技術論文

Friction Measurement of Al-17%Si Monolithic Cylinder with using Newly Developed Floating Liner Device

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要旨

ピストン-シリンダ系はエンジンフリクション低減において重要な役割を果たしている。同系の摩擦挙動を改良するためには、エンジン運転中の摩擦波形を観察・解析することが効果的である。上記要求を満たすため、新たに開発した浮動ライナー装置を用いて摩擦波形を測定した。新開発の浮動ライナー装置における計測は空冷110cc単気筒エンジンをベースとして、過共晶Al-17%Si合金製ダイカストシリンダ(DiASil)、軽量鍛造ピストンおよびDLCピストンリングを使用して行われた。本装置にて観察された摩擦波形は理論的に予測される波形と合致しており、適正に計測がなされていると判断された。各条件の平均摩擦有効圧(FMEP)の値を算出した結果、FMEPはエンジン負荷増加とともに増加することが分かった。これはピストンサイドフォースの増加に起因すると考えられる。さらに慣らし後のFMEPは慣らし前よりも30%減少した。この現象は接触部表面の粗さの変化に加えてトライボフィルム形成も寄与しているためと考えられる。

Abstract

The improvement of fuel consumption is the most important issue for engine manufactures from the viewpoint of energy and environment conservation. A piston-cylinder system plays an important role for the reduction of an engine friction. For the improvement of the frictional behavior of the piston-cylinder system, it is beneficial to observe and analyze the frictional waveforms during an engine operation.

To meet the above-mentioned demand, frictional waveforms were measured with using the renewed floating liner device. In the newly developed floating liner device, an actual cylinder block itself was used as a test specimen.

The measured single cylinder was an aluminum monolithic type made of hypereutectic Al-17%Si alloy using a high pressure die casting process. The combined piston was a light weight forged piston and a DLC coated piston ring was used. For the measurement, 110cc air cooled single cylinder engine was used.

The observed waveforms were considered to be reasonable and proper from the theoretical point of view. A friction mean effective pressure (FMEP) value was also calculated on each measurement conditions. The FMEP was increased with increasing a load of an engine operation. It was considered that the increase of the FMEP was attributable to the increase of a side force of the piston. Furthermore it was proved that the FMEP was decreased through its running-in process. The FMEP after running-in was 30% lower than that before running-in. This phenomenon was due to the change in the roughness of contact surfaces. And also the tribofilm formed on the contact surface would contribute to the reduction of the FMEP.

1 INTRODUCTION

The improvement of fuel consumption is the most important issue for engine manufactures from the viewpoint of energy and environment conservation. The decrease of frictional loss is a fundamental approach for the improvement of fuel consumption. The contribution to the frictional loss of each component in an engine is investigated previously with using a motoring device. It is reported that the frictional loss of a piston-cylinder system is largest among each engine components, and the ratio to the total engine friction is estimated around 35 to 38% [1]. A piston-cylinder system plays a quite important role for the reduction of the engine friction. In order to reduce the friction of the piston-cylinder system, the frictional behavior of the system during each cycle should be observed and analyzed.

For the above-mentioned purpose, a floating liner device is so effective that it has been widely used [2, 3]. The schematic illustration of the conventional floating liner device is shown in Figure 1. A liner which is a cylindrical sleeve is set in a cylinder block. The liner is divorced from the cylinder block and supported by load washers attached to the lower end of the liner. As the result, the liner is sustaining in the cylinder block as if it was floating. Due to this configuration, the frictional force generated in the piston- cylinder system can be measured precisely even in a firing mode.

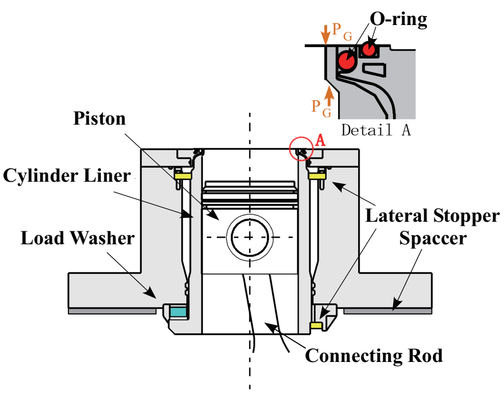


Figure 1: Conventional floating liner device.

