

Methodology for Predictive Friction Modeling in Direct-Acting Mechanical Bucket Valvetrain System

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ABSTRACT

Valvetrain friction can represent a substantial portion of overall engine friction, especially at low operating speed. This paper describes the methodology for predictive modeling of frictional losses in the direct-acting mechanical bucket tappet-type valvetrain. The proposed modeling technique combines advanced mathematical models based on established theories of Hertzian contact, hydrodynamic and elastohydrodynamic lubrication (EHL), asperity contact of rough surfaces, flash temperature, and lubricant rheology with detailed measurements of lubricant properties and surface finish, driven by a detailed analysis of valvetrain system kinematics and dynamics. The contributions of individual friction components to the overall valvetrain frictional loss were identified and quantified. Calculated valvetrain friction was validated against motored valvetrain friction torque measurements on two engines. The system friction was analyzed across the operating speed range and at several oil supply temperatures as well as varying component surface finishes. A good agreement was observed between simulated and measured valvetrain friction torque suggesting that the proposed analytical methodology and the tool can be very useful in guiding new engine design and enhancing the performance of a given valvetrain design.

CITATION: Okarmus, M., Keribar, R., Zdrodowski, R., and Gangopadhyay, A., "Methodology for Predictive Friction Modeling in Direct-Acting Mechanical Bucket Valvetrain System," *SAE Int. J. Fuels Lubr.* 8(1):2015, doi:10.4271/2015-01-0677.

INTRODUCTION

Valvetrain frictional losses and wear of valvetrain components have increasingly become important aspects in the design process of internal combustion engine, which is driven by the changing emission norms, fuel economy and performance requirements. Valvetrain friction can represent a substantial portion of overall engine friction, in particular at low operating speed. In order to minimize frictional losses and the possibility of severe wear, the analysis of tribological phenomena occurring at contact/sliding interfaces (e.g. cam-follower, journal bearing and guide) between valvetrain components is necessary. In fact, the tribological conditions at the concentrated contact between cam and follower are one of the most severe anywhere in an engine, as a result of high loads and small contact area [1]. In case of the direct-acting valvetrain mechanism, the total valvetrain friction is dominated by the losses at this sliding cam-follower contact interface. The extremely thin lubricant film, often within the same scale as the roughness of contacting surfaces, causes the valvetrain to operate primarily in the mixed lubrication regime where load is supported by the partial surface contact. The direct

surface asperity contact results in high frictional energy dissipation, localized heating of the surface material and lubricant and affects the useful service life of the valvetrain [2].

Analytical models of valvetrain dynamics and performance are currently an integral part of the design and optimization process [1]. Nevertheless, accurate prediction of valvetrain friction and wear remains a complex task, which requires careful characterization of component geometry, loads, sliding speeds, lubricant properties, materials, surface finishes and coatings, etc. Modeling methodologies of valvetrain frictional losses range from correlative and simplified to highly detailed. In computationally-efficient but non-predictive models, the value of effective "lubricated" friction coefficient is basically "dialed-in" to produce the Stribeck behavior (see [Figure 1](#)) that will achieve satisfactory correlation with experimental measurements in the operating range. High-fidelity but computationally-intensive three-dimensional models make use of component elasticity and of elastohydrodynamic lubrication in conformal and counterformal contacts [3, 4].

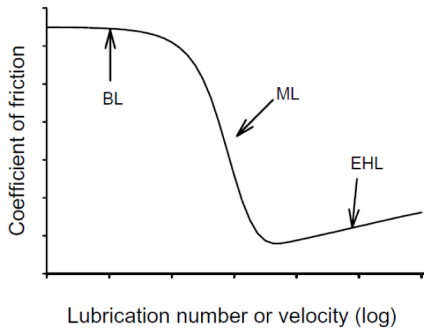


Figure 1. A typical Stribeck curve

This paper describes the methodology for predictive modeling of frictional losses in the direct-acting mechanical bucket tappet-type valvetrain shown in Figure 2. The proposed modeling technique combines advanced mathematical models based on established theories of Hertzian contact, hydrodynamic and elastohydrodynamic lubrication, asperity contact of rough surfaces, flash temperature and lubricant rheology with detailed measurements of lubricant properties and surface finish. It relies on detailed analysis of valvetrain system kinematics and dynamics to simulate local loads, scrubbing and oil entrainment velocities etc. and enables identification and quantification of the contributions of individual friction components to overall valvetrain friction loss.



