PRESSURE AND CAPACITY FORCE IN SLIDE JOURNAL PLANE BEARING LUBRICATED OIL WITH MICROPOLAR STRUCTURE

Paweł Krasowski

Gdynia Maritime University, Faculty Marine Engineering
Morska Street 81-87,81-225 Gdynia, Poland
phone:+48 58 6901331, fax:+48 58 6901399
e-mail:pawkras@am.gdynia.pl

Abstract

Present paper shows the results of numerical solution Reynolds equation for laminar, steady oil flow in slide plane bearing gap. Lubrication oil is fluid with micropolar structure. Properties of oil lubrication as of liquid with micropolar structure in comparison with Newtonian liquid, characterized are in respect of dynamic viscosity additionally dynamic couple viscosity and three dynamic rotation viscosity. Under regard of build structural element of liquid characterized is additionally microinertia coefficient. In modelling properties and structures of micropolar liquid one introduced dimensionless parameter with in terminal chance conversion micropolar liquid to Newtonian liquid. The results shown on diagrams of hydrodynamic pressure in dimensionless form in dependence on coupling number N^2 and characteristic dimensionless length of micropolar fluid Λ_1. Presented calculations are limited to isothermal models of bearing with infinite breadth.

Especially, geometry schema of the slide journal plane bearing gap, the dimensionless pressure distributions p_1 in dependence on coupling number N^2, the dimensionless pressure distributions p_1 in dependence on characteristic dimensionless length of micropolar fluid, the dimensionless maximal pressure p_{1m} in dependence on coupling number N^2, gap convergence coefficient are presented in the paper.

Keywords: *micropolar lubrication, journal plane bearing, hydrodynamic pressure*

1. Introduction

Presented article take into consideration the laminar, steady flow in the crosswise cylindrical slide plane bearing gap. Non-Newtonian fluid with the micropolar structure is a lubricating factor. Materials engineering and tribology development helps to introduce oils with the compound structure (together with micropolar structure) as a lubricating factors. Exploitation requirements incline designers to use special oil refining additives, to change viscosity properties. As a experimental studies shows, most of the refining lubricating fluids, can be included as fluids of non-Newtonian properties with microstructure [3-4, 6]. Presented work dynamic viscosity of isotropic micropolar fluid is characterized by five viscosities: shearing viscosity η (known at the Newtonian fluids), micropolar coupling viscosity κ and by three rotational viscosities bounded with rotation around the coordinate axes. This kind of micropolar fluid viscosity characteristic is a result of essential compounds discussed in works [3] and [4]. Regarding of limited article capacity please read above works. In presented flow, the influence of lubricating fluid inertia force and the external elementary body force field were omitting [3-4].

2. Reynolds equation and hydrodynamic pressure

The constant viscosity of micropolar oil, independent from thermal and pressure condition in the bearing. Quantity of viscosity coefficient depends on shearing dynamic viscosity η, which is decisive viscosity in case of Newtonian fluids. Reference pressure p_0 is also described with this viscosity, in order to compare micropolar oils results with Newtonian oil results. In micropolar oils decisive impact has quantity of dynamic coupling viscosity κ [1, 3]. In some works concerning bearing lubrication with micropolar oil, it's possible to find the sum of the viscosities as a micropolar dynamic viscosity efficiency. In presented article coupling viscosity was characterized with coupling number N^2, which is equal to zero for Newtonian oil:

$$N = \sqrt{\frac{\kappa}{\eta + \kappa}}, \qquad 0 \le N < 1. \tag{1}$$

Quantity N^2 in case of micropolar fluid, define a dynamic viscosity of coupling share in the oil dynamic viscosity efficiency. From the coupling number N^2 we can determine both dynamic viscosity ratio, which is dimensionless micropolar coupling viscosity:

$$\kappa_1 = \frac{\kappa}{\eta} = \frac{N^2}{1 - N^2}, \qquad \kappa_1 \ge 0. \tag{2}$$

From the dynamic rotational viscosities α, β, γ at the laminar lubrication, individual viscosities are compared to viscosity γ, which is known as the most important and it ratio to shearing viscosity η is bounded to characteristic flow length Λ, which in case of Newtonian flow assume the zero quantity. Dimensionless quantity of micropolar length Λ_1 and micropolar length Λ is defined:

$$\Lambda = \sqrt{\frac{\gamma}{\eta}}, \qquad \Lambda\Lambda_1 = \varepsilon. \tag{3}$$

Dimensionless micropolar length, Λ_1 in case of Newtonian oil approach infinity.

