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GENERALIZED NEWTONIAN FLUIDS AS LUBRICANTS IN THE HYDRODYNAMIC CONICAL BEARINGS – A CFD ANALYSIS

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Abstract

Additives, ageing or wear and impurities can cause, that relationship between shear stress and shear rate in a lubricating oil is or becomes non-linear, and due to this, a significant change in the values of operating parameters of slide hydrodynamic bearings may occur. It is important to take into account such dependence during design and simulations of slide bearings. The calculations, which consider the non-linear properties of the lubricating oil, can be carried out by adopting the generalized Newtonian fluid models.

This paper shows the result of CFD simulation of slide conical bearings hydrodynamic lubrication, assuming that the lubricating oil behaves as a generalized Newtonian fluid. The hydrodynamic pressure distributions, load carrying capacities and friction torques were calculated for bearings lubricated with different types of generalized Newtonian fluids and the obtained data were compared. In the study, the following models of fluids were adopted: the Power-law fluid (Ostwald-de Waele), the Cross fluid and the Carreau fluid. The coefficients of mentioned relationships were determined by fitting the curves described by each model to the experimental data using the least squares approximation method. The calculations of hydrodynamic pressure distributions, load carrying capacities and friction torques were carried out using the commercial CFD software Ansys Fluent from the Ansys Workbench 2 platform.

Keywords: *hydrodynamic conical bearing, generalized Newtonian fluid, CFD, non-Newtonian oil, pressure distribution, dynamic viscosity, ferro-oil*

1. Introduction

Changes in physical properties of the lubricating oil in the sliding bearings, and in particular – changes of its viscosity value, significantly affect the hydrodynamic pressure generated in the bearing lubrication gap, thus causing the changes in the operating parameters of bearing. Considering the non-classical lubricating oils with non-Newtonian properties, such as, e.g. containing additives, impurities or ferromagnetic particles (i.e. ferro-oils) with substances, which prevent their coagulation [5], changes in the viscosity of the lubricant in relation to the value of the shear rate, should be also included. Numerical analysis of hydrodynamic lubrication of slide bearings in these cases can be carried out assuming, that the lubricating oil has the properties of a generalized Newtonian fluid [4], then the dependence between the shear stress τ [Pa] and the shear rate $\dot{\gamma}$ [1/s] can be written as follows:

$$\tau = \eta_1(\dot{\gamma}) \cdot \dot{\gamma} . \tag{1}$$

The quantity η_1 [Pa·s] is an effective dynamic viscosity as a function of shear rate. An appropriate definition in the simulations of the effective viscosity function is crucial in certain cases to achieve the values close to the actual. In this paper is shown, how the adopted viscosity model affects the values obtained in the simulations of the hydrodynamic lubrication of the conical slide bearing. Three relationships for generalized Newtonian fluids were considered:

1. for the Ostwald-de Waele type of fluid (power-law fluid) [9, 10]:

$$\eta_1(\dot{\gamma}) = K \cdot \dot{\gamma}^{n-1}, \tag{2}$$

where K [Pa·s^n] is the flow consistency index and n [-] is the flow behaviour (power-law) index,

2. for the Cross type of fluid [10]:

$$\eta_1(\dot{\gamma}) = \frac{\eta_0}{1+(\mu\cdot\dot{\gamma})^{1-n}}\,. \tag{3}$$

where η_0 [Pa·s] is the zero-shear-rate viscosity, μ [s] is a parameter called *"natural time"* and n [-] is a power-law index,

3. for the Carreau type of fluid [10]:

$$\eta_1(\dot{\gamma}) = \eta_\infty + (\eta_0-\eta_\infty)\cdot\left(1+^2\ \mu^2\cdot\dot{\gamma}^2\right)^{(n-1)/2}, \tag{4}$$

where η_0 [Pa·s] is the zero-shear-rate viscosity, η_∞ [Pa·s] is the infinite-shear-rate viscosity, μ [s] is a „*natural time*" and n [-] is a power-law index.

