

Piston Friction Reduction by Reducing Piston Compression Height for Large Bore Engine Applications

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ABSTRACT

Improving engine efficiency and reducing the total cost of ownership demands engine friction loss reduction through optimal design, especially for large bore application considering the amount of fuel the engine consumes during its service life. Power cylinder is a big source for engine friction and piston accounts for about 25% to 47% of the power cylinder friction [1]. Thus the piston design needs to be optimized to minimize friction; and at the same time, not sacrificing the durability. This work focuses on piston friction reduction by utilizing shorter compression height piston for large bore engine application through analytical simulation study. From the simulation study, 12.5% friction reduction has been achieved in the piston skirt to liner interface for the shorter piston with longer connecting-rod compared to the baseline design.

KEYWORDS: IC engine, Power cylinder, Piston modeling, Dynamics, Friction, Asperity contact, Hydrodynamic contact

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INTRODUCTION

Progressive efficiency improvement has been achieved for internal combustion engine development through the past years. Among all the efficiency improvement technologies, reducing friction loss is one way through optimal engine design and use of lower friction components. Power cylinder accounts for about 50% of the total engine friction loss. Within in the power cylinder, piston itself account for 25% to 47% of the loss, which is a significant amount of the total engine friction loss. Piston manufacturers are working on piston design optimization to reduce piston friction loss. With the application of steel piston through friction welding or laser welding, the piston compression height can be further reduced. It is believed that the shorter piston with longer connecting-rod can benefit in friction reduction as it reduces the piston side load, which in turn, helps reduce the friction between the piston and the cylinder liner.

Power cylinder modeling including piston and ring pack has been studied by researchers using both analytical simulation method and experimental measurement [2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26] to better understand the friction mechanism and to reduce it. Marco Maurizi and Daniel Hrdina showed 3 to 5% fuel reduction was achieved for a passenger car engine application by using shorter piston and longer connecting rod [27].

Simulation method has progressed significantly in engine modeling and become the standard approach in new engine development and trouble shooting. It is even more beneficial for this method considering the cost and time efficiency comparing to engine testing.

This study discusses the friction reduction benefit by utilizing a shorter comprehension height piston for a high-speed large bore 4-stroke engine application.

MODEL DESCRIPTION

As the peak cylinder pressure keeps increasing in large bore engine application, piston technology is being improved. This include aluminum piston, cast iron piston, steel piston, friction welded or laser welded steel piston to increase piston strength considering the severe operating condition it is working under. The piston is usually designed tall enough to withstand the high in-cylinder pressure load. With higher strength in steel, steel piston has become the trending choice for large bore engine application. When meeting the load requirement, steel piston can be also design shorter as compared to aluminum or cast iron pistons.

As the piston compression height is reduced, in order to use the same engine block, the connecting rod length needs to be increased such that the distance between piston top and the connecting rod big end center does not change.

In this study, the piston compression height has been reduced by over 20% for the shorter piston and the connecting rod length is increased accordingly. Associated with the height and length change, the mass of the components also change. As a result, the overall piston-pin-connecting rod assembly mass is reduced by over 4% for the shorter compression height piston case through optimal design.

A multi-body dynamics model was built including piston, piston pin, connecting rod and liner. The contact between piston skirt and the liner ID surface is modeled using an EHD joint.

The contact consists two parts: the first part is the hydrodynamic contact with oil film separating the two mating surfaces, in this case, the piston skirt and liner; the other part is the asperity contact, which is considered as metal to metal contact with the clearance between the surfaces in the surface asperity height region.

For the asperity contact, the well-known Greenwood-Tripp model is used. When calculating the asperity friction, the model considers the “lubricated asperity friction and wear (LAFW)” model developed by AVL that enables different coefficient of friction in the boundary friction regime as shown in the Stribeck curve as shown in Fig 1 [32, 33]. In comparison, the traditional approach is having a constant coefficient of friction for the entire boundary friction zone.

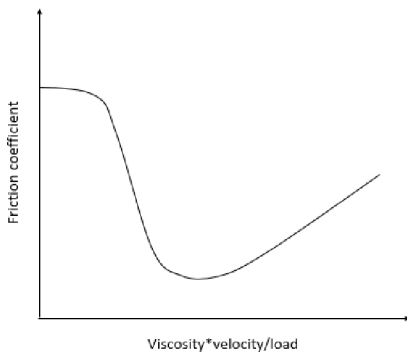


Fig. 1. Stribeck curve

The coefficient of friction depends on the local material and surface properties.

$$\mu_{LAFW} = \mu_0 \cdot A^{-\sqrt{B \cdot L_N}} + C \cdot r_c \cdot L_N \cdot (1 - A^{-\sqrt{B \cdot L_N}})$$

Where A, B and C are parameters and L_N is the characteristic length.

Hydrodynamic contact is not affected by the above LAFW relationship and is modeled by solving the average Reynolds' equation. Oil film thickness and the oil pressure distribution between the skirt and liner contact region can be obtained.

