Energy cost reduction in tilting pad thrust bearings

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SYNOPSIS Hydrodynamic, tilting pad thrust bearings are used in many industrial applications. In every case energy is absorbed by the bearing and dissipated in the form of heat. The purpose of this paper is to explore the energy consumption of thrust bearings for high speed applications under different operating conditions. It is demonstrated that a reduction in the rate at which oil is supplied can lead to significant lowering of the energy loss incurred by the bearing at no risk to its safety. Comparative experiments are reported involving offset pivot thrust pads, subject to forward and reverse rotation of the thrust collar and centre pivot thrust pads. It is found that these changes in thrust pad pivot position do not affect the energy consumed by the bearing.

1. INTRODUCTION

Oil lubricated hydrodynamic thrust bearings rely on a plentiful supply of lubricant being drawn into a convergent space thus generating a load carrying film. In many cases the supply of lubricant is guaranteed by arranging for the working faces of the bearing to be immersed in oil. This arrangement, often referred to as "flooded" lubrication, whilst very satisfactory for lower speeds, is much less suitable for high speed use since it leads to prohibitively large amounts of energy being absorbed by the bearings. Energy consumption derives from two sources; necessary frictional losses caused by shearing in the lubricating film and parasitic losses due to churning of the rim of the thrust collar in the surrounding oil. The effect of churning is not significant at low speeds, but at higher speeds typically above 40 m/s at the mean pitch diameter of the bearing, the associated energy losses increase rapidly to equal or even exceed frictional losses.

Two fundamental approaches have been devised to overcome the problem of parasitic losses and enable high speed bearings to operate successfully with acceptable power loss figures. The first method, referred to in this paper as 'shrouded' lubrication, requires that the thrust collar rim be surrounded by a shroud or baffles. The presence of a restriction at the collar rim substantially reduces the amount of possible churning. The second basic approach, known as 'low loss' or 'directed' lubrication, is a very simple one in which the rim of the collar runs free of oil and churning losses are virtually eliminated.

There have been a number of published reports in which the two design philosophies are described (1,2,3,4,5). Energy consumption in a bearing is manifested principally by the temperature rise between supply and drain of the oil which passes through the system. The ability of energy efficient design to limit this increase when compared with conventional, flooded lubrication has been well established by previous investigations.

The main conclusion of a recently published report (6) on a direct comparison between the two design philosophies was that the low loss approach was the more energy efficient by about 25 per cent when operating conditions were the same for both configurations. Because the principle of the low loss bearing is a very simple one, it was suggested in the same report that bearings of this sort are the most cost-effective arrangement currently available for high speed, energy efficient thrust bearings.

This paper reports the results of some further experiments with an example low loss bearing which have been carried out to determine the beneficial effects in energy terms of changing its operating conditions. In particular the bearing was operated with substantially reduced oil flow and with offset pivot and with centre pivot thrust pads.

2. EXPERIMENTAL APPARATUS

The experiments were carried out using a double thrust bearing selected from the middle of a standard range. Each thrust face comprised eight tilting thrust pads supported by a ring of load equalising segments of conventional design. Basic dimensional information about the bearing is given by Table 1. The bearing was assembled in a new casing mounted on an existing, high speed, horizontal bearing rig which has been described on a previous occasion (1). The bearing configuration and lubrication arrangements are shown by Figure 1.

Oil is supplied under positive pressure from an external source to an annulus around the circumference of the retaining ring which contains the equalising segments. Thence it is directed on to the thrust collar through orifices inserted in the retaining ring. In the low loss bearing the diameter and number of these orifices are used to meter the flow of oil to the thrust surfaces. From a design point of view it is easy to cater for any duty simply by changing the size and/or the number of holes. Overall size of bearing represents no limitation since the number of holes can be increased to ensure adequate supply across the full width of the thrust pad. In the bearing used in the experiments described in this paper there were two orifices, each 3 m.m. in diameter, between each pair of thrust pads. The emergent jets of oil are directed on to the surface of the thrust collar and a proportion of lubricant is drawn into the hydrodynamic films between the thrust pads and collar. Lubricant is discharged from the bearing simply by being allowed to fall, unrestricted, to drain by way of the large opening in the lower half casing.

