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Ranking of Cylinder Liner Materials in Two Stroke Marine Diesel Engines

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Abstract: One of the major prerequisites for an improved combination of cylinder liner material and piston ring material is a good description of the materials tribological performance. Piston rings operate in three different lubrication regimes and the materials should be characterised for all of them before a final selection is made.

A new approach to study the cylinder liner and piston rings (primary drive line in the combustion chamber) is used to characterise five different cylinder liner materials. The utilised test apparatus is working after the block-on-ring principle where the cylinder liner is made into a ring and the piston ring into a block. A short introduction of the test apparatus and its abilities is presented and discussed.

Results from comparison and characterisation of five different cylinder liner materials run with a fixed piston ring material are presented. A preliminary ranking of the materials is given based on the materials tribological performance. The materials are evaluated on basis of friction force, oil film thickness variation, temperature variation and rotational speed.

Keywords: *Piston Rings, Cylinder Liners, Materials, Tribology, Lubrication Regimes, Tribo-Tester*

1. INTRODUCTION

Friction between piston ring and cylinder liner is a major source of power consumption in the primary driveline of two stroke marine diesel engines. Also, the service intervals of large two stroke diesel engine are influenced by the wear of the components. Reduction of the frictional loss requires an excellent knowledge of piston ring and cylinder liner tribology. This knowledge can be obtained by experiments where only the interaction between the piston ring material and cylinder liner material is investigated. By making this isolated investigation it is possible to rank different cylinder liner materials and in a simple way test new materials. In this regard, a test apparatus made after the block-on-ring principle was used for the investigations and the performance of the cylinder liner materials was studied by measuring friction force, oil film thickness, temperature in the contact line and rotational speed of the wheel.

2. SHORT INTRODUCTION TO THE TEST APPARATUS

A special designed test apparatus (presented in [1]) was used to investigate friction between different cylinder liner materials and a fixed piston ring material and wear of the two. The test apparatus is based on the block-on-ring principle. The piston ring is the block and the cylinder liner material is shaped into a ring. The piston ring and cylinder liner materials are delivered by MAN Diesel and they originate from the present engines designed by MAN Diesel SE.

The test apparatus is inspired of the ISO 8251 [3] standard which describes a procedure to test the wear resistance of anodized aluminium and aluminium alloys by mean of an abrasive wheel wear test apparatus. The test apparatus described in this paper is not fully compatible with the ISO 8251 standard but with time it is intended that the developed procedures should be closer to what is described in ISO 8251. Further inspiration was found in [4] where friction and wear in heavy duty diesel engines is reported.

Figure 1. Shows a cross section of the test apparatus. The illustration is taken from the CAD files composed during the development process.

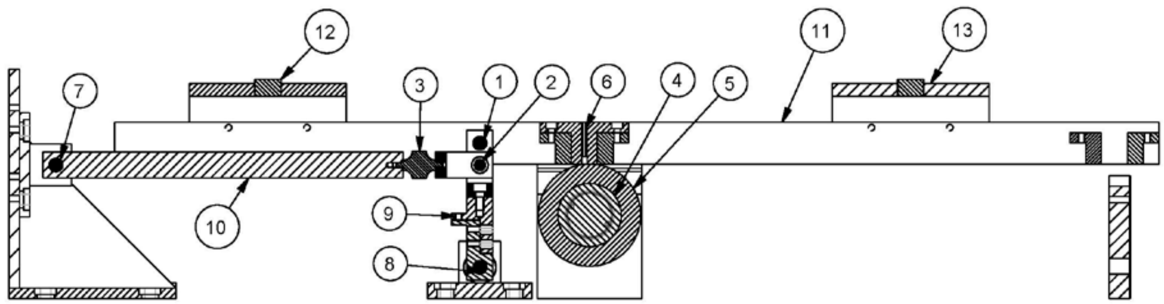

Figure 1. Schematic of the test apparatus. It is seen from the side and as a cross section.

Table 1. Description of the test apparatus as shown in figure 3.

