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EFFECT OF THE PRESENCE OF THE LIFTING POCKET ON THE THD PERFORMANCE OF A LARGE TILTING-PAD THRUST BEARING*

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Abstract

Hydrostatic assistance is a commonly used method to improve limited load carrying ability of tilting-pad thrust bearings during transient states of operation of the vertical shaft hydro-generators. Despite of special hydraulic equipment (as pumps, valves etc.), it also requires manufacturing of special recesses/pockets at the pad sliding surfaces, into which oil is injected under high pressure. It allows to lift the rotor before start-up of the machine and form a hydrostatic film between the pads and the collar. There is a quite wide variety of geometry of recesses (shape, depth, size) met in practical large bearing applications. The presence of a hydrostatic pocket (usually located in the sliding surface above the pivot area, where thin film, high oil pressure and temperature are observed) affects bearing performance under hydrodynamic operation. In theoretical researches, there is an almost common practice not to include hydrostatic recess in THD or TEHD analysis. This is probably due to the problems with obtaining solution for oil film geometry with pocket, with pocket depths order of magnitude larger than gap thickness. In this paper, an attempt was taken to study the effect of the lifting pocket on THD performance of a large tilting-pad thrust bearing of Itaipu power plant. Bearing performance was evaluated including recess shape for several cases of its depth. The results show, that hydrostatic recess changes calculated bearing properties quite significantly, especially in the vicinity of the pocket.

Keywords: Lifting pocket; Tilting-pad thrust bearing; THD regime; Numerical simulations.

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1 INTRODUCTION

A big variety of hydrostatic pocket geometries exist in large thrust bearings - they seem to be designed so as not to deteriorate the bearing surface too much, but on the other hand to provide efficient oil supply and lifting capability, which depends on the pocket size. Thus a pocket diameter is also a compromise between the requirements of hydrostatic and hydrodynamic operation and to some extent a tendency of decreasing pocket size may be observed, accompanied by increased pressures in the hydraulic system. Some data from the literature showing design parameters of hydrostatic jacking systems of various hydrogenerators are presented in Table 1.

Table 1. Comparison of hydrostatic systems of various hydrogenerators

Bearing outer / inner diameters (m)	Rotational speed (rpm)	Flow/pad (l/min)	Pocket diameter (mm) (*% of pad width*)	References
1.80 / 0.80	187.5	1.00	75 (*15.0%*)	HEPP Dychów [1]
1.30 / 0.85	600	1.35	75 (*33.0%*)	PSPP Porąbka Żar [2]
2.30 / 1.30	500	1.42	85 (*17.5%*)	HEPP Kopswerke [3]
2.00 / 1.00	500	5.00	125 (*12.5%*)	[4]
5.20 / 3.25	92	4.70	170 (*17.5%*)	HEPP Itaipu [5]

Various recommendations exist also as far as film thickness and oil flow is concerned. According to [6] assumed film thickness should be proportional to square root of pad size and for the size of Itaipu power plant thrust bearing it should be 45 µm. Abramovitz in [7] gives a more general recommendation of 50-250 µm. The assumed film thickness has a great influence on the system design, as the required pump output is proportional to the third power of the film thickness, so in order to increase film thickness twice an eight-times increase of the pump output is necessary. According to the style of recesses there are also various forms - e.g. shown in the examples presented in [8]. There are pockets of annular style with a circular groove with the oil supply and a small depression in the inner area. The other common style is very shallow pocket with a flat bottom surface, some other manufacturers use a conical shape of the pocket.
Study of the literature showed, that it is a common practice to not include presence of the hydrostatic recess in evaluation of hydrodynamic bearing properties. The only works known to the authors, which take into account this effect are papers of Pajączkowski [3,9] and Heinrichson [10]. Both authors include hydrostatic bearing depression of pad sliding surface in theoretical investigations of large thrust bearing properties under hydrodynamic mode of operation. In works [3,9], Kopswerk II (Austria) power plant hydrogenerator thrust bearing (outer diameter 2.3 m) was analyzed under transient states. Relatively good agreement of calculated and measured temperatures in the pad was obtained, but this was only parameter measured and compared. In work [10], Bieudron (Switzerland) hydrogenerator power plant thrust bearing was investigated (outer diameter ~2.2 m). In this case, film thickness measured by 4 distance sensors fixed to the pad, close to leading and trailing edge was compared to the calculated values. For the selected sensors good agreement was observed, but for sensors placed close to the pad trailing edge at the inner and outer radius, measured values were significantly higher than the calculated ones. This discrepancy of the results could be explained with runner deflection, which was not included in theoretical analysis.

