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Original Paper

Water Lubricated Guide Bearing with Self-aligning Segments

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Abstract

Water lubricated guide bearing was newly released and has been applied to actual hydro turbines with vertical shaft. As a result, they can have not only high bearing performance but environmental advantages in meeting the demand for reducing river pollution by oil leakage from oil lubricated guide bearing. The PTFE composite guide bearing was tested by experimental equipment operated under conditions similar to those of actual hydro turbines. Circumferential and axial tilting bearing segments help to improve the bearing performance and efficiency due to low friction loss in the bearing system. Furthermore, bearing cooling systems could be eliminated and maintenance periods could be extended, thus the initial investment and operating costs of the hydroelectric power plant are reduced.

Keywords: Water lubricated guide bearing, PTFE, Tilting segment, Vertical shaft hydro turbine, Hydrodynamic lubrication

1. Introduction

Hydroelectric power generation, which provides electricity from clean and renewable energy without CO2 emissions, has the highest conversion efficiency and the fastest response in providing a stable power supply at low operating cost. In Japan there is not much room left for development of large capacity hydropower plants. In order to fulfill market demand for clean and renewable energy supply, small and medium capacity hydropower plants have been recently developed.

The service intervals for hydropower plants have been extended and in other words, their long maintenance intervals strongly request that the radial and thrust bearings must have extremely high stability and reliability for supporting the rotating parts such as runner, turbine shaft, generator shaft and rotor. The oil lubricated radial bearings (hereinafter to be called guide bearings instead of radial bearing) have been usually applied to the hydro turbines and generators, but on the other hand the water lubricated guide bearings have recently had a strong background in construction of the hydro plants in order to prevent oil pollution in river water. The turbine guide bearing for multi-nozzle vertical Pelton turbine must have high stability throughout all kinds of operation enough to withstand high bearing pressure during operation driven with only one active nozzle. In case that shaft alignment is out of tolerance, there are possibilities of shaft vibration or partial contact between the bearing segment and the rotating shaft and, therefore, the turbine guide bearing must have preventive measures against abnormal wear and significant damage on the bearing sliding surfaces.

This paper mainly describes the performance of the water lubricated guide bearing which has recently developed to solve the problems mentioned above.

2. Structure of Water Lubricated Guide Bearing

2.1 Structure and Problems of Conventional Water Lubricated Guide Bearing

The conventional water lubricated guide bearing with sliding surface of phenol resin has been applied to Kaplan turbines but it has

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the following problems.

Phenol resin in thermosetting type polymers is lacking in wear resistant and the invasion of the hard particles like river sand into the sliding surfaces of phenol resin is strictly prohibited. Therefore clean water without hard particles through water strainer is supplied to the sliding surfaces and, furthermore, pressure and flow of the supplied water is controlled accurately. The maintenance of this water supply equipment requires high cost and much manpower.

Water absorption rate of phenol resin is so high that the clearance between the shaft journal and the sliding surface of the bearing segment becomes smaller gradually after commencement of the plant operation and finally the bearing performance may drop. Therefore the sliding surfaces of the phenol resin bearing are always finished after the bearing fully expands due to water immersion and absorption. Even if the phenol bearings are set up carefully to ensure the accurate bearing clearance, the clearance may be out of tolerance with the passage of the plant operating time and, if so, the bearing clearance can not be readjusted.

If the misalignment and/or the vibration of the hydro turbine with sleeve type guide bearing like the conventional water lubricated bearing is large, the shaft contacts the upper or lower portion of the bearing partially and/or strongly to cause damage or abnormal wear of the bearing sliding surface.

2.2 Structure and Material of Newly Developed Water Lubricated Guide Bearing

The new water lubricated guide bearing comprises the water basin (7, 8, 9 and 10) for holding lubricating water and the bearing segments (2) for radially supporting the shaft journal (1a) installed to the rotating shaft (1) as shown in Fig.1. Each bearing segment is supported by the adjusting bolt (3) with round tip and the ball transfers (4). Bearing clearance between the bearing segment (2) and the shaft journal (1a) is adjusted by letting the adjusting bolt move forward or backward.