3. OPERATING CONDITIONS

Axial load was applied to the test bearing by a series of hydraulic pistons positioned behind the retaining ring of a completely separate loading bearing adjacent to the test module. The effect of pressurising the hydraulic cylinders in the loading bearing was to create an axial force on the shaft which was absorbed by one of the faces of the test bearing. The main series of experiments reported in this paper, were performed at specific loads of 2 MPa and 4 MPa.

The d.c. motor and thyristor drive of the experimental rig meant that speed was continuously variable with maximum speed being limited only by the power available. However, since the energy savings mechanisms are most effective at higher speeds experiments were conducted at three speeds, 5000 rev/min, 7500 rev/min and 10,000 rev/min, typical of high speed equipment such as compressors, turbines and gearboxes. These rotational speeds are equivalent to sliding velocities at the mean pitch diameter of the thrust ring of 47.3 m/s, 70.9 m/s and 94.6 m/s respectively.

The lubricant used throughout the experiments was a standard ISO VG 32 turbine oil supplied to the casing at a nominal 45°C. In practice experiments took place with oil supply temperatures between 44°C and 46.5°C. It is usual to base recommended rates for oil supply on the expected rise in temperature between the supply point and the drain point from the bearing casing. Values of lubricant temperature increase up to about 17°C are common industrial practice. In the current experiments the oil flow rate was reduced at each operating speed and load from an initial amount to a rate about half that originally supplied with a consequent increase in oil drain temperature. The reductions in oil flow were carried out in stages and the experimental results are reported below.

The oil flow reduction experiments were carried out when the bearing being examined was fitted with offset pivot pads. Pivot position was 0.6 of the distance between the leading and trailing edges of each pad as shown by Figure 2. Subsequently the bearing was operated in reverse for extended periods at the 2 MPa and 4 MPa loadings, with a reduced oil supply. Following this the offset pivot pads were replaced by centre pivot pads and the experiment repeated.

4. INSTRUMENTATION

The temperatures of oil supplied to and leaving the bearing were monitored by thermocouples at inlet and drain respectively. Oil flow rate was measured using a calibrated, variable orifice meter placed in the supply line. These values enabled power absorbed by the test bearing to be calculated by multiplying flow rate and temperature difference (T_{diff}) between supply and outlet with the specific heat of the oil concerned.

The most important and widely used parameter in monitoring hydrodynamic thrust bearing safety is maximum thrust pad temperature. This temperature (T_{max}) was measured by thermocouples embedded 3 m.m. below the whitemetal (babbitt) surface at the centre of the outboard trailing quadrant formed between the mean pitch diameter and the pad pivot as shown by Figure 2 for the offset pivot pad. This location is in agreement with the suggestions of other workers in this field (7,8) and its suitability has been discussed in an earlier paper (1) specifically concerned with measurements of maximum temperature in tilting pad thrust bearings. All the loaded pads were instrumented in the same way and the values given in the results are the arithmetic mean of the readings taken from the eight pads on the loaded side of the bearing.

In the experimental work thermocouple temperatures were recorded continuously by pen plotter. The time it took for values to settle after each change of duty was about 15 minutes. Thereafter the given conditions were maintained for about 1½ hours to ensure that the readings were properly stabilised and not subject to long term variations.

5. EXPERIMENTAL RESULTS

5.1 Reduction of oil flow with offset pivot pads

Figures 3 and 4 show maximum pad temperature, T_{max}, and temperature difference between supply and drain, T_{diff}, plotted against oil flow for the test bearing at the 2 MPa and 4 MPa specific loadings respectively. In each case as the initial oil flow is reduced by up to about half there is a gradual increase in both maximum pad temperature and in the temperature rise of the oil between supply and drain. Maximum pad temperature remains well within acceptable and safe limits for whitemetal thrust pad surface material. The temperature of the lubricant as it passes through the bearing increases from initial values lying between 10°C and 18°C to a maximum value, recorded for the 4 MPa load at 10000 rev/min of 28°C. The bearing performed well throughout these experiments. When it was stripped down subsequently there was no sign of any distress to any of the components and the thrust pad surface material appeared in good condition.

