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A Method for Calculating the Tooth and Oil Sump Temperature of Planetary Gears to Determine the Heat Carrying Capacity

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ABSTRACT

The thermal capacity of a heavy-duty gear is the function of the heat generation from the power losses during operation. In order to lowering the heat generation in the gearboxes it is very important to find the best gearbox construction for a given application and to reach its highest efficiency and load carrying capacity. The efficiency of planetary gearboxes mainly depends on the tooth- and bearing friction losses but in some application also the oil churning and air drag losses have to be determined. This article shows a method for calculating the tooth and oil sump temperature of planetary gears in order to determine the scuffing load carrying capacity.

Keywords: *Tooth friction loss, bearing friction loss, oil churning loss, scuffing, oil sump temperature*

INTRODUCTION

The increasing demands mechanical devices are supposed to meet make it necessary to develop heavy-duty mechanical drives with high power density, and good efficiency, which are of high gear ratio.

According to power transmission, planetary gearboxes belong to the big and varied family of heavy-duty gear drives with permanent gear ratio. To achieve a high load capacity, planetary gearboxes with intensive power sharing are usually coaxial gear drives, made with input and output drives that contain gears making planetary movement. Due to their high performance density and special characteristics (e.g. power summarizing, power dividing gearing, movement equalization), planetary gears are used more and more frequently in different fields of industries. Planetary gearboxes can be found as frequently in precision, medical and food industrial instruments as in the driving systems of various industrial heavy-duty machinery. Planetary gearboxes are normally made up of cylindrical or bevel gears, but in some fields, traction drives are also applied. Bevel geared planetary boxes are used in many fields by the vehicle-manufacturing industry, in different wheel drives, differential gears, and dividing gearing. Cylindrical planetary gearboxes are generally and widely used in the engineering industry, in farm machinery as well as in various power plants [2]-[3]-[4]-[10]-[11]. Improved precision planetary gears with limited tolerance can be found in machine tool drives and robotic drives.

Some important advantageous characteristics of planetary gearboxes are their relatively small size, compact, symmetric structure, high power density, good efficiency under suitable operating conditions, the possibility for power summarizing and dividing, and for increased gear ratio without structural changes.

DETERMINE THE POWER LOSSES

Some important advantageous characteristics of planetary gearboxes are their relatively small size, compact, symmetric structure, high power density, good efficiency under suitable operating conditions, the possibility for power summarizing and dividing, and for increased gear ratio without structural changes.

Geared planetary boxes have many varieties, which can be classified simply and clearly according to what external (E) and internal (I) gear engagement they have. (E) marks the meshing of external/ external tooth wheels while (I) stands for that of external/internal tooth wheels. The operating properties of individual types of planetary gearboxes differ from each other to a significant extent, which has to be taken into consideration when making a choice between them. They are normally chosen on the basis of their load capacity, depending on gear ratio and their efficiency.

The power that can be transmitted by a mechanical drive is restricted by the strength of the elements transmitting load. Scuffing load carrying capacity is normally determined by the highest temperature developing on the surface of the meshing teeth as well as the average tooth temperature, which is also influenced by contact stress, sliding speed, the state of friction and by lubricant.

In planetary gearboxes energy loss resulting from friction almost entirely turns into heat, raising the temperature of the structural elements of the driving gear, and this restricts its load capacity and transmissible power, resulting in the scuffing of gears at the limit of heat capacity. Scuffing appears either at a very low sliding speed and heavy load due to squeezing the lubricant out the contact zone: extended solid contacts and strong adhesion develop resulting strong stuck (cold scuffing), or it appears at too high surface temperatures caused by friction losses, losing the effectiveness of lubrication and causing severe surface scratches (warm scuffing). Warm scuffing is a self-triggering process because the rising temperature of the tooth surface reduces the viscosity of lubricant, deteriorating the efficiency of lubrication increasing friction losses, so the temperature of the tooth surface keeps rising. The load capacity of drives with high power density is largely influenced by their friction losses, their warming up, and the temperature of their load transmitting surfaces. Taking these into consideration, the objective of the research can be composed in the following way.

The objective of the research is to develop methods mainly for determining the losses in different type of planetary gearboxes and for calculating the distribution of the temperature building up in them.

