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Friction and Wear Characteristics of TiC Surface Coatings in a Small Two-Stroke Utility Engine

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ABSTRACT

To quantify the friction and wear characteristics of TiC coatings, a small two-stroke engine is operated with both coated, and un-coated piston and rings on a fan-type dynamometer for a given length of time. Fuel consumption and power were monitored during the runs and motored friction was measured both pre- and post-run. Surface analysis consisting of photography, Scanning Electron Microscopy (SEM) imagery, and x-ray diffraction of the piston and rings were compared before and after the runs to quantify coating durability.

Frictional load results and fuel consumption data were analyzed indicating the approximate efficiency gains for the coatings compared with the stock engine parts.

INTRODUCTION

Small carbureted, crankcase scavenged two-stroke engines are well known for their poor emissions and fuel consumption characteristics. Their low cost and light weight, however, continue their popularity for hand-held power applications. In ongoing efforts to improve the efficiency of such hand held two-stroke engines, we have chosen to focus on reducing frictional losses. Previous studies have indicated that the piston-ring pack rubbing is responsible for the majority of the frictional losses in these engines. In this study we examine the frictional and wear characteristics of titanium and titanium carbide coatings on the piston and rings.

For our study we have chosen a popular back-pack mounted 30cc two-stroke used in grass trimming and leaf blowing operations. Specifications of the engine are shown in Figure 1. The friction of this type of engine was previously reported in (Ripin 2007), where the piston and ring pack friction was estimated to be approximately 0.28 Nm at 3000 rpm, or 53% of the total friction. To separate the contribution from the rings and the piston we chose to test 3 different configurations: standard piston and rings, standard piston and coated rings, and a coated piston and coated rings. A wide variety of coatings are popular for different areas in engines. To reduce friction and wear of rings coatings such as chrome, molybdenum, metal-ceramics or TiCN are popular (Andersson 2002). Piston skirts may be coated with various materials such as molybdenum, tin, and even polymer coatings such as Teflon (polytetrafluoroethylene). For this study we chose a titanium carbide (TiC) coating for the rings, and titanium metal for the piston skirt. TiC coatings are common on bearings due to their low friction and high surface hardness (Bushan 1991). These coatings were sputtered using an innovative, targeted sputtering technique developed by a sponsoring vendor. The standard rings have a zinc phosphate coating to reduce wear scratches on the cylinder, and the standard piston was uncoated.

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

Figure 1 Utility engine specifications.

METHODOLOGY

A new engine was acquired with several spare pistons and ring sets for this study. It was coupled to a fan dynamometer and operated at wide-open throttle (WOT) burning a 25:1 fuel/oil ratio for 5 hours. This initial burn-in was intended to burnish the bore roughness, and reduce its influence in subsequent data. The characteristics of the fan load are shown in figure 2, while the engine and fan are shown in figure 3.

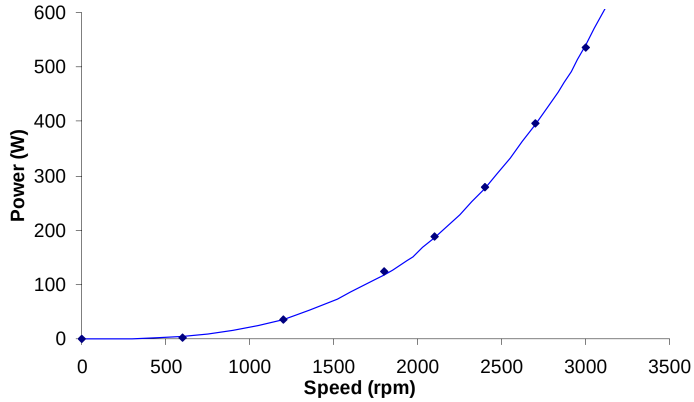

Figure 2 Fan dynamometer load data and best fit curve

The load of the fan was measured on a frictional dynamometer for repeatability. The power required by a fan should be proportional to the cube of the speed. The resulting best fit curve (solid line in figure) is given by Power = $2\times10^{-8} \times \text{rpm}^3$ with a maximum uncertainty of +/- 5% for an 85% confidence level for repeated assembly-disassembly applications. We will use this fit in all subsequent analysis.