Number:	Description:	Number:	Description:
1	Pivot point for the bridge	8	Pivot point for lower support
2	Pivot point for the force transducer	9	Adjustment screw for adjusting the height of the bridge
3	Force transducer (HBM U9B (0,5 kN))	10	The force transducer's connecting rod
4	Shaft holding test ring and connected to the electrical motor	11	Bridge
5	Test ring made of cylinder liner material	12	Counter weight sledge
6	Hole for the thermocouple. Measure the temperature in the contact line.	13	Loading sledge
7	Pivot point for the force transducers connecting rod.	-	-

Table 1 gives a schematic description of the test apparatus. Only the main components of the test apparatus are shown here. Whereas, other features and capabilities are as follows. The generated friction from the contact between the piston ring block and cylinder liner ring is measured with a force transducer. The oil film thickness is measured with two inductive sensors. The temperature in the contact line/oil temperature is measured with a thermocouple mounted in the bracket for mounting the piston ring block. All signals are collected with a NI-9219 card mounted in a NI-cDAQ 9172 chassis which is connected to a PC by a USB wire. The motor's speed in rpm is measured direct from the motor through a RS232 connection to a PC. The system is sampling synchronous from all channels.

3. RESULTS

Five different cylinder liner materials were tested against a fixed piston ring material. This approach gives the possibility to compare the performance of the cylinder liner materials. Each material was tested with lubricated contact at 200, 100 and 50 rpm and with dry contact at 100 rpm and a normal dead weight load which is increased by step of 20 N after every 10 minutes. The characterisation of the materials is divided into two tests. The first one is a test of all the five cylinder materials shaped into rings with ground/burnished surfaces and likewise is the piston ring block manufactured with a ground/burnished surface. The second one is a test of three of the cylinder liner materials shaped into rings with rough surfaces and a piston ring block likewise with a rough surface. This test setup gives the possibility to investigate how the friction is influenced by the asperity contacts. The outcomes of the two test cases are presented in Fig. 2, Fig. 3, Fig. 4, Fig. 5, Fig. 6 and Fig. 7.

The results obtained during test case number one (covering the experiments with ground/burnished surfaces) are presented in figure 2, 3 and 4 together with table 2. The three graphs, with indices (a), (b) and (c) are those representing the lubricated experiments, and (d) represents the experiment run with a dry contact.