In this study, SAE 15W40 is used as the lubrication oil and the viscosity vs temperature relationship can be found in the Fig 2.

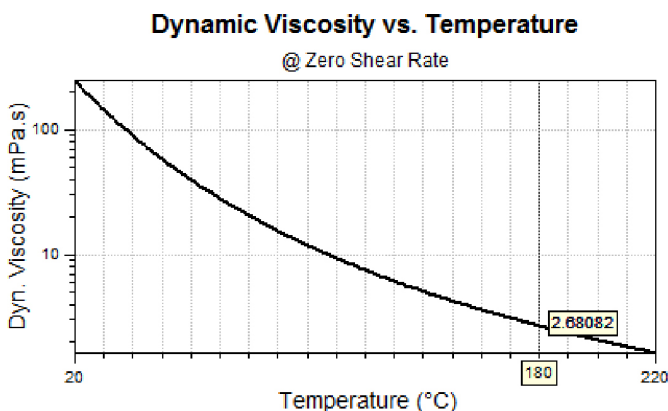


Fig. 2. Oil dynamic viscosity temperature relationship

Considering the complexity of the model and making sure it can be solved in a timely manner, all other joints were modeled in a simpler way using non-EHD joint, including piston pin bore-pin, pin-conrod small end, conrod big end-crank pin.

RESULTS

Due to the confidentiality, only relative normalized results are presented in this work.

As previously mentioned, the piston lateral force can be reduced by using shorter compression height piston pairing with longer connecting rod, without changing the engine block height. Under the same pressure load on the piston, the lateral reaction force from the cylinder liner on the piston can be calculated as (Fig 3):

$$R = F \cdot \tan(\alpha)$$

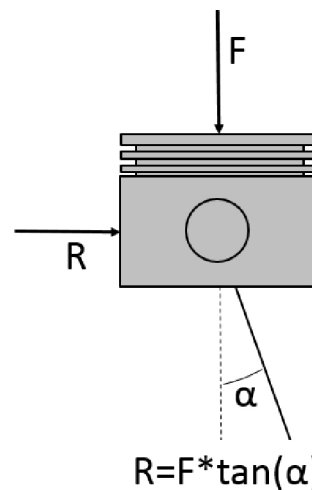


Fig. 3. Piston side load

This lateral forces for both baseline and shorter piston over a cycle can be found in Fig 4. In Fig. 4 and other figures shown later, the 0 crank angle is always the TDC firing condition.

The highest contact force is found around 30 degrees crank angle after TDC firing condition, where the cylinder pressure is still high and the relative angle between the connecting rod and cylinder center line becomes significant for piston side load.

The result shows that 7.3% reduction in the peak lateral force has been achieved for the shorter compression height piston over the baseline. This is believed to be beneficial for friction reduction for the piston.

The total piston friction loss over a cycle can be found in Fig 5. The instantaneous friction loss is reduced significantly during the expansion stroke, especially from 25 to 50 crank angle after TDC firing. This is in good agreement with piston lateral force in Fig 4.

However, in addition to the lateral force, the contact between the skirt and liner, which affects the friction loss, is also dependent on piston secondary motion. Thus the friction loss is not proportional to the

lateral force. For the rest of the expansion stroke and other strokes, the friction loss for shorter compression height piston is only marginal lower than the baseline case.

From the total friction power loss plot in Fig 5, the shorter piston configuration shows 12.5% reduction over the entire engine cycle for the piston skirt. This does not include other frictions from piston pin, crank shaft, etc.

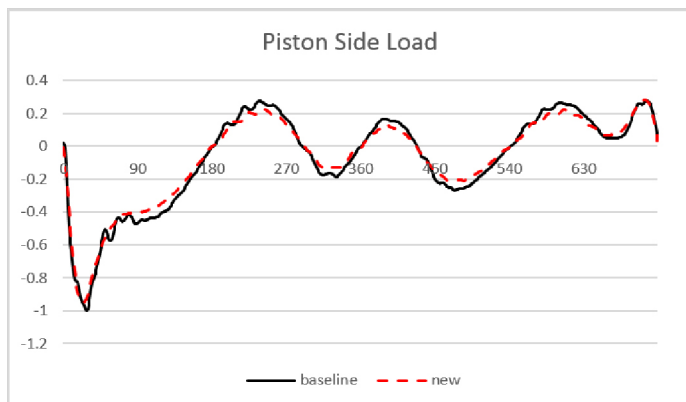
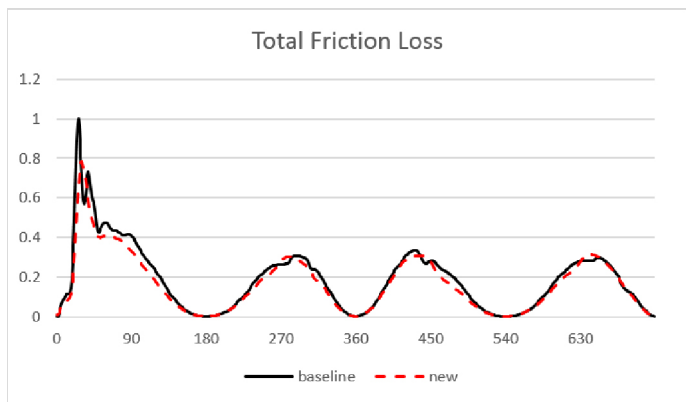
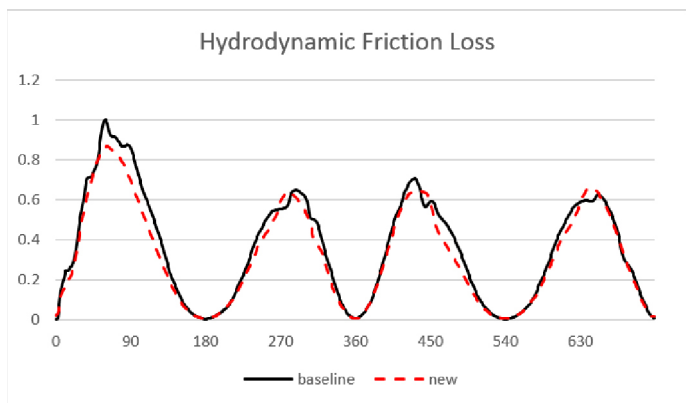


