

Improved Automobile Engine Shaft Thrust Pump Cessation Detection

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Automobile thrust shaft bearing is the most vital module of water pump based on its fatigue cessational consequences that causes the abnormal operation of the system. Conversely, the cessational mechanisms of the bearing are still unclear. In this article, the cessational analysis on the engine thrust shaft bearing was carried out by using the measurement of material composition and properties, observation of macro and micro morphologies, and theoretical analysis of the cessation. The wear behavior and cessational mechanisms were elucidated, and the prevention measures were further proposed. The cessation was identified from the fracture of the bearing cage causing severe wear of the roller and the mandrel originated from the surface fatigue and the cessational attributes to the great radial deflection on load. Additionally, the decreased weight of the impeller and the shaft connector and the increased bearing thickness were recommended to improve the bearing operation.

Keywords: Thrust Shaft Cessation, Detection, Rolling Contact Fatigue, Wear Behavior, Cessational Mechanism

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I. INTRODUCTION

Automobile engine thrust shaft bearing being small in size, compact in structure, and low cost, it has been generally used as the engines driver through the crank shaft to circulate the coolant through water jackets dissipating heat under the operation pressure, which promotes the transformation of kinetic energy into pressure energy (Liu et., al (2022), (Nanee and Kishore (2022)). Due to the superior specific load moving ability, the thrust bearings have been extensively utilized in the system of the water pump to reduce the friction between contacted surfaces, and considered as the major components of water pump (Thamaraiselvan et. al (2016). Equally, as the connection between the rotor and support, the safety and stability of thrust bearings are key to ensure the normal operation of water pump. With the development of science and technology, the increasingly harsh conditions of the thrust bearing require higher performance and the thrust shaft bearing breakdown inevitably results in the major engineering equipment losses, economic losses, and casualties (Zhang et., al 2017) (Ghong et., al 2017). Therefore, it is of significance to understand the breakdown mechanism of the pump shaft bearing during operation in order to prevent long-term cessation or/and catastrophic cessation. Generally, the bearing cessation was attributed to the interacted breakdown factors of the bearing, including human factors, equipment factors, material, processing methods and qualities, and service condition (Nabanita et., al 2022), (Guo et., al 2021). A large number of cessation analysis cases in the previous references that include the power shift transmission shaft Ganesh et., (2024), (Perera 2022) confirmed that the bearing life was obviously influenced by the material quality (large inclusions, defects), heat treatment quality [8] stress concentration unreasonable design, as well as rear axle shaft (Perera 2022), improper process (Mukhopadhyay and Bhattacharya (2024), environmental factors (Liu(2014), (Yang et., al (2017), Yu and Yang (2011), machined surface quality (Turan et., al 2017), and overload (Jalilvand et., al 2026), (Gurumoorthy et., al 2015). To the best of our knowledge, there is no systematic and effective detection analysis method for the water pump shaft bearing. Therefore, it is crucial to find a scientific analysis method to obtain the mechanisms of the thrust bearing. Herein, we proposed a detailed cessational analysis method for the water pump cessation, and the main analysis was sequentially summarized as follows. Firstly, the relevant background information of failed bearing, including the bearing operation records, failed bearings and debris, bearing manufacturing process and service status (the use of bearings and daily maintenance), and the bearing materials based on the relevant quality inspection report and related standards was collected. Secondly, the damage morphology of failed bearing was observed and the tests of lubricating oil performance and bearing technology were carried out. Then, the chemical composition, hardness, and microstructures were also determined. Finally, the wear mechanisms and breakdown mode of the bearing, based on the furrow, cracks, peeling off, and defects, were comprehensively clarified. In this paper, according to the above methods, the diversified breakdown analysis was conducted to elucidate the fatigue cessational mechanisms of the water pump thrust shaft bearing, as well as the proposition of prevention strategy.

II. MATERIALS AND METHODS

2.1 Pump Shaft Bearing

As shown in Figure 1, the automobile engine shaft thrust pump shows a simplified double bearing system. The left side near the impeller is the deep groove ball bearing, while the right power end is cylindrical roller bearing. The structure is suitable for high load and impact load. There is no inner ring about the two supporting bearings, and the rolling body is directly mounted on the shaft. The shaft is generally treated by surface induction hardening. The depth of hardened layer is 1.8–2.5mm, and other items are required to comply with relevant requirements of GB/T 5617-2005.

