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ASPECT OF JOURNAL BEARINGS LUBRICATION USING NON NEWTONIAN FLUIDS

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ABSTRACT

The aim of this work is related to an analysis of journal bearings lubrication, using non Newtonian fluids, which are described by a power-law model. The performance characteristics are obtained for various values of the non Newtonian power-law index ''n'' that is equal to: 0.9, 1 and 1.1. Numerical results indicate that for dilatants fluids ($n > 1$), the load-carrying capacity, the pressure, the temperature, and the frictional force may be greatly increased, while for the pseudo- plastic fluids ($n < 1$), they all decrease. The thermal effects are found to be more pronounced at higher values of the flow behavior index ''n''. The results obtained in this study are compared to those obtained by others. A good agreement is observed between them.

***Keywords**:* Lubrication, Non Newtonian Fluid, Thermo-hydrodynamic Aspect, Numerical Simulation.

1. INTRODUCTION

The evolution of machines with severe operating conditions, following to the number of revolutions increasingly high and shafts strongly charged, has a consequence on the dissipation of energy by shearing in lubricating film, which will generate an increase of its temperature and consequently a reduction of the viscosity of the lubricant fluid, a bearing pressure of the mechanism and a premature wear of the material used. The isothermal theory of lubrication is largely used in the performances calculation of the butted and hydrodynamic bearings. However, the technological requirements, such as the increase in the loads and the number of revolutions, induce an increase in the dissipated energy in the lubricated mechanisms [Frene et al., 1990].

The classical theory of lubrication developed by O. Reynolds for isothermal cases is improved by Kingsbury [Kingsbury et al., 1933] by taking into account heat transfer aspect and by assuming the fluid used as viscous and Newtonian. However, in most mechanisms encountered in real situations, non Newtonian fluids are used in order to increase the lubricants viscosity index by adding additives such as polymers [Harnoy, 1978].

The first approach modelling of the thermal aspect of lubrication was proposed by Kingsbury, in order to take into account the variation of the temperature through the thickness of the film. The method of resolution applied to the case of a conical sleeve viscometer is a graphic method. In his study, Kingsbury has showed that the shearing stress of the bearing surface is about 40% of the constraint value calculated by using the isothermal theory. Then, it can be deduced easily whereas the heating of the film causes a reduction of the load supported by the shaft of 60% compared to the load calculated by the isothermal theory for similar operating conditions.

The behaviour's law of non Newtonian fluids is nonlinear, which has a consequence on the non validity of Reynolds equations commonly used in the traditional hydrodynamic lubrication. The non Newtonian lubricants are encountered in various processes of lubrication. During the four last decades, the interest to lubrication problems with not Newtonian fluids behaviour became extensive, where few works were presented in this field ([Safar, 1979], [Sinha et al., 1983], [Dien and Elrod, 1983], [Sheau- Ming, 1994], [HlavaEek, 1997], [Yürüsoy, 2003]).

The work presented in this paper, is related to the journal bearings lubrication aspect analysis using non Newtonian fluids by taking into account the thermo-hydrodynamic aspect of the problem. It consists in a comparison of the temperature, the pressure distribution and the load, in the various components of the journal bearings, by using different lubricants.

2. MATHEMATICAL MODELLING

2.1 Physical Model

Figure 1 gives a schematic representation of a journal bearing system. It consists of a bearing with a centre O_B and a radius R_B, and a shaft with a centre O_S and a radius R_S. Under the load action, the centres O_B and O_S do not coincide. The distance O_B O_S is called the absolute eccentricity. If the axes of the bearing and the shaft are parallel, and if we neglect the elastic strain of surfaces, under the effect of the temperature and the pressure, these two parameters are sufficient to locate the position of the shaft inside the bearing. The radius R_B is approximately equal to R_S in the contact zone, between the bearing and the shaft, and then we can neglect the curve shape of the film, develop the bearing and compare it to a plan shape.

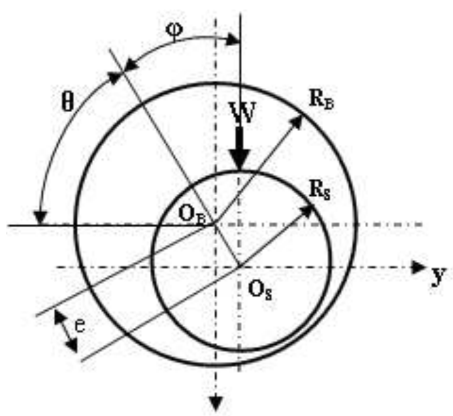

Figure 1 Schematic representation of the journal bearing.