Cooling coil for controlling temperature of lubricating oil is usually installed for oil lubricated bearing, but almost no cooling coil is necessary for water lubricated guide bearing with small friction loss.

Polytetrafluoroethylene (PTFE) composites with filler materials such as carbon fiber was selected as the sliding material of the bearing segments (2) after evaluating the results of wear and friction tests performed on a pin-on-disc device. The bearing components with surfaces in contact with water or moist air are made of stainless steel. In addition rust prevention surface treatment is applied to martensitic stainless steel.

Tap water or deionized water is used as the lubricant.

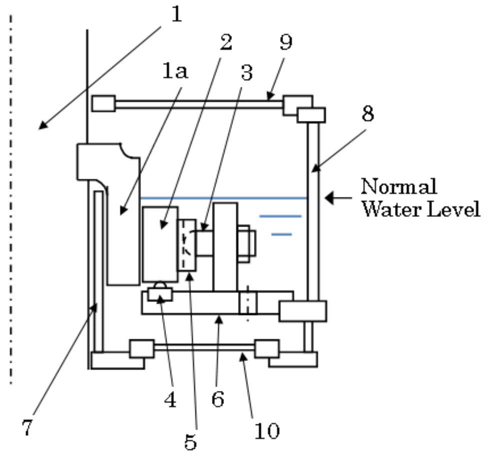

Fig.1 Water Lubricated Guide Bearing with Self-aligning Segments

3. Performance Test Results

Performance test was carried out with the test equipment that has the rotating shaft with 500mm journal diameter as large as one for actual hydro turbine. The shaft was pushed radially by the hydraulic cylinder force imposed on the outer collar of the ball bearing that was installed to the shaft lower end and this radial force was supported with the water lubricated guide bearing and another ball bearings for the test equipment. Bearing force F1 to F4 for the water lubricated guide bearing was separated from the above radial force and transmitted to the bearing segments as shown in Fig.2.

The followings were measured and recorded; rotating speed, shaft torque, bearing force, water film pressure, bearing temperature and so on. The rotating speed and the shaft torque were measured with the torque transducer installed between the induction motor and the rotating shaft. The bearing force was calculated from compressive strain measured with strain gauges bonded to all adjusting bolts. The bearing capacity was calculated by converting radial forces on bearing segments into the same direction as that of hydraulic oil cylinder force “F” and then summing up the converted ones. The average bearing pressure was calculated by dividing the bearing capacity by the projected area of the bearing. The water film pressure on each bearing segment was measured with pressure sensor embedded in the surface of the rotating shaft and its distribution including peak pressure on each bearing segment was also detected accurately by adjusting sampling interval. And the bearing temperature was measured with thermocouples embedded in the bearing segments.

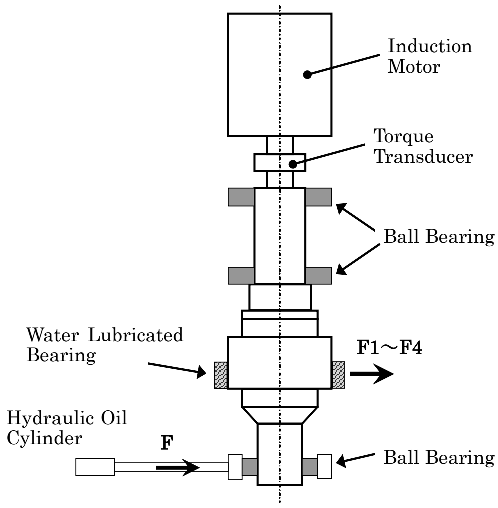

Fig.2 Test Equipment

The performance test was carried out under the condition of low to high bearing pressure at several different rotation speeds that were altered and set with the inverter for the induction motor.

3.1 Test Results of Bearing Characteristics

The relationship between arrangement of bearing segments and direction of bearing force is classified into LBP and LOP as shown in Fig.3. LBP is abbreviation of "Load Between Pads" and LOP is abbreviation of "Load On Pad". Bearing characteristics tests were executed under the condition of both LOP and LBP arrangement, where the parameters such as bearing pressure and rotational speed were changed beyond ones for actual hydro turbines.

