Is Splash Lubrication Compatible with Efficient Gear Units for High-Speed Applications?

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A thermo-mechanical model of a splash lubricated one-stage gear unit is presented. This system corresponds to a first step towards the design of a hybrid vehicle gearbox that can operate up to 40,000 rpm on its primary shaft. The numerical model is based on the thermal network method and takes into account power losses due to teeth friction, rolling-elements bearings and oil churning. Some calculations underline that oil churning causes a high amount of power loss. A simple method to reduce this source of power losses is presented, and its influence on the gear unit efficiency and its thermal capacity is computed.

Introduction

In automotive applications, there is an increased demand for hybrid technology that combines an internal combustion engine and one or more electric motors. In order to increase power densities of these systems, it is possible to use electric motors that run at high rotational speeds: up to 30,000-50,000 rpm. One limitation to this evolution relies on the design of efficient gearboxes at acceptable cost level. Considering medium-speed applications, splash lubrication appears as the most appropriate and economical solution to move the lubricant onto the gear teeth and bearings. Prior to performing tests on actual gear drives, the aim of this study is to investigate if this lubrication technique is worth considering for the above-mentioned, high-speed application.

To this end, a numerical model of a one-stage helical gear unit has been fashioned. This model can predict the temperature distribution and the efficiency of a mechanical transmission via analytical laws. The following sources of dissipation are taken into account:

- Tooth friction
- Rolling-elements bearings
- Oil churning

As far as the thermal behavior is concerned, the gear unit is divided into lumped elements with a uniform temperature connected by thermal resistances that account for conduction, free or forced convection, and radiation. Particular attention is paid to the oil sump behavior, including the use of specific heat exchange relationships between the lubricant and some rotating elements.

Compared to the finite element method, a thermal network model requires less computing time and provides significant information on temperature distribution. It enables testing of many different assumptions, or operating conditions, and the quantification of their influences on both temperature and power losses, by considering the strong coupling between these physical parameters. This method has been often used to study heat transfer in mechanical transmissions (Refs. 1–4).

In this paper, a thermal network model is presented that can predict the temperature distribution and the power losses of a one-stage gearbox lubricated by an oil sump. Calculations were performed to determine the major sources of power losses; a method to reduce oil churning power loss is exposed and some calculations are drawn to emphasize potential savings on the complete geartrain.

System under Consideration

This study deals with a one-stage helical gear unit that will be designed as the first reduction stage of an electric powertrain. The system is composed of two shafts and the gears are designed to withstand a transmitted power of 100 kW on a speed range from 6,500 – 40,000 rpm. The gear ratio is limited to one-third because the final gearbox is planned to have three stages. All shafts are mounted on ball bearings. The main geometrical data of the gear unit are given in Table 1. The whole set is enclosed in a housing. It is parallel-piped with the following external dimensions: $160 \times 90 \times 200\,mm^3$. Assumptions are made concerning the environment of the gearset: the housing is set on the ground and placed in a ventilation stream (these assumptions are in accordance with the future test rig to be designed). The housing is filled with a certain amount of lubricant to ensure splash lubrication. But bearings are not intended to be immerged in the sump; their lubrication is performed by using channels fed by oil projections via the rotating elements. Oil properties considered in this study are given in Table 2.

Modeling of the Thermal Behavior

Thermal network. In order to simulate the steady-state temperature distribution in the above-mentioned system, the thermal network method has been used. The thermal network relies on the decomposition of the test rig into isothermal elements (gears, shafts, bearings, etc.) and the con-

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Table 1 Main features of the gearset

	Pinion	Wheel
Center distance (mm)	88	88
Tooth face width (mm)	28	27
Module (mm)	2	2
Tooth number	20	61
Pressure angle (°)	20	20
Helix angle (°)	14	14
Arithmetic roughness (μm)	0.8	0.8
Mean diameter of bearings (mm)	51	46
Bearings preload (N)	300	300