- To develop an analytic model of tooth friction, which during engagement follows loss point by point and takes into account the state of continuously changing lubrication.
- To develop an algorithm for the calculation of the losses of bearing friction, that allows the calculation of the bearing friction losses in compound planetary gearboxes, depending on the manifesting loads and the determined life expectancy.
- To develop an analytic model for oil churning which takes into consideration the geometry of the teeth, and by describing the amount of delivered lubricant, not only is it suitable for calculating losses, but for calculating the minimal level of lubricant and that of optimal lubrication.
- To develop a numerical algorithm for the calculation of temperature distribution, which through the heat streams allows a more accurate estimate of the temperature of the teeth, and the calculation of the safety factor of scuffing can be more reliable.

In the case of modern gear drives along with the decrease in noise level and vibration, the increase in efficiency, and the effectiveness of the driving gear, lubrication as well as the enhancement of load capacity are also emphasized. It is crucial to know the efficiency of the gear and the distribution of the temperature in the case of heavy-duty planetary gearboxes with high power density whose heat-delivery surface is small, because without being cooled, they can easily get overheated. Nearly all energy losses resulting from tooth and bearing friction and lubricant and air churning convert into heat, which raises the temperature of the gearbox. An excessive increase in heat may cause the deterioration of lubrication, the seizure and scuffing of teeth, the decrease in the life expectancy of lubricant, or it may even cause its damage, scuffing, and the exceeding of the heat-capacity of the driving gear. So to examine the heat capacity of gearboxes, it is essential to calculate the power losses as accurately as possible.

The numerous models revealed during the research into the literature gave a clear insight into the calculation methods of the load dependent and load independent losses of gears. They also allowed me to investigate the calculation methods for the rise in temperature of the driving gear resulting from those losses. On the other hand, it is evident that the models shown almost exclusively only allow calculations of the losses in driving gears with a stationary axis. Reviewing the specialist literature, it can be also observed that in general the rolling efficiency, applied when determining the efficiency of planetary gearboxes, is only interpreted as the product of multiplication of tooth efficiency of tooth engagements, however, apart from tooth friction losses there are other sources of loss, which have a significant influence on efficiency and warming, especially under unfavourable conditions. The methods for the calculation of tooth efficiency in the specialist literature can only be used within quite a limited range of interpretation, and under certain conditions they yield rather different results. This may be the consequence of the tooth friction coefficient being misjudged, and if the structural features influencing the meshing of teeth and the changes in the state of lubrication are scarcely taken into consideration. The calculation of the sources of loss appearing along with tooth friction has only been studied by few so far, and the elaborated calculation methods are either too complicated or they contain correlations mainly generated from readings partly supported by theoretical argumentations, which cannot be widely used. It has not been investigated in detail what influence the structural characteristics of meshing gears have on these losses, on the temperature of gears and, through that, on the scuffing load carrying capacity.

The specialist literature contains several models for the calculation of the average temperature in gearboxes. However, it is not enough to know the average temperature of the gearbox if we want to examine the scuffing load carrying capacity of the wheels of heavy-duty planetary gears, because the contact temperature, the average temperature of the wheels and the average temperature of the planetary gear may differ considerably. It is necessary to know the distribution of the temperature in the driving gear.

CONCLUSIONS AND RESULTS

With the analytic equations developed [5] now the temporary loss of energy in the sliding-rolling contact of gear pairs on the line of action can be examined and analysed, plotted against the turn of angle of the pinion:

$$P_{fs}(\varphi)=P_{fcs}(\varphi)+P_g(\varphi)\ , \tag{1}$$

along with the average energy loss:

$$\begin{aligned}\tilde{P}_{fs} = {} & \frac{1}{3\cdot\left(g_\alpha-\dfrac{m_n\cdot\pi\cdot\cos\alpha_t}{\cos\beta}\right)}\cdot\left[\int_{\varphi_A}^{\varphi_B}P_{fcs}(\varphi)d\varphi+\int_{\varphi_D}^{\varphi_E}P_{fcs}(\varphi)d\varphi\right]+\\ & +\frac{1}{3\cdot\left(\dfrac{2\cdot m_n\cdot\pi\cdot\cos\alpha_t}{\cos\beta}-g_\alpha\right)}\cdot\int_{\varphi_B}^{\varphi_D}P_{fcs}(\varphi)d\varphi\\ & +\frac{1}{g_\alpha}\cdot\int_{\varphi_A}^{\varphi_E}P_g(\varphi)d\varphi\end{aligned} \tag{2}$$

and the efficiency of the gears:

$$\eta_{z_i}=\left(1-\frac{N\cdot\tilde{P}_{fs_i}}{P_{in_i g}}\right). \tag{3}$$

Also it is possible to evaluate the role of efficiency factors.

