figure 1, resulting in failure of the inactive bearing. Signs of extreme heat generation, phase transformation and deep raceway brinelling were evident. A solution similar to that in Case Number 2 was followed to improve the service factor of the pumps operating in this application.

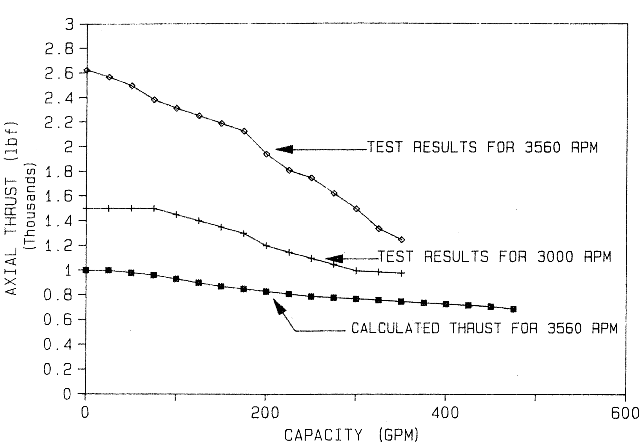

Figure 1. Generated Thrust Load vs *Capacity Obtained from field test.*

Internal Dynamics of Angular Contact Thrust Ball Bearings

The sixth edition of API Standard 610 requires the use of back-to-back mounted 40 degree angular contact ball bearings. A typical installation of this arrangement is shown in Figure 2. Because of this orientation, only one bearing is actively loaded in the thrust direction. During normal operation, the loaded or "active" bearing deflects an amount proportional to the applied load. As shown in Equation (1):

$$\text{Axial Deflection } A = U\ \frac{(C_T \times d)}{2} \quad (1)$$

where

C_T = Total Curvature:
d = Ball Diameter
U = a constant directly proportional to thrust load [3]

Total curvature is defined as the sum of the inner and outer ring raceway curvatures. This is given as a percentage since raceway curvature is the sized relationship between race groove radius compared to the ball radius. The relationship between thrust load and axial deflection for a typical 7313, 40 degree angular contact ball bearing is shown in Figure 3. For example, a single stage API pump generating a thrust load of 3000 lbf with a 7313 sized duplex thrust bearing system installed would experience an axial deflection of 0.00146 in. Since the bearing rings are locked together, the unloaded or "inactive" bearing is relieved by an equal amount which can create an internal clearance in this bearing. Contact angles are a function of internal clearance as shown in Equation (2).

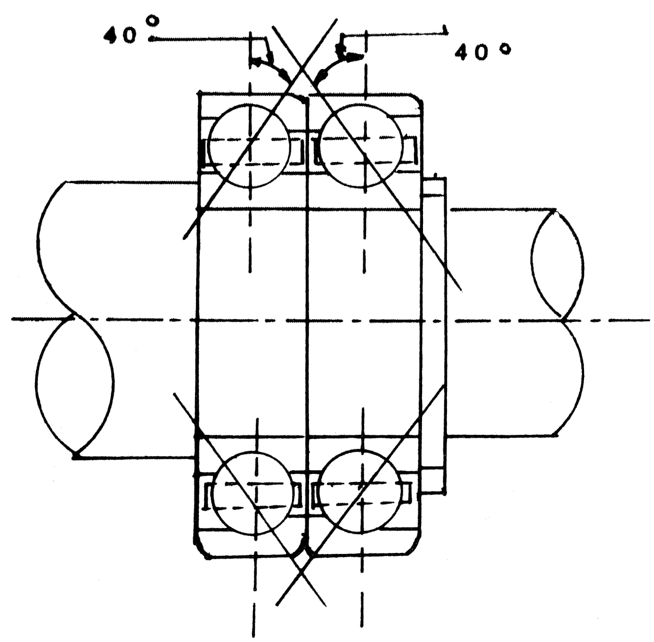

Figure 2. Orientation for an Unloaded 40 Degree Angular Contact Bearing.

