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# NUMERICAL ANALYSIS OF OPERATING PARAMETERS OF HYDRODYNAMIC THRUST BEARINGS LUBRICATED WITH MAGNETIC FLUID

## NUMERYCZNA ANALIZA PARAMETRÓW PRACY HYDRODYNAMICZNYCH WZDŁUŻNYCH ŁOŻYSK ŚLIZGOWYCH SMAROWANYCH CIECZĄ MAGNETYCZNĄ

**Key words:** thrust slide bearing, CFD simulations, magnetorheological fluid, hydrodynamic lubrication.

**Abstract:** The paper presents the results of numerical simulations (CFD) of hydrodynamic thrust slide bearings lubricated with magnetorheological (MR) fluid. The analyses were carried out to evaluate the influence of the rheological properties of the lubricant, as well as the geometry of the bearing's thrust pad surface. Bearing load conditions were considered on the key functional features of the system, i.e. axial force and torque. The paper presents a comparative analysis of various geometries of thrust bearings and points out possible functional features of hydrodynamic thrust bearings lubricated with fluids with controlled rheological properties.

**Słowa kluczowe:** wzdłużne łożysko ślizgowe, symulacje CFD, ciecz magnetoreologiczna, smarowanie hydrodynamiczne.

**Streszczenie:** W pracy przedstawiono wyniki symulacji numerycznych (CFD), hydrodynamicznych wzdłużnych łożysk ślizgowych smarowanych cieczą magnetoreologiczną (MR). Przeprowadzone analizy dotyczyły oceny wpływu właściwości reologicznych cieczy, sposobu ukształtowania powierzchni oporowej łożyska, jak również warunków obciążenia węzła tarcia na kluczowe cechy funkcjonalne układu, tj. siłę osiową oraz moment obrotowy. W pracy przestawiono analizę porównawczą różnych geometrii łożysk wzdłużnych oraz wskazano na możliwe do uzyskania cechy użytkowe wzdłużnych hydrodynamicznych łożysk ślizgowych smarownych cieczami o sterowanych właściwościach reologicznych.

## PREFACE

The source of bearing capacity is usually hydrostatic pressure, which is obtained using an external lubricant supply system, or hydrodynamic pressure, created by relative movement of bearing working surfaces with a suitably shaped geometry. Their load capacity depends on the geometry of the cooperating surfaces, their relative velocity, and the rheological properties of the lubricant. While the shape and dimensions of the lubrication gap and the speed of movement are parameters ususally determined by by the design and operating conditions, the use of a rheological-controlled substance as a lubricant would create the capability of influencing the bearing performance. There are several reasons to consider magnetic fluids for this type of

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application. Such solutions may be characterized by a relatively simple design and there is a possibility of active control over bearing operating parameters.

Magnetic fluids are suspensions of microscopic particles with ferromagnetic properties in the carrier fluid (in technical applications hydrocarbon or silicone oils are generally used). A distinction is made between ferrofluids (FF), which are produced with the use of particles of several nanometres in diameter, and magnetorheological fluids (MR), which contain particles of several micrometres **[L. 1–4]**.

Due to particle sizes, a much wider range of rheological parameter variability is possible in the case of MR fluids than in ferrofluids ones. Moreover, only MR fluids undergo sedimentation. For these reasons, both substances are used in different technical applications. Ferrofluids are mainly used in seals **[L. 5]**, whereas MR fluids are used in vibration dampers, brakes, and couplings **[L. 6]**. There are also research works carried out concerning the development of slide bearings working with these fluids **[L. 7–10]**.

Among the factors indicating the justification behind the potential use of magnetic fluids as a lubricant in sliding bearings, apart from obtaining the capability of controlling the fluid's viscosity by the magnetic field, attention should be paid to the possibility of holding this substance in a specific zone by magnetic field forces. This can be of great importance, e.g., in microgravity operating conditions. Additionally, in a magnetic fluid under the conditions of magnetic field operation, magnetostatic pressure is created **[L. 11, 12]**, which can be used to separate bearing working surfaces, and, moreover, they are capable of generating normal stresses **[L. 12–15]**. Each of these factors can be an additional source of bearing load capacity.

An additional reason is that the MR fluid as a lubricant allows for high bearing stiffness regardless of the height of the working gap, which in turn reduces the requirements for tolerances of the bearing sliding surfaces. Additionally, it is possible to obtain the required bearing stiffness in a much shorter response time to extortion than in conventional solutions **[L. 12]**.

