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Development of measurement apparatus of piston assembly friction in a small motorcycle engine

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This study developed a friction measurement apparatus with a floating cylinder liner in a small motorcycle engine. In this measurement apparatus, joint plates were installed in the grooves on the outer periphery of the floating liner, and then the plates, as well as load washers of piezo type, were mounted in the cylinder block at both the thrust and the anti-thrust sides. A stepped ring, protruding inward, was mounted on the top of the floating liner so that cylinder pressure acting on the stepped portion was balanced in the vertical direction. Thus, it was possible to measure the friction in the sliding directions of the piston. Using this apparatus, the effect of the engine operating period on friction was investigated in a piston micro-dimpled with a fine particle bombarding process. Results indicated that, at low engine speeds, friction decreased with the operating period, but at high engine speeds, friction decreased after 10 hours of operation, and then increased after 20 hours of operation.

Key words: *piston assembly friction, floating cylinder liner, piston, fine particle bombarding, micro dimple*

1. Introduction

Small motorcycle engines are required not only to have excellent acceleration performance but also to have excellent economic efficiency, i.e., less expensive engines with low fuel consumption rates. To improve fuel efficiency, it is effective to reduce engine friction, especially piston assembly friction (the friction between the piston, the piston rings, and the cylinder liner) [2, 8, 10, 12]. Two measurement methods have been mainly used to measure this piston assembly friction during engine operation. One is the floating liner method, in which load washers are fixed between the lower portions of the floating liner and the cylinder block on the thrust and the anti-thrust sides [1, 3, 4, 6, 9, 18, 20–22]. The other is the three-component force sensor method, in which three-component force sensors are fixed, either at 8 points between the outer periphery of the floating liner and the inner periphery of the block on the thrust side, or at 3 points each between the outer periphery of the floating liner and the inner periphery of the block, on both the thrust and the anti-thrust sides [5, 7, 11, 13, 19]. Both methods require a prototype floating liner, but the first floating liner method requires fewer sensors and fewer engine modifications than the three-component force sensor method. So there are more reports on friction measurements using the floating liner method. Our previous study developed a friction measurement apparatus with a floating liner for an eco-mileage vehicle engine, by using components of a small motorcycle engine as much as possible [14, 15]. In this measurement apparatus, joints plates were inserted in the grooves on the outer periphery of the floating liner, machined from the periphery of a commercially available air-cooled cylinder, and then load washers were mounted between the joint plates and the cylinder block, at both the thrust and the anti-thrust sides. To suppress lateral displacement due to piston thrust force, clamping bolts were also mounted to the cylinder block at four sides: thrust, anti-thrust, front, and rear. This enabled measuring the piston assembly friction when the intake and the exhaust valves were not activated, i.e., no cylinder pressure was applied. This current study significantly modified the structure of the upper part of the floating liner from our previous measurement apparatus, so that the force due to the pressure in the cylinder was not transmitted to the floating liner. Thus, it was possible to measure piston assembly friction even when the intake and the exhaust valves were activated, and cylinder pressure was applied. Into this new measurement apparatus, a micro-dimpled piston with a fine particle bombarding (FPB) process [15–17] was installed. Such micro-dimpled pistons enjoy reduced friction compared with untreated pistons [15–17] and reduced total engine friction with increasing engine operating period [16]. However, no report has measured piston assembly friction with a micro-dimpled piston for each cycle and examined the effect of the engine operating period on its friction. This study investigated in detail how piston assembly friction with a micro-dimpled piston changed with the engine operating period compared to before break-in.