Table 2 Lubricant properties

Kinematic viscosity at 40°C (cSt)	Kinematic viscosity at 100°C (cSt)	Fluid density at 15°C (kg/m^3)
47.6	8.3	887.3

nection of these elements by thermal resistances (Ref. 1). The differences in temperature are created by the heat flow between each element. Thermal resistances depend on the kind of heat transfer encountered, i.e. — conduction, free or forced convection, and radiation. Finally, power losses are computed considering nodes temperature to ensure thermo-mechanical coupling. In this study, thermal resistance and power loss are estimated by using analytical formulas.

The proposed thermal network is given in Figure 1; the gearbox has been divided into 15 elements (Table 3). As far as boundary conditions are concerned, some temperatures were considered as input parameters for the thermal model, i.e. — the ambient air (node #1) and the ground (node #2). The casing is decomposed into (a) a lower part that is in contact with the ground (node #5) — (b) a lateral part where bearings are located (node #4) and (c) an upper part (node #3). The bearings are assumed to be isothermal and are represented by a sole element (nodes #7 – #10). Node #6 corresponds to the lubricant. Node #15 represents the meshing zone of gear teeth — a small area shared between the two gears (nodes #13 and #14) where friction occurs.

In order to evaluate heat exchanges, the model uses four types of thermal resistance:

1. The thermal resistance of striction is based on Blok's works (Ref. 5). Heat generated by teeth friction is localized on a very small area compared to the gears' dimensions. The temperature increase is confined to a thin thermal skin whereas the bulk temperature is not affected. In the thermal network, this resistance links a node associated with the meshing of gear teeth and a node which relies on the gear bulk temperature.
2. To evaluate convection and radiation between the surrounding air and gearbox, Newton's and Stephan-Boltzmann's laws are respectively applied (Ref. 4). The casing is assimilated to an assembly of plates and the classic correlations for flat plates are used to quantify the air-casing convection.
3. Conduction between elements is calculated with classical formulations of heat transfer by conduction (Ref. 1). As an example, several elements (bearings, shafts) are represented as cylindrical bodies and the use of Fourier's law enables to determine the corresponding resistances.
4. Convection with oil is quantified by using different relationships (Ref. 4). Heat transfer with gears running partly immersed in the oil sump is characterized by using standard correlations for a rotating disk, but an additional thermal resistance is also introduced that accounts for the heat removal by centrifugal fling off. As previously mentioned, the casing is modeled as an assembly of flat plates. Then classical heat transfer relationships for forced convection and flows over flat plates are used to quantify the coefficient of convection between the lubricant and the housing.

Power losses. Four sources of power losses are identified in a one-stage gear unit lubricated with an oil sump. Power losses in geared transmissions are traditionally decomposed into no-load and load-dependent contributions. Load-dependent power losses are due to the meshing of gear teeth and internal friction of rolling-elements bearings. No-load-dependent power losses are generated by viscous forces in

Table 3 Elements of the thermal network

Number	Element reference
1	Air
2	Ground
3	Upper part of the casing
4	Lateral part of the casing
5	Lower part of the casing
6	Oil sump
7-8	Bearings on pinion's shaft
9-10	Bearings on wheel's shaft
11	Pinion shaft
12	Wheel shaft
13	Pinion
14	Wheel
15	Meshing zone of gear teeth