Previously a lot of investigations are performed with using the floating liner device. Consequently the effects of a resin coating, surface roughness and skirt area on the frictional force of the piston-cylinder system are clarified [4, 5]. However, among these previous investigations, measurement results of an aluminum (Al) monolithic cylinder have never been seen because it is quite difficult to prepare a thin-walled Al liner eliminating considerable distortion due to its low elasticity whereas the conventional liner for the floating liner device is made of cast iron.

Reviewing the cylinder block design, it can be categorized by cylinder bore materials and its structures as shown in Figure 2. Conventionally a cast-in or press-fitted type heterogeneous cylinder is widely used for single cylinder motorcycles mainly used in Asian countries. However, recently the monolithic type Al cylinder, named DiASil, has been widely applied for single cylinder motorcycles [6]. Considering the improvement of frictional loss of DiASil cylinder, the frictional waveform of DiASil cylinder should be measured and analyzed in some way.

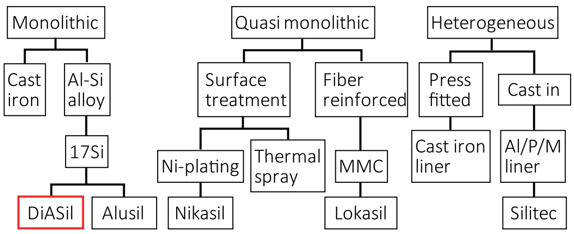


Figure 2: Categorized cylinder block design according to the cylinder bore material and its structure.

In order to meet above mentioned demand, the authors developed a renewed floating liner device using an actual Al monolithic cylinder block [7]. With using the developed floating liner device, waveforms of frictional force were measured at several engine revolutions from 3000rpm to 5200rpm. The measured waveforms showed no irregular fluctuations without any kinds of filtering processing, for example low-pass filtering and/or high-pass filtering. It revealed the enough natural frequency of the developed device for the testing condition. Consequently the obtained waveforms of frictional force were proved to be reasonable and proper.

From the above mentioned investigation, the frictional behavior of Al monolithic cylinder especially under higher engine revolution range was clarified. However, the frictional behavior at full load condition and the change in friction during running-in process have not been described. Considering the usage environment of small motorcycles, these characteristics would be important for the engine performance of the small motorcycles.

In this research, with using the renewed floating liner device originally developed by the authors[7], the frictional forces of DiASil cylinder in combination with a lightweight forged piston and DLC coated piston rings were measured with increasing the load of engine operation, then the effect of the load of engine operation on FMEP was investigated. In this paper, the FMEP was defined as a friction mean effective pressure caused by the friction between the cylinder and the piston & piston rings. Also the change in the FMEP before and after running-in process was investigated by measuring the frictional waveforms and analyzing the surface morphology.

2 EXPERIMENTAL

2-1. FLOATING LINER DEVICE[7]

An appearance of the developed floating liner device is shown in Figure 3. Also a schematic illustration of the structure of the floating liner device is shown in Figure 4. An actual cylinder block was fixed to an outer block by 4 pieces of load washers. A frictional force acted on the cylinder block was detected with these load washers. Here, generally speaking, an increase in the cylinder weight affects the natural frequency of a measurement device, and natural frequency limit the maximum engine speed for friction measurements. The cylinder in the newly developed device was slightly heavier than that in the conventional floating liner because of using the actual cylinder block. Furthermore friction measurements under higher engine speed were required in this study. Therefore 4 pieces of load washers were required to have an enough natural frequency for friction measurements under higher engine speed. A thrust force acted on the cylinder block was supported by a lateral stopper attached at the upper and lower end of the cylinder block.

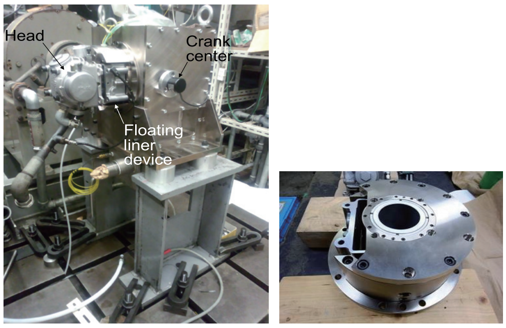


Figure 3: Developed floating liner device. Left photo: Full view of testing apparatus. Floating liner device fixed with head cylinder is attached to crankcase unit. Right photo: Appearance of float-ing liner unit. Cylinder block is covered with outer cylinder.

