

Article

Service-Life Analysis and Effect of Grease Lubrication on Wear of ASB 6302-2RS Ball Bearing in the AST-MPV Machine

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Abstract.

Bearings are critical elements for supporting shaft rotation in industrial equipment. In the AST-MPV detergent-packaging machine at PT. X, ASB 6302-2RS bearings frequently experience jamming because of detergent-dust contamination. This study analyzed bearing service life and evaluated, as an initial comparative observation, the effect of additional grease under detergent-water exposure. The analytical stage used rotational speed, load conversion, dynamic equivalent load, and nominal bearing-life calculation with standardized N and kN units. The experimental stage compared two specimens: bearing A with manually applied Black Grease NLGI 3 LA and bearing B without additional grease. Both specimens were immersed in a 2 kg detergent and 1 L water mixture for five days, rotated at 1000 rpm for 30 min and 1500 rpm for 30 min, and measured using caliper and dial indicator. The corrected spring deflection was 0.006 m, and the calculated service life was 3479.1223 h at 1930 rpm. Additional grease reduced radius wear from 0.52 mm to 0.13 mm and outer-diameter wear from 0.10 mm to 0.03 mm. These findings suggest that scheduled grease lubrication may reduce wear in detergent-contaminated environments, although replicated testing is required.

Keywords: Ball bearing; bearing life; grease lubrication; wear; detergent contamination

INTRODUCTION

Rotating machine components are always associated with friction, temperature rise, wear, and reduced operating efficiency. A bearing is a machine

element used to support a shaft and restrict relative motion between components so that motion remains in the intended direction. Bearing performance determines system stability because failure of this component can stop shaft rotation, increase vibration, and cause further damage to the transmission system [1]. Additional bearing-design references describe bearing life as a function of load, speed, reliability, and rolling-contact fatigue [2], [3].

In industrial environments, bearing failure is not only affected by service life but also by operating conditions, load, rotational speed, lubrication, contamination, and the surrounding environment. Dust, water, process residues, and abrasive particles can enter the contact area and accelerate wear. Therefore, service-life analysis and bearing-condition inspection are required so that a company can schedule replacement and maintenance before total failure occurs [4], [5]. Failure classification standards also emphasize contamination, inadequate lubrication, corrosion, and installation condition as common contributors to rolling-bearing damage [6], [7].

The AST-MPV machine at PT. X is used in detergent-packaging operations. In this machine, ball bearings frequently experience jamming and damage because of accumulated detergent dust. This condition may disrupt production and increase maintenance costs. In addition, detergent-water media around the working area can accelerate corrosion and wear, especially when the lubricant cannot form an adequate protective film on the bearing contact surface [8], [9]. Similar manufacturer troubleshooting guides identify detergent-like dust, moisture, and abrasive contaminants as factors that can disturb grease films and bearing sealing performance [10], [11], [12].

Several previous studies have discussed bearing service life, wear, and maintenance strategies. Anjaswara analyzed bearing failure on a scraper-conveyor loading-ramp system using ANSYS simulation [13]. Irwanda emphasized the importance of grease-type lubricant selection for machines operating at low speed and low temperature [14]. A hoist-simulator study showed a large difference between theoretical and actual bearing life in the field, indicating that direct observation remains necessary [15]. Kuntara et al. developed a bearing-life approach based on wear rate [16], whereas Utomo et al. used the deviation between calculation and actual condition as a basis for preventive maintenance planning [17]. Azis and Zakinura also showed that bearing-life analysis is important for maintaining the precision of rotary test equipment [18]. Studies on used and damaged bearings also show that physical and mechanical condition changes can be observed after service, while radial bearing failure studies support direct damage analysis as part of maintenance evaluation [19], [20].