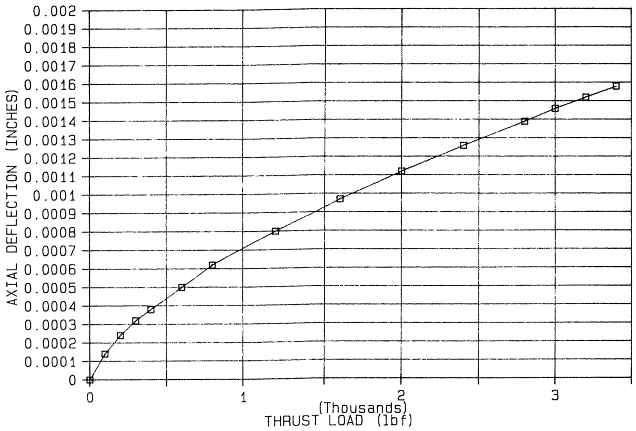

Figure 3. Load Deflection Characteristic for a 7313 40 Degree Angular Contact Bearing.

$$R_C = C_T \times d\ (1 - \cos\alpha) \qquad (2)$$

where

R_C = Radial Clearance

α = Contact Angle

C_T and d are as defined above [3]

From Equation (2), it can be seen that the larger the contact angle, the greater the design internal clearance of the bearing. This internal clearance coupled with relatively high rotational speeds causes a deviation from the designed contact angle orientation as shown in Figure 4. Centrifugal force causes the balls in the inactive bearing to be thrown outwardly and climb the raceways. As the balls gain and lose traction, they shuttle between the races causing vibration and heat. The frictional heat generated causes degradation in the lubrication mechanism resulting in metal-to-metal contact between balls and raceways. This can be recognized by a burnished appearance of the raceways. As bearing operation continues, heat generation is exacerbated by the fact that the raceway is too smooth to support the requisite oil film. Prolonged operation under these conditions will result

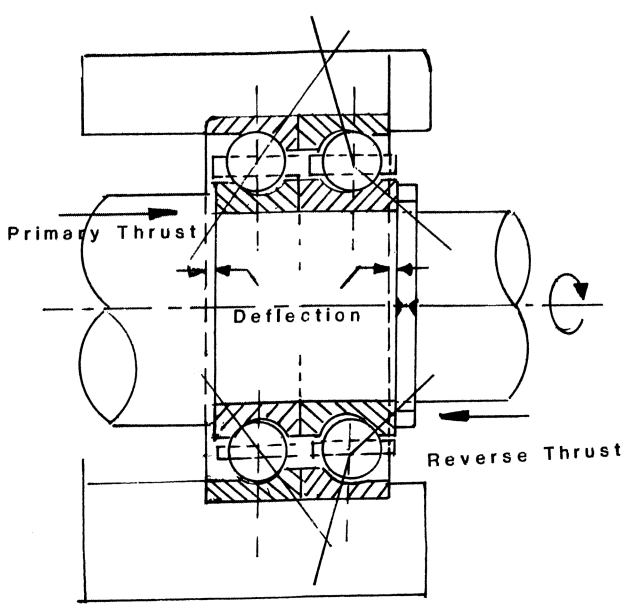

Figure 4. Bearing Orientation with Applied Axial Load.

in phase transformation of the steel allowing for a softening of the metal and imminent failure, the appearance of which closely resembles that of classical lubrication failures. The point worthy of note is that the loss of lubrication is the effect and not the cause. In order to minimize the skidding phenomenon, bearing manufacturers have recommended using preloaded bearings and/or high precision angular contact ball bearings with closer tolerances. Neither of these solutions prevents the total relief of the balls in the inactive bearing. Continuing with the example mentioned earlier where the pump is generating a thrust load of 3000 lbf, the axial deflection for the inactive bearing is calculated as shown below using 7313 duplex 40 degree angular contact bearings mounted back-to-back with a light preload of 310 lbf.

From Figure 3

Axial deflection associated with preload of 310 lbf = 0.00032 inches

Normal axial deflection without preload due to 3000 lbf thrust load = 0.00146 inches

Total deflection of inactive bearing = (0.00146-0.00032) inches
= 0.00114 inches

The inactive bearing deflects a total amount of 0.00114 in, the first 0.00032 in of deflection relieves the preload and the inactive bearing is then relieved by a resulting deflection of 0.00082 in (0.00114 − 0.00032 in). The resulting deflection of the inactive bearing will cause the balls to skid, as described earlier. Increasing the preload will only generate additional heat leading to a thermal runaway situation in the bearings.

The cases discussed herein are classical examples of the previously mentioned phenomena. The heavy thrust loads caused significant deflections of the active bearing and relieved the inactive bearing by an equal amount. The failure mechanism was initiated in a manner similar to that explained earlier. The survival of the split inner ring/7000 Series tandem combination in these applications can best be explained by the fact that since the bearings are acting in tandem, there is no inactive row of balls to be relieved. However, if there is one disadvantage to this design, it would be the inherently large end-play necessitated by bearing design.