Bearings lubricated with magnetic fluids can have a higher load capacity **[L. 9]** and better vibration damping **[L. 9, 16]** compared to conventional solutions. Another aspect of the functioning of these bearings is the ability to use the effect of self-sealing, thanks to which the system can operate without refilling the grease **[L. 17–20]** and, at the same time, obtain a higher load capacity. This effect is related to the possibility of keeping the magnetic fluid in a certain position by means of a magnetic field, thanks to the appropriate geometry of the bearing's thrust pad surface. This allows the fluid flowing out of the working bearing gap to be reduced or completely retained.

The purpose of this paper is to investigate the influence of different bearing thrust pad surface geometry on the friction moment and the value of axial force. The main aim of the study was to determine the possibility of obtaining hydrodynamic pressure in such bearings.

## TEST METHOD AND COMPUTATIONAL MODEL

The paper presents the results of simulation studies and comparative analyses concerning the modelling of the thrust slide bearing, the scheme of which is presented in **Fig. 1**. The research method includes analysing the

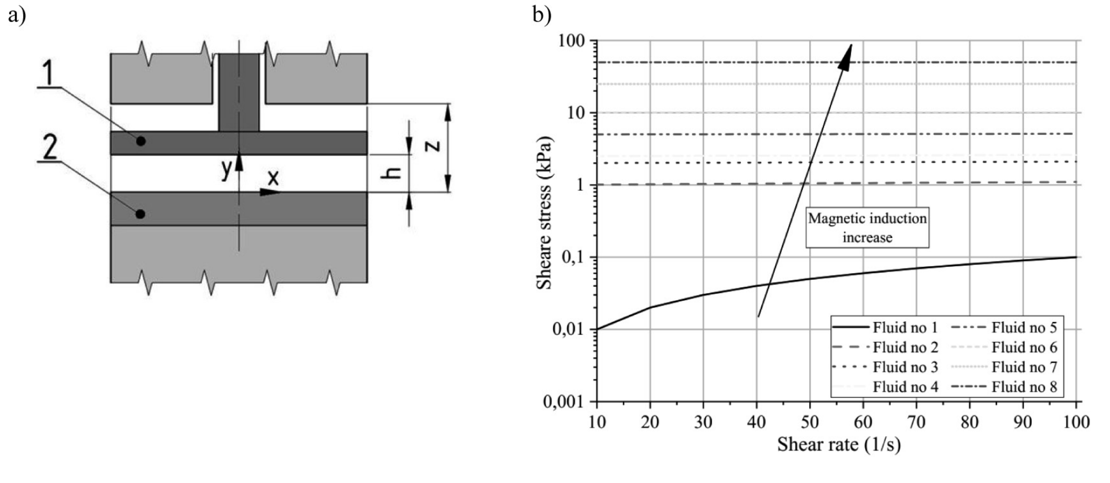



**Fig. 1. Scheme of the tested thrust bearing (a), flow curves of the modeled magnetic fluid (b)**

Rys. 1. Schemat badanego łożyska wzdłużnego (a), krzywe płynięcia modelowanej cieczy magnetycznej (b)

