Thermal Behavior of a High-Speed Gear Unit

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In this paper a thermal network model is developed to simulate the thermal behavior of a high-speed, one-stage gear unit which is jet-lubricated.

Introduction

In the general context of the reduction of energy consumption, there is an increased demand for more efficient gear units. It is possible to design geared transmissions which have a high efficiency: machinery 99 percent. But for high-speed turbo machineries (compressors, steam or gas turbines) the gearboxes may transmit power of several megawatts. In that case, one percent of dissipated power represents hundreds of kilowatts. This energy converted into heat is transferred through the elements and leads to a significant overall temperature rise of the mechanical components (Ref. 1), which can be harmful for the system integrity. Then reducing power losses has a dual objective: save some energy and reduce the overall heating of the gear set.

As for high-speed applications, oil jet lubrication appears as the most appropriate solution for lubrication and cooling gears. On the one hand, only a little amount of oil is sufficient for the formation of an oil film on tooth surfaces. On the other hand, a higher amount of oil flow rate is required for cooling gears. As has been demonstrated by numerous experiments, no-load power losses become prominent when considering high-speed gear transmissions (Ref. 2). Since these sources of dissipation increase with the lubricant flow rate (Ref. 3), the amount of lubricating fluid has to be carefully determined to ensure both sufficient heat transfer and high efficiency.

The aim of this study is to predict temperatures and power losses on industrial high-speed gear units, such as those used in power plants. A typical gearbox has been taken into account: an oil jet lubricated one-stage gear unit which is described in a first section. To simulate the heat transfer phenomena, the thermal network method has been used. The coupling between power losses and thermal calculations is explained in a second section. To validate the developed model, some comparisons between numerical and experimental results are given for different operating conditions. The influence of the oil flow rate on the thermal behavior of the gear unit is investigated in a last section.

Gear Unit Under Consideration

The system under consideration is a one-stage helical gear unit developed by Flender Graffenstaden. The whole set is enclosed in a housing made from cast iron. The gear data are given in Table 1.

The gear unit comprises two shafts which are supported by journal bearings. The unit is oil jet lubricated (kinematic viscosity of 32 Cst at 40° C and 5.4 Cst at 100° C/density of 870 kg/m^3 at 15° C). Different hydraulic circuits are used to lubricate the gears and the bearings. As far as the mating teeth are concerned, four injection nozzles are used along the tooth face width (Fig. 1).

All the tests performed during this study were conducted with no load applied. The rotational speed was imposed by an electric motor which also compensates for the losses in the gear unit. A torque sensor on the motor shaft was used to determine directly the mechanical power dissipated (accuracy of 0.1% of the measured value). Thermocouples were used for measuring temperatures at different locations: housing, inlet and outlet oil circuits, etc. Some temperature sensors were also placed on gears tooth in order to measure the bulk temperature of rotating parts. The data of these sensors were gathered by telemetry.

Table 1 Gear data

		Pinion	Wheel
Number of teeth	[-]	32	113
Module	[mm]	6 8	
Pitch diameter	[mm]	219.5	775
Tooth face width	[mm]	390	400
Pressure angle	[°]	20	
Helix angle	[°]	7.5	

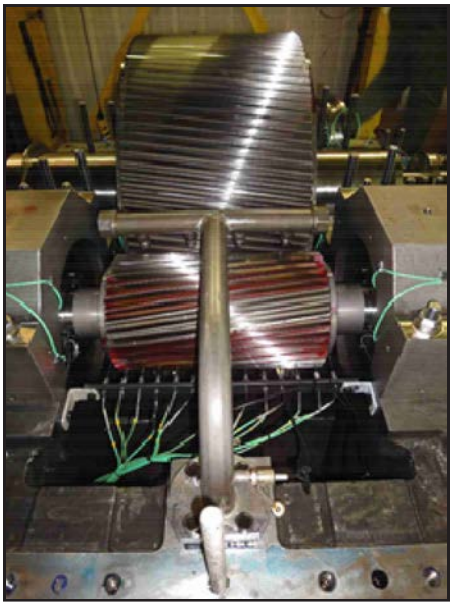

Figure 1 Lubrication of mating teeth.