Lubricating gap is characterize by following geometric parameters: maximal gap height h_o, minimal gap height h_e , gap length L and gap width b (Fig. 1). In presented model the following assumption were made: lubricating gap dimensions along it's width of mating surfaces remain

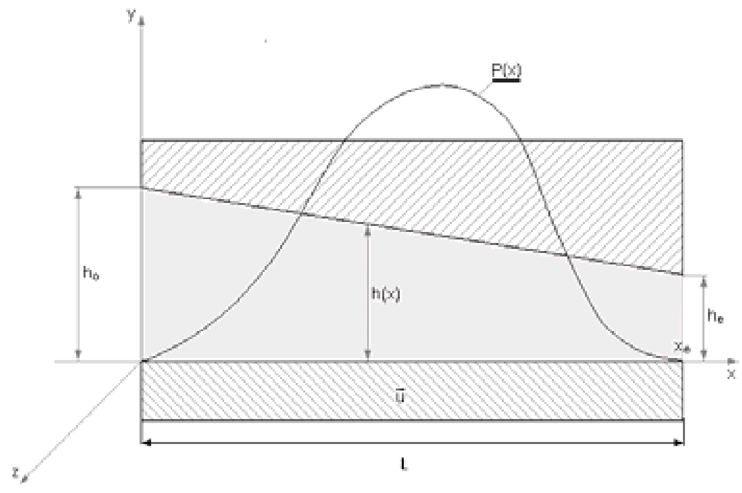

Fig. 1. Geometry schema of the slide journal plane bearing gap

identical. Lubricating gap height after gap length was described in Cartesian co-ordinate system by the following dimensionless form:

$$h_1(x_1) = \varepsilon - (\varepsilon - 1)x_1 \text{ for } 0 \le x_1 \le 1. \tag{4}$$

Dimensionless values [2],[3] that characterize lubricating gap are: length coordinate x_1 , gap height coordinate h_1 and gap convergence coefficient ε :

$$x_1 = \frac{x}{L}; \qquad h_1 = \frac{h}{h_0}; \qquad \varepsilon = \frac{h_e}{h_0}. \tag{5}$$

Reynolds equation for stationary flow of laminar micropolar fluid in slide plane bearing gap can be present [1-2, 7] in dimensional form:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\Phi(\Lambda,N,h)\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\Phi(\Lambda,N,h)\frac{\partial p}{\partial z}\right)=6\frac{dh}{dx}\ , \tag{6}$$

$\Phi(\Lambda,N,h)$ function in form (6) when in case of the Newtonian fluid it has a value 1 and the Reynolds equation (6) change into a non-Newtonian fluid equation.

$$\Phi(\Lambda,N,h)=1+12\frac{\Lambda^2}{h^2}-6\frac{N\Lambda}{h}\coth\left(\frac{Nh}{2\Lambda}\right). \tag{7}$$

Reynolds equation (6) can be presented in dimensionless form [1],[7] using the method of changing into this values:

$$\frac{\partial}{\partial x_1}\left(\Phi_1(\Lambda_1,N,h_1)\frac{\partial p_1}{\partial x_1}\right)+\frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left(\Phi_1(\Lambda_1,N,h_1)\frac{\partial p_1}{\partial z_1}\right)=6\frac{dh_1}{dx_1}\ , \tag{8}$$

$$\text{for } 0\le x_1\le 1;\quad 0\le y_1\le h_1;\quad -1\le z_1\le 1$$

where:

$$\Phi_1=h_1^3+12\frac{h_1}{\Lambda_1^2}-6\frac{Nh_1^2}{\Lambda_1}\coth\left(\frac{h_1N\Lambda_1}{2}\right). \tag{9}$$