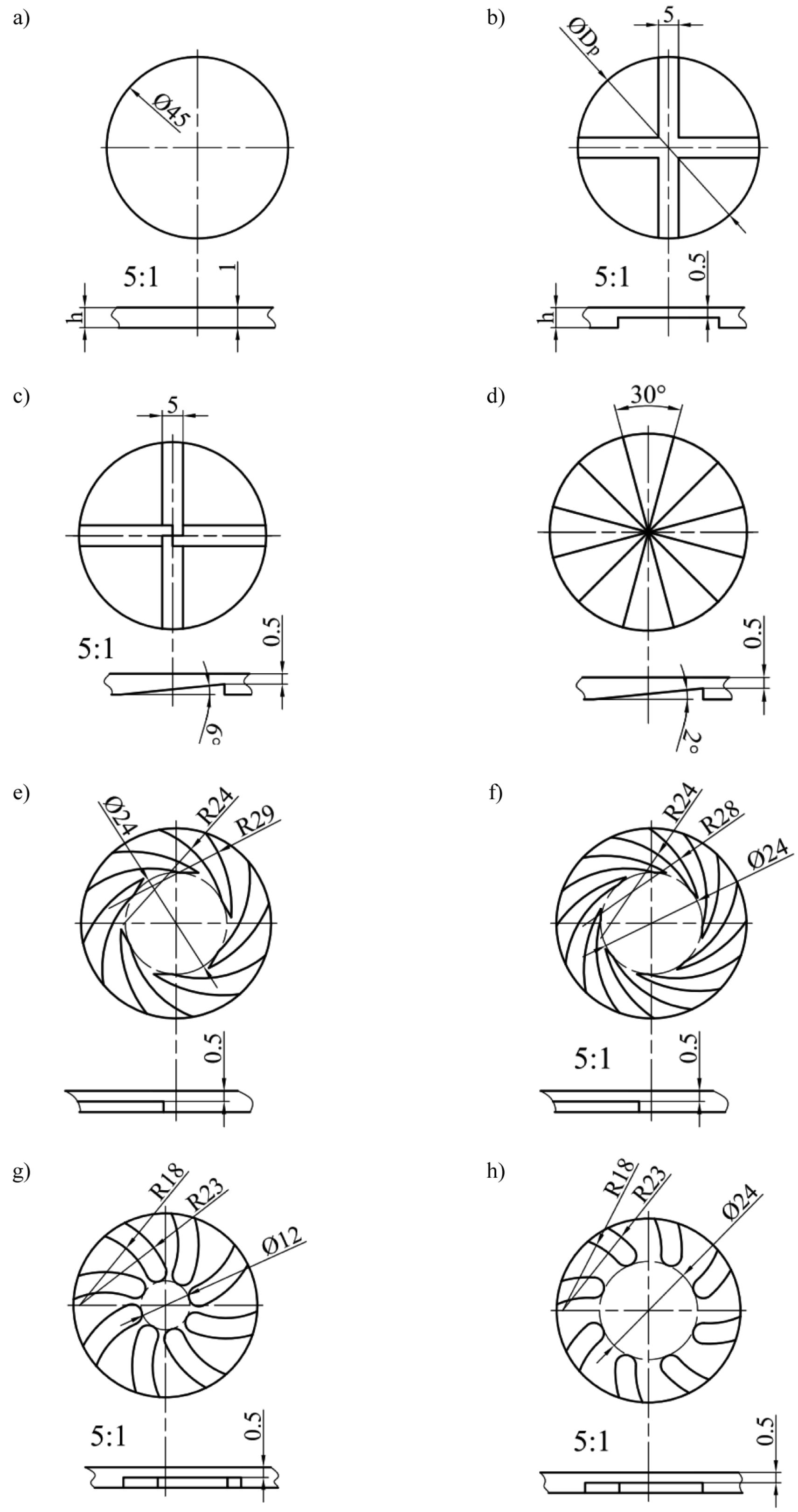



**Fig. 2. Design and dimensions of thrust bearing pads adopted for the simulation studies**

Rys. 2. Konstrukcje i wymiary płytek oporowych przyjętych do badań symulacyjnych

influence of three aspects of the functioning of the tested arrangement:
a) The plate rotational speed,
b) The geometry of the non-rotating bearing pad surface (2, **Fig. 1**), and
c) The rheological properties of the fluid on the bearing friction torque and axial (normal) force measured on the surface of the rotating plate (1, **Fig. 1**).

The first part of this work concerns the determination of the influence of rotational speed on functional parameters of the tested bearing. The tests were carried out on the geometry of two parallel plates (**Fig. 2a**) for four rotational speeds of the upper plate (n = 1, 100, 1000, 10 000 $min^{-1}$) and two rheological parameters of the modelled fluid (items 1 and 5 according to **Table 1**). All parameters refer to normal temperature and pressure according to NIST (20°C and 101.3 kPa). The model of the fluid marked as No. 1 refers to the flow of a Newtonian fluid with a viscosity of 1 Pa·s, while No. 5 is a Bingham fluid with a yield stress of 5 kPa. The values were assumed arbitrarily as typical and expected for the use of MR fluids in bearing systems. The comparison of flow curves of the modelled fluids is shown in **Fig. 1b**. Increasing values of the yield stress can be interpreted as the response of a fluid to an increase in magnetic field induction.

**Table 1. Physical properties of modeled MR fluids**
Tabela 1. Właściwości fizyczne modelowanych cieczy MR

| No. | Yield stress | Apparent viscosity | Density |
|---|---|---|---|
| | Pa | Pa·s | g/cm$^3$ |
| 1 | 0 | 1 | 2.83 |
| 2 | 1 000 | | |
| 3 | 2 000 | | |
| 4 | 2 500 | | |
| 5 | 5 000 | | |
| 6 | 10 000 | | |
| 7 | 25 000 | | |
| 8 | 50 000 | | |

The second stage of work concerned the determination of the influence of the thrust plate geometry on the hydrodynamic buoyancy force, as well as on the value of bearing friction torque and normal force. Variants of the 8 different geometries analysed are presented in **Fig. 2**.