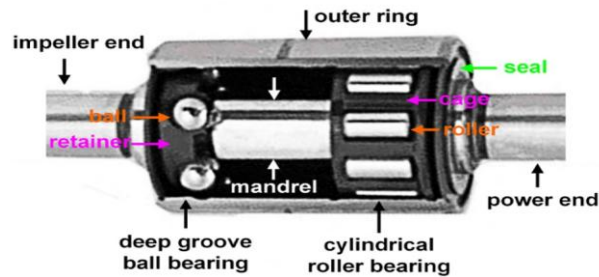


Figure 1. Sectional View of the Integral Bearing Structure of Automobile Engine Shaft thrust Pump

The damaged bearing installed on the water pump shaft of the automobile engine, after running mileage of about 600,000km, was dismantled for failure analysis. The bearing with type of WR2552185 has abnormal vibration and noise, and the materials are GCr15. The macroscopic morphology is shown in Figure 2. The outer ring of the two supporting bearings is made into a whole. Seals are used on both sides of the ring to form the composite parts of the bearing.

2.2 Composition

The chemical composition of mandrel sample and rollers was determined by X-ray spectrometry, using SPECTOR-M9 direct-reading spectrometer according to GB/T18254-2016 standard. Through the comparison with standard sample, the average value was obtained by measuring three times.

2.3 Hardness Measurement

The Rockwell hardness was measured using a Rockwell hardness tester. A load of 150kg and a dwell time of 30s were used on the mandrel surface of the spindle complying with the GB/T34891-2017 standard. Three points were chosen at intervals of 120 in the circumferential direction, and each point was measured three times to obtain the average value. To measure the micro hardness distribution of the mandrel, HTV-1000 Vickers hardness tester was used under a load of 1kg.

2.4 Microstructural Characterization

The metallographic sample were grinded with SiC paper (80–2000 grade), then mechanically polished with 0.5mm diamond particle spray, followed by being etched with a solution containing HNO_3 (4ml) + $\text{C}_2\text{H}_5\text{OH}$ (96ml). The microstructures in the specimen were observed by DMI5000M optical microscopy. For SEM analysis, the surface of the sample was first ultrasonically cleaned. The worn surface morphologies of the mandrel were imaged on the JSM6380LV scanning electron microscope (SEM) operated at 20kV. Elemental analysis of filtered debris was performed by an equipped INCA7582 spectrum of energy dispersive (EDS).

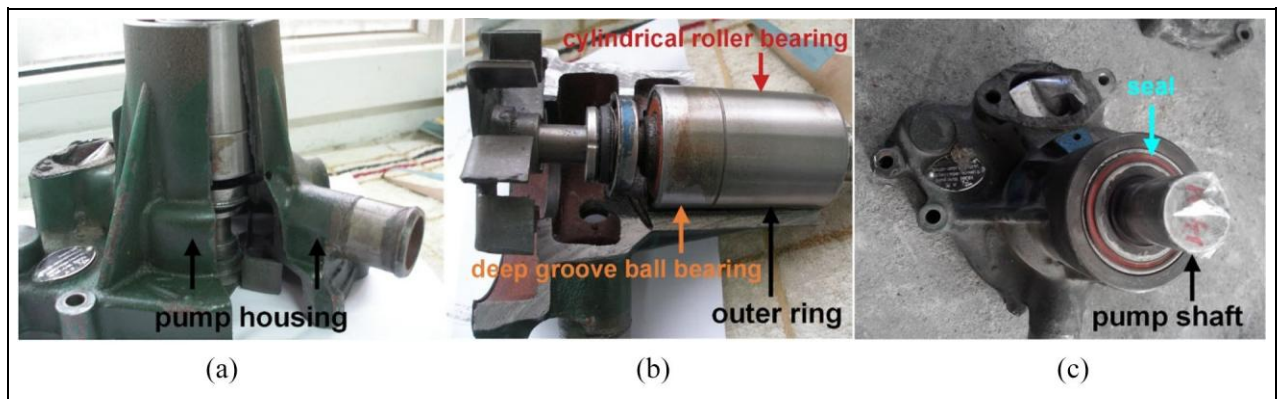


Figure 2. (a) Macroscopic Morphology of Pump Shaft Bearing, (b) Side View, and (c) Top View of Pump Shaft Bearing Without Pump Housing