Figure 3 Fan dynamometer coupled to test engine

Once the engine was burned-in for 5 hours, the upper ring was changed out with a new one for the intended run. The engine was operated at WOT on the fan dynamometer for 20 ml of the oil fuel mixture to get the rings to seat. This took approximately 8 minutes. The pre-test motored friction was then measured. For this measurement the spark plug and carburetor were removed, and the engine was coupled to a frictional dynamometer. Frictional torque was measured at speeds of 10, 20, 30, 40 and 50 Hz. After measuring the friction, the engine was reassembled and coupled to the fan dynamometer. The speed and temperature were monitored while operating at WOT for 5 hours. Fuel consumption was determined by measuring the time required to consume each 1 liter of fuel/oil mixture. At the end of the run the engine friction was again measured as above. After running with the standard piston and rings, the rings were replaced with a TiC coated set. The engine was operated for 20 minutes and the friction of the engine was then measured as above. After 5 hours operating at WOT on the fan dynamometer, the procedure was again repeated with the Ti coated piston and TiC coated rings. This final combination was again prepared and measured as above, then operated at WOT on the fan dynamometer for 5 hours.

Both before and after the runs the pistons and rings were photographed and elemental composition was measured using a Scanning Electron Microscope (SEM) and X-ray Diffraction (EDX). Additional images and roughness measurements were taken with a 3-D optical metrology microscope.

At the end of the experiment the roughness of the bore was measured with a stylus profilometer, and engine emissions were taken from the engine at WOT on the fan dynamometer.

RESULTS

The speed histories of the three runs are shown in figure 4. In each case the engine operates at 2000 rpm +/- 500 rpm. In general the trend is toward higher speeds with time. The nominal combination experiences the smallest increase from 1900 rpm to over 2000 near the end of the run. A relatively small change was expected as only the top ring was replaced for this run. The lower ring, which had previously been present during burn-in of the engine, was left in place to aid in the determination of the wear rate. The increase in operating speed is more obvious with the fresh ring packs of the 2nd and 3rd runs. The standard piston with 2 new TiC rings begins operation at around 1850 rpm, and experiences an unexpected drop in speed to 1500 at approximately 30 minutes. Some scoring of the piston was noted after tear down, and it is believed that this drop in speed is associated with some form of catastrophic wear event. The engine slowly recovers over the course of the next two hours, and finishes the run with a speed of 2150 rpm. The Ti coated piston with a fresh set of TiC rings starts out with a speed of 2050 rpm, and continues up to over 2400 rpm.

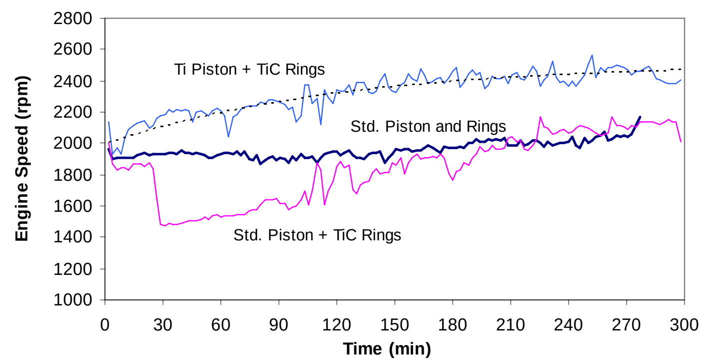

Figure 4 Speed histories of the various runs

Apart form the anomaly in the 2nd run (at 30 minutes), the 2nd and 3rd runs have similar curvatures, indicative of a 1st order system response with a time constant of approximately 120 minutes. This response fit to the data of the 3rd run is also seen as the dotted line in figure 4. This shows that in general the time constant for piston-ring to bore wear on these small engines is approximately two hours. A summary of the initial and final velocities, average power production, fuel consumption, operating cylinder head temperature and overall break thermal efficiency is shown in figure 5.

