Abstract. MHPS and MHI suggest HP/IP/LP separate casing and one bearing/rotor (one bearing between two casings) type steam turbine. To design this type of shaft arrangement, journal bearing that supports high specific load with superior oil film static and dynamic characteristics is demanded. In this paper, it is shown that 610 mm diameter direct lubricated two pads bearing which can operate at high specific load has been developed. Also, it is confirmed that its good bearing characteristics in full scale test.

1. Introduction
In order to meet the requirement of the further improvement of thermal efficiency of steam turbine, the HP/IP/LP individual casing design is suggested as the most effective configuration to increase the thermal heat rate. On the other hand, from the view point of the cost for turbine manufacturing and plant construction, the individual casing design is the most challenging. Therefore, MHPS and MHI suggest HP/IP/LP separate casing and one bearing/rotor (one bearing between two casings) type steam turbine to enhance the turbine efficiency and reduce the cost of construction. To design this type of shaft arrangement, journal bearing that supports high specific load with good static and dynamic characteristics is demanded.

Fig.1 shows the benefit of one bearing for one rotor design against traditional two bearing design. For this case, steam turbine for 1on1 GTCC plant, the turbine total span is decreased by about two meter with one bearing design. This contributes cost reduction greatly. The comparison of the total bearing loss is shown in Fig.2.

History of bearing development at MHPS and MHI is described below. MHPS and MHI have developed various large-diameter tilting pad journal bearings in order to replace the sleeve bearings, which cause the oil whip phenomenon, with the tilt-pad bearings. The performance of static and dynamic characteristics has been confirmed experimentally by using in-house full scale large bearing test rig in MHPS. Also numerical analysis methods for those bearings have been developed. Four pad tilting pad journal bearing with 483mm dia. using steel back metal under flooded lubrication was developed for the HP or HP/IP turbine [1]. After that, two pad tilting pad journal bearing with 533mm dia. using copper pad and cooling ditches was developed for LP turbine also under the flooded lubrication [2]. Recently the nozzle fed type directed lubrication method [3][4] has been applied to two pads journal bearing and four pads journal bearing. Heavy-Load tilted pad bearing of 889 mm dia. operating at half speed for nuclear power plants, has been developed and the good bearing performance was also confirmed [5]. Direct lubrication two pads bearing with 457mm and 610mm dia. for large size gas turbine have also been developed and verified in the test rig and applied commercial operation as the standard bearing [6].

In this paper, it is shown that 610 mm dia. (24inch) direct lubricated two pads bearing which can operate at high specific load has been developed [7]. Bearing performance was estimated by thermos-
hydrodynamic lubrication analysis and structure deformation analysis coupled technique (TEHL), and also confirmed by the full scale test as well. As a result, good performance of the developed bearing was confirmed at high specific load.

**Figure 1.** Total span reduction of steam turbine for GTCC

**Figure 2.** Bearing loss comparison for 200MW class steam turbine

2. Bearing design concept
First, for one bearing for one rotor design, the target specific load in normal operation is set to 1.2 to 1.5 times of the traditional value. This is because to maintain the bearing size even in one bearing one rotor design. Next, the basic design concept of the existing bearing was applied to the developed bearing. MHPS and MHI have already developed the direct lubrication two pads journal bearing and applied some actual plants. This bearing has been proven for its superiority in the low bearing loss due to the bearing upper structure, which eliminate upper pads. Two pads bearing has narrow width guide metal instead of upper pads. Finally, in the developed bearing, a line supported type is adopted in order to make the pivot rigidity higher than the bearing oil film rigidity.

3. Design of developed bearing
The optimization of pad thickness, material, pivot offset and bearing clearance on the two pads bearing was implemented by the MHI in-house 3D-TEHL code [5].

The objective functions were considered as below.
- Pad metal temperature and oil film thickness
- Bearing loss
- Oil film dynamic property ($C_\omega/K$)
- Upstream side pad oil flow ratio

Where, upstream side pad oil flow ratio is an index parameter of flow condition of pad inlet portion [7]. It has been experimentally confirmed that upstream side pad inlet oil flow ratio has strongly relevant to the sub-synchronous vibration characteristics. And, in order to reduce the sub-synchronous vibration, it is effective to use the smaller cavity between upper guide metal and rotor. Therefore, these were adopted in the developed bearings.

