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Frictional Analysis of a Small Two-Stroke Utility Engine via Tear-Down Testing

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Two stroke engines are popular power sources for ambulatory applications because of their superior power to weight ratio. Due to fuel short-circuiting however, they suffer from poor fuel economy and high levels of emissions. Small utility two-stroke engines are typically tuned rich to improve ignition stability and cooling. Additionally the presence of fuel in the crankcase requires high rates of two-stroke oil usage to insure proper lubrication. Given rising petroleum prices and concern for the environment, reducing the fuel consumption and emissions of two-stroke engines is of increasing importance.
With the extreme price sensitivity of small two-stroke engines more sophisticated options such as direct fuel injection, are not possible. Instead, the best starting point for fuel consumption improvement is to insure proper carburetor tuning and lubrication of the engine and optimization of the engine's intake and exhaust tuning.
For this study a typical small utility engine was mounted to a motoring dynamometer. The engine was operated at a standard speed for a given length of time with various oil/fuel ratios. Once the engine ran dry the ignition was switched off and the frictional torque was measured as the engine cooled. Frictional measurements were analyzed as a function of lubricant quantity and engine temperature. Finally friction was measured as the engine was progressively dismantled, removing the exhaust system, carburetor, spark plug, piston and connecting rod. Frictional contributions from each source were then tabulated.

Keywords: Friction, Two-Stroke Engines, Engine Lubrication

1. INTRODUCTION

We chose a typical small utility engine for frictional analysis. The Chinese manufactured Orimas BG380 engine, figure 1, is popular as a back-mounted engine for grass trimming operations. In order to reflect the actual condition of engines in the field we selected a used model in good running condition. The specifications for the BG380 engine are given in figure 2.

Fig. 1 Carburetor side view of the two-stroke analyzed in this study. Dynamometer coupling is on the left.

This engine was the subject of two main studies aimed at investigating the friction of the piston-ring pack in the bore. The engine was subject to tear-down analysis on a motoring frictional dynamometer. Finally the fully assembled engine was operated on various oil/fuel ratios for several minutes. The friction was then measured while motoring the engine at 3000 rpm, and wide open throttle (WOT) to determine the effects of the various lubricant rates on engine friction.

Displacement	30.5 cc
Max Power (6000 rpm)	0.81 kW
Main Bearings (qty 2)	Open cage roller bearings
Connecting Rod Bearings	Needle Bearings
Cooling	Blower forced air
Rings	Compression rings, Qty: 2
Bore	36 mm
Stroke	30 mm

Fig. 2 Specifications of the BG380 engine.

2. TARE-DOWN FRICTION MEASUREMENTS

Prior to testing the engine was instrumented with a thermocouple mounted in the top of the head adjacent to the spark plug. The engine was coupled to a frictional dynamometer capable of spinning the engine at speeds of up to 50 Hz (3000 rpm) while measuring frictional torque with a precision of 0.01 Nm. For tear-down studies the engine frictional torque was measured at various speeds and throttle settings as components were progressively removed. All tests were carried out at ambient temperatures of approximately 25° C, however the head temperatures rose to approximately 50° C when the spark plug was installed due predominantly to heat loss from the compressed air in the combustion chamber. For any given configuration the engine was motored until the temperature became stable, typically several minutes. Throughout the testing of a given configuration the engine temperature was maintained to within +/- 1.5° C to reduce the frictional variation due to temperature.

Fig. 3 Exploded view of the test engine showing the critical components

Initially the engine was tested fully assembled. At each speed the engine was operated at both idle and wide-open throttle (WOT). Subsequently the carburetor and muffler were removed, followed by the spark plug, and finally the piston and connecting rod were removed, leaving only the blower, main roller bearings and an oil seal. The idle curve, figure 4, indicates a sharp decrease in friction from 1200 rpm to 1800 rpm and more gradual drop above 1800 rpm. The WOT curve initially drops, but increases at engine speeds above 2000 rpm. With carburetor and muffler removed the friction roughly parallels the WOT curve, but is slightly higher.

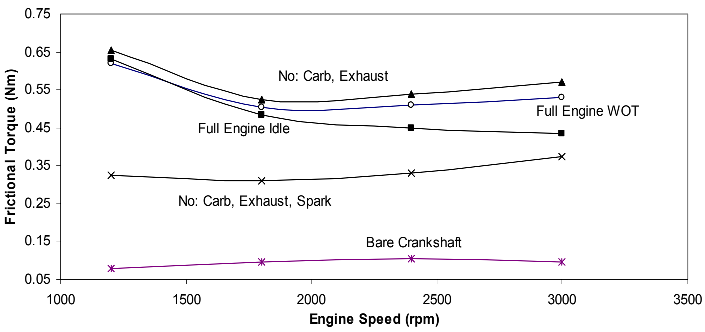

Fig. 4 Frictional torque of the engine versus motoring speed.

With no sparkplug present compression loading of the piston is removed, resulting in lower overall friction. The curve still has a slight drop from 1200 to 1800 rpm, but parallels the no-carb, no-muffler curve above 1800 rpm. Finally the bare crankshaft curve is much lower, rising slightly with engine speed.

