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The body can be viewed from the next page.

Technology to Prolong the Life of Rolling Bearings Used in Steelmaking Plants*


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There are a great number of rotary machines, of which typical examples are guide rolls at continuous casting machine and process rolls at rolling mill in steelmaking plants. Bearing damage reflects directly on the downtime of equipment, and is also one of the most significant issues of mechanical parts for plant management. Though loading conditions and surrounding environment upon each bearing variously depend on each process line, so that several patterns of damages can be discovered among those bearings. To reduce equipment failure, analyses on the damages of rolling bearings occurred in the steelmaking plants and countermeasures for its causes are carried out. Several typical technologies to prolong spherical roller bearing life and to improve other types of rolling bearing are described.

1 Introduction

A large number of various mechanical elements including bearings are preinstalled in steelmaking plants. Because mechanical elements are used in drive equipment, trouble in a single mechanical element stops the whole production line.

Figure 1 shows the actual condition of trouble that occurred in fiscal 1998 in the equipment at Mizushima Works, which is under the control of the Equipment Maintenance Department. In the figure, trouble was rearranged for each mechanical element in terms of the trouble occurrence rate and the downtime rate. Bearings were the 5th worst cause of trouble in terms of the trouble occurrence rate and the worst in terms of the downtime rate. **Figure 2** shows bearing trouble in terms of the trouble occurrence rate, which were classified according to breakdown factors and types of bearing. In trouble other than those to be prevented by equipment management, breakdowns due to axial loads account for about 1/4 and are followed by other breakdowns in the order: tempering and softening of the bearing proper by heat, oxidation and solidification of grease, skew of rollers (bearing behavior), overloads due to vibration impact shock, mechanical constraints in the absorbent part of

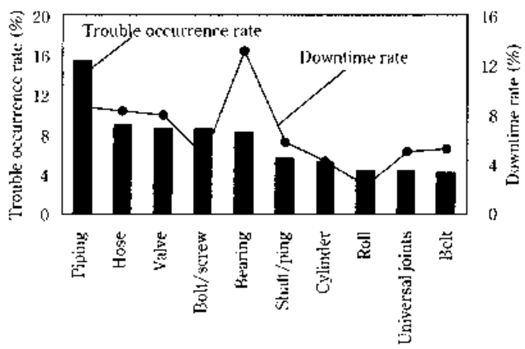

Fig. 1 Actual trouble occurrence classified by mechanical parts (from 1998/4 to 1999/3)

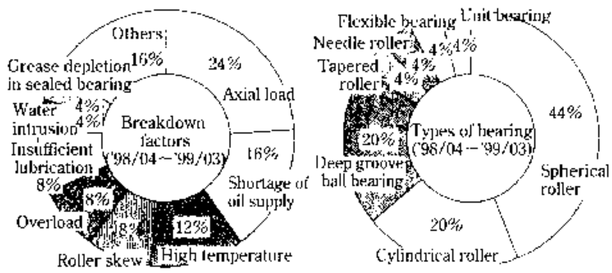

Fig. 2 Classification of bearing trouble

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shaft expansion and contraction, etc., poor lubrication due to the entry of debris and water, and grease depletion in sealed bearings. In terms of the types of bearing, spherical roller bearings account for more than 40% and cylindrical roller bearings and deep-groove ball bearings each account for 20%.

The thickness of the oil film is small in rolling bearings for roll-neck bearings compared with plain bearings and therefore, rolling bearings used as roll-neck bearings are susceptible to the effects of debris and water, and, seizure trouble frequently also occurs due to high speed revolution.

This report focuses on the static behavior and dynamic behavior of spherical roller bearings in an environment in which axial loads work, and describes the development of a new type of bearing that compensates for the weak points of spherical roller bearings and a purified lubrication technique for eliminating damage caused by external factors.