The determination of the coefficients for each of the models involved fitting the curves described by these models to the experimental data by the least squares approximation method.

The result of fitting is shown in Fig. 1.

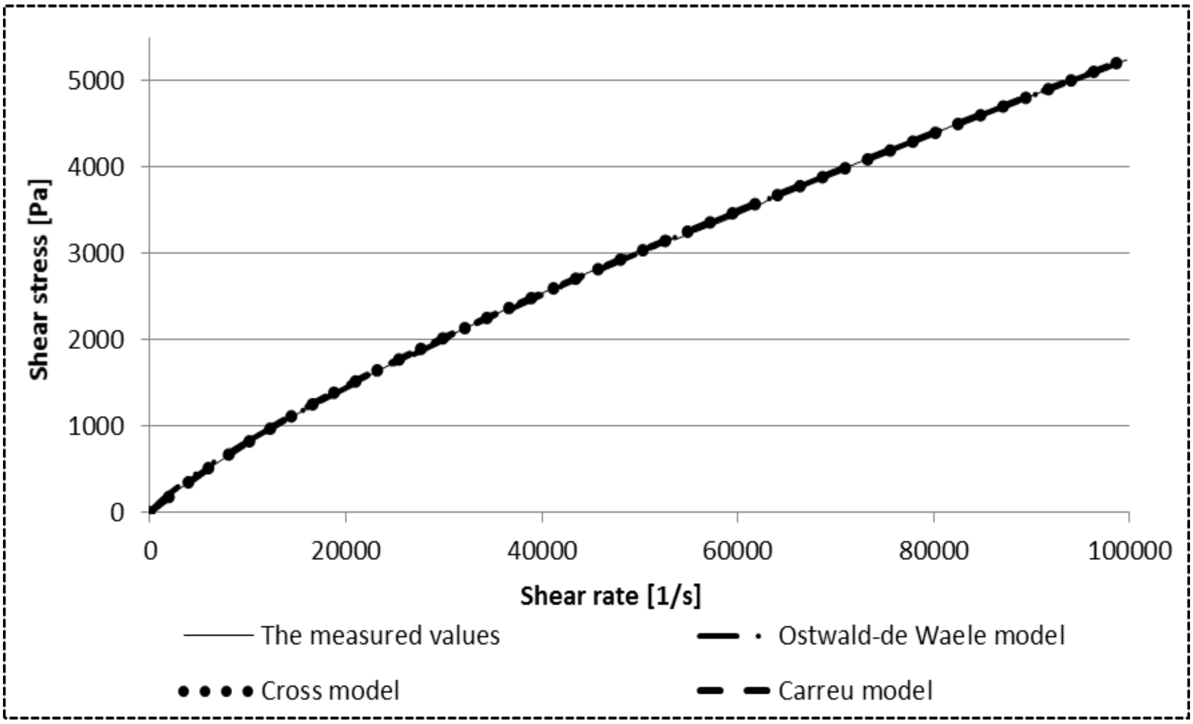

Fig. 1. Fitting of curves described by the considered relationships for generalized Newtonian fluids to the values obtained experimentally

The considered experimental results are related to the ferro-oil, i.e. oil containing magnetic particles and a surfactant [5] (a substance that prevents coagulation of magnetic particles). The measurements were done with the Haake Mars III rheometer (with the concentric cylinders geometry) at ambient pressure (approx. 0.1 MPa) and without the action of an external magnetic field. A special cup and a thermostatic system were used to maintain a constant temperature of the sample, which was (80.0 ± 0.2)°C. The volume fraction of magnetic particles in the tested oil was 6% (in the papers [1, 2] can be found information on an impact of the concentration of magnetic particles in the oil on its viscosity). The measurements could be carried out to a shear rate of about 10^5 [1/s] (above this value, the maximum allowable torque on the rheometer rotor detector was exceeded for this sample at mentioned conditions). Despite the absence of an external magnetic field, just the presence of the magnetic particles and surfactant causes the non-Newtonian properties of the investigated oil are highly relevant.