Pumping losses increase as the square of the engine speed, and hydrodynamic bearing friction increase with linearly with speed (Furguson 2001). The piston and ring assembly typically dominates friction (Heywood, 1988). Through out the stroke the rings pass through various lubrication regimes. At mid-stroke when the piston is traveling fastest the piston and rings may be fully hydrodynamically lubricated. As the piston approaches the end of the stroke it slows, and the lubricant film thins, resulting in mixed or transitional lubrication. At the end of the stroke piston motion ceases and the lubricant film thins further, especially at top dead center (TDC) on the first compression ring where the lubricant film is typically the thinnest (Rabuté 2001). At this point the rings are predominantly in the boundary lubrication.

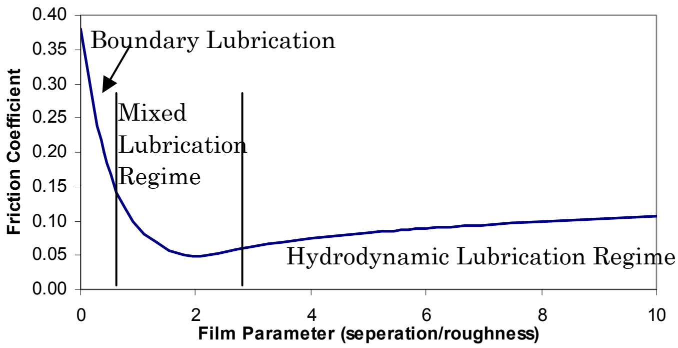

Fig. 5 Stribeck curve. Friction as a function of film thickness parameter (surface separation distance divided by the sum of the surface roughnesses) according to Jocsak (2004)

Lubrication of the piston and rings in two-stroke engines is particular demanding. The interface is constantly being washed with fresh fuel, removing much of the lubricating oil. Additionally the rings are forced to pass over the transfer and exhaust port openings, reducing their ability to dissipate piston heat and, in the case of the exhaust port, exposing them to hot exhaust gasses.

From our data it is apparent that at lower engine speeds the piston and ring assembly is predominantly in the boundary lubrication regime. As speeds increase the assembly is transitions to the mixed lubrication region and friction drops. This is consistent with previous work which shows piston friction to decrease with speed (Patton 1989). The WOT curve indicates an increase in friction above 1800 rpm. As more air is admitted to the combustion chamber, the forces on the piston during compression are increased, raising the lateral loading on the piston, and increasing friction. Pumping losses associated with the larger airflow are also increased. This results in a greater friction than the idle case, which increases with engine speed. Removing the carburetor and muffler all together further increase these losses.

An additional factor potentially contributing to the apparent reduction in friction between 1200 and 1800 rpm is heat loss. During compression the air in the combustion chamber is heated. At low speeds more time is available to conduct this heat to the cylinder wall, resulting in a net negative work, which will show up as friction. At higher speeds less time is available, and this thermal loss is proportionally less important. If this was a major contributor we would expect a significant difference between the idle case and WOT. At WOT much more air is admitted, resulting in greater thermal losses. From figure 4 we see no significant difference between the idle and WOT curves at speeds below 1800 rpm, leading us to believe that thermal losses are not a major factor contributing to the decrease in torque with speed.

Removal of the spark plug greatly reduces the compression loading on the piston though there is still some compression taking place in the crankcase. Friction is greatly reduced in the absence of the spark plug, however there is still a slight drop in friction from 1200 rpm to 1800 rpm. Above 1800 rpm the curve is almost parallel to the WOT and no-carburetor curves, indicating an approximately linear increase in friction with engine speed.

The lowest curve is the friction of the bare crankshaft and blower. It represents frictional losses from the two main roller bearings, an oil seal, and the blower. The general trend is upwards with increasing engine speed.

The fact that the idle curve decreases relative to the WOT curve indicates that the idle volumetric efficiency is decreasing as engine speed increases, reducing the compression loading on the piston. At about 3300 rpm we would expect the idle curve to intersect the no-spark plug curve, and then follow it upwards at higher engine speeds as the unloaded rubbing friction increases.

At 3000 rpm, WOT, the motored engine has a total frictional torque of 0.53 Nm. The crankshaft bearings, oil seal and blower account for approximately 0.095 or 18% of the total. Piston and ring rubbing friction (unpressurized) account for approximately 53% of the total friction at 3000. Including the effect of compression loading this is likely closer to 75%. Finally the motored frictional load of the engine at 3000 rpm, WOT is approximately 167 Watts, giving a frictional mean effective pressure (FMEP) of approximately 109 kPa.

3. OIL-FUEL RATIO FRICTIONAL MEASUREMENTS

Subsequently the effect of the oil/fuel ratio was studied. For this study we used commercially available 97-octane gasoline straight, or mixed with a local brand motorcycle two-stroke mineral based oil. The engine was de-coupled from the frictional dynamometer and operated unloaded on the selected oil/fuel ratio at approximately 2500 rpm for long enough (approximately 4 minutes) to consume 20 ml of the oil/fuel mixture. Once the engine ran dry, the engine was re-coupled to the frictional dynamometer and motored at 3000 rpm. The throttle was held wide-open to allow as much air into the combustion chamber as possible. The resulting frictional torque was measured over the next 3 minutes as the engine cooled. The re-coupling of the engine required approximately 75 seconds. Cylinder head temperatures were approximately 90° C at dynamometer startup and fell to approximately 60° C during the 3 minute test.

