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Part 3: A Study of Friction and Lubrication Behavior for Gasoline Piston Skirt Profile Concepts

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ABSTRACT

This paper deals with the friction performance results for various new concept piston skirt profiles. The program was conducted under the assumption that friction performance varies by the total amount of oil available at each crank angle in each stroke and the instantaneous distribution of the oil film over the piston skirt area.

In previous papers [1,2] it was that lower friction designs would be expected to show higher skirt slap noise. This paper discusses the correlation between friction and skirt slap for each new concept profile design.

Finally, this paper explains the friction reduction mechanism for the test samples for each stroke of the engine cycle by observing the skirt movement and oil lubrication pattern using a visualization engine.

INTRODUCTION

The piston skirt design is one of the most important features within the reciprocating engines power cylinder assembly for the control of friction performance. Therefore, researchers have conducted intensive work and effort on reduction of skirt friction. Of several key features which affect piston friction, piston skirt profiles were selected for this study. Various new conceptual skirt profiles were designed and test samples were made using a special CNC machine tool.

This research has focused on 1) rating the friction and noise performance of each new conceptual profile with comparison against a conventional profile setup and 2) clarifying the cause and mechanisms that contribute to friction and noise by observing oil film behavior through the cylinder bore window of a visualization engine.

TEST ENGINE AND TEST PROCEDURE

Friction and noise measurement

To measure skirt friction and noise, a single cylinder floating liner type engine was used, Tables 1 and 2 show the main specifications of the test engine and the piston respectively. Figure 1 shows the geometry of the piston rings. After full break-in, the same piston rings and cylinder were used for each test.

Table 1. Main specifications of test engine

Engine Type	Single cylinder,4 cycles SI gasoline engine
Displacement	0.4993 L
Bore x Stroke	86 mm x 86 mm
Compression Ratio	10:1
Crank ratio (L/r)	3.5
Crankshaft offset	0 mm
Piston cooling	Oil jet

Table 2. Piston Specifications

Piston alloy	Hypereutectic alloy
Pin offset	0.5 mm to thrust side
Skirt roughness	18Rz
Skirt treatment	No coating
Piston length	53.9 mm

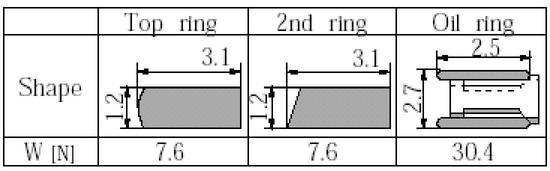

Figure 1. Geometry of piston rings

Five different skirt profiles (patent pending) were selected for the investigation. Each sample was run with a 30 μm piston to bore cold clearance for the friction and noise measurement. Each profile variant is shown in Figure 2. The blue (dark) area of each piston profile represents the area is recessed by 35 to 40 μm.

Figure 2. Skirt profile configuration of test pistons

Observation of oil film formation

After measuring friction and noise using the normal floating liner engine, oil film formation was observed through a sapphire window liner version of a single cylinder research engine (Figure 2). For the purpose of this investigation, each piston sample was made with 50 μm piston to bore cold clearance. The skirt surface was phosphate treated to provide a dark surface for imaging: a thin oil film thus appears dark while a thicker film appears lighter. The principles of the visualization measuring device and image data processing method have been introduced previously [2].

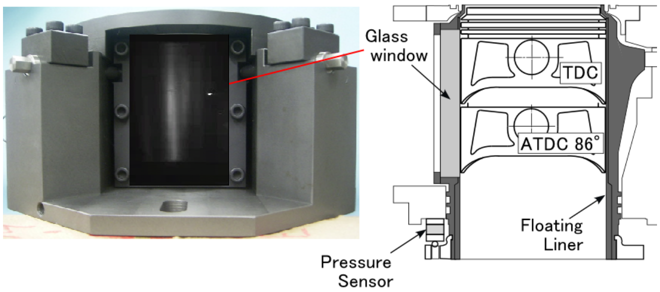

Figure 3. Liner window for simultaneous piston/ring friction and oil film measurement

Table 3. Test conditions

Engine Speed (rpm)	BMEP [kPa]		
	380	500	630
1500	1	2	3
2000		4	
2500		5	

1-5: Measurement order

Table 3 shows the engine test conditions. The engine oil used during each test was SL GF-3 5W-30.

After repeating the break-in schedule four times, the fifth set of friction data was used for judging the friction performance of each profile.

