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Component test for simulation of piston ring – Cylinder liner friction at realistic speeds

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ABSTRACT

The piston ring cylinder liner contact is a large contributor to mechanical friction losses in internal combustion engines. It is therefore important to have methods and tools available for investigations of these frictional losses. This paper describes the design of a novel component test rig which is developed to be run at high speeds with unmodified production piston rings and cylinder liners from heavy duty diesel engines. A simplified floating liner method is used and the test equipment is developed to fill the gap in between a full floating liner engine and typical component bench test equipment. The functionality and repeatability of the test are investigated and an unexpected behaviour of the twin land oil control ring is found.

1. Introduction

Fuel consumption is of high priority for today's engine manufacturer. The frictional losses in a heavy duty diesel engine (HDDE) are responsible for 2–5% of the fuel consumption in normal driving cycles. For these frictional losses, the piston and piston rings are responsible for approximately half [1]. Therefore it is of great importance to have tools available for evaluation of friction in the interface between piston ring and cylinder liner. In this paper such a tool, a new test rig, is described. The test rig can be used for investigation of effects from different components and running conditions and also for validation of numerical simulation models. In [2], Priest et al. suggested that future progress in simulation of the piston ring contact, specifically with consideration to the complex modelling of the cavitation problem, must be based on combined theoretical and experimental approaches. Many different component test rigs have been previously developed and used [3–10]. These types of test rigs usually operate at low speeds which is not optimal for evaluation of friction benefits that can affect fuel consumption. Low speed test rigs are best used for investigating operation close to reversal zones. Examples of this are [11,12] where different low speed reciprocating rigs were used to simulate wear and scuffing behaviour of piston rings at TDC. According to the author's knowledge, today's fastest operating component test rig is the one developed by Akalin and Newaz [7] which has a stroke of 84 mm and maximum rotational speed of 750 RPM. These component test rigs uses sections of cylinder liners and piston rings, this makes the changing of components very fast and also makes it convenient to vary the load from the piston ring on the cylinder liner. However there are a few drawbacks with these types of set-ups. The alignment between the mating components is crucial and time consuming. Also the loading of the piston ring against the cylinder liner will differ from the real engine and the real ring gap effect will not be represented. There are, however many test rigs that can be used to investigate the friction of the complete piston rings. Among these we find the so-called floating liner engines, often in a single cylinder configuration. Some examples of studies where floating liner test rigs have been used are [13–21]. These test rigs represent the engine very well and advanced systems for measuring oil film thickness and piston ring dynamics in realistic conditions can be added such as in the work by Kirner et al. [22]. However the full scale engines result in rather expensive testing and investigating a variety of different components can be much more time consuming than in a component test rigs. According to Furuhama et al. [13] the main challenge of the floating liner is to prevent the gas pressure from leaking out from the combustion chamber. Another difficulty with the floating liner engine is that the gas pressure and dynamic forces will disturb the friction measurement to some extent. The component test rig described in this paper is developed as something in between the commonly used cylinder liner segment type component test rigs and the floating liner test rig. This gives the possibility to quickly investigate friction between different sets of piston rings and cylinder liners with standard HDDE components at engine like operating conditions.

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2. Design of the test rig

This section describes the fundamental design of the test rig and shows the implemented features. The test rig is designed with the requirement to operate with piston speeds close to those of a typical HDDE, also standard production parts should be mounted without extensive modification or machining. A picture of and a schematic view of the entire test rig can be seen in Figs. 1 and 2 respectively.

2.1. Base crank device

The high speed required will generate large dynamic forces compared to the relatively small friction forces that are measured and therefore it is important to have as little vibrations as possible. The test rig uses an inline six cylinder engine as a crank device due to such low vibrations. In theory, an inline six perfectly balances the first two orders of vibrations which is approximately 98% of the vibrations for most engine designs. The engine selected to act as the base for the test rig is a Volvo B6304 [23]. This engine has a bore of 83 mm, a stroke of 90 mm and an effective con rod length of 139.5 mm. The cylinder head was removed from the engine block and the oil supply that would lubricate the cam mechanism was closed since the rig should be rotated with an electric motor instead of combustion. A shaft with a flange was machined and bolted to the engine crankshaft together with the flywheel.

The crank shaft assembly was dynamically balanced to an unbalance of 0.13*g* at 1300 RPM. The engine block was then connected via the shaft to an electric motor with a rubber tire coupling in order to transmit as little as possible from potential misalignment of the shafts. See Fig. 3 for visualisation of the connection of electric motor to crankshaft. An angular position sensor was mounted on the crankshaft for sampling of the crankshaft angle during testing. Lubrication of the crank device is done with the integrated standard oil pump of the engine. The engine block will not be heated other than from the internal frictional heating, which means that the oil will be close to room temperature during the test. Because of this a special lubricant was used in the crank device. This oil was a fully additivated special low viscosity lubricant. At room temperature the special oil has the same viscosity as the engine's standard motor oil at normal operating temperature.

