DETERMINATION OF THE THERMAL GRADIENT INSIDE THE "POCKET" OF LUBRICANT FOR THE CONE DOUBLE ENVELOPING WORM GEAR

Lecturer, dr. Cristian FEDORCIUC – ONIŞA*, dr. Mykola DŽUBINSKÝ**

North University of Baia Mare[1], Christian.Onisa@corusgroup.com

*****Institute of Materials Research of Slovak Academy of Sciences, Watsonova 47, 04353 Košice, Slovakia[1]***

Abstract: The double enveloping worm gear features itself through double enveloping aspect as well as the double contact during running. There is always o contact line and a contact curve simultaneously. Between them a cavity is formed (so called "pocket") in which the lubricant is kept under pressure and hence the high load capacity of this gear under good running conditions is assured.

Due to variation of the worm geometry, the "pocket" of lubricant changes its shape and thickness along the meshing area. The contact conditions, like normal relative curvature, instantaneous contact force, change as well. Consequently, the gradient of instantaneous lubricant temperature inside "pocket" will vary.

The present paper focuses the tribologic study of Cone double enveloping worm gear towards determination of the thermal gradient inside the pocket" lubricant by involving the TEHD lubrication's equations, solving them in an original manner and establishing this variation along the meshing area.

Key words: TEHD lubrication, coefficient of friction, viscosity, lubricant film

1. GENERAL ASPECTS

The problematic tackled into this paper is part of tribological investigation of gears, particularly of Cone double enveloping worm gears. Although known for a while and applied in industry and army logistic, the tackling of it from the lubrication point of view, by involving the TEHD mathematical apparatus in a modern concept of solving them offers new potential ways of improving its efficiency as much as its service life**.**

[1] In present Marie Curie post doc Fellows (EU) at Corus RD&T, United Kingdom

2. THE TEHD EQUATIONS

The TEHD lubrication is described by the following equations, expressed in a cylindrical coordinate of system, the rOθ being tangent to the instantaneous contact point belonging to the contact curve and origin O being the intersection point of the Oz line with the worm symmetry axe:

1. Reynolds equation [7]:

$$\frac{\partial}{\partial\theta}\left(F_2\frac{\partial p}{\partial r}\right)+\frac{F_2}{r}\frac{\partial p}{\partial r}+\frac{1}{r^2}\frac{\partial}{\partial\theta}\left(F_2\frac{\partial p}{\partial\theta}\right)=V_r\frac{\partial p}{\partial r}\frac{F_3}{F_0}+\frac{V_\theta}{r}\frac{\partial}{\partial\theta}\left(\frac{F_3}{F_0}\right)-\rho V_z, \qquad (1)$$

where F_0, F_1, F_2, F_2 are the Dowson's functions [1]:

$$F_0=\int_0^h\frac{dz}{\eta},\qquad F_1=\int_0^h\frac{zdz}{\eta},\qquad F_2=\rho\int_0^h\frac{z}{\eta}(z-\frac{F_1}{F_0})dz,$$

$$F_3=\rho\int_0^h\frac{zdz}{\eta},$$

(2)

The limit conditions for pressure are:

$$p(r_{\min},\theta)=p(r_{\max},\theta)=p(r,\theta_{\min})=p(r,\theta_{\max}) \quad \text{şi} \quad p=0 \text{ pentru } \frac{\partial p}{\partial r}=\frac{\partial p}{\partial\theta}=0. \qquad (3)$$

2. *Energy equation[7]:*

$$\rho c_p\left(v_r\frac{\partial T}{\partial r}+\frac{v_\theta}{r}\frac{\partial T}{\partial\theta}+v_z\frac{\partial T}{\partial z}\right)-k_0\left(\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial\theta^2}+\frac{\partial^2 T}{\partial z^2}\right)=$$
$$=\alpha_T T\left(v_r\frac{\partial p}{\partial r}+\frac{v_\theta}{r}\frac{\partial p}{\partial\theta}\right)+\eta\left[\left(\frac{\partial v_r}{\partial z}\right)^2+\left(\frac{\partial v_\theta}{\partial z}\right)^2\right]. \qquad (4)$$