2 Problems in Spherical Roller Bearings and Development of New-Type Bearing

2.1 Analysis of Problems in Spherical Roller Bearings and Standard for Application

Two representative types of spherical roller bearings, i.e., the symmetric roller type and the asymmetric roller type are shown in **Fig. 3**. In recent years, the symmetric roller type has become the mainstream for the purposes of increasing the load carrying capacity in response to increased roller diameters and lengths, and preventing the edge loads under the action of axial loads.

Under the action of axial loads, the asymmetric roller type differs from the symmetric type in that the posture of the rollers in restrained by the flange of the inner ring, with the result that edge loads are generated mainly between the outer ring and the edges of the rollers. For example, **Photo 1** shows damage to an asymmetric roller type bearing in which flaking occurred only in one row and abnormal wear occurred thereafter. The damage is concentrated on the positions corresponding to the roller edges. Therefore, the difference in characteristics between the symmetric roller type and the asymmetric roller type was evaluated by the distribution of contact stress. The analysis was made by performing an approximate calculation by a two-dimensional FEM method under the conditions of radial and axial loads.

Figure 4 shows the observation points of contact stress variation in the inner and outer rings that are in contact with each corner of the rollers during the analysis. **Figures 5** and **6** show changes in the Von Mises stress at each observation point as a function of the ratio of axial load. From Fig. 5, it is apparent that in the symmetric roller type, the contact stress increases in proportional to the ratio of axial load and that this tendency is independent of the magnitude of the radial load. This means that, in the symmetric type, the rollers and the raceways of the inner and outer rings maintain face con-

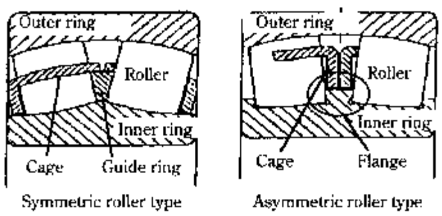

Fig. 3 Prototype of spherical roller bearing

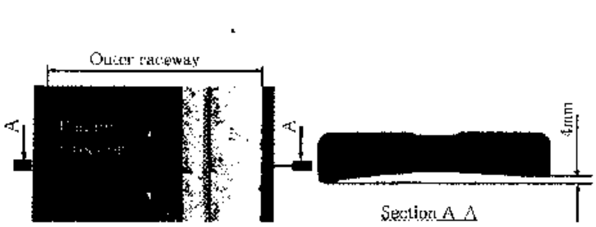

Photo 1 Example of damaged asymmetric roller type bearing under axial load working

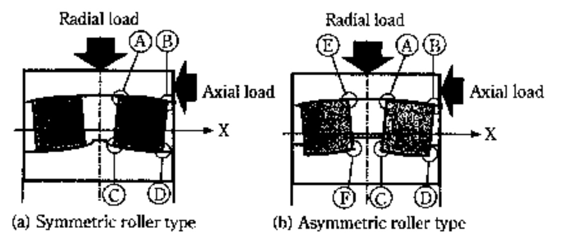

Fig. 4 Observation point of contact stress variation

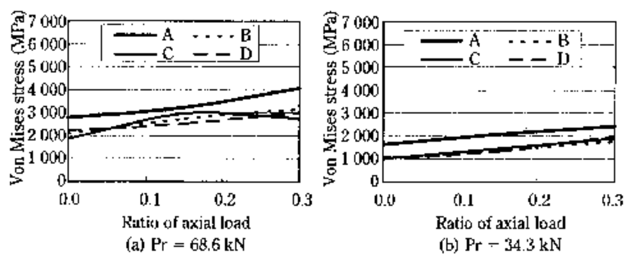

Fig. 5 Contact stress of symmetric roller type

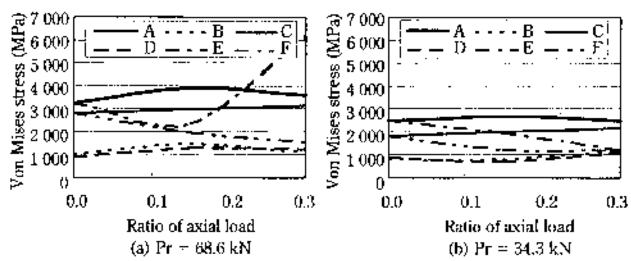

Fig. 6 Contact stress of asymmetric roller type

Table 1 Rules on the selection about spherical roller bearing

		Magnitude of radial load (Cor/Pr) ≦10	Magnitude of radial load (Cor/Pr) >10
Ratio of axial load (Pa/Pr)	<0.3	Symmetric roller	Symmetric roller
	≧0.3	Symmetric roller	Asymmetric roller