Figure 2. Direct-acting mechanical bucket tappet-type valvetrain

Validation of calculated results via comparison to friction torque measurements from a valvetrain system of two different passenger car engines obtained on motored cylinder head test rigs is also presented.

FRICITION MODELING METHODOLOGY

Kinematic Analysis

The description of direct-acting valvetrain friction modeling methodology begins with a brief summary of cam-flat tappet contact kinematics necessary due to its relevance to the lubrication study.

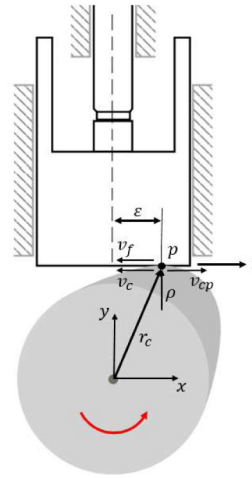


Figure 3. Cam-flat follower contact kinematics

Vector denoted by r_c in Figure 3 is a position vector of cam-follower contact point p measured with respect to the inertial frame (x, y) . ε is the distance from the center axis of flat tappet to the contact point p measured along the direction t tangent to the flat follower surface. The in-plane radius of curvature of the cam profile at the point of contact is denoted by ρ . The tangential components of velocities of material points on the cam and follower surface at location corresponding to the contact point are labeled with v_c and v_f respectively. Finally, v_{cp} represents the tangential velocity of the contact point itself. Using the above nomenclature, the mean entraining velocity is given by:

$$v_e = \frac{1}{2}(v_c + v_f - 2v_{cp}) \quad (1)$$

The tangential scrub velocity is:

$$v_s = v_c - v_f \quad (2)$$

Dynamic Analysis

The multi-body dynamics (MBD) analysis of a valvetrain system described in [5] calculates inter-component contact forces and amounts to solving (and integrating) the following constrained system of equations to obtain the states (i.e. displacements and velocities) of structural components of the valvetrain, including cam, tappet, valve and spring:

$$\begin{bmatrix} M_s & B^T \\ B & 0 \end{bmatrix} \begin{bmatrix} \ddot{u} \\ \lambda \end{bmatrix} = \begin{bmatrix} G \\ \gamma \end{bmatrix} \quad (3)$$

where M_s is the configuration dependent, positive-definite mass matrix, G is a vector that includes damping, elastic and externally applied loads, u is the displacement vector, B is the constraint Jacobian matrix, λ is a vector of Lagrange multipliers associated with

constraints, γ is the vector containing terms associated with the differentiation of constraint equations and $(\cdot)$ represents the 2nd derivative with respect to time. The description of MBD valvetrain models used in this project can be found in the [Appendix](#).

Cam-Follower Contact

Contact Hertz Pressure

The normal contact load at the cam-tappet interface is calculated using the solution of [equation 3](#) (i.e. displacements of structural components are used to compute contact approach) via the application of Hertz theory of elastic contacts [6]. This procedure allows the Hertz contact pressure, deformation and the dimensions of elliptical contact patch to be simultaneously predicted.

Elastohydrodynamic Lubrication

For isothermal, fully flooded elliptical contact conjunction, Hamrock and Dowson [7] proposed a theoretical formulation for minimum and central film thickness. The central film thickness is expressed in dimensionless form as:

$$H_c = 2.69U^{0.67}G^{0.53}W^{-0.067}(1 - 0.61e^{-0.73k}) \quad (4)$$

where k is the point contact ellipticity ratio and the three non-dimensional parameters U , G and W are equal to the following:

$$U = \frac{\mu v_e}{E' R_x} \quad (5)$$

$$G = \alpha E' \quad (6)$$

$$W = \frac{F_h}{E' R_x^2} \quad (7)$$

μ is the dynamic viscosity of lubricant at inlet to the contact conjunction, E' the equivalent Young's modulus, R_x the equivalent radius of curvature in the direction of lubricant entrainment, α is the pressure-viscosity coefficient and F_h is the portion of total contact load carried by hydrodynamic film.

It should be noted that these formulas were developed for steady-state EHL conditions and result in zero film thickness when the entrainment velocity v_e equals to zero. Although squeeze film action has been recognized as being important, especially around the areas of small lubricant entrainment, this model is assumed adequate for use in the qualitative analysis of valvetrain designs [8].