Additional assumptions were made[2]: the dimensionless value for pressure p_1, width L_1 and for remaining coordinates y_1 and z_1 according to the following designation:

$$\begin{aligned} p&=p_0\,p_1\ , & b&=L\,L_1\ ,\\ z&=b\,z_1\ , & y&=h_e y_1\ , \end{aligned} \tag{10}$$

Reference pressure p_o caused by linear velocity U of slide bearing (11) taking into consideration dynamic viscosity of shearing η and relative play ψ ($10^{-4}\le\psi\le10^{-3}$):

$$p_0=\frac{U\eta}{\psi L},\qquad \psi=\frac{h_e}{L}. \tag{11}$$

Below solutions (8) for infinity breadth bearing is presented. The pressure distribution function in case of the micropolar lubrication has a form:

$$p_1(x_1)=6\int_0^{x_1}\frac{h_1-C_1}{\Phi_1(\Lambda_1,N,h_1)}dx_1\,;\ C_1=\frac{\int_0^1\frac{h_1}{\Phi_1}dx_1}{\int_0^1\frac{1}{\Phi_1}dx_1}. \tag{12}$$

In the boundary case of lubricating Newtonian fluid, pressure distribution function is a pressure $p_{1N}(x_1)$.

$$\lim_{\substack{N\to0\\ \Lambda_1\to\infty}}\Phi_1=h_1^3\qquad \lim_{\substack{N\to0\\ \Lambda_1\to\infty}}C_1=1\qquad p_{1N}(x_1)=6\int_0^{x_1}\frac{h_1-1}{h_1^3}dx_1$$

$$p_{1N}=\frac{6(\varepsilon-1)(1-x_1)x_1}{(\varepsilon+1)(\varepsilon-\varepsilon x_1+x_1)^2} \tag{13}$$

Example numerical calculation were made for the infinity breadth bearing with convergence coefficient ε: $\varepsilon_{opt} = 1+\sqrt{2}$ end ε = 1,4 denoted continue and discontinue lines.

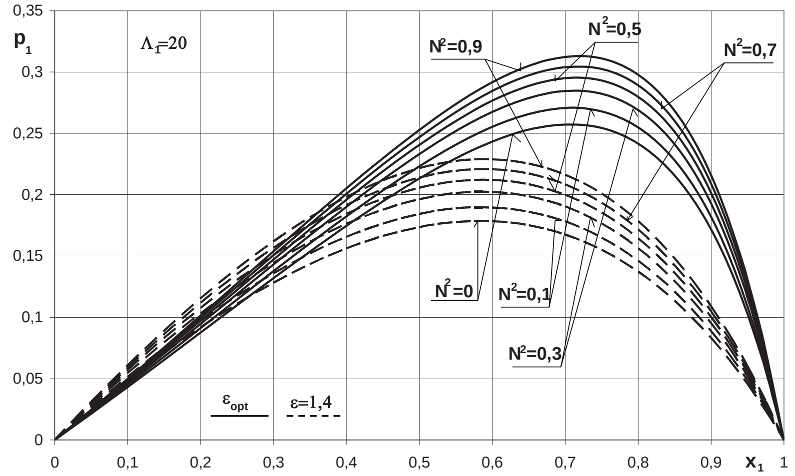

Fig. 2. The dimensionless pressure distributions p_1 in direction x_1 in dependence on coupling number N^2 by micropolar ($N^2>0$) and Newtonian ($N^2=0$) lubrication for characteristic dimensionless length of micropolar fluid $\Lambda_1=20$, gap convergence coefficient ε_{opt} and $\varepsilon=1.4$