For the above-described models, the following values of the coefficients were obtained:

- for the Ostwald-de Waele relation: $K = 0.523$ Pa·s^n and $n = 0.800$,
- for the Cross relation: $\eta_0 = 0.0928$ Pa·s, $\mu = 3.030 \cdot 10^4$ s and $n = 1.075$,
- For the Carreau relation: $\eta_0 = 0.0868$ Pa·s, $\eta_\infty = 0.0299$ Pa·s, $\mu = 8.001 \cdot 10^{-5}$ s and $n = 0.559$.

In the simulations, the viscous heating effect and the influence of temperature changes on the viscosity of the oil also were taken into account. The relationship describing the change in viscosity values depending on the shear rate and temperature is as follows:

$$\eta(\dot{\gamma}, T) = \eta_1(\dot{\gamma}) \cdot H(T), \tag{5}$$

where:

$$H(T) = \exp\left[\alpha_T \cdot \left(\frac{1}{T} - \frac{1}{T_\alpha}\right)\right], \tag{6}$$

is the factor dependent on temperature, while $\alpha_T = E_a/R$ is the ratio of the activation energy E_a [J/kmol] to the thermodynamic constant R = 8314 J/(kmol·K) and T_α [K] is a reference temperature for which $H(T) = 1$.

2. Calculation and results

This research concerns the slide conical bearing with a length of L = 50 mm with a radius of a shaft at lowest cross-section of R = 50 mm. The geometry of investigated bearing is shown in Fig. 1. The angle between the cone generating line and its axis of symmetry is 10° (both for shaft and sleeve) and that gives α = 80°. The radial clearance of that bearing is $\varepsilon = R' - R = 0.02$ mm, where R' is the radius of bearing sleeve at its lowest cross-section. The radial clearance ε is constant value along the bearing (because of the same value of angle α for the shaft and sleeve). The bearing is without misalignment, so the axis of the cone of bearing shaft is parallel to the axis of the cone of bearing sleeve.

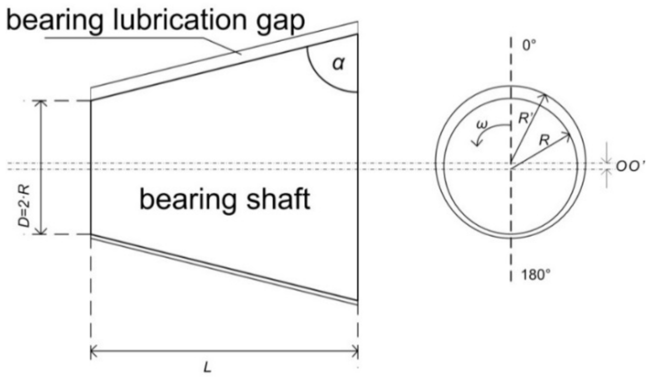

Fig. 2. The geometry of the considered conical bearing

The relative eccentricity ratio λ is defined as [6]:

$$\lambda = \frac{OO'}{\varepsilon}, \tag{7}$$

where OO' is the bearing eccentricity – it is a distance between the axis of the shaft and axis of the sleeve. The adopted assumptions:

- bearing operates in a steady state (constant value of rotational speed, no vibrations, laminar and incompressible flow of lubricating oil, constant value of relative eccentricity),

- smooth and rigid surfaces of the bearing shaft and sleeve, without deformation,
- there is no slip of lubricating oil at bearing surfaces,
- pressure on the side surfaces of bearing gap is equal to the ambient pressure,
- the temperature of the surface of bearing shaft and also the supplying oil is 80°C,
- the bearing sleeve is made of steel and conducts heat from bearing lubrication gap to the surroundings – the parameters of bearing sleeve material: density ρ = 8030 [kg/m^3], specific heat c_p = 503 [J/(kg·K)], heat conduction coefficient κ = 16.27 [W/(m·K)], sleeve thickness δ = 1 [mm],
- the parameters of the lubricating oil: density 1116 [kg/m^3], specific heat 1006 [J/(kg·K)], heat conduction coefficient 0.025 [W/(m·K)],
- the Gümbel boundary condition [6] (half-Sommerfeld condition) for the end of the oil film was used in this simulation.

