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Large hydrodynamic thrust bearing – comparison of the calculations and measurements

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Abstract: Hydrodynamic thrust bearings used to carry axial loads in heavily loaded shafts of water power plants hydro turbines can reach outer diameters even exceeding five meters. In such large objects scale effect could be observed. According to this allowable bearing specific load assuring bearings safe operation has to be decreased leading to increased thrust bearing dimensions. This effect is caused by excessive thermal deflections of bearing pads which change significantly oil gap geometry and in consequence decreases bearing load-carrying ability. Design of hydrodynamic thrust bearing of large dimensions seems to be a demanding engineering challenge, and additional difficulty comes from limited possibilities of experimental testing of these systems due to high costs. Theoretical investigations carried out with the use of specially developed computer models remains a feasible alternative for experimental research. But the accuracy of the models is not often directly validated, because of the lack of appropriate experimental data coming from large objects.
In this paper results of calculations carried out for a large hydrodynamic thrust bearing are shown and compared to measurement data obtained at bearing commissioning stage. Pad temperatures profile sliding surface, oil pressure in hydrodynamic gap and film geometry are compared to the measured values. According to the presented comparisons some conclusions are drawn concerning the accuracy of models used to predict large thrust bearing performance.

1 Introduction

Hydrodynamic thrust bearings used to carry vertical loads as in heavily loaded hydrogenerators are critical machine components. They have to operate reliably in severe conditions: high sliding speeds, extreme loads often at increased temperature of the surroundings. Dimensions of such bearings depend mainly on shaft strength which is limiting their inner diameter and allowable specific load. Hydrodynamic thrust bearings used in hydroturbines reach outer diameters of more than five meters. As it was shown by Ettles in [1], enlarging thrust bearing dimensions decreases simultaneously allowable bearing specific load assuring its safe operation. This effect is known as "scale effect" and is caused by excessive thermal deflections of bearing pads, which change significantly oil gap geometry and in consequence decrease bearing load-carrying capacity. Several bearing designs have been developed in order to reduce thermal deflections, most common being large ring shaped supports analysed first in [2] and spring mattress pad supports [3]. Other designs include double support and double layer pads [4], double row bearings [5] or even water cooling of the pads, described by Kuhn [6] and Chambers and Mikula [7].
Another important problem of large thrust bearings is uneven load sharing between bearing pads [8] - also in this case various remedies have been proposed, summarized in [9]. The third important design problem of the thrust bearings of vertical shaft hydrogenerators is the need to carry load even at very low rotational speeds occurring during starts and stops, which is not possible in hydrodynamic lubrication regime [10]. This problem is usually solved by applying hydrostatic jacking systems comprising high pressure pumps, appropriate installation and pockets machined in the bearing pads [11]. Such systems are activated during start up and slow down periods of machine operation

providing hydrostatic assistance of bearing operation. A more detailed review of the main problems of large thrust bearings and proposed designs can be found in [9].

Because of the above mentioned problems the design of large hydrodynamic thrust bearings seems to be a very demanding engineering challenge especially as experimental investigations of new designs are limited due to their dimensions and high costs. Theoretical investigations of the bearings remain feasible alternative for experimental research. Calculations are usually carried out with the use of specially developed computer models. It is possible to predict thrust bearing pad performance, taking into account phenomena occurring during its operation. However, to obtain solution some additional, usually unknown, boundary conditions have to be assumed. They can change results of prediction significantly, as it was shown in [12] or [13]. On the other hand many models were verified with success with the use of data obtained for experimental bearings of small dimensions [14]. It is difficult to confirm all assumptions without appropriate comparisons to reliable measurements in larger objects. Changes in accuracy of the predicted characteristics for smaller and larger bearings are the effect of different scales of phenomena occurred in bearings with different dimensions (as for example for heat exchange or thermal pad deformations mentioned above). In the literature one can find descriptions of the calculation models devised for large thrust bearings and calculations performed with the use of such models (eg. [14], [15]). Unfortunately, only few direct comparisons between theory and experiments concerning large thrust bearings have been published so far. One of them is the work of Yuan et al. [16] in which a comparison of the calculations results and measurements for a thrust bearing of 1.17 m outer diameter was reported. The comparison showed significant differences in predicted bearing characteristics and measured data - as large as 5°C in case of temperatures and up to 7 MPa (100%) in evaluating film pressure.

In a recent paper by Huang et al. [17] the results of calculations and measurements of a symmetrically supported thrust bearing of OD equal to 2.5 m in a huge test rig in Harbin (China) were described. The authors compared measured temperatures, film thickness and pressure profiles with the results of their 3D TEHD calculations with the use of own FD film model combined with ANSYS structural model. Although the global results show very good agreement between calculation and experiment (~7 μm difference of minimum film thickness, 0.1 MPa for maximum pressure and 5.3°C in maximum temperature), study of the distribution of film thickness, temperature and pressure on the pad surface reveals much larger differences in location of maxima and profiles of the respective fields. This is shown in Fig. 1, in which results for film thickness are illustrated. According to calculations film profile is of regular, almost spherical convex shape, with inlet film thickness of less than 100 μm and minimum of 32.4 μm shifted upstream from the trailing edge. According to measurements and interpolation of measurement results with a method which was not described, the pad film profile is much less regular. At the inner corners the pad is depressed, so that film thickness reaches 200 μm at the inlet and 160 μm at the outlet and minimum film thickness of about 25 μm is located at the trailing edge. The differences should not be attributed to calculation errors only, as during operation in a machine (and also in a large test rig) a real bearing performance is the effect of many factors which are beyond control of the operator and measurement equipment. In reality load is most probably not distributed evenly between the pads and there are machining tolerances comparable to minimum film thickness.