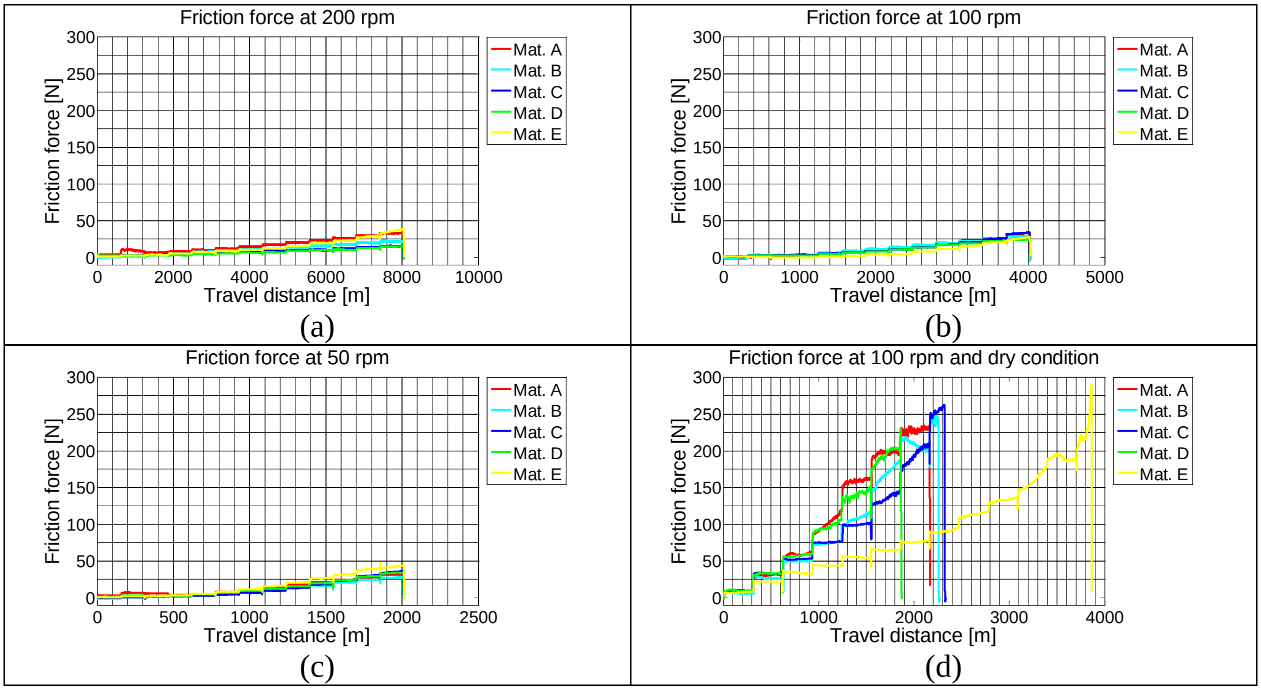

Figure 2. Friction forces measured at 200 rpm (a), 100 rpm (b), 50 rpm (c) and 100 rpm dry contact (d) with ground specimens.

Figure 2. (a), (b) and (c) presents the friction forces obtained at 200, 100 and 50 rpm and oil added for the five materials. No discrepancy between the friction forces measured at the different speeds or materials is clear. The friction forces at the end of the experiments are all in the interval 30 to 40 N which is considered to be the same due to uncertainties.

By evaluating the graphs in figure 2 (d) showing the friction forces measured with dry contact it is clear that material E is fundamentally different from the four other materials[1]. The friction force measured with material E is from the travel distance 0 to 500 meters around the same as for material A to D, but from 500 meters and onwards material E starts to deviate significantly from the other materials. Material E runs for a longer distance (roughly computed 1.75 times longer) than the others. This means, that seizure does not occur with material E in the same scale as with material A to D. Material B and C stand out from material A and D. Material A and D are running for a longer distance and achieving lower friction forces for the same load steps. Material D exhibits the worst behaviour of all the tested materials. Material D runs for only 1.875 meters which indicates that this material is more exposed to seizure when running under dry condition compared to the other materials.

[1] The experiments run with dry contact between the components are continued until the electrical motor’s integrated safety system cuts of the power to the motor due to overload. This is the source to why the different materials do not run for the same distance.

The performance of the materials regarding wear and seizure can also be expressed as a mass loss of the components. In the experiments run here it is only possible to establish the mass loss of the piston ring block and this is only done for the dry condition. The mass losses obtained with the five materials are presented in table 2. Every piston ring block used in the experiments performed at 100 rpm and dry condition was weighed before and after the experiments. By comparing the before and after weight the mass loss of the ring block can be determined. Table 2 shows that material E is experiencing a mass loss which is less than half of the second best material based on the mass loss (material C).

Table 2. Showing the mass loss determined by weighing the piston ring segment before and after each experiment performed at 100 rpm under dry running conditions.

Material	Mass loss [g/20mm]:	Mass loss percent [%]:	Mass loss pr. meter [g/m/20mm]:
A	0,017	0,31	7,82e-6
B	0,0068	0,12	2,99e-6
C	0,0051	0,09	2,19e-6
D	0,0127	0,23	6,77e-6
E	0,00205	0,04	5,29e-7