The geometry of the bearing and mesh were prepared with the ANSYS Workbench 2 platform and the Fluent CFD module was used to calculate the solution. The pressure based SIMPLEC algorithm was applied (Green-Gaus node based, second order pressure, the momentum: second order upwind, the energy: second order upwind). Simulations were carried out for bearings with the relative eccentricities: 0.4, 0.5, 0.6, and with the shaft rotational speeds: 500, 3200, 7200 rpm, for which the hydrodynamic pressure distributions were obtained. The determined values of maximum hydrodynamic pressure p_{max} in the bearing lubrication gap, the transversal C_t and longitudinal C_l load carrying capacities and the frictional angular momentum M_l for each adopted model, are presented in Tab. 1.

Tab. 1. The values of maximum hydrodynamic pressure p_{max} in the bearing lubrication gap, the transversal Ct and longitudinal C_l load carrying capacities, the frictional angular momentum M_l

Viscosity model	λ	0.4			0.5			0.6		
	n_r [rpm]	500	3200	7200	500	3200	7200	500	3200	7200
Ostwald-de Waele	p_{max} [MPa]	3.78	7.72	7.72	6.05	12.14	11.61	10.10	19.78	18.50
	C_t [kN]	8.33	18.74	21.48	12.63	27.28	29.49	19.47	40.44	41.64
	C_l [kN]	1.77	4.16	4.69	2.65	5.96	6.43	4.02	8.65	9.00
	M_l [N·m]	2.80	5.90	5.95	2.95	6.20	6.26	3.16	6.63	6.68
Cross	p_{max} [MPa]	3.96	10.18	10.57	6.59	16.37	10.57	11.55	27.38	26.32
	C_t [kN]	8.51	24.49	29.52	15.64	36.18	29.52	21.32	54.64	58.28
	C_l [kN]	1.80	5.44	6.45	2.77	7.88	6.45	4.35	11.62	12.55
	M_l [N·m]	2.80	5.90	5.95	2.95	6.20	6.26	3.24	8.42	8.69
Carreau	p_{max} [MPa]	3.81	8.43	8.74	6.14	13.41	8.74	10.35	22.18	21.52
	C_t [kN]	8.38	20.34	24.15	12.77	29.86	24.15	19.82	44.72	47.71
	C_l [kN]	1.78	4.51	5.27	2.68	6.51	5.27	4.09	9.54	10.28
	M_l [N·m]	2.82	6.31	6.53	2.98	6.66	6.53	3.21	7.16	7.39

The relative differences between presented values for the concerned viscosity models are greater for higher rotational speeds of bearing shaft, i.e. at higher shear rates of lubricating oil, e.g. by comparing the results obtained when λ = 0.6, with respect to the values obtained from the Ostwald-de Waele model, there is:

Tab. 2. The relative difference of obtained values in relation to the Ostwald-de Waele model for bearing with λ = 0.6

λ = 0.6	Cross model			Carreau model		
n_r [rpm]	500	3200	7200	500	3200	7200
Δp_{max} [%]	14.3	38.4	42.3	2.5	12.1	16.3
ΔC_t [%]	9.5	35.1	40.0	1.8	10.6	14.6
ΔC_l [%]	8.1	34.3	39.5	1.7	10.2	14.2
ΔM_l [%]	2.5	27.0	30.0	1.5	7.9	10.5

