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The Effect of Lubricant Viscosity Variation on Tooth Friction and Hence on the Energetic Behaviour of Gear Unit

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Earlier this year, I created a mathematical model of a gear unit that allowed me to investigate the energy performance of a vehicle gear unit over its full operating range. In the model created, the value of the tooth friction was determined during a precalculation and then entered into the model as a constant parameter. The problem with this approach is that it is not automated, so during the energy analysis or a possible optimisation of the gear unit, the tooth friction is always present in the model as a constant value, whereas in reality, the value of the tooth friction is different for different working points (wheel torque: M_w, wheel speed: n_w) and different geometries. The viscosity of the selected lubricant has a significant influence on tooth friction, which varies partly due to temperature and partly due to load. In this paper, I will investigate the viscosity of the oil required by the gear unit at different operating points for proper operation and how this affects tooth friction and, hence, the gear unit energy loss. Since the studies showed that the effect of oil viscosity is not significant but not negligible, the determination of tooth friction can be incorporated into the gear unit model using a MATLAB function so that it is re-determined at each operating point and for each geometry change. With this modification, the estimation generated by the mathematical model describing the energetic behavior of the gear unit can be made more accurate.

1. Introduction

The lubricant and lubrication method for gear drives should be specified at the design stage, as it has an impact on both the damage-free operation of the gears and the losses of the gear unit. When designing the lubrication system for the gear unit, different solutions are possible, such as oil lubrication, grease lubrication and, in special cases, lubrication with dry solid lubricant applied physically or chemically to the tooth surface (Istenes et al., 2023).

The lubrication method and lubricant quantity for gear drives with the associated lubrication equipment must be determined at the design stage. A damage-free inspection of the gears can only be carried out if the lubricant is known exactly. When designing a gear drive, the power, speed and installation environment are usually taken into account when determining the design of the gear, which also determines the type of lubricant and lubrication. If the structure allows, it is advisable to use oil lubrication. The advantages of oil lubrication are, for example, a continuous supply of lubricant, good heat removal and thus cooling and lubrication of bearings and seals, with the only disadvantage being the need to ensure that the gearbox housing is sealed (Concli, 2016).

The selection of gearbox oil means determining the viscosity and power level required. For cylindrical and bevel gearboxes, a quick and simple method of determining the required viscosity of the oil according to DIN 51509/1 has been applied. It should be noted that this procedure has been automated using a MATLAB function, so that the lubricant selection is part of the objective function (Horváth and Kőrös, 2015).

The numerous loss models identified and presented in the literature survey provide insight into the loss calculation methods for gears and gear units. In many cases, the gear efficiency calculation methods in the literature have a narrow range of validity, as they do not take into account the change in lubrication condition over the full operating range. This study investigates this shortcoming in an attempt to improve the accuracy of energy loss determination methods (Guzzella and Sciarretta, 2005).

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2. Lubricating oil selection

Selecting the transmission oil means determining the viscosity and power level required. Determination of the viscosity of the lubricating oil is necessary to achieve proper elasto-hydrodynamic lubrication (Szalai et al., 2022). For cylindrical and bevel gear drives, a quick and simple method of determining the required oil viscosity according to DIN 51509/1 has been applied, the method being to determine the kinematic viscosity of the oil at 40°C from a diagram based on the load/speed factor (k_s/v) typical of a gear drive (Caporusso et al., 2022).
The Stribeck surface pressure (k_s) is defined by the following relationship Eq(1):

$$k_s = \frac{F_t}{b \cdot d_t} \cdot \frac{u+1}{u} \cdot Z_H^2 \cdot Z_\varepsilon^2 \tag{1}$$

F_t – tangential force on the pitch circle, N
b – gear width, mm
d_1 – small wheel rolling circle diameter, mm
u – gears tooth ratio, -
Z_H – rolling circle factor, -
Z_ε – connection number factor, -
In the pre-planning phase, when some parameters still vary, the following estimation is used changing:
$Z_H^2 * Z_\varepsilon^2 \approx 3$. According to automotive test data, the value of the load speed factor (k_s/v) usually varies in the order of 0...20 therefore the relationship between the load speed factor and the kinematic viscosity of the lubricating oil has been defined in this range is shown in Figure 1. Looking at the figure, it can be seen that for higher speed and lower load working points, a lower viscosity lubricating oil is sufficient, while for higher load working points, a transmission oil with a higher kinematic viscosity is required.