The method of calculation allows a more accurate calculation of the power loss caused by tooth friction than before, and it takes into consideration the influence of the geometrical parameters of the gear mesh while applying such a method for the calculation of the coefficient of tooth friction that takes into account the rules of the EHL lubrication theory as well [5]; [9].

A new method of calculation has developed which makes it possible to choose the right roller bearing for heavy-duty planetary gearboxes in terms of their load capacity and life expectancy at an early stage of design. First the correlation between the load capacity, life expectancy and the mean diameter of certain types of roller bearings has to be determined [6]:

$$d_m(d) = \sqrt[\tilde{d}]{\frac{\sqrt[3]{\frac{16 \cdot M}{\tau_m \cdot \pi}}}{\tilde{c}}} \tag{4}$$

$$d_m(d) = \sqrt[\tilde{d}]{\frac{\sqrt[3]{\frac{32 \cdot M_{h_{max}}}{\sigma_m \cdot \pi}}}{\tilde{c}}} \tag{5}$$

$$d_m(d) = \sqrt[\tilde{d}]{\frac{\sqrt{\frac{16 \cdot V}{3 \cdot \tau_m \cdot \pi}}}{\tilde{c}}} \tag{6}$$

$$d_m(L_h) = \sqrt[\tilde{d}]{\frac{\sqrt[p]{\frac{L_h \cdot 60 \cdot n}{10^6 \cdot a_1}} \cdot F_r}{\tilde{c}}} \tag{7}$$

The resultant mean diameter of roller bearings can be calculated with the following equation:

$$d_{m_{res}} = \left[d_m(d) + \frac{1}{2} \cdot \left(\begin{array}{l} \left|\left(d_m(L_h) - d_m(d)\right)\right| \\ + \left(d_m(L_h) - d_m(d)\right) \end{array} \right) \right] \tag{8}$$

By comparing the results of calculation for individual types of bearing, the most suitable bearing, which meets the given demands, can be chosen [6]; [7].

Taking into consideration the geometrical data of the teeth, new mathematical correlations was developed to determine the oil splashing energy losses (9) and oil expel energy losses (10) of oil-lubricated gears.

$$P_p(\varphi_o) = z_o(\varphi_o) \cdot \dot{m}_{op} \cdot v_{r_{w3}}^2 + P_{ts}(\varphi_o) \tag{9}$$

$$P_{o_{ki}}(\varphi_{\Omega i}) = \frac{\rho \cdot c_{\rho k_i} \cdot \dot{V}_{o\ ki}^3(\varphi_{\Omega i})}{\Sigma A_{e\ yti}^2(\varphi_{\Omega i})} \tag{10}$$

With the correlations it is possible to calculate fairly accurately, from an engineering point of view, the oil churning losses (which are of hydraulic origin) of the gears in planetary gearboxes [8]-[9].

On the basis of the models (Figure 1.) and equations developed, by applying the thermal network model, the temperature of the critical elements in the gearbox can be calculated at an early phase of design.

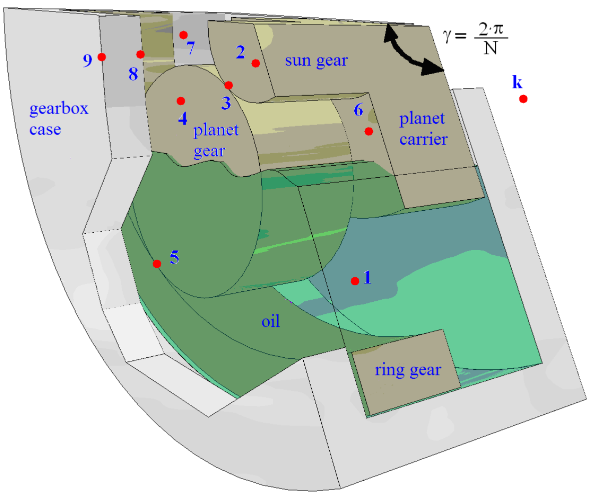

Figure1. *The thermo model and the thermo points of the OI planetary gear*

The important thermo points are described below.

k - The air temperature around the gearbox.

