

Development of a Simplified Instantaneous Friction Model of the Piston-Crank-Slider Mechanism of Internal Combustion Engines

2011-01-0612

Published
04/12/2011

Georgios Livanos
Assistant Professor, TEI Athens

Copyright © 2011 SAE International

doi:[10.4271/2011-01-0612](https://doi.org/10.4271/2011-01-0612)

ABSTRACT

The continuous increase of the green house emissions in conjunction with the limited and finite fuel resources make the improvement of efficiency of all engines, converting fuel chemical energy to mechanical energy, imperative. Even small increase in engine's mechanical efficiency, can be proved significant in economical and environmental terms. Towards this direction, the tribology studies of the Internal Combustion Engines, mainly used as propulsion, power generation and auxiliary drives, are considered important for the design of new engines and the improvement of existing ones.

The objective of this paper is to present a simplified (analytical) model for the prediction of the instantaneous friction, developed on the main components of the piston-crank-slider mechanism of an internal combustion engine, including complete piston ring packs, piston skirts, connecting rod bearings and crankshaft main bearings. In contrast to the detailed numerical models, the developed simplified model requires limited computational resources and can be easily implemented or used in conjunction with existing codes, modeling thermodynamic engine processes or engine control algorithms. On the other hand, the proposed model compared with the semi-empirical friction formulas can be considered more general and non case dependent, since it is based on the physics of engine lubrication (short bearing theory) and not on statistical observations (case related) of motoring friction tests.

The proposed model was used in the case of a typical, small, single cylinder, four-stroke engine. The obtained simulation results were compared with results from a detailed numerical

engine friction simulation code, previously developed and published by the author [1]. Friction measurements were also conducted on simplified friction measurement test cell, consisting of an AC electric motor driving the piston-crank-slider mechanism of an internal combustion engine. Good agreement was found between simulation and experimental results, indicating that the simplified friction model is capable of adequately predicting the dynamic friction behavior of rotating and oscillating engine main components.

INTRODUCTION

Nowadays, the continuously increasing cost of engine fuels in conjunction with the environmental impacts of the operation of internal combustion engines make the improvement of the engine's mechanical efficiency imperative.

A good understanding and measurement of the friction contributions of the various engine rotating or oscillating components is needed, if mechanical efficiency is to be improved.

The piston-crank-slider mechanism is considered as the major contributor to the overall mechanical friction of the engine. Recent literature data of four-stroke Diesel engines [2] suggests that the total friction due to pistons, rings and connecting rods will contribute 1.6% to 8.3% of the total available energy or 4.1% to 20.9% of the power output.

Another more recent analysis [3] states that engine friction loss including piston skirt friction, piston rings and bearings is 66% of the total friction loss, whereas the valve train, crankshaft, transmission and gears accounts for approximately 34%. The same source also suggests that concerning power-train friction loss, sliding of the piston

rings and piston skirt against the cylinder liner is undoubtedly the largest contribution to the friction in a power-train system, whereas frictional losses arising from the rotating engine bearings are the next most significant, followed by the valve train and the auxiliaries.

A similar friction losses distribution is also suggested in [4, 5], where the piston assembly is responsible for the lions share, approaching on-half (45% of total friction losses), followed by bearings (25%), pumping losses (20%) and valve train (10%).

Despite of the discrepancies observed between the several studies, it is generally accepted that the piston - crank - slider assembly has the major contribution to the total engine friction.

In all aforementioned studies it is stated that the exact friction losses distribution strongly depends on the engine design. This observation may lead to the conclusion that detail friction models considering engine geometry should be used if the engine friction is to be estimated, using a single model for several different engine designs.

Significant amount of work has been done through engine tests, experiments and theoretical modeling to understand the different aspects of friction of internal combustion engines. Benefiting from more and more experimental evidence, modeling work has been advancing in many aspects and has played an important role in understanding the physics of engine friction components and giving practical guidance to the design processes.

The engine friction models are generally grouped into two categories: numerical friction models of particular engine components, i.e. piston rings [6, 7, 8, 9], piston skirts [10, 11, 12, 13, 14, 15, 16, 17, 18], connecting rod bearings [19, 20, 21, 22, 23, 24, 25, 26, 27], main bearings and valve train mechanisms; and semi-empirical friction models [28, 29, 30, 31, 32]. The first category models are mainly used for studies of new engine component designs, whereas the second category models are used in more integrated investigations of engine response under several steady-state operating conditions.