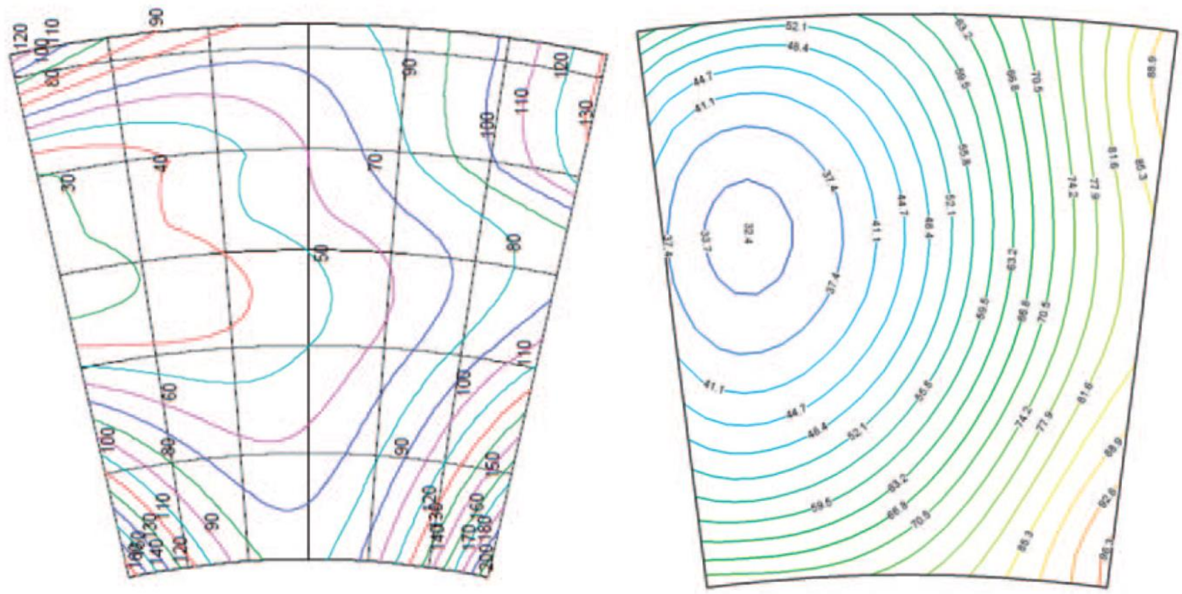

Fig. 1. Film thickness contours [μm] - measured - lefthand side graph and calculated - righthand side graph (direction of sliding from right to left) [17]

In a paper by Pajączkowski et al. [18] calculations of bearing characteristics during start up of a turbine were carried out with the use of FSI (Fluid Structure Interaction) technique. Calculated temperature changes showed good agreement with the results of measurements in a real turbine, but the temperature was the only parameter available for comparison.

2 Objective of the research

The aim of this work was to carry out direct comparisons between results of calculations and detailed measurements of the operational parameters of large tilting pad thrust bearings. As it was mentioned above, only a small number of studies dealing with this subject was published. The main assumption taken for this research was, not to use any measurement data to carry out simulations, especially not for defining necessary boundary conditions. This was intended to illustrate possibilities of correct prediction of bearing performance without any *a-priori* knowledge of its real performance.

Calculations were completed with the use of two different thrust bearing 3D Thermo Elasto Hydrodynamic Models. The first model used in this research was devised by D. Souchet [19] and then modified by M. Fillon (further referred to as Fillon model). The second model developed by Gdansk University of Technology research team is based on Fluid Structure Interaction procedure (further referred to as FSI procedure). The results of calculations obtained with the use of both models were compared to the results of measurements carried out by ALSTOM Hydro in Itaipu power plant hydroturbine large thrust bearing, during commissioning of the machine. During this measurements oil pressure, pad and runner body temperature and oil gap thickness were measured. Comparison of those parameters was presented, and on its basis conclusions were drawn concerning accuracy of models used in this research and identified problems with large thrust bearings properties simulation.

3 Hydrodynamic thrust bearing - object of the analysis

The main object of this research, a large thrust bearing pad with a support system and its main dimensions are shown in Fig. 2.