1 - The oil sump temperature. The source of heat given by the oil churning losses.

2 - Temperature of the sun gear bearing. The source of heat given by the bearing power losses.

3 - Tooth contact temperature in the contact of sun gear and planet gear. The source of heat given by the tooth friction losses.

4 - Temperature of the planet gear bearing. The source of heat given by the bearing power losses.

5 - Tooth contact temperature in the contact of planet gear and ring gear. The source of heat given by the tooth friction losses.

6 - Temperature of the planet carrier bearing. The source of heat given by the bearing power losses.

7 - Air temperature inside the gearbox.

8 - The inner surface temperature of the gearbox case.

9-The outside gearbox case surface temperature.

Assuming that the tooth contact can be described with an average temperature, the oil and the oil sump can be described with an average temperature, the inner side of the case can be described with an average temperature, the outside surface of the case can be described with an average temperature, the average temperature of the bearings can be described with an average temperature and the thermo points are defined in a coordinate system connected to the planet carrier, the heat transfer matrix of an OI planetary gear is given below (Figure 2).

	k	1	2	3	4	5	6	7	8	9
k	-	-	-	-	-	-	-	-	-	**K,S**
1	-	**H**	**K**	**K**	**K**	**K**	**K**	**K**	**K**	-
2	-	**K**	**H**	**V**	-	-	-	**K**	-	-
3	-	**K**	**V**	**H**	**V**	-	-	**K**	-	-
4	-	**K**	-	**V**	**H**	**V**	-	**K**	-	-
5	-	**K**	-	-	**V**	**H**	-	**K**	**V**	-
6	-	**K**	-	-	-	-	**H**	**K**	-	-
7	-	**K**	**K**	**K**	**K**	**K**	**K**	**H**	**K**	-
8	-	**K**	-	-	-	**V**	-	**K**	-	**V**
9	**K,S**	-	-	-	-	-	-	-	**V**	-

Figure2. *Heat transfer matrix of an OI planetary gear. H – source of heat, S – heat radiation, V – heat conduction, K – heat convection*

The developed equations for calculating the average temperature of the thermo points can be derived from the heat transfer matrix.

Equation for thermo point „**k**":

$$\alpha_6 \cdot A_8 \cdot (T_k - T_9) + 0{,}0507 \cdot \varepsilon \cdot A_8 \cdot \left[\left(\frac{T_k}{100}\right)^4 - \left(\frac{T_9}{100}\right)^4\right] - \dot{Q} \ = 0 \tag{11}$$

Equation for thermo point „**1**":

$$\begin{aligned} &\dot{Q} \ + \alpha_1 \cdot A_1 \cdot (T_1 - T_2) + \alpha_2 \cdot A_2 \cdot (T_1 - T_3) + \alpha_1 \cdot A_3 \cdot (T_1 - T_4) + \alpha_2 \cdot A_2 \cdot (T_1 - T_5) + \\ &+ \alpha_1 \cdot A_4 \cdot (T_1 - T_6) + \alpha_3 \cdot A_5 \cdot (T_1 - T_7) + \alpha_4 \cdot A_6 \cdot (T_1 - T_8) = 0 \end{aligned} \tag{12}$$

Equation for thermo point „**2**":

$$\alpha_1 \cdot A_1 \cdot (T_2 - T_1) + \dot{Q} \ + \frac{2 \cdot \pi \cdot b \cdot \lambda_1}{\ln\left(\frac{d_{w2}}{d_{m2}}\right)} \cdot (T_2 - T_3) + \alpha_5 \cdot A_1 \cdot (T_2 - T_7) = 0 \tag{13}$$

Equation for thermo point „**3**":

$$\alpha_2 \cdot A_2 \cdot (T_3 - T_1) + \frac{2 \cdot \pi \cdot b \cdot \lambda_1}{\ln\left(\frac{d_{w2}}{d_{m2}}\right)} \cdot (T_3 - T_2) + \dot{Q} \ + \frac{2 \cdot \pi \cdot b \cdot \lambda_1}{\ln\left(\frac{d_{w3}}{d_{m3}}\right)} \cdot (T_3 - T_4) + \alpha_5 \cdot A_2 \cdot (T_3 - T_7) = 0 \tag{14}$$

