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Development of a mathematical tool to predict engine in-cylinder friction

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KEYWORD

Piston top compression ring
Engine cylinder liner
Friction force
Lubricant film thickness
Reynolds equation

ABSTRACT

A better fuel-efficient automotive engine is more sought-after to promote greener environment in the era of global warming. One of the factors to cause the increase of fuel consumption in vehicles is the frictional loss within an internal combustion engine. In this study, the focus is to determine the tribological behaviour between the piston top compression ring and the engine cylinder liner for a full engine cycle. Mathematical models are derived from a 1-D Reynolds equation, assuming Half-Sommerfeld and Reynolds boundary conditions. Greenwood and Tripp rough surface contact model is applied to predict frictional properties along the ring-liner contact, considering viscous and boundary friction. It is found that the Half-Sommerfeld boundary condition predicts minimum lubricant film thickness that correlates well with literature data. However, the friction force predicted by the Reynolds boundary condition along dead centres correlates better with literature data. With friction along the cylinder liner dead centres being very significant, it is, therefore, suggested that the Reynolds boundary condition be the better mathematical model in studying the piston ring-liner tribological conjunction.

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1.0 INTRODUCTION

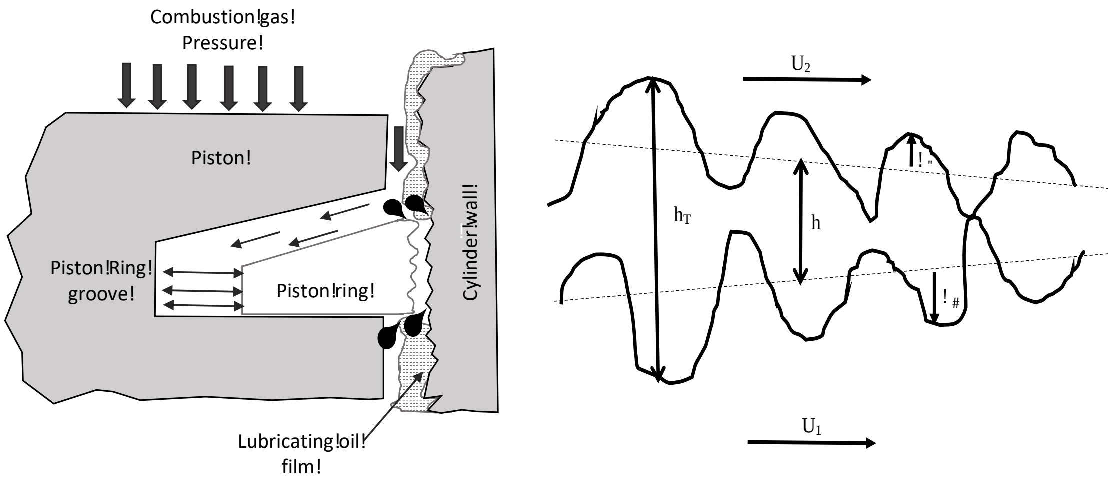
The tribological behavior of piston rings has long been recognized as an important influence on the performance of internal combustion engines in terms of power loss, fuel consumption, oil consumption, blow-by and harmful exhaust emissions (Kapsiz et al., 2011). In an engine system, the piston ring pack provides sealing between the combustion chamber and the crankcase, while the lubricant film separating the ring pack and liner assists in dissipating heat from the combustion chamber.

The heart of the reciprocating IC engine is the piston assembly, forming a critical linkage in transforming the energy generated by combustion of the fuel and air mixture into useful kinetic energy (Tung and McMillan, 2004). Therefore, tribological studies of the piston assembly, particularly the piston ring/cylinder liner contact, have dominated the interest of researchers in the fields of lubrication, friction and wear (Taylor, 1998). During engine operation, the piston ring-liner contact will be exerted by the pressure from the ring tension and the pressure behind the ring as shown in Figure 1(a) (Chong et al., 2011). The ring tension is designed for effective sealing of the combustion chamber in reducing possible combustion gas blow-by, which could significantly affect the engine efficiency. The pressure acting behind the piston ring is from the combustion gas pressure during the spark ignition of air-fuel mixture.