International standards and manufacturer catalogues provide additional guidance for rolling-bearing rating life, equivalent dynamic load, failure classification, and maintenance decisions. ISO 281 describes dynamic load

ratings and rating life, while ISO 15243 classifies bearing damage and failure characteristics. Manufacturer catalogues and troubleshooting guides from SKF, NSK, NTN, and Timken are relevant for interpreting load ratings, lubrication practice, contamination, and bearing-damage mechanisms [3], [6], [10], [11], [12], [21].

Based on this background, this research focuses on the service-life analysis of ASB 6302-2RS ball bearings in the AST-MPV machine and an experimental test of the effect of grease application on bearing wear after immersion in detergent solution. The main contribution of the study is the combination of a field-data-based service-life calculation with a simple experimental comparison between a greased bearing and a non-greased bearing under detergent exposure.

The objectives of this study are to determine the service life of the ball bearing used in the AST-MPV machine, evaluate the effect of detergent immersion on dimensional and flatness changes, and identify the more suitable lubrication treatment for ASB 6302-2RS bearings in detergent-exposed operating environments.

METHODS

Research Design

This study used a quantitative approach by combining theoretical calculation and laboratory experiment. The theoretical calculation was performed to determine bearing service life based on load and rotational-speed data from the AST-MPV machine. The experiment was conducted to compare bearing wear and flatness after lubrication treatment and immersion in detergent solution. The research object was the ASB 6302-2RS deep-groove ball bearing used in the AST-MPV machine at PT. X. The calculation framework follows the general nominal-life approach used in bearing design and ISO-based rolling-bearing selection [2], [3], [12].

The independent variable was lubrication treatment, consisting of a bearing with additional grease and a bearing without additional grease, under detergent-solution exposure. The dependent variables included radius wear, outer-diameter wear, and flatness changes measured using a dial indicator. The control variables included bearing type, Black Grease NLGI 3 LA lubricant, detergent-water mixture, immersion time, test speed, and the rotary test device.

This experiment used one bearing specimen for each treatment condition. Therefore, the experimental results were analyzed as an initial comparative observation rather than as a statistically conclusive test. The initial specimen history was not fully recorded in the laboratory notes; however, the pre-test radius and outer diameter of both specimens were measured before immersion. In the interpretation of inner-bearing dial-indicator data, bearing A

was noted as having previously been used in preliminary rotary trials, which may have contributed to runout and localized wear.

The equipment used in this study included a tachometer to measure rotational speed, a hanging scale to estimate the load applied through spring extension, a caliper to measure radius and diameter, an INSIZE dial indicator model 2307-10 to measure relative flatness/runout, and a rotary test device modified from a washing-machine motor. The dial indicator had a measuring range of 10 mm and a resolution of 0.01 mm per division. The materials included ASB 6302-2RS ball bearings, Black Grease NLGI 3 LA, powdered detergent, and water. The use of dial-indicator-based runout or geometric deviation measurement is consistent with common engineering inspection practice for rotating components [22].

Table 1. Field and bearing-test data.

Parameter	Description/Value
Machine type at PT. X	AST-MPV machine / detergent-packaging machine
Testing machine	Washing-machine motor used as a rotary test device
Bearing type	ASB 6302-2RS
Motor speed at PT. X	1930 rpm
Motor speed on testing machine	1000-1500 rpm
Normal spring length on test device	7.2 cm
Normal spring length at PT. X	9.7 cm
Spring length under load on test device	10.0 cm
Spring length under load at PT. X	10.3 cm

Bearing-Life Calculation

Bearing-life calculation began by determining spring deflection, force received by the bearing, axial load, radial load, dynamic equivalent load, and nominal life. Spring deflection was calculated from the difference between the spring length under load and the initial spring length. Under AST-MPV operating conditions, the initial spring length was 0.097 m and the loaded spring length was 0.103 m; therefore, the corrected deflection was 0.006 m, not 0.06 m. Next, this is described through equation 1.

$$\Delta l = l_2 - l_1 \quad (1)$$

The force received by the bearing was calculated by multiplying the measured equivalent mass by gravitational acceleration. Based on the measurement, the equivalent mass received by the bearing was 2.8 kg, resulting in a force of 27.44 N. This value is force, not mass. Next, this is described through equation 2.