Equation for thermo point „**4**":

$$\alpha_1 \cdot A_3 \cdot (T_4 - T_1) + \frac{2\cdot\pi\cdot b\cdot\lambda_1}{\ln\left(\frac{d_{w3}}{d_{m3}}\right)} \cdot (T_4 - T_3) + \dot{Q} \; + \frac{2\cdot\pi\cdot b\cdot\lambda_1}{\ln\left(\frac{d_{w3}}{d_{m3}}\right)} \cdot (T_4 - T_5) + +\alpha_5 \cdot A_3 \cdot (T_4 - T_7) = 0 \tag{15}$$

Equation for thermo point „**5**":

$$\alpha_2 \cdot A_2 \cdot (T_5 - T_1) + \frac{2\cdot\pi\cdot b\cdot\lambda_1}{\ln\left(\frac{d_{w3}}{d_{m3}}\right)} \cdot (T_5 - T_4) + \dot{Q} \; + \alpha_5 \cdot A_2 \cdot (T_5 - T_7) + + \frac{2\cdot\pi\cdot b\cdot\lambda_1}{\ln\left(\frac{l_{ház}}{d_{w4}}\right)} \cdot (T_5 - T_8) = 0 \tag{16}$$

Equation for thermo point „**6**":

$$\alpha_1 \cdot A_4 \cdot (T_6 - T_1) + \dot{Q} + \alpha_5 \cdot A_4 \cdot (T_6 - T_7) = 0 \tag{17}$$

Equation for thermo point „**7**":

$$\begin{aligned} &\alpha_3 \cdot A_5 \cdot (T_7 - T_1) + \alpha_5 \cdot A_1 \cdot (T_7 - T_2) + \alpha_5 \cdot A_2 \cdot (T_7 - T_3) + \alpha_5 \cdot A_3 \cdot (T_7 - T_4). \\ &+ \alpha_5 \cdot A_2 \cdot (T_7 - T_5) + \alpha_5 \cdot A_4 \cdot (T_7 - T_6) + (\dot{Q}) + \alpha_5 \cdot A_6 \cdot (T_7 - T_8) = 0 \end{aligned} \tag{18}$$

Equation for thermo point „**8**":

$$\alpha_4 \cdot A_6 \cdot (T_8 - T_1) + \frac{2\cdot\pi\cdot b\cdot\lambda_1}{\ln\left(\frac{l_{ház}}{d_{w4}}\right)} \cdot (T_8 - T_5) + \alpha_5 \cdot A_6 \cdot (T_8 - T_7) + \frac{\lambda_2 \cdot A_7}{\tilde{s}} \cdot (T_8 - T_9) = 0 \tag{19}$$

Equation for thermo point,,**9**":

$$\frac{\lambda_2 \cdot A_7}{\tilde{s}} \cdot (T_9 - T_8) + \alpha_6 \cdot A_8 \cdot (T_9 - T_k) + 0{,}0507 \cdot \varepsilon \cdot A_8 \cdot \left[\left(\frac{T_9}{100}\right)^4 - \left(\frac{T_k}{100}\right)^4\right] = 0 \tag{20}$$

Solving these equations the oil sump and tooth temperatures can be determined.

With the method shown above the scuffing load carrying capacity of the gears can be evaluated, and knowing the oil temperature and energy loss, it is possible to choose the most suitable lubricant. The calculations made with the changes in the formation and the size of the structural elements, it is possible to develop a more advantageous gearbox in terms of its thermal capacity.

Calculated Results and Conclusions

By applying the correlations presented above, we can calculate the tooth friction power for O and I tooth connections in an OI planetary gearbox (figure 3) as well as the oil churning losses. By using the thermal network model, also the temperature distribution can be calculated in the planetary gear.

Main parameters of the OI planetary gear: z_2 = z_3 = 36; z_4 = 108; m_n = 0,8 [mm]; a_w = 28,8 [mm]; β = 0 [°]; x_1 = Σx = 0; h_a* = 1; c_0 = 0,25; N = 3; b = 15 [mm]; R_a = 0,4 [μm]; M_2 = 30 [Nm]; n_1 = 1500 [1/min]; T_0 = 60,5 [°C]; η_0 = 0,162 [Pas]; dip lubrication, b_{m3} = 23 [mm].

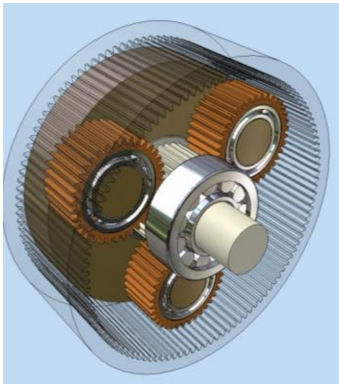

Figure3. *The OI planetary gearbox investigated in the research*

The following figures show the results in the sun gear – planet gear mesh.