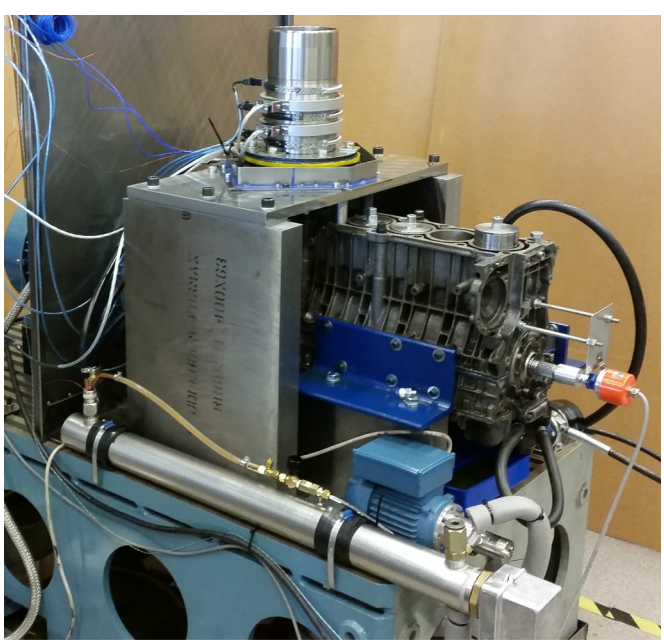


Fig. 1. The test rig.

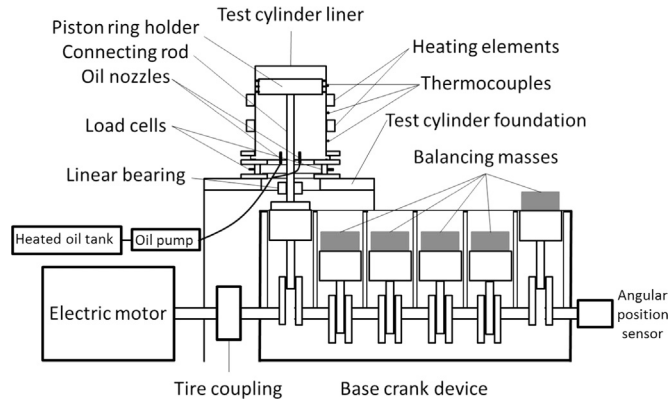


Fig. 2. Schematic view of the test rig.

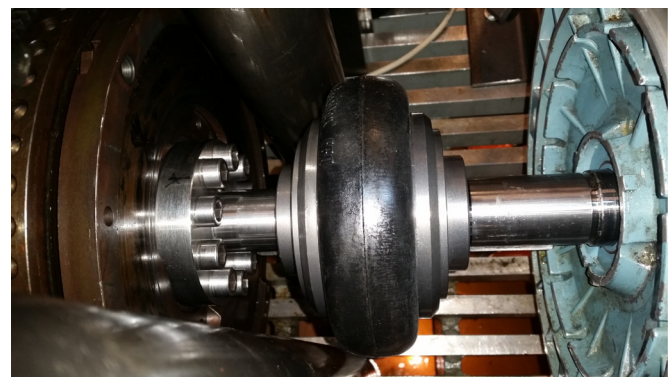


Fig. 3. Coupling of crank device to electric motor.

2.2. HDDE piston ring holder

In order to perform measurements on the HDDE piston rings a steel rod was mounted on the crank device piston closest to the electric motor. On this rod a piston ring holder machined from a HDDE piston was mounted. In order to keep the balance of the engine, extra weight was added to the other pistons equivalent to the weight of the entire piston ring holder assembly with all the piston rings. A thin layer of polyurethane was cast into the top of each of the six pistons to spread the load of the mounting bolts and holes were drilled through the piston top. The rod and the balancing weights were then bolted from inside of the pistons with special washers against the load spreading polyurethane cast in the pistons. Fig. 4 shows the crank device pistons with the entire ring holder assembly and balancing weights mounted.

Since the test rig is supposed to only measure friction in the piston ring – cylinder liner contact a linear guide was machined for the rod connecting the crank device piston with the piston ring holder to keep the ring holder from contacting the liner. The linear guide was made from PTFE filled with 25 vol% carbon fibre with a fibre diameter and length of 10 μm and 150 μm respectively. The linear guide with holder mounted on the crank device can be seen in Fig. 5.