Fig. 4. Piston side load over a cycle

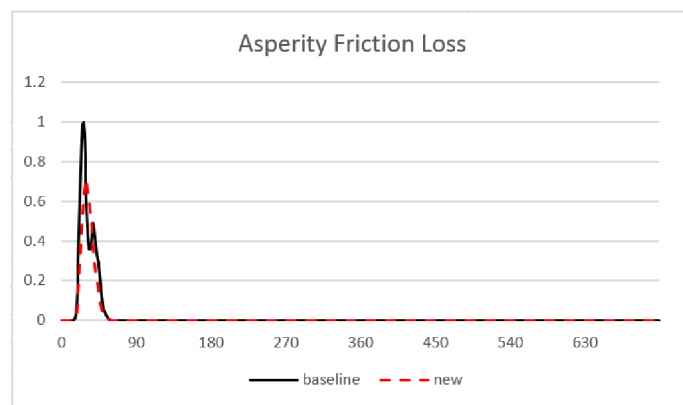


a. Piston total friction loss



b. Piston hydrodynamic friction loss

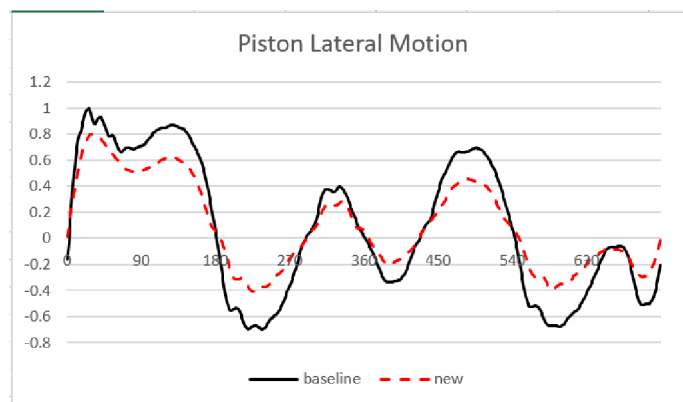
Fig. 5.



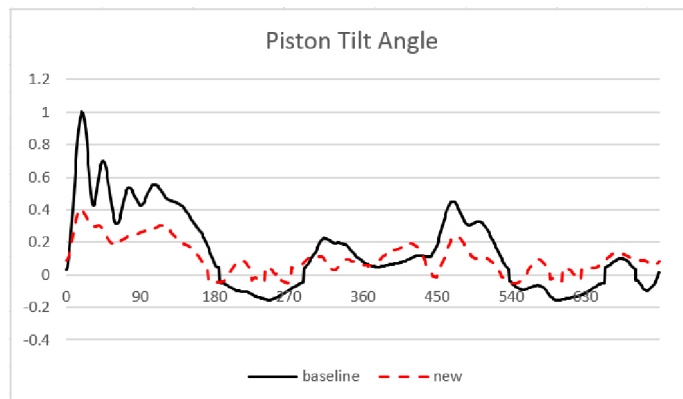
c. Piston asperity friction loss

Fig. 5. (cont.) Piston friction loss over a cycle, (a) total loss; (b) hydrodynamic loss; (c) asperity loss

As mentioned in the previous paragraph, the piston secondary motion also influences how the piston skirt interacts with the liner. The skirt profile needs to be optimized to well guide the piston. In this study, both pistons don't show top land contact. Secondly, the skirt is desired to promote the oil film build-up and eliminate the metal to metal asperity contact.



a. Piston lateral motion



b. Piston tilting angle

Fig. 6. Piston secondary motion