The 2 MPa results of Figure 3 are repeated as broken lines on Figure 4 so that the effect of increasing load on T_{max} and T_{diff} can be seen. For a given speed and oil flow rate T_{max} increased by 20°C to 25°C for the specific load change from 2 MPa to 4 MPa. T_{diff} increases by a very much smaller amount in absolute terms but one which is still significant as a proportion of the 2 MPa value. The change in T_{diff} with load is reflected in Figure 5 which shows power absorbed by the low loss bearing against oil supply rate for both specific loadings. Results are plotted for the three speeds at which experiments were carried out. At each speed approximately 15 per cent more energy is absorbed when the load is doubled. At the lowest speed, 5000 r.p.m., there is little variation in power loss with change in oil flow. However as speed increases the change in power loss becomes more significant. At 10000 r.p.m. the decrease in power loss is approximately 15 per cent at both 2 MPa and 4 MPa loadings given a 40 per cent reduction in oil flow.

5.2 Reverse rotation with offset pivot pads

The major part of the experimental work described in this paper was carried out using bearings with thrust pads which had offset pivots. It is well-known that offset pivot pads run significantly cooler (1) than equivalent centre pivot pads given the same operating conditions. Because of this advantage offset pivot pads are a frequent choice for machinery in which the duty requirement calls for a single direction of rotation. However sometimes such machinery may be called upon to act in reverse. Example situations may be at machine rundown under pumped load or under certain emergency conditions. In these circumstances the reverse running capacity of offset pivot pads is important.

In order to establish the reverse running capability of the experimental bearing, it was operated in reverse for the standard 2 MPa and 4 MPa loads over the full range of speeds using one set of oil flows reduced by between 15 per cent and 30 per cent compared with the starting point of the experiments described above. The bearing performed satisfactorily in every way. The results of the reverse rotation experiments are given by Table 2 together with the results of the equivalent series of forward rotation experiments. It can be seen that the maximum pad temperature for reverse rotation was some 20°C higher than that for forward rotation. Nevertheless, the maximum pad temperature remained within acceptable bounds for such an extreme situation. The temperature difference between supply and drain did not increase with reverse rotation and consequently the power absorbed by the bearing shows no significant difference when operated in reverse.

5.3 Comparison with centre pivot pads

The advantage of offset pivot pads in comparison with centre point pads is borne out by the final set of results given in Table 2. The difference in the value of T_{max} for the offset pad in forward rotation and the centre pivot pad is about 10° for the 2 MPa loading and 20°C for the higher 4 MPa load. These results compare well with those quoted in reference (1) for the difference to be expected between offset and centre pivoted pads.

Examination of the values of T_{diff} for the bearing when fitted with centre pivot pads shows that there is almost no difference between the values recorded and those recorded for the bearing when the offset pads were fitted. Hence the energy absorbed is the same notwithstanding the much higher maximum pad temperature in the case of the centre pivot pad bearing.

It is interesting to compare the centre pivot pad results with those of the offset pad in reverse rotation. In each case as already indicated there is little or no difference in the values of T_{diff}. There is a difference however in maximum pad temperature. At the lower, 2 MPa applied load the centre pivot pad is 6°C to 12°C cooler than the offset pad operating in reverse. However when the 4 MPa load is applied, there is remarkable similarity between the results.

6. DISCUSSION

6.1 Effect of oil flow

Figure 5 shows the effect of decreasing oil supply rate on energy consumed by the bearing. In every case the reduction in oil supply leads to decrease in power absorbed. At 5000 rev/min there is little change when oil supply is reduced while at 10000 rev/min there is a significant decrease in energy loss given a substantial reduction in oil supply.

These results of the experiments with reduced oil flow demonstrate the tolerance of low loss bearings to a decrease in oil supply. Maximum pad temperatures certainly increased gradually with diminishing oil flow rate but not to unsafe levels. Temperature difference between supply and drain also increased but again not to a hazardous extent. Considerable experience with other similar bearings suggests that there should be no danger in operating for long periods with temperature differences up to 30°C provided oil is supplied at about 45°C and maximum pad temperatures do not exceed 130°C. Figure 5 shows that as oil supply was reduced power loss decreased. It is proposed that this is an indication that hydrodynamic lubrication persisted throughout the experiments. If this were not the case then there is likely to have been an increase in power loss to accompany the break up of the fluid film. The actual minimum oil supply level when this would have occurred as a precursor to failure exists at some point beyond the range of the present work. The nature of fluid film bearings makes the experimental or theoretical determination of final limits of this sort very difficult. What the experimental work has done, however, has been to increase knowledge of the operation of low loss bearings beyond what are present day normal boundaries for industrial applications.