When the piston reciprocates from top dead centre to bottom dead centre, the frictional properties between the piston ring and the cylinder liner is affected by the lubricant fluid film formation. The effect is far more evident for the piston top compression ring. This is because the top compression ring is directly exposed to combustion chamber, which leads to this ring being highly loaded when compared with the other rings during power stroke of the engine cycle.

The frictional losses of the piston ring-liner are at its peak for engine operation near the top dead center during the power stroke. This is due to the combined effect of high combustion pressure and low piston velocity, giving rise to direct interactions between the surface asperities, as shown in Figure 1(b). This indicates that piston ring-liner contact operates under mixed and boundary lubrication regimes along the power stroke top dead center region. Along such lubrication regimes, surface roughness profile influences the friction behavior. Surface asperity contacts between piston ring and cylinder liner cannot be neglected because the lubricating film is thin (Chong et al., 2011; Chong et al., 2012; Chong and De La Cruz, 2014). Such observation implies that surface topography and shear characteristic of the contact surface affect boundary friction contribution (Styles et al., 2014).

The friction and wear performance of vehicles are highly dependent on the piston ring-liner contact in the engine as it contributes to the largest frictional losses per unit contact area in the automotive system. It is, therefore, important to better understand the mechanical friction along the piston ring-liner conjunction in an internal combustion (IC) engine, in order to further reduce in-cylinder frictional losses. One of the earliest calculations on piston ring and cylinder liner lubrication were made by Castleman Jr (1936). He pioneered the hydrodynamic lubrication theory in the piston ring analysis. Later, Eilon and Saunders (1957) conducted another lubrication analysis of a piston ring to calculate the lubricant film thickness and the friction force. In 1959, Furuhama (1959) developed a dynamic hydrodynamic analysis of the piston ring lubrication for a ring profile consisting of a flat central land bounded by two half parabolas. Furuhama related his work to the Reynolds' equation with consideration of squeeze film effect.



(a) (b)

Figure 1: Piston ring-liner contact properties. (a) Piston ring-liner conjunction, (b) Surface roughness profiles.

Ting and Mayer also (Ting and Mayer, 1974a; Ting and Mayer, 1974b) developed an analytical model that uses hydrodynamic lubrication theory to analyse the lubricant flow between ring and cylinder liner. They calculated the fluid pressure, lubricant film thickness and friction in the piston ring-liner using a simplified Reynolds' equation. Furthermore, Dowson et al., 1979 also used the Reynolds' boundary condition in the hydrodynamic lubrication model for piston ring analysis. The predicted pressure distribution and lubricant film thickness from the analysis is applied to calculate the exact amount of lubricant supply needed. Miltsios et al. (1989) developed a finite element model to solve the governing equations of piston ring-liner lubrication and to calculate the friction force on each of the rings. Instead of Reynolds' boundary condition, they also used Sommerfeld type boundary condition in their piston ring analysis.

From the literature, it could be surmised that to analyse the piston ring-liner tribological contact, one has to be conduct a dynamic hydrodynamic analysis, involving the solution of Reynolds' equation. Through solving the Reynolds' equation, it is then possible to determine the lubricant film thickness, which is essential in determining the frictional properties along piston top compression ring- liner contact. In this study, mathematical models based on 1-D Reynolds equation assuming Half-Sommerfeld and Reynolds boundary conditions are derived to determine the tribological properties for a piston top compression ring sliding against an engine cylinder liner. Greenwood and Tripp rough surface contact model is used to obtain the friction force of the investigated tribological conjunction.

2.0 MATHEMATICAL MODEL

2.1 Reynolds equation

Reynolds equation (Reynolds, 1886) is commonly used in lubrication models for piston ring-liner analysis in order to determine the contact pressure distribution and the lubricant film thickness. Swift and Stieber (Swift, 1932) used the Reynolds boundary conditions to predict the lubricant film pressure without the consideration of the cavitation effect. The equation is commonly used to study fluid film lubrication of lubricated machine elements such as journal bearings (Nathi 2016a ; Nathi 2016b). In order to predict the frictional properties along the piston ring-liner contact, there are a number of assumptions that have to be considered when using the Reynolds equation to solve for the contact pressure distribution. As an approximation, the piston ring-liner conjunction is considered to be an infinitely long sliding bearing without side leakage. This assumption is valid as long as the ring length to width ratio is over 100 (Chong et al., 2011).