Pr: Radial load Pa: Axial load Cor: Basic static load rating

tact independent of the ratio of axial load. Conversely, as is apparent from Fig. 6(b), in the asymmetric roller type under small radial loads, the stress change at each corner is small because the contact region expands toward the center due to the wedge effect and stresses become uniform. However, when the radial load is large (i.e., the absolute value of the axial load is large), as shown in Fig. 6(a), edge loads are locally generated at point E on the outer ring side due to the restraining of the roller posture in the flange. This means that a uniform contact stress distribution cannot be maintained when the absolute value of the axial load is large. Furthermore, the roller diameter of the asymmetric roller type is small compared with that of the symmetric type and, for this reason, the contact stress is high even in a case of radial load alone. Thus, it is considered that the asymmetric type is favorable only in the case of a low radial load and a high axial load ratio.

In view of the foregoing, a standard for the selection between the symmetric roller type and the asymmetric roller type in an environment of axial loads was set, as shown in **Table 1**, and is application has already begun.

2.2 Stabilization of Behavior of Spherical Roller Bearings

When a nonuniform contact stress distribution with respect to the center of contact occurs in a spherical roller bearing, the balance of the frictional moment working between the rollers and the outer ring and between the rollers and the inner ring is lost and the skew phenomenon occurs. When skew begins, supplementary moment due to slip in contact portions and due to contact with the cage and flange works, and a balance is recovered in that state and stabilizes (**Fig.** 7(a)). Furthermore, even when the contact stress distribution is uniform, a relative slip (differential slip) state is always present, except at two constant-velocity points, due to the difference in the peripheral velocity between the rollers and the raceways of the inner and outer rings (Fig. 7(b)). These behavior problems cannot be avoided due to the structural characteristics of spherical roller bearings. However, the driving force of the skew moment is a frictional force on a contact face in the presence of an oil film and therefore does not cause actual damage as long as the frictional force is small. In addition, the effect of differential slip manifests itself as an amount of wear only when the formation of the oil film is insufficient.

The examples of damage shown in **Photos 2** and **3** are considered to have been caused by structural defects of the spherical roller bearings. Normal and reverse operation at 0 → 150 → 0 rpm is repeated in cycles of approximately 10 s in the former example, and the bearing rotates continuously at extremely low speeds of 1–2 rpm in the latter example. The common point of these two cases is that formation of the oil film is insufficient with both bearings.

Therefore, the effect of the oil film thickness on the behavior of rollers was experimentally reproduced. The results of this experiment are shown in **Fig. 8**. Figure 8(a) shows the bus line of the outer ring of spherical roller bearing #23022 after use for 350 h in an extreme boundary lubrication region in which only an oil film

