On thermal calculation for helical gear transmission system

I S Gabroveanu[1]**, S Cananau**[2,3,]**, R F Mirica**[2] **and A A Ilies**[1]

[1] National Research and Development for Gas Turbines SC COMOTI SA, 220 D Iuliu Maniu Bd., 061126, Bucharest, Romania
[2] Politehnica University of Bucharest, Machine Elements and Tribology Department, 313 Spl.Independentei, 060042, Bucharest, Romania

Abstract. The purpose of the study in the paper is to evaluate the thermal comportment of the transmission systems with helical gears. Also, the influences of heat on characteristics of gear transmission system, based on the meshing thermal stiffness of gears are analyzed. An experimental testing stand of the system will be used to obtain data. The respond of the transmission under different working conditions is also analyzed. The study concerns the comparison of theoretical results obtained using calculation methods (from literature) and the experimental results, in order to obtain an evaluation of the friction in the meshing gears. The heat (temperature) affects the dynamic characteristics of the system by the meshing stiffness of the gear. The experimental results are used to verify the correctness of the theoretical model.

1. Introduction

In the last years, the design of modern gear transmissions is concerning to improve energy-dense property of the new product. As consequence the development of new mechanical systems with gears require much detailed knowledge and experiments. One of interesting area of the gears transmissions research is to evaluate all kind of energy losses in the all detailed components in function in the system for a better running performance. Energy losses in a gearbox are converted into heat which must be dissipated. Thus, this heat is the main responsible for the temperature increase while the system is running. In this paper we are going to investigate, within a thermal model who evaluates each kind of power loss, the thermal calculation for gears.

Many authors were concern of this area of research: Durand de Gevigney et alt. are concern to study thermal *modelling* of a back-to-back gearbox test machine with an experimental FZG test rig [1], Hlebanja and Kulovec focusses on the differences in thermal load of cylindrical, spur, plastic involute and S-gears [2], Kanatnikov et alt. presents a model for predicting the thermal processes arising during shaping gears with internal non-involute teeth [3]. The variations of the temperature in the function time are also a subject of research interest. Luo and Fei focus on some common mechanical parts in precision technology and propose mathematical models for hollow piece, gear and cube. The experimental results also make it clear that these models are more logical than traditional models [4]. Other authors are involved in the study of comportment of the lubricants in these mechanical systems. Badrinarayanan et alt. said that fluid transmission losses and inherent friction, heat energy is generated in the system which rises the fluid temperature. This heat has to be dissipated by proper means to

---

[3] Author to whom any correspondence should be addressed

maintain the system temperature within safe limits. In their paper the authors attempt to study the thermal aspects of hydraulic system, heat generation and dissipation to estimate temperature rise profile and steady state temperature [5]. Other authors are interested to establish the dynamic behaviour of the gear pairs under different operating conditions, as Radu et alt. [6]

2. Thermal balance and heat dissipation

By hypothesis the gearbox at a *thermal balance*, heat evacuated by conduction, convection and radiation through the all mechanical elements of the transmission is a result of a power loss generated inside the system. So, the balance will be expressed as:

$$Q_{diss} = P_{loss} \tag{1}$$

Each part of this equation has the following explanation:

2.1. The heat dissipation

2.1.1. The heat dissipation by radiation. The heat dissipation Q_{diss} is a sum of evacuated heat by conduction, convection and radiation for all elements of the system that in time are supposed to be in the process of dissipation

$$Q_{diss} = \sum_{i=1}^{n} Q_{i,rad} + \sum_{j=1}^{m} Q_{j,conv} + \sum_{k=1}^{p} Q_{k,cond} \tag{2}$$

According to othe*r* works, example [1], the internal heat transfer is not taken into account for the stationary gearboxes. The dissipation by radiation is due to the heat flow from the surfaces of the gearboxes housing.

2.1.2. The heat dissipation by convection. The convection is due to the contact between the oil and the mechanical parts interaction, at various temperatures

$$Q_{conv} = c_{conv} \sum_{j=1}^{m} A_{j,conv} (T_{wall} - T_{oil}) \tag{3}$$

where c_{conv} is the oil convection coefficient, A_j is the area of a surface j involved in heat dissipation by convection, T_{wall} is gearbox housing temperature, T_{oil} is the fluid (oil) temperature.

2.1.3. The heat dissipation due to conduction is not taken into account in this paper.
Concerning the second part of the Eq.(1) we will take into account: *2.2.1.* teeth friction loss, 2.2.2. shaft-bearing power loss, *2.2.3.* shaft seal power loss, *2.2.4.* churning losses.