III. RESULTS

3.1 Analysis of the Bearing

The thrust shaft bearing of the pump was intact without obvious damage, and the bearing seal was unbroken, as shown in Figure 2. After disassembly, the overall bearing consisted of the deep groove ball bearing on the impeller side and the cylinder roller bearing on the power side as shown in Figure 3(a). To clarify the cessational mechanism of the pump bearing, the macroscopic morphologies of bearing groove, steel ball, and bearing retainer were examined separately. The core shaft groove surface and steel ball surface were bright without obvious running traces, as shown in Figure 3(b) and 3(c). In addition, there was no obvious damage on the cage as depicted in Figure 3(d). Note that the lubricant of deep groove ball bearings was dark brown and bright without no obvious discoloration. Therefore, it was concluded that the deep groove ball bearing on the impeller side operated normally.

The components of the cylinder roller bearing including outer ring, cylindrical roller, and bearing cage were disassembled and examined respectively, as well as the mandrel surface. The raceway colour of outer ring was dark gray with local circumferential wear traces and pits, as shown in Figure 4(a). There were many circumferential wear traces on the surface of the raceway of mandrel, which were distributed along the circumference and accounted for about 2/3 of the whole circumference (Figure 4(b)). Among the seven rollers, surface colour was gray-black and there were evenly distributed pits and wear traces. At the same time, the end faces of cylindrical rollers were bright without obvious damage as shown in Figure 4(c). The cage was seriously damaged and unable to be completely spliced together, and there were more fractured parts on the both ends and less damage to the longitudinal girder, Figure 4(d). Therefore, it was inferred that the damage of bearing cage occurred first at the both ends. Consequently, the vibration and noise abnormalities of the water pump shaft bearing were mainly attributed to the failure of the cylinder roller bearing on the power side. Therefore, the following failure analysis mainly focused on the hardness distribution, crack propagation, and fatigue behavior of the cylindrical roller bearing.

3.2 Composition Analysis and Hardness Distribution

The chemical composition of mandrel and rollers on the power side were measured and the findings are listed in Table 1. The results showed that the contents of C, Si, Mn, and Cr in mandrel were higher than that in rollers, but are within the range of standard ingredients, complying with the relevant provisions of GB/ T18254-2016.

The measured hardness on the failed roller and mandrel surfaces are also given in Table 2, as well as the standard hardness according to the standard GB/ T34891-2017. The surface hardness was approximately to be 64 HRC. The hardness and uniformity of the tested parts meets the requirements of the standard.

3.3 Filtered Metal Debris without Pollution

It was worth noting that the lubricating grease of cylindrical roller bearing parts was seriously oxidized and turned black. After cleaning and filtering with clean gasoline, only a small amount of metal debris was found to be free of foreign bodies as shown in Figure 5(a). Through the energy spectrum analysis, the main chemical elements were detected to be Fe, Cr, and O without other abnormal elements, as shown in Figure 5(b).

