

# MODELING LUBRICANT FLOW BETWEEN THRUST BEARING PADS

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## 1. INTRODUCTION

### 1.1. Flow phenomena between bearing pads and in the bearing grooves

In 1961 Arwas and Sternlicht noticed that the temperature of the oil entering the film and the mixing of oil between bearing pads have a great influence on bearing characteristics [1]. Research on oil flow through a lubricating groove carried out by Ettles [2] showed that about 85% of hot oil leaving the gap enters the next oil gap in the case of laminar flow. Later in his subsequent papers [3, 4] Ettles proposed an idea of the “hot oil carry-over factor”. Values for this factor were assessed on an experimental basis as a function of sliding speed and size of the gap between bearing pads.

Other models of oil flow in the bearing groove were proposed by Vohr [5], who presented a model including a variety of heat exchange phenomena in the groove, and Heshmat and Pinkus [6] and Kiciński [7], who calculated both the flow of oil and heat balance. Some phenomena in the oil gap are described with increasing accuracy by these derived models; however, they consider bath lubrication only, i. e. the situation where the bearing is submerged in oil. In contemporary bearing designs oil is quite frequently supplied directly to the area between pads, either in the groove or forced from an external cooler directly to the film. In these cases, the phenomena in the area between pads can not be described by existing models and the inlet temperature is difficult to evaluate.

The importance of inlet temperature value for the results of bearing calculations may be illustrated by an example of the authors’ study [8] of the performance of a thrust bearing of a water turbine. Outer diameter of the bearing was 1780 mm, rotational speed 214 rpm (sliding speed 14.9 m/s). Axial load was equal to 6 MN (specific load 3.05 MPa), the bearing was fitted with a system to directly supply cool oil between the pads (Fig. 1). A drawing of a similar arrangement is shown in Fig. 2 (after [9]).

Data in Table 1 show the results of thermo-elasto-hydrodynamic (TEHD) calculations for two cases: one with the inlet oil temperature set as 40°C and the other with inlet

temperature equal to 55°C. One can see large difference of the results. Although the maximum bearing temperature differs by only 7°C, the minimum film in Case A is almost two times thicker than that in Case B. The film profiles for both cases (shown in Fig. 3) are completely different.

The practical problem is that there is no basis for the assumed inlet temperature. In these calculations, assuming a 40°C inlet temperature can be justified as this was slightly above the temperature of the cool oil injected between pads. However, after a comparison of calculation results with experimental data, a better agreement was obtained with the use of a 55°C inlet temperature.

## **1.2. Use of CFD in fluid film bearings calculations**

A work of Zhang and Rodkiewicz [10] was one of the first in which calculations of a complete model of a bearing (i. e. oil film and the space between bearing pads) were carried out with the use of CFD techniques. Chen and Hahn used CFD code (CFX 4.1 Flow Solver) for solving the full Navier-Stokes equation for an inclined slider bearing, a Rayleigh-step bearing and a journal bearing [11]. A very good agreement of the results was observed. Kosasih and Tieu carried out numerical calculations of the journal bearings with a lubricating groove and pressurized oil supply [12]. They used the commercial codes CFX and PHOENIX. The authors noticed that supply pressure and lateral size of the groove are the primary factors which affect the efficiency of supplying cool oil to the oil film. Glavatskih and his collaborators using SOLVIA [13] carried out a TEHD analysis of a thrust bearing model including bearing pad, fluid film and the runner. The authors noticed a good agreement of the results and experimental data but pointed out that the use of a 3-D film element and consideration of cavitation could further improve the accuracy of the calculations. Inlet temperature was calculated with the use of Ettles' concept of hot oil carry over factor. The value of the factor was adjusted to obtain a good agreement with experimental results.

## **1.3. Special lubricant supply methods in bearing design**

The problem of hot oil entering the film of the next pad seems to have been recognized by bearing designers long before the attempts of modeling the flow between pads. Various arrangements aimed at improving scoring of hot oil layer moving with the runner have been proposed, [15, 16].