DESIGN FEATURES OF THE NEW THRUST BEARING

The drawbacks and benefits experienced by typical angular contact bearings were examined in depth resulting in a new bearing system designed with the following criteria:

- Reduction of the active bearing deflection by tightening the race curvatures, thus spreading the contact forces over a wider area reducing internal stresses and decreasing the amount of relief in the inactive bearing.
- Utilization of the benefits of a back-to-back mounting arrangement which eliminates shaft end-play, therefore improving mechanical seal life.
- Minimization of ball shuttling in the inactive bearing. This is achieved by designing a bearing with less internal clearance and, therefore, a lower contact angle as shown in Figure 5. This also translates to lower inherent vibrations and operating temperatures.

The new thrust bearing system utilizes one 40 degree angular contact ball bearing as the active bearing and is manufactured with closer race curvatures to maximize axial rigidity, thus decreasing axial deflection and the amount the inactive bearing is relieved. This is mounted in a back-to-back orientation with a 15 degree angular contact ball bearing as the inactive bearing (Figure 5). The 15 degree contact angle ball bearing was designed to further improve ball stability by preventing shuttling, as a result of the smaller internal clearances. The inactive bearing is still capable of handling reverse thrust loads, but at slightly reduced capacity.

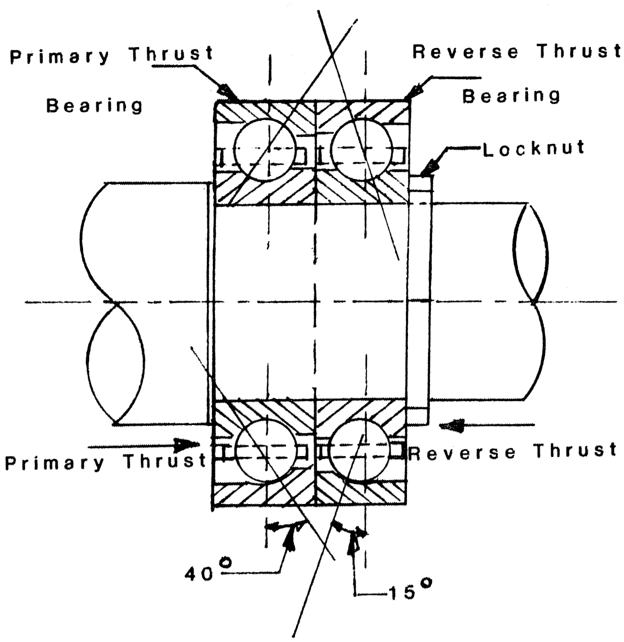

Figure 5. Normal Orientation of New Thrust Bearing.

These bearings have been successfully applied in centrifugal pumps operating in hydrocarbon and slurry services. There are at least ten applications in which these bearings are presently operating in excess of 12 months.

CONCLUSIONS

• Testing as well as field experience have corroborated with analytical predictions for this new bearing. This has resulted in lower vibration levels and operating temperatures.

The mandatory adherence to a general bearing specification for all API centrifugal pump applications can limit the selection for thrust bearing optimization. The guidelines given by API 610 is much too rigid in trying to accomplish a criteria that would satisfy all aspects of thrust load applications. As such, we would suggest full understanding of the thrust loads which might be experienced and recommend that the following thrust bearing selection criteria for centrifugal pump applications be considered.

A. Pump Type:

• Overhung double or single suction pump.

Conditions:

• Low Speed (≤ 1750 cpm)

• Any load condition.

Recommendation: A duplex 40 degree angular contact ball bearing system or this new thrust bearing system is acceptable. Since at these low speeds centrifugal effects on the balls is not as significant as operating at higher speeds, shuttling is not a concern for the 40 degree angular contact bearing.

B. Pump Type:

• Overhung double or single suction pump.

Conditions:

• Speed in excess of 1750 cpm.

• Large thrust loads or heavy thrust loads coupled with radial load.

Recommendation: Use a bearing system similar to the new design.

C. Pump Type:

• Double Suction between bearings.

Conditions:

• Speeds in excess of 1750 cpm.

• Predominantly radial loads low thrust loads.

Recommendation: Two 15 degrees angular contact bearings mounted back-to-back are preferred, if thrust bearings are to be used in this type of application.

• Finally, there are criteria other than theoretical fatigue life which must also be considered in thrust bearing selections. This challenges the Hydrocarbon Processing Industry and bearing manufacturers to jointly develop other parameters, in addition to fatigue, which will more accurately predict actual bearing life. Because bearings may fail from unstable motion of the balls [2] then one such parameter could be based on ball stability.

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