2. Experimental apparatus and method

Figures 1 and 2 show the friction measurement apparatus with the floating liner. As in our previous study, this measurement apparatus employed a four-stroke, air-cooled, horizontal, single-cylinder, gasoline engine for a commercially available small motorcycle (bore × stroke = 39 mm × 41.4 mm) [14, 15]. The crankcase of this engine was cut, and a cover was attached to the cut of the crankcase. The engine was then rotated from horizontal to vertical. For the floating liner, the outer periphery of the air-cooled cylinder (an aluminum finned cylinder with a cast-iron liner) was turned, and the grooves for the joint plates were machined on its outer periphery at the thrust and the anti-thrust sides. In addition, the top of the liner was machined, and an aluminum alloy stepped ring was mounted on the top of the liner, protruding inward. Cylinder pressure acting on the stepped portion was balanced in the vertical direction, so that the force due to the pressure in the cylinder was not transmitted to the floating liner. Joint plates were installed in the grooves on the outer periphery of the floating liner, and then piezo type load washers were mounted between the joint plates and the cylinder block, at both the thrust and the anti-thrust sides. Thus, only vertical force (piston assembly force) was applied to the joint plates, and from there to the load washers. An O-ring was installed between the stepped ring of the floating liner and the spacer immediately above it for gas sealing. The compression ratio became 7.3 after installing the stepped ring and the spacer. In addition, to suppress lateral displacement due to piston thrust force, circular thin-disk springs were attached to the upper and lower sides between the floating liner and the cylinder block. A strain-gage pressure transducer was also attached to the spark plug hole when measuring the cylinder pressure.

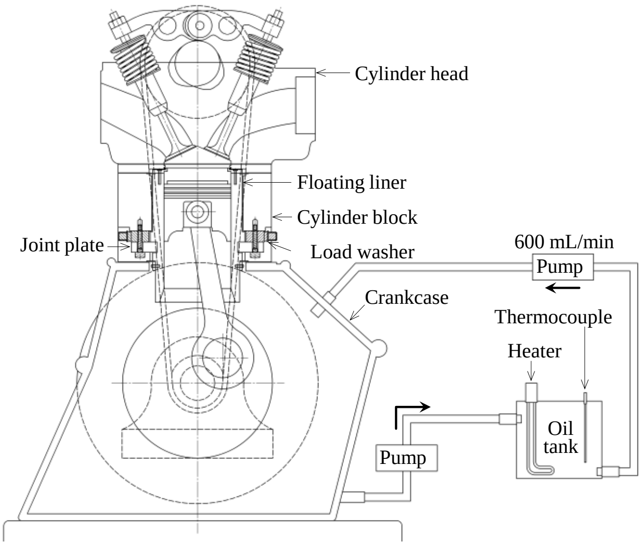

Fig. 1. Measurement apparatus of piston assembly friction

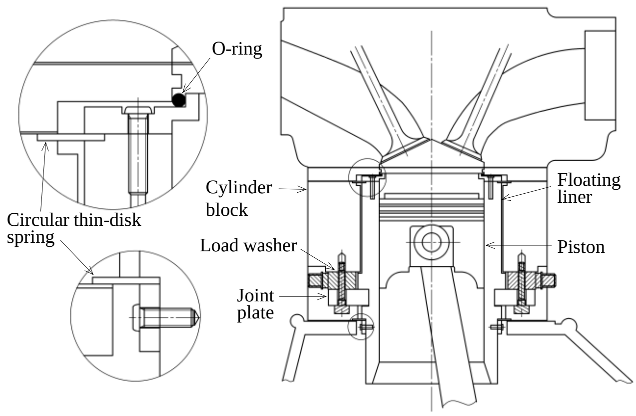

Fig. 2. Friction measurement system with floating liner

Bore surface temperature was adjusted with a temperature controller, by installing heaters in the cylinder block at the thrust, the anti-thrust, and the rear sides, and thermocouples at the front and the rear sides. The oil temperature was also adjusted with a temperature controller by installing a heater and a thermocouple in an oil tank outside the engine, as in our previous study [14, 15]. The lubricating oil, at fixed temperature, was pumped from the oil tank to a pipe on the upside of the crankcase, and then supplied from its pipe to the crankshaft at a flow rate of 600 mL/min, as shown in Fig. 1.

Figure 3 shows the micro-dimpled piston and its surface shape on the piston skirt. In this micro-dimpled piston, ceramic particles with a diameter of 45 μm were air-blasted onto a commercially available standard piston, which had a streaked sliding surface on the skirt, except for three grooves in the skirt. The top land of the piston was also machined away from contact with the stepped ring of the floating liner.

