Thermal Modelling of Mixed non-Newtonian thermo-elastohydrodynamics in Dry Sump Lubrication Systems

Ehsan Fatourehchi[1)], Mahdi Mohammadpour[1)*], Stephanos Theodossiades[1)], Homer Rahnejat[1)]

[1)] Wolfson School of Mechanical, Electrical and Manufacturing Engineering, Loughborough University, Loughborough, LE11 3TU, UK
[*]Corresponding author: m.mohammad-pour@lboro.ac.uk

1. Abstract
Improved fuel efficiency is the primary objective in the optimization of modern drivetrain systems. Recently, the dry sump lubrication system is regarded as the lubrication system for high performance transmission systems. Dry sump lubrication enhances the system efficiency by reducing the churning losses, whilst providing sufficient lubrication for the tribological contacts.
One of the most important aspects of any dry sump system is assessment of the thermal performance. The generated heat in the contacts should be dissipated through impinging jets and air-oil mist in the transmission casing in an efficient manner. The present work incorporates a tribological model and a 3D CFD model into a finite element model. The aim is to evaluate the quantity of generated heat in the lubricated gear pair contacts, as well as heat removal rate due to an impinging oil jet. Furthermore, the transient circumferential temperature distribution on gear surfaces is determined. This provides an accurate input temperature for the entrant lubricant in the gear teeth-pair contacts. Such an approach has not hitherto been reported in literature.
To perform time-efficient system level analysis in the finite element model, extrapolated equations are obtained from a transient 3D CFD model using regression formulae.
Keywords: Thermal Analysis, generated heat in Transmissions, Dry Sump Lubrication.

2. Introduction
Efficiency and durability are key objectives for modern transmissions. Power losses are categorized as load-dependent and load-independent losses. The load-dependent losses are subject to the generated friction in gear teeth contacts and the supporting bearings. Churning and windage losses are the main sources of the load-independent power losses. The drag of lubricant, adherent on the rotating surfaces is the main underlying cause of these losses. Therefore, the type of lubrication system has the main effect on this category of losses. Most transmission systems in automotive applications are cooled and lubricated under a dipped (splash) lubrication system. In such a system, rotating gears are partially immersed into an oil sump. Churning losses are the main source of load-independent losses in splash lubrication systems. Seetharaman and Kahraman [1] developed a numerical model to determine the churning losses of gear pairs, validated against experimental measurements [2]. The effects of both spin power losses due to the interaction of individual gears with a bath of lubricant, as well as the power loss due to pumping of the lubricant into the gear mesh were considered.
To minimize/eliminate the churning losses, dry sump (no oil bath) lubrication system is usually introduced.
In these cases, the lubricant feed into the gear meshing conjunctions is provided through directed impinging oil jets, which also contribute to cooling. This is to dissipate the generated contact heat. In dry sump gearboxes, it is essential to ensure that an adequate amount of lubricant penetrates into the meshing teeth pair contact conjunctions for lubrication and cooling purposes. Gear bulk temperature is a controlling factor in preventing surface scoring, scuffing and other failure modes [3]. Therefore, proper cooling of gear contact prevents excessive surface temperature rise, which would lead to failure of the gear teeth.
DeWinter and Blok [4] introduced an analytical model to determine the heat removal rate through an impinging jet on the surfaces of the gears. The model assumed a laminar-adhered film layer formed on the gear flanks. This assumption over-estimated the film layer thickness, and as a result also the heat dissipation rate. Long et al. [5] carried out an analytical-numerical investigation of temperature distribution over a range of applied loads on a gear tooth temperature. They assumed temperature equilibrium for all parts of any gear. The same assumption was also used in Wang and Cheng’s study [6].
In this study, an integrated multi-disciplinary method is proposed to determine the transient temperature distribution of all surfaces of a rotating gear, operating under dry-sump lubrication system. The heat generation is obtained through a tribological contact model of meshing gear teeth. Heat dissipation is estimated assuming convection from the gear surfaces as well as conduction through solid gear bodies. The transient flank temperature provides an estimate for the inlet temperature of the lubricant into the contact conjunction. Therefore, an integrated approach in heat generation and removal is achieved.

3. Methodology
The developed method combines three models; a tribological contact model for meshing teeth pairs (including heat generation), a thermo-fluid dynamics (CFD) model of heat removal through impinging jets and air-mist environment and a thermal finite element model (to evaluate temperature of solid bodies).