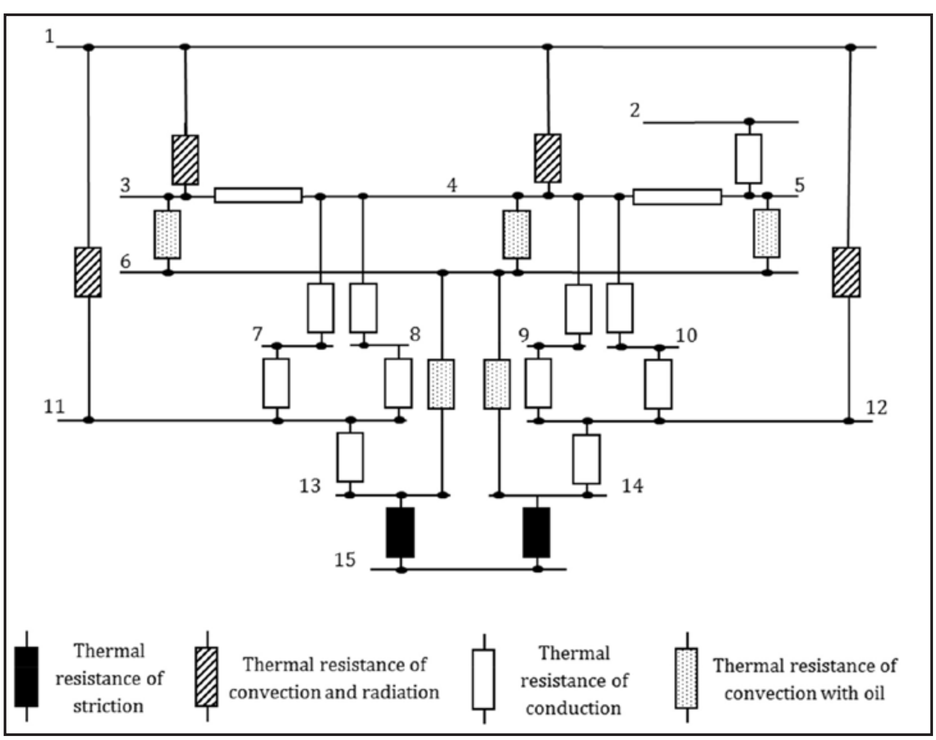

Figure 1 Thermal network of the one-stage gear unit.

rolling-elements bearings, oil churning, and seals friction. In the studied system it is assumed that labyrinth seals are used. They provide non-contact sealing action and then no-friction torque.

For each individual dissipation source implemented in the thermal network, a description is given hereinafter (P refers to power loss and C to friction torque):

(a) *The power losses due to teeth friction*: The heat-generated-per-unit-of-time by the friction between the mating teeth is calculated as a function of the following parameters: (i) the input power P_{input}, (ii) a geometrical factor H_v determined with equation from Velex and Ville (Ref. 6) and (iii) an average friction coefficient f determined with ISO14179-1 (Ref. 7). These power losses are injected to node #15:

$$P_{teeth\,friction} = P_{input} * Hv * f \tag{1}$$

(b) *The power losses due to friction in rolling-elements bearings*: To evaluate this source of power loss, the classical formulas developed by Harris are used (Ref. 8). It depends on the mean diameter of the bearing d_m, a parameter that takes into account bearing characteristics f_1, and the load P_1. These power losses are injected to nodes 7–10:

$$C_{friction} = f_1\, P_1\, d_m \tag{2}$$

(c) *The power losses due to viscous forces in rolling-elements bearings*: Harris' formulas are also used to quantify no-load-dependent power losses in the bearings. As shown (Ref. 1), this source of power loss is a function of oil kinematic viscosity v, the rotational speed n, the bearing mean diameter d_m and a factor f_0, which depends on the type of bearing and lubrication. These power losses are injected to nodes 7–10.

$$C_{hydrodynamic} = 10^{-7} f_0\,(vn)^{2/3}\, d_m^3 \tag{3}$$

(d) *The power losses due to churning phenomenon*: Formulas from Changenet et al are used to determine oil churning power loss (Ref. 9). The drag torque is expressed as a function of a dimensionless torque $C_{dimentionless}$ that depends on the fluid flow around a rotating gear, its rotational speed w, its pitch diameter D_p and the submerged surface area of the gear S_m. These power losses are injected to node #6.