The continuously increasing interest in engine dynamics during transient operation had led to the development of more sophisticated friction models, capable of predicting the instant friction torque of the several engine modules. The observation made by Winterbone and Tennant in [33] that friction levels under transient performance are appreciable higher than the steady state levels at the same load and speed, increased the motivation for the development of instant torque friction models [34, 35, 36, 37, 38, 39, 40, 41, 42].

In this paper, a simplified friction model capable of predicting the instantaneous friction losses of the piston-crank-slider mechanism of an internal combustion engine is presented. The proposed model is based on the short bearing approximation of the hydrodynamic lubrication theory and, in contrast with the existing friction models, considers the dynamic interconnection of the parts of piston-crank-slider mechanism, by solving the coupled system of differential equations which describes the mechanism motion.

THEORETICAL MODEL

The detailed friction models [1, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26, 27], which describe in detail the friction development on internal combustion engine main components, require increased computational resources and may be proved difficult to be implemented or used in conjunction with existing codes modeling thermodynamic, dynamic engine processes and control schemes. On the other hand, simplified analytical models are not only easily used, but additionally can provide a fast insight into the tribological conditions prevailing on elements under consideration, and thus help in the calculation of initial conditions applied on detailed literature models [1, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26, 27]. For these reasons, a simplified friction model was developed based on short bearing lubrication theory. As described below, the developed model considers that the piston-crank-slider friction contributors are the piston rings, the piston skirts, the main bearings, the small end and big end connecting rod bearings.

PISTON RING PACK

The model solves the Reynolds equation
$$\frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \right) = \frac{1}{2} u \frac{\partial h}{\partial x} + \frac{\partial h}{\partial t}$$
 describing the lubricant flow around each ring of the pack by making two simplified assumptions.

The first assumption states that the ring profile features a symmetrical barrel face described by the following function

$$h(x) = (h_{\min} + R_{\text{ring}}) \left(1 - \frac{R_{\text{ring}}}{h_{\min} + R_{\text{ring}}} \cos \theta \right) \quad (1)$$

This assumption is very common in the literature [43, 44], corresponds to measured piston ring profiles of typical test engines and enables the analytical solution of Reynold's equation as will be explained below. It is also to be noted that ring face profiles, significantly change with the engine operation, due to the phenomenon of wear and very usually settle down to parabolic shapes, although their original shape

was completely different, as a result of the oscillating motion of piston rings.

The second assumption states that the ring oil wetted surface extended from x_1 to x_2 is known either from experimental observations or detailed simulation results (Figure 1).

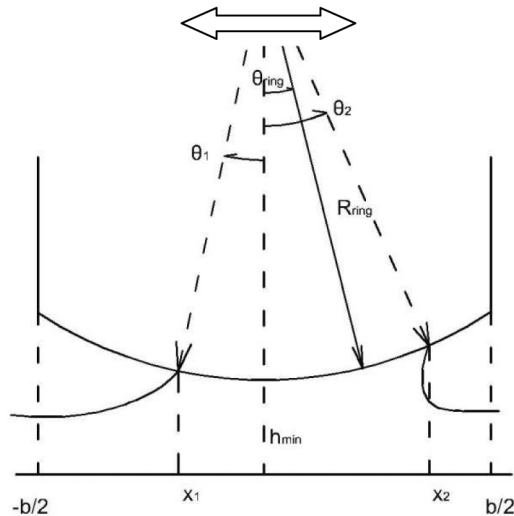


Figure 1. Piston Ring Assumed Profile

Integrating the Reynolds equation and applying the aforementioned assumptions the following expression is obtained for pressure distribution along the ring width:

$$p(x, t) = -6\mu u [J_1(x) - h(x_1)J_2(x)] + 12\mu \frac{\partial h}{\partial t} \left[(J_3(x) - x_1 J_2(x)) - \frac{J_3(x_2) - x_1 J_2(x_2)}{J_2(x_2)} J_2(x) \right] + \frac{p_2 - p_1 + 6\mu u (J_1(x_2) - h(x_1)J_2(x_2))}{J_2(x_2)} J_2(x) + p_1 \quad (2)$$