Along with scrapers, different lubricant supply methods were devised in order to help fresh lubricant enter the film, thus reducing inlet oil temperature. These provide a forced supply of oil directly to the space between bearing pads (Fig. 1 and 2) supplying the oil directly to a groove machined close to the leading edge or in a special element situated close to the leading edge (known under trade marks as Leading Edge Groove®, Pocket Feed® or Controlled Inlet®).



Another idea is to spray cool oil directly on the runner through special high-velocity jets in order to scour the layer of hot oil leaving the film of the upstream pad before it enters the film of the downstream pad. The efficiency of various directed lubrication systems has been demonstrated in experiments. Remarkable effects including lower bearing temperature, lower power loss and lower oil consumption are observed especially for high-speed bearings with sliding speeds of 50-60 m/s and above [16].

## **2. TWO-DIMENSIONAL CALCULATIONS**

Improving capabilities of CFD tools and other research attempts involving CFD applications in bearing modeling encouraged the authors to take up their own studies on the use of CFD. In [17, 18], the evaluation of ANSYS 6.1 for the analysis of flow phenomena in bearings was carried out on the basis of a simple 2-D model of a fixed geometry slider. A good agreement with analytical solutions was observed after getting through a variety of “teething problems”. In the next effort, the gap between the bearing pads was included in the model. In Fig. 4 circulation of the oil between the pads in a bath-lubricated thrust bearing is compared with that in a bearing having a direct oil supply.

One may observe an intensified circulation in the case with direct oil supply which improves the supply of cool oil to the inlet of the downstream pad. In the analyzed case the average inlet temperature was decreased by 4°C.

## **3. THREE-DIMENSIONAL CALCULATIONS**

### **3.1. Geometry and boundary conditions**

Thrust bearings are the main interest of the research team at the Gdansk University of Technology. In a thrust bearing, due to centrifugal forces acting on the oil between pads, a radial component of speed is important. This component is not included in 2-D models of the bearing. To include this component, it was decided to switch to 3-D modeling despite its increased complexity and the longer computing times involved. A model used for 3-D calculations is shown in Fig. 5. Again, two systems of oil supply were compared: traditional bath lubrication and a direct supply of oil between the pads (Fig. 5). Adiabatic conditions were assumed for the bearing pad and the runner on all metal-oil interfaces. At the bottom and sides of the groove a temperature was imposed. An inclined plane was assumed as the bearing pad surface. In both models, nodes at the outlet from the fluid film and inlet to the groove were coupled.

Flow has been assumed as turbulent. A Shear Stress Transport Model which consists of a standard  $k-\varepsilon$  model and  $k-\omega$  model was used. The latter provides better modeling of the turbulent boundary layer near the wall than the standard  $k-\varepsilon$  model alone. [19]

### 3.2. Mesh density

The authors have made some solutions in order to compare influence of mesh density (in the range of 1 000-40 000 elements) on the results. The difference between maximum pressure and temperature comparing fine and coarse meshes was about 5%. On the other hand, better definition of the velocity field between pads has been obtained when more elements were used (more than 2 500 elements). Table 2 shows mesh density used in FEM calculations.

### 3.3. Results

Calculations for both cases were carried out for data of a large tilting pad bearing, defined in Table 3, from a water turbine. The two bearing designs differed only in the lubrication method: bath lubrication or direct oil supply through a pipe situated between the pads with the jets directed towards the runner (see Fig. 1 and 2).

The results of calculations for the bearing with bath lubrication are shown in Fig. 6. Vectors depict oil velocity distribution in selected cross-sections of the space between the pads. It can be seen that oil motion across the space caused by a centrifugal force is the important component of motion, which clearly shows the necessity of 3-D modeling of the oil flow in the space between pads in a thrust bearing, since in 2-D models this component is neglected.