$$C_{churning} = \frac{\rho\,\omega^2\, S_m\,(D_p/2)^3\, C_{dimentionless}}{2} \tag{4}$$

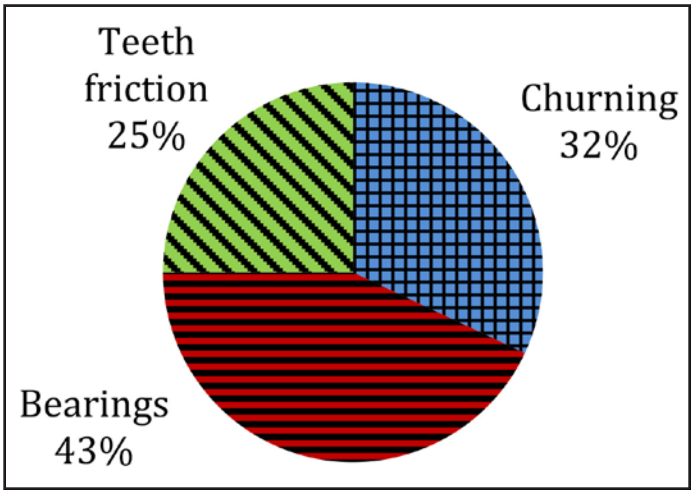

Figure 2 Power losses repartition at 100 kW and 32,000 rpm.

First results. To investigate the power losses repartition, some calculations were performed using different operating conditions. Figure 2 presents the results obtained for an input power of 100 kW at 32,000 rpm, and when oil level is assumed to be at 40 mm under the gears' axis. In this case the teeth friction represents only 25% of the global power losses. The drag torque — due to rolling-element bearings and oil churning — dominates and consequently highlights the strong influence of no-load-dependent power losses.

To improve the overall performance of the gear unit, the power losses must be controlled. A first step to reduce heat generation consists in limiting the dissipation associated with oil churning. Of course it is possible to reduce churning losses by lowering the oil level in the sump, but this action will also lead to higher gear bulk temperature. For example, at 100 kW and 6,400 rpm, if the oil level is decreased by 10 mm, churning loss is lowered by a factor of two. Simultaneously, the temperature difference between oil and the gear wheel increases by 10°C. Another solution to minimize churning losses without modifying temperatures consists of using axial flanges (Ref. 10).

Tests to Reduce Churning Losses

Test rig. A specific test rig was developed to investigate the housing influence on churning (Ref.10). A pinion shaft is operated by an electric motor that allows speeds up to 7,150 rpm. Churning losses are measured with a strain gauged contactless sensor with a full-scale range of 2 Nm. Because the pinion shaft is supported by two pairs of ball bearings, their contribution to the total drag torque has been experimentally determined and subtracted from global torque measurements. The housing is a parallelepiped with a face made of Plexiglas (Fig. 3). Thermocouples are used for measuring the ambient and lubricant temperatures. Several heating covers have been installed on the external bottom face of the housing in order to perform measurements up to temperatures near 100°C. Some movable walls can be inserted in the gearbox to modify the clearances between a wall and a gear face or top land.

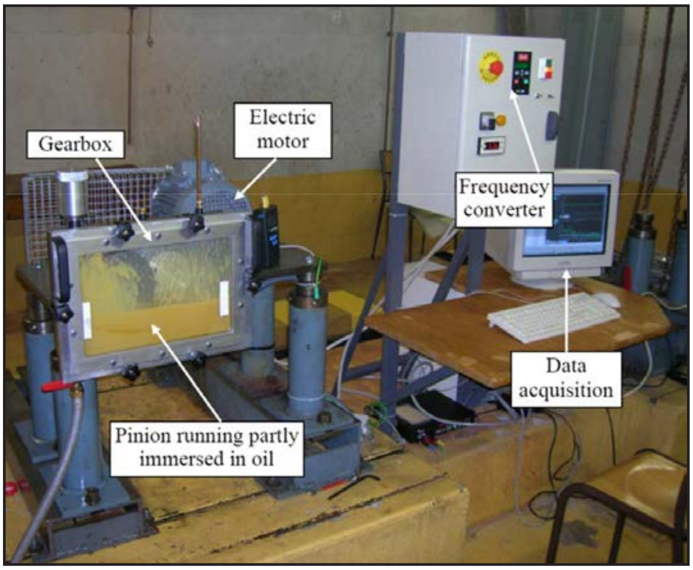

Figure 3 Test rig.