Where

$$J_1(x) = \int_{x_1}^x \frac{1}{h^2(x)} dx \quad (3)$$

$$J_2(x) = \int_{x_1}^x \frac{1}{h^3(x)} dx \quad (4)$$

$$J_3(x) = \int_{x_1}^x \frac{x}{h^3(x)} dx \quad (5)$$

These integrals can be calculated analytically if the following substitutions are applied:

Considering and substituting:

$$x = R_{ring} \sin \theta \quad (6)$$

$$J_1(\theta) = \frac{R_{ring}}{(h_{min} + R_{ring})^2} \int_{\theta_1}^{\theta} \frac{\cos \theta}{\left(1 - \frac{R_{ring}}{h_{min} + R_{ring}} \cos \theta\right)^2} d\theta \quad (7)$$

$$J_2(\theta) = \frac{R_{ring}}{(h_{min} + R_{ring})^3} \int_{\theta_1}^{\theta} \frac{\cos \theta}{\left(1 - \frac{R_{ring}}{h_{min} + R_{ring}} \cos \theta\right)^3} d\theta \quad (8)$$

$$J_3(\theta) = \frac{R_{ring}^2}{(h_{min} + R_{ring})^3} \int_{\theta_1}^{\theta} \frac{\sin \theta \cos \theta}{\left(1 - \frac{R_{ring}}{h_{min} + R_{ring}} \cos \theta\right)^3} d\theta \quad (9)$$

Finally we consider the following substitution which introduce a new variable ψ :

$$1 - \frac{R_{ring}}{h_{min} + R_{ring}} \cos \theta = \frac{1 - \left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2}{1 + \frac{R_{ring}}{h_{min} + R_{ring}} \cos \psi} \quad (10)$$

So, that the analytical calculation of the integrals (7), (8), (9) could be done more easily as follows:

$$\cos \theta = \frac{\frac{R_{ring}}{h_{min} + R_{ring}} + \cos \psi}{1 + \frac{R_{ring}}{h_{min} + R_{ring}} \cos \psi} \quad (11)$$

$$\sin \theta = \frac{\sqrt{1 - \left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2} \sin \psi}{1 + \frac{R_{ring}}{h_{min} + R_{ring}} \cos \psi} \quad (12)$$

$$d\theta = \frac{\sqrt{1 - \left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2}}{1 + \frac{R_{ring}}{h_{min} + R_{ring}} \cos \psi} d\psi \quad (13)$$

The following expressions are obtained:

$$J_1(\psi) = \frac{R_{ring}}{(h_{min} + R_{ring})^2} \left[1 - \left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2 \right]^{\frac{3}{2}} \left(\frac{R_{ring}}{h_{min} + R_{ring}} (\psi - \psi_1) + \sin \psi - \sin \psi_1 \right) \quad (14)$$

$$J_2(\psi) = \frac{R_{ring}}{(h_{min} + R_{ring})^3} \frac{1}{\left[1 - \left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2 \right]^{\frac{5}{2}}} \cdot \left[\left(\left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2 + 1 \right) (\sin \psi - \sin \psi_1) + \frac{3}{2} \frac{R_{ring}}{h_{min} + R_{ring}} (\psi - \psi_1) + \frac{1}{4} \frac{R_{ring}}{h_{min} + R_{ring}} (\sin 2\psi - \sin 2\psi_1) \right] \quad (15)$$

$$J_3(\psi) = \frac{R_{ring}^2}{(h_{min} + R_{ring})^3} \frac{1}{2 \left[1 - \left(\frac{R_{ring}}{h_{min} + R_{ring}}\right)^2 \right]^2} \cdot \left(\frac{-2R_{ring}}{h_{min} + R_{ring}} (\cos \psi - \cos \psi_1) - \cos^2 \psi + \cos^2 \psi_1 \right) \quad (16)$$