The influence of coupling number N^2 and the influence of dimensionless micropolar length Λ_1 on hydrodynamic pressure distribution in the bearing length direction are analyzed. At the Fig. 2 pressure distribution for individual coupling numbers at constant micropolar length $\Lambda_1=20$. The pressure increase effect is caused by oil dynamic viscosity efficiency increase as a result of coupling viscosity κ. At $N^2= 0.5$, coupling viscosity is equal to shearing viscosity. Pressure graph in the Fig. 2 for micropolar oil lubrication ($N^2>0$) find themselves above the pressure graph at the

Newtonian oil lubrication ($N^2=0$). Pressure distribution is higher for higher coupling number. It is caused by oil viscosity dynamic efficiency. In the Fig. 3 the course of dimensionless pressure $p_1$for few micropolar length quantity Λ_1 is shown: decrease of this parameter determine the increase of micropolar oil rotational dynamic viscosity. Pressure distribution are presented at the constant coupling number $N^2=0.4$. Newtonian oil pressure in the course 1 and rotational viscosity increase determines the pressure distribution increase and is caused, because both the oil flow and microratation velocities are coupled. Quantities of coupling number N^2 and dimensionless micropolar length where taken from works [1-2].

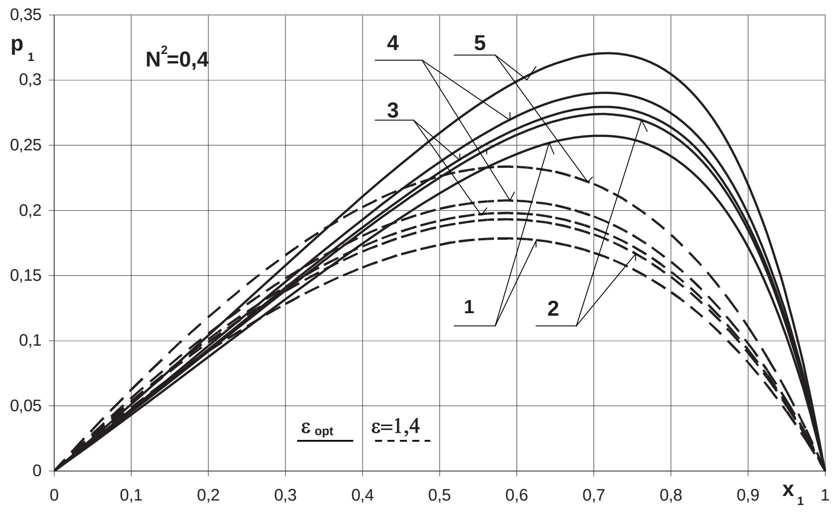

Fig. 3. The dimensionless pressure distributions p_1 in direction x_1 in dependence on characteristic dimensionless length of micropolar fluid Λ_1: 1) Newtonian oil, 2)$\Lambda_1=40$, 3) $\Lambda_1=30$, 4) $\Lambda_1=20$, 5) $\Lambda_1=10$, for coupling number $N^2=0.4$, gap convergence coefficient ε_{opt} and $\varepsilon=1.4$

Based on given hydrodynamic pressure distribution p_1 on direction x_1 , the numerical quantities of maximal pressure p_{1m} and the length coordinate x_{1m} (at the maximal position) were obtain. Quantities p_{1m} are presented in the Fig. 4 in the coupling Number N^2 function for chosen