In general, the 1-D Reynolds equation can be defined as the partial derivative equation (P.D.E) for contact pressure distribution of thin viscous film in lubrication theory. The contact pressure distribution of piston ring-liner contact can be predicted using a 1-D Reynolds equation as given below:

$$\frac{\partial}{\partial x}\frac{h^3}{\eta}\frac{\partial P}{\partial x} = 6\left\{\frac{\partial}{\partial x}(U_1 + U_2)h + 2(W_{s1} - W_{s2})\right\} \tag{1}$$

The Reynolds equation given in equation (1) can be further modified into:

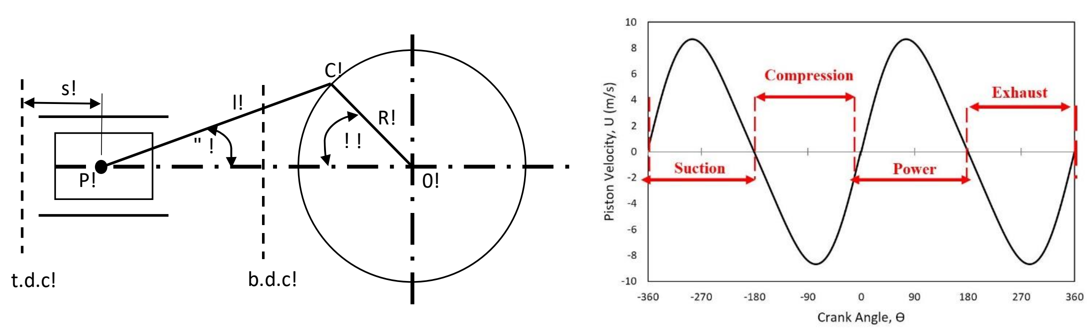
$$h^3\frac{\partial P}{\partial x} = 6U\eta_o h + 12W_s\eta_o x + C_1 \tag{2}$$

Appropriate boundary conditions are needed to solve for the constant of the integration in equation (2). There are two commonly used boundary conditions, namely: Half-Sommerfeld and Reynolds boundary conditions. The profile of the piston top compression ring face is assumed to be a parabola in this study, where lubricant film thickness can be expressed as below:

$$h = h_o + \frac{x^2}{2R} \tag{3}$$

The kinematics of the piston motion during full engine cycle is an essential parameter in solving for the Reynolds equation as discussed above. With the help of the crank mechanism in Figure 2 (a), the piston sliding velocity, which is assumed to be the sliding velocity of the piston ring, can be calculated using equation (4). Figure 2(b) shows the piston velocity when the IC engine undergoes a four-stroke cycle at an engine speed of 2000 rev/min.

$$U_P = R_c\Omega\left[\sin\theta + \frac{1}{2}\left(\frac{R_c}{l}\right)\sin\theta\right] \tag{4}$$



(a) (b)

Figure 2: Kinematics of piston sliding along engine cylinder liner. (a) Crank mechanism with the parameters (Evans and Johnson, 1986) and (b) the piston velocity plotted against the crank angle.

2.2 Friction force

Solving for the Reynolds equation gives only the contact pressure distribution and also the lubricant film thickness for the piston ring-liner contact. In order to predict the friction force along this conjunction, the frictional losses of the piston ring are calculated using the Reynolds solution as an input. The friction force is assumed to have a boundary component (F_b), due to direct surface asperity interaction and a viscous component (F_v), due to lubricant shearing as shown below (Chong et al., 2011):

$$F_f = F_b + F_v \tag{5}$$

The viscous friction force can be computed as:

$$F_V = \int \tau_v \, dA = \sum \tau_v (A - A_a) \tag{6}$$

Where τ_v is the viscous shear stress ($\tau_v = \frac{\eta_o U}{h(x)}$), A is the contact area covered by the piston ring and A_a is the asperity contact area.