The limit conditions for energy express two physical phenomena [7]:

- The convection heat exchange between lubricant and the mating surfaces (5)
- The continuity of lubricant flow at inlet and outlet for the contact point (6).

$$k_0\left[\frac{\partial T}{\partial z}\right]_{z=0} = k_{OL}\left[\frac{\partial T_1}{\partial z}\right]_{z=0} \quad k_0\left[\frac{\partial T}{\partial z}\right]_{z=h} = k_{Bz}\left[\frac{\partial T_2}{\partial z}\right]_{z=h} \tag{5}$$

$$T(r,\theta,z)_{0,j,k} = T(r,\theta,z)_{i,j,k} = T(r,\theta,z)_{i,0,k} = T(r,\theta,z)_{i,j,0.} \tag{6}$$

3. *The elasticity equation.* The integral of surface elastic deformation in the p(r, θ) contact point is, according to Timoshenko şi Godier [9],:

$$d(r,\theta) = k_d \int_{R_1}^{R_n}\int_{\zeta_1}^{\zeta_n} \frac{p(r,\zeta)\ dr\ d\theta}{\sqrt{(r\cos\zeta - R\cos\zeta)^2 + (r\sin\zeta - R\sin\zeta)^2}}. \tag{7}$$

4. *The equation of pressure and temperature related variation of lubricant viscosity*, as proposed by Roeland (1963) is [4]:

$$\eta = \eta_0 \exp\left\{[\ln(\eta_0) + 9.67]\cdot\left[-1 + \left(1 + 5.1\cdot 10^{-9}\cdot p\right)\right]^{\zeta} + \left(1 - \frac{T_{max}}{T_0}\right)\right\}.$$

(8)

5. *The equation of pressure and temperature related variation of lubricant density,* as proposed by Dowson şi Higginson (1966) is [3]:

$$\rho = \rho_0\left(1 + \frac{0.6\cdot 10^{-9}\cdot p}{1 + 1.7\cdot 10^{-9} p}\right)\cdot\left(1 + \beta\cdot T_0\left(1 - \frac{T_{max}}{T_0}\right)\right) \tag{9}$$

3. NUMERICAL METHOD FOR SOLVING THE TEHD EQAUTIONS

The TEHD equations form a highly non-linear system of equations. In order to solve it, a program has been developed in a MathCAD2000 environment and named AMGLUB. The separate blocks, each of which solves each equation, have been implemented and linked together in MathConnex2000 environment.

Reynolds and energy equations have been solved iteratively by Levenberg-Marquardt [11] method after finite differences discretion. In case of energy equation, which is function of three variables *(i, j, k),* it resorted to separate it in *n* elementary equations, which describe the energy balance in the lubricant "pocket" for these *n* levels (along the *Oz* axe). These *n* elementary equations together with the limit conditions form the above mentioned highly non linear system of equations, solvable by using the Given-Minner method available in MathCAD2000.

In this way one get the plane thermal gradient for each of those *n* levels inside the "pocket" of lubricant. If only the maximum values are kept, the matrix of the gradient of the maximum instantaneous temperature inside the "pocket" of lubricant is obtained. The flow chart of AMGLUB is schematic presented in figure 1.