2.3. Cylinder liner assembly

The cylinder liner is mounted upside down in the test rig by clamping the upper part of the liner between two steel discs. The cylinder liner assembly was mounted on three piezo-electric load cells which were mounted on a steel plate. The steel plate which can be moved to centre of the cylinder liner against the piston ring

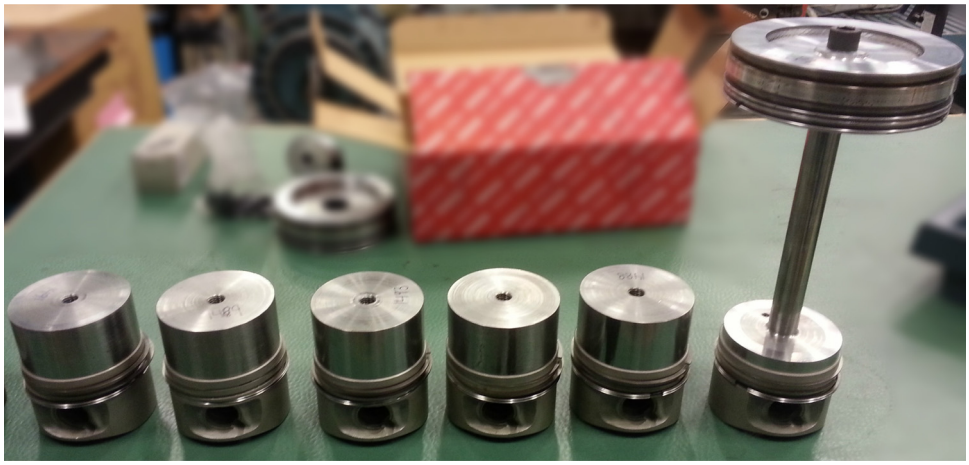


Fig. 4. Crank device pistons with piston ring holder and balancing weights mounted.

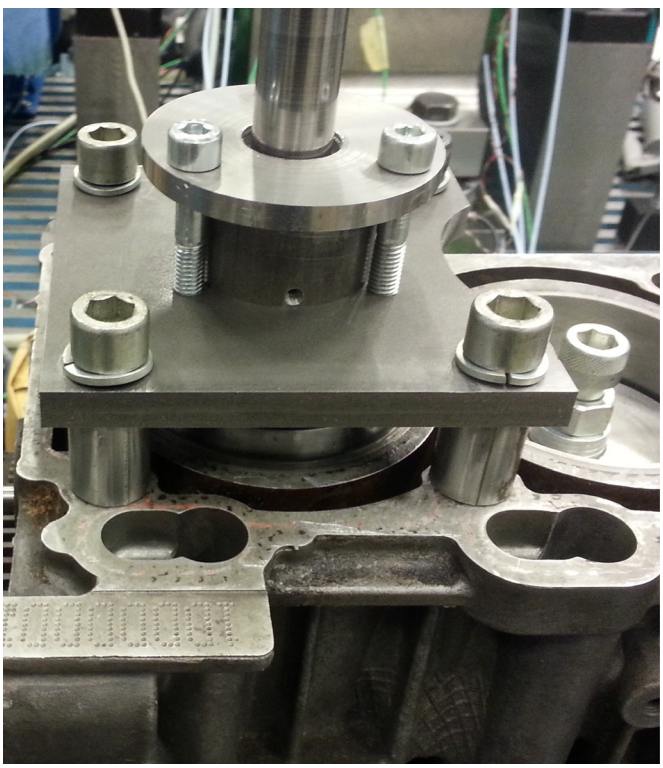


Fig. 5. Linear guide with holder mounted on the crank device.