The boundary friction force occurs when the piston ring has a minimal lubricant film coverage in the boundary lubrication regime. The boundary shear can be predicted using the classic Eyring model (Evans and Johnson, 1986). Thus, the boundary friction force is given as:

$$F_b = \sum A_a(\tau_b) \quad ; \quad \tau_b = \tau_o + m\left(\frac{W_a}{A_a}\right) \tag{7}$$

Where τ_o is the Eyring shear stress of the lubricant, m is the pressure coefficient of the boundary shear strength and W_a is the load carried by the asperities. The asperity contact area and the load carried by the asperities are predicted in this study using the classical Greenwood and Tripp model (Greenwood and Tripp, 1971), where:

$$\begin{aligned} A_a &= \pi^2(\zeta\beta\sigma)^2 f_2(\lambda) \\ W_a &= A\frac{8\sqrt{2}}{15}\pi(\zeta\beta\sigma)^2\sqrt{\frac{\sigma}{\beta}}E_r f_{\frac{5}{2}}(\lambda) \end{aligned} \tag{8}$$

With the statistical functions f_2 and $f_{\frac{5}{2}}$ calculated using the polynomial approximation proposed by Teodorescu et al. (2003):

$$\begin{aligned} f_2 &= -\frac{1}{10^4}(18\lambda^5 - 281\lambda^4 + 1728\lambda^3 - 5258\lambda^2 + 8043\lambda - 5003) \\ f_{\frac{5}{2}} &= -\frac{1}{10^4}(46\lambda^5 - 574\lambda^4 + 2958\lambda^3 - 7844\lambda^2 + 776\lambda - 6167) \end{aligned} \tag{9}$$

Therefore, the total friction force along the piston ring-liner contact is:

$$F_{total} = \int (F_b + F_v)\, dx \cdot L \tag{10}$$

Lubricant properties, engine parameters, piston ring geometry and friction model parameters are shown in Table 1-3.

Table 1: Lubricant properties (SAE5W30@120°C) (Reynolds, 1886).

Parameters	**Values**
η_0	0.00689 $Pa.s$

Table 2: Engine parameters and piston ring geometry (Styles et al., 2014).

Parameters	**Values**
Crown Height, c	14.9 μm
Crank Radius, R_c	0.04 m
Piston Radius, R	0.01825 m
Connecting Rod Length, l	0.1419m
Bore Diameter, D	0.0889 m
Engine Speed, N	2000 rev/min
Ring Tension, σ_r	0.341MPa

Table 3: Friction model parameters (Swift, 1932).

Parameters	**Values**
σ	0.37 μm
m	0.08
τ_o	2.0 MPa
$\zeta\beta\sigma$	0.055
σ/β	0.001

3.0 RESULTS AND DISCUSSION

During engine operation, the piston ring will be pressed onto the liner through the ring tension and the combustion pressure acting behind the piston ring. The ring tension is designed for effective sealing of the combustion chamber in reducing possible combustion gas blow-by. The pressure acting behind the piston ring is from the combustion gas pressure during the spark ignition of air-fuel mixture.

The combustion pressure has a significant contribution to the contact load along the piston ring-liner contact. However, there is no exact mathematical formula to determine the combustion pressure in the full engine cycle. This is beyond the scope of this study and it requires a full analysis of all in-cylinder chemical phenomena. Therefore, the combustion pressure measured by Mishra et al. (2008) is taken for this study and is shown in Figure 3. This pressure will be used as an input together with the ring tension in the current study. The contact load, W along the piston top compression ring can be computed as:

$$W = f_{ring} + f_{comb} = (\sigma_r + P_{comb}) \times t_r L \quad (11)$$

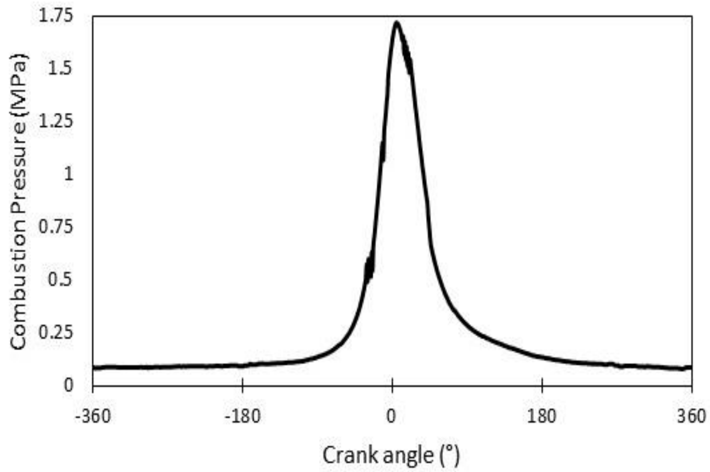


Figure 3: Combustion pressure at 2000 rev/min (Mishra et al., 2008).