Table 2 Elements of thermal network

Number	Element reference
1	Air
2	Gearbox housing
3	Injected oil
4	Mixture of air and lubricant
5	Oil trapped in the tooth interspaces
6, 7, 8, 9	Bearings
10	Primary shaft
11	Secondary shaft
12	Pinion
13	Wheel
14	Pinion's teeth
15	Wheel's teeth
16	Meshing of gear teeth

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Thermal Network

In order to simulate the thermal behavior of the tested gearbox, the thermal network method has been used. This method consists of dividing the geared unit into isothermal elements which are connected by thermal resistances. The one-stage jet lubricated gear unit under consideration has been divided into 16 elements, as detailed in Table 2.

The pinion and the gear wheel have been divided into two elements (nodes 12 and 14 for the pinion, nodes 13 and 15 for the gear wheel) in order to simulate the radial temperature gradient which may occur between the bulk temperature of a gear and the one of its teeth. Similarly, the lubricant has been separated in a number of nodes: the temperature of injection may be different to the one of the air/oil mist inside the casing, or the one of the oil that is trapped in the tooth interspaces. On the contrary, the gearbox housing has not been divided into several elements, since the objective was not to calculate the temperature distribution in the housing, but to determine a bulk temperature aimed at quantifying the heat exchanges with the air surrounding the gear unit.

The corresponding thermal network of the studied gear unit is described (Fig. 2) with the element labels as defined in Table 2. These elements are connected by thermal resistances depending on the kind of heat transfer, i.e. — conduction, free or forced convection and radiation. Following Changenet et al. (Ref. 4), the gearset is considered as an assembly of elements that have simple geometric shapes (as cylinders, flat plates, etc.) Thus, usual relationships of heat transfer can be used to quantify the associated thermal resistances. As an example — to evaluate the thermal resistance of convection and radiation between the housing and the surrounding air, Newton's and Stephan-Boltzmann's laws are respectively applied. In a similar manner the thermal resistance of conduction (for instance, between a gear and its shaft) is determined by applying Fourier's law. The oil jet flows used to lubricate the gearbox generate forced convection heat transfers: (a) along the casing walls, (b) with rotating parts and (c) by centrifugal fling-off on gear tooth faces. As a consequence, the thermal resistances of convection with oil (or air/oil mist) are determined according to the fluid flow considered. Finally, for a pinion and wheel in mesh under load, the gear tooth temperature consists of the addition of the bulk temperature and the flash temperature (Ref. 5); i.e., the Hertzian contact zone is very small in comparison with the characteristic dimensions of the gear tooth, and leads to a constriction of the thermal current from the surface to the body center. To account for this phenomenon, thermal resistances of striction are used in the thermal network (Ref. 4).