4. NUMERIC CASE

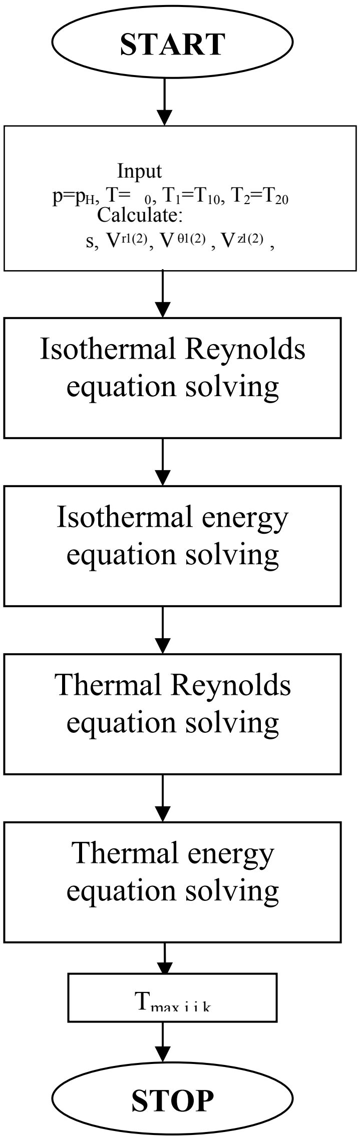
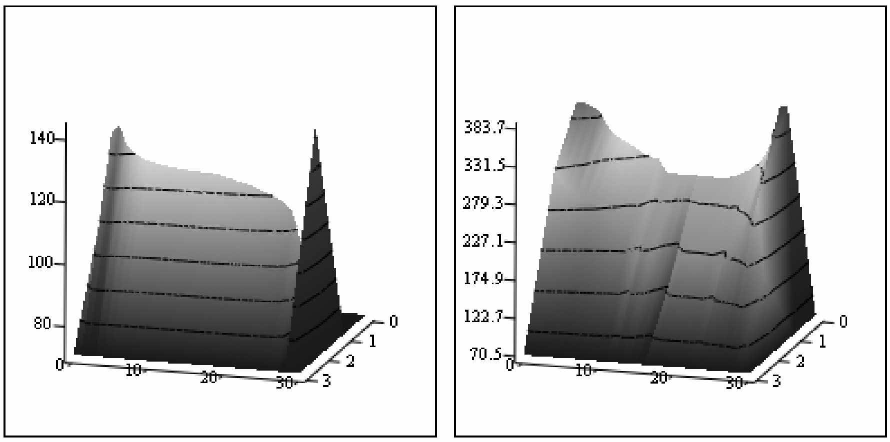


Fig.1 [2]
The AMGLUB's flow chart

As numeric case, a Cone double enveloping worm gear with the following features was considered: centre distance A=125 mm, ratio i=9,75, z_1=4, n_1=1500 RPM, M_{t2}=100 daNm. The following three oils, with a pre-contact temperature of 70,5 ^{0}C, assure lubrication:

a) Shell Tivella WB, ν_{40^0C} = 234 cSt,
b) Shell Morlina 10, ν_{40^0C} = 10 cSt,
c) Petrom T90 EP2, ν_{40^0C} = 195 cSt.



The first contact curve The forth contact curve

Fig. 2 [2]. The gradient of instantaneous maximum temperature inside the "pocket" of lubricant [^{0}C]

The mating surfaces roughness are: R_{a1}=0,63 μm for worm thread and R_{a1}=0,32 μm for wheel flank.

The gradient of maximum instantaneous temperature inside the “pocket” of lubricant for four contact points generated by the same value *u* of linear parameter of surface generation, belonging to the first and, respectively, the last instantaneous contact curve from the meshing area is showed in figure 2.

As one can see, the maximum instantaneous temperature inside the lubricant during the contact increases towards the exit from the meshing, respectively towards the zone of minimum worm diameter. The explanation of this variation resides in instantaneous contact force (related to worm diameter on its turn). Decreasing of the specific sliding inside lubricant film towards the minimum worm diameter zone should lead to a decreasing of the maximum instantaneous temperature. By combining the effect of the instantaneous contact force and the gear geometry lead to the conclusion that this force has a significant influence on the increasing the instantaneous temperature, which vary from 70,6 ^{0}C at the inlet into the meshing area up to 343,1 ^{0}C at the outlet.

The variation of thermal gradient inside the meshing area is presented in figure 3. Alongside the contact curve the maximum instantaneous temperature of the lubricant decrease from the bottom towards the top of the wheel tooth, as one see in figure 4.