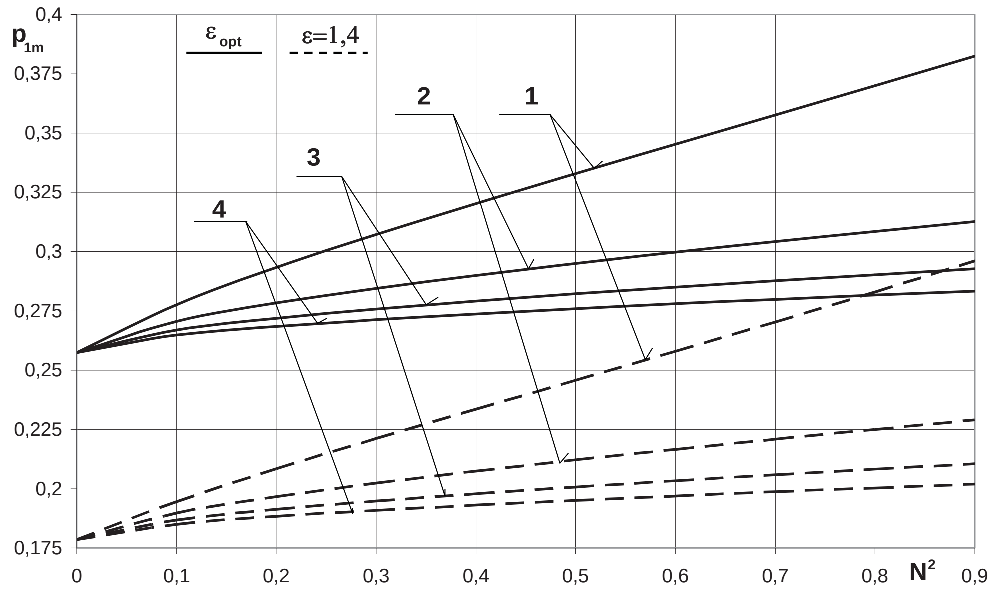

Fig. 4. The dimensionless maximal pressure p_{1m} in dependence on coupling number N^2 for characteristic dimensionless length of micropolar fluid Λ_1:1) Λ_1=10, 2) Λ_1=20, 3) Λ_1=30, 4) Λ_1=40 , gap convergence coefficient ε_{opt} and ε =1.4

micropolar length Λ_1. All lines are coming out from the maximal pressure point in case of Newtonian fluid flow. We observe maximal pressure increase when the coupling number N^2 increases (coupling viscosity increases κ) and the micropolar length decreases Λ_1 (rotational viscosity increases γ).

3. Capacity forces

Hydrodynamic capacity force in the bearing comes from the hydrodynamic pressure integral on bearing surface slide. In dimensionless form:

$$W_1 = \frac{W}{W_0} = \int_0^1 p_1(x_1)dx_1 \; ; \; W_0 = bLp_0 , \tag{14}$$

where: W_0 - characteristic value of capacity force.

In case of lubricating oil flow with constant viscosity independent from pressure, the capacity load W_{10} by stationary flow is determined [2] by equation:

$$W_{10} = \frac{6}{(\varepsilon-1)^2}\left(\ln\varepsilon - 2\frac{\varepsilon-1}{\varepsilon+1}\right). \tag{15}$$

Quantities capacity forces W_1 are presented in the Fig. 5 in the coupling Number N^2 function for chosen micropolar length Λ_1. All lines are coming out from the capacity force point in case of Newtonian fluid flow. We observe maximal capacity force when the coupling number N^2 increases (coupling viscosity increases κ) and the micropolar length decreases Λ_1 (rotational viscosity increases γ) . Full range of coupling number change, that covers the range [0;1), apply to coupling viscosity κ change from small to very high quantities. In most of the works, the hydrodynamic parameters of the bearing graphs are given in the function, which is nonlinear scale for coupling