During the runs a strong correlation between smoke and the oil/fuel ratio was noted. At oiling rates below 1:25 very little exhaust smoke was noticed, while at 1:5 and 1:2 the engine demonstrated an impressive potential as a mosquito fogger. Additionally the 1:2 ratio required application of the choke which was not needed at any of the other ratios.

The results, shown in figure 6, indicate that as the engine cools the frictional torque tends to increase slightly. As the lubricant film cools, its viscosity will increase, increasing the rubbing friction. Additionally the crankshaft-mounted blower cools the outside of the aluminum cylinder, causing it to contract and decreasing the piston-bore clearance, which will tend to increase frictional loads on the piston.

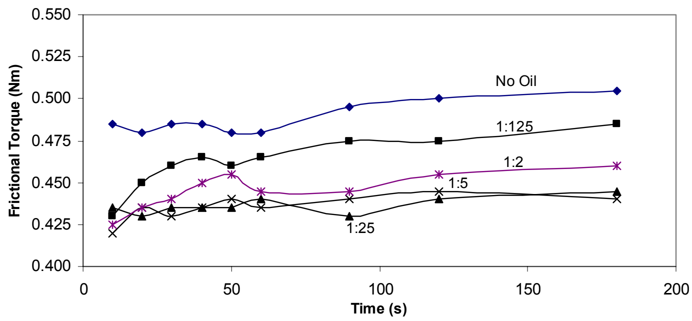

Fig. 6 WOT motored friction at 3000 rpm after operating on various oil/fuel ratios.

The effect of oil/fuel ratio can clearly be seen in figure 6. With strait gasoline the motored friction is the highest, at approximately 0.5 Nm. Increasing the lubricant rate to 1:125 reduces the friction to approximately 0.475 Nm. At the nominal oil/fuel ratio of 1:25 we get the lowest friction of approximately 0.43 Nm. The friction of the 1:5 mixture is very close to that of 1:25, however at 1:2 the friction begins climbing again to around 0.45 Nm.

Taking the data recorded at 2 minutes after dynamometer startup and plotting friction versus oil/fuel lubricant concentration, ie. 1/26 for the 1:25 mixture, we get the curve shown in figure 7. This is essentially the Stribeck curve comparable to figure 5. Significantly the lowest friction occurs at the nominal lubricant ratio of 1:25. Figure 7 also indicates high levels of friction, and probably high rates of wear, in the oil-starved interface at ratios below 1:25.

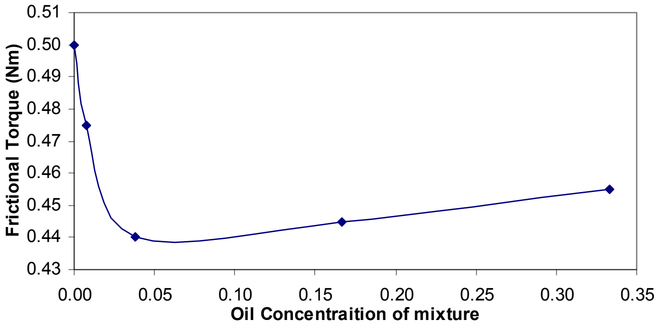

Fig. 7 WOT friction at 3000 rpm, 50°C, after 2 minutes of motoring versus relative oil concentration of oil/fuel mixture.

As the oil/fuel ratio goes higher than 1:25 friction rises again as the interface becomes flooded, and power is being dissipated in shearing the lubricant layer.

This again confirms that at 1:25 oil/fuel ratio and 3000 rpm, WOT, the piston-ring assembly is operating primarily in the mixed lubrication regime. As engine speeds decrease we would expect friction to increase as piston speeds decrease, and the piston-ring assembly transitions to primarily boundary lubrication. This effect can be seen in the data of figure 4, as engine speed is reduced below 2000 rpm frictional torque increases.

4. CONCLUSIONS

Based on our studies we have found that:

(1) Small two-stroke utility engine's friction is dominated by piston-ring pack rubbing friction in the bore.
(2) The minimum friction occurs at an oil/fuel ratio of approximately 1:25.
(3) At lower speeds, or lower oiling rates, the ring-bore interface is likely to be predominantly in boundary lubrication regime.
(4) At oiling rates (above 1:25) friction will increase, as well as visible smoke emissions.
(5) At 3000 rpm, WOT approximately 18% of the motored friction is due to the 2 main bearings, oil seal and blower. The rest is mainly piston-ring pack rubbing friction, with perhaps 2% contributed by the connecting rod needle bearings.

Future work in this area will focus more directly on the piston ring-bore lubrication and friction, rather than relying on whole-engine frictional torque measurements. Additionally new components suitably worn in will be used to reduce the exposure to uncontrolled variables, such as engine usage history.

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