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1.1 Objective of The Research

The aim of this work was to investigate theoretically how hydrostatic recess influences large hydrodynamic bearing properties under its hydrodynamic action. As an object of the analysis, Itaipu power plant hydrogenerator thrust bearing was selected (Figure 1). This bearing, with its outer diameter equal to 5.2 m, makes it one of the biggest thrust bearing which is under operation in the world. Bearing specific load is equal to 2.6 MPa and rotational speed is 92 rpm. During commissioning of the machine, detailed measurements of the hydrogenerator were completed. Within machine testing program, all important bearing parameters were monitored, especially pad and collar temperatures, hydrodynamic pressures profiles and oil gap thickness profiles.
In previous works [11,12], the authors completed TEHD analysis of Itaipu thrust bearing without including hydrostatic recess. Comparison of predictions and measurements revealed, that there were results discrepancy, especially in the vicinity of the hydrostatic recess. That is why, an attempt to include hydrostatic recess depression into analysis of Itaipu thrust bearing was taken. Additionally, it was decided to check the influence of the recess size on predicted bearing properties.

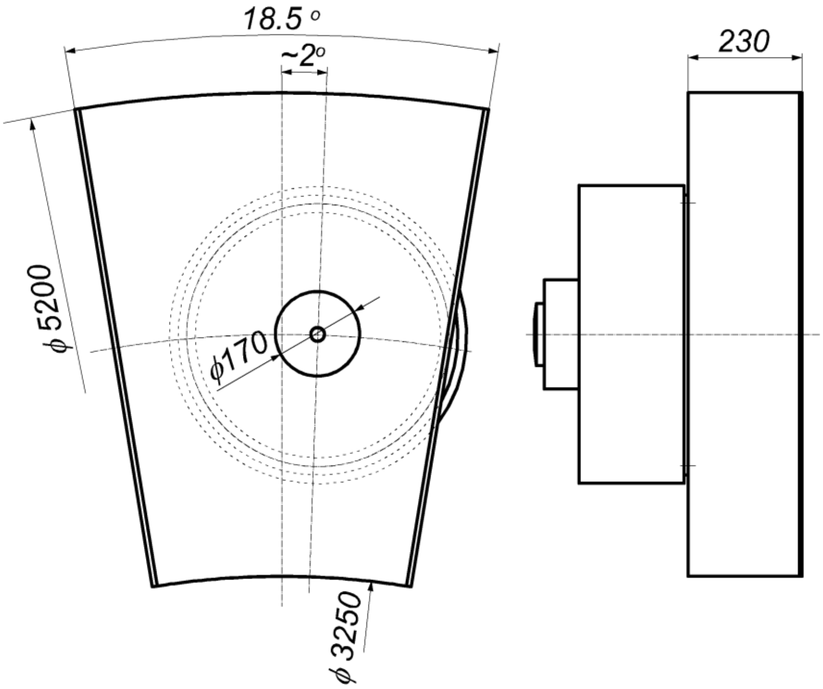

Figure 1. Itaipu thrust bearing pad and support arrangement [11].

2 MATERIAL AND METHODS

To reach the goal of this work, it was decided initially to use the bearing modelling which was applied with success in previous TEHD analysis of the Itaipu pad (without including the hydrostatic recess). Generally, this model is based on the solution of 3D Reynolds equation solved with the use of finite difference method, and can include all the most important bearing performance phenomena: pad tilting, thermal and elastic effects. Detailed description of the model can be found in [11,12]. To include hydrostatic recess depression, oil gap geometry was modified in the area of pocket. It was assumed similarly as in [3,9] that recess has conical shape, and nominal recess

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depth is 2 mm. It was planned to study both influence of the recess depth and diameter on predicted bearing parameters.
During calculation process, it was proved that in order to obtain a converged solution, quite dense mesh grid had to be used. In previous calculations completed for Itaipu pad without presence of hydrostatic, recess grid 31x49 (radial x circumferential) was used. In the present analysis, this was enlarged to 121x121 respectively (for some cases radial mesh was reduced to 51 divisions to shorten time of analysis). Improved mesh allowed to achieve converged solution only for hydrostatic recess depth variation. Attempts carried out to analyze different recess diameter with the use of the mentioned model were not successful.
In these circumstances, it was decided to try to analyze influence of recess dimensions with the use of another method - Rigid Body Solver (RBS) implemented in CFX software [13]. This allows to carry out FSI analysis (Fluid Solid Interaction). In FSI technique both physical fields: fluid and solid interact exchanging loads between fields automatically. Limitation of RBS method is, that interacting bodies can not deform under fluid or thermal loads, but can move freely, for example to reach equilibrium position. Rigid body can be defined within analysis as immersed solid or as set of surfaces. For the needs of this paper, 3D oil gap was modeled with the use of RBS (Figure 2). Mesh density was specified to obtain a uniform grid, 141x101x11 divisions in radial x tangential x thickness were used. Calculations were completed for isothermal regime, with the assumed representative oil viscosity μ = 0.018 Pas (Itaipu oil viscosity at 66°C – average temperature of the pad in the results of THD calculations). At the side of the oil gap, opening boundary condition was imposed. Upper wall of the model (touching the collar) was assumed as stationary wall with rotational speed equal to n = 92 rpm. Lower wall of the model (touching the pad) was assumed as a Rigid Body region. This wall can move in z direction, and has rotational degrees of freedom (around x and y axis). At the pivot position at Rigid Body region of the model (pad/oil wall) vertical force Fz = 1.73 MN was applied. This force is a representation of the bearing pad load, and causes Rigid Body wall movement and tilt (around x and y axis) to find its equilibrium position.
THD calculations were carried out for 5 different depths of hydrostatic recess: 30%, 50%, 70%, 85% and 100% of initial recess depth and nominal recess diameter.
Isothermal calculations with the use of RBS technique were completed for three recess depths: 30%, 70% and 100% of initial recess depth (recess diameter in those cases was equal to 170 mm) and three different recess diameter: 100%, 150% and 200% of initial recess diameter (recess depth in those cases was equal to 2 mm). Additionally, one case without the recess was also calculated.