Asperity Interactions

In highly loaded lubricated contact of rough surfaces, the total load is shared between hydrodynamic pressure and asperity contact. The surface asperity interaction model used in the proposed methodology is that developed by Greenwood and Tripp [9]. The model is based on

the assumption of a Gaussian distribution of asperity heights and constant radius of curvature of symmetric asperities. The load carried by the asperities can be calculated using the following equation:

$$F_a = \frac{8\sqrt{2}}{15} \pi (\sigma \beta \eta)^2 \sqrt{\frac{\sigma}{\beta}} E' A F_{\frac{5}{2}}\left(\frac{d}{\sigma}\right) \quad (8)$$

where σ is the composite standard deviation of asperity peak (or summit) distributions, β is the radius of curvature at the peak, η is the asperity density, A is the apparent area of contact and d denotes the separation between the mean levels of asperity height distributions of the contacting rough surfaces. The statistical function $F_{\frac{5}{2}}$ is defined by the following integral:

$$F_n(s) = \frac{1}{\sqrt{2\pi}} \int_s^\infty (u-s)^n e^{-\frac{u^2}{2}} du, n = \frac{5}{2}, s = \frac{d}{\sigma} \quad (9)$$

An iterative procedure is employed to split the total contact load into its hydrodynamic component F_h and the asperity component F_a via the repeated application of [equations 4 and 8](#), effectively combining the theories of elastohydrodynamic lubrication of two smooth surfaces in contact and the asperity contact between rough surfaces. The division of load between the film and the asperities in the circumstances where both contribute to supporting the load - the condition frequently referred to as partial or mixed elastohydrodynamic lubrication - has been investigated in the past [10, 11]. The area of elliptical contact patch is taken as the apparent area of contact S . The value of equivalent Young's modulus E' for the cam-follower contact was calculated from cam and tappet material properties to be equal to 153 GPa.

An important thing to point out, something that many valvetrain tribologists and authors take for granted, is that there is a distinction between the separation of mean levels of asperity height distributions of rough surfaces d used in the Greenwood-Tripp model and the separation between mean levels of surface distributions taken to be equivalent to the central oil film thickness h_c predicted in dimensionless form by [equation 4](#). This is illustrated in [Figure 4](#).

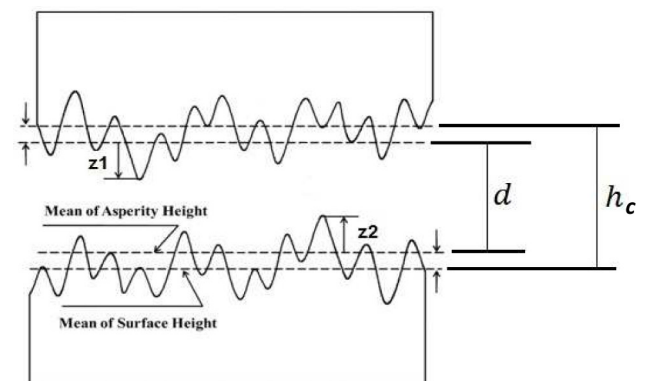


Figure 4. Illustration of separation distance of mean surfaces (h_c), used for EHD model, and distance between summit height means, used in the Greenwood-Tripp asperity contact model

This important distinction is considered in the proposed modeling methodology as it has shown to have a significant influence on valvetrain friction predictions. Asperity peak contact separation d can be computed using [equation 10](#):

$$d = h_c - g\sigma \quad (10)$$

The standard deviation of the composite asperity peak height distribution σ is computed by statistically combining the standard deviations of individual summit height distributions of each contact surface σ_1 and σ_2 as follows:

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2} \quad (11)$$

A non-dimensional distance parameter g referred to as the composite normalized asperity peak mean height and used in [equation 10](#) is computed by again combining the corresponding parameters of individual surfaces g_1 and g_2 in the following manner:

$$g = \frac{g_1\sigma_1 + g_2\sigma_2}{\sigma} \quad (12)$$

For an individual rough surface, shown in [Figure 5](#), parameter g_i represents a distance between the mean of asperity peak height distribution $\bar{z}_i$ and the mean level of the surface distribution $\bar{z}_{si}$ normalized by σ_i of that surface:

$$g_i = \frac{\bar{z}_i - \bar{z}_{si}}{\sigma_i}, i = 1, 2 \quad (13)$$

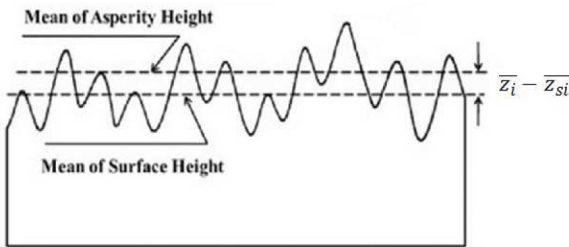


Figure 5. Distance between mean lines of asperity summit height and surface height distributions for a single rough surface