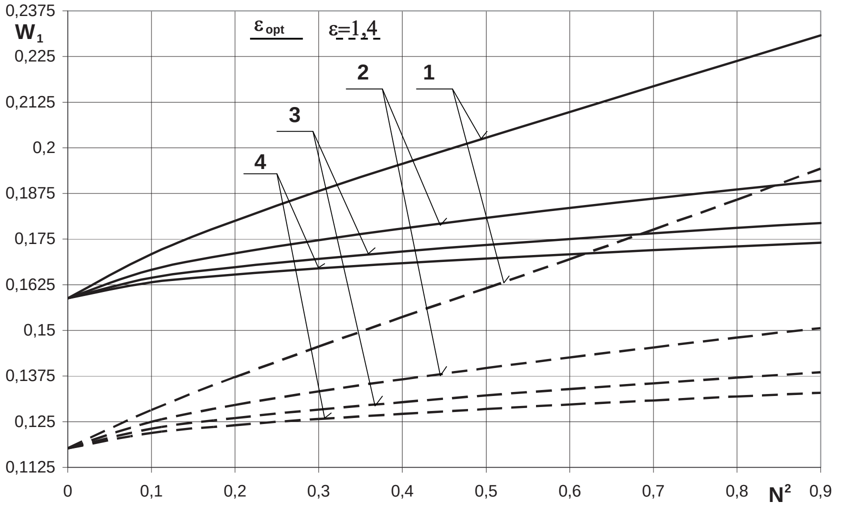

Fig. 5. The dimensionless capacity forces W_1 in dependence on coupling number N^2 for characteristic dimensionless length of micropolar fluid Λ_1:1) Λ_1=10, 2) Λ_1=20, 3) Λ_1=30, 4) Λ_1=40 , gap convergence coefficient ε_{opt} and ε =.,4

viscosity κ_1. In the Fig. 6 the same graph is given in the dimensionless viscosity κ_1 function. Change range N^2 from the Fig. 5 comply to κ_1 changes in the Fig. 6. Capacity force courses presented in the Fig. 6, can be more suitable for small quantities for parameter κ_1. All lines approach asymptotic to the broken line when the micropolar length increases (rotational viscosity decreases γ). Together with coupling number increase, maximal pressure increases (coupling viscosity increases).

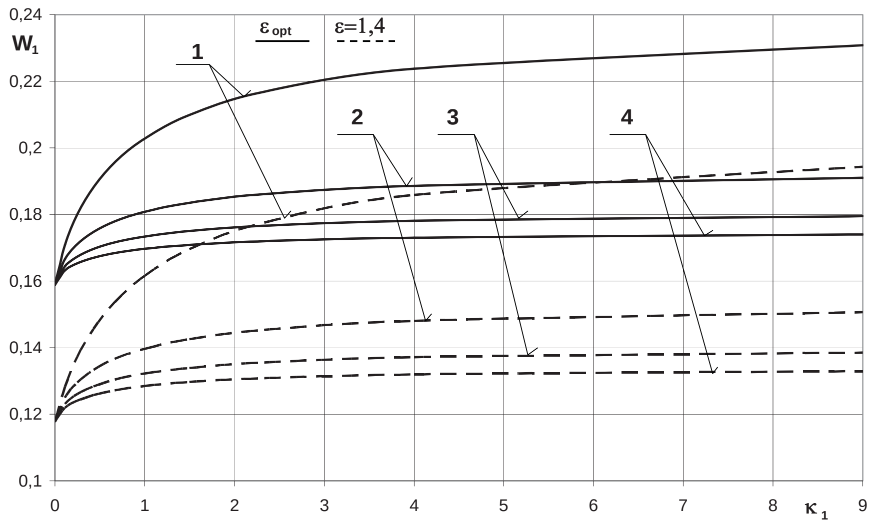

Fig. 6. The dimensionless capacity forces W_1 in dependence on coupling viscosity κ_1, for characteristic dimensionless length of micropolar fluid Λ_1:1) Λ_1=10, 2) Λ_1=20, 3) Λ_1=30, 4) Λ_1=40 , gap convergence coefficient ε_{opt} and ε =1.4

Change of length coordinate Δx_{1W} situated capacity forces W_1 in dependence on coupling number N^2 for characteristic dimensionless length of micropolar fluid Λ_1 presented Fig. 7. Change Δx_{1W} coordinate x_1 position of capacity force are defined:

$$\Delta x_{1W} = x_{1Wm} - x_{1WN}, \tag{16}$$

where:
x_{1Wm} - coordinate position of capacity force by micropolar oil lubrication,
x_{1WN} - coordinate position of capacity force by Newtonian oil lubrication.