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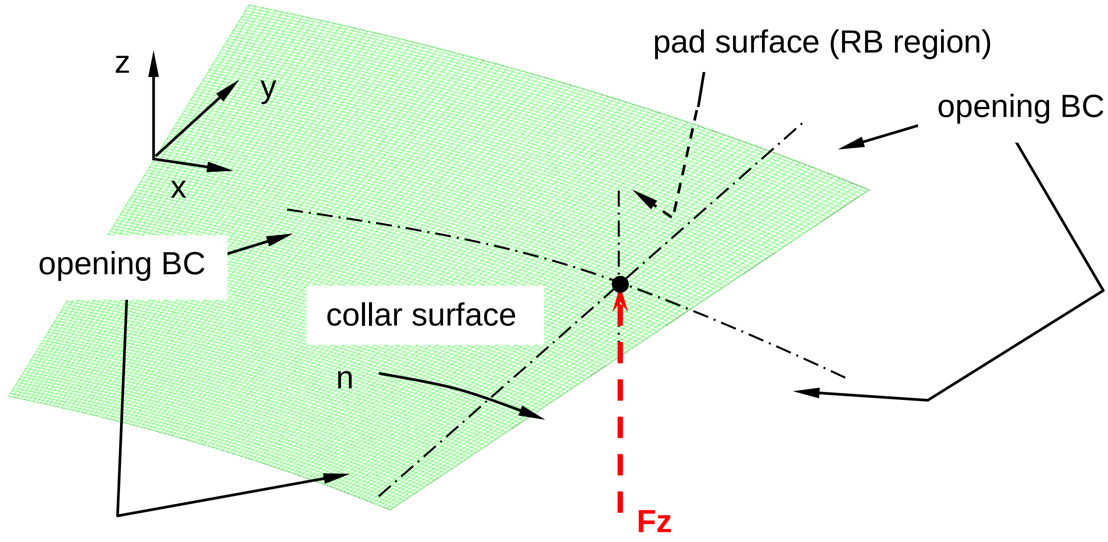

Figure 2. Mesh of the oil gap used in RBS analysis (recess is not visible, due to large in plane dimensions comparing to model thickness).

3 RESULTS AND DISCUSSION

Results of calculations are presented in form of profiles, to make it easy to compare them and find differences in calculated bearing parameters. Additionally, some selected bearing parameters are also shown as contour plots.

3.1. THD Results

THD calculations were carried out for different recess depth. Extreme values of calculated parameters were collected in Table 2.

Table 2. Comparison of calculated bearing parameters for different recess scaling, THD regime.

THD	unit	h scale				
		mesh 121x121			mesh 51x121	
		30 %	*50 %*	*70 %*	*85 %*	*100 %*
p_{max}	[MPa]	5.97	5.76	5.70	5.62	5.60
T_{max}	[°C]	81.8	81.4	81.1	81.7	81.6
h_{min}	[μm]	72.7	73.3	73.9	72.5	72.6

The largest maximum oil pressure p_{max} was calculated in case of smallest recess depth (5.97 MPa, 30% of nominal depth). Increase of recess depth caused decrease of calculated maximum oil pressure, to its minimum value equal to 5.6 MPa for 2 mm recess depth. In case of maximum pad sliding surface temperature T_{max}, the trend is not constant. Increase of recess depth leads to a small decrease of calculated maximum temperature. But for cases of 85% and 100% of the recess depth, calculated maximum temperature of the pad was higher. This is probably caused by the change of grid density (from 121x121 on 51x121). Similar results inconsistency is observed also in calculations of minimum oil gap thickness h_{min}. Slightly larger minimum film thickness was calculated for deeper recesses, but change of mesh density disturbed this trend.

In Figure 3, results of THD calculations for three recess depths in form of contour plots are shown. Differences in calculated parameters are not so significant, but

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presence of hydrostatic recess is clearly visible in the contours of hydrodynamic pressure and temperature of pad sliding surface.

