Development of Low Temperature Rise Technology for Oil Bath Tapered Roller Bearings Adopted in High-speed Train Axles

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Manufacturers of axle bearings adopted in high-speed trains face the issue of increasing temperature rise due to faster travel speeds. In particular, oil bath lubrication generates a large amount of heat due to oil agitation. In response to this issue, we have developed additional parts which decrease temperature rise by 10% through controlling oil flow in an axle box. This paper introduces low temperature rise technology.

Key Words: *high-speed train, oil bath, low temperature, tapered roller bearing, oil agitation, Shinkansen*

1. Introduction

Since the inauguration of the Tokaido Shinkansen in 1964, the Shinkansen ("bullet train") has become an indispensable part of Japan's transportation infrastructure and has increased its travel speed, and by doing so, has contributed to economic growth. The maximum speed of the Tokaido Shinkansen at the time of its inauguration was 210 km/h, but as of 2021, the speed has been increased to 285 km/h. Also, the Tohoku Shinkansen is currently being developed for achieving an even higher speed of 360 km/h when it is extended to Sapporo (**Fig. 1**). As the speed increases, the temperature rise (bearing temperature minus environmental temperature) of the axle bearing (**Fig. 2**) also increases, and the key issue is how to minimize the generated heat.

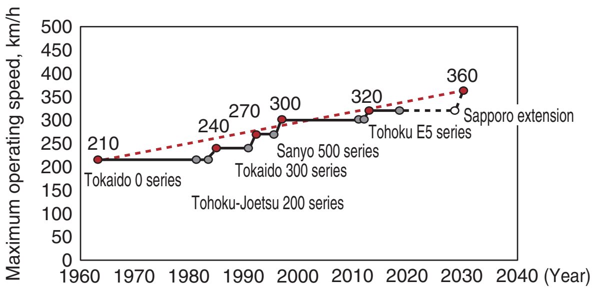

Fig. 1 Transition of maximum travel speed

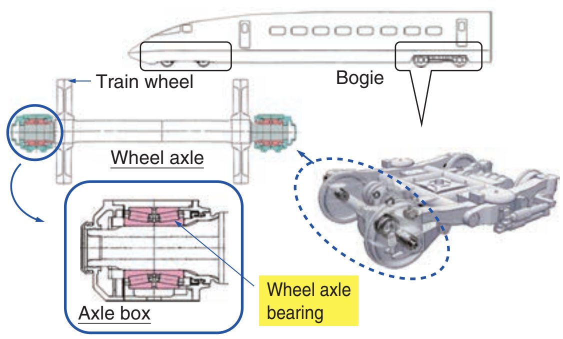

Fig. 2 Axle bearing mounted position

There are three main types of specifications for the Shinkansen axle bearings (**Fig. 3**). Among them, the oil bath lubrication system is a highly reliable system that has been used since the start of the Shinkansen bullet train service because the state of the bearings and oil inside the housing can be checked without disassembly from an oil window and magnetic plugs. However, one disadvantage is that the temperature is higher than the grease lubrication system. We have worked on the development of an oil bath lubricated tapered roller bearing to reduce heat generation, and we have succeeded in reducing the temperature rise of the outer ring by 6℃ (10%) compared to the current product. This paper presents this temperature rise reduction technology.

	Development target bearing		
	Type A	Type B	Type C
Bearing model	Double-row cylindrical roller bearing	Double-row tapered roller bearing	Double-row tapered roller bearing
Lubri-cation	Oil bath lubrication	Oil bath lubrication	Grease lubrication
Struc-ture			
Advan-tages	• Oil window and magnetic plugs for monitoring the bearing state • High-reliability lubrication system		• Low heat generation • Simplified axle box structure
	• Easy disassembly and inspection	• High running stability	
Disadvan-tages	• Higher temperature than grease lubrication		• Check of bearing state requires disassembly

Fig. 3 Axle bearing specifications

2. Development Objectives

Oil bath lubricated tapered roller bearings for Shinkansen axles consist of double rows and are surrounded by a housing. The housing has an indentation called an oil pool, which is filled with lubricant up to 80mm below the center of the axle. When stopped, the bottommost roller is half submerged in lubricant, and when running, it rotates while agitating this lubricant. During the test, the temperature rise was measured and monitored by thermocouples mounted to the top of the outer ring outside diameter surface and thermocouples mounted in the lubricant in the bottom of the housing (**Fig. 4**).