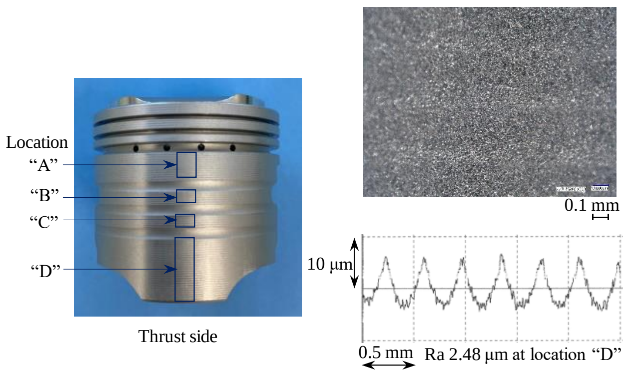

Fig. 3. Micro-dimpled piston and surface shape on the piston skirt

Table 1 shows the experimental rings, which are standard parts of a commercially available engine. These rings had been fully run in our previous study [15].

Table 1. Experimental piston rings

[mm]

Top ring	Second ring	Oil ring
1.7; 0.78	1.7; 1°46'; 0.78	1.04; 0.3; 0.3
End clearance 0.08 mm Tension 3.2 N Chrome plating	End clearance 0.15 mm Tension 4.4 N Phosphate	End clearance 0.52 mm Tension 11.2 N Chrome plating

The floating liner had been fully operated in advance, with a standard untreated piston, and its roughness of the sliding surface on the liner bore was Rz_{JIS} 0.90~1.56 μm (Ra 0.12~0.19 μm).

The experimental lubricating oil was Honda genuine Ultra Green with a lower viscosity for low-fuel consumption engines. The kinematic viscosities of the lubricating oil measured with the Redwood viscometer were 30.3 mm^2/s at 40°C, 15.8 mm^2/s at 60°C, and 10.4 mm^2/s at 80°C.

The engine was operated by motoring, and the experiments were conducted according to the procedure shown in Fig. 4. That is, a new dimpled piston was installed in this measurement apparatus, and then without any break-in operation, the piston assembly friction was measured at engine speeds of 800 rpm, 1000 rpm, 1200 rpm, and 1400 rpm, at temperatures of 40°C, 60°C, and 80°C, respectively. Next, after running at 80°C and 1400 rpm for 10 hours, the friction was measured under these same conditions. Furthermore, after running at 80°C and 1400 rpm for another 10 hours (meaning after 20 hours in total at 80°C and 1400 rpm), friction was again measured under these same conditions.