The analysed bearing consists of 16 pads (pad angle β=18.5°) supported on annular ring shaped support. Main goal of such design is to compensate thermal pad deformation by opposite elastic deformations, which occur in the pad area inside annular ring of the support. The bearing is unidirectional, with off-set pivot. Thrust bearing pads are steel and their thickness is 230 mm including babbitted sliding surface of 4 mm thickness. Total thrust load applied to the thrust bearing measured during machine operation is equal to 27.7 MN (specific bearing pressure p_m = 2.6 MPa), sliding velocity v_m = 20.4 m/s (at mean bearing radius). Thrust bearing is equipped with hydrostatic jacking system to facilitate start ups and shut downs of the machine. Main thrust bearing data concerning its operational conditions are gathered in Table 1.

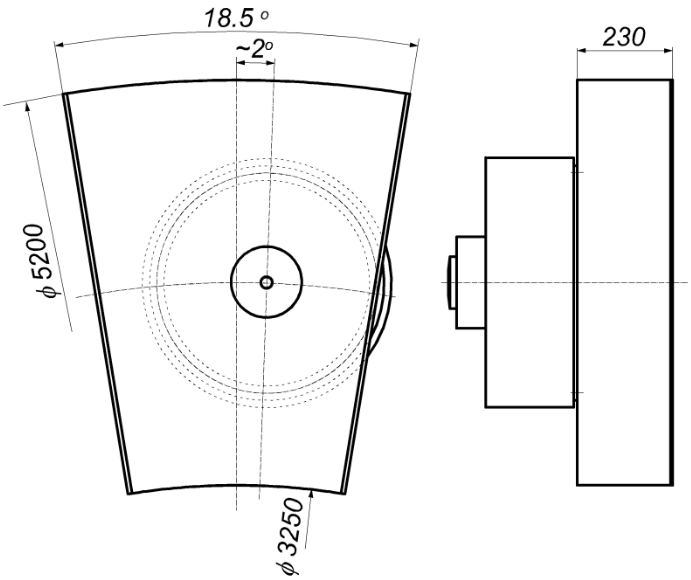

Fig. 2. Object of the research, analyzed thrust bearing pad with its main dimensions

Table 1. Operational conditions of analyzed thrust bearing

Quantity	Unit	Value
Rotational speed	[rpm]	92
Total bearing thrust load	[MN]	27.7
Oil bath temperature	[^{o}C]	39.3

3.1 Monitoring of bearing performance

During installation and commissioning the hydroturbine was subjected to several tests. Test were carried out for various operating conditions, the axial load, rotational speed, high pressure oil system on/off status and machine power output were changed. An important part of the field tests was monitoring of thrust bearing properties. To complete this, many sensors were installed in the runner and bearing pads (thrust and guide) to measure all important parameters of the bearing system from the point of view of its safe operation. These were primarily: temperatures in the pads and in the runner, pressures in hydrodynamic film and hydrodynamic oil gap thickness.

In this research, measurements for rated load and speed conditions were used to carry out comparisons with the theoretical results obtained with the use of TEHD calculations.

The main sensors arrangement is shown in Fig. 3. Single bearing pad was equipped with 25 temperature sensors placed in two layers, one layer close to the bearing sliding surface (symbols with the letter “s” - surface in the subscript) and the other close to the pad bottom (symbols with the letter “b” - bottom in the subscript). Additionally in the collar, three pressure sensors (marked pr1 to pr3) and three distance sensors (marked hr1 to hr3) were placed to monitor oil gap pressure and oil film profiles. In Fig. 3 dotted lines indicate paths of pressure and distance sensors.