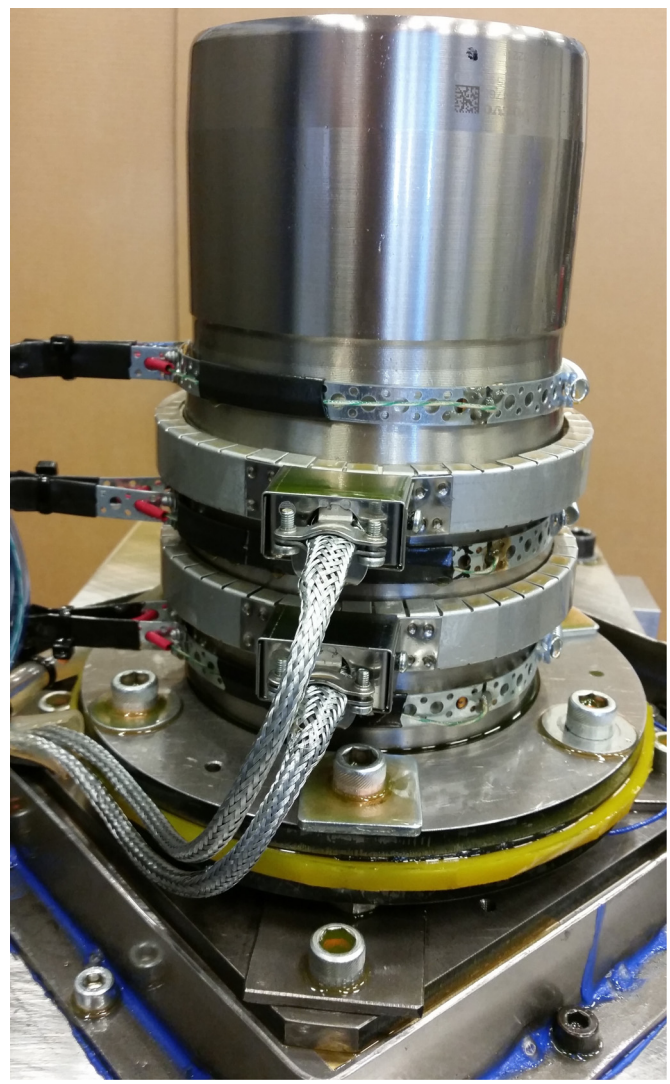


Fig. 6. Cylinder liner with heating elements and thermocouples mounted.

holder is then mounted on a foundation built around the crank device. The cylinder liner assembly is only supported by the load cells thus mimicking the floating liner method of measuring friction.

The HDDE piston rings and cylinder liner are lubricated by an oil spray from underneath of the piston ring holder in order to mimic the lubrication of the piston rings in a real engine. The lubrication system for the HDDE piston rings and cylinder liner is completely separated from the lubrication of the crank device. It consists of a heated oil tank with a pump that distributes oil to the test cylinder which are then lead back to the heated oil tank. The temperature of the oil is regulated by measuring the temperature of the oil close to the exit inside of the spraying nozzle.

The cylinder liner is heated by two ceramic heating elements clamped around the circumference. In total, nine thermocouples are fitted on the outside perimeter of the cylinder liner on three different locations, top-, mid- and bottom-parts of the stroke, three on each stroke location evenly distributed around the circumference. The average temperature around the circumference at the top- and bottom-parts of the stroke is then used for regulation of the heating elements. The thermocouples on the mid-part of the stroke is only used for measurement of the temperature. A cylinder liner assembly mounted on the foundation with heating

Table 1
Components in measuring system.

Component	Type
Thermocouples	Type K
Load cells	PCB 208C02
Signal conditioner	PCB 442B104
Angular position sensor	Fritz Kübler 8.58
Controller	NI cRIO-9068

elements and thermocouples mounted can be seen in Fig. 6.

2.4. Measuring system

Table 1 shows a specification of the components used in the measuring system. The software for temperature control and sampling of data was written in LabView.

3. Experimental parameters

In order to test the capability of the test rig a number of different experiments were performed. In a typical HDDE three piston rings are mounted on each piston, two compression rings at the top and one twin land oil control ring (TLOCR) at the bottom. In one of the test of the test rig capability the amount of piston rings was varied. Tests were performed with three different configurations; both compression rings, only the twin land oil control ring and all of the piston rings mounted. All these tests were performed with the same cylinder liner and piston rings. The test was run at 1200 RPM with the temperature set to 80 °C for both the cylinder liner and the oil spray. All three ring configurations was performed with two different assemblies where the entire cylinder liner assembly and piston ring holder was disassembled and then reassembled in between the tests in order to find the accuracy and repeatability of the test. Hereafter the two different assemblies are referred to as Assembly 1 and Assembly 2. In some of the tests performed at Assembly 1, only the piston ring holder was disassembled and then reassembled without removing the cylinder liner from the rig. These different set-ups are referred to with an additional index as Assembly 1. *i*, where *i* describes how many times the piston ring holder has been reassembled.

A test where the speed was varied from 300 RPM to 1500 RPM with 300 RPM increments was also performed. In this test all three piston rings were mounted on the ring holder. Before running the tests shown in this work, run-in of the components was performed at low speed, 300 RPM, until the friction converged to a constant level.

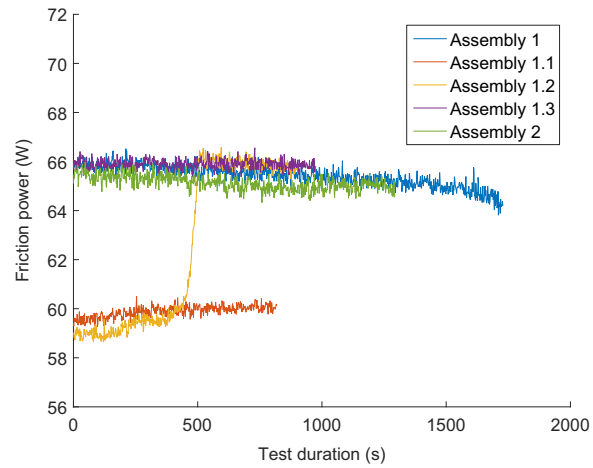


Fig. 8. Friction power as a function of test duration for tests with only TLOCR mounted on the piston ring holder.