The relation between coefficient of kinetic friction and Stribeck Number (or Bearing Number) ηN/P (η; lubricant viscosity, N; rotation speed, P; average bearing pressure) is defined as Stribeck curve (Fig.4A). The relation between coefficient of kinetic friction and Sommerfeld Number ηN/P(R/C)2 (R; radius of journal, C; radial bearing clearance) is also defined as Stribeck curve (Fig.4E). Coefficient of kinetic friction is calculated by balancing measured torque with friction torque on bearings. The state of lubrication under realistic operating conditions of actual hydro turbine were plotted in Fig.4B for LBP and Fig.4D for LOP. They showed that all test results were in hydrodynamic lubrication. Severer operating condition implied the possibility of the transition from hydrodynamic to mixed film lubrication (Fig.4C). Computer analysis of water film pressure distribution and water film thickness on the sliding surface of the bearing segment was exemplified in Fig. 6A and Fig.6B respectively. Computer software applied to the analysis is for internal use only.

The test results were summarized as follows.

① Lubricated condition in all tests was classified into hydrodynamic lubrication. (Fig.4A, Fig.4B and Fig.4D)

② In case of eight bearing segments, the bearing with the arrangement LBP had larger bearing capacity than that for LOP. (Fig.8A and Fig.8B)

③ The bearing segments for supporting the load imposed through the rotating journal had water film pressure on their sliding surfaces regardless of the bearing arrangement LBP or LOP. (Fig.5A and Fig.5B)

④ Water film pressure on each bearing segment was distributed over its whole sliding surface and pressure peak existed near the center of the segment while the pressure decreased gradually in both circumferential and axial direction. (Fig.5A and Fig.6B)

⑤ Defining the transition condition from hydrodynamics to mixed film lubrication was dependent on the operating condition such as average bearing pressure and rotational speed. Lubricated condition in the test with bearing pressure 0.16MPa and peripheral velocity above 1.5m/s was in hydrodynamic lubrication. (Fig.4C)

⑥ The newly developed bearing has a lower coefficient of kinetic friction than that of conventional phenol resin sliding bearing lubricated in water. (Fig.4E)

⑦ Bearing temperature was affected by test condition such as bearing pressure, peripheral velocity and environmental condition including water temperature, room temperature and wind velocity. It was affected mainly by peripheral velocity in case of fixed bearing pressure 0.5MPa. In addition to this, bearing temperature of actual hydro turbine could be affected by river water temperature and flow velocity. (Fig.7)

⑧ In case of hydro turbine with circumferential velocity of about 20.9m/s of shaft journal, the estimated temperature of the bearing segment imposed with maximum radial force is around 60 degrees Celsius at room temperature of 30 degrees Celsius in summer and also the estimated temperature of other bearing segments is at most 60 degrees Celsius. Judging from the bearing performance, cooling equipment for bearing is not always necessary. (Fig.7)