The reference geometry is a configuration of two flat parallel plates (**Fig. 2a**) with the diameter D = 45 mm and a gap height between them of h = 1 mm (see **Fig. 1**). In the case of Geometry (b), four rectangular steps with 0.5 mm high, perpendicular to each other, radially spaced outlets were modelled. In Geometry (c), the step surfaces were inclined at an angle of 6° with a minimum gap height of 0.5 mm. In the case of Geometry (d), the steps of the pad surface were divided into 12 identical segments arranged radially, giving them an opening angle of 30°, and, additionally, the segment surfaces were tilted to the measuring plate at an angle of 2° to obtain a minimum gap height of 0.5 mm. Geometry (e) is based on one of the typical geometries used in hydro- and aero-dynamic thrust bearings. Geometry (f) is a modification of System (e) with the addition of two outlets. In Geometries (g) and (h), the shape of the outlets was modified. The shape of radial arcs was adopted, whose radial centres are not in the bearing axis. In Geometry (h), the inner diameter was increased from 12 mm to 24 mm. In Geometries (e), (f), (g), and (h), the step height was 0.5 mm.

The third stage of the research concerned the determination of the influence of rheological properties of the modelled MR fluid. For this purpose, five different sets of rheological parameters of the fluid in the bearing (according to **Table 1**) and a bearing variant type (f) (**Fig. 2**) were modelled, operating at a constant speed of n = 1000 $min^{-1}$.

All numerical analyses, the results of which are presented in the paper, were carried out using the laminar flow model, without the influence of gravity (this was justified by the low height of the modelled fluid). Thermal phenomena were omitted. As finite elements of the modelled fluid volume, pyramids with an average distance between grid nodes equal to 0.1 mm were used. In the regions of the analysed geometries, where large flow velocity gradients were expected, the mesh size was concentrated. The program used is ANSYS CFX.

## RESULTS

### Study of the effect of rotational speed on system operation

The results of the simulation studies carried out for the geometry of two flat parallel plates (**Fig. 2a**) are shown in **Fig. 3**. At a rotational speed of up to 100 $min^{-1}$, no axial force is observed (**Fig. 3a**). When the speed is increased to 1000 $min^{-1}$, a small force (–1.47 N) arises, increasing at 10 000 $min^{-1}$ to close at –150 N. The justification for such an observation is the occurrence of centrifugal force acting on the modelled fluid. Mass forces acting in the axial direction tend to eject the fluid from the working zone, and the related force acts in such a way that it tries to bring the plates closer to each other. The results of the simulation did not show any differences in the observed values of axial force for the two modelled fluids. This indicates that the force value depends only on the density of the fluid and not on its rheological properties (**Table 1**), which confirms the correctness of the used numerical model.

In the case of friction torque, as expected, higher values were observed for fluid No. 5. In particular, at the highest analysed rotational speed, a torque close to 3 Nm was obtained, with such a result showing that the operation of the bearing system under such conditions would be highly ineffective.

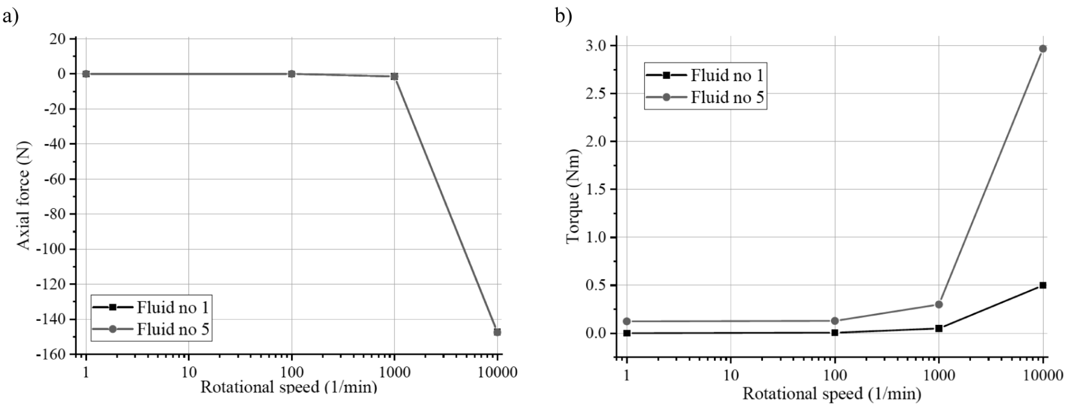



**Fig. 3. Plots of: a) normal force, b) friction torque, as a function of rotational speed**
Rys. 3. Wykresy: a) siły normalnej, b) momentu tarcia, w funkcji prędkości obrotowej

## STUDY OF THE INFLUENCE OF BEARING GEOMETRY SURFACE

A comparison of simulation results concerning axial force and friction torque for 10 tested bearing variants, for two speeds: 100 and 1000 min$^{-1}$, and two fluids (No. 1 and No. 5) is shown in **Fig. 4**.