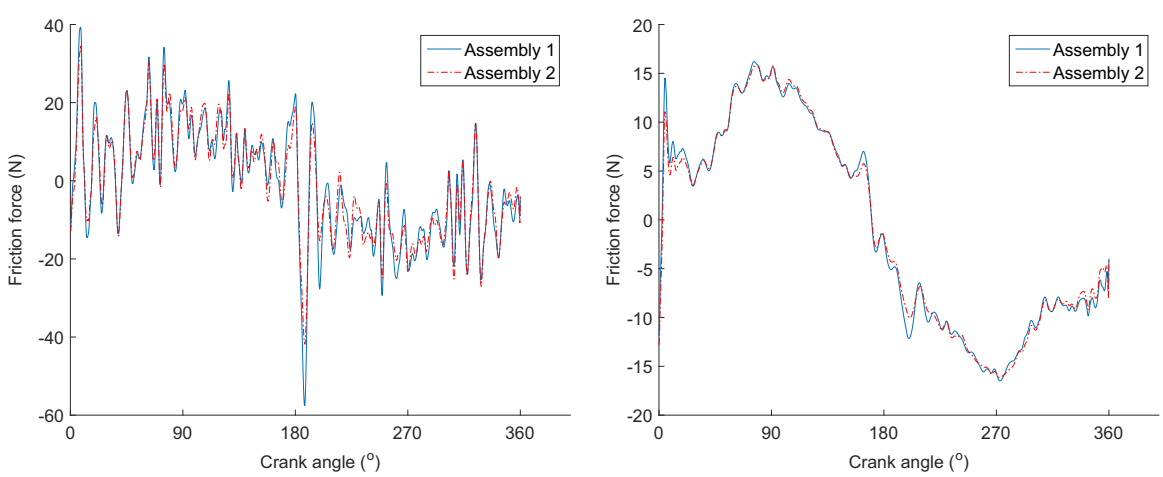
4. Results and discussion

This section shows the results from the measurements. In all test results shown in this section, the initial part of the sampled data is discarded. This is because the friction stabilises sometime after starting the test and comparison between different tests are better made with results from when the friction has stabilised. When filtered data are shown the data is processed with the moving average method which is the same method as used by Kikuchi et al. [14].

4.1. 1200 RPM

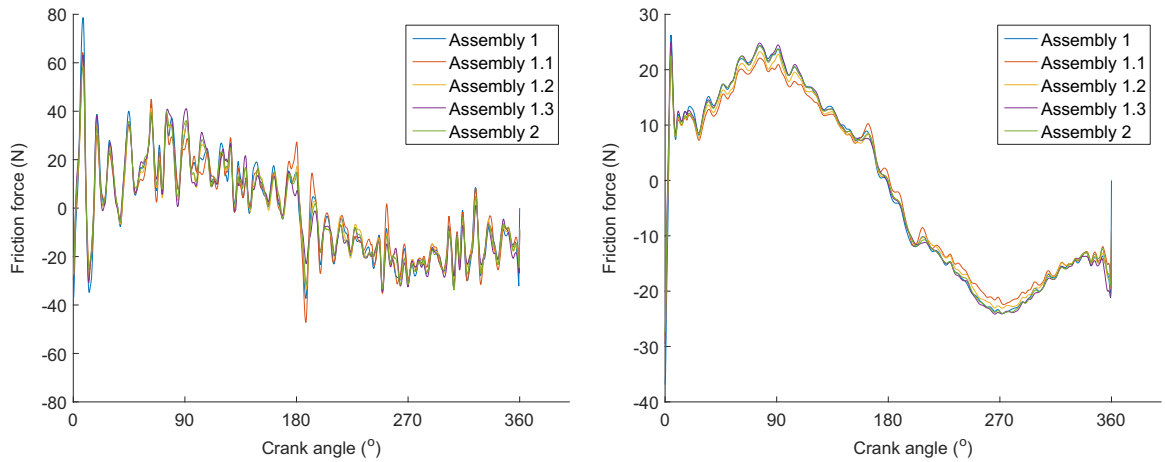
Fig. 7 shows the crank angle resolved friction force for Assembly 1 and Assembly 2 for the two top rings, both sampled raw data and filtered data are shown. The friction force in the figures are the mean force at each location of the stroke for the test. As can be seen in the figure the repeatability is very good in-between the assemblies, especially around the mid-stroke region.