2.2. The power losses

2.2.1. Teeth friction losses. In the functioning conditions the pinion (driving) and the wheel (driven) teeth are in meshing. In this paper we assume the calculus for the teeth friction losses in conditions of mixed film lubrication, which means that the normal load for the teeth contact is not fully supported by EHD lubricant film. In the real case of functioning there are different stages of meshing with possible various film thickness, due to variation of the contact geometry, pressure and load value along the meshing line. For general purpose we consider the loss power in mesh as is considered in [1]:

$$P_{mesh} = \mu_f \cdot P_{in} \cdot H_v \tag{4}$$

where H_v is the geometry parameter:

$$H_v = \frac{\pi(u+1)}{(z_1 \cdot u \cdot \cos \beta_b)(1-\varepsilon)} \tag{5}$$

and P_{in} is the input power, μ_f friction coefficient.

Regarding the friction coefficient μ_f there is a lot of discussions and researches done to agree with an expression to fit a sustainable value of this parameter. We take for reference the expression of Höhn at al. [7] and developed by Castro and Seabra [8]. In this paper we adopt the following expression:

$$\mu_f = 0{,}171 \left(\frac{W_L}{R'_{eq(C)} \cdot V_R} \right)^{0,2} m^{-0,05} \left(\frac{R_{aEq}}{d_1} \right) X_L X_C \qquad (6)$$

where W_L is specific normal load, V_R is the rolling speed, $R'_{eq(C)}$ is the equivalent radius of curvature of the contact points of teeth (C), m is dynamic oil viscosity at oil temperature, d_1 is the pinion pitch diameter, R_{aEq} is the equivalent arithmetic average of the roughness of surfaces in contact, X_L is lubricant correction factor (function of the nature of additives) and X_C is a correction factor for gears with coated surfaces. (in our case, X_C is equal to unit).

2.2.2. Rolling bearing power loss. Taking into account the considerations due to Niemann et. all. [9] we will consider only the following causes for the friction in shaft-bearings system:
- rolling at a contact area, having different radiuses with elastic deformations and material hysteresis,
- sliding between guiding surfaces and rolling elements,
- lubricant internal friction, drag losses, churning, splashing,
- air resistance by high speed,
- rolling resistance caused by various contaminations in contact area.

In [9] a simplified calculation method is obtained. The frictional moment can be estimated with sufficient accuracy using a constant coefficient of friction, $\mu_{bearing}$, the mean diameter of rolling bearing, d_m, and the equivalent dynamic bearing load, F_{eq}: According to SKF [10], the recommend following formula to estimate the frictional torque, T_{fr}, is:

$$T_{fr} = 0{,}5 \cdot \mu_{bearing} \cdot d_m \cdot F_{eq} \qquad (7)$$

where $\mu_{bearing} = 0.002$ (for spherical roller bearings that are used in experimental rig) is an approximated constant coefficient of friction for the bearing, F_{eq} is the equivalent dynamic bearing load. SKF [10] recommend the same formula but using $\mu_{bearing} = 0.0018$.

The power loss is estimated using the angular velocity of shaft, ω:

$$P_{fr} = T_{fr} \cdot \omega \qquad (8)$$

2.2.3. Shaft seal power losses. In many studies researchers evaluate the seal power loss with less influence compared to other losses in the period of functioning. Seal losses are the power losses due to friction between the shaft and its seal. We must pay attention to consider, in our study, both gearboxes with possible heat dissipation. We take into account in this paper the expression from Höhn et al. [7]:

$$P_{seal} = 7{,}69 \cdot 10^{-6} \cdot d_{sh} \cdot n \qquad (9)$$

where d_{sh} is the diameter of the seal.

2.2.4. Churning losses. Churning losses is also a subject of various opinions from many researches. This is a consequence of the experimental set up of the test gear, for each research considered. For example, there are different designs for the case of the gears. By other hand, the expression of this calculation is in direct relation with the geometry of housing, the design and also the dimensions. In this paper we had as reference the work of Hai Xu and Kahraman [11], Changenet and Velex [12].

The expression we will use is obtained from [12]:

$$P_{churn} = \frac{1}{4}(d_{w2} \cdot \omega)^3 \cdot \rho \cdot A \cdot C_m \tag{10}$$

where ω is the angular velocity, ρ is the oil density, A is the lateral and teeth immersed surface of contact between gear and lubricant, C_m is the dimensionless drag torque. In our case, for the calculus, we will consider that the gear is immersed in lubricant with a dimension of three times the gear module.