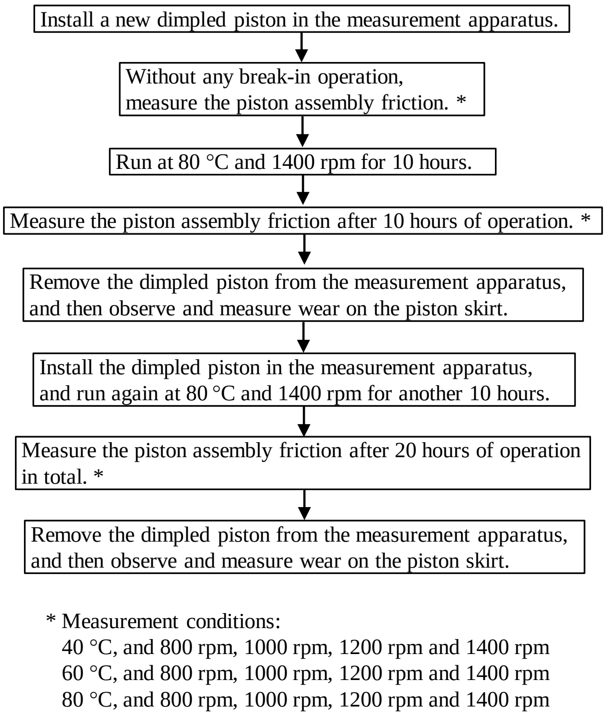

Fig. 4. Experimental procedure

3. Results and discussion

Figures 5 and 6 show the effect of the engine operating period on piston assembly friction at 60°C, first at 1000 rpm and then at 1400 rpm, respectively. In Figs. 5 and 6, crank angles of –360º, 0º, and 360º represent engine top dead center (TDC); –180º and 180º bottom dead center (BDC). During the piston downward stroke (from –360º to –180º in the intake stroke, and from 0º to 180º in the expansion stroke), a downward force is applied to the liner, and during the piston upward stroke (from –180º to 0º in the compression stroke, and from 180º to 360º in the exhaust stroke), an upward force is applied to the liner. So these forces are indicated as negative force and positive force, respectively. In Fig. 5, at 60°C and 1000 rpm, as the engine operating period increased, friction decreased around TDC and BDC, as well as during the compression and the expansion strokes. In addition, the longer the operating period, the lower the friction reduction rate. In Fig. 6, at 60°C and 1400 rpm, friction after 10 hours of operation decreased around TDC and BDC and during the compression stroke, compared to before break-in operation (0 hours of operation). However, friction after 20 hours of operation was almost the same as that after 10 hours of operation, around TDC and BDC and during the compression and expansion strokes. During the intake and the exhaust strokes, except around TDC and BDC, friction after 20 hours of operation increased beyond that after 10 hours.

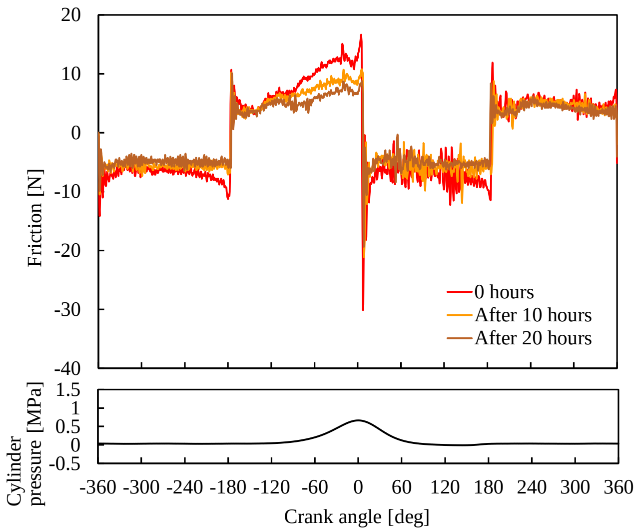

Fig. 5. Piston assembly friction at 60°C and 1000 rpm

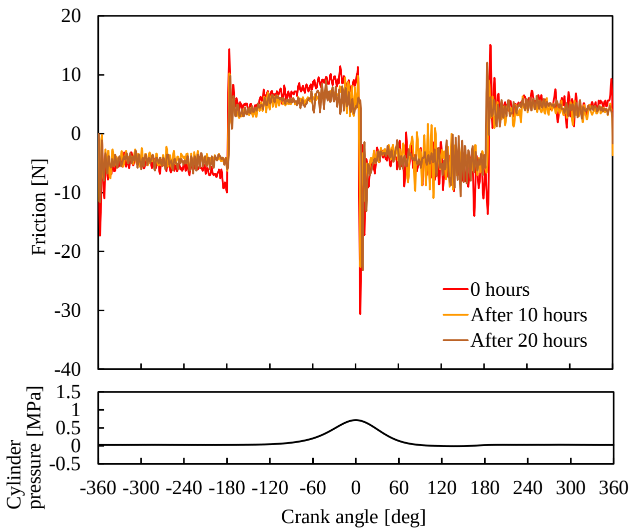

Fig. 6. Piston assembly friction at 60°C and 1400 rpm

Figures 7 to 9 show the effect of the engine operating period on the friction mean effective pressure (FMEP) at 40°C, 60°C, and 80°C, respectively. Here the FMEP was obtained by integrating the absolute value of friction at crank angles of –360° to 360° and dividing by the stroke volume. As the engine operating period increased, the FMEP decreased in each temperature, except at 1200 rpm and 1400 rpm after 20 hours of operation. At 1200 rpm and 1400 rpm, the FMEP after 20 hours of operation was higher than that after 10 hours of operation. When the engine speed increased, the FMEP decreased, both before break-in operation (0 hours of operation) and after 10 hours of operation. After 20 hours of operation, at 40°C and 60°C, the FMEP decreased as the engine speed increased from 800 rpm to 1200 rpm, and at 80°C, the FMEP decreased as the engine speed increased from 800 rpm to 1000 rpm. But at 40°C at more than 1200 rpm, the FMEP hardly changed, and at 60°C at more than 1200 rpm, and at 80°C at more than 1000 rpm, the FMEP increased.