Several velocity distributions for the bearing with direct oil supply are shown in Fig. 7. Vectors of oil velocity are shown in two cross-sections: one radial through the supply pipe (A-A) and the other circumferential (B-B). The radial view shows the radial flow of oil directed outwards caused by the centrifugal force. In the circumferential view, one can notice the oil flow from the supply pipe directed to the runner. The speed of the oil coming from the jets, even though comparable to the sliding speed, does not seem adequate as the kinetic energy of the fresh oil is dissipated before it reaches the surface of the runner.

Other results are shown in Table 4. For calculations performed for the same film geometry (175  $\mu\text{m}$  at the inlet and 65  $\mu\text{m}$  at the outlet), the mean temperature at the inlet to the film changed by 3°C and the maximum bearing temperature by 5°C. Colder, more viscous oil gives a larger load-carrying capacity for the bearing with direct oil supply. In a real bearing, the effects would probably be more pronounced because at the same load, the oil film in a bearing with direct oil supply would have been thicker which would result in further decrease of the temperature. Due to lower temperatures at the



surface, the thermal deformations (not taken into account in our model) would decrease which in turn would further improve the film profile and increase the load-carrying capacity (or increase oil film thickness for the same load).

Oil temperature at the inlet of the pad is shown in Fig. 8 – for bath lubrication at the upper drawing and direct oil supply at the bottom drawing.

## 4. CONCLUSIONS

The results from CFD analyses of the oil flow through the gap between thrust bearing pads illustrate the possibilities for modeling phenomena within the bearing with this technique. Such analyses can improve inputs to traditional bearing modeling since current models treat the flow between bearing pads in a very simplified way, despite its great influence on bearing performance. Simplified modeling of the phenomena between pads is justified for bearings provided with bath lubrication, while it is insufficient for bearings with direct oil supply to the gap between the pads. Although a variety of bearing designs with improved oil supply are being applied, it seems to the authors that these modifications are far from optimized and can be further improved with the use of CFD analysis.

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file with figures to the paper:

## **MODELING LUBRICANT FLOW BETWEEN THRUST BEARING PADS**

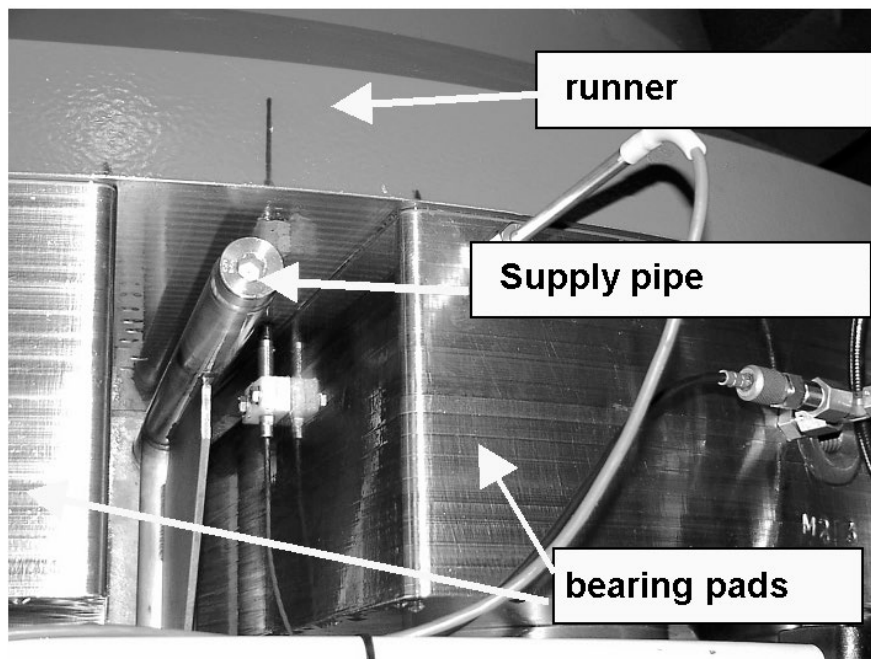


Fig. 1. Thrust bearing with a system of direct oil supply [8]