$$F = m \times g \quad (2)$$

The load-related parameters were standardized in N or kN. The technical bearing data used for calculation were $C = 9.95 \text{ kN}$ and $C_0 = 5.4 \text{ kN}$, corresponding to the SKF W 6302-2RS1 technical catalogue for a sealed 6302-series deep-groove ball bearing used as a traceable reference for the 6302-2RS specification [23]. If the actual ASB manufacturer catalogue differs from this value, the service-life calculation should be recalculated using the exact ASB data sheet. Additional catalog references were used to support the traceability of the 6302-series bearing rating data [10], [12], [21].

The axial load was calculated from C_0 using a factor of 0.056, producing $F_a = 0.3024 \text{ kN}$. The radial load recorded in the original calculation was 118.60 kgf and was converted to $F_r = 1.163 \text{ kN}$. The X and Y factors referred to the load-factor table for ball bearings, with $X = 0.56$ and $Y = 2.30$ [1]. The dynamic equivalent load was calculated using Equation (3), resulting in $P = 1.347 \text{ kN}$, which is equivalent to approximately 137.34 kgf in the previous calculation basis. This approach is consistent with standard use of radial and axial load factors in bearing equivalent-load estimation [1], [2], [3]. Next, this is described through equation 3.

$$P = XF_r + YF_a \quad (3)$$

The nominal bearing life was then calculated based on the basic dynamic load rating, dynamic equivalent load, rotational-speed factor, and motor speed. The general life equation for ball bearings uses exponent $p = 3$ because the rolling elements are balls [3]. Next, this is described through equation 4.

$$L_h = (10^6/60n) \times (C/P)^3 \quad (4)$$

Grease, Immersion, And Rotary-Test Procedure

The experiment used two bearing specimens. Bearing A was manually coated with additional Black Grease NLGI 3 LA before immersion, whereas bearing B received no additional grease. The grease was applied to the accessible bearing surface and seal-contact area as a practical protective layer before immersion. The exact grease mass was not weighed in the original laboratory record; therefore, the treatment should be interpreted as manual grease coating rather than a controlled grease-mass dosage. Grease lubrication was selected because it is commonly recommended for rolling bearings where sealing simplicity, contamination resistance, and relubrication practicality are required [10], [11], [12], [23], [24].

The immersion mixture was prepared using 2 kg of powdered detergent and 1 L of water, then mixed until the detergent was dispersed. The detergent was recorded as detergent used in the field/laboratory simulation, but its exact surfactant and additive formulation was not available. Both bearing specimens were immersed in the detergent-water mixture for five days. This limitation was added to improve transparency and reproducibility. Corrosion and tribological references note that water, oxygen, and dissolved chemical species can accelerate surface degradation when protective films are insufficient [24].

After immersion, both bearings were installed on the rotary test device and rotated at 1000 rpm for 30 min and 1500 rpm for 30 min. Before and after testing, the bearing radius and outer diameter were measured using a caliper to determine dimensional wear. Flatness/runout was measured using the dial indicator at eight angular sectors: 0°-45°, 45°-90°, 90°-135°, 135°-180°, 180°-225°, 225°-270°, 270°-360°, and 360°-0°.

For dial-indicator measurement, the probe was positioned on the tested bearing surface and the indicator was zeroed at the initial reference sector. The bearing was then rotated clockwise through each 45° sector, and the deviation from the zero reference was recorded. The values in Tables 3 and 4 are presented as dial-indicator divisions; one division corresponds to 0.01 mm for the instrument used. Positive and negative signs indicate movement above or below the zero reference point, respectively. Measurements were taken on the outer diameter and inner bearing surfaces to observe geometric changes after treatment.