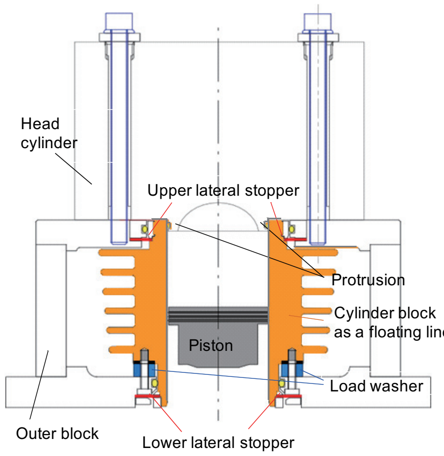


Figure 4: Schematic illustration of the structure of developed floating liner device. Connecting rod and crankcase are abbreviated from the illustra-tion for simplification.

Generally in the floating liner device, a metallic gasket for sealing a combustion gas pressure cannot be used because the cylinder block should be "floated". So, in the developed device, a sealing method originally proposed by Furuhama et al. [8] was applied. That is to say, the lower surface of a cylinder head mating to the cylinder block was partially protruded into the cylinder bore, and then the O-ring was put in the groove machined on the outer surface of the protrusion of the cylinder head.

The influence of the friction between the O-ring and the cylinder bore was corrected based on a relation-ship between a pressure in a combustion chamber and a power output of the load washer. Furthermore the pres-sure in a crankcase influences the power output of the load washer. The influence was also corrected with using an actual pressure measured in the crankcase.

A cooling air was blown into the space between the cylinder block and an outer block. The temperature of the cylinder block during an engine operation was measured with the thermocouple embedded near the cylinder bore surface.

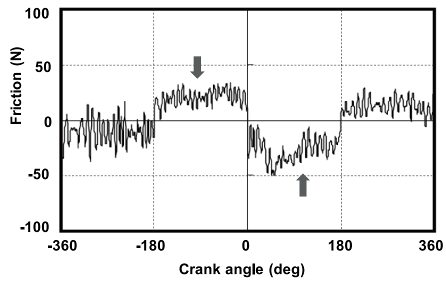
The specification of tested engine is shown in Table 1. The tested engine is 110cc air cooled single cylinder type. The cylinder bore diameter is 50mm and the stroke is 57.9mm.

Table 1: Specification of tested engine.

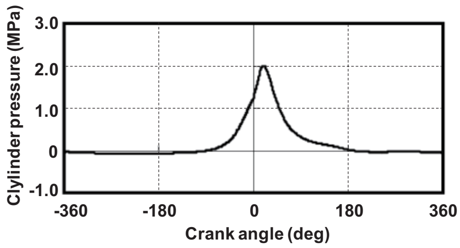
Bore x Stroke (mm)	50 x 57.9
Displacement (L)	0.11
Connecting rod length (mm)	93.5
Reciprocating parts mass (kg)	0.136

A typical waveform of the frictional force quoted from the reference [7] is shown in Figure 5(a). The horizontal axis shows a crank angle from -360 to 360°. 0° shows the point of a compressive top dead center (TDC). So, each 180° of crank angle corresponds to each engine stroke, intake, compression, expansion and exhaust. The vertical axis shows the frictional force generated between the cylinder and the piston / the piston ring set. Positive or negative value of the vertical axis shows the direction of the friction force. When the frictional force toward the TDC is generated on the cylinder bore surface, the measured frictional force is presented in positive value, and vice versa. A corresponding cylinder pressure is also shown in Figure 5(b). As well as Figure 5(a), the horizontal axis shows the crank angle. The vertical axis shown as "cylinder pressure" indicates the pressure inside of a combustion chamber. At this moment, the engine revolution was 5200rpm and the indicated mean effective pressure (IMEP) value was 494kPa.