6.2 Effect of pivot position

The results given in Table 2 show that for the experimental bearing and the operating conditions chosen, there is little significant variation in temperature difference between supply and drain resulting from change in thrust pad pivot position. In other words the energy absorbed by the bearing is the same whether offset pivot pads are used, in forward or reverse rotation, or if centre pivot pads are used.

The offset pivot pads, in forward rotation, offer values of maximum pad temperature which are substantially less than those of the centre pivot pads. The experiments carried out with the offset pivot pads in reverse show that they operate satisfactorily to provide acceptable values of T_{max} and T_{diff}. At the 4 MPa loading, the results for the offset pivot pads in reverse are no worse than for the centre pivot pads. This suggests that for all applications in which there is a predominant direction of rotation there is every reason to select offset pivot pads and take advantage of the reduced T_{max} in comparison with centre pivot pads. Centre pivot pad bearings remain an appropriate design choice for applications in which there is an equal requirement for rotation in both directions.

It was interesting to note the effect of applied load on the maximum temperatures of offset pivot pads operating in reverse in comparison with centre pivot pads.

We summarise that the performance of offset pads in reverse is aided by their ability to bend under the effects of load and temperature to form the necessary convergent hydrodynamic film in the changed direction of rotation. Evidently pad bending is related to the size of the applied load. In the case of the 4 MPA loading it seems that the deflection is such that the offset pad operating in reverse acts in a very similar way to a conventional centre pivot pad bearing.

6.3 Comparison with theory and previous experimental work

The minimum, unavoidable power loss in a hydrodynamic bearing is that due to shearing the fluid flims between pads and collar. In the case of a double thrust bearing the majority of losses are incurred on a loaded face but nevertheless a small and significant proportion is due also to the unloaded face.

Theoretical results were obtained for both faces for conditions corresponding to the current experiments using programs developed at Michell Bearings based on established design principles. The theoretical results for the 4 MPa load are shown on Figure 6. Comparison between theory and experiment shows good agreement and supports the idea that this design is one in which parasitic losses have been effectively eliminated. The considerable size of the parasitic losses in comparison with the fluid film losses can be seen from the theoretical energy consumption of a fully flooded bearing which is also shown on Figure 6. The flooded bearing includes both shear losses from the fluid film and parasitic losses.

A previous paper (6) included results for a "shrouded" energy saving bearing in direct comparison with the low loss bearing described in this paper. The results for the shrouded bearing, given on Figure 6, show how this design approach represents a significant improvement on the energy cost of a fully flooded bearing. Nevertheless on energy cost alone the shrouded bearing remains approximately 25 per cent more expensive than the low loss bearing for the same operating conditions.

7. CONCLUSIONS

The energy absorbed by the low-loss bearing conforms closely with the theoretical minimum loss expected due to shearing of the hydrodynamic films on the loaded and unloaed faces. This is considerably less than the loss which would be incurred by a fully flooded bearing and, as shown by previous experimental work, significantly less than that incurred by an equivalent shrouded bearing.

The low loss bearing has been shown to be extremely tolerant of major reductions in oil supply. It has been found that reducing oil supply by about 40 per cent resulted in the energy absorbed by the bearing reducing by up to 15 per cent. The greatest reduction occurred at the highest speeds of operation.

The ability of offset pivot pad bearings to run satisfactorily in reverse has been demonstrated and a comparison with an equivalent centre pivot pad bearing has been carried out. It was found that the energy absorbed by the low loss bearing is independent of thrust pad pivot position.

In conclusion it is suggested that the offset pivot pad, low loss bearing operating with a reduced lubricant supply represents the optimum energy efficient choice currently available to designers for all high speed operations in which there is a predominant direction of rotation.