Piston	Std.	Std.	Ti	
Rings	Std.	TiC	TiC	
Initial	1919	1868	2062	**rpm**
Final	2061	2131	2426	**rpm**
Average Power	151	130	254	**W**
Fuel Consumption	0.073	0.061	0.057	**gm/s**
Temperature	145	144	166	**C**
Effeciency	4.7	4.8	10.2	**%**

Figure 5 Summary pertinent data from the various runs.

Results of the frictional measurements are shown in figure 6. Several observations can be made from this data. Except for the nominal piston and ring set at low speeds, each combination gives higher friction with engine speed. This is similar to what was measured in previous studies.

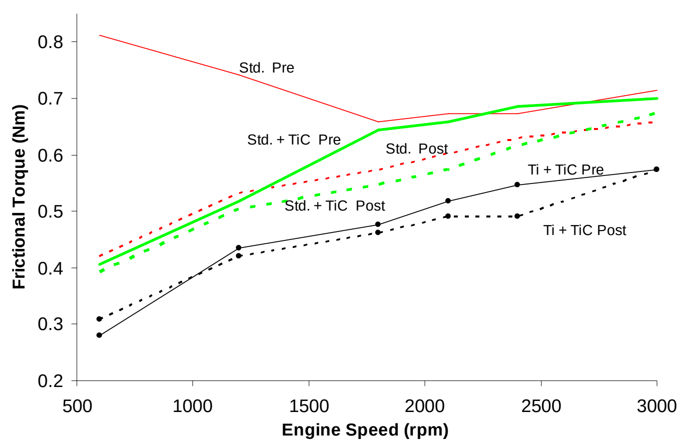

Figure 6 Motored engine friction versus speed for the various piston + ring combinations. Post run data is indicated by dotted lines.

Additionally the correlation between lower friction and higher engine speed can be seen. Taking the friction data at 2100 rpm from figure 6 and plotting it against the corresponding engine speed we get the data of figure 7. The initial engine speed is plotted against the pre-run friction, and the final engine speed is plotted against the post-run friction. Lower engine friction means that more of the indicated power is available as shaft power. This results in an increase in the operating speed of the fan dynamometer. Taking a best fit line from this we can model the increase in engine power as a function of engine friction. This will also allow us to estimate the possible efficiency improvement achievable through friction reduction.

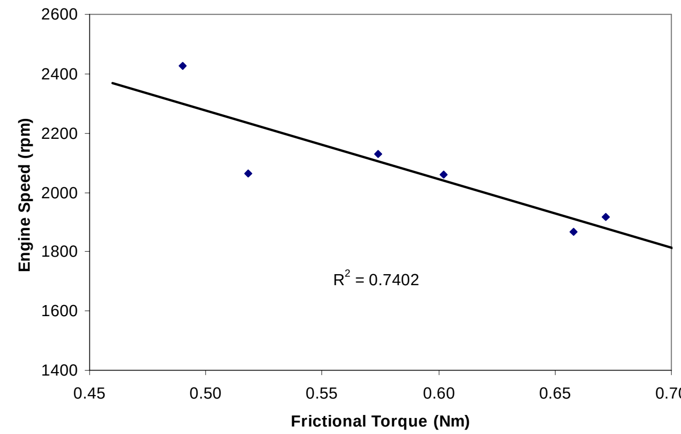

Figure 7 Engine motored torque (sans carburetor and spark) versus loaded operating speed

The effects of burnishing can clearly be seen in the frictional data in figure 6. For the standard piston with both ring sets the post run data shows an approximate 10% decrease in friction, most noticeable at 1800 rpm and above. A final interesting feature of both the frictional data and the speed data is the apparent effect of frictional coatings. Comparing the post-run data we notice that the TiC coated rings give an average of 3.6% less motored friction than the nominal rings, and the coated piston and ring pack give an average of 20% less friction than the nominal set. While the 3.6% improvement for the coated rings is in the expected range, the 20% reduction for the Ti coated piston and TiC rings was a surprise. Metrology gave insight into all of these issues.

MATERIAL ANALYSIS

The piston is an aluminum alloy with large amounts of silicon to enhance the hardness and durability. Trace amounts of other materials are also present. The bore is also an aluminum alloy with large amounts of silicon. The bore is textured with honing scratches for oil retention. This rough surface undergoes significant wear, especially in the area rubbed by the piston rings. The rings are a cast iron substrate with an anti-stick zinc phosphate surface treatment. Figure 8 lists the approximate weight percentages of the various elements as measured by the USM School of Materials X-ray diffraction SEM.