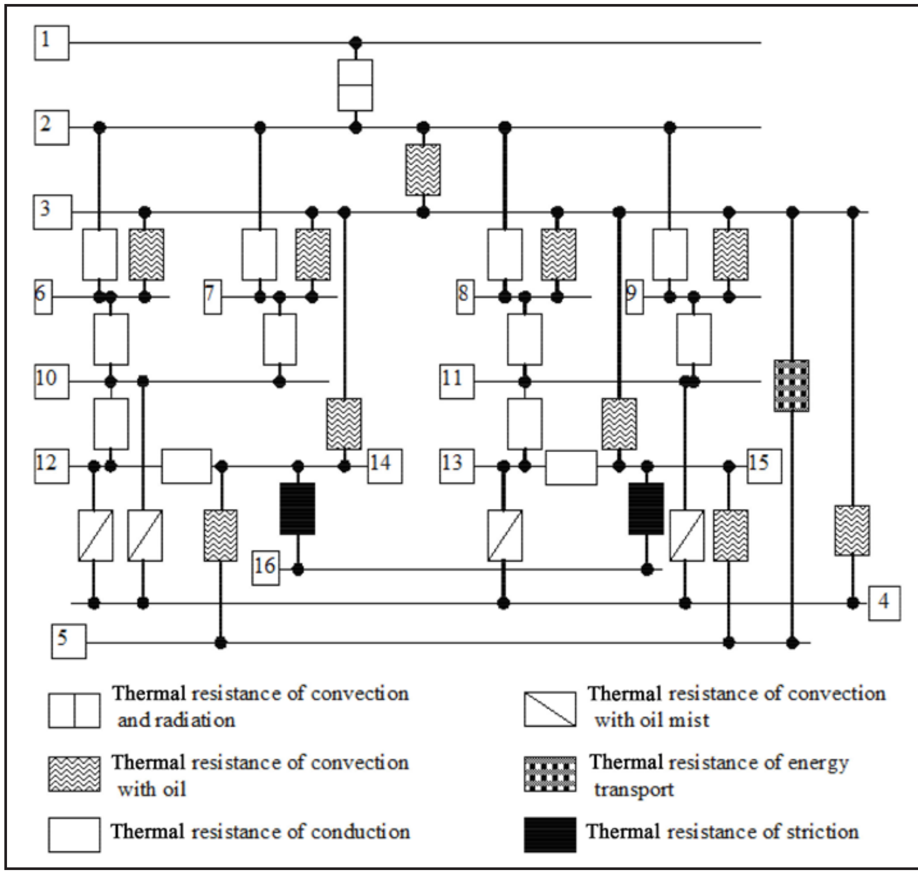

Figure 2 Thermal network of oil jet-lubricated, one-stage gear unit.

The heat generated by power losses in geared systems must be injected at certain nodes of the thermal network; five sources of dissipation have been taken into account in the gear set:

1. **Friction at the mating teeth of loaded pinion and wheel.** Tooth friction losses are calculated with relationships of Velex and Ville (Ref. 6). They are injected at node #16.
2. **Bearing power losses.** The resisting torque generated by a journal bearing can be estimated by using equations developed by Pierre and Fillon (Ref. 7). These power losses are injected into nodes #6 through #9.
3. **Windage effects.** The windage power loss generated by gears rotating in air/oil mist is evaluated according to Diab's formulas (Ref. 8). This source of dissipation is injected at node #4.
4. **Oil acceleration.** The oil jet flow that is used to lubricate gears is accelerated in the circumferential direction (Ref. 9). This phenomenon requires energy which is taken into account at node #3.
5. **Oil trapping.** The lubricant trapped in the tooth interspaces is successively compressed and expanded in these spaces, giving rise to power loss. Such power loss is quantified according to Butsch's relationships (Ref. 10) and injected into node #5.

The temperatures and heat flux distributions can be estimated by solving the equations generated by the thermal network (Ref. 4). The numerical solving procedure can be synthesized as follows: the different temperatures are initialized, power losses and thermal resistances are calculated, the bulk temperatures are determined and an iterative loop is made in order to revalue these parameters. Convergence is reached when the relative difference between two iterations is less than 0.1% for power loss and temperature.

Comparisons between Numerical and Experimental Results

The results presented in this section are written in a dimensionless form. For each node *i* of the network, the temperatures are expressed as:

$$\theta_i = (T_i - T_{air})/(T_{oil} - T_{air}) \quad (1)$$

Where T_{oil} and T_{air} are, respectively, the temperatures of the oil jet at the inlet and of the ambient air. As far as power losses are concerned, their dimensionless values are defined as follows:

$$P_i = Q_i/(M.c.\Delta T_{max}) \quad (2)$$

Where Q_i represents heat generated by node *i*, *M* the lubricant mass flow rate, *c* the oil-specific heat and ΔT_{max} the maximum temperature difference measured between the outlet and the inlet of the oil jet flow.

As mentioned, the tests presented in this section were performed with no applied load. Figure 3 presents power losses for different rotational speeds. In this figure the solid line (Num) represents the numerical results, whereas the marks (Exp.) account for the experimental findings. The results show that the prediction of power losses is satisfactory.

In Figure 4 the dimensionless temperature is plotted against the rotational speed for two different elements — a) the outlet oil circuit and b) the tooth of the gear wheel. This figure shows good agreement between the experimental and numerical results. One can notice that the bulk temperature of the tooth wheel is much higher than the one of the oil jet flow. This kind of result is logical for tests under load but seems surprising when no load is applied. The thermal network model can be used to analyze this result. For the studied operating conditions, the power losses due to oil trapping are very important; at 4,000 rpm this source of dissipation represents almost 33 percent of the total power lost. As a consequence, the oil trapping generates local temperature rises both for the gear wheel tooth and the pinion one.