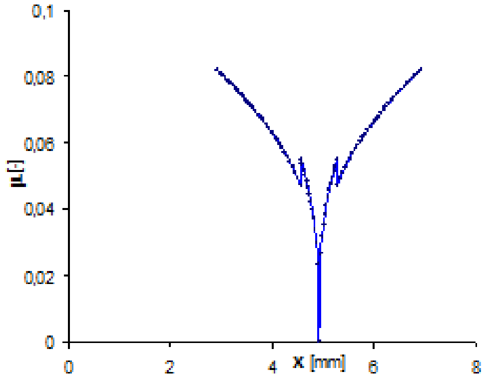

Figure4. *The coefficient of friction as a function of the contact line in the external connection (n2=1500 [rpm])*

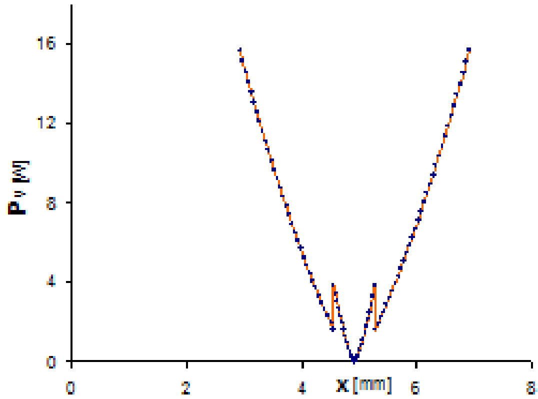

Figure5. *The tooth power loss as a function of the contact line in the external connection (n2=1500 [rpm])*

The following figures show the results in the planet gear – ring gear mesh.

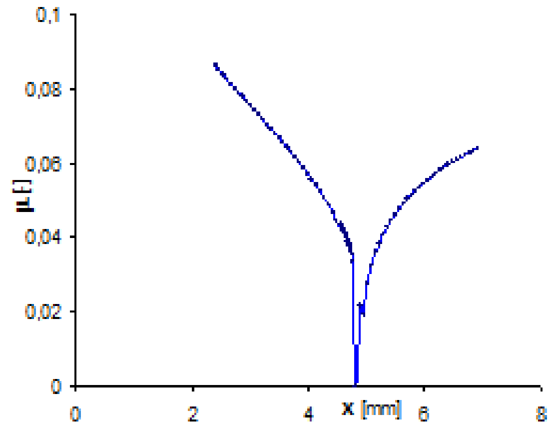

Figure6. *The coefficient of friction as a function of the contact line in the internal connection (n2=1500 [rpm])*

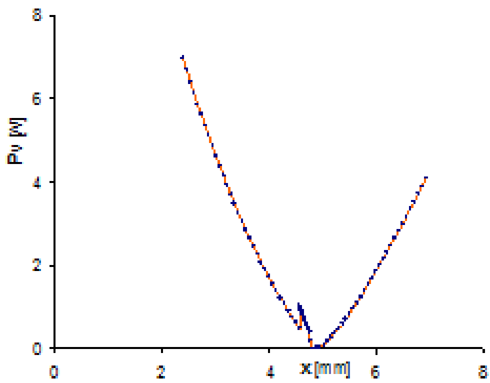

Figure7. *The tooth power loss as a function of the contact line in the internal connection (n2=1500 [rpm])*

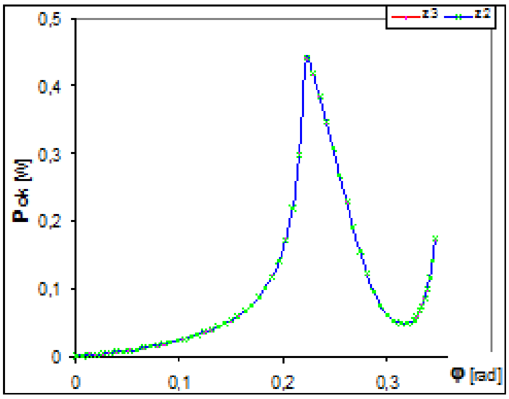

Figure8. *The power loss of oil expel as a function of the gear rotation in the external connection (n2=1500 [rpm])*