For the test with only TLOCR two different friction levels was observed. Because of this several tests were performed with different set-ups at Assembly 1 with reassembly of only the piston



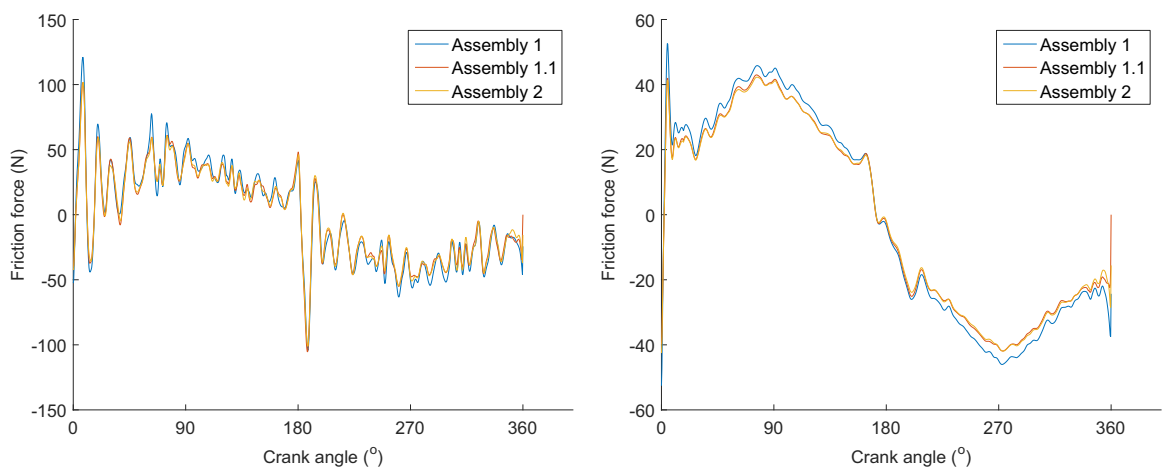
(a) Sampled data (b) Filtered data

Fig. 7. Friction force as a function of crank angle degree for both assemblies with the two top rings mounted on the piston ring holder.



(a) Sampled data (b) Filtered data

Fig. 9. Friction force as a function of crank angle degree with only TLOCR mounted on the piston ring holder.



(a) Sampled data (b) Filtered data

Fig. 10. Friction force as a function of crank angle degree with all of the rings mounted on the piston ring holder.

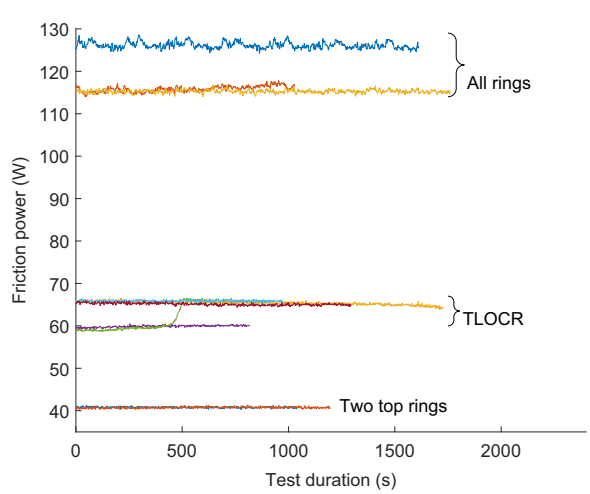
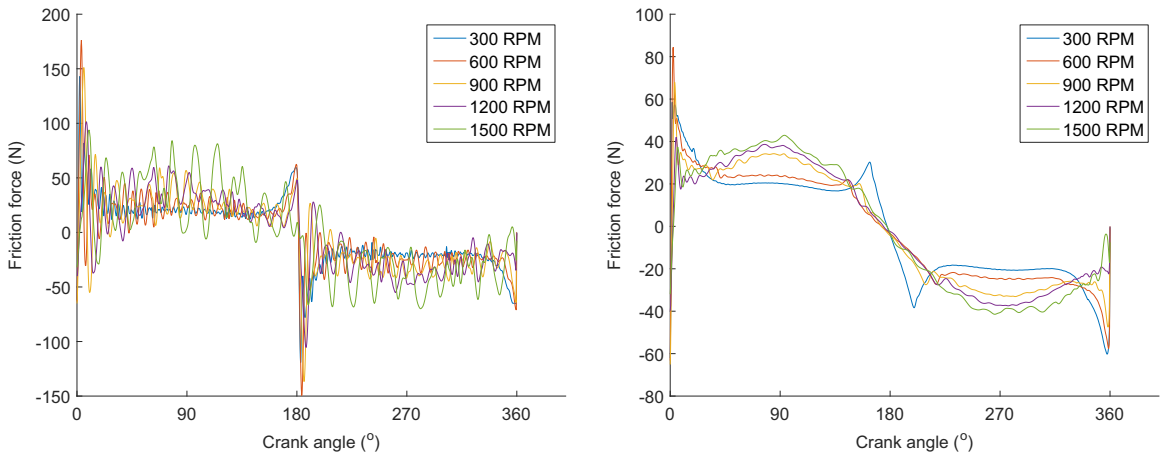


Fig. 11. Average friction power for each stroke as a function of test duration for all of the results shown in Section 4.1.