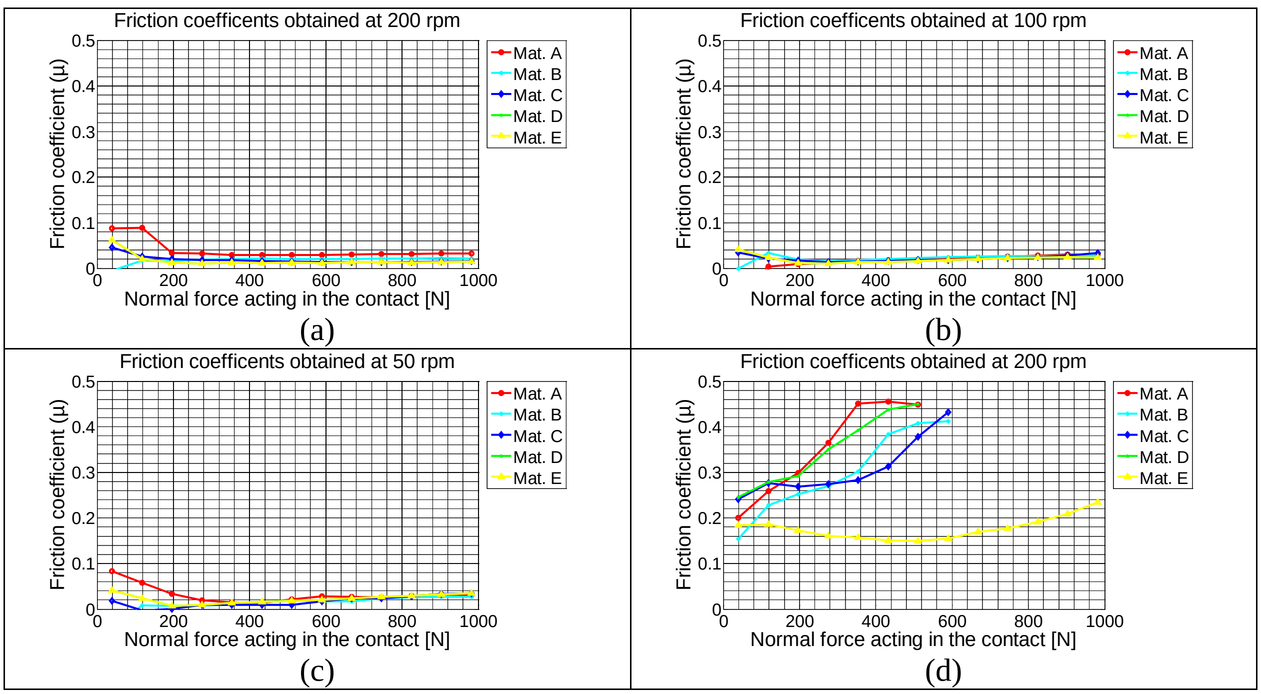

Figure 3. Graphs representing the friction coefficients obtained at 200 rpm (a), 100 rpm (b), 50 rpm (c) and 100 rpm dry contact (d).

In figure 3, the graphs are representing the calculated friction coefficients for the experiments run with ground/burnished cylinder liners ring and piston ring segment. The first three points (interval from 40 to 200N) in figure 3(a), (b) and (c) have to be discarded due to the uncertainties in the measurement. From 200N and onwards the graphs are correct. When figures 3(a), (b) and (c) are compared it becomes clear that the friction coefficients at 50rpm increases steadier than those at 200 and 100rpm. This indicates a higher contribution from hard asperity contacts. From 200N and onwards the friction coefficients are in the interval from 0.01 to 0.03 and these values are considered to be realistic and the five materials are stated to performing equally under lubricated conditions. Presented in figure 3(d) are the graphs for the experiment done at dry running conditions. Under dry conditions the materials differ from each other regarding seizure and thereby friction coefficients which is already established by the explained differences in the measured friction forces. It can be seen in figure 3(d) that material E has a friction coefficient which is up to a factor of 3 lower than for the rest. This is, however, only the case under dry running conditions. When an oil film is present (figure 3. (a), (b) and (c)) the friction coefficients are all the same for the five materials.