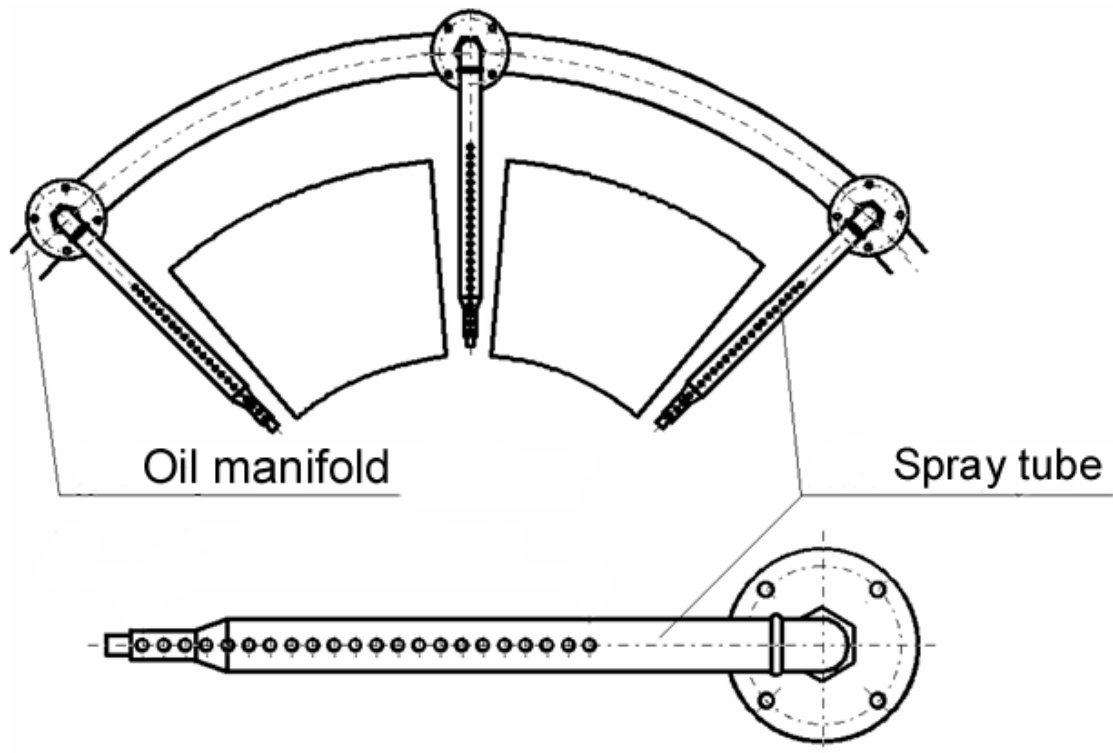


Fig. 2. Schematic view of the direct oil supply system



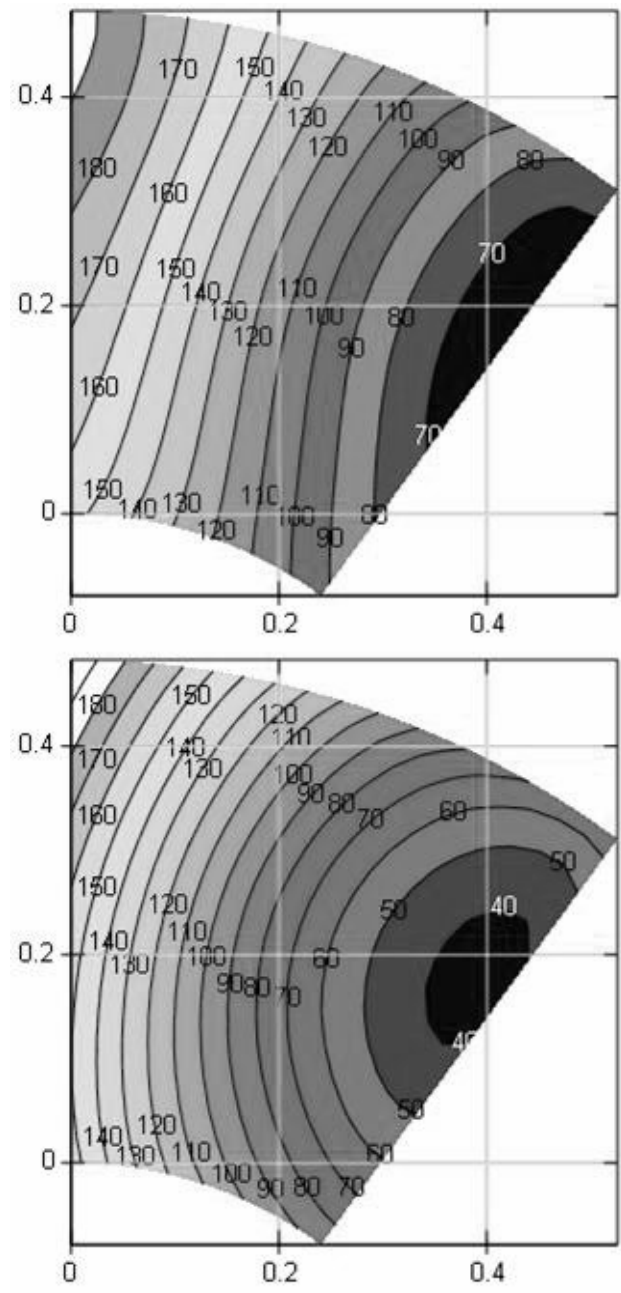


Fig. 3. Film profile for inlet temperature of 40°C (upper diagram) and 55°C (lower diagram) [8]