8. ACKNOWLEDGEMENTS

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Table 1 Dimensional information for the low loss experimental bearing

Number of Pads per Thrust Face	8
Inside Diameter	114.3 mm
Outside Diameter	228.6 mm
Surface Area per Thrust Face	25994 mm²
Thrust Collar Diameter	231.7 mm
Shaft Diameter	100 mm
Axial Clearance in Bearings	0.7 mm
Area of Drain	7650 mm²
Number of housing inlets per thrust face	2
Diameter	20 mm
Number of orifices per pad	2
Diameter	3 mm

Table 2 Operating conditions and results for the test bearing under forward and reverse rotation when fitted with offset pivot pads, and when fitted with centre pivot pads

Speed	Specific	Total Oil Flow	Forward T_{max}	T_{diff}	Offset Pivots T_{max}	T_{diff}	Centre Pivots T_{max}	T_{diff}
rev/min	MPa	l/hr	°C	°C	°C	°C	°C	°C
5000	2.0	1920	90	13.5	106	14	100	14.5
7500	2.0	2840	95	18.5	117	19	107	19.5
10000	2.0	3960	100	21	122	23.5	110	24
5000	4.0	2240	106	13.5	113	14	116	15.5
7500	4.0	3300	116	19	128	18	130	19
10000	4.0	4520	115	21	138	21.5	137	22

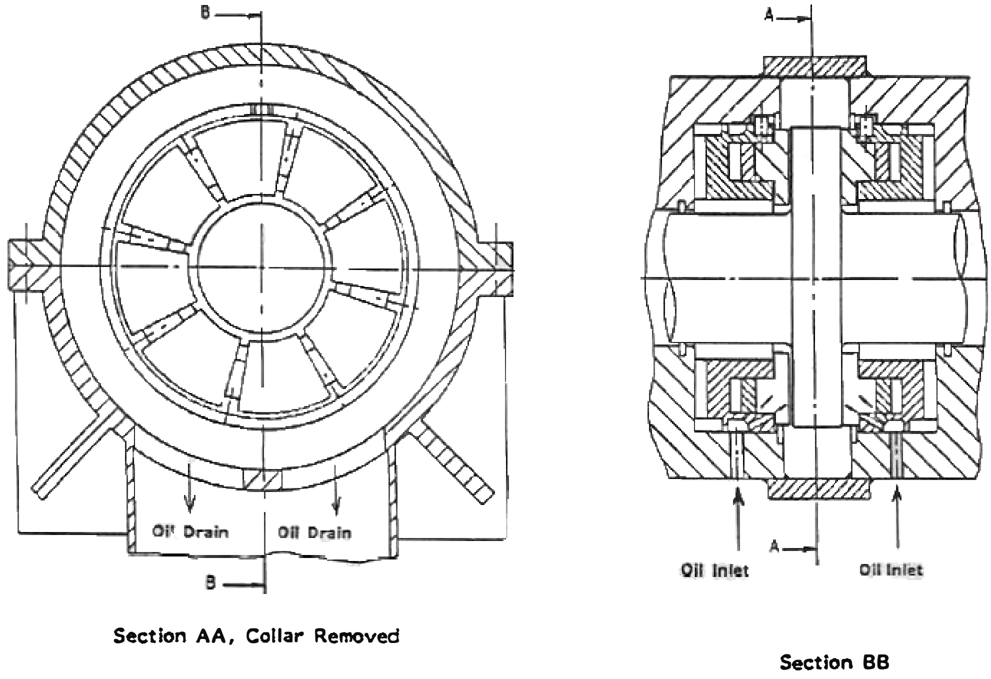

Fig 1 Experimental bearing

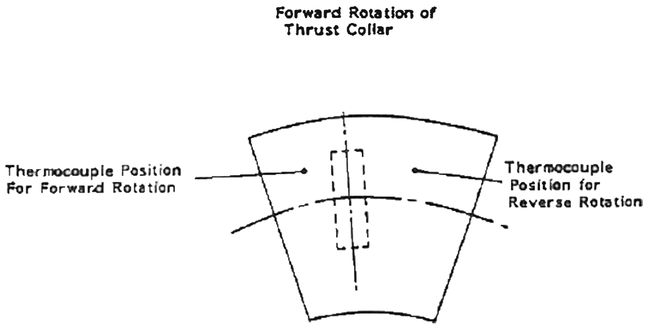

Fig 2 Sketch showing location of thermocouples measuring Tmax for forward and reverse rotation