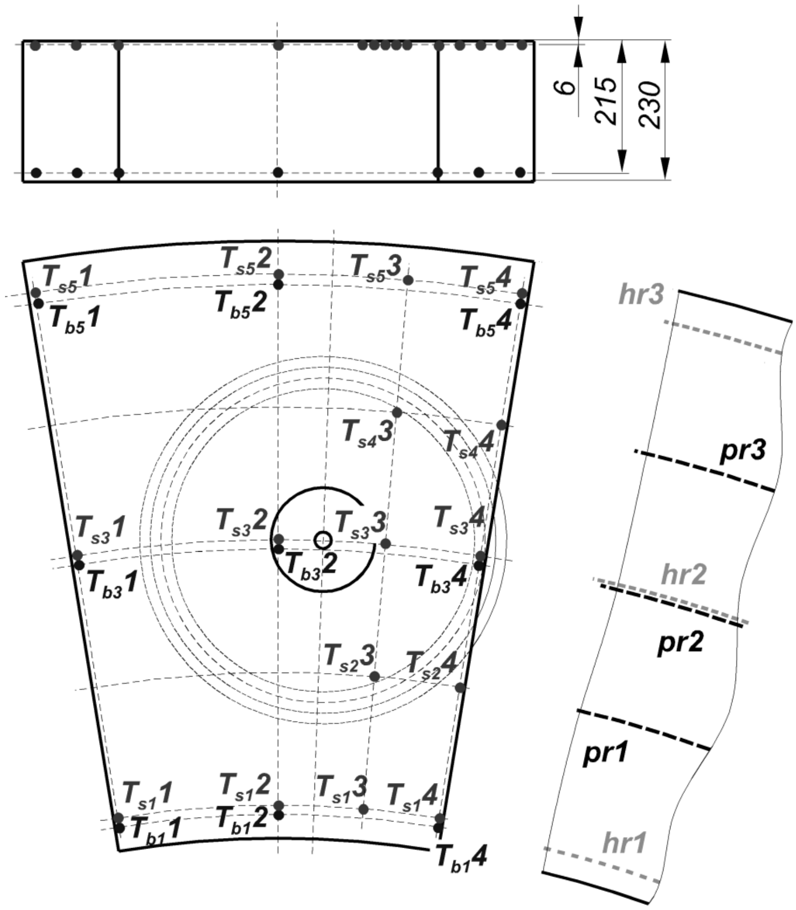

Fig. 3. Sensors arrangement in analyzed thrust bearing

Measurements were carried out during commissioning of the hydrogenerators in Itaipu power Plant which took place in 1980's. Exact evaluation of the measurement errors results from sensors accuracy is difficult after many years, however some technical details of the monitoring system used for Itaipu bearing were described in the report [20]. Thermistors were used for temperature monitoring, eddy current type displacement sensors for oil gap geometry and piezoelectric pressure transducers were applied for oil pressure measurements. In the report, the accuracy of the displacement sensors is assessed as ± 10 μm, and the pressure transducers as 1% of the full scale of 25 MPa, i. e. 0.25 MPa. As to the data acquisition accuracy, important for the transducers installed in the thrust collar - the data was acquired with a frequency of about 1500 Hz, allowing for approximately 50 data points at a single sweep over one pad. It was also possible to assess the measurement accuracy, with the use of data acquired by reserve sensors.

Reserve sensors were installed both, in the pads and runner with the intention to avoid loss of data in case of damage of any sensor and to control influence of shaft angular position and differences between pads on the measurements. The discrepancy of the parameters measured by main and reserve sensors was probably caused by external sources, as for examples: limited sensor accuracy or zero adjustment errors and in the case of sensors installed in different pads also the uneven load distribution among the pads.

In addition to the main instrumented pad shown in Fig. 3, another, reserve pad was also instrumented with 4 thermocouples situated in the hottest parts of the pad. Similarly, reserve sensors of pressure, film thickness and temperature were installed in the runner.

In Fig. 4 comparison of the measurements collected with the use of main and reserve sensors at the bearing mean diameter are presented. Compared oil pressures and gap geometries were obtained when sensors moved over the instrumented pad (equipped with thermocouples showed in Fig. 3) and under normal condition of bearing operation (axial load and shaft speed as in Table 1). In Fig. 4, difference Δp between pressure sensors (main and reserve) and distance sensors measurements difference Δh are plotted. Similar comparison was also done for the inner and outer radii sensors, but the results of these measurements is not presented in the paper for the sake of consistency.

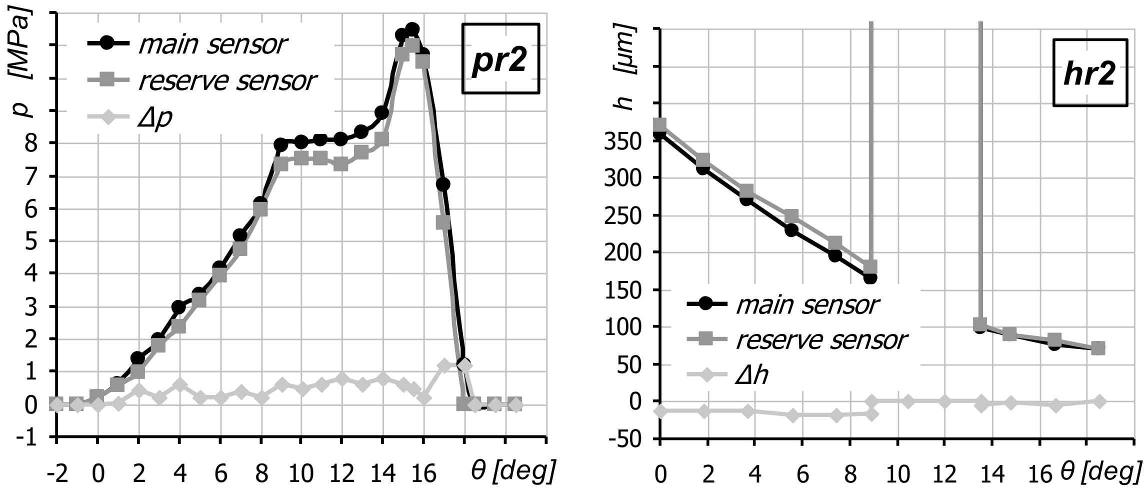

Fig. 4. Comparison of the measurements obtained with the use of main and reserve sensors installed in the runner: measured oil film pressures (left column) and measured oil gap thickness profiles (right column). Difference between the sensors signals as a function of pad angle was also marked as Δp (for pressure sensors) and Δh for oil distance sensors

Concerning pressure sensors, measured trends are repeatable but with some noticeable deviations. Maximum value of difference Δp is equal to about 10-12% of maximum value of the oil pressure. The biggest difference Δp for all of the analyzed profiles can be observed at the trailing edge of the pad. It can be caused by local negative pressure which usually appears at trailing edges of pivoted thrust bearings pads. At the rest of angular extent of the pad, pressures measured with the use of main and reserve sensors, differed from each other not more than 0.5-0.7 MPa, which is approximately twice as much as the error of the transducer.