The main parameter generating a decrease in churning power losses is the axial distance between a flange and a gear face (*Ja* in Fig. 4). By mounting flanges close enough to the gear lateral faces, it is possible to divide by two the oil churning losses. In (Ref. 10) several measurements have been conducted and the influence of flanges has been quantified by establishing relationships based on dimensional analysis.

As tests performed in (Ref. 10) deal only with spur gears, some additional experiments have been conducted in order to extend previous results to helical gears. Figure 6 presents some typical results obtained for two different gears: a spur gear and a helical gear. These gears have the same module (3 mm), the same face width (24 mm) and a similar outside diameter (about 162 mm). For a peripheral speed of 58 m/s, churning losses can be substantially reduced and it is noted that the helical gear has the same behavior as the spur gear. Other measurements have been performed on gears with smaller module (1.5 mm). These tests confirm the above-mentioned evolutions.

Numerical results on the complete gear unit. Formulas developed in (Ref. 10) represent a good approximation to estimate the decrease in churning losses. They have been used in the thermal network model to estimate potential savings associated with flanges on the complete geartrain. The following parameters have been used to obtain numerical results: (i) presence of flanges with an axial clearance equal to 4 mm; (ii) flanges with a clearance of 1 mm.

By adding flanges, the power losses repartition is modified. Table 4 presents the total power loss and the relative importance of the dissipation sources for each test. It appears that churning losses can be decreased up to 50 percent with the smallest axial clearance, whereas total power loss can be reduced by 13 percent.

The evolution of power losses leads to a modification in temperature distribution, which is presented in Table 5. As the oil bulk temperature depends on the total power loss in the gear unit, it can be noticed that the presence of flanges generates a decrease of 10°C in the lubricant temperature. Because of high heat transfer coefficient of convection, the bulk temperature of the gear wheel is found to be very close to that of the lubricant. As far as the pinion is concerned, this element is not immersed in the oil sump and stabilizes at higher temperature.

Compared to oil churning and teeth friction, bearings-related losses become predominant; they vary from 657 W in the initial configuration to 750 W with flanges at 1 mm. This behavior can be explained by viscous forces in rolling-element bearings: the evolution of temperature distribution in gear units modifies the local oil viscosity at certain points in the transmission; it highlights the role of temperature-related power loss sources.

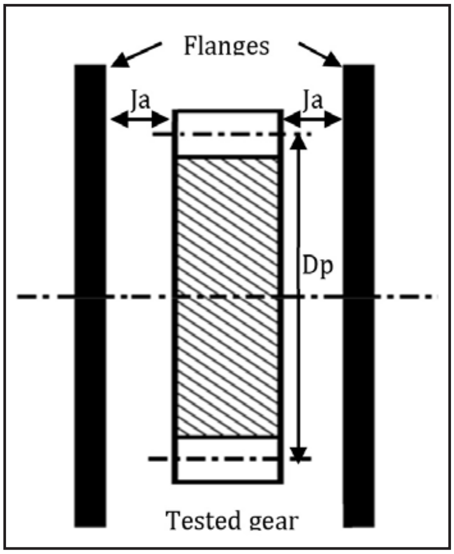

Figure 4 Flanges position relative to the gear.