The experimental data were analyzed descriptively by comparing wear values and dial-indicator readings between bearing A and bearing B. Because the study used one specimen per treatment, the analysis emphasizes relative trends and maintenance implications rather than statistical proof of grease effectiveness.

RESULT AND DISCUSSION

Bearing-Life Calculation Results

Based on the calculation at 1930 rpm, the corrected spring deflection in the AST-MPV machine was 0.006 m and the force received by the bearing was 27.44 N. With $C_0 = 5.4$ kN, $C = 9.95$ kN, axial load = 0.3024 kN, radial load = 1.163 kN, and factors $X = 0.56$ and $Y = 2.30$, the resulting dynamic equivalent load was approximately 1.347 kN.

This equivalent load was used to calculate the nominal bearing life. The calculation result showed that the bearing service life in the AST-MPV machine at PT. X was 3479.1223 h. This value represents the estimated operating time under the analyzed load and speed conditions. On the testing machine operating at 1000-1500 rpm, the recorded bearing life was 4556.58186 h. This difference indicates that a lower rotational speed tends to provide a longer operating life

when the workload remains within the bearing capacity. This relationship between speed, equivalent dynamic load, and rating life is also consistent with ISO 281 and rolling-bearing catalog procedures [3], [10], [12].

These results support the concept that bearing life is strongly affected by rotational speed and workload. The higher the speed and the greater the load, the higher the contact frequency between rolling elements and the raceway, which increases the potential for surface fatigue. Therefore, service-life estimation should be linked with inspection and relubrication schedules so that bearings are not operated beyond safe conditions [14], [17], [18].

Radius And Outer-Diameter Wear

Table 2 and Figure 1 show that the bearing without additional grease experienced greater dimensional wear than the bearing with additional grease. In the radius measurement, bearing A had a wear value of 0.13 mm, whereas bearing B had a wear value of 0.52 mm. Thus, the radius wear of the bearing without grease was about four times higher than that of the greased bearing. In the outer-diameter measurement, bearing A decreased by 0.03 mm, whereas bearing B decreased by 0.10 mm.

Table 2. Comparison of radius and outer-diameter wear of ASB 6302-2RS bearings.

Parameter	Bearing A - with grease (mm)	Bearing B - without grease (mm)
Initial radius	14.02	14.02
Final radius	13.89	13.50
Radius wear	0.13	0.52
Initial diameter	42.00	42.00
Final diameter	41.97	41.90
Outer-diameter wear	0.03	0.10

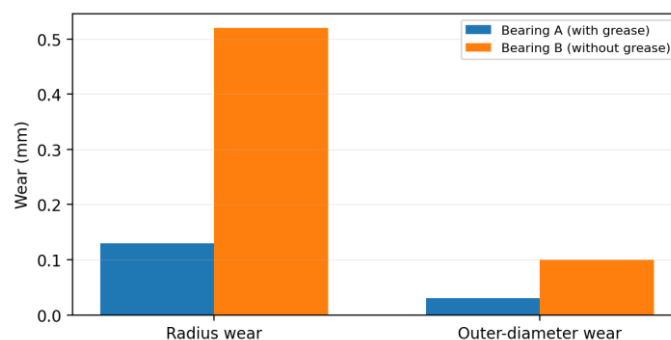


Figure 1. Wear comparison between bearings with and without additional grease.

This difference suggests that grease can form a separating layer in the contact area, thereby reducing direct contact between the bearing surface and detergent particles. In the bearing without additional grease, the surface was more exposed to detergent-water media and solid particles; therefore, friction

and abrasion tended to increase. However, because only one bearing was tested in each condition, this result should be understood as an initial comparative indication and should be validated using repeated specimens before it is generalized as a definitive effectiveness claim. This interpretation follows established tribological principles in which lubricant films reduce direct asperity contact, wear, heat generation, and contaminant ingress [7], [11], [23], [24].