In case of distance probes, the biggest difference between main and reserve sensors can be observed at pad inlet (up to mid pad angular extend), while the smallest at the pad outlet. Maximum value of distance sensors signals difference Δh was found for hr2 sensor position, and was equal to ~18 μm, which is in good relation with an error of eddy current proximity probes estimated in an internal report concerning measurement setup. In case of this sensor, lack of signal was observed when it was moving over hydrostatic recess.

Maximum difference of temperature measurements between main and reserve sensors in the pads was equal to 0.7-7.5°C, while the difference between main and reserve temperature sensors installed in the runner did not exceed 0.6°C, but the latter results are not presented in the paper. One of the identified reasons of the differences between temperatures of the main and reserve pads is the variation of loads acting on particular pads. This discrepancy is known because the load on each pad was also monitored. According to this measurement it was equal to ±10%, but there are also other reasons in pad support or geometry (manufacturing inaccuracy) since the temperature distribution, e.g. location of the maximum is different on both pads.

To conclude, on the basis of comparison of the measurements results between main and reserve sensors, uncertainty can be assessed as not exceeding: ~0.7 MPa for oil pressure, ~20 μm for distance sensors and ~7.5°C for pad temperature sensors and 1°C for the runner.

4 Thrust bearing performance prediction

For the needs of the presented research two 3D thermoelastohydrodynamic models (TEHD) developed for thrust bearing performance prediction were used. In this paragraph their short description were presented.

4.1 FSI procedure

TEHD calculations with the use of Computational Fluid Dynamics (CFD) and FSI procedures are recently becoming more and more popular method of fluid film bearing calculations ([21], [22], [23], [24]). The authors of this paper have also started to apply this method for calculations of tilting pad thrust bearings. In general, FSI is coupling Finite Element Method (FEM) software environment for bearing structure calculations and Computational Fluid Dynamics (CFD) software environment for oil flow calculations. The main idea is that the model comprises both, fluid and structure models,

complementary to each other. Both areas of TEHD analysis are connected internally, and during iterative solution between both fields of calculations (and both programs – CFD/FEM) loads are exchanged automatically. Inside FEM software bearing structure (pad, support and runner) is modelled while in CFD software surrounding oil and oil gap is described. Loads are transferred between both models through interface surfaces. Deformations (thermal and elastic) of the bearing pad and runner change oil gap geometry. In turn oil pressure and heat flux generated in the oil gap are carried out from fluid flow model an applied to solids as boundaries. One of the characteristic features of FSI calculations is the fact that heat exchange coefficients on fluid solid boundaries are not input data assumed at the beginning of the calculations, on the contrary they are automatically evaluated in the course of calculations and can be different in different locations of the fluid solid interface, as shown in [25]. More details of FSI procedure used in these calculations can be found in [26] and [18] One detail of the modelling, which should be mentioned is the fact that no hydrostatic recess was modelled on a pad surface, because of the difficulty in obtaining convergence, encountered once the recess had been included in the model. A literature study on this issue revealed that it is a common practise, as the only large thrust bearing calculations in which hydrostatic recess was included in pad geometry were the works of Pajączkowski et al. [26], and Heinrichson PhD dissertation and papers [27], [28]. A PhD and papers by Heinrichson were the first systematic study of the influence of hydrostatic pockets on bearing performance. In many other papers the existence of the pocket is neglected in the calculations of hydrodynamic regime of operation - Ettles et al. in their paper [11] studied bearing operation in hydrostatic regime, but it seems that no depression was geometrically modelled in the pad sliding surface. Jiang et all in their recent paper [15] and Huang et al [17], as well, did not include any geometrical discontinuity on the surface.
FSI analysis was applied before, with success in several researches concerning thrust bearing properties evaluation carried out at Gdansk University of Technology in cooperation with ALSTOM company. However it should be mentioned that in this research for the first time FSI procedure was adopted for TEHD calculations of thrust bearing of such a large size and considerable specific pressure.

4.2 TEHD model of M. Fillon

Thrust bearing TEHD model used in this research was built by D. Souchet [19] and then modified by M. Fillon. It allows to carry out TEHD analysis for an individual bearing pad (with the assumption of equal load sharing by each pad). TEHD analysis of the bearing pad takes into account influence of strongly connected phenomena occurring during bearing operation on its performance. The most important of them are: hydrodynamic effect of the fluid flow in the oil gap, elastic pad deformations under hydrodynamic oil pressure, heat generation and its distribution in the oil and bearing elements, and thermal deformations due to temperature differences caused by heat flow. It is possible to include influence of collar deformations on bearing performance, but in this research they were not considered. Reliability of Fillon thrust bearing TEHD model was checked with the use of experimental data especially for small thrust bearings. Some results of those researches and additional information concerning details of the model can be found in [13], [29] and [30]. Calculations results obtained with the use of Fillon model were obtained with the assumption of ambient temperature equal to 39.3^{o}C and oil convection coefficient of 750 W/kgK for all bearing pad surfaces. Similarly to most other papers the hydrostatic recess was not modelled on the surface of the pad.