Analysing the qualitative influence of rheological properties of the modelled fluids on the bearing operating conditions, one can pay attention to the significant similarity of the results obtained for each of the modelled fluids. Due to the maximization of axial force, the most favourable results were obtained for Geometries (e), (f),

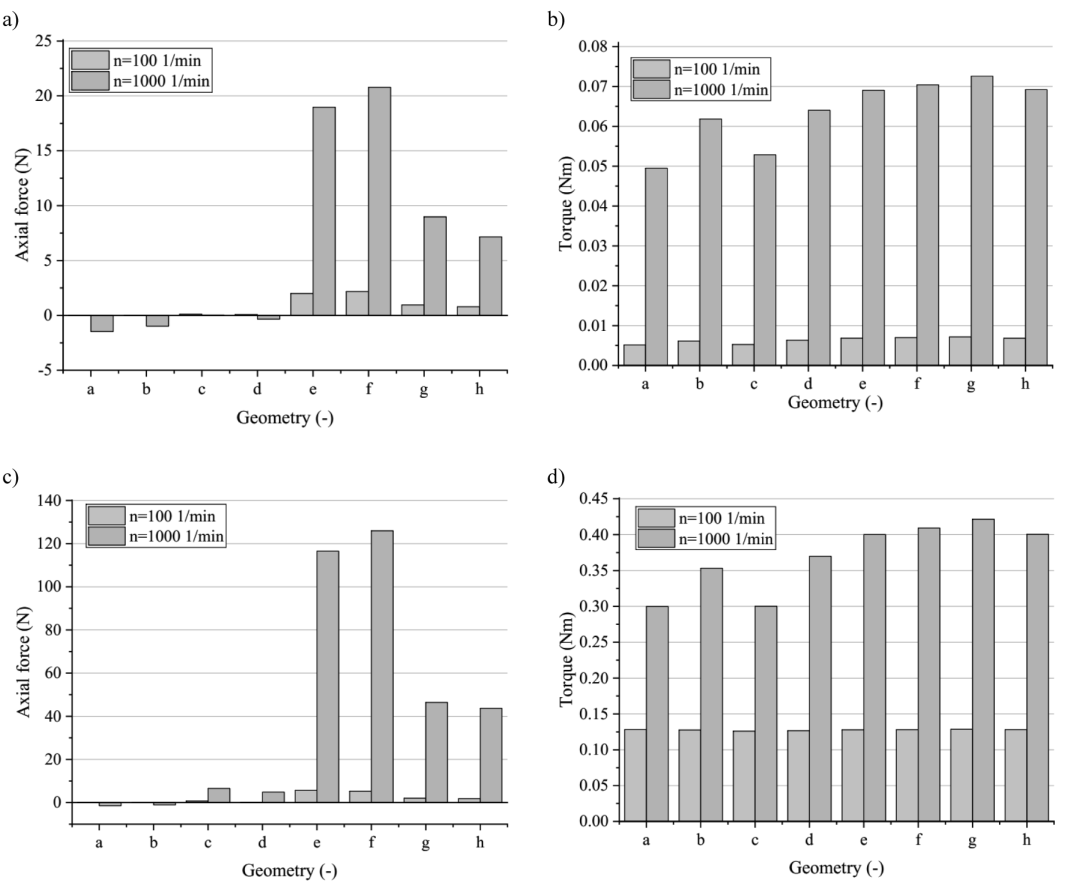



**Fig. 4. Results of simulation studies of axial force and friction torque for the tested geometries: a, b) fluid No. 1, c, d) fluid No. 5**
Rys. 4. Wyniki badań symulacyjnych siły osiowej i momentu tarcia dla badanych geometrii: a, b) ciecz nr 1, c, d) ciecz nr 5

(g), and (h). In the case of torque, the designs which promise the best results in thrust bearing are (e) to (h).

However, there are significant differences in terms of quantity, both in terms of axial force and torque. The effect of speed is visible here. A change in speed from 100 to 1000 min$^{-1}$ results in an average 25-fold increase in the observed axial force and a 3-fold increase in torque. Such proportions are maintained for each of the two tested fluids.