2.2 Governing equations

The mathematical modelling of the problem is based on conductive heat transfer equations, for the bearing and the shaft, and on energy and momentum equations, formulated by Reynolds equation, for the lubricating film. The generalized Reynolds equation is given by [Tsann-Rong, 1991]:

$$\frac{\partial}{\partial x}\left[G\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[G\frac{\partial p}{\partial z}\right]=U\frac{\partial}{\partial x}\left[h-\frac{I_2}{J_2}\right] \qquad (1)$$

Where $I_2=\int_0^h \frac{y}{\mu_a}\,dy$, $J_2=\int_0^h \frac{dy}{\mu_a}$, $R=\int_0^y \rho(x,\xi,z,t)\,d\xi$

$F=\frac{1}{J_2}\int_0^h \frac{R}{\mu_a}\,dy$, $G=\int_0^h \frac{R\,y}{\mu_a}\,dy - I_2\,F$, $U=\omega R_S$

The thickness of the film, h, is (see figure1).

$$h=C(1+\varepsilon\cos\theta) \qquad (2)$$

With $C=(R_B-R_S)$, $\varepsilon=e/C$, $\theta=x/R_S$

In Cartesian coordinates system, balance energy equation in lubricating film is given by:

$$\rho\cdot C_p\left[u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right]=K\frac{\partial^2 T}{\partial y^2}+\mu_a\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right] \qquad (3)$$

Within the bearing, the thermal phenomena are governed by the conductive heat equation given by:

$$\rho C_p\frac{\partial T}{\partial t}=K\,\Delta T \qquad (4)$$

Taking into account the cylindrical shape of the bearing, the above equation will be then:

$$\rho\cdot C_p\frac{\partial T}{\partial t}=K\left[\frac{1}{r}\frac{\partial}{\partial r}\left(\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial r^2}+\frac{\partial^2 T}{\partial z^2}\right] \qquad (5)$$

The heat transfer process in the shaft is governed by the equation of energy in a steady state. According to the experimental results of Dawson [Frene et al., 1990], the temperature of the fast revolving shaft is independent of the angular coordinate, θ. Under these conditions the equation of heat is written [Tsann- Rong et al, 1991] as:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial r^2}+\frac{\partial^2 T}{\partial z^2}=0 \qquad (6)$$

2.3 Boundary conditions

The boundary conditions used for the film are those of Swift- Stieber [Tsann-Rong et al, 1991]. They take into account the conservation of the flow at the rupture of film, and they are expressed by the pressure conditions as follows

$$P(\theta_s,z)=0,\ \frac{\partial P}{\partial\theta}(\theta_s,z)=0 \qquad (7)$$

About the bearing, the boundary condition is given by the continuity of flow between ambient air and the external surface of the bearing. It is given by:

$$-K_B\left.\frac{\partial T}{\partial r}\right|_{r=R_B}=h_B\left[\left.T\right|_{r=R_B}-T_A\right] \qquad (8)$$

The interface condition, between the lubricant and the internal surface of the bearing, is given by the below condition

$$K_F \left.\frac{\partial T}{\partial y}\right|_{y=0} = -K_B \left.\frac{\partial T}{\partial r}\right|_{r=R_B}$$

(9)

For the shaft- film condition, because the shaft is fast in rotation, this flow is independent of the angular co-ordinate, thus we integrate the heat flow leaving the film on a crown of a radius equal to the radius of the shaft and with a width, dz. Then, it is given as follows:

$$K_S \left.\frac{\partial T}{\partial r}\right|_{r=R_S} 2\pi R_S\, dz = -K_F\, dz \int_0^{2\pi} \left.\frac{\partial T}{\partial y}\right|_{y=h} R_S\, d\theta \quad (10)$$

The boundary condition between the shaft and the film fluid is given by the continuity of heat flow at the interface. It is given by:

$$-K_S \left.\frac{\partial T}{\partial z}\right|_{z=0,l} = h_S \left[T|_{z=0,l} - T_A \right] \quad (11)$$

3. NUMERICAL SIMULATION

The solution of the problem requires the resolution of Eqs. (1) - (6) with the boundary conditions (7)- (11). These equations are solved numerically by using the finite difference method [Carnahan, (1969)].

4. RESULTS AND DISCUSSION

The journal bearing used is that of [Sheau-Ming et al., 1994]. It has two components, one representing the shaft and the other the bearing, the system is supplied by lubricant fluid through openings, which emerge in an axial groove. The experimental data used are given by table 1.

Table 1 Technical data used for the numerical simulation.