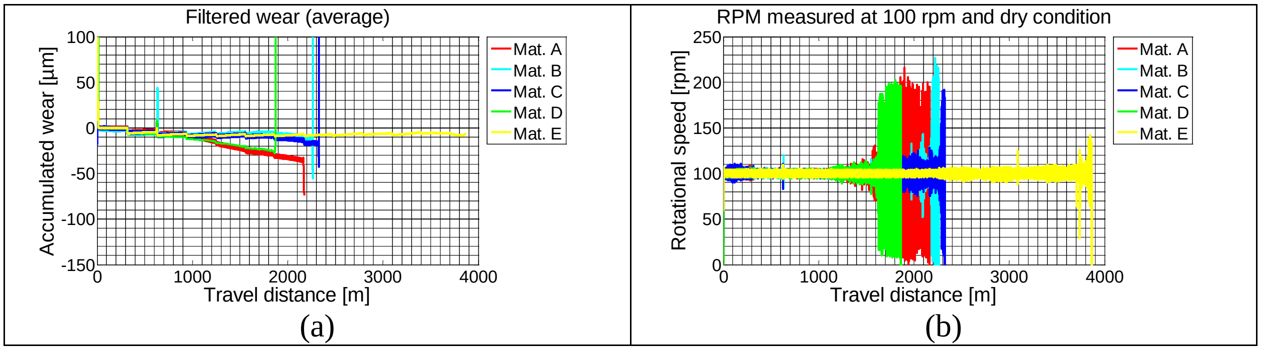

Figure 4. Graphs representing the combined measured wear of the piston ring block and cylinder liner (a) and the rotational speed of the cylinder liner ring (b).

Figure 4(a) shows the graphs drawn on the basis of the measured combined wear[2] of the cylinder liner ring and piston ring block (the accumulated wear). Again is material E standing out from the other materials with a combined wear of 7.5µm. Upon careful observation of the figure 4(a), it reveals that the wear is decreasing after 1200 m of sliding distance. This is unexpected but can be explained by the thermal expansion of the cylinder liner ring together with the high wear resistance of material E.

The second test is carried out with the same materials as material A, B and C in the first test. The change from the first test is that the surfaces of the cylinder liner rings and piston ring segments have higher relative roughness values, Ra. This gives the opportunity to investigate the influence from asperity contacts.

Figure 5 shows the change in the friction forces as the load is increased. Graphs (a), (b) and (c) represent the experiments performed with a lubricated contact between the cylinder liner ring and the piston ring block. There is no significant difference in the three materials performance when oil is present in the contact line and the surfaces are rough. This is equivalent to what was established out from Fig. 2 (a), (b) and (c). The graphs end on approximately 125 N in case (a), (b) and (c). The small deviations between the graphs are ascribed to the variation in amount of oil available in the contact line.

By comparing figure 2(a), (b) and (c), with ground/burnished surfaces, with figure 5 (a), (b) and (c), with rough surfaces, a significant difference in measured friction forces is observed. With the ground/burnished surfaces the friction forces are in the interval 30 to 40 N and with rough surfaces the friction forces are fluctuating around 125 N. This is an increase in the friction forces of a factor of 3 to 5 from ground/burnished surfaces to rough surfaces. This clearly stresses the importance of the surfaces topology. It also stresses the importance of having similarly manufactured components when the materials are ranked with the method developed.

Under dry running conditions there is no pronounced difference in friction force between rough and

[2] Combined wear means: the total wear which takes place of both the piston ring block and test-wheel. The test rig is capable of measuring a distance between the sensor and the steady base plate.

burnished surfaces. This reveals that the topology of the surfaces is influencing the formation of the oil film but it does not have an influence on the materials dry performance when the friction forces are evaluated.