Then the waveforms at 4400, 4800 and 5200 rpm were analyzed and then FMEP values were calculated in the reference [7]. From the analysis, it was revealed that the FMEP value increased with increasing the engine revolution. This phenomenon showed that a thicker fluid lubrication film which was formed at a higher engine revolution resulted in the increase of the viscosity resistance. So the feature of hydrodynamic lubrication condition was observed through above mentioned. Under the hydrodynamic lubrication condition, friction force theoretically shows proportional trend to sliding velocity under a constant load. However the friction waveform shown in figure 5 shows a flat profile at the mid part of the stroke, although the piston speed was increased, as pointed with arrows. It was suggested that oil film thickness could not reach the theoretical value because of the shortage of supplied oil volume [9] and oil film in a constant thickness was generated at around the mid-stroke, hence the friction wave form showed the flat profile.



(a) Waveform of the frictional force.



(b) Corresponding cylinder pressure.

Figure 5: Typical waveform of the frictional force measured with using renewed floating liner device (7).

2-2. TESTING CONDITIONS

Testing conditions are summarized in Table 2. Every measurement was performed at 3000rpm. The IMEP which showed the load of the engine operation was changed from 300 to 900kPa, which was a full load condition. Cylinder temperature was set at 373K in case 300kPa IMEP and at 393 K in case 500 and 900kPa IMEP respectively. The difference of the cylinder temperatures was due to the difficulty of a temperature control only by air flow. Precise control of the cylinder is a future task. Oil temperature was kept at 343 K. In case of 3000 rpm and 300kPa

IMEP, the measurements were done before and after running-in. The rest of measurements were done after running-in.

Table 2: Testing conditions.

Engine revolution (rpm)	IMEP (kPa)	Cylinder temperature (K)	Oil temperature (K)
3000	300	373+/-5	343
3000	500	393+/-5	343
3000	900	393+/-5	343

The definition of running-in in this research is as follows. As soon as the engine operation was started, the measured friction value is gradually declining. After several period of engine operation, the decline is saturated at a certain value. The period is defined as running-in in this research.

2-3. TESTED CYLINDER BLOCK, PISTON AND PISTON RING

The appearance of tested cylinder block, piston and piston rings are shown in Figure 6. The cylinder block is made of hypereutectic Al-17%Si alloy by using a high pressure die casting process [6]. The chemical composition of the cylinder block is shown in Table 3. After die casting, the cylinder block was T5 heat treated. Then the cylinder bore surface was honed subsequently etched with alkaline solution in order to expose Si particles dispersed in an Al matrix. The surface morphology of alkaline etched surface is shown in Figure 7. These exposed Si particles play a role for preventing a wear and/or a scuffing phenomenon [10, 11].

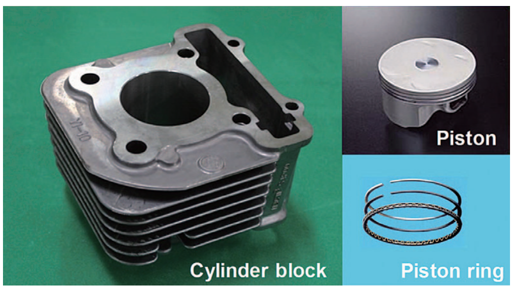


Figure 6: Appearance of tested components, cylinder block, piston and piston rings.

Table 3: Chemical composition of DiASil cylinder.

Si	Cu	Mg	Zn	Fe	Mn	Al
17	4.5	1.2	0.05	0.5	0.5	REM.

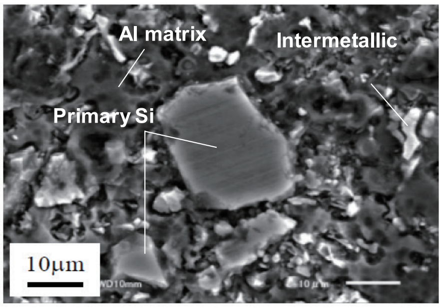


Figure 7: The surface morphology of alkaline etched DiASil cylinder surface. Si particles and several kinds of intermetallic compounds are dispersed in Al matrix. Other fine dispersive particles are not identified.

The piston is made of Al-12%Si-4Cu alloy by using a forging process. The detailed chemical composition of the piston is shown in Table 4. The forged piston was T6 heat treated then machined. On the piston skirt, Fe-Sn plating was treated to prevent a direct contact between the cylinder bore material and the piston skirt material.