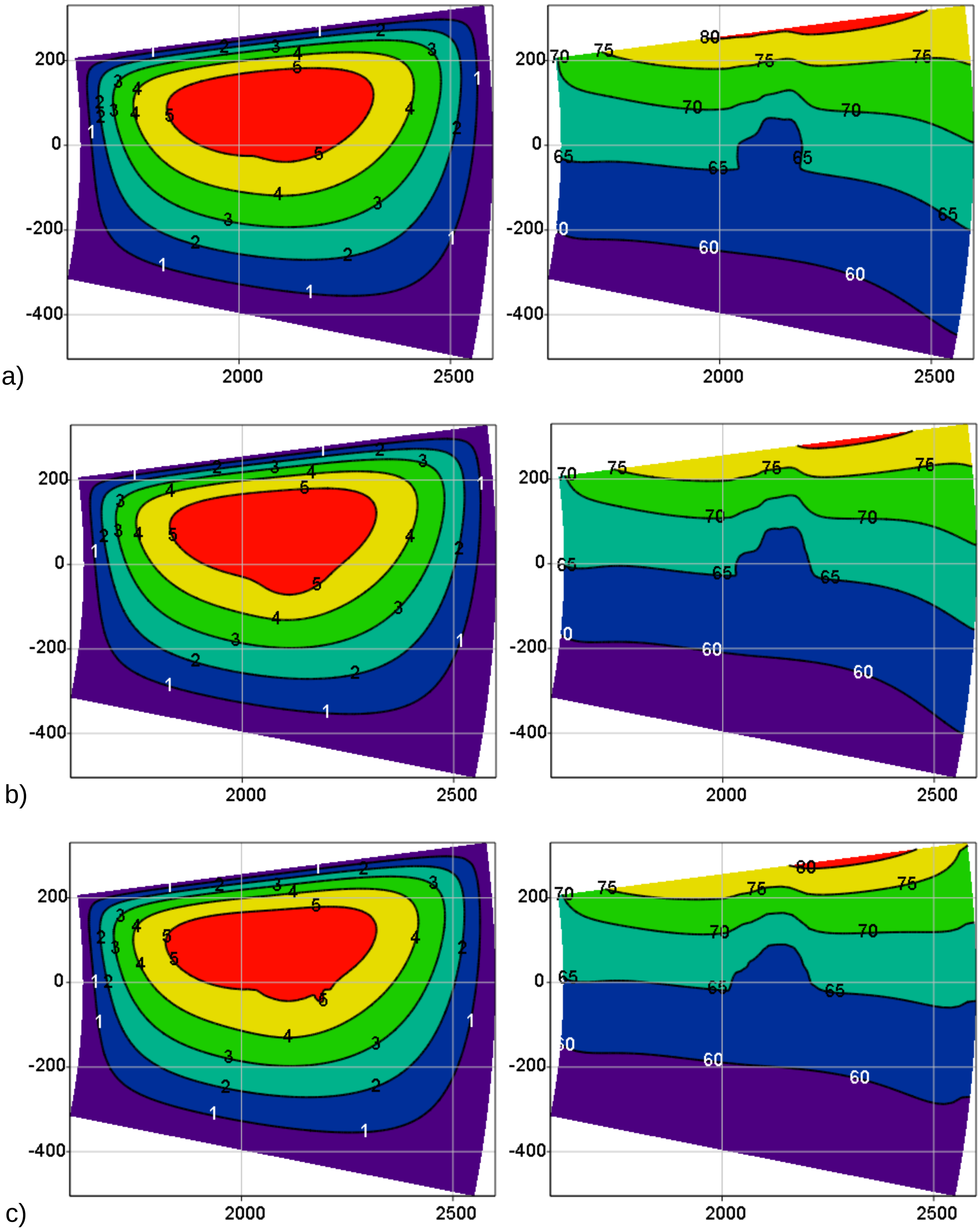

Figure 1. Contour plot of hydrodynamic pressure [MPa] (left-hand column) and pad sliding surface temperature [ºC] (right-hand column); a) for 30% of recess depth, b) for 70% of recess depth, c) for 100% of recess depth.