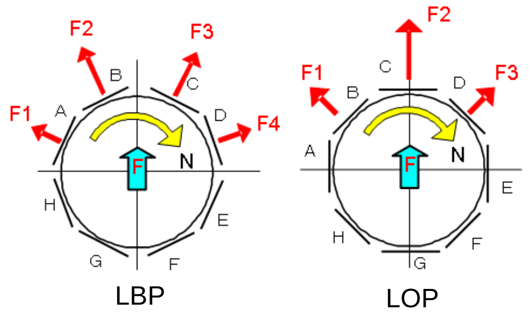

Fig.3 Arrangement of Bearing Segments

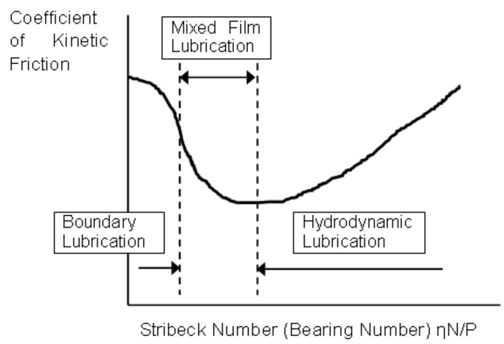

Fig.4A Stribeck Curve

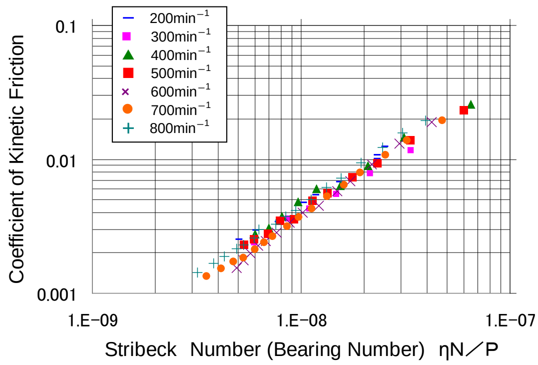

Fig.4B Stribeck Curve (LBP)

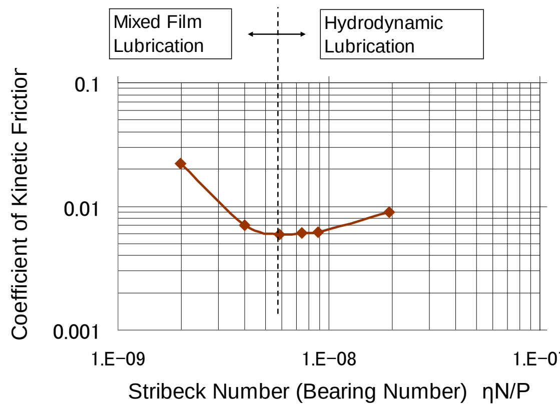

Fig.4C Stribeck Curve (LBP, 0.16MPa)

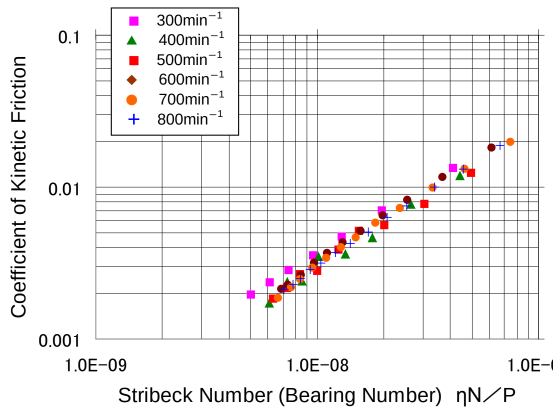

Fig.4D Stribeck Curve (LOP)

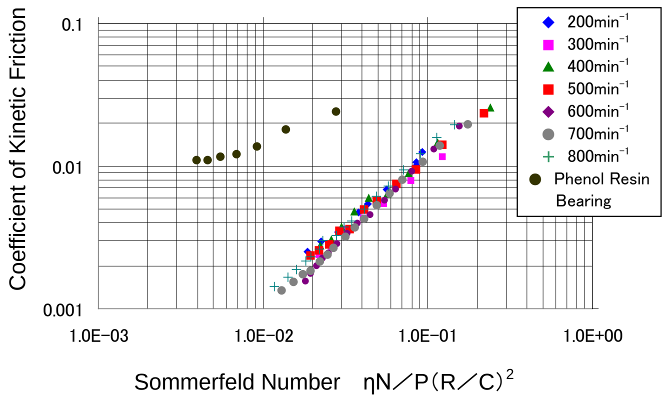

Fig.4E Stribeck Curve (LBP)

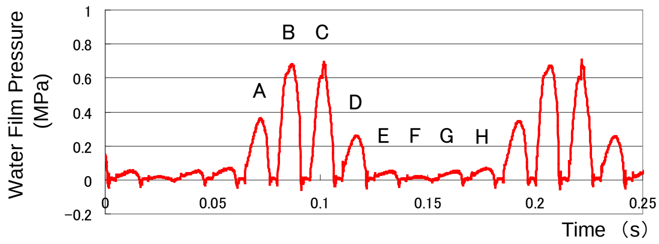

Fig.5A Water Film Pressure Distribution
(LBP, 500min^{-1}, 0.45MPa)