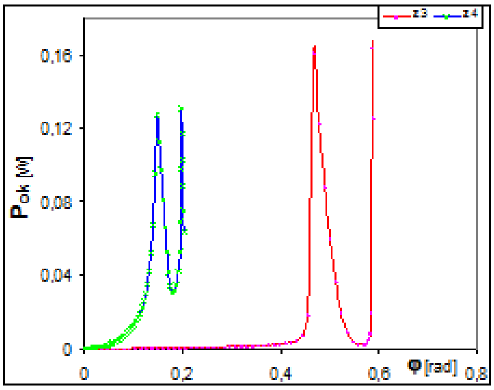

Figure9. *The power loss of oil expel as a function of the gear rotation in the internal connection (n2=1500 [rpm])*

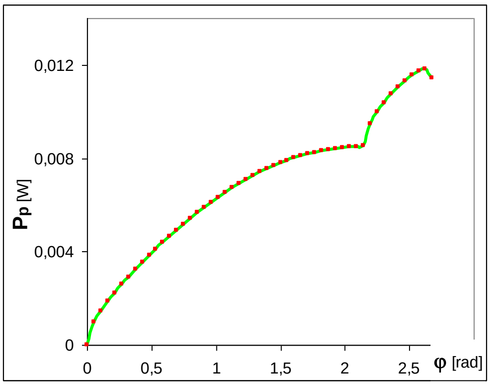

Figure10. *The power loss of splashing as a function of the planet carrier rotation (n_2=1500 [rpm])*

Figure 11. show the calculated temperatures in each thermo points. The results show that increasing the input speed the temperatures of the element are increasing relatively fast.

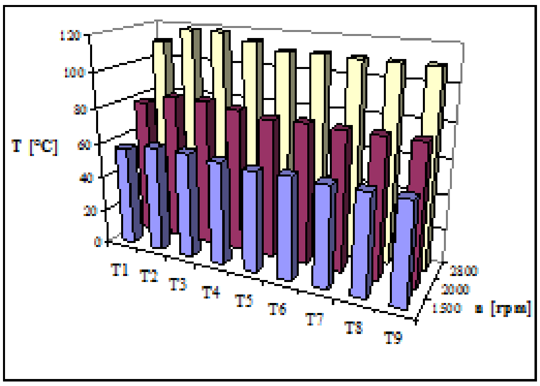

Figure11. *The calculated temperatures of the thermo points in the OI gear*

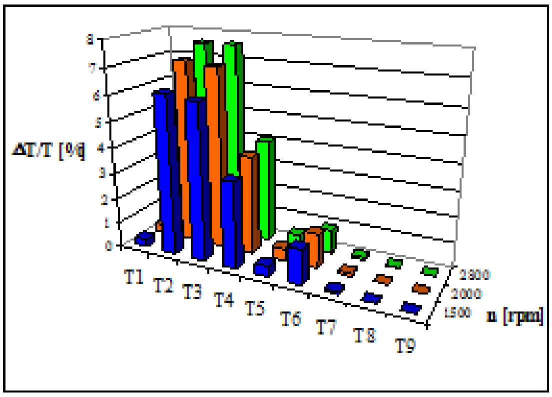

Figure12. *Relative temperature differences between thermo points and the average gearbox case temperature as a function of input speed (ΔT/T= (Ti-T9)/T9]*

Figure 12. show the temperature difference between the elements comparing to the gearbox case temperature. It can be seen that the tooth temperatures are about eight percent higher than the case temperature and this difference is getting bigger increasing the input power. This is why it is important to calculate the tooth, the oil and the bearing temperatures as precisely as it is possible to determine the heat carrying capacity of the planetary gear.

A good agreement between the calculated and measured temperatures of the oil sump is shown in the figure below (Figure 13).