Outer-Diameter Flatness Measurement Results

In the outer-diameter measurement, bearing A with additional grease produced readings of 0, +1, 0, 0, -4, 0, 0, and 0 divisions. Bearing B without additional grease produced readings of +1, +3, +1, +1, 0, +1, +3, and 0 divisions. Because the dial-indicator resolution was 0.01 mm per division, the largest recorded outer-diameter deviation magnitude was approximately 0.04 mm in bearing A and 0.03 mm in bearing B relative to the zero reference.

Table 3. Comparison of dial-indicator readings on the bearing outer diameter.

Angle	Bearing A (with grease)	Bearing B (without grease)	Difference
0-45°	0	+1	1
45-90°	+1	+3	2
90-135°	0	+1	1
135-180°	0	+1	1
180-225°	-4	0	4
225-270°	0	+1	1
270-360°	0	+3	3
360-0°	0	0	0

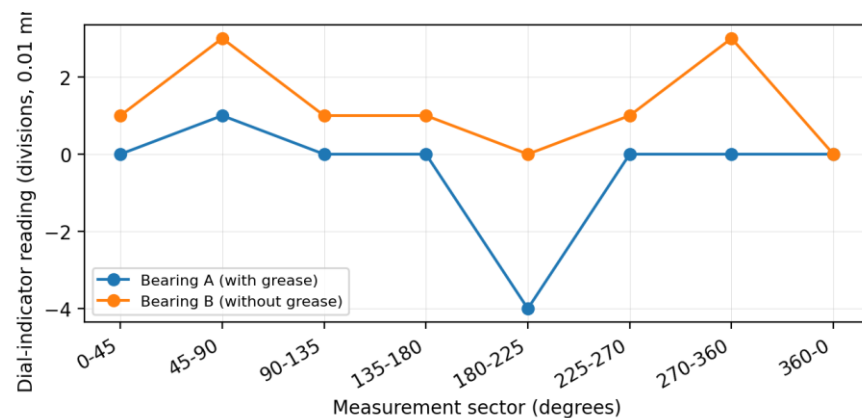


Figure 2. Dial-indicator readings on the bearing outer diameter.

In general, bearing B showed more frequent positive readings and tended to be less stable in several angular sectors, especially at 45°-90° and 270°-360°. The dial-indicator readings reveal geometric changes caused by the combination of immersion and rotary testing. The largest difference in outer-diameter measurement occurred in the 180°-225° sector, while another relatively high difference appeared in the 270°-360° sector. These results indicate that detergent

contamination and lubrication condition affected the stability of the outer bearing surface after rotary loading.

Inner Bearing Measurement Results

The inner bearing measurement showed a different pattern from the outer-diameter measurement. Bearing A produced readings of +6, +3, +2, +4, -5, -15, -13, and +5 divisions, whereas bearing B produced +3, +5, -5, -8, -1, +5, -2, and -2 divisions. With the 0.01 mm/division resolution, these values correspond to relative deviations from approximately +0.06 mm to -0.15 mm for bearing A and +0.05 mm to -0.08 mm for bearing B.

Table 4. Comparison of dial-indicator readings on the inner bearing.

Angle	Bearing A (with grease)	Bearing B (without grease)	Difference
0-45°	+6	+3	3
45-90°	+3	+5	2
90-135°	+2	-5	7
135-180°	+4	-8	12
180-225°	-5	-1	4
225-270°	-15	+5	20
270-360°	-13	-2	11
360-0°	+5	-2	7