For a full engine cycle, the findings of the minimum lubricant film thickness are essential because these could attribute to significant friction force values. Figure 4 shows the minimum lubricant film thickness for a full engine cycle while the highlighted region at A and B are the area of interest within this study. It is interesting to note that the minimum lubricant film thickness predicted by the Half-Sommerfeld boundary condition follows more closely as compared to the literature data solved using the modified Elrod solution, especially along the mid-stroke regions (higher film thickness regions).

The minimum lubricant film thickness computed above has a direct correlation with the friction force magnitude. Decreased lubricant film thickness will result in the increase of friction force along the piston top compression ring-liner contact. In this study, the friction model based on Greenwood and Tripp rough surface contact model is used to predict the frictional losses along the piston ring-liner conjunction. Figure 5 shows the boundary friction and the viscous friction components experienced by the piston ring-liner contact for the full engine cycle.

In Figure 5, it can be observed that the boundary friction is as high as 101 N while the viscous friction force is an order smaller than the boundary friction values. It is also to note that the viscous friction predicted using the Half-Sommerfeld and Reynolds boundary conditions are almost twice higher than the one provided by literature data. This is expected because boundary friction involves the interaction between asperities from opposing surfaces, which increases significantly the frictional losses along the ring-liner contact, possibly leading to higher wear.

Figure 6 shows the total friction force for the ring-liner contact along the suction stroke BDC (location A). It can be observed that the predicted friction forces using the mathematical models proposed within this study follows closely the trend obtained from literature data. The peak friction predicted by the Half-Sommerfeld boundary condition is observed to be closer to the literature data. However, overall, the Reynolds boundary condition is observed to correlate better to the literature data in terms of total friction force, where it follows more closely the frictional losses given by the literature data. Similar observation can be surmised along the power stroke TDC (location B).