4. Experimental set up for full scale test

4.1. Test rig
The prototype of newly developed two pads bearing and test rig was manufactured, and full scale test was conducted in the in-house facility of MHPS. The prototype bearing is of 610mm shaft diameter for 50s⁻¹ machine. The specification of the full scale test is listed in Table 1. The developed bearing and existing bearing are also different in pad dimensions, offset ratio, oil feeding nozzle, oil supply distribution to each pad, etc. Fig.3 shows prototype of newly developed bearing.

The test rig is an internal force balanced type that is not restricted by the base. The test bearing is fixed in the center, and load bearings connected with load levers are arranged on both sides. The load
bearing is connected to the loading lever with connecting link and loading lever is connected to hydraulic jack fixed to base plate of test rig. And also each load bearing is flexibly supported individually by the air bellow actuators (not depicted in this figure) for adjusting the inclination of the test rotor by applying moment force to the loading bearings. The load bearing and its link mechanism, including the test rotor, have a balancing toy structure supported by the center test bearing. A photo of bearing test rig overview is shown in Fig.4.

In this test, both static and dynamic bearing characteristics were measured with changing rotational speed, bearing load, supply oil pressure, and shaft tilt angle, considering the operation of actual plants, and the major results of static and dynamic performance are mentioned in this paper.

### Table 1. Specification of the full scale test

| Units          | Existing                     | Developed                     |
|----------------|------------------------------|------------------------------|
| Bearing type   | Tilting pad                  | ←                            |
|                | Direct lubrication           | ←                            |
| Oil supply method | (spray bars)               | ←                            |
| Oil viscosity grade | ISO VG32                  | ←                            |
| Direction of load | Load between pads           | ←                            |
| Number of pads | 2                            | ←                            |
| Diameter       | mm 610                       | ←                            |
| Pad support type | Pivot supported             | Line supported               |
| Rated speed    | rpm 3000                     | ←                            |
| Supply oil temperature | ℃ 46                    | ←                            |

**Figure 3.** The newly developed two pads bearing

**Figure 4.** Test rig for full scale test
4.2. Measuring method of Dynamic characteristics

In this subsection, equation of motion to calculate stiffness coefficients and damping coefficients of test bearing is indicated. The spring-damper-mass system of the test rig for the dynamic characteristics identification is shown in Fig.5. In this test, the rotor is excited by attaching weights to both ends of the rotor (unbalance force). Each equation of motion of the rotor and loading bearing pad in time domain in the each orthogonal direction is derived from Fig.5 as follows.

For rotor:  \[ M \ddot{x}_0 + c_1 (\dot{x}_0 - \dot{x}_1) + k_1 (x_0 - x_1) = f \]  \( i = 0 \sim 4 \) (1)

For pad:  \[ m_p \ddot{x}_3 + c_3 (\dot{x}_3 - \dot{x}_0) + k_2 (x_3 - x_0) + k_3 (x_3 - x_4) = 0 \]  \( i = 0 \sim 3 \) (2)

\[ \chi_i = X_i \cdot e^{j \omega t} \]  \( i = 0 \sim 4 \) see Fig.5, \( f = F \cdot e^{j \omega t} \) (3)

The equation of motion to obtain test bearing stiffness coefficients \( k_1 \) and damping coefficients \( c_1 \) is derived from Eq.1 and Eq.2 as follows.

\[ k_1 (x_0 - x_1) + c_1 (\dot{x}_0 - \dot{x}_1) = f - M \ddot{x}_0 - m_p \ddot{x}_3 - k_3 (x_3 - x_4) \]  \( i = 0 \sim 4 \) (4)

Eq.5 of frequency-domain is derived from Eq.4 with Fourier transform.

\[ (k_1 + j \omega c_1) (X(\omega)_0 - X_1(\omega)) = F(\omega) + M\omega^2 X_0(\omega) + m_p \omega^2 X_3(\omega) - k_3 (X_3(\omega) - X_4(\omega)) \]  \( i = 0 \sim 4 \) (5)

\[ Z(\omega) = (k_1 + j \omega c_1) = \frac{F(\omega) + M\omega^2 X_0(\omega) + m_p \omega^2 X_3(\omega) - k_3 (X_3(\omega) - X_4(\omega))}{X(\omega)_0 - X_1(\omega)} \]  \( i = 0 \sim 4 \) (6)

\[ Re(Z(\omega)) = k_1 \quad Im(Z(\omega)) = \omega c_1 \]  \( i = 0 \sim 4 \) (7)