	Ring Base	Nominal coating	TiC Coating	Piston Base	Ti Coating
Fe	76	9			
C	22	54	80		53
Ti			20		35
Al				82	
Si	2			18	
O		16			12
Zn		13			
P		8			

Figure 8 Elemental breakdown of the various components (% wt)

The dominant surface structure of the rings are annular machining marks covered by fine crystals of the original surface coating. Even with proper lubrication the rings experience significant contact with the surface of the bore. Figure 9 is a SEM image of the worn standard ring surface. The dark horizontal striations indicate the approximately 10 µm deep annular machining grooves on the surface of the ring. The fine material in the grooves are zinc phosphate crystals. The lighter areas are plateaus of ring base material where the over lying material has been abraded away by the cylinder wall. Fine vertical scratches indicate the relative motion of the ring against the bore.

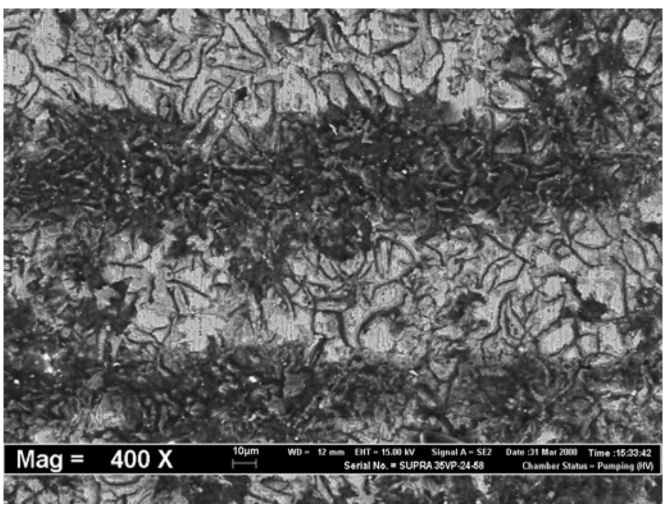

Figure 9 SEM image of ring surface.

Figure 10 shows the virgin surface of a standard ring (top) and the warn surface (bottom). The annular machining marks are clearly seen as the horizontal intensity variations. The TiC coated rings are compared before and after the runs in figure 11. Again a similar mechanism appears to be operating even with the TiC coating, however the wear is less uniform. The TiC coated rings had more wear occurring near the upper and lower edges of the ring, with slightly less wear overall compared to the standard rings.

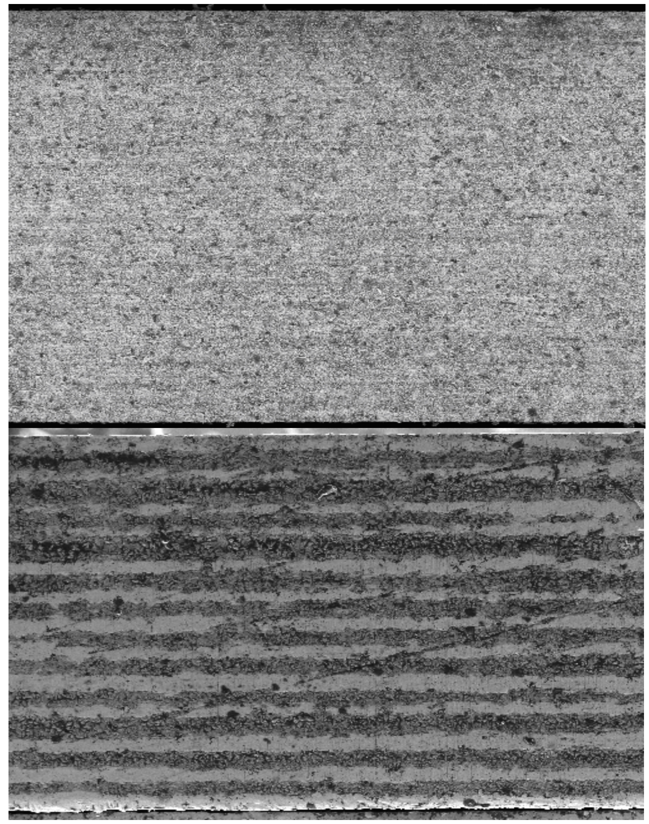

Figure 10 SEM images of the nominal top ring, pre-run (top) and post-run (bottom)