Table 4: Chemical composition of forged piston.

Si	Cu	Mg	Fe	Mn	Cr	Ni	Zn	Al
11	4.5	0.6	0.3	0.05	0.04	0.02	0.08	REM.

The piston ring set consists from a top ring, a second ring and an oil ring. The top ring was made of Si-Cr steel (JIS-SWOSC). The sliding surface of the top ring had a barreled face with a DLC coating. The second ring was made of conventional ductile cast iron (JIS-FCD700). The sliding surface had a tapered face. The oil ring consisted from side rails and an expander. The side rails were made of stainless steel (JIS-SUS440) having the barreled face with the DLC coating as with the top ring.

2-4. SURFACE ANALYSIS BEFORE AND AFTER TESTING

Surface roughness was measured with using a surface roughness tester. Surface morphology was observed by a scanning electron microscopy (SEM) and a laser microscopy. The cylinder bore surface was cut out subsequently analyzed by an electron probe microscopy (EPMA).

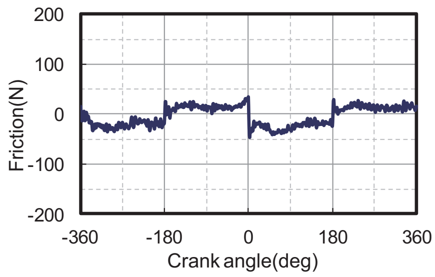
3 RESULTS AND DISCUSSION

3-1. EFFECT OF LOAD OF ENGINE OPERATION

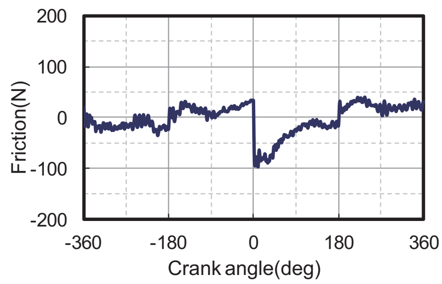
Considering the usage environment of small motorcycles, the motorcycles are quite often used under high loaded condition. Therefore, the frictional behaviors of DiASil cylinder under several load conditions were investigated. The frictional waveforms under 300, 500 and 900kPa of IMEP are shown in Figure 8. The frictional forces of compression and expansion stroke are increased with increasing the IMEP value, especially around 0°, compressive TDC. Also the frictional force at the early period of exhaust stroke is increased with increasing the IMEP. On the other hand, the frictional force at the mid stroke is kept lower even under 900kPa of IMEP same as shown in Figure 5 (a). Then FMEP values were calculated by integrating the frictional waveforms from -360 to 360°. The relationship between the IMEP and the FMEP is shown in Figure 9. The FMEP value increases proportionally with increasing the IMEP.

Then in order to investigate the effect of a side force exerted by the piston, an averaged pressure distribution on the piston skirt of each stroke was calculated with RecurDyn developed by Function Bay K.K. The average pressure is defined as the value which is the integrated pressure generated during each stroke divided by the number of calculating steps, 720. The calculation was done at every 1°crank angle. The calculation results of a thrust side are shown in Figure 10. The averaged pressure is increasing with increasing the IMEP value especially in the expansion stroke.

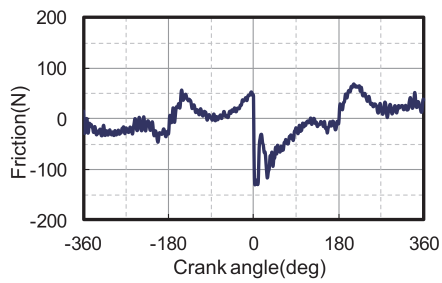
Then the side force through the engine stroke was calculated by integrating the averaged pressure. At this moment, not only the side force of the thrust side but also that of the anti-thrust side were included in the calculated side force value. The relationship between the averaged side force on the piston skirt and the FMEP is shown in Figure 11. It is shown that the FMEP is correlated highly with the averaged side force. Nakanishi et al.[12] revealed by a floating liner device using tri-axial force sensors that the friction of piston was larger than that of piston rings and it accounted for more than 50% of total friction, in some case it reached 90% of total friction. In this research, same as above-mentioned result, the side force of the piston skirt would make a dominant influence on the change in the FMEP value.