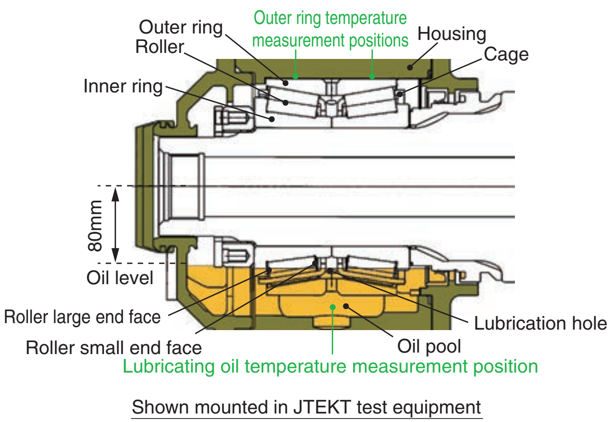

Fig. 4 Oil bath tapered roller bearing

Past in-house test results have shown that the speed dependence for temperature rise is significantly different between the oil bath and grease lubrication systems, with a difference of about 30℃ in temperature rise at 400 km/h (**Fig. 5**).

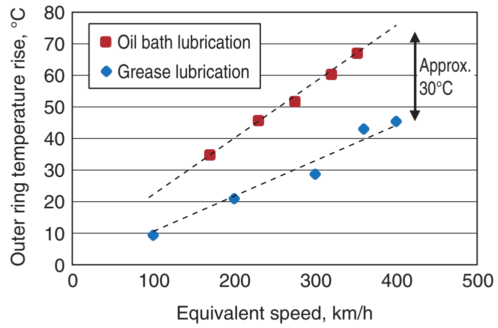

Fig. 5 Comparison of lubrication methods (tapered roller bearing)

The factors in heat generation for tapered roller bearings include the rolling viscous resistance, agitation resistance of the lubricant, and sliding resistance between the cone back face rib and the roller large end face. In oil bath lubricated bearings, the agitation resistance of the lubricant is considered to be a major factor. In the past, we developed an LFT® (Low Friction Torque) bearing[1] used in the pinion shafts of automobile differentials by devising a cage structure to reduce the agitation resistance of lubricant. This enabled us to reduce the torque (reduce the agitation resistance) by up to 50% or more compared to conventional products. This technology was used as a reference in this development for lowering the heat generation at high-speed running by reducing the agitation resistance of the lubricant with only structural changes around the bearing without changing the housing structure or lubricant level.

3. Estimating Lubricant Behavior under High-speed Running Conditions

3. 1 Procedure for Estimating Lubricant Behavior (under High-speed Running Conditions)

To reduce the agitation resistance of the lubricant, we must know the state of the lubricant inside the housing during rotation, and so we considered observation using a transparent acrylic case. However, due to concerns about the durability of the transparent acrylic case, we determined that it would be difficult to conduct and observe engine bench tests under high-speed running conditions equivalent to those of an actual train, and so we decided to use CAE analysis technology. The first step in this process is to visualize the system using an engine bench test that uses a transparent acrylic case and a high-speed camera under low-speed running conditions (rotation speed: 500 min^{-1}). CAE fluid analysis is also performed under low-speed running conditions in the same way, and fitting is performed. After confirming that the lubricant flow could be reproduced, CAE fluid analysis was conducted under high-speed running conditions (rotation speed: 2 418 min^{-1}, equivalent to train speed of 360 km/h) to estimate the agitation state (**Fig. 6**).

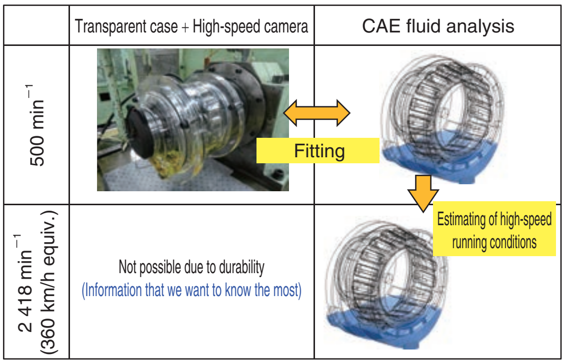

Fig. 6 Procedure for estimating lubricating oil behavior (high-speed travel)

3. 2 Engine Bench Visualization Test Results (Low-speed Running Conditions)

Observations using a transparent acrylic case and a high-speed camera were conducted in two directions: the front side, where the lubricant was agitated, and the opposing rear side (**Table 1, Fig. 7**). On the front side, where the lubricant was agitated, it was confirmed that the lubricant flowed in from the roller large end face side at the bearing lower section and passed through the rolling surface toward the small end face side (**Fig. 8**). On the rear side of the bearing upper section, it was confirmed that the lubricant that was moving toward the roller small end face side as described above was pushed back and discharged from the large end face side (**Fig. 9**).