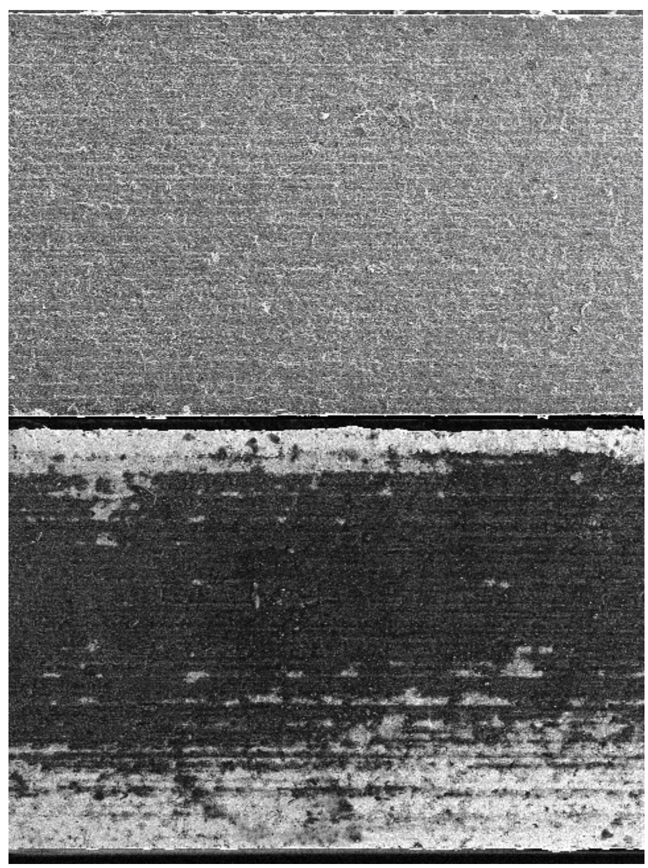

Figure 11 SEM images of the TiC coated rings, pre-run (top) and post-run (bottom)

Surface roughness was analyzed using an Aliconia "Infinite Focus" 3-D optical microscope. All measurements were performed on a large two-dimensional area in the center of the major thrust side face of each ring. Rubbing between the bore and ring faces should abrade the tops of the peaks, reducing surface roughness, and resulting in plateaus. We are therefore taking the surface roughness as a measurement of surface wear. The average surface roughness of the standard and TiC coated rings is compared in figure 12. Surface roughness (Ra) or the standard rings is reduced by approximately 75% in the first 5 hours of operation, and a further 58% in the next 5 hours. The TiC rings wear rate is not as great, suffering only 44% reduction in Ra in the first 5 hours of operation.

Ring	**Std.**	**TiC**	
New	2.13	1.04	µm
5 hours	0.55	0.58	µm
10 hours	0.23	NA	µm

Figure 12 Surface Roughness (Ra in µm) average for rings as a function of run time.

Three dimensional images of the standard ring surface is shown in figure 13, both pre-run and post-run. The annular groves are clearly visible as the "valleys" in the direction of view. The dark areas indicate the original surface finish, and the lighter areas are exposed ring base material. Abrasion of the peak areas of the ring is clearly evident.