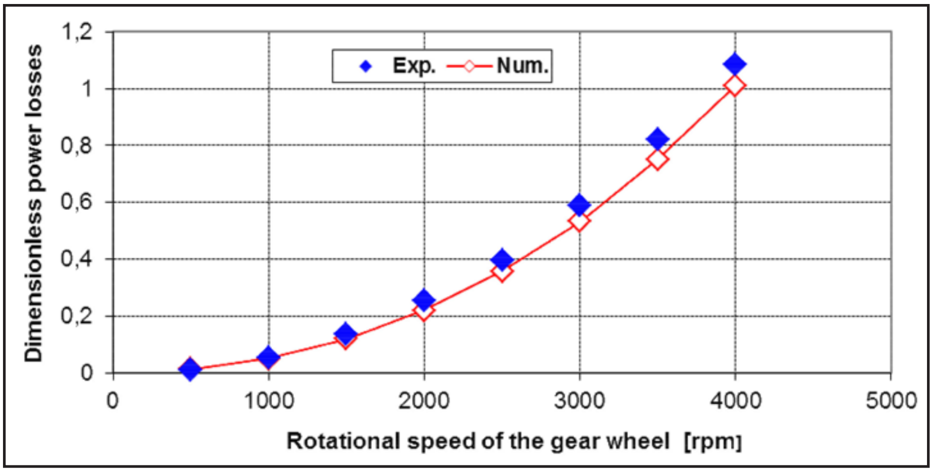

Figure 3 Dimensionless power losses vs. rotational speed.

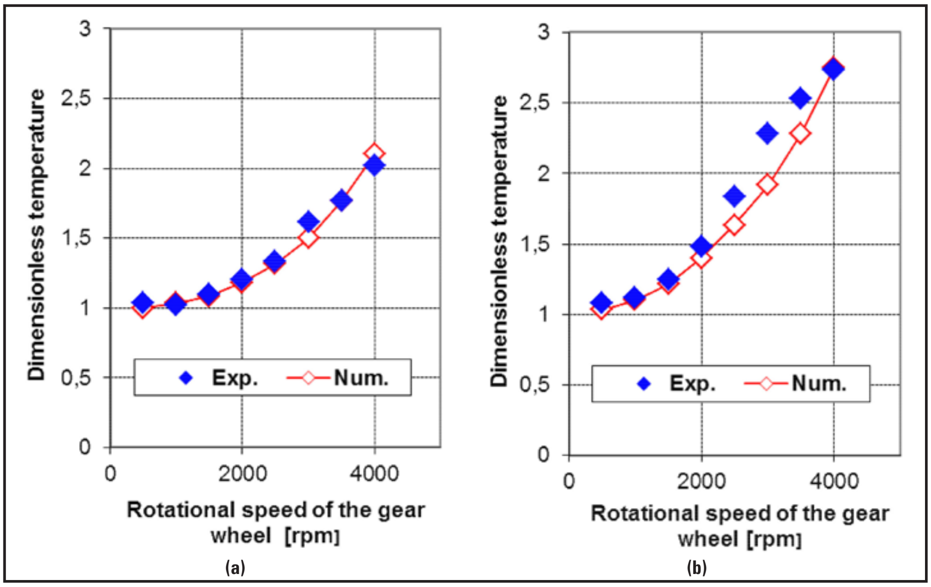

Figure 4 Comparison of measured and calculated temperatures for (a) oil jet at the outlet, and (b) tooth of the gear wheel.

Influence of Oil Flow Rate

The previous section has demonstrated that the thermal network model gives accurate results. So it can be used to simulate tests under load and to study the oil flow rate influence on the thermal behavior of the enclosed gear drive. For an input power equal to 30 MW and a rotational speed of the gear wheel equal to 4,000 rpm, the power loss distribution estimated through the thermal network is given in Figure 5. From these results it appears that power losses generated in the gear unit can be divided into four main sources of dissipation — tooth friction; bearings losses; windage effects; and oil trapping — which are almost equally distributed.

For high-power conditions an important part of the lubricant has to be used for cooling the gear teeth. Among the sources of dissipation highlighted in Figure 5, oil trapping and the torque required to accelerate the oil jet flow in the circumferential direction increase directly with the lubricant mass flow rate. On one hand, if too much oil entered the gear mesh, excessive losses may occur because of oil being trapped in the gear teeth and pumped out of the mesh. On the other, if not enough oil is injected the gear cooling can be insufficient.