Table 1 Test condition (low-speed travel)

Specification	Engine bench test	CAE fluid analysis
Rotation speed	500 min^{-1}	
Lubricant temperature	40℃	
Load	Extremely small load	No load
Analysis subject	–	Single row (shaft end side)

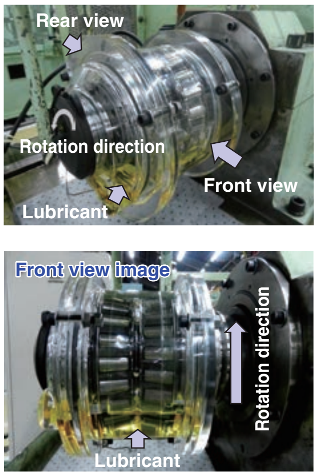

Fig. 7 Video recording state

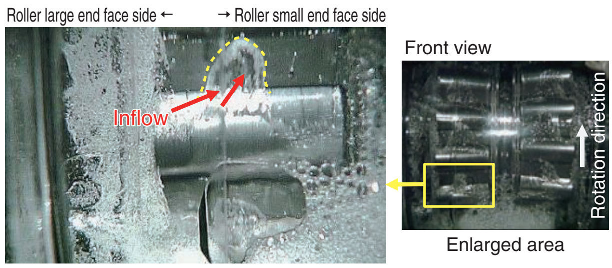

Fig. 8 Inflow of lubricating oil (front)

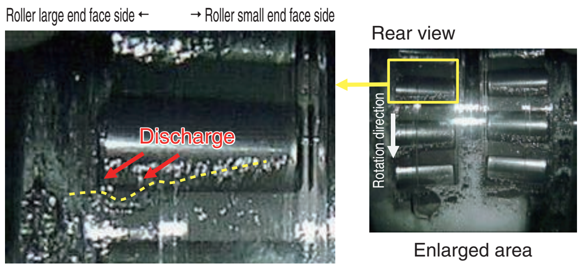

Fig. 9 Outflow of lubricating oil (rear)

3. 3 CAE Fluid Analysis Results (Low-speed Running Conditions)

Fluid flow analysis using CAE was conducted under the same conditions as the engine bench test. It was confirmed that, on the front side where the lubricant is agitated, in the same way as the engine bench test, the lubricant flowed in from the roller large end face side and passed through the rolling surface toward the small end face side (**Fig. 10**), and on the rear side, the lubricant was pushed back and discharged from the roller large end face side (**Fig. 11**).

The engine bench test and CAE fluid analysis were also used to confirm that the direction and phase of lubricant inflow and discharge were similar. From these results, it was determined that the CAE fluid analysis was able to reproduce a flow of lubricant equivalent to that of the actual machine under low-speed running conditions. Consequently, it was considered feasible to estimate the lubricant behavior by CAE fluid analysis for high-speed running conditions.

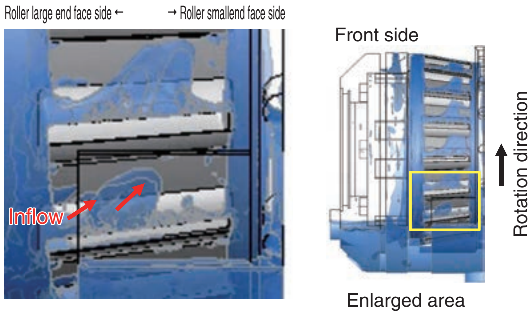

Fig. 10 Inflow of lubricating oil (front)

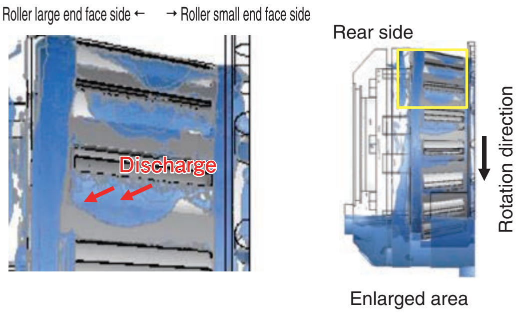

Fig. 11 Outflow of lubricating oil (rear)