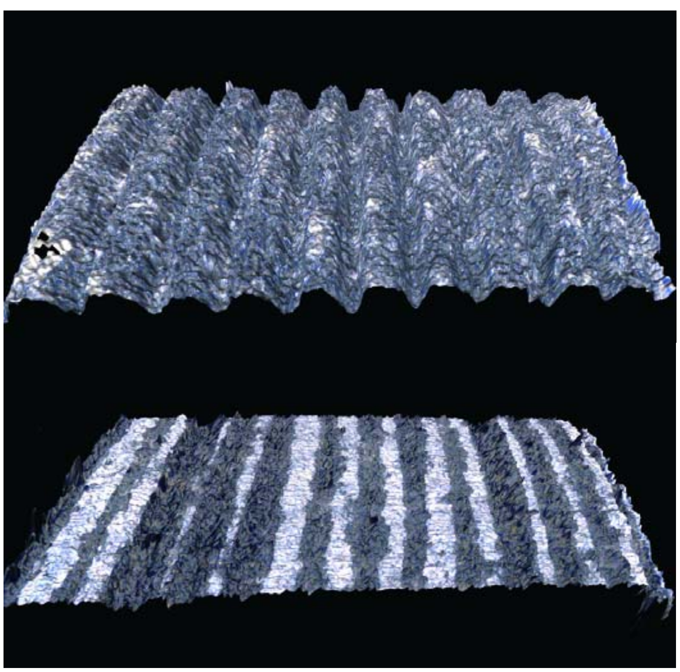

Figure 13 3D surface images from the standard ring, pre-run (top) and post-run (bottom). Top of ring is to the left.

Simple visual inspection of the Ti coated piston indicated significant wear post-run. Under SEM magnification, figure 14, it became clear that the Ti layer was largely delaminated in areas experiencing significant rubbing. Such delamination indicates that the coating was improperly adhered to the surface of the piston. This is often a result of unsatisfactory surface preparation (ie. cleanliness or chemical activation) or incompatible materials structure, which would require a seed layer to insure good adhesion.

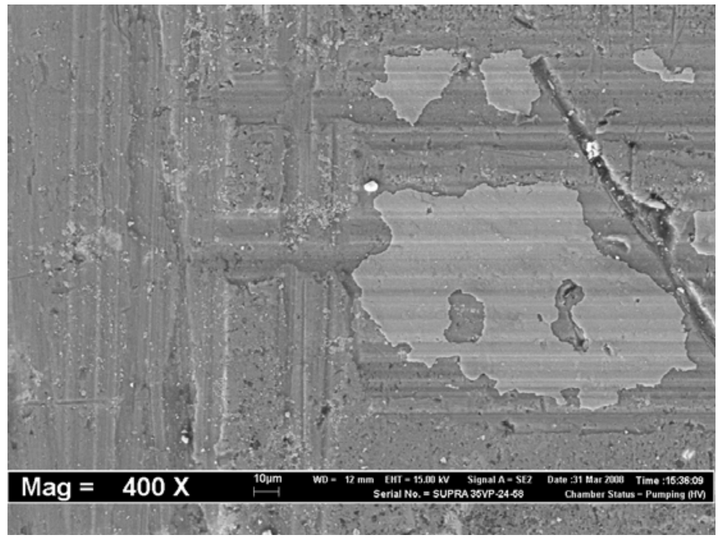

Figure 14 SEM image of the Ti coated piston post run. The large "island" of material on the right is Ti, while the remaining material is piston base material.

The horizontal lines in figure 14 are machining marks, while the vertical marks are scratches from the bore, most likely from the edge of a port. The reduction in friction previously noted was considered unusually large. The piston and ring friction is expected to be about half of the measured motored friction, so the 20% reduction in motored friction would represent approximately a 40% reduction in piston-ring friction. This was larger than anticipated, so further investigation was initiated. Dimensional verification of the piston and rings indicated that despite being a replacement part there were significant differences in the piston, while the rings were confirmed to be functionally identical to those used in the rest of the study. The major difference in the piston was a slightly smaller (0.14 mm) diameter. This gave a looser fit in the bore, and lowered the sliding friction. It also resulted in a higher "cocking" angle, and greater concentration of lateral forces on the top and bottom parts of the piston side. This wear pattern was observed on the piston, and in these areas the Ti coating was completely gone by the end of the run. This dimensional variation is believed to be the major contributor to the observed reduction in friction for the Ti coated piston.