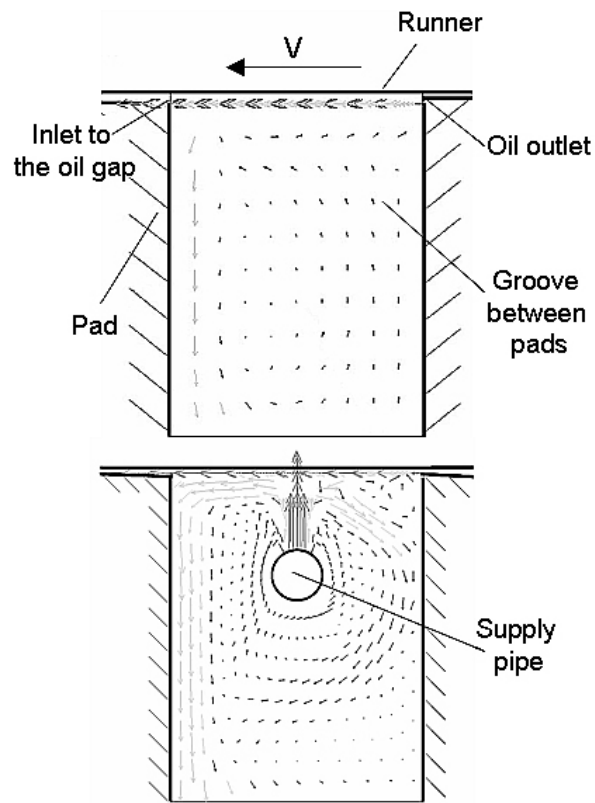


Fig. 4. Oil circulation between the pads in a thrust bearing with bath lubrication (upper drawing) and a direct cool oil supply system (lower drawing)