Engine Temperature Control

The cold state condition, used to measure piston noise, had a liner temperature set at 50°C by controlling coolant flow. Oil temperature was set at 20°C at start and was allowed to rise to 30°C at the end of each test.

A hot state test was used to measure friction. The liner temperature and oil temperature were set at 100°C and 85°C respectively.

TEST RESULTS

Friction measurement

Friction measured at the five different load and speed conditions (Table 3) showed similar trends. Figure 4 shows a typical friction force diagram.

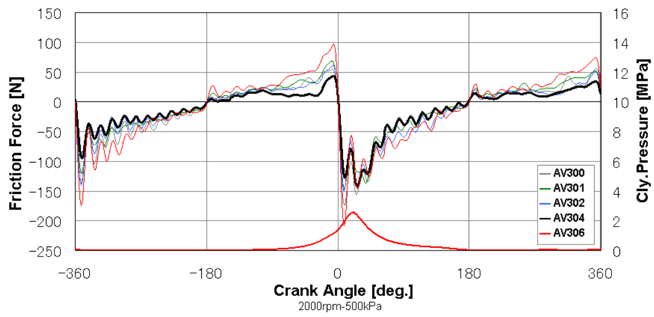

Figure 4. Friction force at 2000 rpm-500 kPa BMEP

Intake stroke: The oil is commonly introduced from the bottom of piston skirt and spreads up the skirt due to the motion of oil relative to the piston [2]. The intake stroke is not usually a focus for friction reduction; however, the test results show that there is an opportunity to lower friction during this stroke. Figure 5 shows the friction force diagram and a summary of friction loss in joules.

Figure 6 shows the oil spreading pattern at an early stage of the intake stroke. It is apparent that friction is dominated by the amount of oil retained on the piston skirt during this period.

The conventional profile, AV300 and profile AV306 show the highest friction. For profile AV300, oil is fed in through the relatively small open-end clearance. The AV306 piston was designed with an '**X**' shaped skirt bearing surface, which was designed to reduce contact area. Although it has a recess on the bottom of the skirt, oil is blocked from spreading to the area where lubrication is needed.

For profiles AV301and AV302, oil flows through the gap, but cannot flow into the non-recessed skirt contact area, resulting in hard contact.

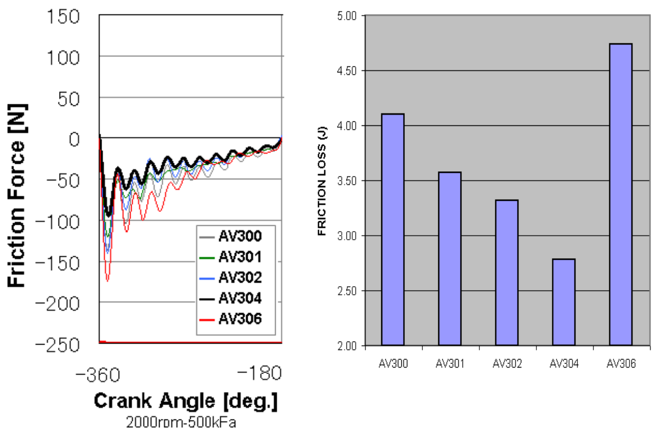

Figure 5. Friction force diagram and friction loss in Joules during intake stroke

AV304 shows the lowest friction during the intake stroke. It is observed that when the intake stroke begins, oil is collected through the recessed area formed at the bottom of the skirt. It then spreads onto the main load bearing area in the middle of the skirt. The whole skirt surface is flooded with oil throughout the stroke. The recessed area retains oil throughout the stroke.

After most of the oil passes the middle of the skirt, towards the end of the stroke, there is little difference in friction between the profile types.

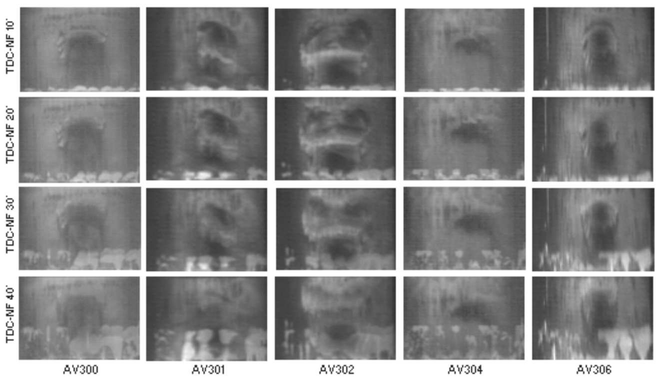

Figure 6. Oil spreading pattern during intake stroke

Compression stroke: Oil moves down the piston during the compression stroke. It was concluded that there is a relationship between piston friction and skirt oil film thickness [2]. Figures 7 and 8 show the friction force diagram and oil spreading pattern for this stroke.