The thermal network of the gear unit is then used to study the evolution of bulk temperatures versus the lubricant flow rate. As an example, Figure 6 presents the calculated temperature of the gear wheel. One can notice that above 300 l/min the gear wheel bulk temperature decreases slowly with oil flow rate. For this operating condition, the heat-transfer coefficient of convection between oil and gear wheel is high. Thus the associated thermal resistance is small and a change of its value no longer determines the thermal equilibrium of this element. It can be concluded from this evolution that there is no interest to significantly increase the

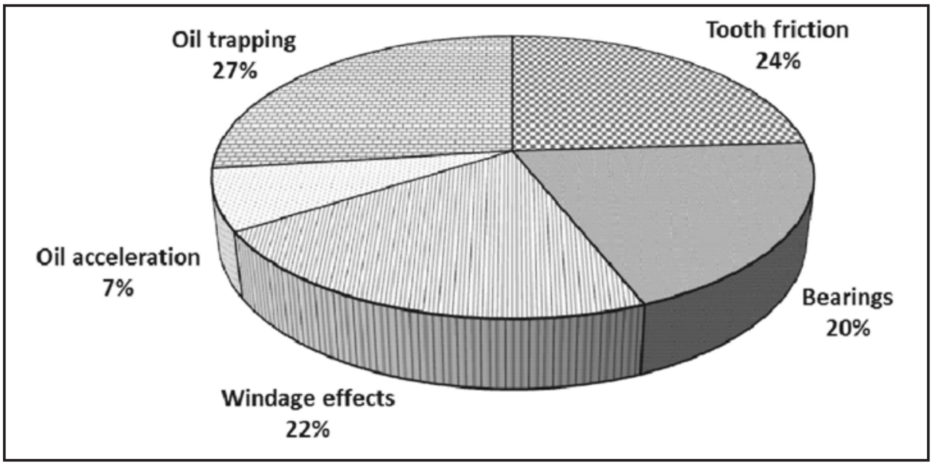

Figure 5 **Power loss distribution.**

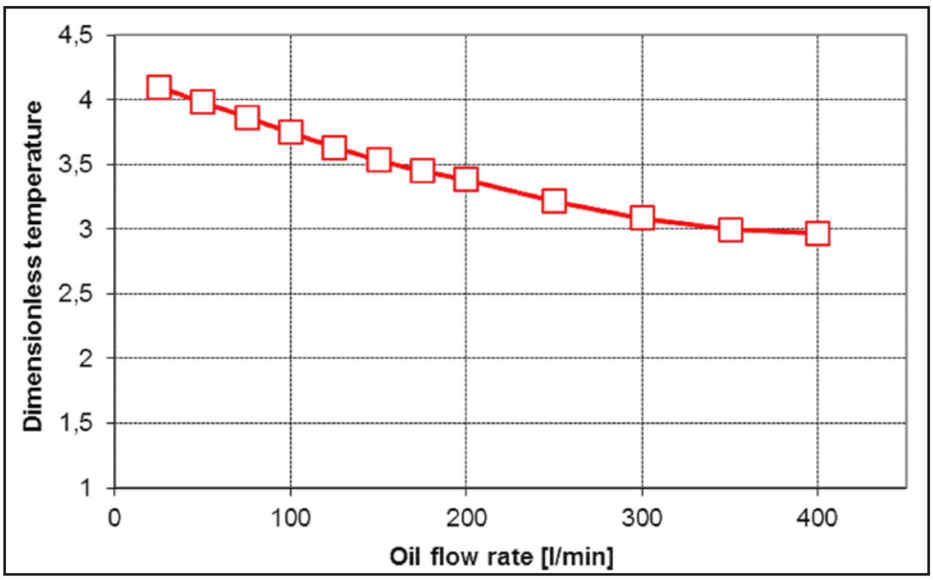

Figure 6 **Evolution of gear wheel bulk temperature vs. oil flow rate.**

lubricant mass flow rate. Moreover, this limitation will also induce a more efficient gear unit.

Conclusion

A model of a high-speed gear unit is presented that combines bulk temperature predictions and power loss calculations using a thermal network. A series of measurements was carried out on a specific test rig in order to validate this numerical model. Based on comparisons between experimental and numerical results, it is shown that the proposed model can accurately predict power losses generated in the gear unit and its steady state thermal behavior. As the model gives access to a detailed thermal mapping in the gearbox, it is used to study the evolution of rotating parts bulk temperature versus lubricant mass flow rate. It appears that above a given value, increasing the oil flow does not induce a significant decrease in gear bulk temperatures. Then an upper limit of the lubricant mass flow rate can be defined and used to design more efficient enclosed gear drives.

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