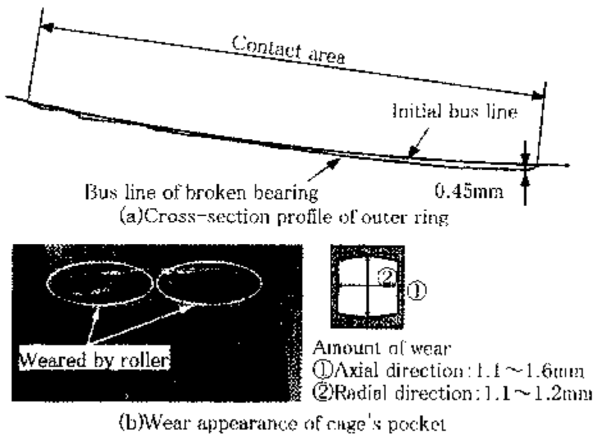

Photo 2 Example of damaged bearing by skew skidding

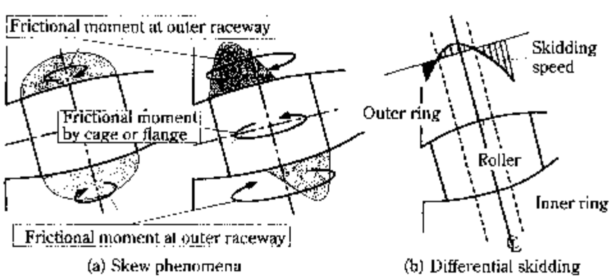

Fig. 7 Roller behavior of spherical roller bearing

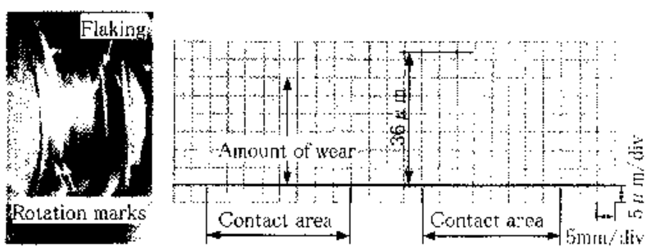

Photo 3 Example of damaged bearing by skew skidding

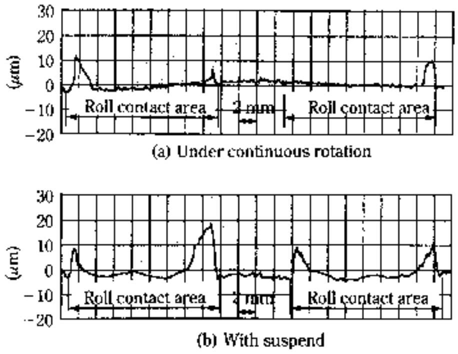

Fig. 8 Wear profile under typical boundary lubrication

thickness of approximately 0.01 μm can be expected under the conditions of Co/P = 2, speed of revolution of 10 rpm, and bearing box temperature of 100°C. In this case, only wear of the roller edges can be observed. However, when a condition of no oil film is forcedly caused at high frequencies by suspending the operation of a spherical roller bearing operating under the same operating conditions, a remarkable tendency toward wear of portions other than the two constant-velocity points manifests itself, as shown in Fig. 8(b). Furthermore, the same phenomena applies to skew, and it is thought that this phenomenon is remarkable in a case where the frictional force in portions of contact between the rollers and the raceway face and flange is large.

In view of the foregoing, a tapered roller bearing with spherical seat and a complete rolling bearing for both radial and axial loads were developed as bearings that avoid the structural problems peculiar to spherical roller bearings used when an oil film is not sufficiently formed.

2.2.1 Development of tapered roller bearing with spherical seat

In order to avoid skew moment and differential slip structurally, it is vital to change the roller type to a complete line contact type, such as cylindrical rollers and tapered rollers. Particularly when bearings are used on the fixed side, where the position of rolls which are subject to axial loads is determined, the tapered roller type is advantageous because axial loads can be supported not only by sliding contact portions between the rollers and the flange, but also by the face contact between the raceway and the roller. Furthermore, in order to realize a self-alignment mechanism, it is necessary that a spherical seat be provided outside the bearing. In view of the foregoing, the double-row tapered roller bearing[1,2] shown in **Fig. 9**(a) was developed.

Figure 9(b) shows the bus line of the outer ring of the newly developed bearing, which was used in the same location for the same period as the spherical roller bearing shown in Photo 3. A substantial reduction in the

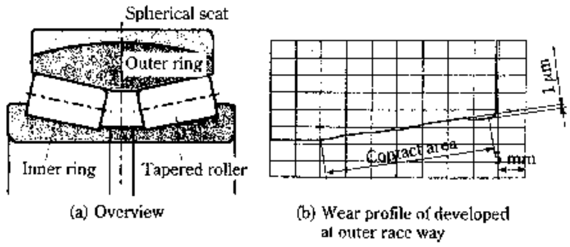

Fig. 9 Double raw tapered roller bearing with spherical seat

amount of wear, and uniform wear could be achieved. When wear is uniform, the problems of nonuniform contact stress distribution and slip, which provide an initiation point for skew, can be avoided and, therefore, the same effect against skew can be expected.