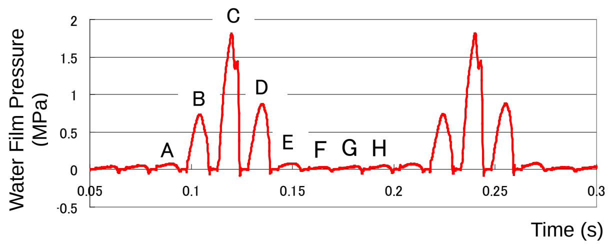

Fig.5B Water Film Pressure Distribution
(LOP, 500min^{-1}, 0.48MPa)

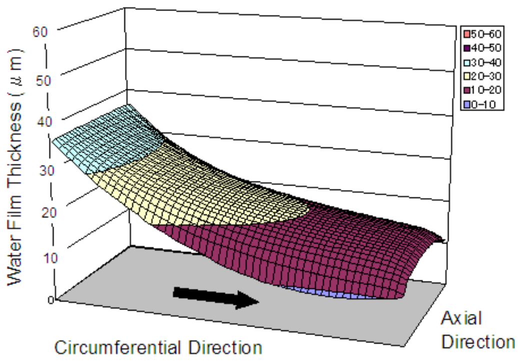

Fig.6A Analysis of Water Film Thickness
(LBP, Bearing Segment “B”)

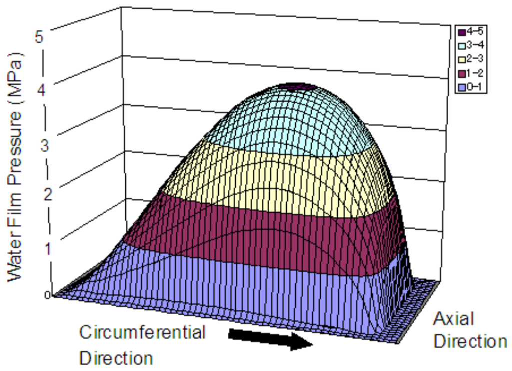

Fig.6B Analysis of Water Film Pressure
(LBP, Bearing Segment “B”)

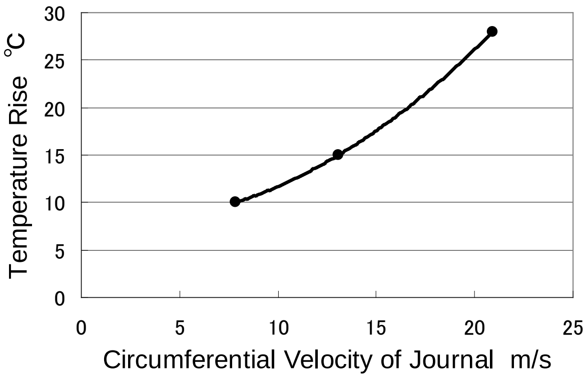

Fig.7 Temperature Rise of Bearing
(LOP, 0.5MPa, Bearing Segment “C”)