Finally wear was observed in the bore as well. Figure 15 shows a view looking into the inverted bore near the intake port. The intake port is the rectangular feature centrally positioned, and the transfer ports (exposed to the side of the piston) are at the periphery right and left. Honing scratches can be seen on the surface of the bore making an angle of around 15 degrees with the horizon. At Bottom Dead Center (BDC) the rings of the piston just enter the intake port. To the left and right of the intake port at the BDC locations we can observe dark areas indicating wear caused by the rings. Although the bore experiences wear over the whole stroke, the wear is generally worse at the ends of the stroke where piston speeds are reduced and boundary lubrication dominates.

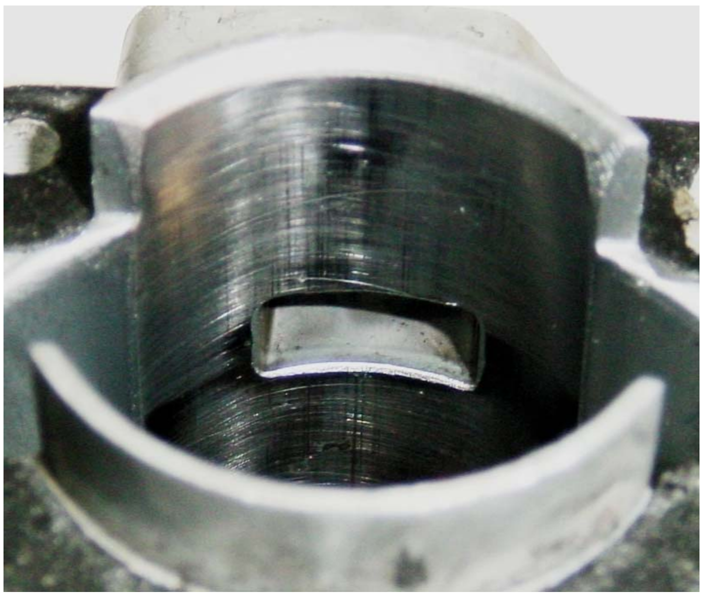

Figure 15 Inside of the bore looking at the intake port (center). Diagonal scratches are honing marks while dark areas (left and right of intake port) indicate wear caused by the rings near BDC.

The roughness of the bore was measured using a Mitutoyo SV-400 stylus profilometer. The surface scan from TDC to below BDC is shown in Figure 16. Notice the increase in roughness below the BDC mark. This indicates an area of the bore not traversed by the rings, which therefore suffers less wear.

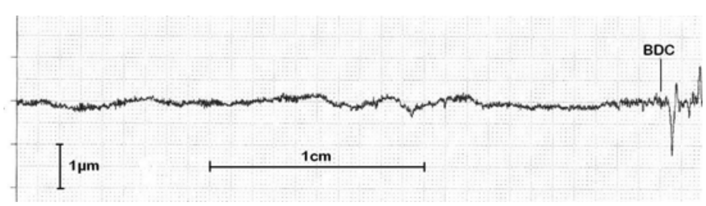

Figure 16 Surface profilometer trace of the bore from approximately TDC (left) to just below BDC (right). The length of the trace is approximately 3cm, and the vertical scale is 1μm per major division.

DISCUSSION

Although the engine speeds were not identical for each of the tests, the setup was. We believe this is gives a realistic comparison of the frictional coatings effects. Utility engines of this type are typically operated at a constant throttle setting (usually WOT) for a given length of time, eg. the shift duration of a highway crew cutting grass. A more efficient engine will be able to cut more grass, or consume less fuel cutting the same amount of grass than a less efficient engine.

While the rate of wear of the TiC coated rings was slightly less than standard rings (44% reduction in Ra compared to 75% for the standard rings) the reduction in friction was a modest 3.6% with the TiC rings. It is therefore believed that the TiC surface coating was not exceptionally successful at reducing friction. Two factors may be contributing to the lack of coating performance. Frictional piston ring coatings are typically very thick, on the order of ten to a hundred microns, while our TiC coating was on the order of 1 micrometer thick. Additionally it was applied on top of the pre-existing anti-sticking coating. It is believed that a thicker TiC applied to a properly prepared substrate would be required to gain the full frictional and wear benefits of such a coating.

Additionally the Ti piston obviously suffered from poor adhesion and better preparation would be required for complete frictional and wear assessment.