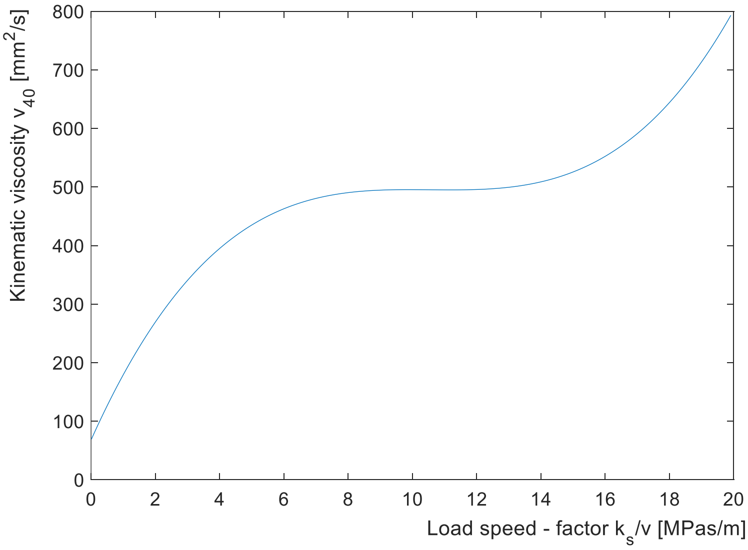

Figure 1: Relationship between load speed coefficient and kinematic viscosity of lubricating oil

In order to select the transmission oil, it is useful to know the critical points for lubrication, which has been done by first defining the full operating range of the vehicle with its corresponding working points (De Guido et al., 2019). The gear unit would be installed in a small urban electric vehicle, so the vehicle data was determined accordingly.
The parameters of interest for the gear unit test are the vehicle mass (500 kg), the tyre size (145/70 R12 (dg=509 mm)) and the maximum speed of the vehicle (v_{max}=60 kg). Based on the above data, the test range is n_w=0...700 rpm wheel speed range and the torque range from the tractive effort demand M_w=1...50 Nm.
The parameter vector of the gear unit was determined as follows Eq(2):

p=*[6, 1.5, 78.75, 15, 20, 0]* (2)

2.1 Drive unit parameter vector determination

Drive unit model is constructed so, that drive unit loss is determined by steady-state examination at work points. This work point examination makes both dynamical and analytical mathematical model application possible during optimization. Parameter determination has key importance in model construction, by which drive unit can be optimized (Polák and Lakatos, 2015). These parameters are such independent variables, which can unambiguously determine a given drive unit (Polák and Lakatos, 2016). Parameter vectors create a space, where drive unit optimization is realized. Drive unit parameter vector Eq(3):

$\mathbf{p}[i, m, a_w, b, \alpha, \beta]$ (3)

where:

- transmission ratio, - i
- module, mm m
- centre distance, mm a_w
- gear width, mm b
- pressure angle, rad α
- helix angle, rad β

2.2 Selecting the lubricating oil

Using the parameters defined above, the oil_selection function was run, which calls the oil_viscosity function during its execution and determines the viscosity of the gear oil for each working point, from which the highest viscosity value can be selected to select the particular gear oil. The oil viscosity value determined at the given working points is shown in Figure 2. From the tribological point of view, the unfavourable situations where the torque is high but the speed is low are mostly the acceleration situations at start-up.