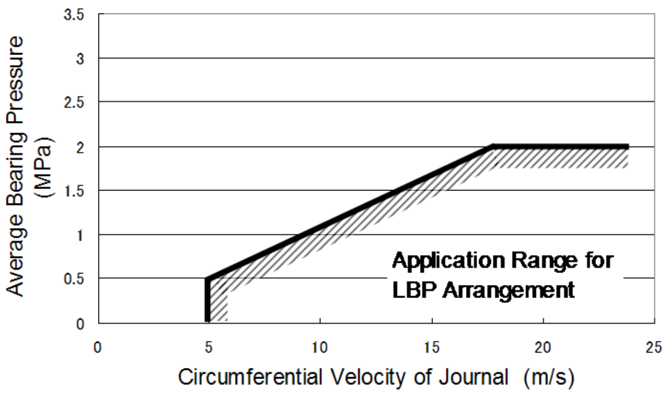

Fig.8A Application Range
for Water Lubricated Bearing

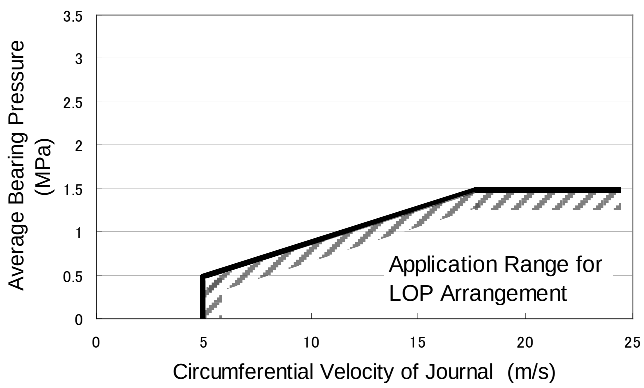

Fig.8B Application Range
for Water Lubricated Bearing

3.2 Test Results of Endurance Characteristics

It was due to similar operating condition to that of actual hydro turbine that condition Ⅰ (500min^{-1}, 0.5MPa), condition Ⅱ (648min^{-1}, 1.0MPa) and condition Ⅲ (648min^{-1}, 1.9MPa) were selected for endurance test. Bearing segment arrangement was LOP for condition Ⅰ and LBP for condition Ⅱ and condition Ⅲ.

Test results were as follows.

① Coefficient of kinetic friction was 0.005 to 0.006 through the test under condition Ⅰ (Fig.9) and 0.001 to 0.002 through the test under condition Ⅲ. Each coefficient was low and stable.

② High bearing load was safely supported with four bearing segments on the active side of the journal in case of the condition Ⅲ. (Fig.10)

③ Bearing temperatures at some specific points were stable and their change was within approximately one degree Celsius. (Fig.11)

④ The sliding surface of the bearing segment had slight wear after the test under the condition Ⅰ. The estimated wear after ten years operation under the condition Ⅱ and the condition Ⅲ was smaller than the design bearing clearance. (Fig.12)

⑤ Radial force of Pelton turbines operated under the condition of use of pressurized water ejecting from only one nozzle is so large that average bearing pressure of turbine guide bearing rises to about 1MPa, but they can continue to maintain stable and commercial operation. Maximum increase in bearing clearance after ten years operation of these Pelton turbines is expected to be smaller than the designed one. (Fig.4B, Fig.8A, Fig.12)

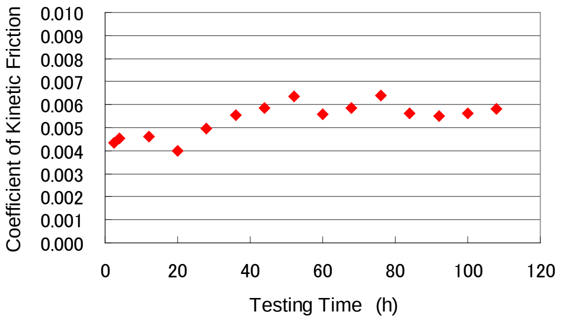

Fig.9 Change of Coefficient of Kinetic Friction (LOP、500min^{-1}、0.50MPa)

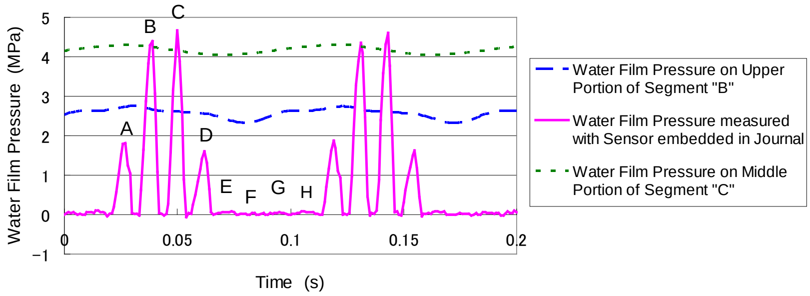

Fig.10 Water Film Pressure Distribution (LBP, 648min^{-1}, 1.9 MPa)