3.1. Tribological model
Heat is generated due to friction in lubricated contacts

of meshing teeth pairs. The proposed method enables prediction of frictional losses. The method integrates TCA (Tooth Contact Analysis) with an analytical elastohydrodynamic lubrication (EHL) model to predict the load-dependent power losses [7]. The TCA model is based on the approach of Vijayakar [8], which determines the instantaneous contact geometry kinematics of the mating surfaces, as well as contact and load distribution between simultaneous meshing teeth pairs.

Chittenden et al [9] provided an extrapolated lubricant film thickness formula, taking into account the variation of lubricant film thickness under the instantaneous operating conditions. The predicted film thickness is then used to determine viscous friction in the EHL contacts, based on the approach of Evans and Johnson [10].

In addition to viscous friction of a thin lubricant film, direct boundary interactions of rough teeth surfaces promote boundary friction. This is because the height of surface asperities is quite comparable with the thinness of the lubricant film. The boundary friction within the contact is determined, based upon the approach highlighted by Greenwood and Tripp [11].

Finally, the instantaneous power loss for a meshing cycle is determined by taking into account the calculated viscous and boundary friction contributions as:

$$P_{loss} = (f_v + f_b)U_s \tag{1}$$

where, U_s, is the instantaneous contact sliding velocity, which is determined through TCA. f_v and f_b are viscous and boundary friction contributions, respectively.

It is assumed that the total generated power loss in the contact would convert to heat which is conducted through the contacting teeth surfaces, increasing their surface temperatures. The increased surface temperature, calculated using the approach highlighted by Crook [12], is the heat source in the computational fluid dynamics model.

3.2. CFD model

To model an impinging turbulent lubricant jet, a standard k-ε computational fluid dynamics (CFD) model is used. The governing equations for conservation of mass and momenta for each phase in a 3D Newtonian and incompressible turbulent flow are based on the approach outlined by Hou and Zou [13]. The interface between the immiscible liquid lubricant and the liberated vapour phase is monitored using the Volume of Fluid (VOF) method [14]. Figure 1 is a schematic of the developed model, including the boundary conditions for each face of a gear body in front and side view elevations. The oil jet impinges upon the contact exit of a pair of meshing teeth for improved heat dissipation from a single gear pair only (in this analysis). Hence, the effect of the impinging oil jet can be examined on a single gear body, assuming to have received half the generated contact heat.

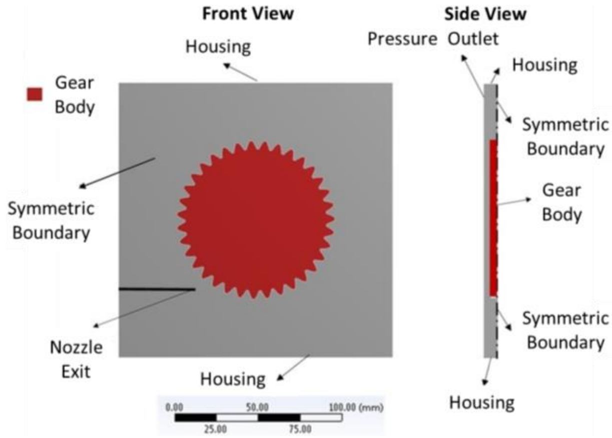

Figure 1 Schematic of the developed CFD model

To minimize the impact of the computational boundary upon the results, it is assumed that the gear rotates in a large box (Figure 1). The gear flank temperature is obtained through the tibological model and it is assumed to be constant. The nozzle exit is defined with lubricant mass flow rate of 0.0168 kg/s flow rate at 100°C. The outlet boundary condition is specified to be at atmospheric pressure.

3.3. Finite element model

The heat transfer mechanism for all tooth segments is assumed to be the same. Consequently, the temperature distribution on each tooth segment is identical. Therefore, a finite element model for one tooth segment is developed. A typical configuration for a single tooth segment with defined boundaries is shown in Figure 2.

3.3.1 Thermal analysis and boundary conditions

For a pair of gear blanks rotating at high speeds under dry sump lubrication system, the main cooling is usually provided by a mixture of oil mist surrounding the gear surface [6]. Therefore, the conducted heat through the supporting bearings is neglected.