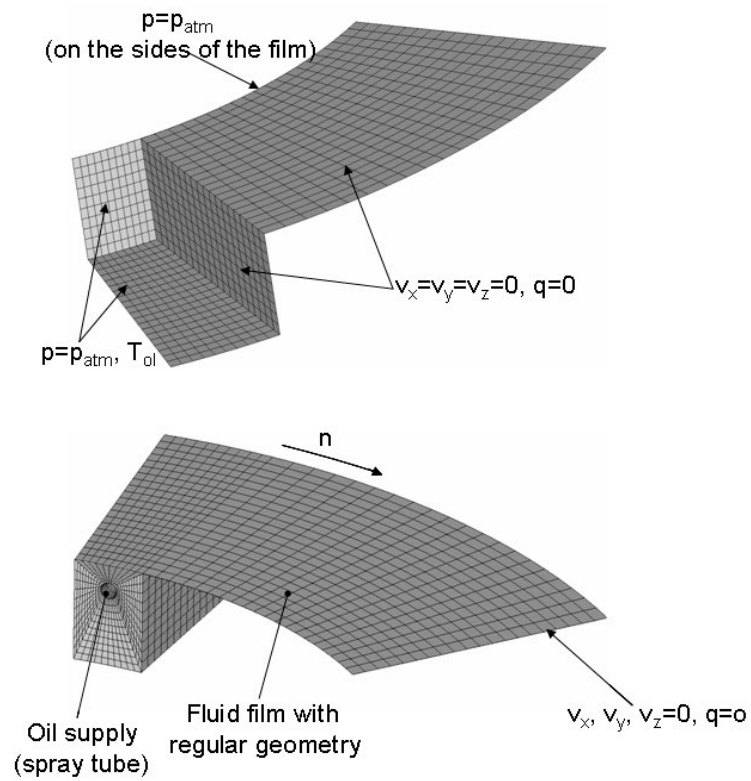


Fig. 5. Geometry and boundary conditions for 3D calculations: bearing with bath lubrication (upper drawing) and bearing with direct cool oil supply (lower drawing)

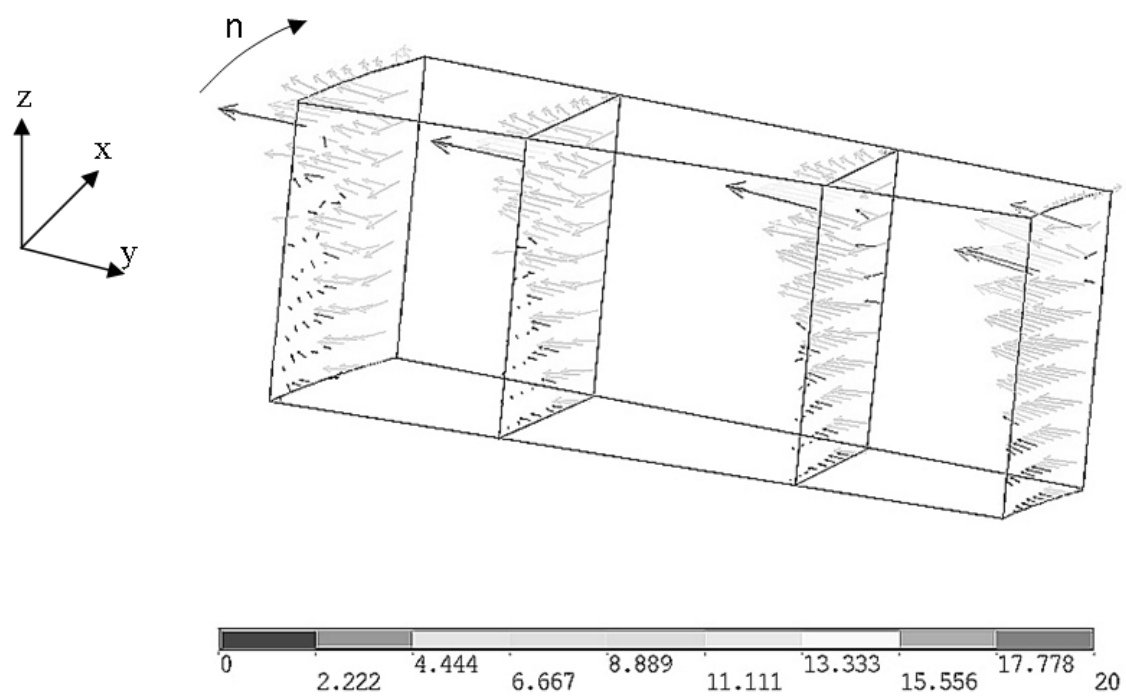


Fig. 6. Oil velocity distribution in the gap between pad (in selected cross-sections) – bath lubrication case. Shades of grey of the vectors show oil speed in  $[\text{m/s}]$ , according to the legend bar