Oil retained in the recessed area formed on AV302 is spread to the skirt bearing surface when the piston reaches top dead center. Oil flow to the bottom of the AV306 skirt is blocked by the non recessed areas. So, at TDC, very little oil remains on the bottom of the skirt and the friction force spike is the highest. This spike correlates with the friction power loss trend for the stroke as shown in the bar chart (Figure 7).

AV304 retains a lot of oil throughout the stroke. Oil retained in the reservoir formed in the upper skirt provides lubrication to the middle and bottom of the skirt as the skirt begins to tilt toward the non-thrust side.

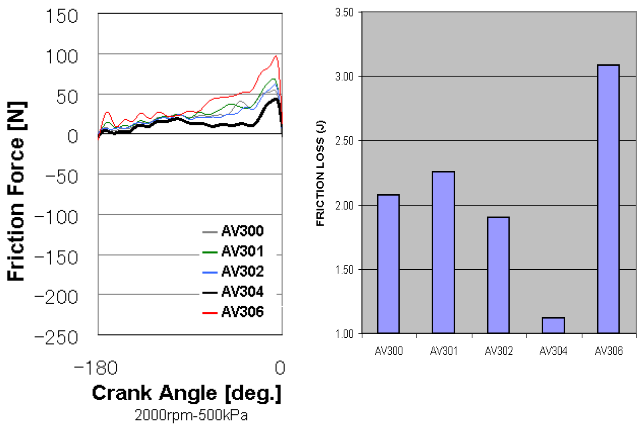

Figure 7. Friction force diagram and friction loss in Joules during compression stroke

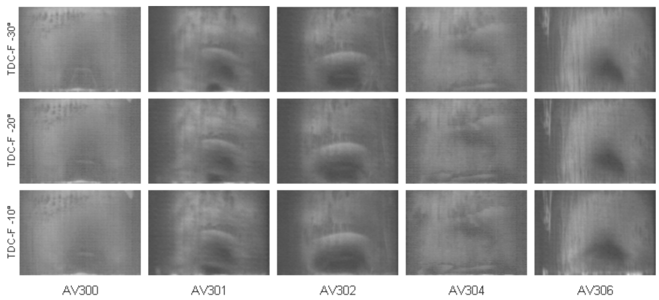

Figure 8. Oil spreading pattern during compression stroke

Expansion stroke: Figures 9 and 10 show friction force diagrams and oil spreading patterns for the expansion stroke. Friction during the expansion stroke is the highest of the four strokes. In addition to friction reduction, the avoidance of skirt scuffing should be also considered.

The test results do not show as large a variation in friction during the expansion stroke as originally expected. However it is possible to rate the friction performance for each profile by observing the oil film behavior. The key to reducing friction in this stroke is to utilize the abundant lubricating oil that is available below the skirt. As stated in the ‘Intake stroke’, oil enters the bottom of the skirt and spreads up the skirt.

<u>AV300 profile</u>: The conventional piston profile was optimized to have ideal skirt contact. As shown in Figure 8, the piston starts moving downwards with a limited

amount of oil retained on the skirt surface. The oil is rapidly squeezed out towards the top and sides of the skirt. The bottom center of the skirt blocks upward oil flow due to spring-back against the bore surface. Oil is evenly spread over the entire skirt surface area. This profile is rated with a moderate level for friction performance. The top of the skirt has a very thin oil film but a large quantity of oil is collected above this hard contact zone.

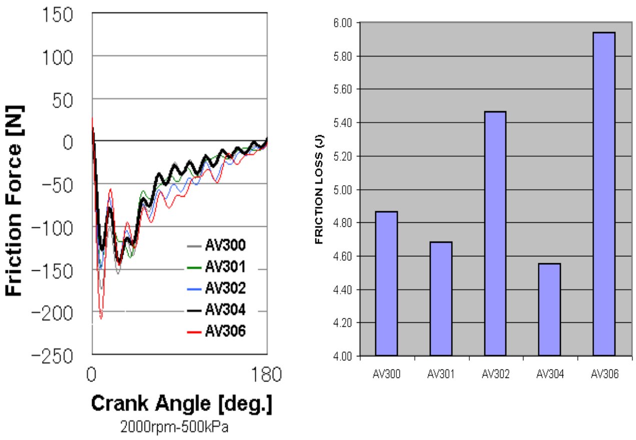

Figure 9. Friction force diagram and friction loss in Joules during expansion stroke