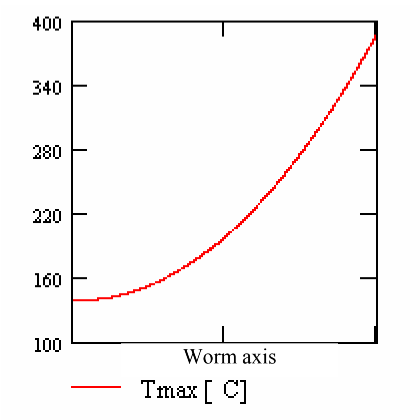


Fig.3
Variation of maximum instantaneous temperature of lubricant along the meshing area

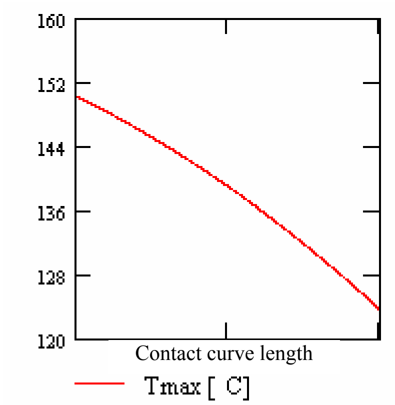


Fig. 4
Variation of maximum instantaneous temperature of lubricant along contact curve (second contact cureve)

5. CONCLUSIONS

Determination of maximum instantaneous temperature of lubricant inside the “pocket” close by an instantaneous contact line and an instantaneous contact curve is important for a better understanding of the mechanism by which the lubrication is assured in case of Cone double enveloping worm gear. The chosen approach is useful not only for establishing new ways of improving the performances for this kind of gear, but also could lead to developing new kinematics, based on the tribological results with the aim of building new kind of worm gears with higher efficiency and/or longer life service.

Due to very small contact time (the magnitude of nanoseconds), these high instantaneous temperatures inside the lubricant do not affect the chemical stability of it, but instead the pressure and the fluid coefficient of friction are affected. Due to the shown variation along the meshing area, the wear is expected to develop in a non-uniform manner.

Tackling the thermal aspect of the EHD lubrication, despite the level of difficulty brought in, permits a more realistic evaluation of the phenomena occurring at the microscopic contact level, enhancing the current models of gears lubrication with obvious consequences in finding new alternatives for increasing the performances either of an existent gear or a designed one.

REFERENCES

1. **Dowson, D.,** A Generalized Reynolds Equation for Fluid Film Lubrication, *International Journal of Mechanical Science, Pergamon Press Ltd., vol. 4, 1962*
2. **Fedorciuc-Onişa, C.,** Influenta unor factori tribologici asupra functionarii angrenajelor melcate globoidale**,** *Teza de doctorat, Universitatea Tehnica Cluj Napoca, 2001*
3. **Hsu, C. H., Lee, R. T.,** An Efficient Algorithm for Thermal Elastohidrodinamic Lubrication Under Rolling / Sliding Line Contacts, *Transactions of the ASME, Journal of Tribology, October, 1994, vol.116, pp.762-769*
4. **Kim., K., Sadeghi, F.,** Three–Dimensional Temperature Distribution in EHD Lubrication: Part II – Point Contact and Numerical Formulation, *Transactions of the ASME, Journal of Tribology, January, 1993, vol.115, pp.36-45*
5. **Lubrecht, A.A., ş.a.,** Multigrid, An Alternative Method for Calculating Film Thickness and Pressure Profiles in Elastohydrodynamically Lubricated Line Contacts, *Transactions of the ASME, Journal of Tribology, vol. 108, 1986*
6. **Olaru, N. D.,** Tribologie. Elemente de bază asupra frecării, uzării şi ungerii, *Universitatea Tehnică "Gh,. Asachi", Iaşi, 1995*
7. **Simon, V.,** Elastohydrodynamic Lubrication of Hypoid Gears. *În: ASME, Journal of mechanical Design, vol. 103, 1981, p. 195-203*

8. **Tallian, T. E.,** The Theory of Partial EHD Contacts, *Wear, 21, 1972*
9. **Timoshenko, S. and Godier, J. N.,** Theory of Elasticity, *McGrow –Hill, New York, 1951*
10. *** The MathCAD 2000 Treasury