Table 2
Mean friction power for all of the tests shown in Section 4.1.

Set-up	Mean friction power (W)	Level
Two top rings Assembly 1	40.8	–
Two top rings Assembly 2	40.7	–
TLOCR Assembly 1	65.4	High
TLOCR Assembly 1.1	59.9	Low
TLOCR Assembly 1.2	62.4	Low and high
TLOCR Assembly 1.3	65.9	High
TLOCR Assembly 2	65.1	High
All rings Assembly 1	126.1	High
All rings Assembly 1.1	115.8	Low
All rings Assembly 2	115.3	Low

ring holder. Fig. 8 shows the average friction power at each stroke as a function of test duration for four different set-ups at Assembly 1 and one at Assembly 2. The two different levels of friction for the TLOCR are very distinct and in Assembly 1.2 both levels were observed in the same test. The reason for the different friction levels is believed to be caused by the spring which is forcing the ring against the cylinder liner. The spring can get stuck at one or several of the lubrication holes or ring gap of the TLOCR and therefore not distribute the load evenly around the circumference. Since the friction increased during the test for Assembly 1.2 it seems reasonable to assume that the spring was initially stuck in the TLOCR and then broke free resulting in a more evenly distributed load. If the load is not evenly distributed it could potentially cause the friction to be high at some sections of the ring contact and very low in other sections resulting in a total reduction of friction. With this assumption the low level friction condition would also let



(a) Sampled (b) Filtered

Fig. 12. Friction force as a function of crank angle with all of the rings mounted on the piston ring holder.

more oil pass the TLOCR since it must have a greater mean separation around the circumference compared to the high friction level. Fig. 9 shows the crank angle resolved mean friction force for the TLOCR tests. It can be seen that the friction around midstroke is lower and friction around bottom reversal zone (180° crank angle) are higher for the low friction level test which would strengthen the theory of the load being unevenly spread around the circumference.

The crank angle resolved mean friction force result for the test with all of the rings mounted on the ring holder are shown in Fig. 10. These results are of course affected by the different friction levels of the TLOCR. Also here the piston ring holder was disassembled and reassembled in-between Assembly 1 and Assembly 1.1. It can be noticed that the result for Assembly 1.1 and Assembly 2 is very similar to each other while the result from Assembly 1 shows higher frictional losses. This is believed to be an effect from the different friction levels found for the TLOCR.

In order to compare the difference in friction power from the different ring set-ups, all of the earlier shown results are compiled into Fig. 11. This figure shows the mean value for the average friction power at each stroke as a function of test duration. The mean values for the friction power results shown in this section can be seen in Table 2. From the results it can be noted that the repeatability is superb for the tests with the two top rings, less than 0.3% difference in friction power between the two assemblies. If one considers only the difference at the same level of friction the repeatability is acceptable for the tests with only TLOCR and all of the rings mounted as well. For the TLOCR high level friction power Assembly 2 showed the lowest friction and Assembly 1.3 the highest, the difference between those tests was approximately 1.2%. When comparing the difference for the results with all of the rings at the same level (low) the difference was approximately 0.6%. This is in between the difference for the configuration with two top rings and only TLOCR which would be the expected result. This indicates that the repeatability of this test method is good. However when the TLOCR is mounted the test is less repeatable than with the two top rings but still within a reasonable range for future investigations. Another interesting result is that when adding the highest friction power measured for the TLOCR to the friction power for the two top rings the sum, 106.7 W is still far from the lowest friction power measured with all of the rings mounted at the same time, 112.7 W. Because the friction with all of the rings mounted is higher than the sum of adding the two components, it is indicated that the TLOCR significantly affects the amount of oil available for the two top rings.

When studying the results even further it can be noticed that the difference between the two levels of friction is greater for the tests with all of the rings compared to the test with only TLOCR. This furthermore strengthens the assumption that the spring of the TLOCR is distributing the load unevenly around the circumference during the low friction level tests. Thus more oil is able to pass the TLOCR in the low friction level tests. This results in reduced friction also for the two top rings in the test with all rings. The difference in the friction levels for the different ring set-ups is also another indication that the two top rings are significantly affected by the oil left from the TLOCR. Which again implies that the two top rings operate in starved lubrication conditions in these tests. This would also indicate that when performing numerical simulations of piston ring friction an oil availability model is necessary for the two top rings.