2.2.2 Development of complete rolling bearing for both radial and axial loads

Because a cylindrical roller bearing has no contact angle, both radial and axial loads can be supported in separate portions and complex load of the two loads does not work on the contact face. Thus, this type of bearing has an advantage in terms of life. Furthermore, with the exception of the problem of edge load, it is easy to obtain a uniform contact stress distribution along the full length of the rollers, and this is also favorable for reducing contact stresses under the same rolling element load.

Conversely, because a cylindrical roller bearing has no fixed orbital axis, the skew phenomenon occurs easily when imbalance of frictional moment occurs. Particularly, because axial loads are supported by the sliding contact portions between the roller end faces and the flanges, this is unfavorable for lubrication. Imbalance of frictional moment tends to occur in these portions and roller skew promotes the generation of edge loads between the outer ring and the rollers and causes breakage of the flanges.

The complete rolling bearing for both radial and axial loads[3] shown in **Fig. 10** was developed as a bearing which makes the most of the advantages of the cylindrical roller bearing while overcoming its disadvantages.

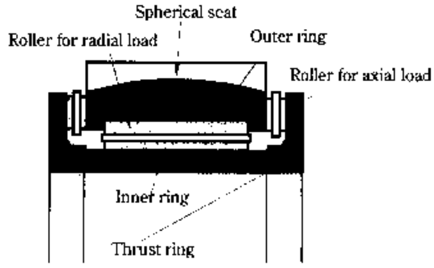

Fig. 10 Complete rolling bearing for both radial and axial load

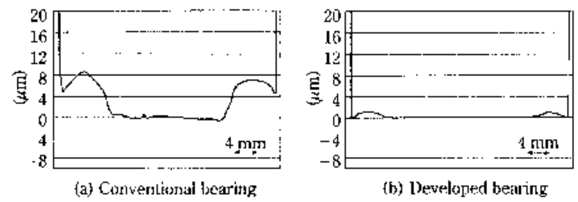

Fig. 11 Wear profile at outer raceway

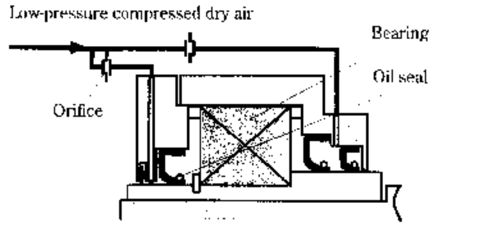

Fig. 12 Diagram of grease-air lubrication system[4)]

Figure 11 shows the bus lines of the outer rings of a conventional cylindrical roller bearing with a spherical seat and the newly developed bearing after use for the same period. Edge wear could be greatly reduced. Because there is no difference in the crowning profile between the two bearings, this decrease in wear is indicated to be due to a decrease in skew.

Table 2 Result of grease analysis

	Water content (wt%)	Debris content (wt%)
Conventional	1.8~9.6	0.44~1.98
Grease air system	<0.01	0.07~0.13

3 Life Prolongation by Elimination of External Factors and Optimization of Oils and Greases

3.1 Purified Lubrication Technique for Bearings at Ultra-low Speed Revolutions

3.1.1 Development of grease-air lubrication system

Under the conditions of ultra-low speed revolutions and high load, the contact portions between the rollers and the inner and outer rings are in a boundary lubrication state that produces metal contact. In the boundary lubrication region, life is substantially shorter than fluid lubrication in which an oil film is sufficiently formed. In commercial production equipment such as continuous casting machines, damage to bearings is further accelerated by the entry of water and debris into the bearings and the wear of the raceway faces. Therefore, a technique for prolonging bearing life by improvement of seals was developed.