In order to compare the influence of fluid rheological properties on the obtained results, **Fig. 5** shows the quotient values of the measured parameters obtained for fluids No. 5 and No. 1.

The Bingham fluid model (No. 5), instead of Newton's (No. 1), results in a more than 2-fold increase in the axial force for 100 min$^{-1}$ and even 6 times for 1000 min$^{-1}$ (Geometries (e) to (h)). For Geometry (c), there is an increase in axial force by more than 160 times and for (d) a decrease by almost 15 times. Such large changes take place because, for these two geometries, very low values of the axial force were obtained in the case of modelling Newtonian fluid.

The influence of fluid rheological properties on the friction torque is shown in **Fig. 5b**. During the operation at the rotational speed of 100 min$^{-1}$, the increase in yield stress from 0 to 5 kPa (change from fluid No. 1 to No. 5) causes a value about 20 times higher, and at higher bearing speed (1000 min$^{-1}$), about 6 times. It can be seen that, for Geometries (e) to (h), increasing the rotational speed from 100 to 1000 min$^{-1}$ causes a 6-fold increase in axial force, but this is related with an approximately 6-fold increase in friction torque.

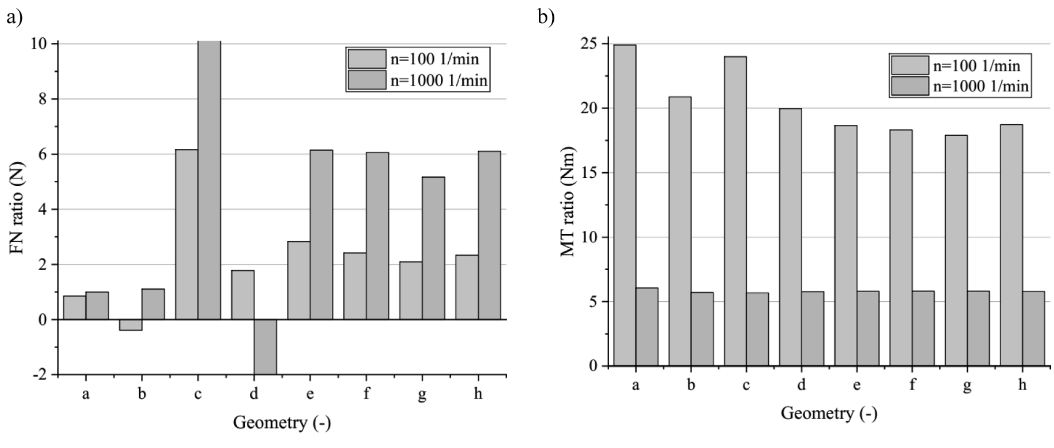



**Fig. 5. Plots of the relative change; a) axial force; b) friction moment**
Rys. 5. Wykresy względnej zmiany; a) siły osiowej, b) momentu tarcia

In order to analyse the behaviour of the tested geometries in more detail, especially in the context of qualitative explanation of the axial force, contour plots of total pressure distributions and vector diagrams of fluid flow velocity are presented in **Fig. 6**. The charts were developed for the case of modelling the bearing operating with fluid No. 5 for 100 min$^{-1}$. They are presented in the plane in which the highest velocity gradients occurred. In the diagrams, it is possible to distinguish places of negative pressure (shades of blue) which reduce the value of the observed axial force. The areas with overpressure (red) that are responsible for generating the lifting force are also visible.

For Cases (b), (c), and (d), relatively large negative pressure zones and relatively small areas of overpressure are observed, resulting in a small positive or negative axial force (see **Figs. 4c, d**).