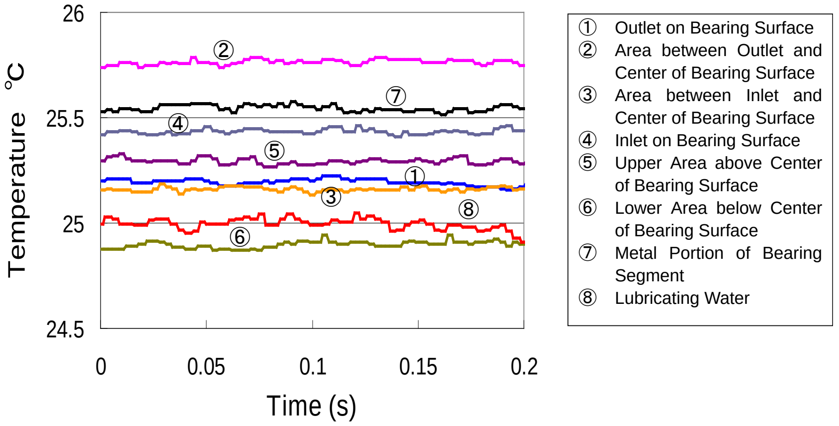

Fig.11 Change of Temperature of Bearing (LBP, 648min^{-1}, 1.9MPa, Bearing Segment "C")

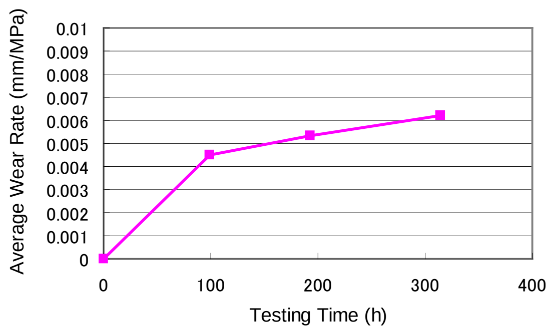

Fig.12 Change of Wear Rate (LBP, Bearing Segment "B")

3.3 Test Results after Start-Stop Operation

Start-stop operation test was carried out under the cyclic condition that had bearing pressure 0MPa and rotational speed 0min^{-1} during standstill state and 0.18MPa and 500min^{-1} during no-load operation. The test results verified the stability and reliability of the bearing under severer operation including mixed film lubrication.

In case that hydro turbine is operated on average twice a day, 7300 and more start-stop cycles tested is equal to 10 years or more after use.

Test results were as follows.

① Coefficient of static friction on the start of sliding was about 0.10 to 0.17. (Fig.13)

② Roughness of the sliding surface became a little large but it caused no abnormal wear on the surface. (Table 1)

③ There was no peeling of PTFE material from the sliding surface.

④ If small and hard particles such as metal, rust or sand do not invade into the bearing clearance between the journal and bearing segment, the sliding surface of each segment has no abnormal wear and no peeling of sliding material.(Fig.12, Table 1)

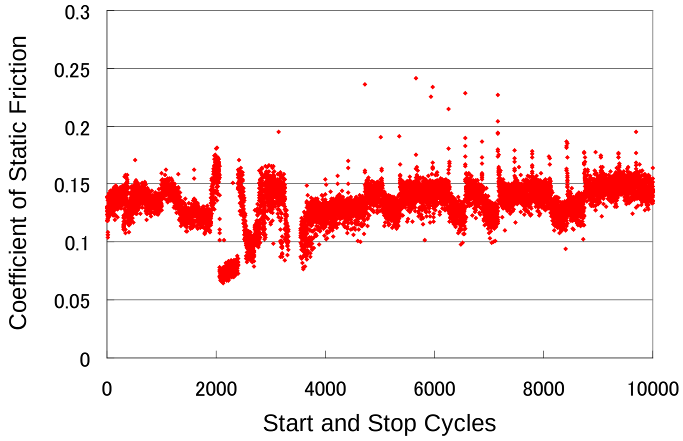

Fig.13 Change of Coefficient of Static Friction
(LOP, 0⇔0.18MPa, 0⇔500min^{-1})

Table1 Change of Sliding Surface Roughness of Bearing Segments

		Bearing Segment "C"	Bearing Segment "E"	Bearing Segment "F"	Bearing Segment "G"	Journal
Ra	Before	0.64	0.73	0.78	0.40	0.10
μm	After	1.03	0.97	1.03	1.04	0.20