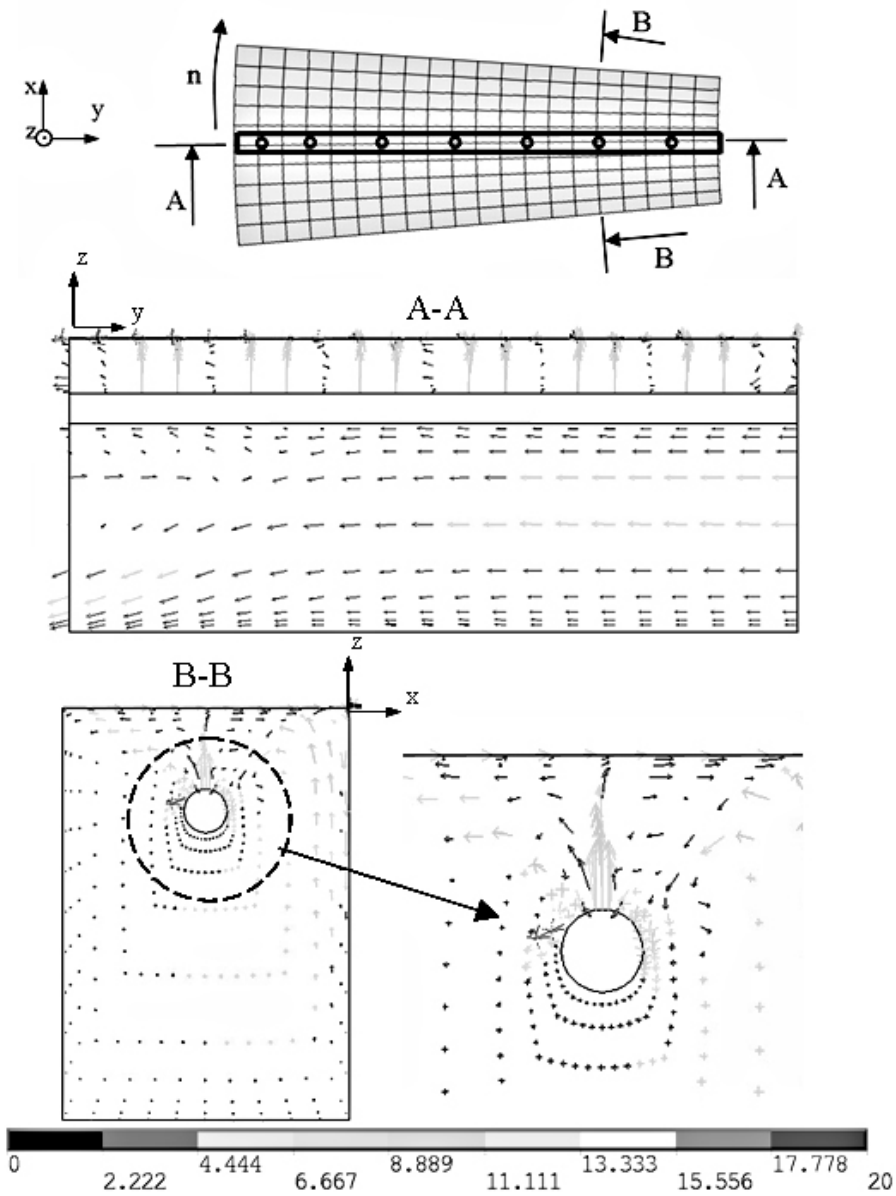


Fig. 7. Oil speed distribution in the gap between pads of the bearing with direct oil supply. Shades of grey of the vectors show oil speed in [m/s] according to the legend bar