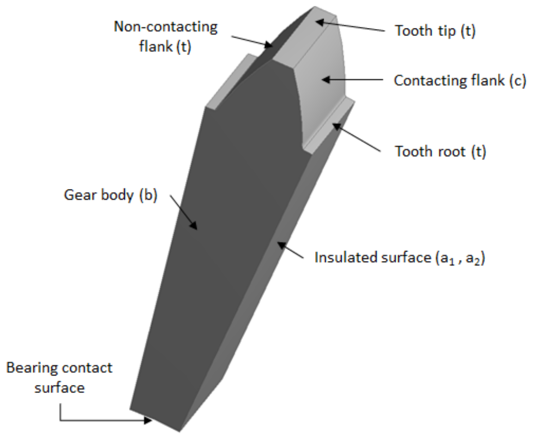

Figure 2 A tooth segment with definition of FE model boundary conditions

In each gear revolution, an identical heat flux is applied to a tooth flank segment. The governing equation to obtain the temperature distribution uses the Fourier heat conduction equation as:

$$k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) = \rho c \frac{\partial T}{\partial t} \tag{2}$$

where temperature, T = T (x, y, z, t), changes with time t and position x, y, z. The associated boundaries of different surfaces of tooth segment (Figure 2) are as follows:

$$-\left.\frac{\partial T}{\partial n}\right|_{c} = h_t(T - T_{amb}) + P_{loss}$$
$$-\left.\frac{\partial T}{\partial n}\right|_{t} = h_t(T - T_{amb})$$
$$-\left.\frac{\partial T}{\partial n}\right|_{b} = h_b(T - T_{amb}) \tag{3}$$

where, h_t and h_b are the heat transfer coefficients for the gear surface and body, respectively. P_{loss} is the generated contact heat, n is the length of coordinate in the direction of the outward normal to the surface.

The insulated heat condition on the gear model is:

$$\left.\frac{\partial T}{\partial n}\right|_{b_1} = \left.\frac{\partial T}{\partial n}\right|_{b_2} = 0 \tag{4}$$

3.4. Solution procedure

The proposed method integrates a tribological model of heat generation and a CFD simulation of an impinging oil jet on a rotating gear with a transient finite element model of a tooth segment in order to determine the transient temperature distribution on the gear surfaces.

The solution procedure comprises:

Step 1: The generated heat in the meshing contacts and surface temperature rise in a meshing cycle, obtained through solution of mixed-EHL model.

Step 2: The temperature rise is an input to the developed 3D CFD model in order to calculate the heat transfer coefficients on gear surfaces and body.

Step 3: The generated heat and the heat transfer coefficients on gear surface and body are input to the transient finite element model to obtain the transient temperature distribution.

Step 4: At the end of each gear revolution, the finite element model is interfaced with a Matlab program to update the finite element model's initial nodal temperatures for the next gear revolution.

Step 5: The process continues until the amount of dissipated heat through convection equals to the amount of generated heat. Under certain harsh drive cycles, this equilibrium may not be achieved.

4. Results and discussion

For the purpose of analysis, the dry sump system of a high-performance racing transmission is considered. Table 1 provides the lubricant and gear properties.

Table 1 Lubricant and gear geometrical properties

Pressure viscosity coefficient (Pa^{-1})	2.383×10^{-8}
Lubricant dynamic viscosity at 100°C (Pa.s)	0.0171
Number of teeth	35
Face width (mm)	13.5

A typical racing condition with gear speed of 11000 RPM is chosen. Figure 3 shows the predicted frictional power loss in a meshing cycle for the chosen conditions.

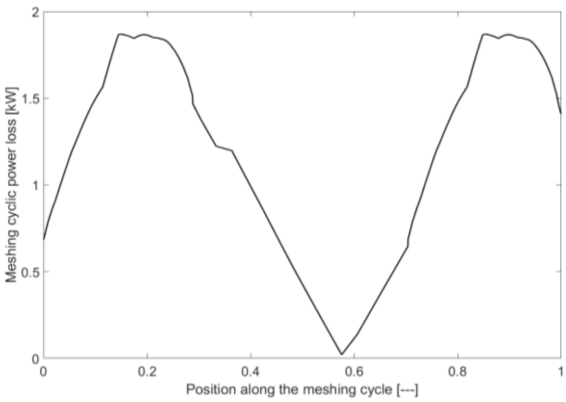

Figure 3 Meshing cyclic power loss

In this case, the average power loss for a meshing cycle is 1000W. According to Crook [12], the generated heat can be assumed to be conducted equally to the two mating gear teeth surfaces (i.e. 500W per tooth flank in this case). Considering the presence of a thin elastohydrodynamic lubricant film, any heat dissipation due to convection is neglected [15]. Having obtained the generated heat on the meshing teeth flanks, the average heat transfer coefficient on gear teeth and body (Figure 2) is obtained using the extrapolated equation from the 3D CFD model. These form the input to the finite element thermal model.