As shown in Photo 3 and Fig. 8, when the formation of an oil film is insufficient in a spherical roller bearing, nonuniform wear occurs in the raceway of the outer ring, and flaking occurs from a convex part where wear is slight as the point of initiation. Particularly, the amount of wear is increased by water entering the grease and stress concentration on the convex part also becomes more pronounced. In commercial production equipment, there are cases where the grease contains approximately 1.8 to 9.6% water. In spherical roller bearings, it is necessary to minimize the amount of water entering the grease.

On the other hand, flaking sometimes occurs in cylindrical roller bearings near a large indentation formed on the raceway. Although indentations occur due to both wear particles from the bearing itself and debris from outside the bearing, and the large diameter debris in the latter case cause early flaking. Therefore, in order to prolong the life of cylindrical roller bearings, suppressing the entry of debris from the outside the bearing is important.

In view of the foregoing, in order to prevent the entry of water and debris into bearings, the company developed a grease-air lubrication system[4)] in which a slight positive pressure is maintained by supplying dry air to the gap between seals arranged in a double layer (**Fig. 12**). This system performs seal cooling through the use of air ejected from between the seals and prevents the entry of water and debris from outside by sucking dry air when a negative pressure is generated in the bearing. As shown in **Table 2**, the water and debris contents of the grease used in greasing by the grease-air lubrication system decrease substantially. This demonstrates that the grease air system is very effective in preventing the entry of water and debris from outside.

3.1.2 Application of face-contact seal

Attempts have been made to date to improve the sealing performance of oil seals by adopting a multi-seal design and changing the materials of the seals. However, the fastening quality deteriorated due to the degradation of the seal rubber proper, such as heat, wear and permanent set, and substantial improvement of the sealing performance could not be achieved. Therefore, the application of a face-contact seal made of PTFE (polytetrafluoroetylene) shown in **Fig. 13** was tested. Because PTFE can be used at a temperature (220°C) higher than the thermostable temperature of fluorine rubber, which has the highest temperature resistance among rubbers, it

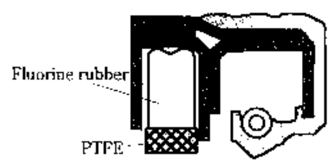

Fig. 13 Face contact PTFE seal

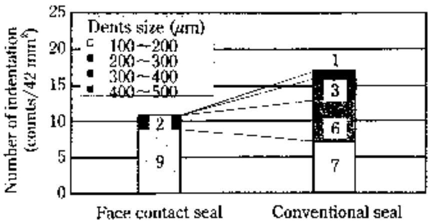

Fig. 14 Comparison of dents size on outer raceway

does not undergo heat hardening or permanent set in fatigue. It is possible to improve substantially resistance to entry of debris from outside for the long term by using PTFE in the sliding surface of a seal and, at the same time, by adopting a face-contact design.

Figure 14 shows the number of indentations formed on the raceway of the outer ring in bearings for the guide rolls of a continuous casting machine. In the face-contact seal, the number of indentations with a size of more than 200 μm decreased substantially. Thus, it has become possible to reduce damage to raceways due to debris in terms of both the size and number. Furthermore, a decrease in the half-value width was measured with an X-ray diffractometer, and from the results, it was ascertained that it is possible to prolong life to about 1.7 times that of the conventional seal.

3.2 Lubrication and Purifying Techniques for High-Speed Revolutions

In rolling mills, it is an exceedingly important task to ensure that compatibility is maintained between high precision and high-speed rolling. In the company's tin-plate mills, gauge accuracy was first improved by changing the plain bearings of back-up rolls to roller bearings. To achieve further increases in speed, it was necessary to improve reliability by upgrading the environment in which the bearings were used. For this purpose, monitoring of equipment conditions was enhanced and a lubricating oil purifying system was established.

Figure 15 shows a flow chart of a lubrication system for the bearings of backup rolls[5]. In the system, a flow indicator is installed for each bearing in order to ensure the amount of oil supply, and the temperature of the returning lubricating oil is also measured for each bearing for online monitoring of abnormalities such as heat generation.