3. 4 CAE Fluid Analysis Results (High-speed Running Conditions)

CAE fluid flow analysis was conducted as described in **Section 3.3** using high-speed running conditions (**Table 2**). The flow of lubricant into and out of the rollers was tracked over time by broadly dividing into four surfaces (between the outer ring and cage on the roller small end face side, between the outer ring and cage on the roller large end face side, between the inner ring and cage on the roller small end face side, and between the inner ring and cage on the roller large end face side). As a result, it was found that the inflow and discharge of lubricant were most prominent between the outer ring and cage on the roller large end face side. In addition, there was a tendency for the agitation resistance to increase when the lubricant flowed between the outer ring and cage on the roller large end face (**Fig. 12**).

Table 2 Test condition (high-speed travel)

Specification	Contents
Rotation speed	2 418 min^{-1} (equivalent to train speed of 360 km/h)
Lubricant temperature	95.5℃
Load	No load
Analysis subject	Single row (shaft end side)

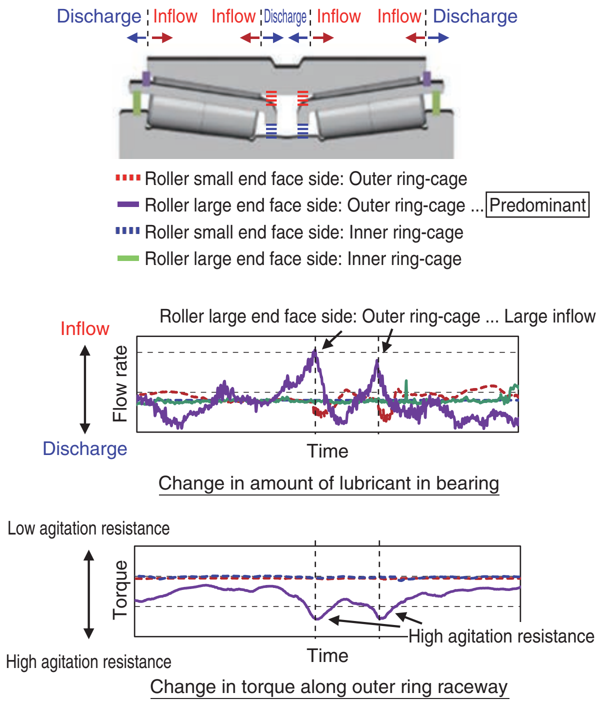

Fig. 12 Relationship between lubricating oil behavior and agitating resistance

And so, we focused mainly on the roller large end face side, where the inflow and discharge of lubricant were large and observed the respective phases, and we noticed the following tendencies (**Fig. 13**).

- Inflow phase: 135° to 225°
  When the rollers rotate in the phase below the lubricant surface, they agitate lubricant from the roller large end face side, which then flows to the roller small end face side.
- Discharge phase: 0° to 135°
  The pumping action of the bearing generates a constant force toward the roller large end face side, and the flow starts to be pushed back from the roller small end face side to the roller large end face side at around 0°.

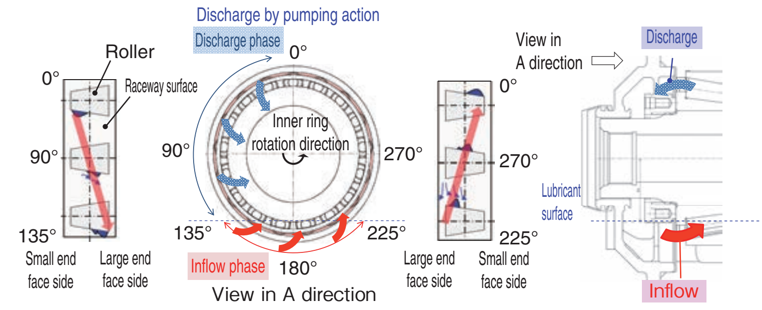

Fig. 13 Lubricating oil inflow phase and outflow phase

4. Design Proposal (1)

4. 1 Concept of Design Proposal (1)

From the results in **Section 3.4**, it was found that the agitation resistance increased when the lubricant flowed in from the roller large end face. The inflow phase was at the 135° to 225° location. If the inflow of lubricant for this phase could be reduced, the agitation resistance would be reduced and the heat generation would be lowered. Therefore, Design Proposal (1) was verified with shield plates (**Fig. 14**) to suppress the inflow of lubricant. The shield plates were designed to be installed on both ends of the outer ring (**Fig. 15**).