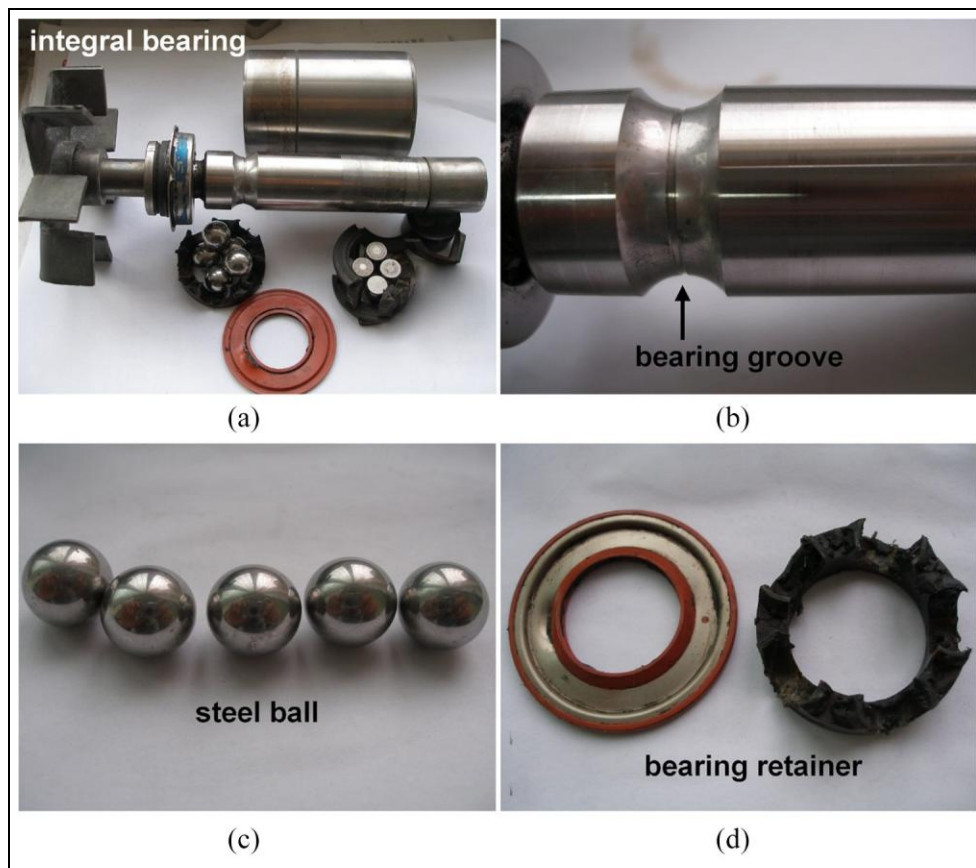


Figure 3. Disassembly Morphology of Pump Shaft Bearing: (a) Integral Bearing, (b) Groove of Deep Groove Ball Bearing, (c) Bearing

IV. DISCUSSIONS

4.1 Wear Behavior and Mechanism

Figure 4 presents uneven worn surface of the failed mandrel at two randomly selected locations. The wear characteristics, including furrow, peeling off, and micro-cracks, were mainly found on the worn surface. During the bearing operation, the friction and wear occurred on the contact surface. The wear produced debris and wear particles as depicted in Figure 9(c) and (f) with high hardness, which could not be discharged in time and remained on the contact surface. The abrasive dust rubbed repeatedly between the contact surfaces and played a role of ploughing, resulting in the generation of furrow, as shown in Figure 9(a) and (d). Due to the eccentric loading, the local stress at the surface increased. The high temperature generated as a result of the friction process led to the increased temperature of the contact surface and the formation of secondary quenching and tempering layers. Additionally, the abrasion resulted in the uneven surface and the stress concentration at the local contact points, which caused the surface peeling and micro-crack as shown in Figure 9(b) and (e). As the bearings continued to roll, the micro-cracks propagated and produced the peeling off. Consequently, it was concluded that the bearing had surface origin fatigue failure after rolling.