4.2. Varied speed

The sampled and filtered friction force as a function of crank angle with all of the rings mounted on the piston ring holder for different speeds can be seen in Fig. 12. As expected the friction force at mid-stroke increases and decreases close to the reversal zone which indicates that more hydrodynamic lubrication and less boundary lubrication takes place with increased speed.

5. Conclusions

A novel component test rig has been developed, the test rig can operate at high speeds close to actual engine running conditions. Repeatability of the test rig has been investigated and good capability for this was shown. The deviation in friction result from removing and remounting the cylinder liner was shown to be 0.3% when running the two top rings, 0.6% when running all three rings and 1.3% when running only the oil control ring. It was found that the oil control ring is significantly affected by the contact with the spring loading it against the cylinder liner. If the spring sticks in the back of the ring, friction of the oil control ring and oil left for the two top rings are affected. Because the oil left for the two top rings is affected, the friction of the two top rings is also affected by the spring sticking in the back of the oil control ring. The results from the different ring configuration tests furthermore showed that the oil control ring significantly affects the running condition of the two top rings. Because of these results it can be concluded that an oil availability model is necessary when the two top rings are studied by the means of numerical simulations.

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References

[1] Richardson DE. Review of power cylinder friction for diesel engines. J Eng Gas Turbines Power 2000;122(4):506–19.
[2] Priest M, Dowson D, Taylor CM. Theoretical modelling of cavitation in piston ring lubrication. Proc Inst Mech Eng Part C: J Mech Eng Sci 2000;214(3):435–46.
[3] Ting LL. Development of a reciprocating test rig for tribological studies of piston engine moving components – part I: rig design and piston ring friction coefficients measuring method. Technical report. SAE Technical Paper; 1993.
[4] Hill Stephen H, Hartfield-Wünsch Susan E, Tung Simon C. Bench wear testing of common gasoline engine cylinder bore surface/piston ring combinations. Tribol Trans 1996;39(4):929–35.
[5] Dellis PS. Effect of friction force between piston rings and liner: a parametric study of speed, load, temperature, piston-ring curvature, and high-temperature, high-shear viscosity. Proc Inst Mech Eng Part J: J Eng Tribol 2010;224(5):411–26.
[6] Dearlove Janice, Cheng Wai K. Simultaneous piston ring friction and oil film thickness measurements in a reciprocating test rig. Technical report. SAE Technical Paper; 1995.
[7] Akalin Ozgen, Newaz Golam M. A new experimental technique for friction simulation in automotive piston ring and cylinder liners. Technical report. SAE Technical Paper; 1998.
[8] Bolander NW, Steenwyk BD, Sadeghi F, Gerber GR. Lubrication regime transitions at the piston ring–cylinder liner interface. Proc Inst Mech Eng Part J: J Eng Tribol 2005;219(1):19–31.
[9] Spencer Andrew, Avan Emin Yusuf, Almqvist Andreas, Dwyer-Joyce Rob S, Larsson Roland. An experimental and numerical investigation of frictional losses and film thickness for four cylinder liner variants for a heavy duty diesel engine. Proc Inst Mech Eng Part J: J Eng Tribol 2013;227(12):1319–33.
[10] Johansson Staffan, Nilsson Per H, Ohlsson Robert, Rosen Bengt-Göan. Experimental friction evaluation of cylinder liner/piston ring contact. Wear 2011;271(34):625–33 [The 12th international conference on metrology and properties of engineering surfaces].
[11] Obert Petra, Müller Torben, Füßer Hans-Jürgen, Bartel Dirk. The influence of oil supply and cylinder liner temperature on friction, wear and scuffing behavior of piston ring cylinder liner contacts – a new model test. Tribol Int 2016;94:306–14.
[12] Olander Petra, Jacobson Staffan. Scuffing resistance testing of piston ring materials for marine two-stroke diesel engines and mapping of the operating mechanisms. Wear 2015;330:42–8.
[13] Furuhama Shoichi, Sasaki Shinichi. New device for the measurement of piston frictional forces in small engines. Technical report. SAE Technical Paper; 1983.
[14] Kikuchi Takashi, Ito Shinichiro, Nakayama Yoshinori. Piston friction analysis using a direct-injection single-cylinder gasoline engine. JSAE Rev 2003;24(1):53–8.
[15] Kohashi Ken-ichi, Kimura Yuichiro, Murakami Motoichi, Drouvin Yann. Analysis of piston friction in internal combustion engine. Technical report. SAE Technical Paper; 2013.
[16] