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The Efficiency of a Simple Spur Gearbox - A Thermally Coupled Lubrication Model

By A.I. Christodoulis, A.V. Olver,
and A. Kadiric, Imperial College
London, A.E. Sworski,
A. Kolekar, and F.E. Lockwood,
Valvoline/Ashland

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

A thermally coupled efficiency model for a simple dip-lubricated gearbox is presented. The model includes elastohydrodynamic (EHL) friction losses in gear teeth contacts as well as bearing, seal and churning losses. An iterative numerical scheme is used to fully account for the effects of contact temperature, pressure and shear rates on EHL friction. The model is used to predict gearbox efficiency with selected transmission oils whose properties were first obtained experimentally through rolling-sliding tribometer tests under representative contact conditions.

Although the gearbox was designed using standard methods against a fixed rating, the model was used to study efficiency over a much wider range of conditions. Results are presented to illustrate the relative contribution of different sources of energy loss and the effect of lubricant properties on the overall gearbox efficiency under varying operating conditions.

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Introduction

The efficiency of drivetrain components is quickly becoming a significant research area pushed by the ever intensifying quest for improved fuel economy in automotive vehicles. Emission-controlling regulations are becoming stricter, owing mainly to environmental issues such as air contamination but also due to the depletion of oil deposits and the resulting high fuel prices. Pursuing augmented efficiency of a drivetrain or its components can also have the positive side effect of decreasing the frictional heat that is generated inside the gearbox, differential or axle component therefore improving scuffing or pitting behavior.

The drive train in passenger cars absorbs around six per cent of the total fuel energy in combined city-highway driving - equivalent to about thirty per cent of the mechanical energy delivered to the wheels [1]. The biggest part of this energy loss ends up as heat in the axle or transmission lubricant resulting from friction, windage and churning. As a result the reduction of lubricant-related losses in a vehicle's drivetrain can lead to significant improvements in both the fuel economy and the environmental. Thus, this efficiency increase is a much sought-after goal for lubricant suppliers of OEM factory fill and aftermarket lubricants.

Background

The efficiency of spur gears is started receiving significant attention since about 1980's with the work of Anderson [2] [3]. Recent years have seen a boom in such publications owing mainly to the energy crisis. Li and Kahraman [4] and more recently Chang and Jeng [5] focused on a spur gear pair while Michaelis [6] considered a more integrated approach which included churning losses as well as bearing and seal losses. Churning losses play a very important role in the prediction of a dip lubricated component's efficiency and recent studies from Changenet and Velez [7] [8] have shown that the accurate determination of churning losses and how these are affected by design parameters is a challenging problem.

Thermal behavior of the components and thermal response of lubricants are dominant factors of efficiency enhancement. There are several published models of spur gear pairs that analyze the thermal behavior of the pair like those developed by Long and Lord [9] and Taburdagitan [10] which use finite element methods to predict the overall and surface temperatures. A more integrated approach by Changenet and Velez [11] considered lump thermal elements to study and model a six-speed gearbox, but with no experimental validation. In addition, the thermal response of transmission lubricants was extensively studied by Olver [12] who also developed a comprehensive model to predict traction in Elastohydrodynamic (EHL) contacts which included thermal effects [13]. However, there are currently no efficiency models that consider full spur gearbox including the all-important thermal coupling, taking into account the mesh and bulk temperature rise of all drivetrain components or that of the oil surrounding them as well as all sources of energy loss including bearings, seals, churning and EHL traction.

Additionally, the type of lubricant itself is a crucial factor in determining the efficiency of a drivetrain. Despite this, there is currently very limited ability to predict the relative fuel economy arising from the use of different drive train lubricants. Petry-Johnson [14] have included more than one type of lubricants in their studies in an attempt to pinpoint the possible effect that a specific combination of lubricant, component design and operating condition may have on the overall gearbox efficiency. The results from their studies showed a linear relationship between the gear power loss and the rotational speed of the gears and also highlighted the effect of surface finish on the efficiency of the gearbox. Their lubricant comparison used three different lubricants to indicate that there is a possible change in overall efficiency depending on the lubricant type. However, no complete method has been developed that is able to simultaneously account for specific lubricant characteristics under EHL contact conditions and relevant

gearbox parameters. Such an approach should be able to provide a more accurate estimation of the possible efficiency gain.