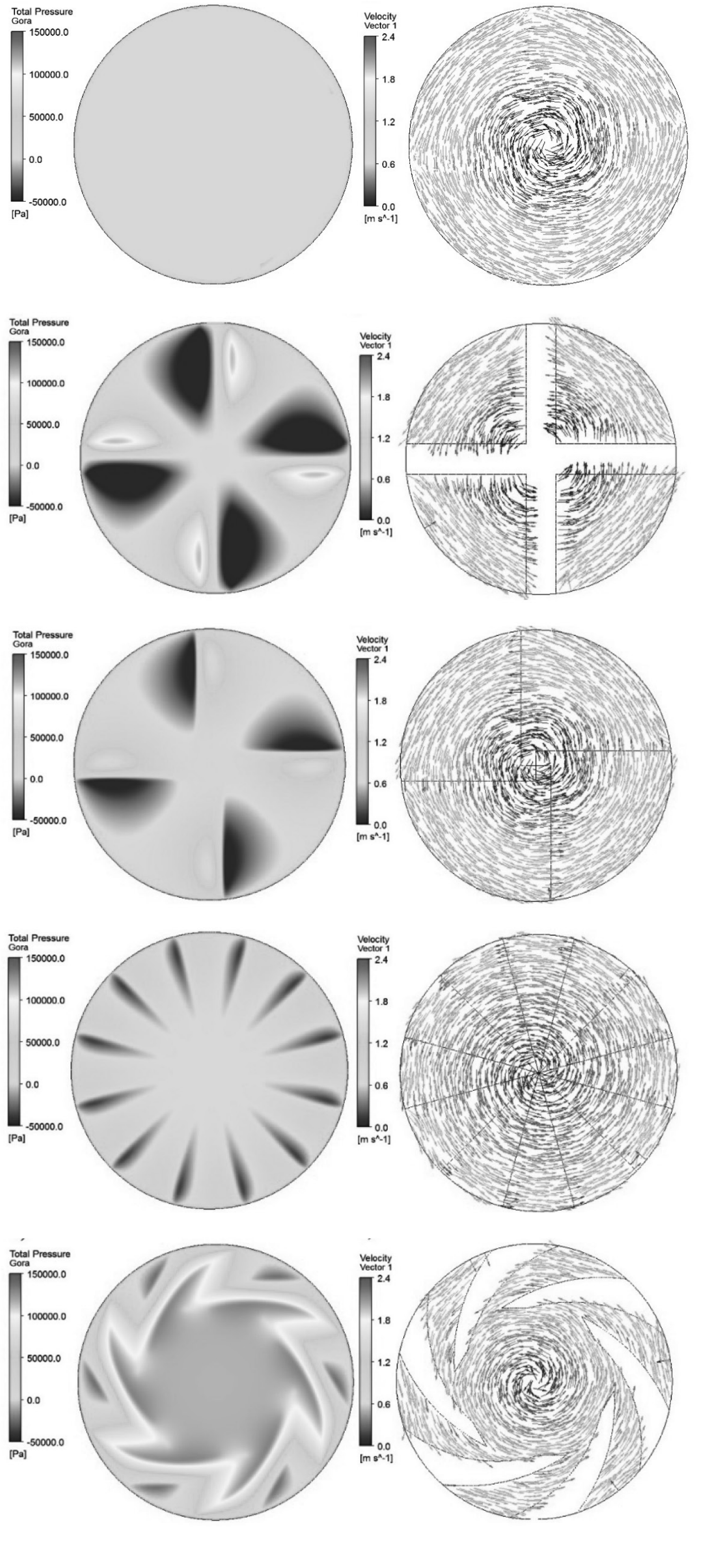
Bearing versions (e) and (f) were characterized by obtaining the highest values of axial forces (**Figs. 4c, d**), which is reflected in the contour plots of total pressure. The change in the number of outlets on the surface of the thrust plate did not significantly affect the distribution of flow velocity in the modelled arrangement.

The distributions of total pressure in bearings (g) and (h) indicate the presence of numerous regions of high and low pressure, but unlike (a) to (d), there is a large area of higher pressure in the central part, which results in a significant buoyancy force. Bearing (g), for whom a higher value of axial force in relation to (h) was obtained, showed the highest value of torque. This may be due to the presence of low-velocity areas of the fluid on most of the plate area.

## Study of the influence of the lubricant rheological properties on the system operation

In order to investigate the influence of rheological properties of the modelled magneto-rheological fluid on bearing functioning, further simulations were performed. They were carried out at the speed of 1000 min$^{-1}$ for the bearing variant type (f) because the highest force values were obtained for it. The results are shown in **Fig. 7**.

a)
b)
c)
d)
e)


Total Pressure
Gora
150000.0
100000.0
50000.0
0.0
-50000.0
[Pa]
Velocity
Vector 1
2.4
1.8
1.2
0.6
0.0
[m s^-1]