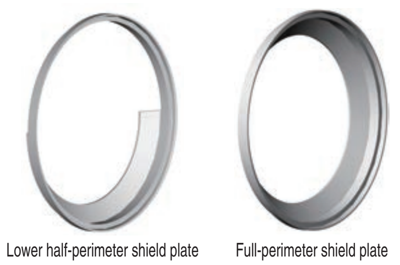

Fig. 14 Shield plate structure

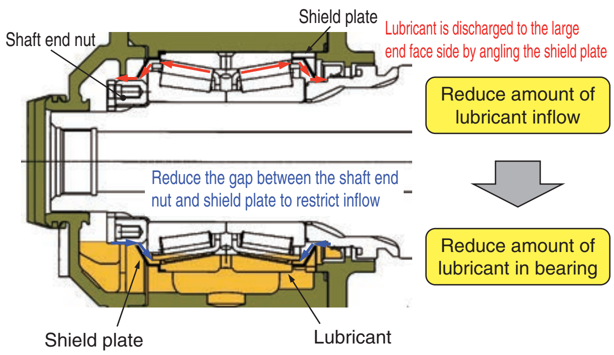

Fig. 15 Image diagram of shield plate installation

4. 2 Engine Bench Temperature Rise Test (High-speed Running Conditions)

A test was conducted under high-speed running conditions (**Table 3**) using bearings fitted with these shield plates. The results of the test were contrary to our expectations and showed no reduction in the temperature rise of the outer ring. On the other hand, a decrease was found in the lubricant temperature measured at the bottom of the housing (**Fig. 16**).

It is thought that the warm lubricant inside the bearing was blocked from being discharged by the shield plates or the protruding part of the housing, and so the lubricant did not circulate to the oil pool side. As a result, the temperature rise of the outer ring was the same as that of the current product and did not decrease, and the lubricant temperature did not decrease (**Fig. 17**). Because there was no change in the temperature rise of the outer ring, it was presumed that the amount of lubricant agitated inside the bearing did not change.

Table 3 Test condition

Specification	Contents
Rotation speed	2 418 min^{-1} (equivalent to train speed of 360 km/h)
Radial load	53.1 kN
Axial load	15.9 kN
Test time	24 hours
Cooling	Wind cooled at 10 m/s

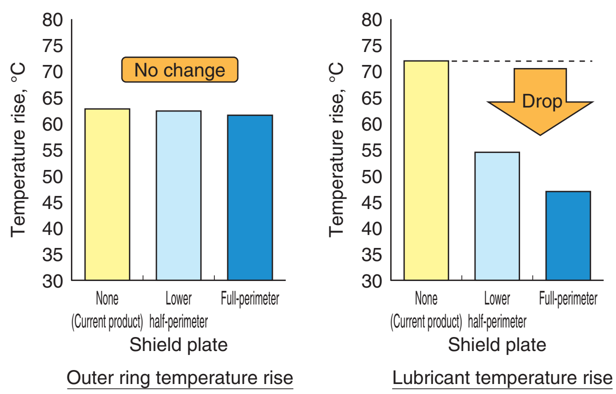

Fig. 16 Test result

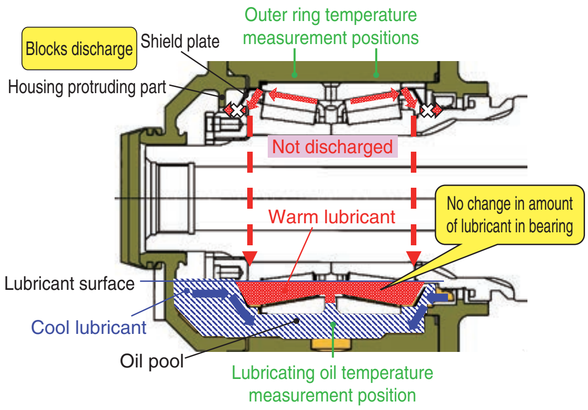

Fig. 17 Image of lubricating oil flow

5. Design Proposal (2)

5. 1 Concept of Design Proposal (2)

Taking into account the findings from the test results of Design Proposal (1), Design Proposal (2) was prepared based on the following concept for reducing the amount of lubricant that is agitated inside the bearing.

- A labyrinth structure is installed in the inflow phase to prevent the discharged lubricant from returning to the inside of the bearing.
- In the discharge phase, a discharge window is provided to avoid blocking of the discharge as much as possible.