3. Discussion

The results show, that the examined viscosity models give significantly different values, particularly in cases where the high shear rates occur. This is due to the effect of shear rate on the dynamic viscosity. The curves described by these models were fitted to the experimental data, but in the shear rate range up to ~10^5 [1/s], while the shear rates of oil in the investigated bearings were much greater, e.g. for bearing with $\lambda = 0.6$ and $n_r = 7200$ rpm the shear rate of lubricating oil at the lowest gap height location was about $\dot{\gamma}_{max} = 37.7 \cdot 10^5$ [1/s]. Even though, that over the measured range of shear rates, these models well reflect the change in viscosity, the extrapolation in such a large extent caused great differences between the obtained values. The shear stress dependence on shear rate in the range up to $30 \cdot 10^5$ [1/s], on the basis of the mentioned viscosity models and the determined coefficients, is shown in Fig. 2.

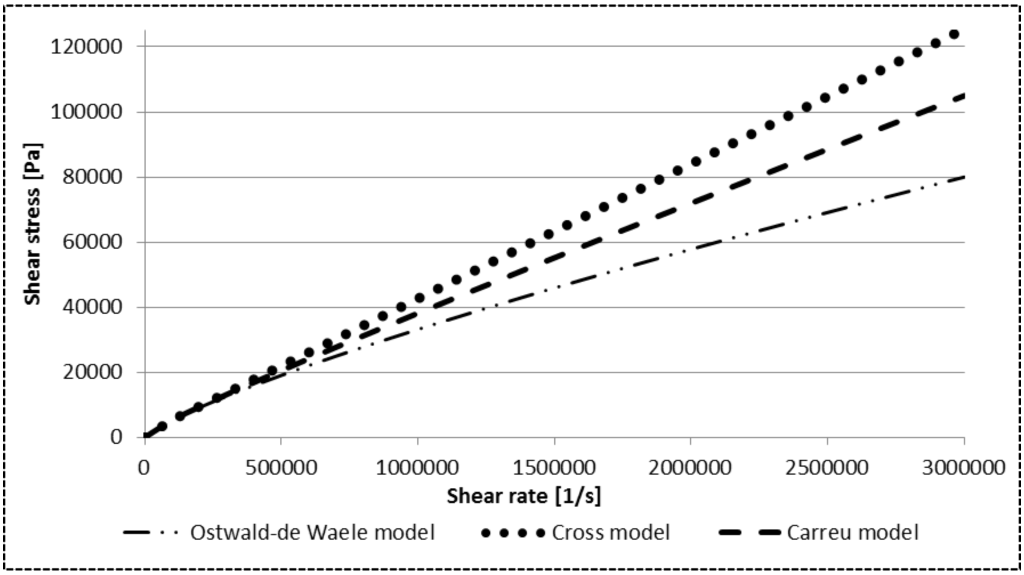

Fig. 3. The extrapolation of shear stress dependence on shear rate based on the concerned models

This graph shows that with increasing share rate, the difference between the computed values of shear stress become greater. How this affects the calculation of the viscosity values, in accordance with the examined models, is shown in Fig. 4. The comparison of the value of viscosity determined with the Corss model, to a value determined by the Ostwald-de Waele model, at shear rate $30 \cdot 10^5$ [1/s], gives the difference of $\Delta\eta = 57\%$.

The presented charts show only the changes in the viscosity from the shear rate at constant reference temperature $t_\alpha = 80$°C, while the simulations also includes the effect of temperature on the viscosity according to the Eq. 6, which shows, that with increasing temperature the viscosity decreases its value. The internal friction of lubricating oil at high shear rates produces large quantities of heat, which increases the temperature. The maximum values of temperature generated in the lubricating oil, for bearing with $\lambda = 0.6$, are shown in Tab. 2. The observed differences of temperature are due to the different viscosity values, which is the measure of internal friction.