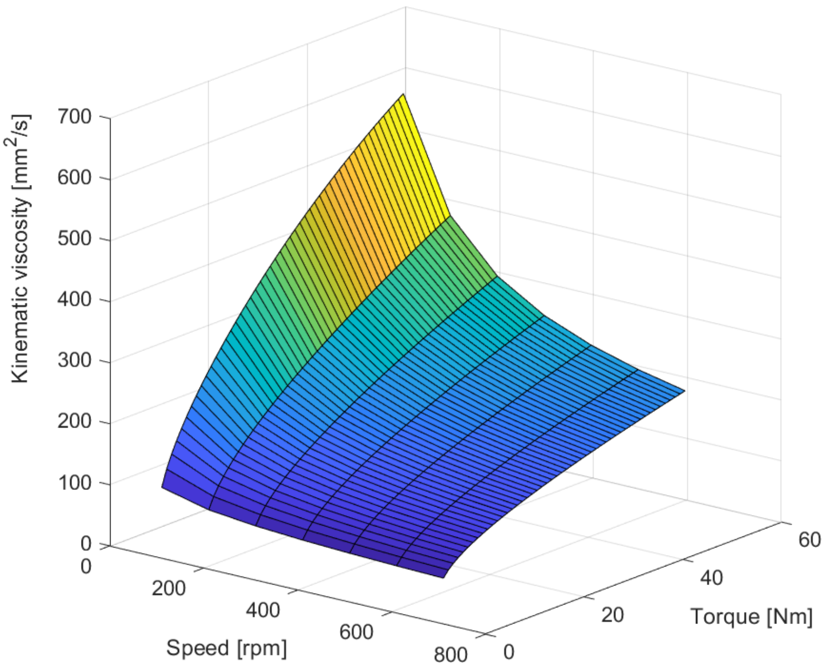

Figure 2: Variation of the viscosity of the lubricant under different loading conditions

3. Determination of tooth friction

The tooth friction models, from which the model used was selected, are suitable for determining the average tooth friction coefficient (Németh and Fischer, 2021). The value of μ_m was calculated for the same geometry, lubrication and operating conditions to test the comparison of the different procedures. As a result of the study, the Schlenk model was chosen to determine the value of μ_m, which is described by the following relationship Eq(4):

$$\mu_m = 0{,}048 \cdot \left(\frac{F}{b \cdot \mathrm{v}_{\Sigma C} \cdot \rho_{redc}}\right)^{0{,}2} \cdot \eta^{-0{,}05} \cdot R_a^{0{,}25} \cdot X_L \tag{4}$$

F – tangential force on the pitch circle, N
b – gear width, mm
$v_{\Sigma C}$ – the speed of rolling of the tooth profiles together along the contact line; m/s
ρ_{redC} – equivalent radius of curvature at the principal point, mm
η – dynamic viscosity of the lubricant at operating temperature, Pas
R_a – the average of the mean surface roughness of the contact teeth of the gears connected; μm
X_L – lubricant factor, -

In the case of the previously created gear model, the value of tooth friction was determined during a pre-calculation and then entered into the model as a constant parameter. The problem with this approach is that it is not automated, so during the optimization the tooth friction is always present in the model as a constant value, while in reality the value of tooth friction is different for different working points (M_w, n_w) and different geometries. Due to the problem described above, the tooth friction is incorporated in the gearbox model using a MATLAB function, so that the tooth friction is re-determined at each working point and for each geometry change (Lakatos and Titrik, 2015).

Main parameters of the study

- The vector of gear unit parameters: p = [6 1.5 78.75 15 0.349 0]
- Load range of the gear unit: wheel speed n_w=25...700 rpm, the torque range from the tractive effort M_w=2...50 Nm.
- Selected lubricating oil viscosity: v=680 mm^2/s.

During the model run, the tooth friction coefficient per working point was determined and its evolution is shown in Figure 3. From this figure, the minimum and maximum values of tooth friction and the nature and extent of its variation can be determined. The difference between the two extremes of tooth friction is almost 4 times are listed in Table 1.

Table 1: Maximum and minimum value of tooth friction

Torque, Nm	Speed, rpm	tooth friction, [-]
50	25	0.1051
2	700	0.02836

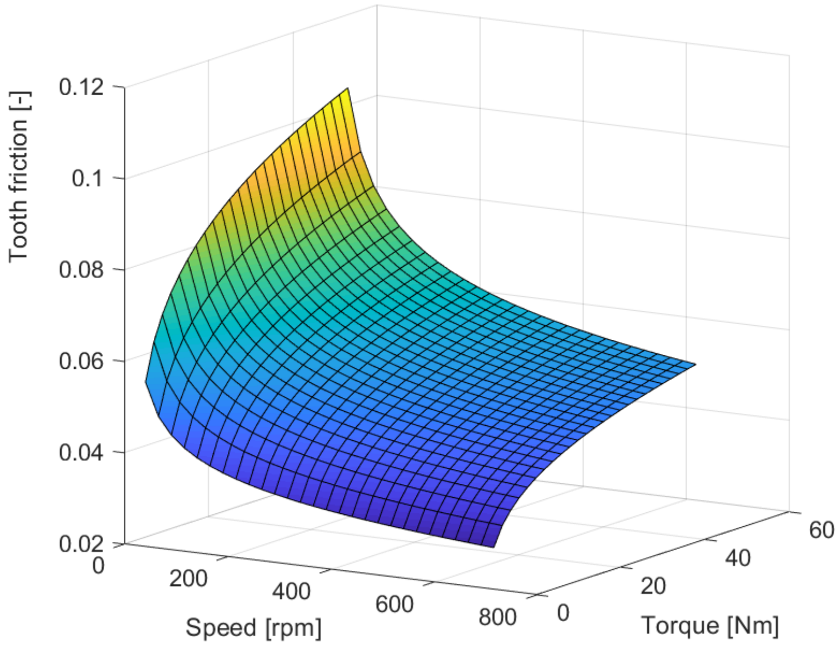

Figure 3: Change in tooth friction over the full load range

4. Parameter sensitivity test

To carry out the sensitivity test, the working points (M_w, n_w), the dwell time at working points (t) and the vector of the actuator parameter (**p**) were defined, these values are given in Table 2.