3.4 Test Results of Bearing Function under Low Water Level Condition

When water as a lubricant leaks from the water basin and evaporates from water surface, water level drops below the designed one. The test executed at this abnormally low water condition confirmed that hydro turbine could be kept running and stopped under the condition of bearing pressure 0.5MPa and rotational speed 500min^{-1} without the damage of the bearing segments. It was because the water basin was radially divided into two sub-basins, the inside reservoir had higher water level than that of the outside one and the bearing segments installed in the inside one were immersed fully or safely in the water enough to attain the hydrodynamic lubrication.

① Even if the water level was low in the water basin, water could be supplied to upper one of the bearing sliding surface with the help of self pumping function of the bearing. (Fig.14)

② When the water level in the basin falls below the lower sensor or the threshold level, water needs to be added to the water basin until the water level is equal to or a little above the upper water level. (Fig.15)

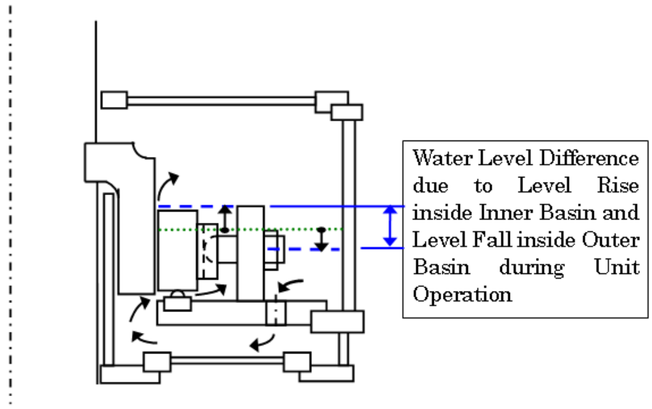

Fig.14 Lubrication of Bearing Segments under Low Water Level Condition

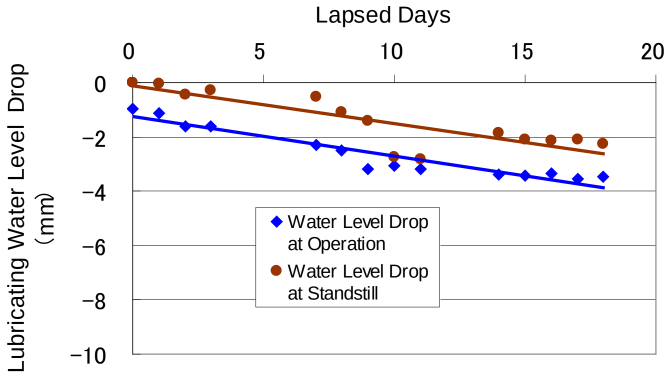

Fig.15 Change of Lubricating Water Level Drop through Evaporation

4. Conclusion

Authors have concluded the followings through all tests with similar operating condition to that for actual hydro turbine.

① This guide bearing can be applied to many of hydro turbines because all tests for the bearing covered a wide range of bearing pressure and rotational speed of actual hydro turbine and all test conditions were included in hydrodynamic lubrication.

② Bearing segments can support the rotating shaft against radial force and in case of eight segments, four segments for LBP share the force and three ones for LOP share it, and the bearing with LBP arrangement has larger bearing capacity than that of the bearing with LOP arrangement.

③ Coefficient of kinetic friction is below 0.1, and coefficient of maximum static friction is below 0.3.

④ Even if water level in the water basin falls too low, hydrodynamic lubrication could be maintained enough to stop hydro turbine safely.

Nomenclature

Abbreviations

LBP Load Between Pads
LOP Load On Pad

Notations

F Hydraulic cylinder force imposed on collar of lower ball bearing around shaft [N]
F1~F4 Bearing forces due to F [N]
η Lubricant viscosity [MPa・s]
N Rotation speed [s^{-1}]
P Average bearing pressure [MPa]
R Radius of journal [mm]
C Radial bearing clearance [mm]

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