Tab. 3. The maximum value of temperature generated in the lubricating oil, for bearing with $\lambda = 0.6$

model	n_r [rpm]		
	500	3200	7200
Ostwald-de Waele	83°C	114°C	145°C
Cross	83°C	123°C	162°C
Carreau	83°C	116°C	150°C

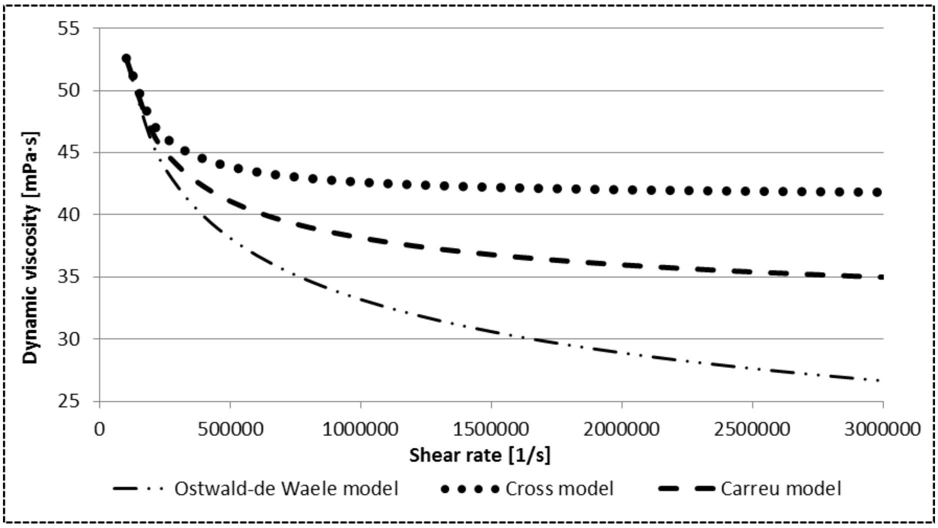

Fig. 4. The viscosity vs. shear rate, calculated with accordance with the investigated models and the determined coefficients

In some cases, for example at $\lambda = 0.5$, for Cross and Carreau models, the drop in oil viscosity value due to temperature rise caused, that the pressure and load carrying capacities at 7200 rpm have lower values, than at 3200 rpm.

The results show, that the effect of shear rate on the dynamic viscosity of the lubricating oil is a critical issue during simulation and determination of operating parameters of the conical slide bearing. It is important to select the appropriate model describing the physical properties of the lubricating oil. This article only includes the effects of shear rate and temperature on the viscosity of the oil; however, the effect of pressure may also be important [3]. Furthermore, the ferro-oils are liquids, which viscosity depends strongly on the magnetic field and the magnetic particles concentration [1], but ferro-oils in a magnetic field may also exhibit viscoelastic properties. Such a liquid begins to flow only when the stress exceeds value of yield stress τ_o, then the dependence of stress on shear rate could be represented, for example, as Herschel-Bulkley fluid [7, 8, 10]:

$$\tau = \tau_o + K \cdot \dot{\gamma}^n . \tag{8}$$

The modelling of the hydrodynamic lubrication of bearings, in which the ferro oil is a lubricant, is especially important, because it gives the potential to impact with the external magnetic field and control, to a certain extent, the operating parameters of the bearing.

4. Conclusions

1. The effects of temperature and shear rate are important factors influencing the parameters of the hydrodynamic conical bearing.
2. The ferro-oils, even without the presence of an external magnetic field, exhibit non-Newtonian properties.
3. The non-Newtonian lubricating oils can be simulated in the numerical calculations as the generalized Newtonian fluids.
4. The selection of the appropriate viscous model is essential in the numerical calculations of hydrodynamic lubrication with non-Newtonian oils.
5. The determination of the coefficients for the selected model should be carried out in an appropriate range of shear rates – but there occurs the limitation associated with the technical capabilities of a measuring equipment.
6. The modern CFD software is a good tool, which enables to study the effects of various physical properties of the lubricating oil and their influence on the hydrodynamic lubrication of slide bearings.

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