3. Validation of presented method

With a view to validate the previously presented method, some experiments have been effectuated. The used test rig is a testing machine with closed power circuit, ensuring the application of different torques, in the range 0 - 200 Nm and number of revolutions in the range of 1000 - 3000 rpm. The test rig was instrumented with: optical precision encoder, ROC 425 type, Heidenhain production, Germany, two pieces; torque converter without contact T10FS Hottinger Baldwin Messtechnik, precision class 0.05%; lines for measuring the oil temperature in the lubrication baths of the two gearboxes, test and return; AC electric motor speed measurement. In figure 1 is shown the test rig. The geometry of tested gears and other characteristics used in the calculation method are given in table 1.
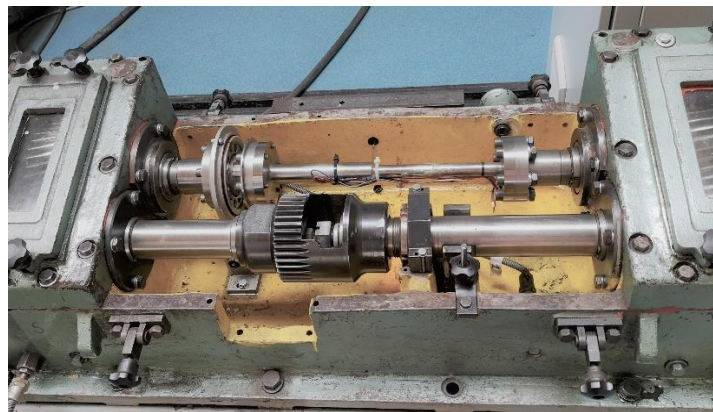

Figure 1. Test rig.

Table 1. Geometry and other characteristics of the test rig.

Characteristic	Notation	Data	Characteristic	Notation	Data
Centre distance [mm]	a_w	125	Pinion hardness	HB	270-290
Pinion teeth number	z_1	15	Wheel hardness	HB	290-310
Wheel teeth number	z_2	46	Flank roughness [μm]	Ra	0.4
Helix angle [°]	β	10	Type of the spherical roller bearings	-	21310 E
Profile shift coefficient at pinion	x_1	0.427			
Profile shift coefficient at wheel	x_2	-0.138	Distance between bearings [mm]	-	200
Face width [mm]	b	72.37	Oil type	-	H46EP
Transverse contact ratio	ε_α	1.4532	Oil volume [l]	V_{oil}	6
Overlap ratio	ε_β	0.871	Oil density [kg/m³]	ρ	877
Gear accuracy grade	-	5	Mean of thermal capacity of oil [kJ/kg K]	c_p	2.121
Gear material (case hardened)	-	41MoCr11			

At pinion torque values of 50 Nm, 100 Nm and 150 Nm are determined the heat quantity accumulated in oil sump at number of revolutions of 1000 rpm and 2000 rpm. The temperature increase is registered. The convectively dissipated heat through housing of test rig was neglected in this phase.

Using the experimental thermal capacity of oil

$$c_p = 1.826 + 4.213 \cdot t_{med}/1000 \tag{11}$$

the temperature variation, Δt and test duration (time), the power loss gets into oil sump is calculated:

$$P = Q/time = \rho \cdot V_{oil} \cdot c_p \cdot \Delta t/time \tag{12}$$

The experimental data have been centralized in the table 2. In the figures 2 and 3 are shown the calculated power losses using the presented method.

Table 2. Experimental power loss [kW] / at mean temperature [º C].

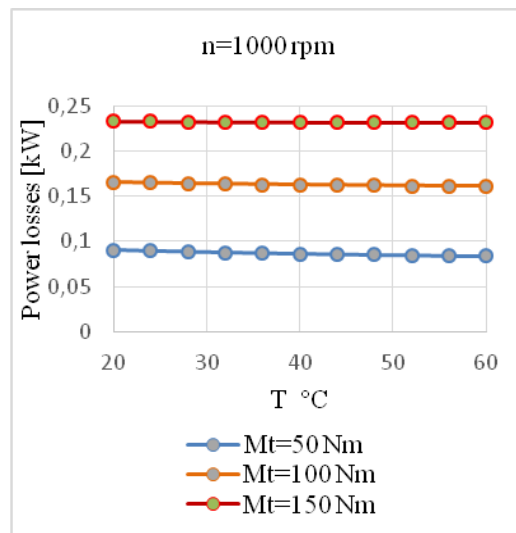| Torque [Nm]	50 Nm	100 Nm	150 Nm
Number of revolutions [rpm] / time [min]			
1000 / 10	0.019 / 38.85	0.053 / 25.25	0.053 / 32.75
1000 / 30	0.04 / 27.5	0.045 / 35.7	0.064 / 29.95
2000 / 10	0.089 / 27.675	0.091 / 29.45	0.096 / 37.05
2000 / 30	0.067 / 36.75	0.08 / 46.35	0.102 / 44.3