In this test, mass of rotor \( M \), mass of loading bearing pad \( m_p \) and back spring stiffness \( k_3 \) are obtained in advance. For the test, measurement data is derived through vector filter. Unbalance force \( F \), displacements of rotor \( X_0 \) and loading bearing \( X_3 \), relative displacement of test bearing pad to rotor \( (X_0 - X_1) \) and relative displacement of test bearing pad to loading bearing carrier ring \( (X_3 - X_4) \) are measured. The displacement \( X_1 \) calculated by subtracting the vibration vectors measured by two test runs is used. These measurement results are substituted to Eq.6, bearing stiffness \( k_1 \) and damping coefficients \( c_1 \) are obtained. Then, the inertia force of rotor \( M \ddot{X}_0 \) is actually considered the mode shape of rotor by multiple sensors which measure the rotor displacement.

5. Results and discussions

5.1. Measurement results (comparison between developed and existing bearing)

The result of the verification test is shown in Fig.6 and Fig.7. The pad metal temperature was compared to the existing two pads bearing in Fig.6. The metal temperature in actual unit was verified to have enough safety margins all over the operation range. The target specific load has surely been achieved. Fig.7 shows the measurement result of the broadly range sub-synchronous vibration. The Y-axis is the amplitude of broadly range sub-synchronous vibration, the overall amplitude eliminated synchronous component [7]. The developed bearing’s amplitude of low frequency vibration was greatly improved than the existing two pads bearing in all over the specific loading range. This result indicates that the design optimization for this type of vibration was correct and adequate.

Moreover, the mechanical loss of this bearing was precisely predicted by the empirical method based on the existing two pads bearing, and it is verified that it has the quite low bearing loss characteristics through the existing direct lubricating bearing. Fig.8 shows the bearing loss comparison, and the developed two pads bearing loss is about 0.6 times as the existing two pads including the size down effect brought by the countermeasures of higher specific load.
Figure 5. The spring-damper-mass system of the test rig

Figure 6. Result of pad metal temperature

Figure 7. Result of broadly range sub-synchronous vibration
Figure 8. Comparison of the bearing loss including the size down contribution

5.2. Measurement results (comparison with THEL analysis)

The measurement results are compared with TEHL analysis results in Fig.9 and Fig.10.

Fig.9 shows the comparison with analysis results oil film pressure and oil film thickness of upstream side pad from top to bottom @3000rpm. As shown in Fig.9, oil film pressure and oil film thickness are in good agreement with TEHL analysis results.

(a) Normalized bearing specific load 1.0 @3000rpm

(b) Normalized bearing specific load 0.83 @3000rpm

Figure 9. Comparison with Oil film pressure and thickness by TEHL analysis (Upstream side pad)
In Fig. 10, Comparative result of bearing dynamic characteristics is shown. The measurement result is the value calculated by Eq. 7. As shown in Fig. 10, measurement results of stiffness coefficient and damping coefficient are in good agreement with TEHL analysis results. In particular, damping coefficients is remarkable.

Summarizing these results, it is confirmed that the results of TEHL analysis show good agreement with full scale test results of 610mm diameter, direct lubricated two pads bearing. But, at the same time, various differences point are revealed. For more accurate estimation, it is necessary to try to grasp of the mixing condition between pads, prediction of more exact pad deformation, and grasp of the influence of partial oil starvation at pad inlet. Also it is necessary to measure the flow condition / pattern of bearing various cavity space and boid ratio of oil and air mixture of such portion.

![Diagram](image)

(a) Normalized bearing specific load 1.0

![Diagram](image)

(b) Normalized bearing specific load 0.83

Figure 10. Comparison with bearing dynamic coefficient by TEHL analysis

6. Conclusions

The new two pads bearing for high specific loading was developed to realize the one bearing for one rotor steam turbine shaft configuration. This configuration has advantage in the shorter total span and smaller mechanical loss. The prototype developed bearing was manufactured and tested in full scale test bearing size (610mm dia.). And the result of the test was totally satisfying for the design target. Furthermore, it is confirmed that the results of TEHL analysis show good agreement with full scale test results.
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