As can be seen from [equations \(2\), \(14\), \(15\) and \(16\)](#) an analytical expression for the oil pressure distribution along the ring width was obtained. Integrating [expression \(2\)](#) and applying force equilibrium the following expression is obtained for the oil squeeze speed

$$\frac{\partial h}{\partial t} = \frac{W_{external} - W_{asperity} - p_1(x_2 - x_1) - 6\mu u \left(\iint_A J_1 dx - h(x_1) \iint_A J_2 dx \right)}{12\mu \left(\iint_A J_3 dx - x_1 \iint_A J_2 dx - \frac{J_3(x_2) - x_1 J_2(x_2)}{J_2(x_2)} \iint_A J_2 dx \right)} - \frac{p_2 - p_1 + 6\mu u (J_1(x_2) - h(x_1) J_2(x_2))}{J_2(x_2)} \iint_A J_2 dx \quad (17)$$

$$\tau = \frac{h}{2} \frac{\partial p}{\partial x} - \mu \frac{u}{h}, \text{ where } \tau \text{ is the fluid (oil) shear stress}$$

After integrating [expression \(17\)](#) with respect to the time, the minimum oil film thickness (h) can be calculated for a complete cycle, and as a result the friction losses can be predicted integrating [equation \(18\)](#) along ring width. It is to be noted that the friction losses are calculated directly for each time step without solving any non-linear equation as in the case of the detailed model [1]. Consequently, significantly reduced computational effort is required.

PISTON SKIRTS

The developed simplified friction model, incorporates a hydrodynamic lubrication model of the piston skirt, and allows the piston motion and the associated skirt friction to be calculated as function of engine crank angle.

It is well known from the existing literature [[10](#), [11](#), [12](#), [13](#), [14](#), [15](#), [16](#), [17](#), [18](#), [45](#)] that the piston executes small translations and rotations within the confinement of the cylinder clearance (Piston Secondary Motion). These motions can be defined by the piston eccentricities “et” and “eb” at the top and bottom of the skirt, respectively ([Figure 2](#)). It is to be noted that et and eb are functions of time, and to be predicted by solving the equilibrium equations for the piston ([equations 21](#)), which are analytically described in the literature [[10](#), [11](#), [12](#), [13](#), [14](#), [15](#), [16](#), [17](#), [18](#), [45](#)].

The development of the simplified friction model of piston skirts was based in the assumption of “short bearing” approximation of Reynolds equation describing the lubrication of piston skirts. According to this assumption there is no pressure relief in the axial direction and thus the pressure gradient term (in the axial direction) of the Reynolds equation is considered negligible.

Another assumption also made in the simplified friction model of piston skirts is that the oil film thickness along the circumferential direction of piston remains constant. Adopting this assumption means that the curvature of piston and piston liner in the area of piston skirts is negligible. This assumption is described by the [expression \(18\)](#) which holds and is depicted in [Figure 2](#).

$$h(y) = C \pm e_t \pm \frac{e_b - e_t}{L} y \quad (18)$$

In [equation \(18\)](#) the positive sign holds for the thrust side whereas the negative sign holds for the anti-thrust side of cylinder liner, as is defined in [Figure 2](#).