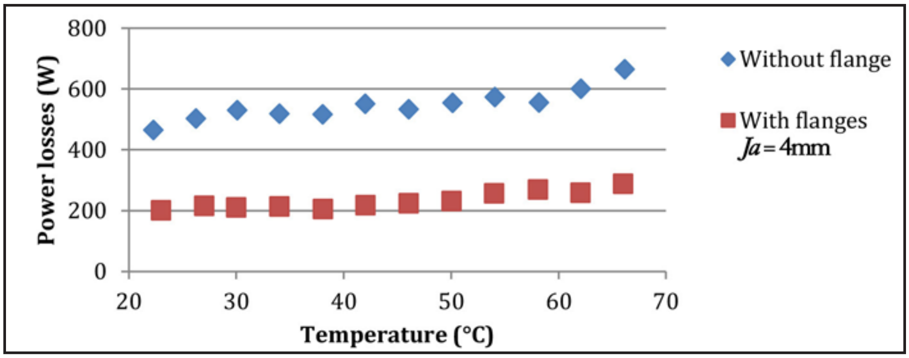

Figure 5 Power losses with and without flanges at 7,000 rpm for the spur gear.

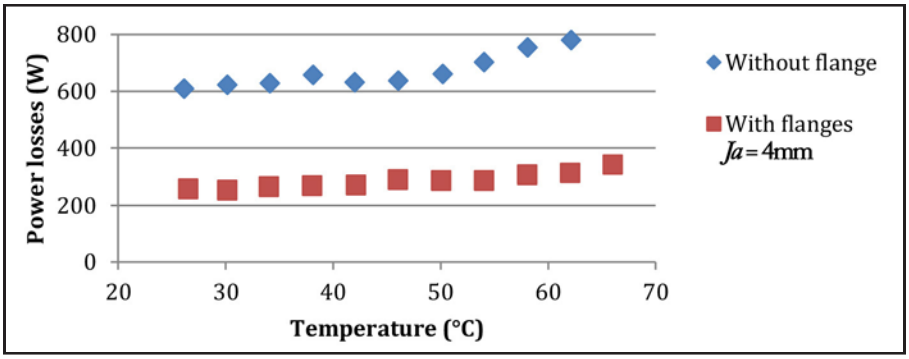

Figure 6 Power losses with and without flanges at 7,000 rpm for the helical gear.

Table 4 Power losses repartition (32,000 rpm–100 kW)

	Initial configuration	Use of flanges *Ja* = 4 mm	Use of flanges *Ja* = 1 mm
Total power loss (W)	1,512	1,389	1,315
Churning (%)	32	22	14
Bearings losses (%)	43	51	57
Teeth friction (%)	25	27	29

Table 5 Temperatures distribution (32,000 rpm–100 kW)

Calculated temperatures	Initial configuration	Use of flanges *Ja* = 4 mm	Use of flanges *Ja* = 4 mm
Oil (°C)	87	81	77
Pinion (°C)	94	89	86
Wheel (°C)	87	81	77

Conclusion

To investigate power losses generated by splash-lubricated gears at high speed, a thermo-mechanical model of a one-stage gear unit was created. Particular attention was given to churning power losses and to heat transfer between gears and the oil sump. The numerical model shows that for high rotational speeds, churning losses represent a major source of dissipation.

A simple solution to reduce churning losses — without decreasing convection heat transfer with the oil sump — is to insert some flanges in the gearbox in order to modify axial clearances between a wall and a gear face. This solution is presented and implemented in the thermo-mechanical model. The numerical results show that it seems possible to reduce by 13 percent the global power losses at nominal operating conditions. The presence of these flanges also generates a decrease of 10°C in the oil bulk temperature.

It can be noticed that rolling-element bearings also represent an important source of dissipation. For the moment these elements have been considered as isothermal in the numerical model. As an example — no temperature difference is calculated between the inner and outer rings. A more realistic modeling will be implemented in the near future by using the thermal network method (Ref. 11). **PTE**

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