Whitehouse and Archard [12] reported the typical values of g_i observed in freshly ground and bead-blasted surfaces to be about 1.14. Experimental surface roughness measurements acquired on various valvetrain components (see the Experimental Measurements section below) and used in this study revealed the range of values between 0.7 and 1.8.

As mentioned above, one simplifying feature of the Greenwood-Tripp asperity contact model is the Gaussian distribution of asperity summit heights. In practice, this is an approximation as the peak heights distribution of typical surfaces can deviate significantly from the Gaussian assumption although there are evidences that the summit heights indeed have Gaussian distribution even if the surface,

as a whole, does not [13]. Another simplifying assumption made in the model is that the radius of curvature of asperity peaks β on both surfaces in contact is constant and so is the asperity density η . In reality, the two surfaces in contact will usually have different topography. Nevertheless, very little experimental data and evidence in literature is available about the values of these two surface parameters. The product of $\sigma\beta\eta$ is normally taken to be in the range of $0.03 < \sigma\beta\eta < 0.05$ although some recent works obtained different values [13]. An alternative approach to the statistical definition of rough surfaces shows that a given surface can be characterized by two parameters: standard deviation σ and the correlation length l , which is a measure of the effective wave length of the main surface texture [12]. According to this approach, the density η is proportional to l^{-2} and the mean radius of asperity summits is proportional to l^2/σ . It then follows that the three roughness parameters are not independent but, instead, their product $\sigma\beta\eta$ is roughly constant.

Much attention was devoted in this study to the analysis of roughness parameters σ , β and η , their variability observed in the numerous measurements taken on both new and worn surfaces as well as methods for combining the parameters from individual surfaces into composite values for use in the Greenwood-Tripp asperity model and their effect on friction predictions. [Equation 11](#) provides a standard way of computing the standard deviation of the composite asperity peak heights distribution. In accordance with Hertzian solution for the contact of two paraboloids, the composite β was computed as:

$$\frac{1}{\beta} = \frac{1}{\beta_1} + \frac{1}{\beta_2} \quad (14)$$

The composite asperity density η was taken as:

$$\eta = \frac{1}{2}(\eta_1 + \eta_2) \quad (15)$$

Further discussion concerning the variability of roughness parameters and issues encountered during the application of above equations in valvetrain friction studies will be provided in the following sections of the report.

Temperature in EHD Contact

Most of the frictional loss is converted into heat resulting in local increase of the surface and lubricant temperature [1]. The temperature rise in the cam-tappet contact conjunction due to frictional heating was estimated using the concept of flash temperature developed by Blok [14]. The maximum conjunction temperature T_c is given as:

$$T_c = T_b + 1.064 \frac{F_f v_s}{A} \frac{\sqrt{2b}}{b_c \sqrt{v_c + b_f \sqrt{v_f}}} \quad (16)$$

T_b is the bulk temperature of cam and tappet, F_f is the magnitude of cam-follower friction force, A is the apparent area of contact, b_c and b_f are the thermal contact coefficients of cam and tappet, respectively. Tangential velocities v_s , v_c and v_f have already been defined in [equation 2](#).



Tappet Rotation

It has been demonstrated experimentally [15, 16] and through the application of numerical simulation [17, 18] that the flat tappet rotation about its axis of symmetry can have an effect on valvetrain friction. With the cam offset axially away from the center axis of the tappet (see Figure 6), the friction force developed in the cam-follower contact conjunction, when multiplied by the offset, generates a torque which can cause the tappet to spin in its bore.