The limitations of current approaches include limited treatment of gear churning losses and not accounting for the transient conditions arising from variable vehicle duty cycle. Kolekar and Olver [15] have recently worked on this issue concentrating on hypoid axles. A transient thermal model coupled to a quasi-steady state lubricant traction and churning formulation has been used together with lubricant bench tests in order to predict the energy that is dissipated during specified drive cycles. The results highlight the high influence that the properties of axle oils have and that the ranking order of lubricant composition and properties depends greatly on the specified duty cycle, with high viscosity, friction modified oils being favored for high power use. In contrast, lower viscosity fluids than are currently in use provide lower losses for city and light highway duty.

In addition to treating only the axle, this work has other significant shortcomings including the lumped mass axle temperature that does not capture the bulk lubricant temperature, the lack of a change gearbox model as well as the lack of provision to determine the effect of drive train efficiency on the engine behavior and therefore on fuel consumption.

The present work is designed to offer significant improvements in the accuracy of efficiency predictions for simple spur gearboxes including the effects of lubricant rheology on gearbox efficiency. The model accounts for EHL friction losses in gear teeth contacts, bearing and seal losses, and losses due to oil churning. The EHL friction losses are calculated based on lubricant properties that have been obtained through extensive measurements on a ball-on-disk tribometer, while also fully accounting for the effect of thermal coupling so that any increase in lubricant temperature (and the corresponding change in lubricant properties) due to power losses, is fed back to EHL friction calculations. The ambient temperature of the lubricant in the gearbox is calculated using a multi-physics finite element model which includes all conductive and convective heat transfers within the gearbox. Such a holistic approach, particularly the inclusion of thermal coupling, will enable the model to account for the transient conditions due to particular usage history and therefore predict the efficiency for a given drive-cycle.

Methods

This section first outlines the basics of the EHL model used to predict tooth frictional losses, followed by the methods to predict bearing, seal and churning losses. Finally the FEM approach used to calculate the ambient (oil bath) temperature rise, which is fed back to the EHL model, is outlined.

The experimental method used to extract lubricant characteristics that are used as input to the EHL model is also described.

Basic EHL model

A thermally coupled EHL model was devised adopting the approach of Olver and Spikes [13] which predicts the friction coefficient in the rolling-sliding EHL contact of a lubricated disc pair.

The model uses the EHL regression equations developed by Chittenden [16] to calculate the minimum and central film thickness:

$$\frac{h_0}{R'_x} = 3.63 \bar{U}^{0.68} \bar{G}^{0.49} \bar{W}^{-0.073} \quad (1)$$

$$\frac{h_c}{R'_x} = 4.30 \bar{U}^{0.68} \bar{G}^{0.49} \bar{W}^{-0.073} \quad (2)$$

where

- h_0 is minimum EHL film thickness, m;
- R'_x is reduced radius of contact, m;
- $\bar{U}$ is entrainment velocity, m/s;
- $\bar{G}$ is gravitational acceleration, m/s²;
- $\bar{W}$ is total contact load, N;
- h_c is central EHL film thickness, m.

Where the non-dimensional parameters

$$\bar{U} = \frac{U\eta_0}{E'R'_x} \quad (3)$$

$$\bar{G} = \alpha E' \quad (4)$$

$$\bar{W} = \bar{W}_L = \frac{W}{E'R'_x L} \quad (5)$$

are the speed, load (L = line contact) and material parameter respectively.

where

- η_0 is inlet viscosity (at $p = 0$), Pa.s;
- E' is reduced modulus of elasticity, Pa;
- α is pressure-viscosity coefficient;
- $\bar{W}_L$ is total contact load in line of contact;
- L is length of EHL contact, m.

The mean shear stress is calculated based on the Ree-Eyring approach as adapted by Evans and Johnson [17], and the traction regime is decided using the following three-stage process described by Olver and Spikes which is based on the non-dimensional Deborah and S numbers (Figure 1).