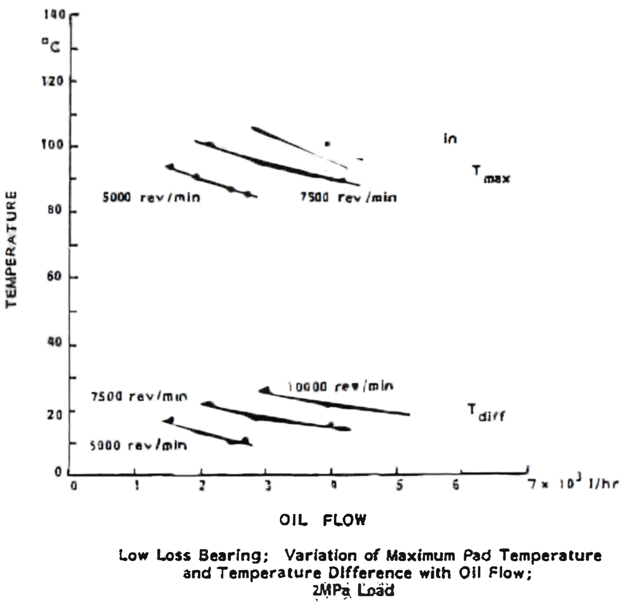

Fig 3 Variation of maximum pad temperature and temperature difference with oil flow; 2MPa load

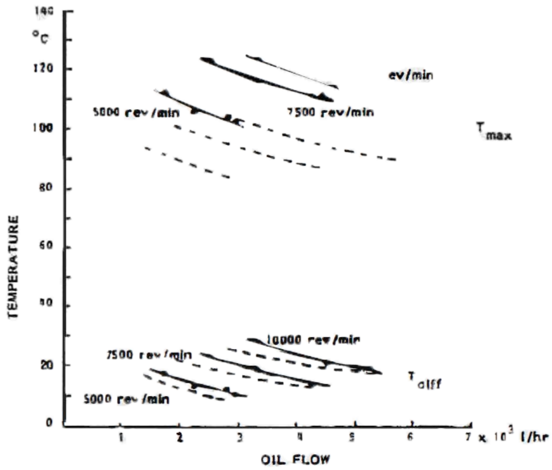

Fig 4 Variation of maximum pad temperature and temperature difference with oil flow; 4MPa load (--- 2 MPa results from fig.3)

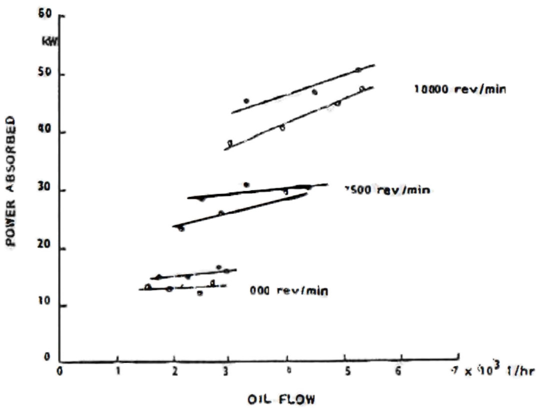

Fig 5 Variation of power absorbed by bearing with oil flow; o, 2MPa load; ●, 4MPa load

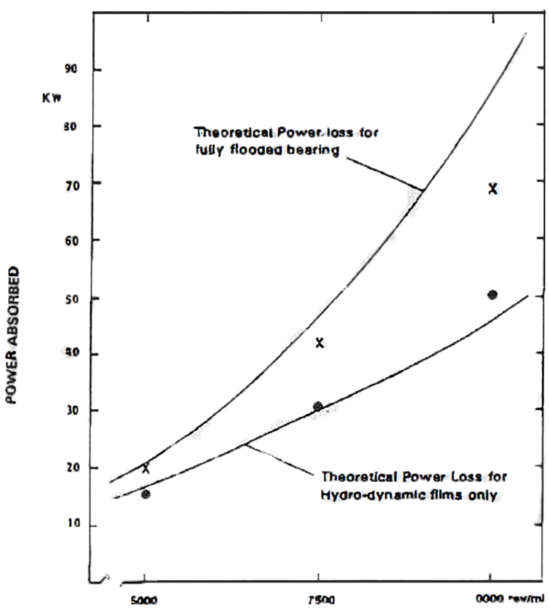

Fig 6 Variation of power absorbed with speed; comparison of experimental and theroretical results; 4MPa load (Experimental results; ●, low loss bearing; x, shrouded bearing, ref.6)