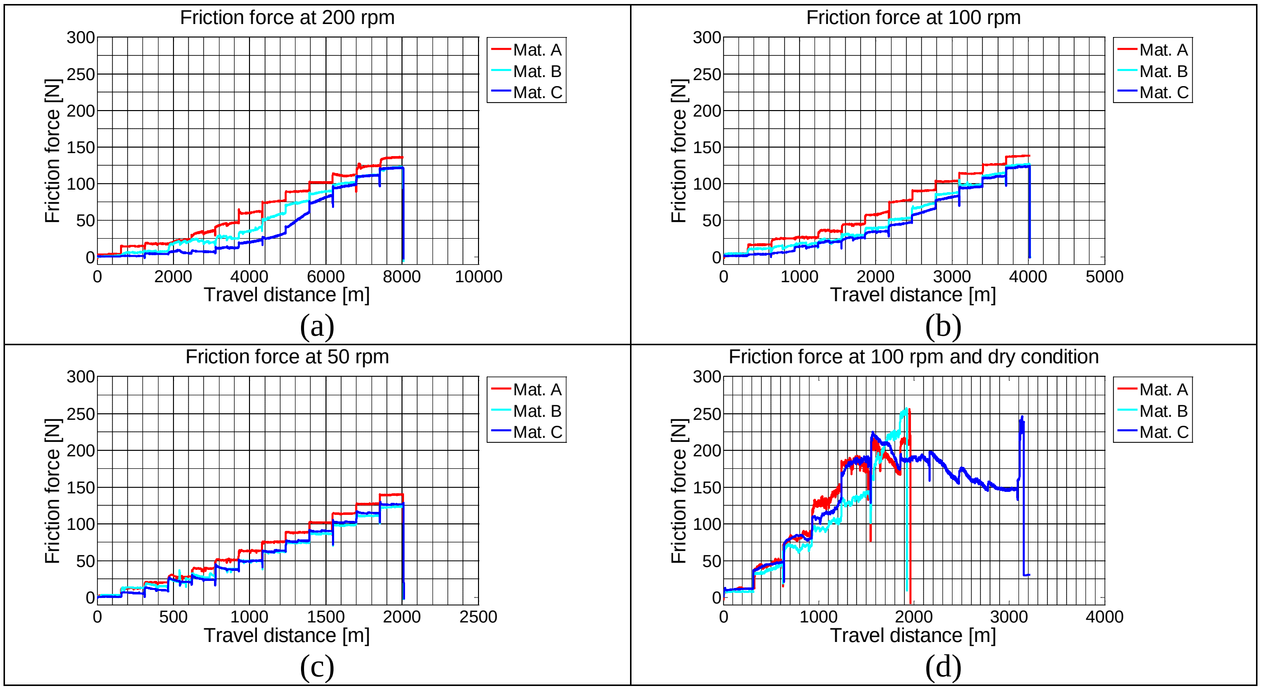

Figure 5. Friction forces measured at 200 rpm (a), 100 rpm (b), 50 rpm (c) and 100 rpm dry contact (d).

The experiments with dry contact between the components are continued until the electrical motor's integrated safety system cuts of the power to the motor due to overload. Figure 5(d) shows the friction forces measured in the case where the components are in dry contact and with rough surfaces. Materials A and C are travelling the same distance. Material C runs for approximately 1.100m more than material A and B. This is due to oil from the construction oozed down on the cylinder ring and thereby destroyed the experiment. The results for material C should therefore be ignored. Probably the pollution started at approximately 1.550m where the curve for the friction force starts to decrease.

Figure 6 presents the friction coefficients calculated on the basis of the measured friction forces and the load step in question. The graphs a, b and c shows that material A has the highest friction coefficient. The graphs also show that the friction coefficients are rising throughout the experiments. This indicates influence from asperity contact which means that the oil film thickness is not sufficient separate the surfaces. The increase in the friction coefficients are larger than those calculated for the ground/burnished surfaces. Since the roughness is smaller for the ground surfaces the required oil film thickness to separate the surfaces is smaller. The friction coefficients for the three materials are in the interval from 0.05 to 0.15. Figure 6(d) presents the friction coefficients from the experiment run with dry contact between the piston ring block and cylinder liner ring and rough surfaces. This graph clearly shows that material B is performing better than material A. Material C is also performing worse than material B from a normal force of 40 N to 350 N but this observation should be made with care due to the oil pollution of the experiment mentioned earlier. The friction coefficients are in the interval from 0.2 to 0.5. This is equivalent to what was obtained with the ground/burnished surfaces.