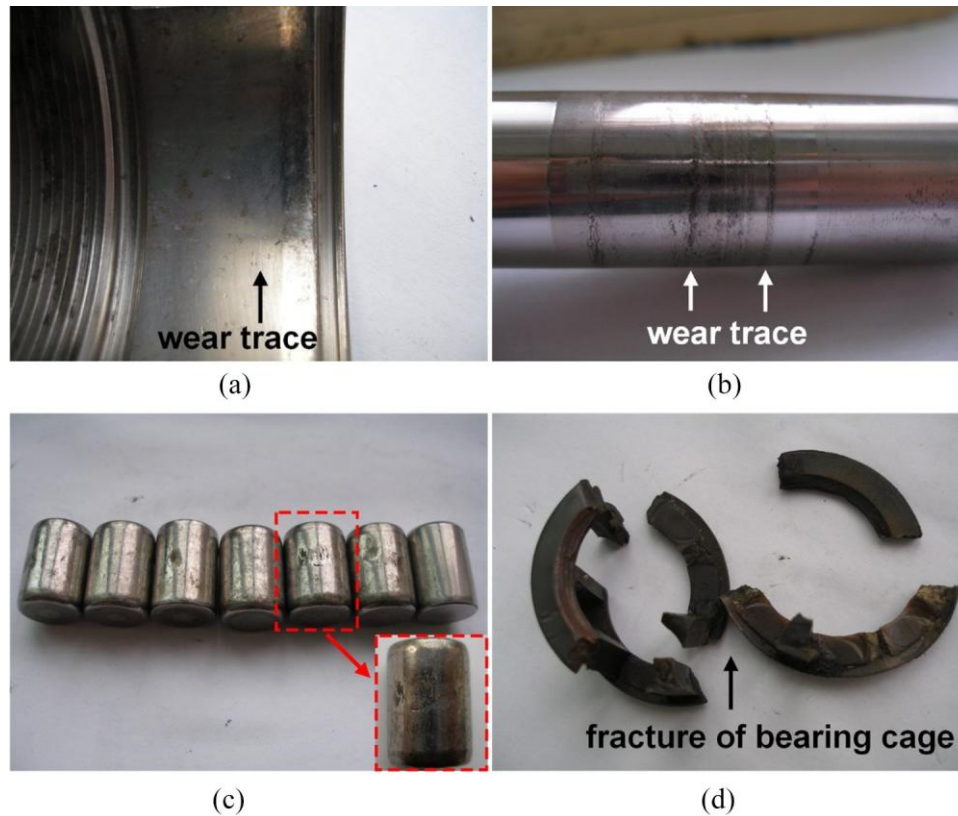


Figure 4. Disassembly Morphology of Cylindrical Roller Bearing: (a) Wear Trace on Outer Raceway Surface, (b) Wear Marks On Mandrel Surface, (c) Damage Morphology of Roller Surface, and (d) Fracture Morphology of Bearing Cage.

Table 1. Chemical Composition of Failed and Standard Mandrel and Roller.

Chemical compositions of samples (wt. %)										
	C	Si	Mn	Cr	Mo	P	S	Ni	Cu	O
Mandrel	0.983	0.225	0.279	1.43	0.0037	0.0123	0.0098	0.0116	0.0108	0.0009
Rollers	1.02	0.273	0.349	1.46	0.0134	0.0121	0.0002	0.0452	0.0538	0.0011
Standard	0.95–1.05	0.15–0.35	0.25–0.45	1.40–1.65	<0.10	<0.025	<0.020	<0.25	<0.25	<0.0012

Table 2. Surface Hardness of Failed Bearing Parts And Standard Requirements

Samples	Point 1 (HRC)	Point 2 (HRC)	Point 3 (HRC)	Deviation (HRC)
Severely worn roller	64.0	64.5	64.0	0.5
Roller without wear trace	64.0	64.3	64.0	0.3
Mandrel surface	63.7	64.0	63.8	0.3
Standard requirement	61–66 HRC of rollers, ϕ 62 HRC of mandrel			<1.0

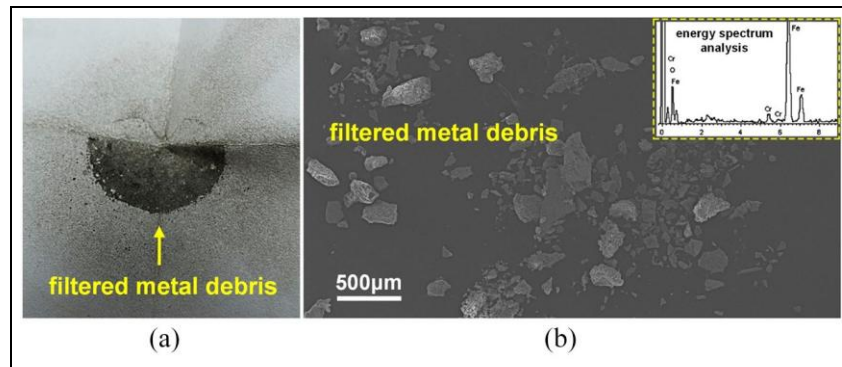


Figure 5. Morphology of Filtered Metal Debris (a) and SEM Images of the Debris and Corresponding Energy Spectrum Analysis (b).

The Morphology of the filtered metal debris are shown in Figure 5(a) and SEM images of the debris and corresponding energy spectrum analysis also shown in Figure (b) rubbed repeatedly between the contact surfaces and played a role of ploughing, resulting in the generation of furrow as shown in Figure 9(a) and (d). Due to the eccentric loading, the local stress at the surface increased. The high temperature generated due to the friction process led to the increased temperature of the contact surface and the formation of secondary quenching and tempering layers. Additionally, the abrasion resulted in the uneven surface and the stress concentration at the local contact points, which caused the surface peeling and micro-crack as depicted in Figures 9(b) and (e). As the bearings continued to roll, the micro-cracks propagated and produced the peeling off. Consequently, it was concluded that the bearing had surface origin fatigue failure after rolling.