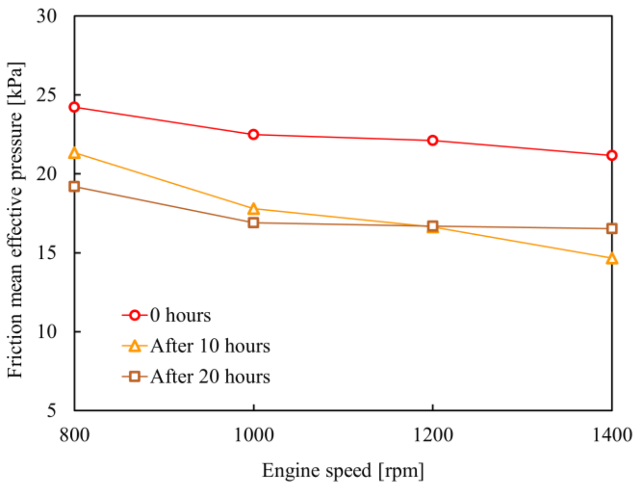

Fig. 7. Friction mean effective pressure at 40°C

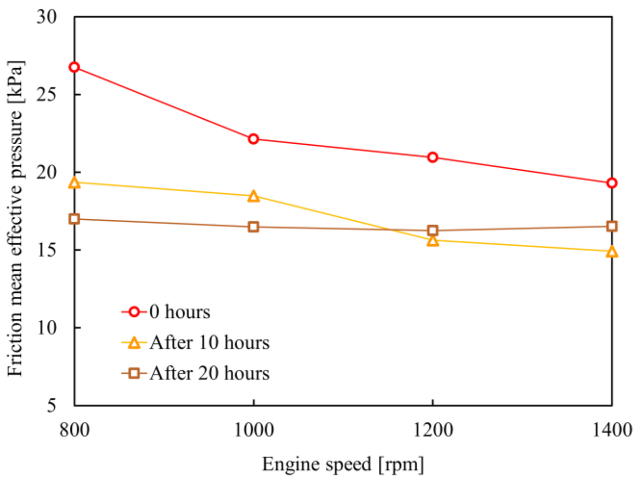

Fig. 8. Friction mean effective pressure at 60°C

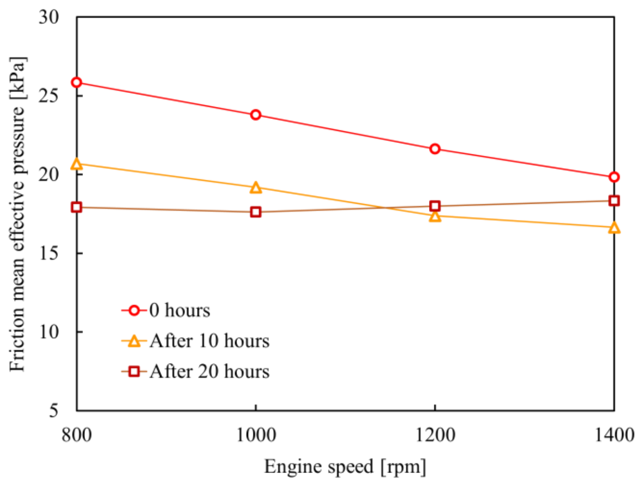

Fig. 9. Friction mean effective pressure at 80°C

When the sliding surface of the piston was observed after 10 hours and after 20 hours of operations, the wear of streaks was visually confirmed in locations "C" and "D" of the skirt. Therefore, the amount of wear on streaks in skirt "C," "D1" (upper part of "D") and "D2" (lower part of "D") was measured by using a surface roughness measuring instrument. Figure 10 shows the maximum amount of wear on streaks in skirt "C," "D1" and "D2" after 10 hours and after 20 hours. In Fig. 10, the maximum amount of wear after 10 hours was 1.5~3.0 μm on the thrust side, and 1.0~3.0 μm on the anti-thrust side. The maximum wear after 20 hours increased by 0.5~1.0 μm compared to that after 10 hours, but the amount of wear per hour tended to decrease.