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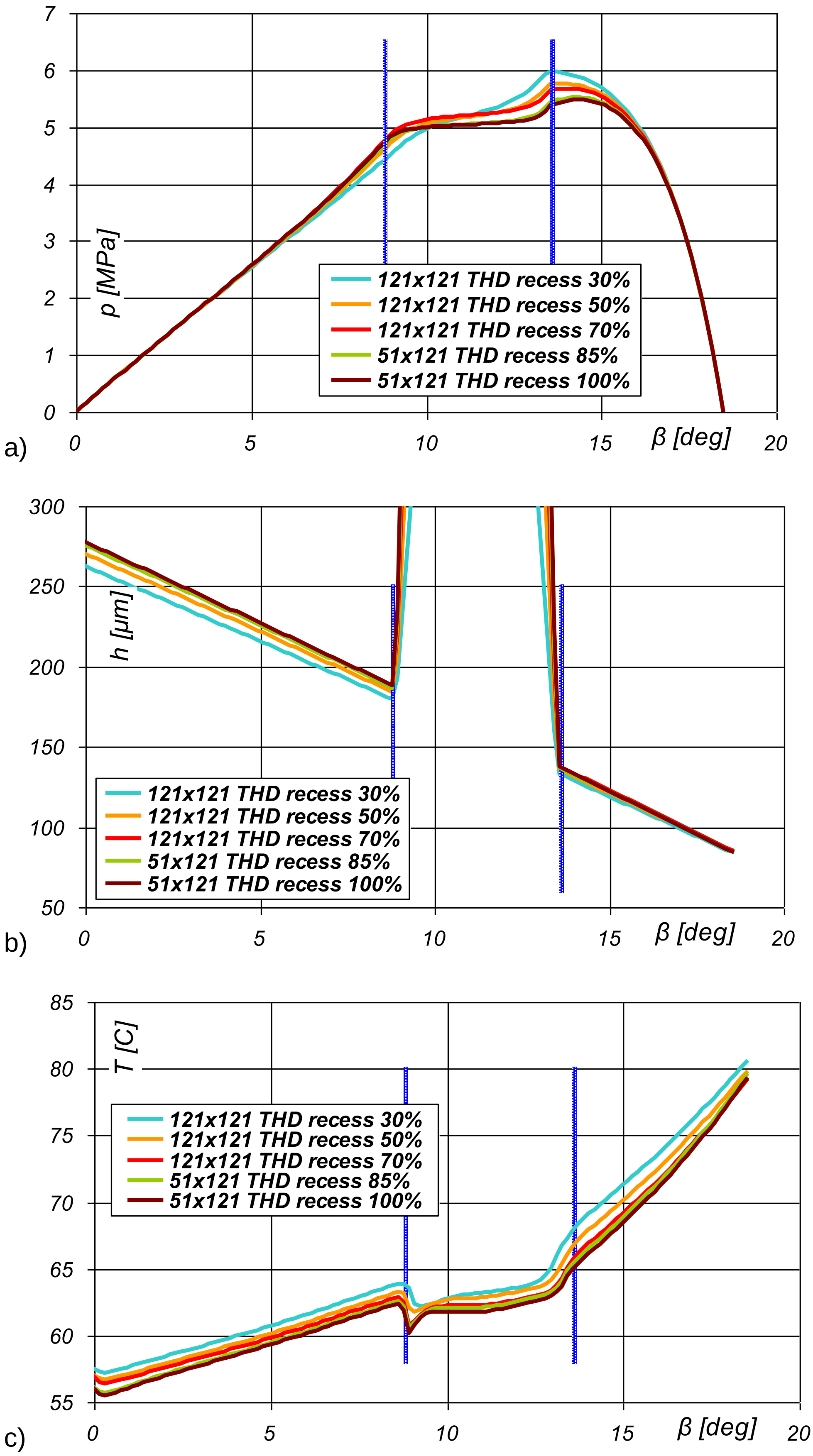

Figure 2. THD results of Itaipu pad parameters profiles at mean pad radius; a) oil gap pressure [MPa], b) oil gap thickness [μm], c) pad sliding surface temperature [^{o}C].

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Differences in calculated bearing parameters can be more easily observed in Figure 4, where profiles of oil gap pressure, oil gap thickness and pad sliding temperature at pad mean radius are compared. Comparing oil gap pressure profiles it is clearly visible, that shallow recess (30%) influences the pressure profile less, than deep recess (100%). In case of shallow recess, pressure increased continuously, even in the zone of hydrostatic recess. While for deep recesses (85% and 100%) there is an area with almost constant pressure in the pocket (form of “plateau”).
Comparing calculated oil gap profiles, one can observe, that shallow recess lead to lower circumferential pad tilt. On the opposite, deep recess caused the largest pad tilt. Recess depth had almost no influence on minimum film thickness calculated with the use of THD model.
Concerning temperature profiles at pad mean radius, irrespective of recess depth, temperature profile is very similar. The highest temperature of pad sliding surface was calculated for the case with shallow recess (30%), the smallest for the case of deep recess (100%). Temperature of the pad increased from the inlet to the pad to the beginning of the recess. Then, in the area of the pocket, small temperature drop was observed for all analyzed cases of recess depths. In the recess area, temperature increased slightly, and approaching recess outlet significant temperature increase was calculated. In the area between recess and pad outlet, constant temperature gradient can be seen for all analyzed cases.

3.2. Rigid Body Solver Solution – Isothermal Case

Isothermal calculations were carried out for different recess depths and recess diameters. Extreme values of calculated parameters were collected in Table 3.

Table 3. Comparison of calculated bearing parameters for different recess scaling, Isothermal regime.

RBS Isothermal	d scale	*100 %*				*150 %*	*200 %*
	h scale	*0 %*	*30 %*	*70 %*	*100 %*	*100 %*	*100 %*
p_{max}	[MPa]	5.85	6.02	5.71	5.63	5.33	5.17
h_{min}	[µm]	83.4	82.1	81.5	81.2	78.8	74.2

Analyzing data collected in Table 3, clear trends are visible. Presence of the shallow recess increased maximum hydrodynamic pressure comparing to case without recess (h scale 0 %). For deeper recesses, maximum hydrodynamic pressure was calculated smaller than for shallow recesses. What seems interesting, for cases with 70 % and 100 %, maximum hydrodynamic pressure was calculated smaller than in case without hydrodynamic recess. Concerning minimum oil gap thickness, it was calculated the smallest for deep recesses, and highest for case without recess. However, in this case, differences were not significant (~2 µm).
Comparing obtained results for different recess diameters, one can see, that larger recess diameter (200 %) caused reduction in maximum oil gap pressure, and also in minimum film thickness comparing to case with nominal pocket diameter (d scale 100 %).
In Figure 5, contour plots of calculated hydrodynamic pressure for different recess depths and diameters are shown. The same contour scale were used for all graphs, to make it easy to compare the results. Depression of the hydrostatic recess has a significant influence on hydrodynamic pressure profile, and is visible as an isobar shape change in contour plots shown in Figure 5.

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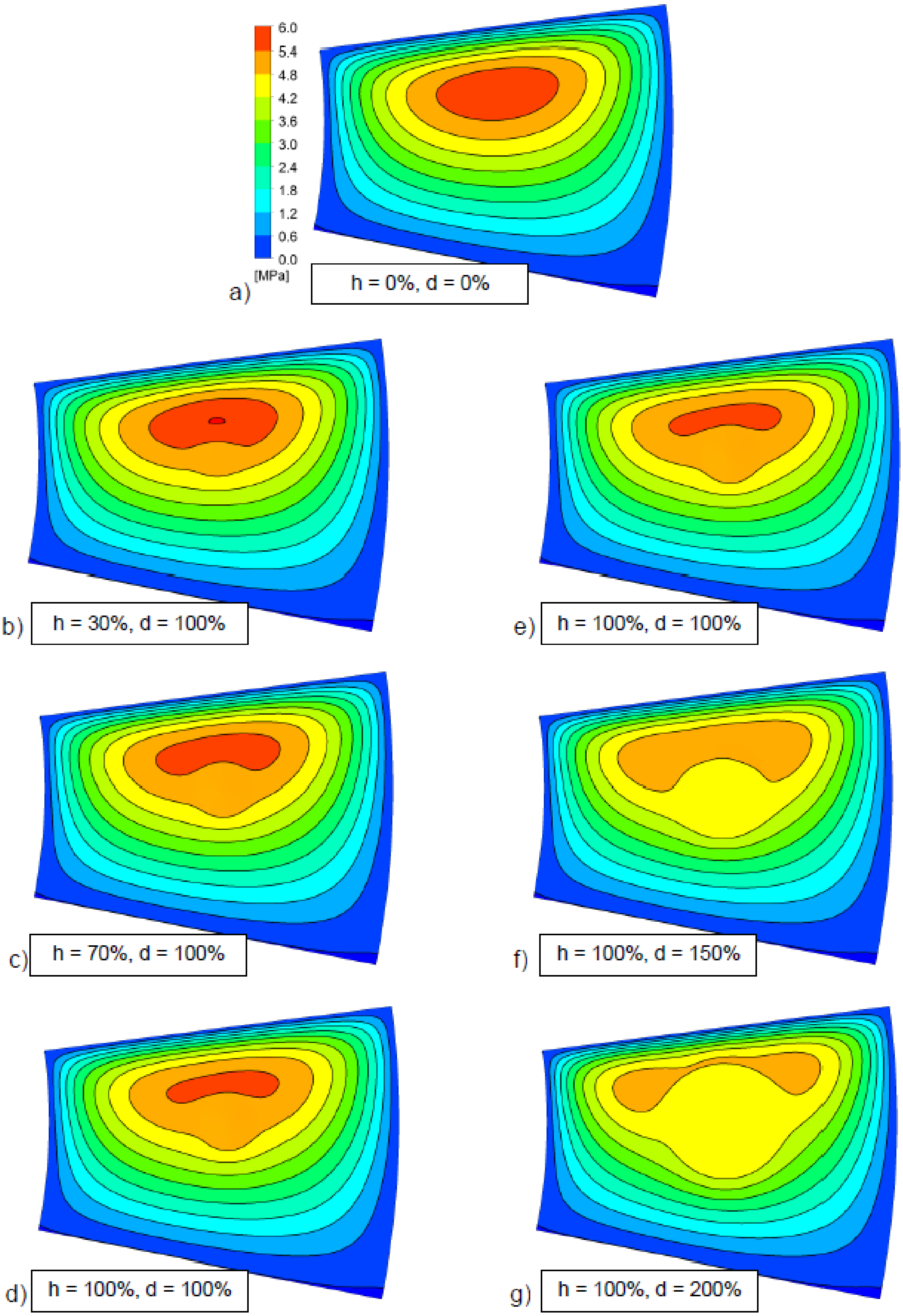

Figure 5. Contour plot of hydrodynamic pressure [MPa] obtained for Isothermal calculations; a) pad without recess, b) to d) for recess depth variation (left-hand column), e) to g) for recess diameter variation (right-hand column).