EFFICIENCY CALCULATIONS

For a given indicated power, a reduction in friction improves the mechanical efficiency, resulting in a corresponding improvement in break power. For an application requiring a fixed power, this improved mechanical efficiency will be translated into a reduction in fuel consumption. Small carbureted, crankcase scavenged two-stroke engines have exceptionally poor efficiencies due in a large part to fuel short circuiting.

Hydrocarbon (HC) emissions were measured using dispersive Inferred (IR), during testing of our engine. The resulting average emissions were approximately 10,000 ppm HC (hexane). Assuming a volumetric efficiency of 80% with a displacement of 30cc and an average speed of 2000 rpm and air density of 1.18 kg/m^3, we calculate an air flow rate of 0.94 g/sec (after Heywood, 1988). At the measured fuel flow rate (for the nominal piston combination) of 0.0731 g/sec this gives a rich air/fuel ratio of 12.9. Carbureted two-stroke engines are typically tuned rich to improve cooling and ignition stability (Blair 1996). Taking the HC emissions as hexane, and the fuel as C_8H_{18}, we can calculate the approximate exhaust HC flow of 0.028 g/sec. Fuel short circuiting accounts for approximately 0.028/0.731 or 38% of the ingested fuel.

According to Ripin 2008, the total motored friction for a similar engine was approximately 0.50 Nm at 2000 rpm. This results in a frictional power loss of approximately 105W. The distribution of engine power is broken down into various categories in figure 17 for the standard piston and rings. Of the total fuel power, approximately 54% is lost directly as heat, either from the cylinder, or in the exhaust gasses. The next largest loss is fuel short circuiting (HC emissions) which accounts for approximately 38%. Of the remaining power about 60% is available as break power, and 40% is lost to friction.

Power	W	%
Input	3216	100
Heat Loss	1748	54.4
HC Loss	1212	37.7
Break	151	4.7
Friction	105	3.3

Figure 17 Fuel power distribution of the engine.

The engine life time fuel consumption can be calculated by the total operating time multiplied by the operational power rating divided by the chemical energy of the fuel over the efficiency of the engine. Using the previously calculated efficiencies, and assuming a 500 hour life operating at 500W with a fuel chemical energy of 44,000 J/gm the nominal engine should consume a total of 849 liters of fuel. The TiC coated rings would bring this down to 824 liters, saving 25 liters of fuel, or about 25$ at a fuel price of 1$ per liter. Even with such a modest gain in efficiency the savings in fuel cost indicate that a friction reducing coating should be cost effective. Much more impressive are the gains with the Ti coated piston. While the gains in efficiency not solely due to the Ti surface coating, the life time fuel savings of approximately 460 liters of fuel (or 460$) certainly indicates the validity of further investigation.

CONCLUSIONS

Based on the above data we can conclude the following:

Initial burnishing of fresh rings or piston has a time constant of approximately 2 hours of operation. During this time the motored friction tends to drop on the order of 20%.

For standard zinc phosphate coated rings the Ra is reduced by approximately 75% in 5 hours of operation. A TiC coating on the rings can reduce this to a 44% reduction in Ra in the same time.

TiC coated rings can reduce the motored friction by approximately 3.6%, which may reduce the life time fuel cost of the engine by 25$. Care must be taken with surface preparation and coating thickness to insure long-term performance.

The overall efficiency of the engine can be greatly affected by part-to-part variation, even with allegedly interchangeable parts.

FUTURE WORK

Work in this area is continuing including long-term testing of multiple engines for life time cost/benefit analysis of various coatings.

ACKNOWLEDGMENTS

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REFERENCES

1. Ripin, Mohd. Zaidi et al. “Frictional Analysis of a Small Two-Stroke Utility Engine via Tear-Down Testing”, SAE 2007-32-0076

2. Andersson, Peter et al. “Piston Ring Tribology”, Helsinki University of Technology, 2002

3. Bushan, B. et al. “Handbook of Tribology”, McGraw-Hill, 1991

4. Heywood, John B., “Internal Combustion Engine Fundamentals”, McGraw-Hill, 1988

5. Blair, Gordon, “Design and Simulation of Two-Stroke Engines”, 1996

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