(a) Frictional waveform of 3000rpm, 300kPa IMEP.



(b) Frictional waveform of 3000rpm, 500kPa IMEP.



(c) Frictional waveform of 3000rpm, 900kPa IMEP.

Figure 8: Frictional waveforms of 3000 rpm under 300, 500 and 900kPa IMEP.

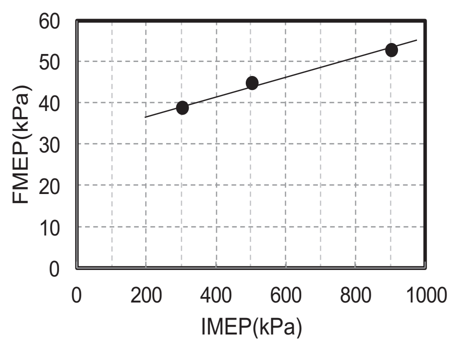


Figure 9: Relationship between IMEP and FMEP.

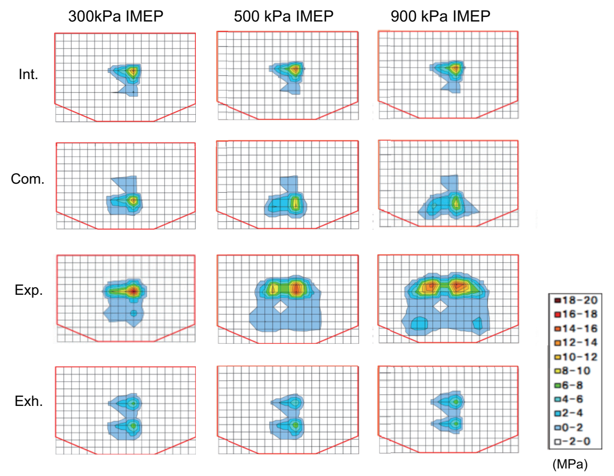


Figure 10: Averaged pressure distribution on the thrust side of piston skirt of each stroke. Each row corresponds to each stroke, intake, compression, expansion and exhaust. Pressure distributions are calculated at 3000 rpm under 300, 500 and 900kPa IMEP respectively. Red lines represent calculated piston skirt area.

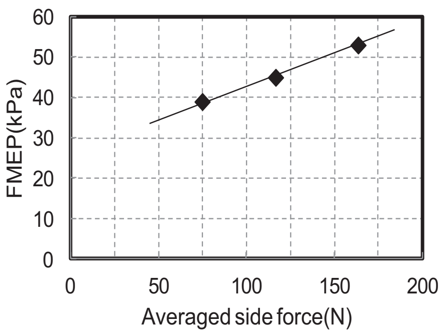
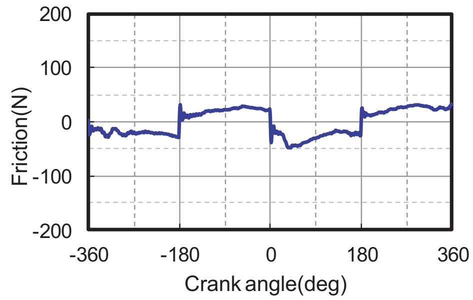


Figure 11: Effect of averaged side force of piston skirt on the increase of FMEP value.

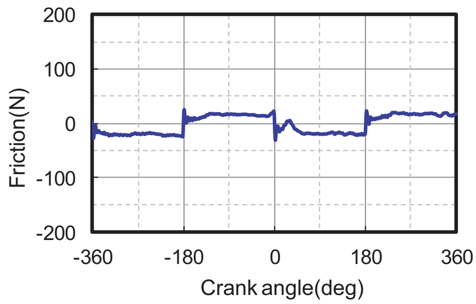
3-2. CHANGE IN FRICTION THROUGH RUNNING-IN PROCESS

The running-in process is important for the engine performance of the small motorcycles. DiASil cylinder has a characteristic cylinder surface morphology as shown in Figure 7, that is to say, a lot of exposed Si particles are existing to prevent tribological problems during the early stage of engine operation [11]. The exposure height of these Si particles is decreased with the time of engine operation [6]. So, in this research, the friction behaviors before and after running-in were measured then the effect of surface condition was discussed.