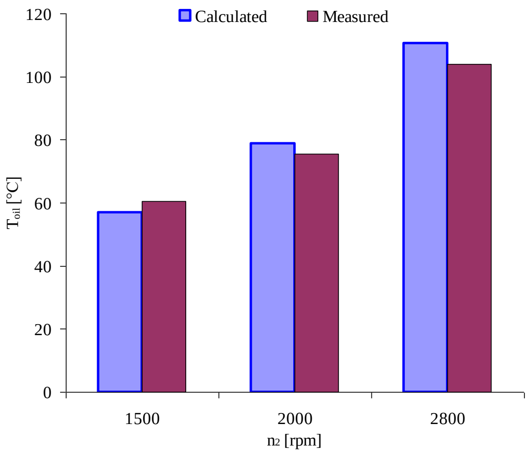

Figure13. *The changes in the temperature of the oil bath plotted against the revolutions per minute of the input drive (sun gear axis)*

Nomenclature

a – exponent for bearing calculation [-]

a_1– factor for bearing life correction [-]

A_1– area of the sun gear bearing devided by N [m^2]

A_2 – area of gear tooth [m^2]

A_3 – area of planet gear bearing devided by N [m^2]

A_4 – area of carrier devided by N [m^2]

A_5 – area of oil connection with the air inside the gaerbox [m^2]

A_6 – area of the inner side of gearbox case devided by N [m^2]

A_7 – average area of gearbox case devided by N [m^2]

A_8 – area of the outside of gearbox case devided by N [m^2]

Ae_{yti} – cross sections of the oil flow [m^2]

a_w – center distance [mm]

α_τ – working pressure angle [°]

α_1 – heat transfer coefficient between bearing and gear [W/m^2°C]

α_2 – heat transfer coefficient between oil and gear teeth [W/m^2°C]

α_3 – heat transfer coefficient between oil and air

[W/m^2°C]

α_4 – heat transfer coefficient between oil and gearbox case [W/m^2°C]

α_5 – heat transfer coefficient between oil and air [W/m^2°C]

α_6 – heat transfer coefficient between gearbox case and air [W/m^2°C]

$\alpha_o(\varphi_o)$ – maximal immersion angle [°]

b – pinion with [mm]

b_m –immersion depth [mm]

β – helix angle [°]

c~ – developed constant for bearing calculations [-]

$c_{\rho k}$ – correction factor for oil expel calculation[-]

d – inner diameter of bearing [mm]

$d_m(d)$ – average bearing diameter as a function of the inner diameter[mm]

$d_m(L_h)$ – average bearing diameter as a function of the prescribed bearing lifetime[mm]

$d_{m\ res}$ – resultant average bearing diameter [mm]

d – developed constant for bearing calculations [-]

ε – thermal emission coefficient of the gearbox case [-]

η_M – viscosity at operating temperature [mPas]

η_z – tooth efficiency [-]

F_r – bearing radial load component [N]

g_α – length of the contact line [mm]

φ – parameter, pinion or gear turn angle [°]

i – gear ratio [-]

λ_1 – thermal conductivity of the gear [W/m°C]

λ_2 – thermal coductivity of the gearbox case [W/m°C]

M – torque [Nmm]

M_{hmax} – maximal bending torque [Nmm]

$M_{2;4}$ – sun or ring gear torque [Nm]

$\dot{m}_{op}$ – splashed oil mass flux [kg/s]

m_n – normal modul [mm]

μ – coefficient of tooth friction along the contact line [-]

n – bearing rotational speed [rpm]

N – number of planet gears [-]

n_{in} – input speed [rpm]

$P_{fs}(\varphi)$ – tooth power loss along meshing [W]

$P_{fs}(\varphi)$ – average tooth power loss along meshing [W]

$P_{fcs}(\varphi)$ – power loss of sliding between the teeth along meshing [W]

$P_g(\varphi)$ – power loss of rolling between the teeth profiles along meshing [W]

P_{in} – input power [W]

$P_{in\ g}$ – input rolling power [W]

$P_{oki}(\varphi)$ – power loss of oil expel [W] $P_p(\varphi_o)$ – power loss of splashing [W]

$P_{ts}(\varphi)$ – disc churning loss [W]

Q` – heat flux [W]

R_a – average surface roughness (CLA)

ρ – oil density [kg/m^3]

σ_m, τ_m – allowable normal and shear stress components [MPa]

T_0– air temperature [°]

V – shear load of planet gear pin [N]

v_{rw3} – tangential speed of planet gear [m/s]

$\dot{V}_{ok}(\varphi_\Omega)$ – flow rate, oil volume flux for oil expel calculations [m^3/s]

x – tooth profile shift factor [-]

$z_o(\varphi_o)$ – number of submerging teeth [-]

z – tooth number [-]

$_2$ – index of sun gear

$_3$ – index of planet gear

$_4$ – index of ring gear

$_k$ – index of carrier

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