AV301 profile: Oil is fed in from the recesses formed at the bottom of the skirt throughout the stroke. Oil is flowing through the '^' shaped skirt bearing area. Side load is supported by the limited surface area. As a result, oil is squeezed out while the area is lubricated by the previously described oil flow mechanism. The friction with AV301 profile is the 2nd lowest due to consistent lubrication on main contact area throughout the engine stroke. The recess at the top of the skirt collects a lot of oil which passed through the '^' shaped skirt bearing zones. It is also observed that oil is retained in the valleys between these bearing zones. At the end of the stroke, there is no difference in friction compared with other profile designs.

AV302 profile: Oil feeding from the lower skirt is blocked due to the pad positioned low and in the center of the profile. Although this design has the advantage of contact area reduction, the load is supported by this limited area, resulting in higher contact pressure. Oil supply to these contact zones is not sufficient. The two upper pads start to support the side load just after the lower pad contact begins. Lateral movement at top of the skirt is distinctive. Three areas of contact can be clearly seen (Figure 10). Friction with this profile is ranked as the 2nd highest. It is thought that friction performance with this profile type can be adjusted by controlling the pad contours and locations.

AV304 profile: A recess formed at the bottom edge is designed to introduce, retain and provide more oil faster to the main contact area. As shown in Figure 10, the stroke starts with plenty of oil on the entire skirt surface, partly due to oil retained in the oil reservoir at the top of the skirt. It is observed that oil continues to flow to the middle of the skirt through the bottom recess throughout the stroke. As the skirt is loaded by the side force, the main contact area is fully lubricated by the oil collected in the lower recess. More oil is introduced as the piston speed increases. This design shows the lowest friction during the expansion stroke.

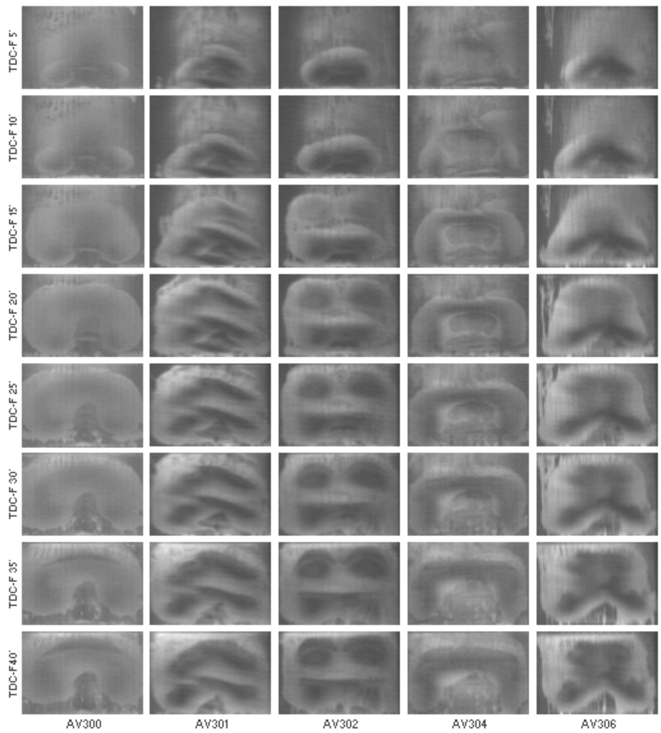

Figure 10. Oil spreading pattern in expansion stroke

AV306 profile: This profile, which has a '**X**' shaped bearing area, shows the highest friction. As described above, just after the stroke starts, oil penetration to the skirt is blocked due to the '**X**' shaped feature at the bottom. Test results and observation of the oil film formation pattern suggest that minimizing the skirt contact area does not contribute to lowering friction at least for expansion stroke. It is interesting that oil collected at the bottom center provides oil to the skirt center throughout the stroke whereas oil on the other contact areas is squeezed out, resulting in oil starvation. The center of the piston is lubricated by oil penetration as a result of the wedge effect of the lower '^' shaped feature.

Exhaust stroke: At the early stage of the exhaust stroke, there is plenty of oil at the top of skirt, collected by oil ring scraping and piston movement during the previous stroke. This oil starts to move downwards to cover the piston skirt surface. Any protrusion on the profile tends to disturb the oil flow. As a result, the amount of oil coverage on the skirt contact surface depends on the type of skirt profile.