Journal bearing length	$l = 10^{-1}$ m
Shaft radius	$R_S = 5\ 10^{-2}$ m
External bearing radius	$R_B = 10^{-1}$ m
Radial clearance	$C = 10^{-4}$ m
Ambient temperature	$T_A = 40$ °C
Initial temperature of lubrication	$T_{al} = 36.8$ °C
Coefficient of equation (13)	$m_0 = 0.0323$ Pa/s
Coefficient of equation (13)	$\beta_f = 0.037$ °C^{-1}
Initial pressure of lubrication	$Pa = 70\ 10^3$ Pa
Lubricant density	$\rho = 860$ kg/m^3
Lubricant specific heat	$C_p = 2000$ J/kg.K
Bearing thermal conductivity	K_B =50.84 W/m.K
Shaft thermal conductivity	$K_S = 52.0$ W/m.K
Film thermal conductivity	$K_F = 0.13$ W/m.K
Bearing convective heat transfer coefficient	h_B =80W/m .K
Shaft convective heat transfer coefficient	$h_S = 100$ W/m .K
Revolution speed	N= 2000 rev./min

The law of the lubricating oil viscosity used in experiment is given by [Sheau-Ming et al., 1994]:

$$\mu_a = m(t) \cdot \left[\left(\frac{\partial u}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial y} \right)^2 \right]^{\frac{(n-1)}{2}} \quad (12)$$

With

$$m(t) = m_0 \cdot \exp\left(\beta_f \cdot (T - T_0) \right) \quad (13)$$

Fig. 2 shows the evolution of the pressure in the film, according to the circumferential co-ordinate, and for various values of index of structure n, where we can note that the pressure increases with the increase of this index.

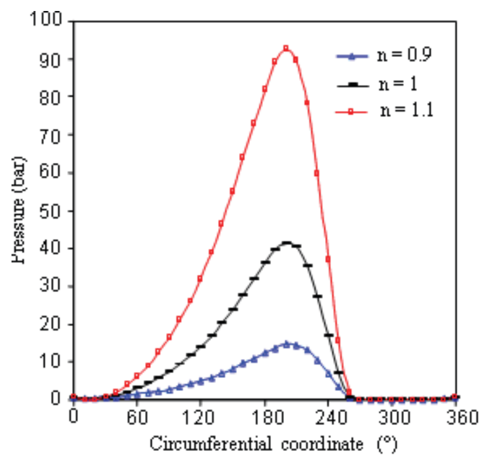

Figure 2 Evolution of the pressure versus the coordinated circumferential for different value of n.

The use of the dilating fluids gives an increase in the pressure of load, which reaches up to 122 % the value obtained by the use of Newtonian fluids for the same operating conditions.

Figure 3 gives the pressure evolution in the film versus the circumferential co-ordinate, for different values of structure's index n, and for two different cases (thermo hydrodynamic and isotherm).

The variation of the pressure between the isothermal and the thermo-hydrodynamic cases increases with the increase of the index of structure. We can also note that the heating effect is more outstanding for the dilating fluids, where the variation of the results between the two cases, isotherm and thermo-hydrodynamic is very large (reached up to 116 %), that is justify the importance of the thermo-hydrodynamic aspect analysis in the case of such fluids. However, for the pseudo-plastic fluids, the variation is the very weak, the thermo-hydrodynamic aspect have not a great influence.

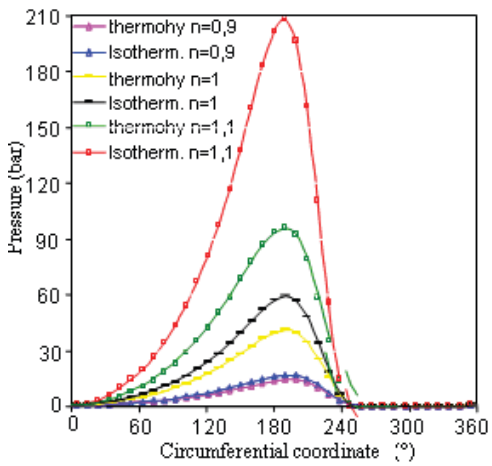

Figure 3 Evolution of the pressure versus the coordinated circumferential for different value of n. Comparison between isothermal and thermo-hydrodynamic cases.

Figures 4 and 5 show the influence of the revolution number by minute on the isothermal lines for a fixed index structure. The temperature increases with the revolution number by minute.

Figures 6 and 7 represent the bearing and the film temperature evolutions, according to the circumferential coordinates and the film thickness, and for various values of index of structure n, where we can note that the temperature of the film and the bearing increases with the increase of index n.

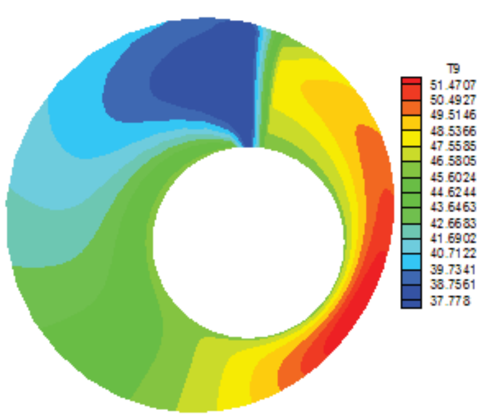

Figure 4 Isothermal lines of the journal bearing for : N= 1000 rev./min and n=1.1.