The frictional waveforms before and after running-in process are shown in Figure 12. Comparing these waveforms, the frictional force of compression and expansion stroke especially around the compressive TDC is significantly reduced. The friction of the mid of exhaust stroke is also decreased. The FMEP values calculated from the waveforms are shown in Figure 13. The FMEP value after running-in is 30% lower than that before running-in.



(a) Frictional waveform before running-in.



(b) Frictional waveform after running-in.

Figure 12: Frictional waveforms before and after running-in process.

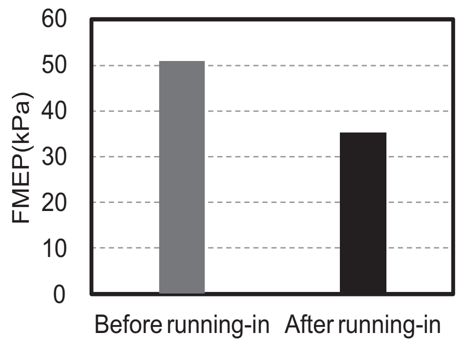


Figure 13: Effect of running-in on the FMEP value.

In order to investigate the reason why the FMEP value after running-in was decreased comparing to the FMEP before running-in, the sliding surfaces of the piston and the cylinder bore were measured and analyzed.

Figure 14 shows a surface profile of the piston skirt before and after running-in. A regular triangular shape having 0.56 μm Ra is observed before running-in as shown in Figure 14 (a). On the other hand, after running-in, peaks of the triangular shape are worn out and become plateau having 0.26 μm Ra as shown in Figure 14 (b). However, even after running-in, valleys are still remaining. These valleys would make a role of oil retention.

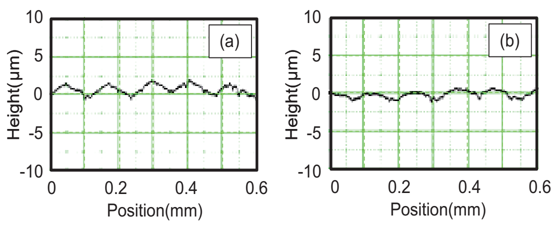


Figure 14: Surface profile of piston skirt before and after running-in. (a); before running-in. (b); after running-in.

Regarding the effect of the surface roughness of piston skirt on engine friction, Takiguchi et al.[5] revealed that the reduction of the surface roughness of the piston skirt resulted in more than 10% reduction of frictional loss with the conventional floating liner device using a combination of an Al casted piston and a cast iron liner. In the present research, the change in the surface profile of the piston skirt would contribute to the reduction of the FMEP.

On the other hand, the effect of a cylinder bore roughness was also investigated. Kawanishi et al. [13] measured the change in friction during an engine operation with using a conventional floating liner device, then, they revealed that the decrease of the cylinder bore roughness was a dominant factor for the reduction of FMEP through running-in process between the piston ring and the cast iron cylinder bore. The change in surface morphology of an Al monolithic type cylinder bore during running-in process was also investigated. Slattery et al. [14] characterized the surface evolution during running-in process comparing the worn and unworn surfaces of the cylinder bore, and then, they revealed that the evolved cylinder bore surface consisted of exposed Si particles with smooth edges and tribofilm coating over the Al matrix. Dienwiebel et al. [15] also investigated the Al-Si cylinder bore after an engine operation. They concluded that the wear particles were embedded and several chemical elements from engine oil were mixed into the Al matrix, consequently the Al matrix was strongly modified during running-in.

In this research, the surface morphology of tested cylinder bore surfaces before and after running-in process were observed with the laser microscopy. Observed images are shown in Figure 15. The Si particles having around 1 μm of exposure height, 0.39 μm Ra, before running-in were worn out, consequently became a smooth surface after running-in. However the exposure height of Si particles was still remaining around 0.4 μm, 0.18 μm Ra.

The surface roughness of a top ring and a second ring was also measured. The surface roughness of the top ring decreased from 0.31 μm Ra before running-in to 0.05 μm Ra after running-in. That of the second ring also decreased from 0.25 μm Ra to 0.14 μm Ra. Regarding side-rails of an oil ring, the width was too narrow to be measured.