For a driving cycle which takes 84 s, the analysed gear is loaded for 5.5 s in the considered transmission system (i.e. equivalent to 1000 rotations). This means for the time period which the gear is not loaded, convection heat transfer from the gear surfaces is the only mechanism of heat transfer.

Figure 4 presents the average temperature on the contacting teeth flanks for a complete driving cycle.

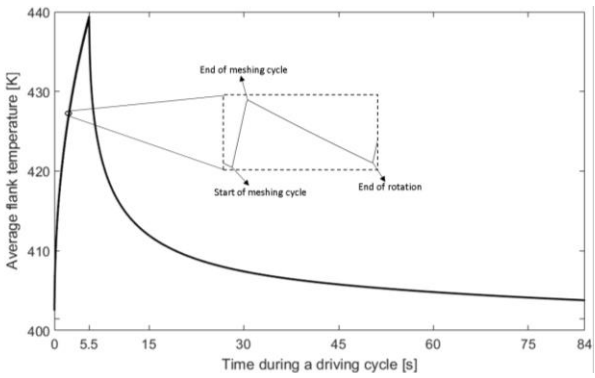

Figure 4 Average tooth flank temperature

The figure shows that at the beginning of the drive cycle where the gear is engaged the flank temperature rises sharply. After 5.5 s, where the analysed gear in the transmission is disengaged, convection to the ambient remains the only heat transfer mechanism. The temperature difference between the ambient (403 K, in the transmission housing) and the gear flank is the highest just after 5.5 s at the beginning of the drive cycle. Therefore, the rate of convection heat transfer is higher in this part. From 30 seconds onwards, the temperature drops only slightly because the temperature difference with the ambient and consequently the rate of convection is only marginal.

Figure 5 shows the temperature contours at the end of the gear engaged period and 11 s thereafter.

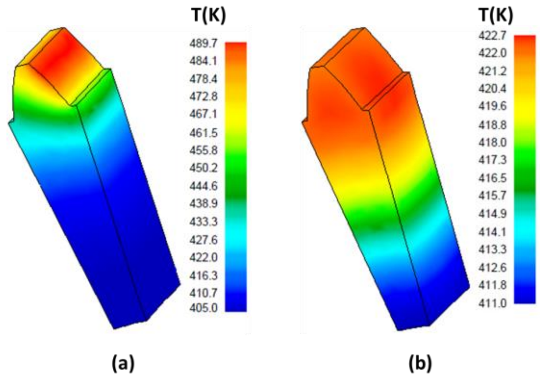

Figure 5 Temperature distribution after (a) 1000 rotations (5.5 s) and (b) 3000 rotations (16.5 s)

During gear engagement, the conducted heat almost penetrated into the gear topside. During gear disengagement, the temperature contours are more uniform, since convection cooling occurs from all gear sides.

Finally, Figure 6 shows the heat transfer through the gear during a complete drive cycle.

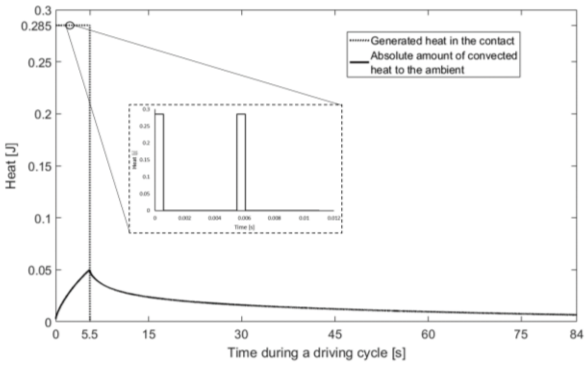

Figure 6 Heat transfer through and from the gear

As a result of higher temperature rise at the beginning of simulation, the rate of convection heat transfer to the ambient is also higher. This is shown in Figure 6. The higher the temperature rise, the higher the temperature difference with the ambient environment of the transmission casing and therefore the greater is the amount of convection heat transfer.

5. Conclusion

An integrated numerical method, comprising a tribological heat generation model, a CFD heat dissipation model, and a thermal finite element model of solid boundaries is presented. Transient heat and temperature distribution through a gear tooth segment lubricated by an impinging oil jet of high-performance racing application is investigated. The results indicate that the generated heat in the contact can be dissipated by convection heat transfer and a single oil jet. However, the jet must be sprayed at all times, including during the disengaged gear periods.

6. References

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