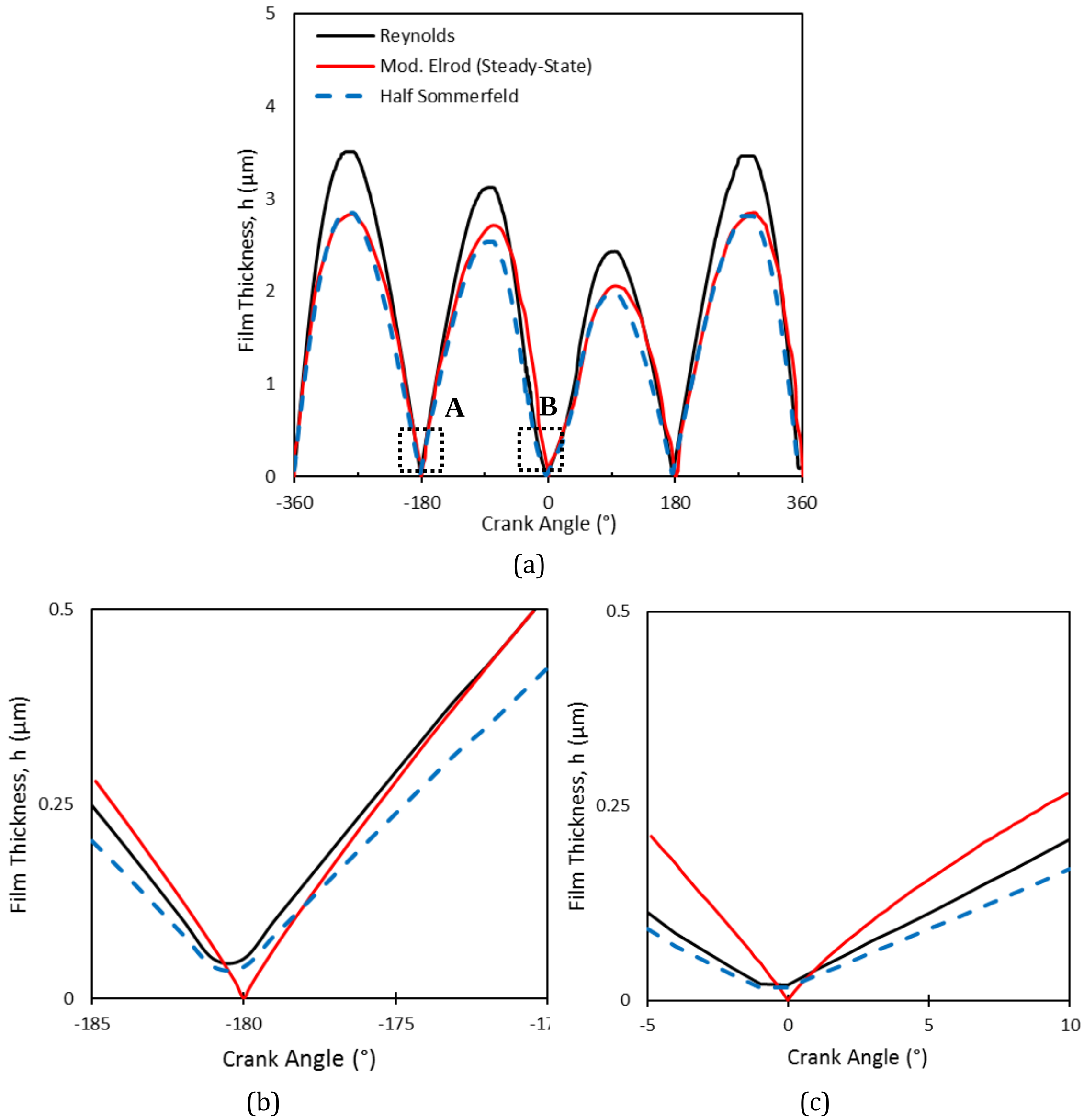
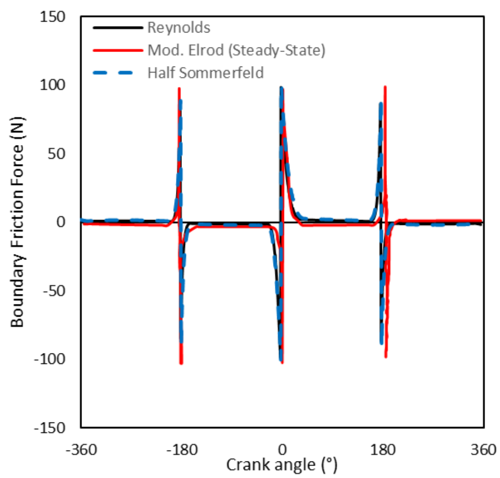
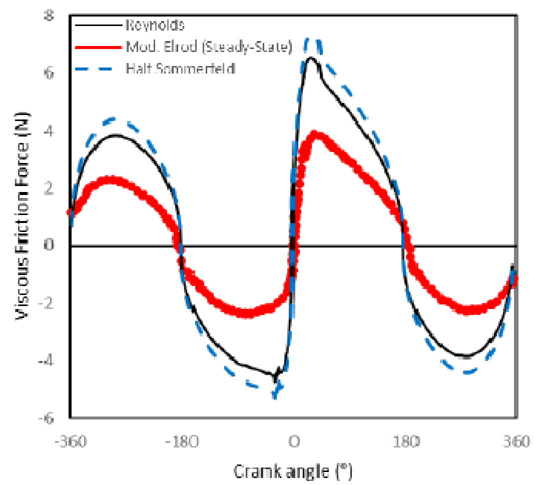


Figure 4: Minimum lubricant film thickness in an engine cycle. (a) Full engine cycle, (b) In the vicinity of the suction stroke BDC (location A), (c) On the vicinity of the power stroke TDC (location B).





(a) (b)

Figure 5: Boundary and viscous friction force at 2000 rev/min. (a) Boundary friction force. (b) Viscous friction force.

CONCLUSION

In this study, 1-D Reynolds equation is solved analytically using the Half- Sommerfeld and the Reynolds boundary conditions for a piston top compression ring sliding against the engine cylinder liner. Half-Sommerfeld boundary condition predicts minimum lubricant film thickness that correlates better with the literature data. Greenwood and Tripp rough surface contact model is used to consider for surface asperity contacts especially along the dead center regions. The friction force predicted by the mathematical model taking Reynolds boundary condition along the dead center correlates better when compared with the literature data. Overall, it can be concluded that the Reynolds equation assuming Reynolds boundary predicts better frictional forces along the piston top compression ring-liner contact, especially where frictional losses are most critical and dominated by boundary friction, which is at dead center along the engine cylinder liner.