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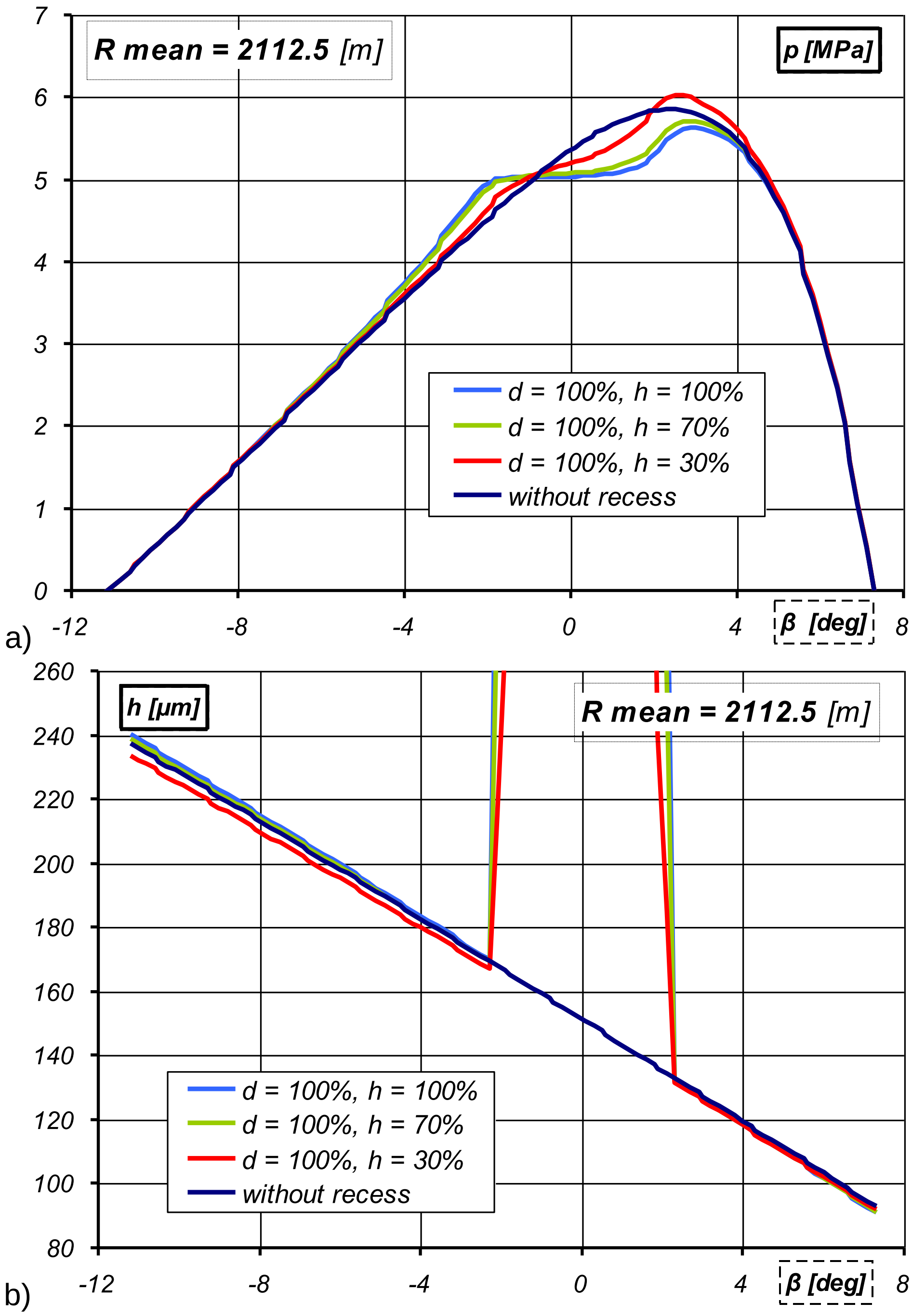

Figure 6. Isothermal results of Itaipu pad parameters profiles at mean pad radius for recess depth variation (nominal recess diameter); a) oil gap pressure [MPa], b) oil gap thickness [μm].

In Figure 6 and Figure 7, calculated at mean pad radius profiles of hydrodynamic pressure and oil gap thickness were compared for different recess dimensions.
Comparing oil gap pressure (Figure 6.a and Figure 7a), stronger influence on pressure profile had variation of the recess diameter then its depth. In case of variation of recess depth, the same trends were observed as for THD calculations (see Figure 4a). The highest hydrodynamic pressure was calculated for shallow recess, the smallest for deep recess. In case of oil film thickness calculated with the use of isothermal regime, the same trends in influence of recess depth on oil film thickness was noticed as for THD results. The smallest pad tilt was calculated for shallow recess (30%), the highest for deep recess. Nevertheless, what should be mentioned, differences in tangential oil film profiles were not significant.