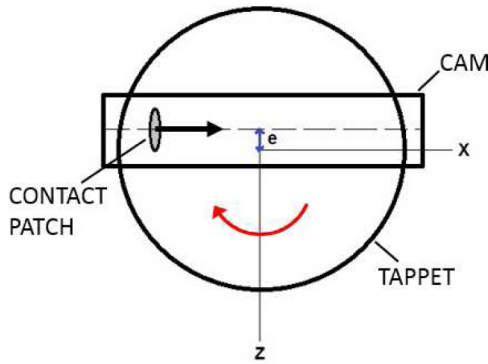


Figure 6. Bottom view of the cam-tappet geometry with the cam offset axially from the center axis of the tappet

A detailed analysis of tappet rotation and its influence on entrainment and scrub velocities at contact point has been carried out to understand and quantify the effect of tappet angular velocity on cam-follower friction. Introduction of tappet rotation has resulted in a reduction of magnitude of scrub velocity as well as an increase of magnitude of entrainment velocity when the contact point is on cam flanks and a decrease of magnitude of entrainment velocity with the contact point located over cam nose. Because majority of frictional power dissipation occurs when the contact point is at or in vicinity of cam nose, the final conclusion was that the overall cam-follower power loss will increase somewhat when tappet rotation is modeled. Nevertheless, because of the lack of experimental data for lifter angular velocity and due to conflicting information about tappet rotational behavior (which can be erratic and difficult to measure) found in literature, the effect of tappet spin was not included in the proposed modeling methodology.

Camshaft Journal Bearings

The analysis of dynamically loaded journal bearings is necessary for the proper design of bearings in many industrial applications [19]. The behavior of camshaft journal bearings is critical to the valvetrain's operation from the point of view of friction, performance and durability. Although these journal bearings are expected to operate mostly in the hydrodynamic lubrication regime, valvetrain loads can create journal misalignment that causes metal-to-metal contact with the bearing shell. Hence, in addition to pure hydrodynamic lubrication, the mixed lubrication modeling is an important aspect of the bearing analysis methodology [20]. The simulation technique to analyze camshaft journal bearings used in this study was the mobility solution [21]. This method provides extremely efficient and robust journal orbit/trajectory calculation through use of bearing characteristics stored in the form of "mobility maps", but is limited to analyzing ideal, smooth and rigid circular

bearings. To obtain meaningful predictions of frictional losses in the bearing during operation in the mixed lubrication regime, the mobility solution was combined with Greenwood-Tripp asperity contact model. Additionally, influences of oil feed features on oil flow and load carrying capacity of the bearing as well as a thermal model to predict the rise in lubricant temperature resulting from frictional heat dissipation were included [5].

Tappet and Valve Guides

Investigation of simulation results confirmed that the contribution of tappet and valve guide friction components to the overall valvetrain frictional loss is small as compared to the friction generated in the cam-follower contact interface and camshaft bearings. Based on this observation and for computational-efficiency reasons, the friction force in tappet and valve guides was modeled using a simple relationship involving a normal load-dependent and speed-dependent friction coefficients:

$$F_f = c_f F_n + c_d \Delta v \quad (17)$$

where F_f is the friction force in the guide, F_n is the normal load, Δv is the relative sliding velocity, c_f is the dry friction coefficient and c_d is the speed-dependent viscous friction coefficient computed using oil viscosity μ , guide diameter D , guide length L and nominal clearance in the guide c as:

$$c_d = \frac{\mu \pi D L}{c} \quad (18)$$

Lubricant Rheology

In HD and EHD contacts, lubricant properties play a very important role from the point of view of film formation and friction [1]. Oil viscosity affects film thickness and influences when transitions occur between boundary, mixed, and hydrodynamic friction regimes. The viscosity of multi-grade oil depends on temperature, shear rate, and pressure, all of which vary over the range of engine operating conditions. Even for a given engine operating point, the apparent viscosity varies between different engine components, such as a cam-tappet contact (high oil shear rate, high contact pressure), a journal bearing (high oil shear rate, low contact pressure), or a piston ring at top-dead-center (low oil shear rate, high contact pressure).

In the proposed methodology, the effect of shear rate on viscosity was represented using the shear thinning model based on the Carreau equation [22]:

$$\mu(\dot{\gamma}) = \mu_\infty + (\mu_0 - \mu_\infty)[1 + (\lambda\dot{\gamma})^2]^p \quad (19)$$

where μ_0 , and μ_∞ are the reference viscosity at zero shear and the viscosity at infinite shear, respectively, $\dot{\gamma}$ is the shear rate and λ and p are model parameters, which were determined from experimental measurements of viscosity via the application of least-squares method.