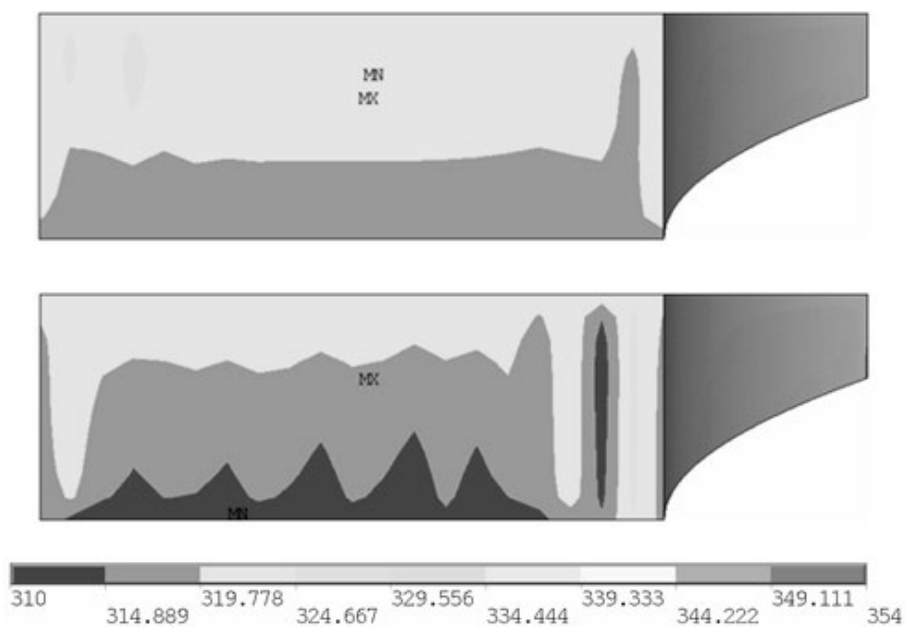


Fig. 8. Temperature distribution at the inlet to the film for bath lubrication (upper graph) and direct oil supply (lower graph); shades of grey show oil temperature in [K] – according to the legend bar

Table 1. Bearing performance for different inlet temperatures

Case	Inlet temp.	Maximum film pressure	Minimum film thickness	Maximum temperature	Maximum pad deformation
A	40°C	9.3 MPa	62.6 $\mu\text{m}$	64.8 °C	24.9 $\mu\text{m}$
B	55°C	11.2 MPa	36.6 $\mu\text{m}$	71.7 °C	57.3 $\mu\text{m}$

Table 2. Mesh density used in 3D calculations

Coordinate	Fluid film mesh	Thin layer over groove/Groove meshes	Total number of elements	Calculation time (1500 iterations)
Bath lubrication				
$\varphi \times r \times h$	20×20×10	10×20×10/10×20×10	8000	1h 5min
Directed lubrication				
$\varphi \times r \times h$	20×20×10	10×20×10/(10×20×10) ×4	14000	1h 25min

Table 3. Data used for 3D calculations

Quantity	Bath lubrication	Direct supply
Rotational speed	214 rpm	
Bearing diameters	OD 1780 mm/ID 814 mm	
Width of the gap (in circumferential direction)	200 mm	
Number of pads	8	
Oil	ISO VG-46	
Bath temperature	45°C	
Film thickness inlet/outlet	175 $\mu\text{m}$ /65 $\mu\text{m}$	
Supply temperature	-	30°C
Supply flow for the bearing	-	1000 l/min
Supply speed	-	10 m/s



Table 4. Results of 3D calculations

Quantity	Bath lubrication	Direct oil supply
Bearing load	868.8 kN	928.8 kN
Bearing max. temperature	86 °C	83 °C
Mean inlet temperature	47 °C	44 °C