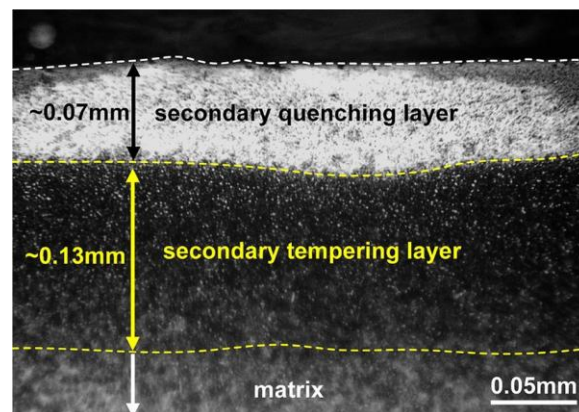


Figure 6. Surface Metallographic Structure of Worn Cylindrical Rolle

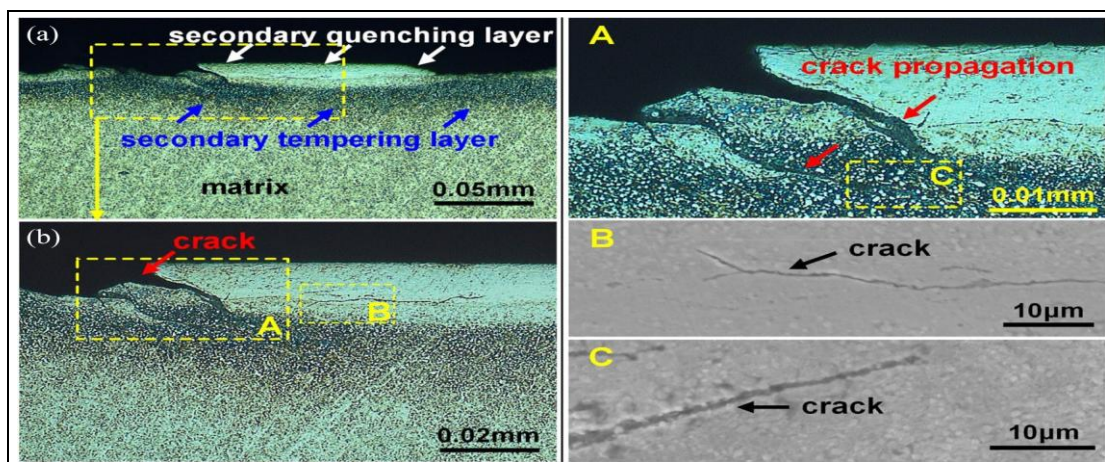


Figure 7. (a) Crack Morphology and (b) Corresponding Magnification in the Secondary Quenching and Tempering Layers of the Worn Mandrel. A, B, and C Represents the Local Magnification of the Dotted Yellow Regions

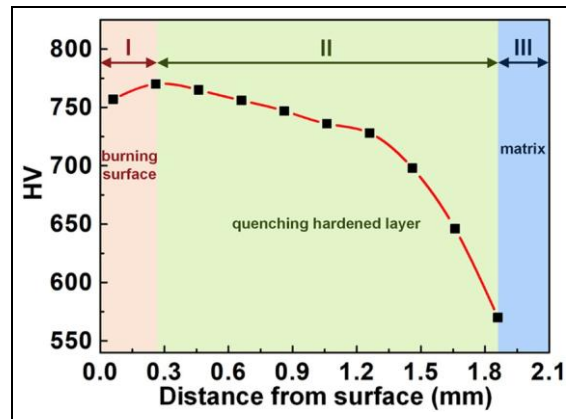


Figure 8. Gradient Curve of Micro Hardness Distribution of the Worn Mandrel.

4.2 Fatigue Failure and Prevention

Automobile pump bearing is characterized by one end of the belt as a power input, driving the other end of the blade rotation to achieve drainage purposes. From the above analysis, the cylindrical roller bearing at the power end had serious wear and a lot of surface cracks. And, the burn layer and superficial peeling emerged on the surfaces of cylindrical roller and mandrel. Especially, the wear marks of the mandrel are found along the circumferential direction and accounted for about two thirds of the ring track, indicating the abnormal operation of cylindrical roller.