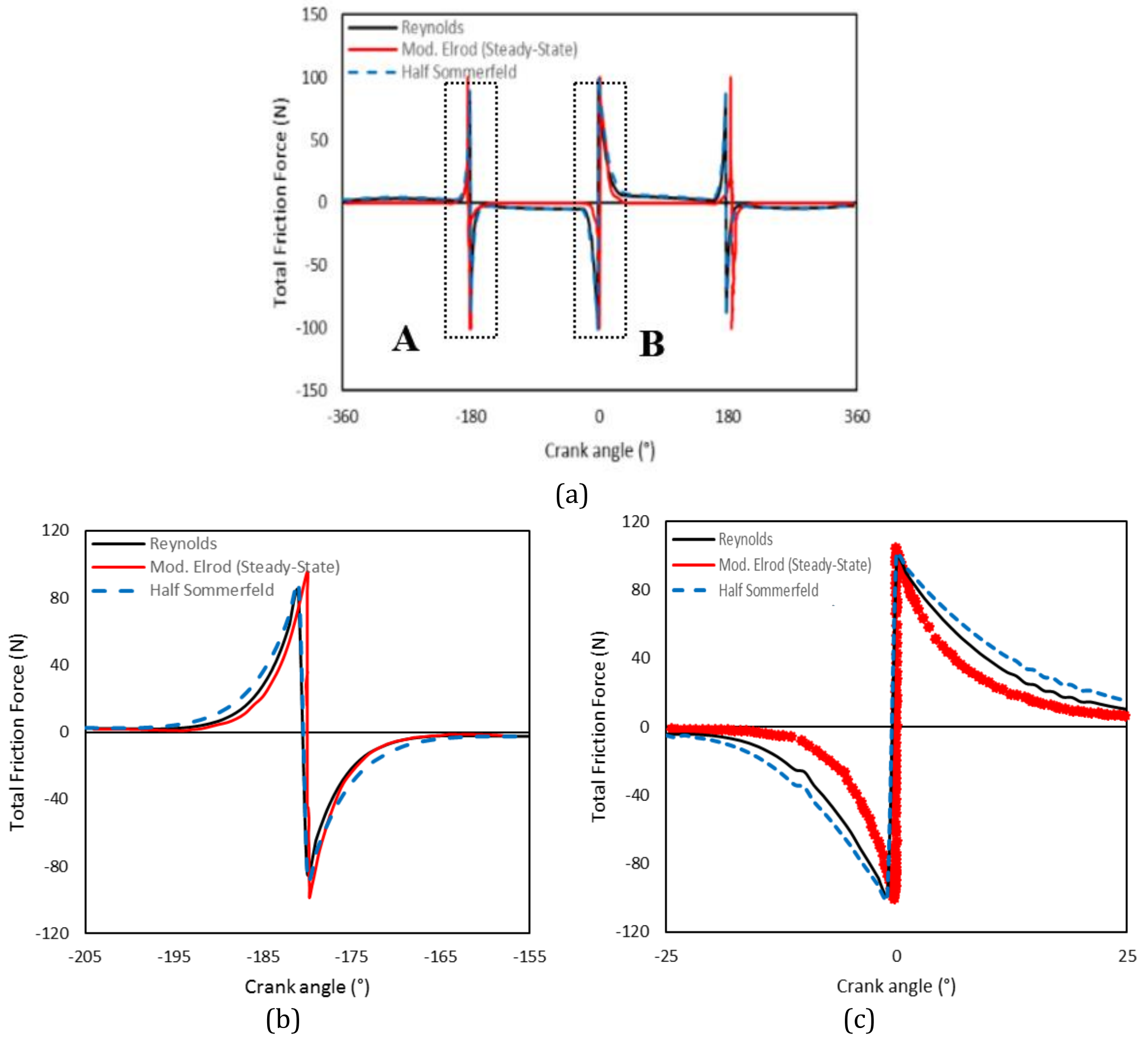


Figure 6: Total friction force at 2000 rev/min. (a) Full engine cycle, (b) In the vicinity of the suction stroke BDC (location A), (c) On the vicinity of the power stroke TDC (location B).

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