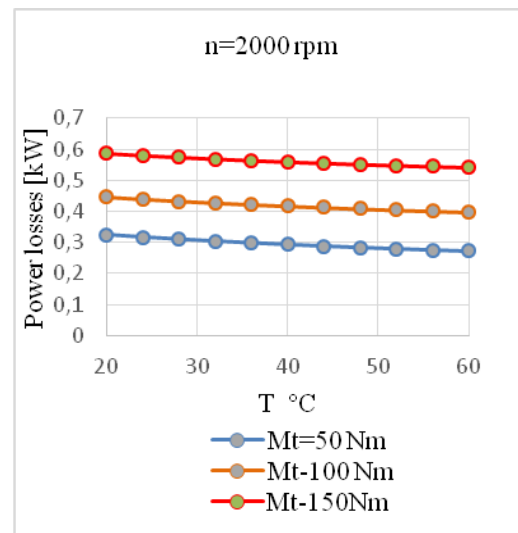
Figure 2. Calculated power loss at 1000 rpm.   **Figure 3.** Calculated power loss at 2000 rpm.

Comparing the experimental results from table 2 with the calculated ones from figures 2 and 3, it can be observed that the experimental values are considerably lower than the calculated ones. The possible explanation is the use of simplified calculation methods (that must give sure results) and the influence of convective heat transfer from test rig housing. It is to observe, that at same torque and number of revolutions the experimental power loss (table 2) is lower than the calculated one (figures 2,3) at higher mean temperature. The explanation is the difference of temperature between test rig and environment which is greater and, as consequence, also the convection is greater.

4. Conclusions
1. The thermal calculation of gearboxes was analyzed.
2. A test rig for experimental study of thermal behavior of gearboxes was realized.
3. Preliminary results of experiments are compared with calculation method. The calculated results are greater than the experimental ones. An explanation is the convective dissipation, which was neglected. Therefore the experimental assessment of convective heat dissipation of test rig is necessary.
4. An improvement of calculation especially of power losses in gears is needed.

5. References
[1]   Durand de Gevigney J, Changenet C, Ville F and Velex P, 2012, Thermal modeling of a back-to-back gearbox test machine: Application to the FZG test rig. *Proceedings of the Institution of mechanical Engineers, Part J: Journal of Engineering Tribology*, vol **226** issue: 6, pp 501-515, https://doi.org/10.1177/1350650111433243

[2]   Hlebanja G and Kulovec S, 2018, Thermal properties of S-gears in comparison with involute gears, *IOP Conf. Series: Materials Science and Engineering* 393 012067 doi:10.1088/1757-899X/393/1/012067

[3]   Kanatnikov N V, Kanatnikova P A, Vlasov V V and Pashmentova A S, 2018, Modeling of thermal processes arising during shaping gears with internal non-involute teeth, *IOP Conf. Series: Materials Science and Engineering* 327 022056 doi:10.1088/1757-899X/327/2/022056

[4]   Luo Z and Fei Y T, 2006, Research on Non-Similarity about Thermal Deformation Error of Mechanical Parts in Highaccuracy Measurement, *J. Phys.: Conf. Ser*. **48 41**, doi:10.1088/1742-6596/48/1/008 *International Symposium on Instrumentation Science and Technology, Institute of Physics Publishing Journal of Physics: Conference Series* **48** (2006) pp 41–45

[5]   Badrinarayanan S, Bhinder Kuldeep Singh, Ramesh Kumar V and Anupprasad, 2018, Thermal analysis of hydraulic system of landing gear, IOP Conf. Series: Materials Science and Engineering 402 012100 doi:10.1088/1757-899X/402/1/012100

[6]   Radu M C, Andrei L and Andrei G, 2018, A perspective on gear meshing quality based on transmission error analysis, *IOP Conf. Series: Materials Science and Engineering* 444 052011 doi:10.1088/1757-899X/444/5/052011

[7]   Höhn B-R, Michaelis K, and Vollmer T, 1996, *Thermal rating of Gear Drives: Balance Between Power Loss and Heat Dissipation*, AGMA Technical paper

[8]   Castro, M J D and Seabra, J, 2007, Coefficient of friction in mixed film lubrication: Gears versus Twin-discs, Proceedings of the Institution of Mechanical Engineers, *Part J: Journal of Engineering Tribology*, vol **221**, No. 3, pp 399-411

[9]   Niemann, G, Winter, H, 1985, *Maschinenelemente*, Band II, Springer-Verlag, Berlin

[10]   *** SKF rolling bearings catalogue, 2013, ed. SKF Group

[11]   Xu, H., Kahraman, A, 2005, A Frictional Efficiency Loss Model for Helical Gears, DETC2005-85243, Proceedings of IDETC/CIE 2005, *ASME 2005 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference*, Long Beach, CA

[12]   Changenet C, Velex P, 2007, A model for the Prediction of Churning Losses in Geared Transmissions-Preliminary Results, *Journal of Mechanical Design,* **129(1),** doi:10.1115/1.2403727.