5 Comparison of measurements and calculations of bearing performance

According to the assumption, the results of TEHD calculations were obtained without any knowledge about the results of measurements, thus it could be a good indication how close theoretical prediction of bearing performance with the use of modern tools to the measured data can be. Selected results of measurements and calculations are compared below. They comprise temperatures in the pad, oil film pressure and oil gap geometry. They show general trends observed in the results obtained with the use of both theoretical models.

5.1 Pad temperatures

Calculated tangential temperature profiles at pad sliding surface and pad bottom surface are compared to measured data for pad mean radius in Fig. 5. In Fig. 6 radial pad sliding surface and pad bottom surface calculated profiles are compared to the measured data for pad outlet. Maximum values of calculated and measured temperatures at the bearing pad sliding surface are gathered in Table 2.

Analysing results shown in Fig. 5 one can notice, that temperatures predicted for sliding surfaces with the use of Fillon model are higher at the inlet to the pad than temperatures predicted with the use of FSI procedure. This changes after crossing mean pad angle, and finally higher temperatures were predicted with the use of FSI procedure.

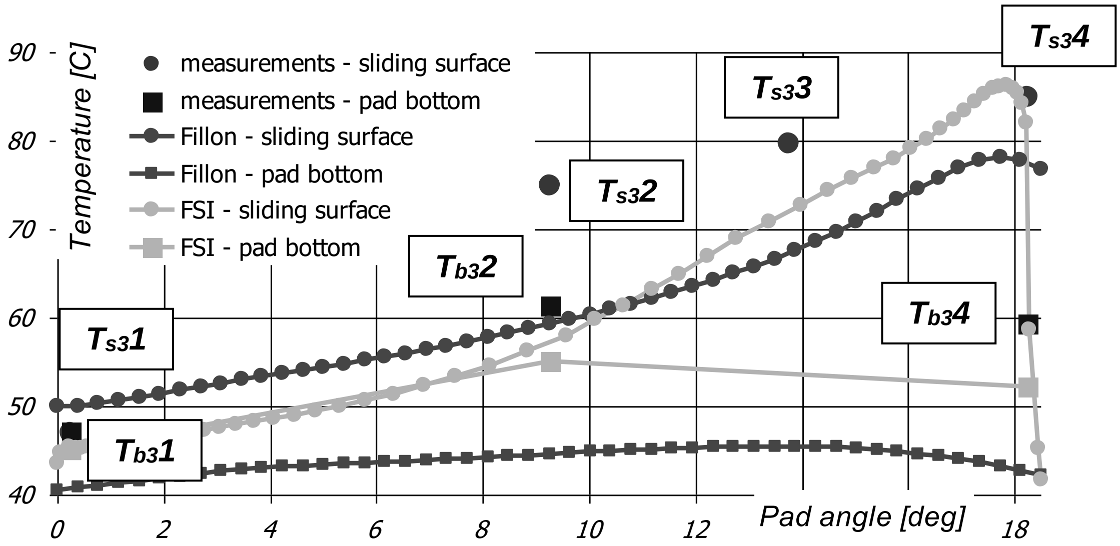

Fig. 5. Pad temperatures profiles along the circumference at mean radius R=2112.5 mm, comparison of the measurements and calculation results (sensors positions see Fig. 3)

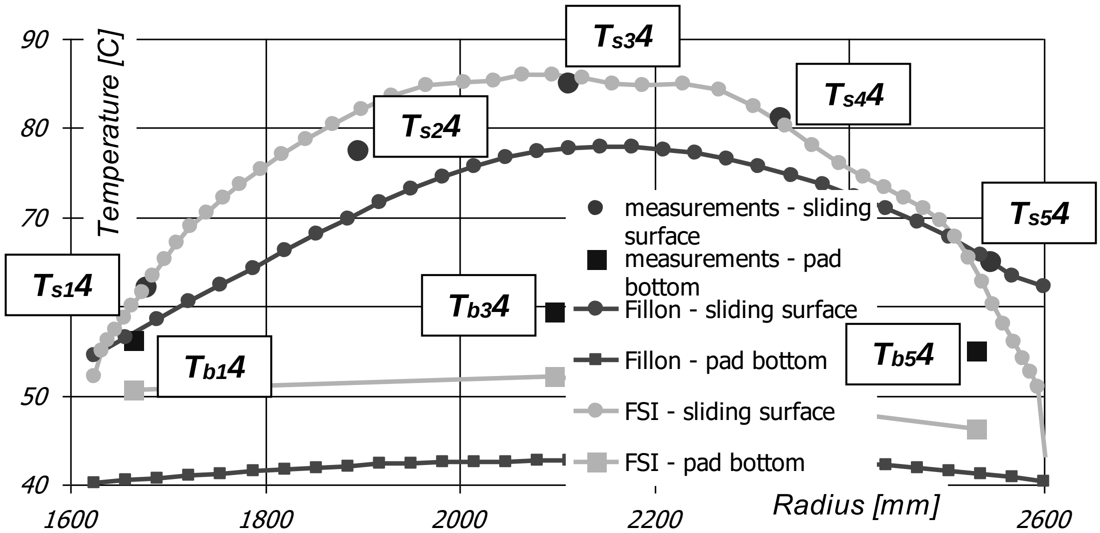

Fig. 6. Radial pad temperature profiles, comparison of the measurements and calculation results (sensors positions see Fig. 3) for pad outlet (β=18.25°)