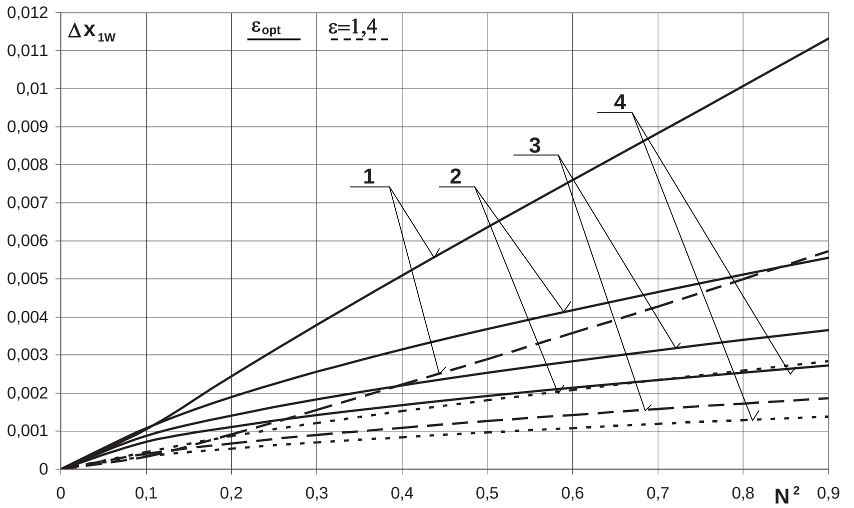

Fig. 7. Change coordinate Δx_{1W} situated capacity forces W_1 in dependence on coupling number N^2 for characteristic dimensionless length of micropolar fluid Λ_1:1) Λ_1=10, 2) Λ_1=20, 3) Λ_1=30, 4) Λ_1=40 , gap convergence coefficient ε_{opt} and ε =1.4

4. Conclusions

Presented example of the Reynolds equation solutions for steady laminar non-Newtonian lubricating oil flow with micropolar structure, enable the hydrodynamic pressure distribution introductory estimation as a basic exploitation parameter of slide bearing. Comparing Newtonian oil to oils with micropolar structure can be used in order to increase hydrodynamic pressure and also to increase capacity load of bearing friction centre. Micropolar fluid usage has two sources of pressure increase in view of viscosity properties: increase of fluid efficient viscosity (coupling viscosity increase) and the rotational viscosity increase (characteristic length parameter Λ). Author realize that he made few simplified assumptions in the above bearing centre model and in the constant parameter characterizing oil viscosity properties. Despite this calculation example apply to bearing with infinity breadth, received results can be usable in estimation of pressure distribution and of capacity force at laminar, steady lubrication of slide plane bearing with infinity breadth. Presented results can be usable as a comparison quantities in case of numerical model laminar, unsteady flow Non-Newtonian fluids in the lubricating gaps of slide plane bearings.

References

[1] Das, S., Guha, S. K., Chattopadhyay, A. K., *Linear stability analysis of hydrodynamic journal bear ings under micropolar lubrication,* Tribology International 38, pp. 500-507, 2005.

[2] Krasowski, P., *Stacjonarny, laminarny przepływ mikropolarnego czynnika smarującego w szczelinie smarnej poprzecznego łożyska ślizgowego,* Zeszyty Naukowe Nr 49, pp. 72-90, Akademia Morska, Gdynia 2003.

[3] Łukaszewicz, G., *Micropolar Fluids. Theory and Aplications* – Birkhäuser, Boston 1999.

[4] Walicka, A., *Reodynamika pzepływu płynów nienewtonowskich w kanałach prostych i zakrzywionych,* Uniwersytet Zielonogórski, Zielona Góra 2002.

[5] Walicka, A., *Inertia effects in the flow of a micropolar fluid in a slot between rotating sufrages of revolution,* International Journal of Mechanics and Engineering, Vol. 6, No. 3, pp. 731-790, 2001.

[6] Wierzcholski, K., *Mathematical methods in hydrodynamic theory of lubrication,* Technical University Press, Szczecin 1993.

[7] Xiao-Li Wang, Ke-Qin Zhu, *A study of the lubricating effectiveness of micropolar fluids in a dynamically loaded journal bearing,* Tribology International 37, pp. 481-490, 2004.