Because large amounts of rolling mill oil and cooling water splash on the bearing boxes of backup rolls, there is concern that water and debris may enter the lubrication system. In order to improve the reliability of bearings, debris removing filters and water removing filters were installed and the lubricating oil was purified. The cleanliness of the lubricating oil after the application of the filters is shown in **Fig. 16**[6]. Debris contents of less than 20 ppm and water contents of less than 500 ppm were achieved.

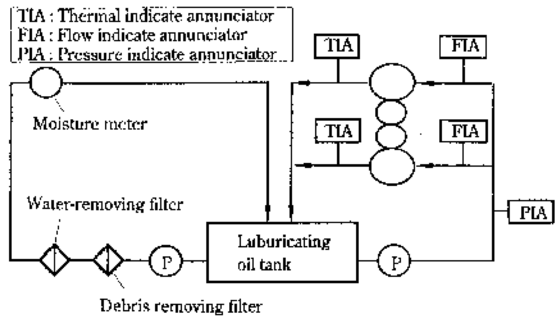

Fig. 15 Lubrication system for bearings of backup rolls[5]

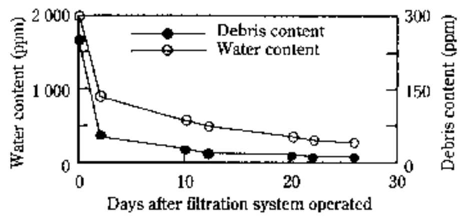

Fig. 16 Transition of lubricant purity[6]

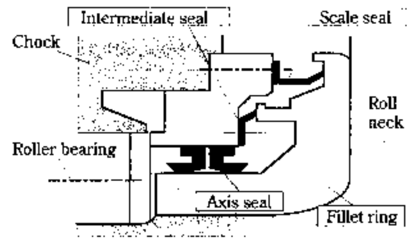

Fig. 17 Structure of seals in roller bearing of backup roll

The wear resistance of seals becomes important as the speed of the backup rolls increases. Therefore, the conventional structure composed of a shaft seal alone was reviewed, and a three-stage seal structure composed of a scale seal, an intermediate seal, and a shaft seal, as shown in **Fig. 17**, was developed[7]. In addition, fluorine rubber was adopted as the material for the seals and the shaft-side fillet ring that slides with the all seals was plated with hard chromium to provide wear resistance[7].

In addition to the above lubrication oil purification technique and seal technique for backup-roll bearings, a high-PV grease was developed for work rolls[5] and oil-air lubrication auxiliary rolls were carried out with the result that the rolling speed could be increased to the world's highest speed of 2 800 m/min from the conventional speed of 2 260 m/min[8].

4 Conclusion

Various damage factors are related to bearings. The company has preferentially made improvements in the factors in which the improvements were expected to

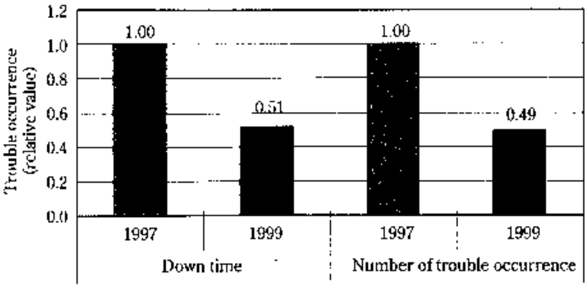

Fig. 18 Performance of bearing stabilization and life prolongation activity

have a great effect in the stabilization and prolongation of bearing life. As a result, trouble related to bearings in fiscal 1999 decreased to about half that in fiscal 1997, before the start of improvement activities, in terms of both downtime and the trouble occurrence rate (**Fig. 18**). In main tinplate mills, compatibility between high precision and high-speed rolling was realized, and operation has been stable, without trouble such as flaking, although more than seven years have passed since the adoption of roller bearings.

The authors would like to thank those concerned at the three bearing manufacturers, NTN Corp., Koyo Seiko Co., Ltd. and NSK Ltd. for their collaboration in the development of the bearings described in this report and the collection of experimental data.

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