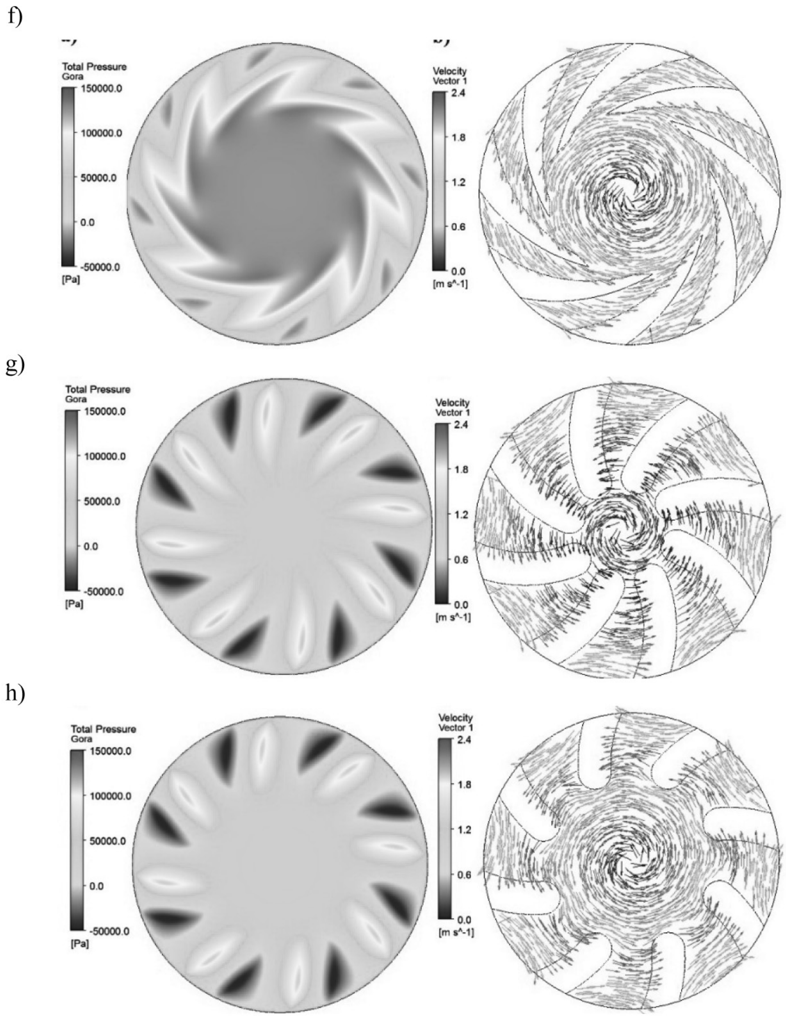



**Fig. 6. Contour diagrams of the spatial distribution of total pressure and the fluid velocity vector in the analysed bearings**

Rys. 6. Wykresy konturowe przestrzennego rozkładu ciśnienia całkowitego oraz wektora prędkości przepływu cieczy w analizowanych łożyskach

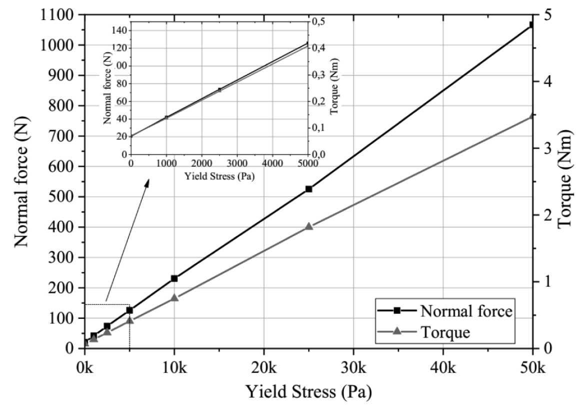



**Fig. 7. The relationship between the axial force and the friction torque as a function of the modelled fluid yield stress**

Rys. 7. Zależność siły osiowej oraz momentu tarcia od granicy płynięcia modelowanej cieczy

A linear dependence of the normal force and the friction torque on the yield stress of the modelled fluid is observed. The results indicate that probably higher normal forces can be obtained for higher values of magnetic field strength. In the range of yield stress 0–5 kPa, axial force in the range 20 to 120 N (12.5 to 75.5 kPa) with a change in resistance torque from 0.1 to 0.4 Nm is achieved.

## CONCLUSIONS

The paper presents a comparative analysis of the results of fluid flow simulation in a thrust bearing. The rheological parameters of the modelled fluid were chosen in such a way that its characteristics and the values of the parameters correspond to those typical for magnetorheological fluids.

The main result of the performed analyses is to show the possibility of generating axial forces due to the occurrence of hydrodynamic phenomena in the modelled bearing systems, as well as to indicate the most advantageous ways to modify the thrust plate surface.

For similar bearing systems operating with the same extortion, the use of a lubricating fluid characterized by the yield stress allows several times higher hydrodynamic load capacity values to be obtained than in the case of Newtonian fluid lubrication.

The study also shows that the increase in the modelled fluid yield stress favourably influences the hydrodynamic buoyancy force; however, it results in the expected increase in movement resistance of the modelled system.

## ACKNOWLEDGMENTS

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