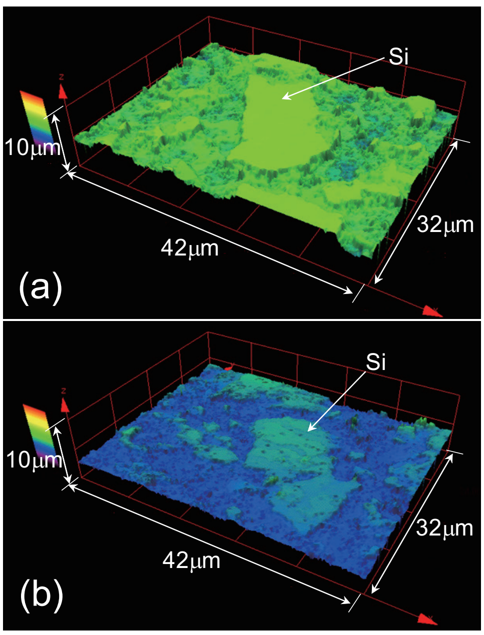


Figure 15: Surface morphology before and after running-in. (a); before running-in. (b); after running-in.

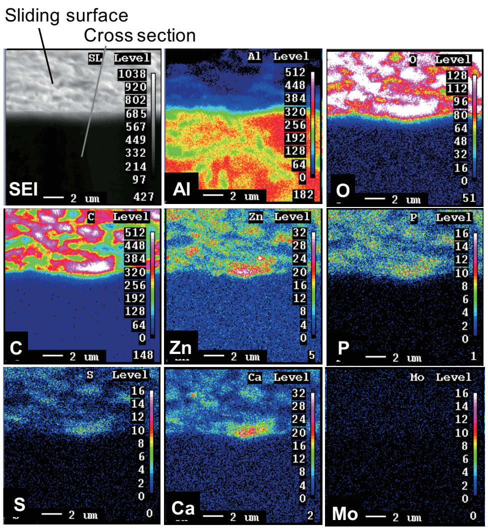


Figure 16: Surface analysis of the cylinder bore after running-in.

From these measurement results, it was suggested that the surface roughness of the contact surfaces would have a large influence on the FMEP value. In addition, the surface change from the chemical point of view should be considered as described in references [14] and [15]. So, in this research, the sliding surface of the cylinder bore after running-in was cut out and analyzed with the EPMA. Prior to the analysis, the surface was degreased with acetone in order to clean up residual engine oil. The result is shown in Figure 16.

C, Zn, P, S and Ca which were the constituents of engine oil were detected from the sliding surface. This result suggests that chemical elements from engine oil could react with Al surface generating some kinds of tribofilm. Also O was strongly detected from the sliding surface compared to the cross section of the cylinder bore specimen. The detected O from the sliding surface was not coming from absorbed oxygen or natural oxide film. This is suggested that the surface was oxidized through running-in. Mo was not detected because the tested engine oil does not contain MoS2 for preventing the slippage of a clutch system.

From these results, it was suggested that not only surface roughness but also the chemical reaction between the constituents of engine oil and Al surface would play an important role for the reduction of FMEP through running-in process. The contribution of these factors on the reduction of the FMEP is not clarified quantitatively, so it should be future issues to deal with.

4 CONCLUSIONS

The frictional forces of Al monolithic cylinder (DiASil) in combination with a lightweight forged piston and DLC coated piston rings were measured with using a renewed floating liner device at several load of engine operating conditions and also before and after running-in process. Through the analysis of the obtained friction data in combination with the calculation of contact condition and the detailed analysis of the tested surfaces, following conclusions are derived.

The FMEP was increased linearly with increasing the load of engine operation. For investigating the dominant factor, the average pressure on the piston skirt was calculated. In consequence, a high correlation between the averaged side force of the piston skirt and the FMEP was revealed. It was suggested that the side force of the piston skirt would be a dominant factor for the increase of the FMEP.

The FMEP after running-in was decreased compared to that before running-in. The FMEP value after running-in was 30% lower than that before running-in. This result would be attributable to the reduction of the surface roughness of the contact surfaces. In addition, it was suggested that the chemical reaction between the constituents of engine oil and the Al surface would contribute to the reduction of the FMEP.

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