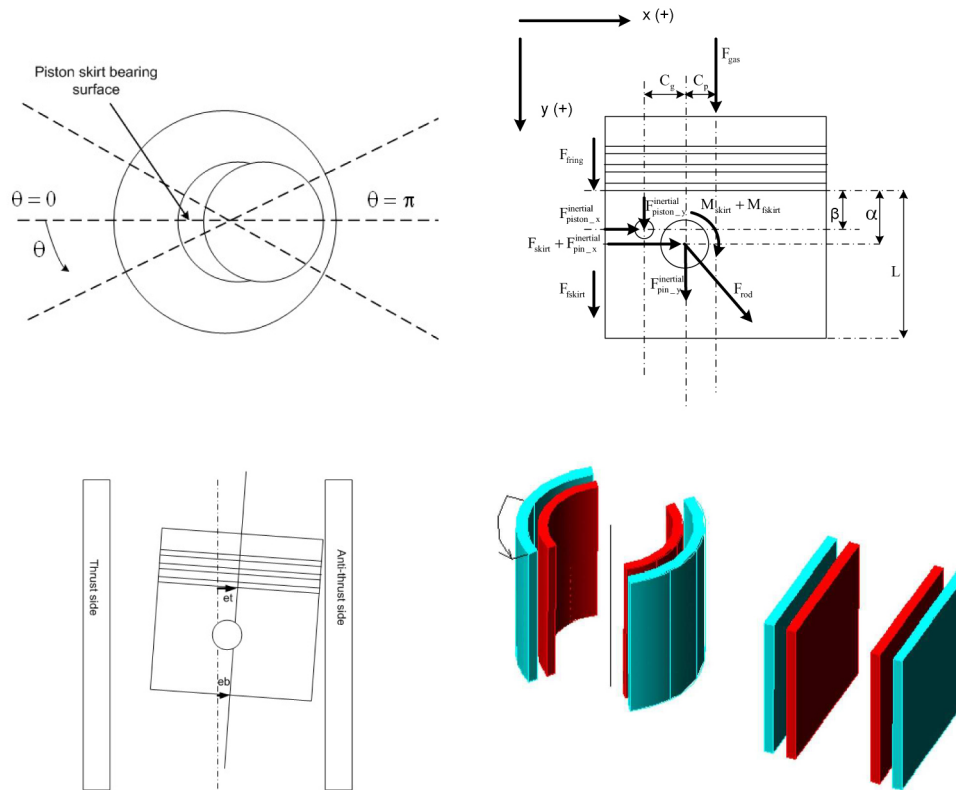


Figure 2. (Left) Piston Skirts Eccentricity Terminology, (Upper Right) Forces and moments acting on piston body, (Lower Right) Piston Skirt Shape Assumption in the Simplified Friction Model

Using these assumptions, the Reynolds equation describing the lubrication of piston skirts [10, 11, 12, 13, 14, 15, 16, 17, 18] yields:

$$\frac{1}{R^2} \frac{\partial}{\partial \theta} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial \theta} \right) = \pm \frac{1}{2} u \frac{e_t - e_b}{L} \pm \left(\dot{e}_t \left(1 - \frac{y}{L} \right) + \dot{e}_b \frac{y}{L} \right) \quad (19)$$

Integrating equation (19) and applying boundary conditions along the boundaries of the piston skirt the following equation for pressure distribution is obtained for thrust (positive sign) and anti-thrust sides (negative sign).

$$p(\theta, y) = \pm \left(3\mu u R^2 \frac{e_t - e_b}{L h^3} + 6\mu R^2 \frac{\dot{e}_t \left(1 - \frac{y}{L} \right) + \dot{e}_b \frac{y}{L}}{h^3} \right) (\theta^2 - \theta^2) \quad (20)$$

Where θ is measured from the center line plane of thrust or anti-thrust side and y is measured from the top of the piston skirt. After the integration of the obtained pressure profile, all forces and moments of the right vector of equation (21), which describes the piston secondary motion, can be obtained. Subsequently, this equation can be solved and the piston trajectory can be predicted, leading to the

determination of piston skirts friction losses. A detailed description of the solving procedure can be found in [1].

$$\begin{bmatrix} m_{\text{piston}} \left(1 - \frac{\beta}{L} \right) + m_{\text{pin}} \left(1 - \frac{\alpha}{L} \right) & m_{\text{piston}} \left(\frac{\beta}{L} \right) + m_{\text{pin}} \left(\frac{\alpha}{L} \right) \\ \frac{I_{\text{piston}}}{L} + m_{\text{piston}} (\alpha - \beta) \cdot \left(1 - \frac{\beta}{L} \right) & -\frac{I_{\text{piston}}}{L} + m_{\text{piston}} (\alpha - \beta) \cdot \left(\frac{\beta}{L} \right) \end{bmatrix} \begin{bmatrix} \ddot{e}_t \\ \ddot{e}_b \end{bmatrix} = \begin{bmatrix} F_{\text{skirt}} + (F_{\text{fskirt}} + F_{\text{fring}} + F_{\text{gas}} + F_{\text{pistony}} + F_{\text{piny}}) \tan \phi \\ M_{\text{skirt}} + M_{\text{fskirt}} + F_{\text{fring}} C_p + M_{\text{gp}} + F_{\text{gas}} C_p - F_{\text{pistony}}^{\text{inertial}} C_g \end{bmatrix} \quad (21)$$

It is to be noted that the adopted assumptions lead to the direct calculation of oil pressure developed around the piston skirts, without applying any complex and time consuming numerical scheme (Finite Element Method or Finite Differences Method), since the two dimensional Reynolds equation is approximated by an one dimensional Reynolds equation, which can be easily solved.

ENGINE BEARINGS

As in the case of piston skirts, the assumption of “short bearing lubrication” is also adopted in the case of dynamically loaded journal bearings. In this theory, it is considered that the pressure flow in the axial direction (width