Table 2. Calculated and measured values of maximum pad temperature

	FSI procedure	Fillon model	Measurements
T_{max} [°C]	86.3	78.3	85.0

Comparing measured temperatures of the sliding surface to calculated values it should be observed that there are no clear rules. At the inlet and outlet of the pad, measured values are close to the calculated ones, while at the centre of the pad measurements are generally higher (even by about 15°C - see sensor $T_{s3}2$ in Fig. 5 than calculated temperatures). At the inlet to the pad, temperatures predicted with the use of FSI procedure are very close to the measurements.
Comparing results obtained for pad bottom surface one can observe that temperatures predicted with the use of FSI procedure are closer to measurements then those obtained with the use of Fillon model. Temperatures of the pad bottom calculated with the use of Fillon model are very low in

comparison to measured values. This is caused by arbitrary assumed relatively high value of heat exchange coefficient in Fillon model. This led to excessive heat transfer across pat thickness to the oil, and in consequence to lower calculated pad bottom surface temperatures.

Analysing results of radial temperature profiles collected in Fig. 6 the same tendency can be observed as in tangential profiles for pad bottom temperatures. They are predicted very low with the use of Fillon model and temperatures obtained with the use of FSI procedure are closer to the measurements. Quite good agreement of calculations of FSI procedure results and measurements could be observed for pad outlet radial temperature profiles (see Fig. 6), as well as at the inlet profile not shown in the graph. Better agreement of FSI results with measurement (than that of Fillon model) at the pad borders - outlet, inlet and pad bottom probably illustrates the fact that automatic evaluation of heat exchange coefficients within the FSI procedure reflects the real heat exchange conditions more accurately than arbitrarily assumed coefficients of a constant value.

Comparing maximum temperature values presented in Table 2 one can see that better agreement of results of calculations and measurements was obtained for FSI procedure, but anyway, as it was shown at the figures, profiles of calculated temperatures differ much from the profiles obtained from measurements.

5.2 Oil film thickness

Calculated oil gap thickness profiles are compared to measured data for mean radius in **Błąd! Nie można odnaleźć źródła odwołania.**. Extreme (i. e. minimum) values of calculated and measured oil film thickness in the oil film gap are collected in Table 3.

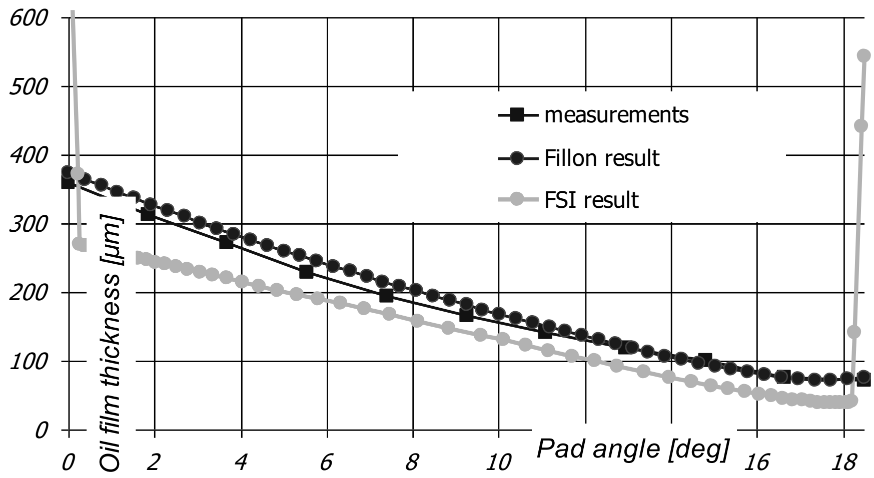

Fig. 7. Oil gap profiles along the circumference at mean radius R=2112.5 mm, comparison of the measurements and calculations results (sensors positions see Fig. 3) at sensor hr2 position

Good convergence of measured data and calculations can be observed for results of Fillon model especially for the mean radius (Fig. 7) and outer radius, not shown here. Worse agreement was obtained for inner pad radius (also not shown), where both models represent the same predictions of minimum film thickness while at the inlet to the film FSI procedure shows slightly smaller gaps than Fillon results. Generally at the inner bearing radius both models underestimate the oil gap thickness. For all analysed radii, the smallest values of oil film thickness are calculated in FSI procedure and the difference were larger than estimated measurement uncertainty. On the other hand, measurement of film thickness is a very difficult task and especially in a real large machine various factors may have affected its accuracy, uneven load distribution and different temperature distribution causing different thermal deflections.