During the fatigue failure of the bearing, some factors influencing bearing life were summarized and excluded. Firstly, no foreign body was found in the lubricating grease of cylindrical roller bearing, and the grease contaminated by foreign matters was invalid. Secondly, the hardness and quenching and tempering microstructures of the bearing mandrel and cylindrical roller complied with related standards, and the substandard heat treatment quality was ruled out. Although the local burn layer (secondary quenching layer and high temperature tempering layer) on the surface of the roller and the mandrel was not excluded from the insufficient lubrication, it would not lead to the breakage of the dynamic end cage.

Experimentally discussed above, the roller and mandrel were considered as two elastic cylinders with curved radius of R_1 and R_2 , and lied in contact with parallel axes, as shown in Figure 10. The maximum pressure (P_0) could be expressed by Equations 1, 2 and 3 from Hertzian theory (Hertz, 1882)

$$P_0 = E^* / 2 \left(d/R \right)^{1/2} = (E^* F / LR)^{1/2}$$

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$$

$$\frac{1}{E^*} = \frac{1 - \nu_1}{E_1} + \frac{1 - \nu_2}{E_2}$$

3

where, E^* is the moduli of elasticity, F is the normal force, L is the length of roller, E_1 and E_2 are the elastic modulus of roller and mandrel, ν_1 and ν_2 are the poisson's ratio of roller and mandrel.

During operation, the large radial force of the pulley at the dynamic end increased the bearing deflection, and the uncompleted contact decreased the length, further leading to the increased contact force. The abnormal local stress promoted the abnormal operation and serious burning, finally resulting in the fracture of bearing the cage, serious wear of the roller and the mandrel as well as the generations of vibration and abnormal noise.

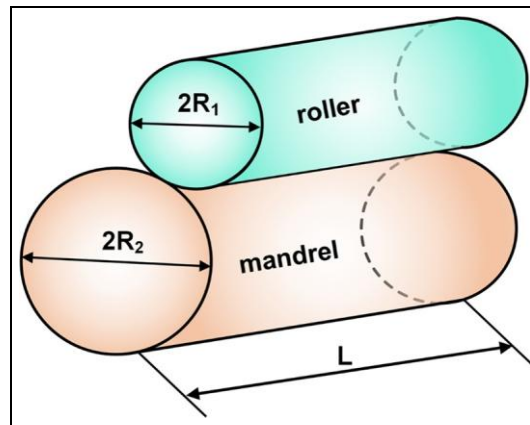


Figure 10. Schematic Diagram of the Paralleled Contact between Roller and Mandrel.

The radial load of pump shaft bearing mainly consisted of tension of driving belt and the weights of water pump impeller and shaft connector. Hence, the following recommendations and measures are proposed: reducing the weight of water pump impeller and shaft connector, increasing the bearing thickness to improve the radial bearing capacity. The bearing thickness represented the mandrel diameter. As the thickness increased, the mandrel diameter (R) in Equation (1) increased, resulting in the decreased contact force. On the other hand, the increased mandrel diameter promoted the reduced the bending deformation of the mandrel and prevented the reduction of length.

V. CONCLUSIONS

The fatigue damaged analysis of the engine shaft water pump bearing was conducted by using the chemical composition detection, hardness distribution testing, wear morphological characterization, and theoretical analysis methods. The wear behavior and cessational mechanism of the engine water pump shaft bearing was in-detail elucidated, and the subsequent prevention measures were further proposed. The main conclusions of the research are as follows:

The abnormal operation of the water pump shaft bearing was mainly attributed to the damage of the cylinder roller bearing on the power side, rather than the deep groove ball bearing on the impeller side. Excluding the composition, hardness, and microstructures complying with the standard requirements, the fracture of the bearing cage and the serious wear of the roller and mandrel mainly resulted in the bearing cessation.

The wear characteristics on the failed mandrel surface, including furrow, peeling off, and micro-cracks, confirmed the rolling bearing having surface origin fatigue cessation.

The abnormal large radial force from bearing deflection promoted the abnormal operation, the fracture of bearing cage, serious burning, and the wear of the bearing roller and mandrel, accompanied by the abnormal vibration and noise. In addition, the decreased weight of water pump impeller and shaft connector, and the increased bearing thickness were proposed to prevent the fatigue cessation.

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