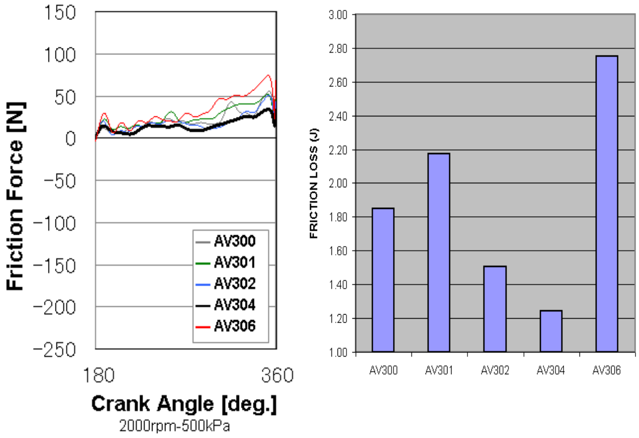

Figure 11. Friction force diagram and friction loss in Joules during exhaust stroke

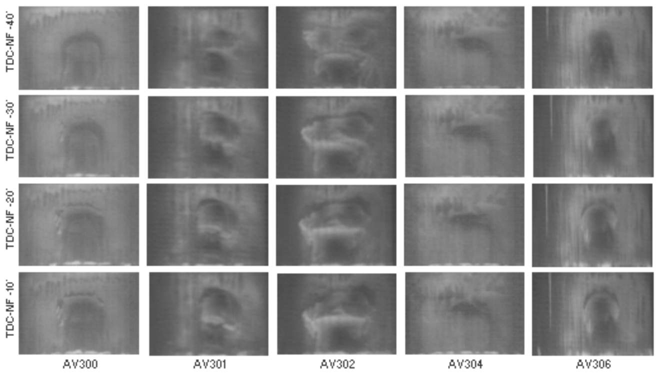

Figure 12. Oil spreading pattern in exhaust stroke

<u>AV300 profile</u>: Oil is spread towards the bottom of the skirt through the gap between the cylinder bore and the skirt surface. The oil film is observed to be thin but evenly spread. The center of the skirt has a relatively small amount of oil because the bottom center of the skirt blocks upward oil flow due to “spring-back” against the bore surface.

<u>AV301 and AV306 profiles</u>: Oil flow is blocked due to the ‘ʌ’ or ‘**X**’ shaped protruded surface regions. As shown in Figure 11, the AV301 and AV306 pistons displayed the highest friction values during this stroke.

<u>AV302 profile:</u> Oil flow is partially disturbed by the size and location of the pads. It is, however, observed that the lower pad receives more oil as the piston reaches top dead center. The AV302 piston shows the second lowest friction.

<u>AV304 profile:</u> It is observed that the oil reservoir at the top of skirt retains a lot of oil throughout the exhaust stroke, even though oil keeps flowing down to the lower part of the skirt. The oil spreading pattern of AV304 is quite different from other profile types. The skirt is fully flooded with oil and oil is flowing down through the lower recess without disturbance.

TEST SUMMARY

EFFECT OF ENGINE LOAD

The effect of engine load for AV304 was analyzed. Figure 12 compares the friction performance for AV304 measured at three different engine load conditions.

As the engine load increases, the thin oil film area increases due to the higher thrust load (this characteristic is visually confirmed in Figure 10), also reported on in [1, 2]. Also, the oil film becomes thinner and the friction force increases. The same trend is observed for the other profiles.

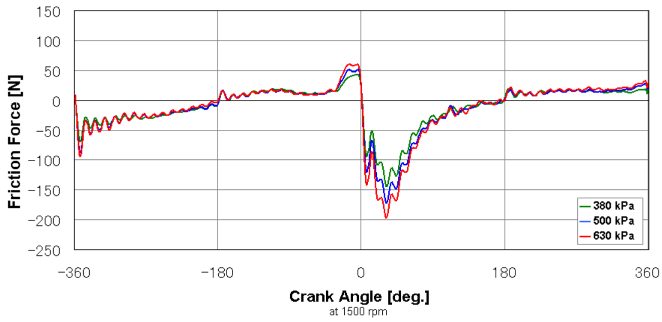

Figure 13. Effect of engine load for AV304 Profile

EFFECT OF ENGINE SPEED

From the end of compression stroke to 40° after TDCF, AV304 friction shows a different pattern from the other profiles. It is well known that, as engine speed increases, friction force during the expansion stroke increases. However, AV304 friction decreases with increased engine speed. As mentioned earlier, more oil is introduced as the piston speed increases due to the oil recess at the bottom of the skirt.