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In case of the results obtained for different recess diameters – it had stronger influence on calculated pad tilt, than the recess depth. Increased recess diameter resulted in larger pad tangential tilt and smaller oil gap thickness at the pad outlet. Concerning pressure profiles, in all analyzed cases of recess diameter, in the area of the recess pressure plateau was observed. What is characteristic, close to the recess end pressure rise was calculated.

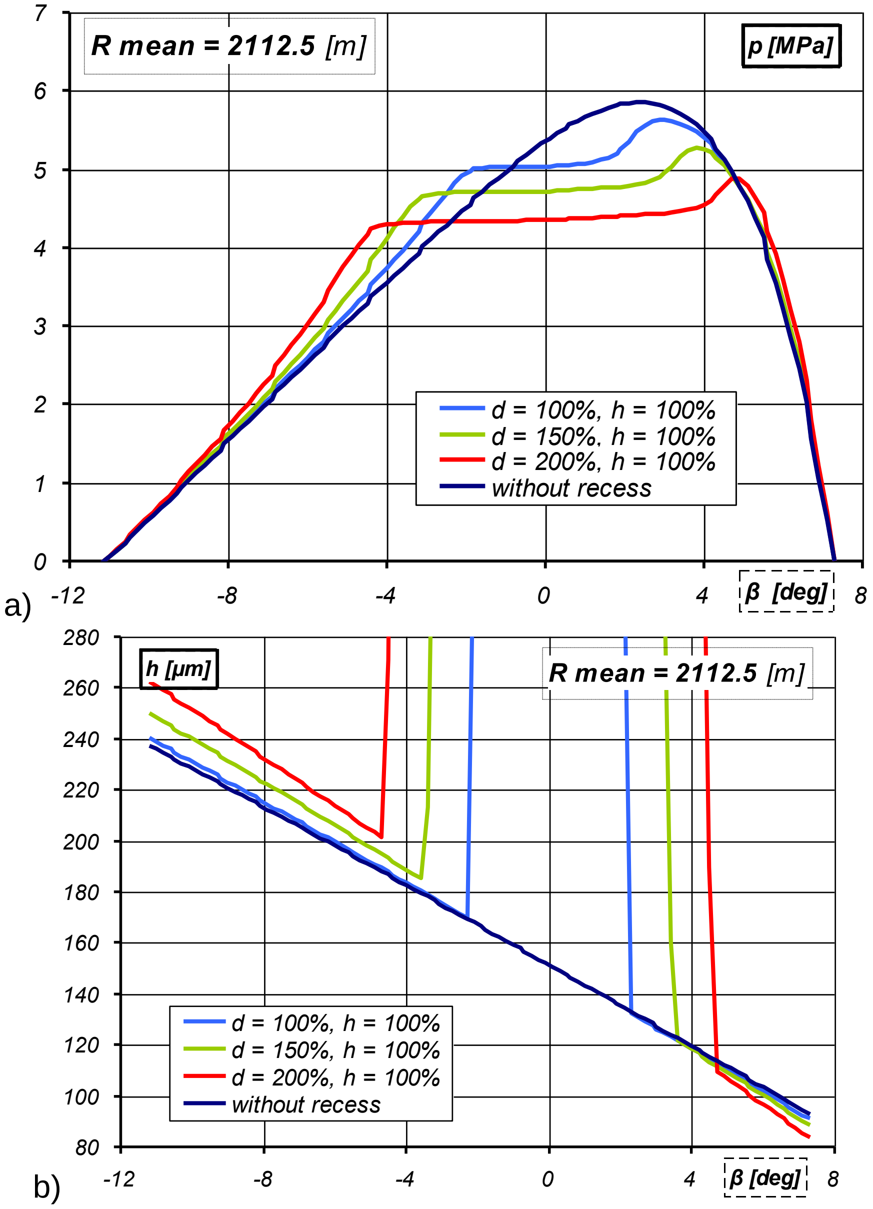

Figure 3. Isothermal results of Itaipu pad parameters profiles at mean pad radius for recess diameter variation (nominal recess depth); a) oil gap pressure [MPa], b) oil gap thickness [μm].

4 CONCLUSIONS

In this paper, a modelling of lifting pockets in tilting-pad thrust bearings has been presented in both isothermal and thermohydrodynamic regimes. The influence of the pocket depth and pocket size on the main bearing characteristics has been analyzed. The main conclusions are the following:

- Significant problems with the solution convergence were caused by including the hydrostatic pocket into theoretical analysis of the thrust bearing.

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- The presence of the lifting pocket has an significant effect on both pressure and temperature fields on the pad. In the pressure field, a plateau is observed in the pocket and a pressure jump is noticeable at the trailing edge of the pocket. In the temperature field, a general small decrease and remarkable drop at leading edge of the pocket are noted.
- The maximum pressure and the maximum temperature are reduced.
- The tilt angle of the pad is increased compared to the operation without a lifting pocket. Thus, the minimum film thickness decreases.
- The influence of the pocket increases with the increase of the depth and the size of the pocket.

Future work will concern the analysis of the influence of the lifting pocket on the TEHD performance of large tilting-pad thrust bearings and the comparison of the numerical results with the experimental data.

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