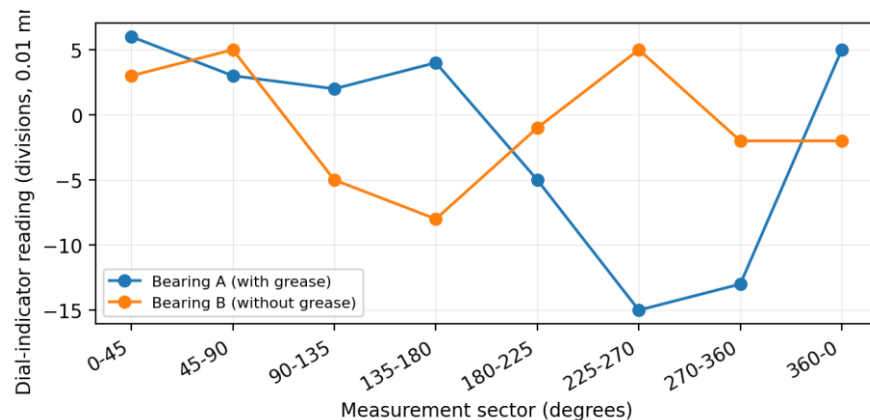


Figure 3. Dial-indicator readings on the inner bearing.

The inner bearing results should be interpreted carefully because the high variation in bearing A may have been influenced by runout and the condition of the bearing side that had previously been used for trial testing, causing part of the surface to be worn. Therefore, the inner-bearing result should not be used as the sole basis for concluding grease effectiveness. Nevertheless, the data still show that internal geometry inspection is necessary because damage is not always visible from the outer diameter alone.

Overall, the radius and outer-diameter wear data indicate the potential benefit of grease application, while the dial-indicator data highlight the

importance of periodic inspection of both outer and inner bearing surfaces. Grease lubrication is suitable for bearings operating at relatively non-extreme speeds and temperatures because it is easy to apply, helps prevent corrosion, and reduces direct contact between metal surfaces and contaminants [8], [9], [14], [23], [24].

Maintenance Implications

The practical implication of this study is the need to establish a lubrication and bearing-inspection schedule for the AST-MPV machine. With an estimated service life of 3479.1223 h at 1930 rpm, the company can set an inspection interval before this value is reached. Inspection should not rely only on noise or jamming but should also include temperature measurement, vibration observation, grease condition evaluation, and signs of contamination around the bearing area.

In a working environment containing detergent dust, bearing seals and cleanliness around the bearing area should be carefully maintained. If possible, adding a relubrication path or grease nipple can simplify periodic maintenance. Lubricant selection should consider temperature, rotational speed, load, and potential contaminant exposure so that the lubricant does not degrade quickly and can continue to protect the bearing contact surface. Manufacturer maintenance guides also recommend controlling contamination, maintaining seals, and using appropriate relubrication intervals to reduce premature bearing damage [10], [11], [12].

The calculation and experimental interpretation also have limitations. First, the ASB-specific manufacturer catalogue was not attached in the original experimental record, so a traceable 6302-2RS1 catalogue value was used for the calculation reference. Second, the amount of grease and detergent formulation were not quantified. Third, only one bearing was used per treatment. These limitations mean that future research should use replicated specimens, measured grease mass, controlled detergent chemistry, and manufacturer-specific ASB bearing data. Therefore, future work should follow replicated experimental design, controlled lubricant quantity, and traceable catalogue or standard-based bearing data [3], [12], [21], [25].

CONCLUSION

The corrected spring deflection of the ASB 6302-2RS ball bearing system in the AST-MPV machine was 0.006 m. Using standardized load units, the calculated service life of the bearing at PT. X was 3479.1223 h at 1930 rpm, whereas the bearing life on the testing machine operating at 1000–1500 rpm was 4556.58186 h. Additional grease application reduced radius wear from 0.52 mm to 0.13 mm and outer-diameter wear from 0.10 mm to 0.03 mm after detergent immersion and rotary testing. However, because only one specimen was tested for each condition, this finding should be considered an initial comparative

observation. The results indicate that rotational speed, workload, lubrication condition, and working environment are the main factors affecting bearing durability, with higher speed and load potentially shortening bearing service life. Therefore, scheduled grease lubrication is recommended for ASB 6302-2RS bearings in the AST-MPV machine because it can reduce friction, inhibit wear, and help protect bearing surfaces from detergent contamination. Further replicated testing is required to confirm the magnitude of the grease effect.

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