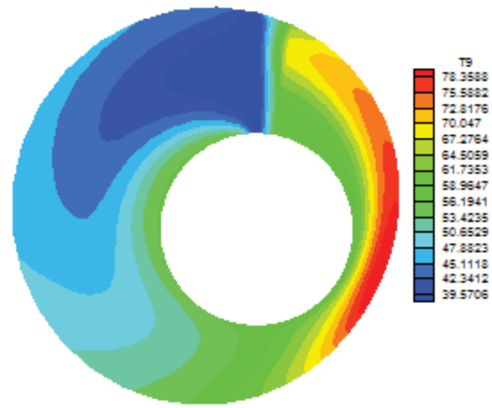

Figure 5 Isothermal lines of the journal bearing for : N= 3000 rev./min and n=1.1.

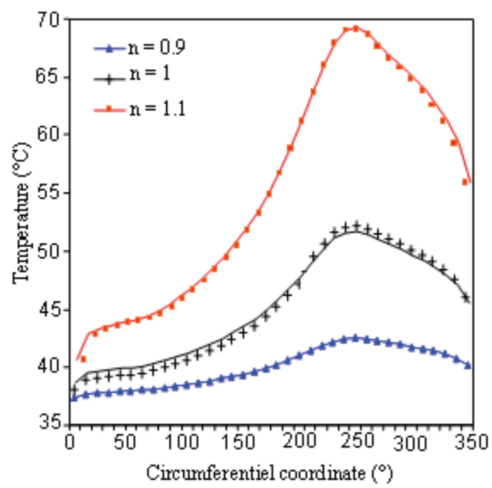

Figure 6 Evolution of the bearing internal surface temperature according to the circumferential coordinate and for different value of n.

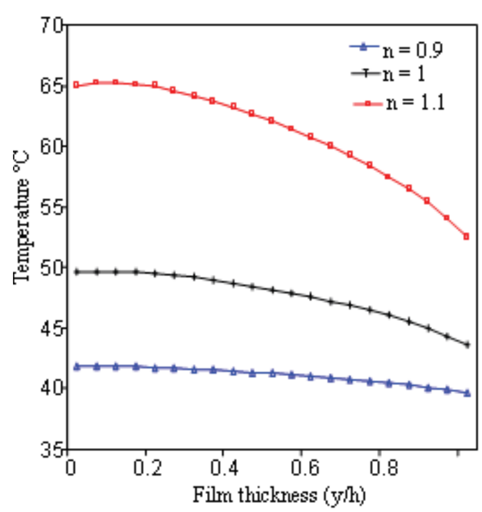

Figure 7 Evolution of the lubricating film temperature according to its thickness and for different value of n.

5. CONCLUSION

The work presented in this study is related to a thermo-hydrodynamic analysis of a smooth journal bearing, using a non Newtonian behaviour lubricant, with a power law formula. The results presented showed that:

- The increase of the structure index, n, generates an increase in the pressure. This increase becomes more significant for the case of the dilating fluids.
- For the same operating conditions, the increase of index induces an increase in the pressure, the temperature and the load.
- For the dilating fluids ($n > 1$), the load, the temperature and the pressure are more significant than those of the Newtonian fluids.
- For the pseudo- plastic fluids ($n < 1$), the load, the temperature and the pressure are weaker than those of the Newtonian fluids.
- The thermal effects are important in the dilating fluids cases. The difference between results obtained by using isothermal theory of lubrication and non isothermal one is very large (reached up to 116 %), which justifies the importance of the thermo-hydrodynamic study in the case of such fluids.

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NOMENCLATURE

l	Length of the journal bearing, m
R_S	Radius of the shaft, m
R_B	Internal radius of the bearing, m
C	Radial distance between shaft and bearing, m
x, y, r, z	Space coordinates, m
u, v, w	Velocities, $m.s^{-1}$
T_A	Ambient temperature, °C
P	Pressure, Pa
ρ	Density, $kg.m^{-3}$
C_p	Specific heat, $J.kg^{-1}.K^{-1}$
K_B	Bearing thermal conductivity, $W.m^{-1}.K^{-1}$
K_S	Shaft thermal conductivity, $W.m^{-1}.K^{-1}$
K_F	Film thermal conductivity, $W.m^{-1}.K^{-1}$
h_B	Bearing convective heat transfer coefficient, $W.m^{-2}.K^{-1}$
h_S	Shaft convective heat transfer coefficient, $W.m^{-2}.K^{-1}$