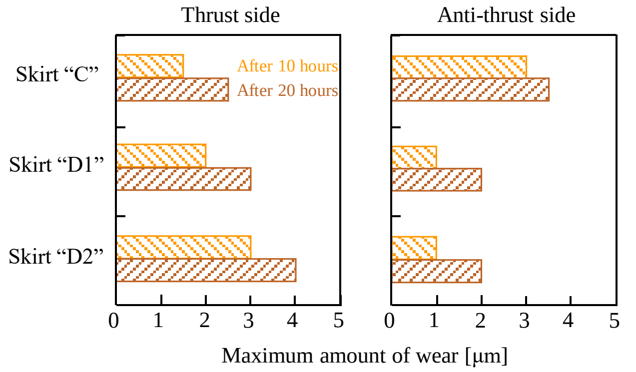

Fig. 10. Maximum amount of wear on streaks in lower skirt of piston after 10 hours and after 20 hours

Figure 11 shows the observation results of the worn part of skirt "C" after 10 hours and after 20 hours. In Fig. 11, the wear on both the thrust and the anti-thrust sides tended to be wider after 20 hours than after 10 hours, but after both 10 hours and 20 hours, micro-dimples remained even on the worn surfaces of the streak, and no significant difference was observed on them.

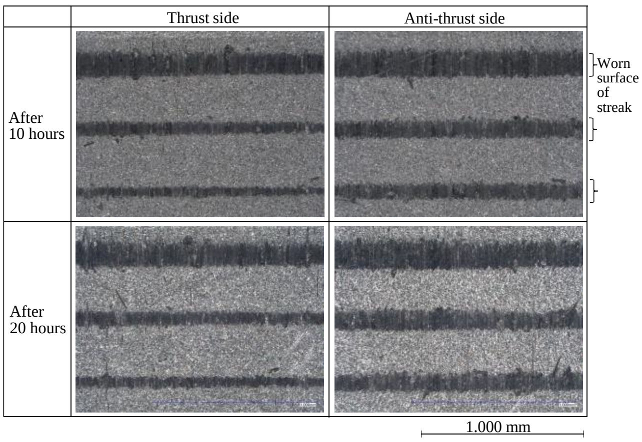

Fig. 11. Observation results of worn part at piston skirt "C" after 10 hours and after 20 hours

It seemed that, at 800 rpm and 1000 rpm, the longer the engine operating period, the greater the wear of the sliding surface, decreasing pressure (the normal load per area) on the sliding surface, and the less dominant the severe mixed lubrication, reducing the FMEP. In this measurement apparatus, the lubricating oil was supplied to the crankshaft, and was then splashed onto the cylinder by the rotation of the crankshaft. Therefore, as the engine speed increased, the amount of oil supplied to the piston per unit time increased. It was thought that, with increased engine speed, the amount of oil supplied to the piston increased, and then friction around TDC of compression stroke (around 0º) decreased, reducing the FMEP except after 20 hours of operation. It also seemed that, after 20 hours of operation, because the wear on the streak of skirt had increased, forming a suitable sliding surface, hydrodynamic lubrication became dominant at 1200 rpm and 1400 rpm, and then friction increased during the intake and the exhaust strokes (where the piston thrust force weakened), except around TDC and BDC, increasing the FMEP.

4. Conclusions

Using a small motorcycle engine, a friction measurement apparatus with a floating liner was constructed, so that the force due to the cylinder pressure was not transmitted to the floating liner. Utilizing this measurement apparatus, the influence of engine operating period on friction with a micro-dimpled piston was examined. Results indicated that, at engine speeds of 800 rpm and 1000 rpm, friction decreased with the engine operating period. At more than 1000 rpm, friction decreased after 10 hours of operation, but then friction increased after 20 hours (to a level greater than after 10 hours). In addition, both before break-in operation (0 hours of operation) and after 10 hours, friction decreased with increasing engine speed. However, after 20 hours, friction increased when engine speed exceeded 1200 rpm at 60°C, and 1000 rpm at 80°C.

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Nomenclature

BDC	bottom dead center	Ra	calculated average roughness
FMEP	friction mean effective pressure	Rz_{JIS}	10-point average roughness
FPB	fine particle bombarding	TDC	top dead center

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