Table 3. Calculated and measured values of minimum oil film thickness

	FSI procedure	Fillon model	Measurements
h_{min} [μm]	36.5	71.8	70.0

5.3 Oil film pressure

In Fig. 8 calculated profiles of oil film pressure are compared to measured data for mean radius of bearing pad. The influence of hydrodynamic recess presence on measured pressure data can be clearly observed. Extreme values of calculated and measured oil film pressure noticed in the oil film gap are shown in Table 4.

The calculated pressure profile differs much from the measurement result especially because in both calculation models the hydrostatic pocket was not included, while a plateau in the area of the pocket can be noticed in measurement results. Analysing results presented in the figure one can also observe that maximum oil film pressure calculated with the use of FSI procedure is moved closer to the trailing edge of the pad. Pressures calculated with the use of Fillon model were lower at mean radius comparing to measurements and FSI results. At the mean radius smaller predicted pressures are probably the effect of not including influence of hydrostatic recess in the theoretical analysis.

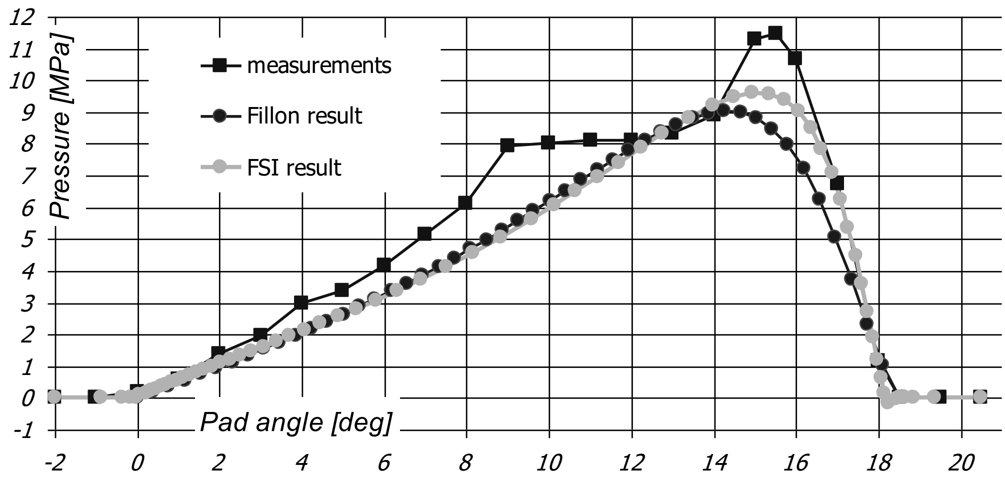

Fig. 8. Oil pressure profiles along the circumference at mean radius R=2112.5 mm, comparison of the measurements and calculations results (sensors positions see Fig. 3) at sensor pr2 position

Table 4. Calculated and measured values of maximum oil film pressure

	FSI procedure	Fillon model	Measurements
p_{max} [MPa]	10.2	9.02	11.8

6 Conclusions

Theoretical evaluation of the detailed operational characteristics of a large thrust bearing still seems to be a difficult task. Both, examples from the literature and the results presented in the paper show that discrepancies between the theoretical results and measurements are considerable. The differences can be attributed to simplifying assumptions in the calculations and measurement inaccuracy, as well. In the calculations, in fact an ideal single bearing pad is calculated, one of the simplifications was omitting hydrostatic recess, due to encountered calculation convergence problems. In turn, in a real bearing, despite equalizing mechanisms the load distribution between pads is not even, in addition, the real bearing pads show manufacturing imperfections, so that the measurements carried out in a selected pad reflect the parameters in this particular pad. Another problem is a certain level of measurement uncertainty, which was also discussed in the paper.

Discussing the comparison between bearing performance predictions obtained with the use of both theoretical models and experimental data in detail, it should be noticed that temperature distribution measured in the bearing pad was different from distribution based on the results of simulations. In case of Fillon calculations, temperature of the sliding surface were too low close to the trailing edge, and a little too high close to the leading edge, while temperatures of the bottom of the pad were too low over the whole pad bottom surface. Temperatures obtained with the use of FSI procedure show similar discrepancy. The largest differences of predicted pad sliding surface temperatures for both models in comparison to measured data were observed in the hydrostatic recess area (pad centre). Better agreement of calculations and measurements for pressure profiles could be found in the results obtained with the use of FSI procedure (for all available circumferential profiles). Very good agreement

of measurements and calculation results for oil film thickness was obtained for predictions with the use of Fillon model. Only the calculation results of the oil gap profile for inner bearing radius differ from measurements significantly.
It seems that some work should be done to find appropriate boundaries for both models, which shift results of theoretical predictions closer to the measurement results, however relatively good agreement of temperatures at the pad peripheries (inlet to the film and pad bottom) in the FSI model shows that the CFD procedure is efficient in simulating heat exchange phenomena around the pad.

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