Table 2: Setting parameters to perform the sensitivity test

working points	1	2	3	4	5	6
M_w, Nm	15.02	17.84	22.46	28.96	37.27	47.52
n_w, rpm	104.23	208.46	312.7	416.93	521.16	625.93
t, s	300	300	300	300	700	300
Vector of gear unit parameter				**p** = [6 1.5 78.75 15 0.349 0]		

The results of the study are shown in Table 3 and Figure 4.

Table 3: Sensitivity of the gearbox model to changes in lubricating oil viscosity

Standard viscosity levels v, mm^2/s	Cumulative energy loss W_l, J
22	129,081.165
32	128,392.09
46	127,740.005
68	127,053.574
100	126,390.533
150	125,705.143
220	125,063.963
320	124,437.487
460	123,828.297
680	123,170.558
1000	122,527.075

From the analysis, it can be concluded that a unit change in viscosity level resulted in an average cumulative loss energy change of ~0.5 % (~596 J), but since the change function is non-linear, this interpretation is negligible. Considering the whole viscosity range, the change is ~5 % (~6,554 J). Analysing Figure 4, it can be concluded that the effect of viscosity change on the energetic behaviour of the gear unit is not significant but noticeable and, therefore, relevant for optimisation.

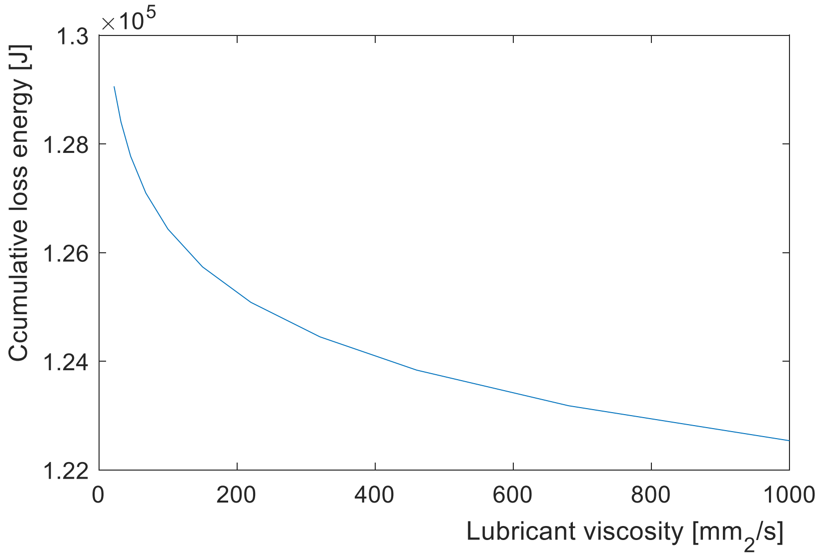

Figure 4: Relationship between cumulative loss energy change and lubricant viscosity

5. Conclusions

Earlier, a mathematical model of the gear unit was created, which allows for the energetic analysis of a vehicle gear unit over its entire operating range. The shortcoming of this model is that the value of the tooth friction was determined during a precalculation and then entered into the model as a constant parameter, thus not taking into account that, in reality, the tooth friction varies with the change in tooth geometry and the working points.
Tooth friction is also significantly influenced by the viscosity of the lubricant selected, which varies partly as a result of temperature and partly as a result of load. The examinations presented in this paper have shown the viscosity of the oil required by the gear unit at different operating points for proper operation and how this affects tooth friction and, hence, the gearbox energy loss.
The studies showed that the effect of oil viscosity is tangible, so the determination of tooth friction and oil viscosity is incorporated into the gearbox model using a MATLAB function so that it is re-determined at each working point and for each geometry change. With this modification, the estimation of energy loss generated by the mathematical model describing the energy behaviour of the gear unit has been made more accurate, helping designers to define near-optimal gear unit parameters already in the pre-design phase.

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