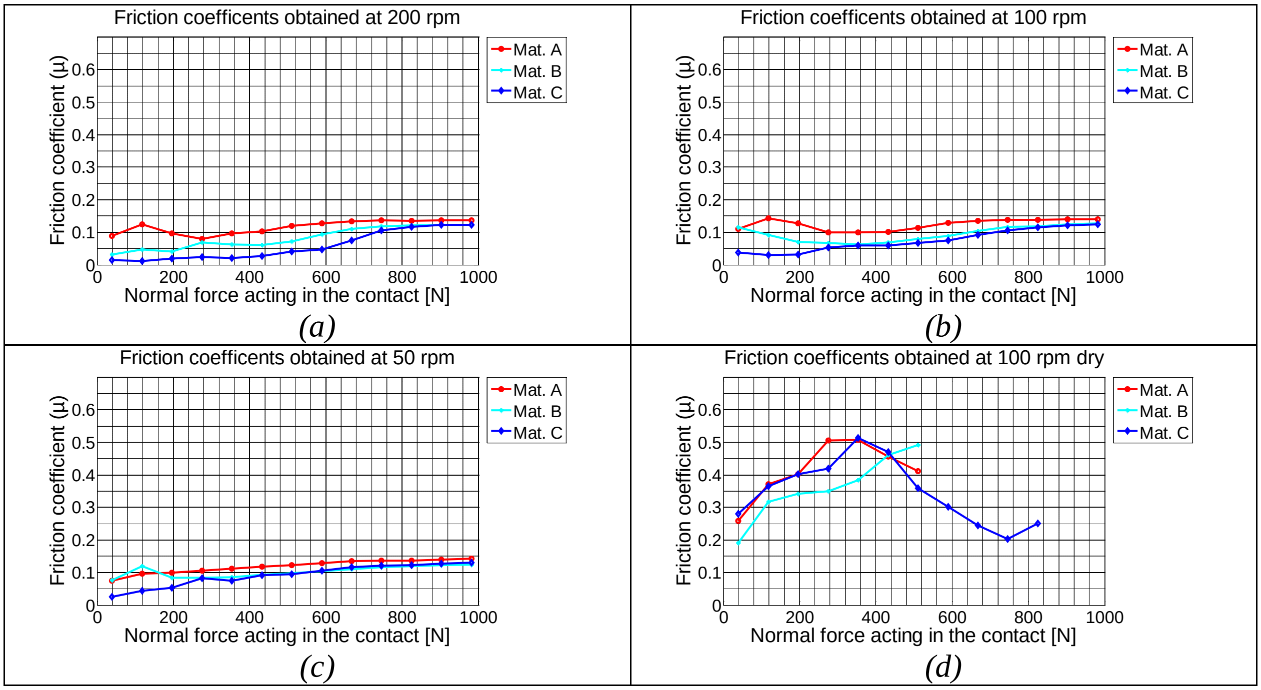

Figure 6. Graphs representing the friction coefficients obtained at 200 rpm (a), 100 rpm (b), 50 rpm (c) and 100 rpm dry contact (d).

Table 3. Showing the mass loss determined by weighing the piston ring segment before and after each experiment performed at 100 rpm and dry running conditions.

Material	Mass loss [g/20mm]:	Mass loss percent [%]:	Mass loss pr. meter [g/m/20mm]:
Mat. A	0,024	0,44	1,22e-5
Mat. B	0,014	0,25	7,26e-6
Mat. C	0,035	0,63	10,87e-6

Table 3 gives an overview over the mass losses obtained with the materials under dry running conditions. The mass losses are given as a mass loss pr. width (20 mm), mass loss percent and mass loss per meter (travel distance) pr. width (20 mm). When evaluating the materials based on the mass loss or mass loss percent material B is performing the best. Due to the entrapped oil entering into the dry experiment the results regarding mass loss should be discarded for material C. Consequently material B is exhibiting the best wear resistance based on all three parameters. When comparing the mass losses for the rough surfaces with the ground/burnished surfaces for material A, B and C it becomes clear that the ranking of the materials are the same but the mass loss for the same material is higher when running with rough surfaces.

Figure 7(a) shows the accumulated wear measurement. This is a measure of how much both the piston ring block and cylinder liner ring are worn all together. The graphs can be utilized to get information regarding incubation time for the materials. Consistent with what is presented in Table 4 material A is worn the most. The new information is that material A also has the shortest incubation time compared to material B.