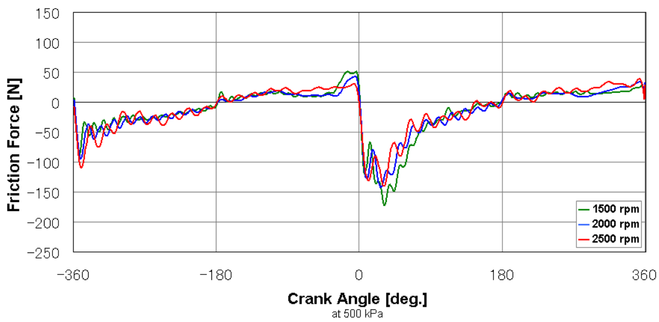

Figure 14. Effect of engine speed for AV304 Profile

EFFECT OF TYPE OF SKIRT PROFILE ON FRICTION

Figure 15 is a summary of friction performance for the selected test piston profiles. AV304 profile shows the lowest friction throughout the engine cycle for all five test conditions.

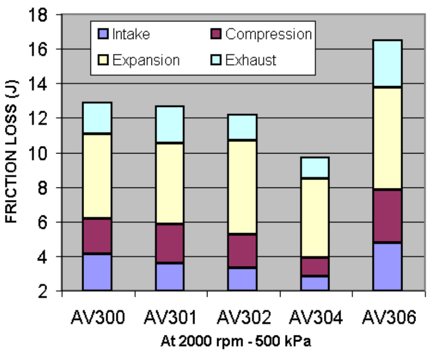

Figure 15. Effect of type of skirt profile on friction

EFFECT OF TYPE OF SKIRT PROFILE ON NOISE

Cylinder block vibration due to piston slap was measured with an accelerometer, at each of the five load and speed conditions, in the cold state. Measured block acceleration data was converted into frequency domain using the Fast Fourier Transform method (FFT). The characteristic skirt slap frequency was identified as 1.5 kHz [1].

Figure 16 ranks the noise performance for each skirt profile variant. AV304 is remarkably quieter than the other profile variants. It has been observed from a series of these and other engine tests that skirt slap noise is related to the hydrodynamic lubrication condition. Oil is also known to have a noise damping effect.

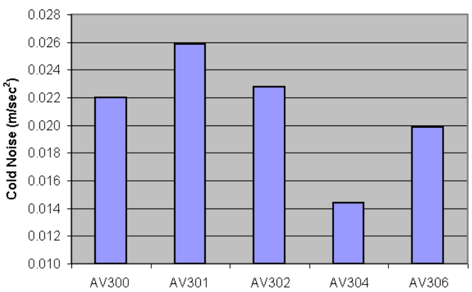

Figure 16. Ranking of skirt slap noise by profile type

CONCLUSION

1. The friction test results for each engine stroke have been well correlated with the observed oil spreading pattern for each profile concept.
2. Oil supply timing and the amount of oil retained on piston skirt dominate skirt friction and slap noise.
3. A uniform oil spreading pattern without oil flow disturbance is very important in reducing piston skirt friction.
4. Friction reduction can be achieved by proper management of the vertical movement of oil relative to the piston.
5. Friction measurement results show that there is an opportunity to reduce friction during the non-firing strokes.
6. AV304 profile, displaying recesses at the top and bottom of the skirt, shows the lowest reported friction results. A unique oil spreading pattern is observed throughout the engine cycle for this variant.
7. Reduction in skirt contact area is not always an effective way to reduce skirt friction. Oil can be squeezed out of the bearing area, resulting in oil starvation.
8. It is confirmed from all the profile tests that, as engine load increases, piston friction increases.
9. As engine speed increases, the piston skirt friction of AV304 decreases whereas the friction of the other profiles increases. This is because AV304 has better lubrication of the load bearing surface.
10. These conceptual skirt profiles were selected for research work. Further verification studies on production feasibility, oil consumption, blow-by, durability and scuffing are required.

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REFERENCES

1. Madden D. et al.,' Part 1: Piston Friction and Noise Study of Three Different Piston Architectures for an Automotive Gasoline Engine 'SAE Paper No.2006-01-0427, 2006'
2. Kwang-soo K. et al.,'Part 2:The Effects of Lubricating Oil Film Thickness Distribution on Gasoline Engine Piston Friction' SAE Paper No. 2007-01-1247'