A shield plate is mounted on the outer ring outside diameter surface, and a labyrinth collar is mounted on the rotating ring side (shaft end nut) to provide a labyrinth structure with these two parts (**Fig. 18**).

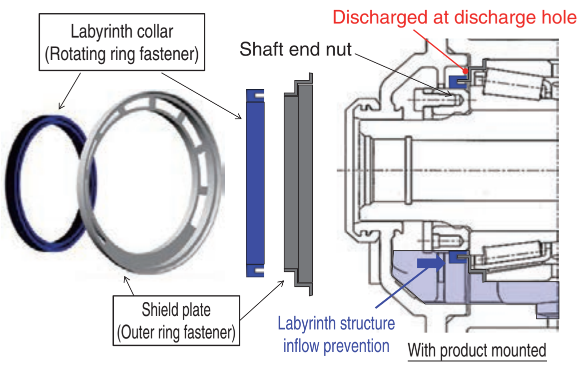

Fig. 18 Structure of developed product, diagram of embedded state

5. 2 Engine Bench Visualization Test (Low-speed Running Conditions)

Before comparing the temperature rise values under high-speed running conditions, observation was conducted using a transparent acrylic case to confirm the inflow reduction effect. Compared with the current product, the amount of lubricant flowing from the roller large end face to the roller small end face was reduced in the bearing lower section for Design Proposal (2) (**Fig. 19**).

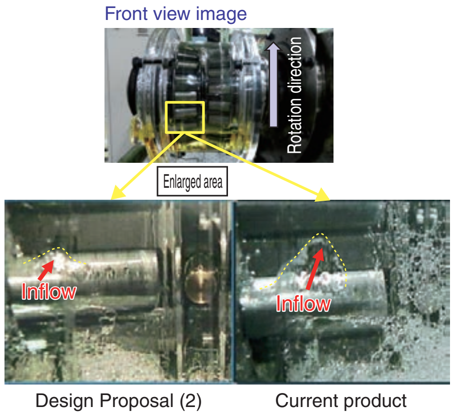

Fig. 19 Observation result

5. 3 Engine Bench Temperature Rise Test (High-speed Running Conditions)

Next, a test was conducted under high-speed running conditions to confirm the performance of Design Proposal (2). For the outer ring, the temperature was successfully lowered by about 6℃ (10%) compared to the current product. The temperature rise of the lubricant was also reduced by about 16℃ (25%), and this result is also expected to lead to reduced thermal degradation of the lubricant (**Fig. 20**).

This resolved the problem found in Design Proposal (1) where the amount of lubricant agitated inside the bearing without being discharged remained unchanged. As a result, both the outer ring and the lubricant showed lower temperature rises, and this enabled reduction of the temperature rise for the oil bath lubricated tapered roller bearings for Shinkansen axles.

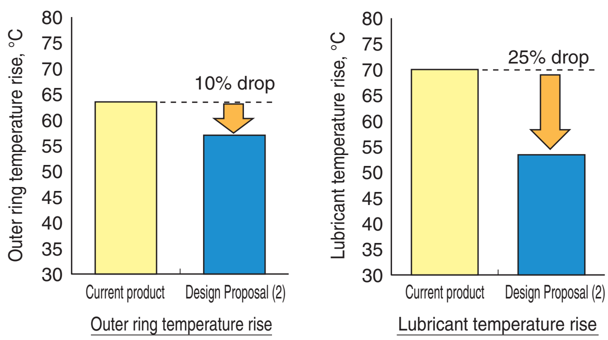

Fig. 20 Test result

6. Conclusion

To achieve lower temperature rises for oil bath lubricated tapered roller bearings for Shinkansen axles during high-speed running, we focused on the behavior of the lubricant inside the bearings. Specifically, in order to observe the behavior of the lubricant inside the bearing under low-speed running conditions, visualization was performed using a transparent acrylic case, and it was confirmed that the results of CAE fluid analysis and the actual behavior of the lubricant were almost identical. Therefore, the flow of lubricant under high-speed running conditions was estimated by taking into account that CAE fluid analysis can be applied even to high-speed running conditions. Based on the results of this study, a labyrinth collar and shield plates were developed to reduce lubricant agitation inside the bearing, and the temperature rise of the outer ring was reduced by about 6℃ (10%) compared to the current product. Looking forward, we would like to further improve the technology for lowering the temperature rise of axle bearings to handle higher speeds for the Shinkansen and for contributing to the stable operation of the Shinkansen.

* LFT is a registered trademark of JTEKT Corporation.

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