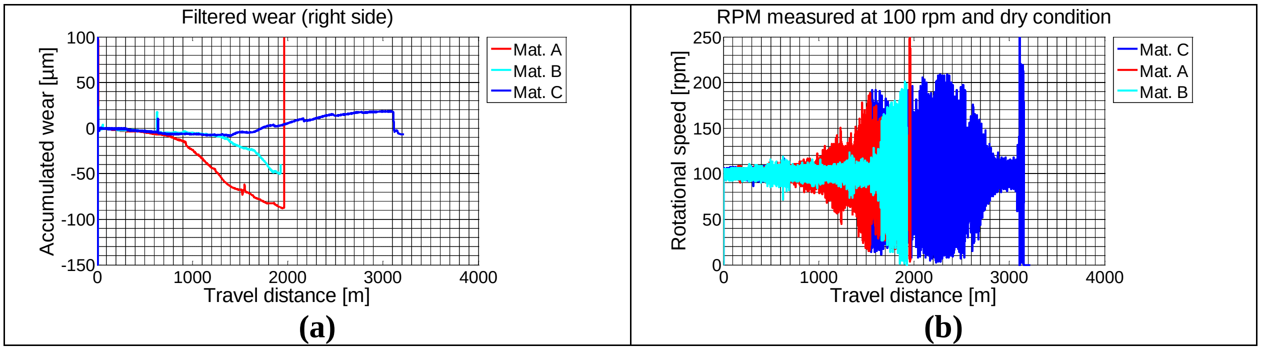

Figure 7. Graphs representing the combined measured wear of the piston ring block and cylinder liner (a) and the rotational speed of the cylinder liner ring (b).

Figure 7 (b) shows the rotational speeds of the cylinder liner rings measured during the experiments at 100 rpm and with dry contact between piston ring block and cylinder liner ring. The graph gives information about how seizure evolves. Material A experiences seizure earlier than material B which is consistent with the observation in figure 7(a) where B has a longer incubation time. What also can be observed is that material B has a faster increase of seizure which indicates the material has poor seizure resistance.

When the five materials with ground/burnished surfaces are compared and ranked based on the above presented results it is clear that material E is ranking as number one. Material E exhibits the best frictional and wear resistant behaviour. The second best material is material B, third is material C and on a shared forth position are material A and D. Material A to D are performing almost equally well meaning that no significant differences in properties are established. But material E is having outstanding properties compared to the four other materials.

When materials A to C with rough surfaces are compared for the same materials with ground/burnished surfaces, it becomes clear that the surface topology of the components is significantly influencing the behaviour of the materials. The three materials have the same ranking as with ground/burnished surfaces (the ranking is in the mentioned order: material B, (material C)[3] and material A). Due to the higher surface roughness the measured mass loss is higher for the rough surfaces and the oil film formation is also influenced in a negative way by the higher surface roughness.

4. CONCLUSION

Five cylinder liner materials are tested with a fixed piston ring material. The five materials are ranked after how they perform when it comes to friction force, friction coefficient and wear.

Three of the materials (material A, B and C) are tested with both ground/burnished and rough surfaces to make it possible to establish the influence of asperity contact.

Based on the two test cases material E is by far the best of the tested materials. Material E exhibit significantly lower friction force, friction coefficient and wear rate. It can also be established that the surface roughness of the cylinder liner ring and piston ring block is of importance for the oil formation, friction force and friction coefficient during lubricated condition.

ACKNOWLEDGEMENTS

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[3] Mat. C is put in parentheses due to the pollution of the experiment and thereby the results obtained with Mat. C under dry condition is not completely true. The parentheses symbolize the uncertainties.

REFERENCES

1. M. T. Pedersen, P. Klit, T. Imran, C. L. Felter, A. Vølund. Novel Tribotester for Cylinder Liner/Piston Ring Contacts of Two Stroke Marine Diesel Engines. In Balttrib 2009: Kaunas, 2009. DTU.

2. M. T. Pedersen. Experimental Investigation of the Tribological Behaviour of Piston Rings. Technical University of Denmark, DTU Mechanical Engineering, Department of Mechanical Engineering. 2009. (Master Thesis)

3. ISO 8251

4. John J. Truhan, Jun Qu and J. Peter Blau. A rig test to measure friction and wear of heavy duty diesel engine piston rings and cylinder liners using realistic lubricants. Tribology International 2005.