The friction coefficient is then predicted by means of a convergence loop; the loop is initiated by assuming an initial friction coefficient μ_0 and then the temperature rise due to shear heating of the lubricant $(\Delta T_{oil})_{av}$ and contact of asperities $(\Delta T_f)_{av}$ are calculated using the approach developed by Olver [13].

$$(\Delta T_f)_{av} = \frac{1.06 \alpha_h \dot{q}}{Ak_1} \left(\frac{\chi_1 \alpha_0}{U_1} \right)^{1/2} = 1.06 B_1 \alpha_h \dot{q} \quad (6)$$

Question	Criterion	Mean shear stress formula
Stage 1		
Is the response viscoelastic?	i.e. $D_0 > 1$	
Is there shear thinning?	i.e. $S > \sinh 1$	
Stage 2		
If the response is <i>not</i> viscoelastic <i>and</i> there is <i>no</i> shear thinning, then		$\bar{\tau}^* = S$
If there is shear thinning <i>but</i> the response is <i>not</i> viscoelastic, then		$\bar{\tau}^* = \sinh^{-1} S$
If the response is viscoelastic <i>but</i> there is <i>no</i> shear thinning, then		$\bar{\tau}^* = S[1 - D_0(1 - e^{-1/D_0})]$
If the response is viscoelastic <i>and</i> there is shear thinning with $\tau^* > 2$, then		$\tau^* \approx \ln(2S) - \ln(1 + 2 S - \frac{1}{2} e^{-S^*/(2D_0)})$ (see text)
Stage 3		
Is the plastic limit reached?	i.e. $\bar{\tau}^* > \tau_c^*$	
If so, then discard previous value and put		$\bar{\tau}^* = \tau_c^*$

Figure 1. The rules for determination of the mean shear stress [13]

(D_0 is Deborah number ($= \eta_0 U / (2\alpha G e)$), S is non-dimensional strain rate ($= \dot{\gamma} \eta_0 / \tau_E$), τ^* is non-dimensional shear stress ($= \tau / \tau_E$) and τ_c is the limiting shear stress)

$$(\Delta T_{oil})_{av} = \frac{\mu W U_s h_c}{8 A k_{oil}} = \frac{U_s \bar{\tau} h_c}{8 k_{oil}} \quad (7)$$

where

- α_h is heat partition;
- $\dot{q}$ is heat generated by sliding in the contact, W;
- A is EHL contact area, m^2 ;
- k is thermal conductivity, W/m-K;
- χ_1 is thermal diffusivity of material, m^2/s ;
- α_0 is EHL semi contact width, m;
- B is transient thermal resistance;
- U_s is sliding velocity, m/s;
- τ is shear stress, Pa;
- k_{oil} is thermal conductivity of the oil, W/m-K.

Using these temperatures in conjunction with the skin (boundary) temperature rise, T_B , calculated using a heat partition α_h between the two surfaces, the mean film temperature, $\bar{T}$, can be calculated.

$$\alpha_h = \frac{1.06 B_2 + \frac{h_c}{2 k_{oil} A} + M_2}{1.06 (B_1 + B_2) + \frac{h_c}{k_{oil} A} + (M_1 + M_2)} \quad (8)$$

$$T_B = \frac{U_1 T_{B1} + U_2 T_{B2}}{U_1 + U_2} \quad (9)$$

$$\bar{T} = T_B + (\Delta T_f)_{av} + (\Delta T_{oil})_{av} \quad (10)$$

where

- M is steady state thermal resistance of contacting body.

The temperatures are repeatedly evaluated until the desired convergence is achieved (usually within 0.1 °C or less). Once these are known, the dissipated power in the form of heat can be calculated by simply multiplying the load with the friction coefficient and the sliding speed.

The EHL model assumes that the basic lubricant properties do not change within the contact and are calculated either in respect to the inlet conditions or to the mean film temperature, based on the assumption by Evans [18]. The lubricant properties that are used are based on a synthetic 75W90 type gear lubricant and they are calculated using a linear approach for the variation of density and pressure viscosity coefficient with temperature. The shear modulus is calculated using an exponential approach quoted by Muraki [19]. The ASTM equation based on the ASTM D341-722 standard is used to calculate the low shear rate viscosity of the lubricant, based on measurements at 40 °C and 100 °C.

$$\log \log (v + 0.7) = b - c \log T \quad (11)$$

where

- v is kinematic viscosity, m^2/s ;
- b is tooth face width, m;
- c is constant.

The model can successfully predict the traction regime and friction coefficient of a combination of materials and lubricants. Roughness of the surfaces is also taken into account by means of the non-dimensional lambda (λ) value which necessitates the use of a boundary modified friction coefficient of the form shown below, suggested by Smeeth and Spikes [20] for $m = 2$.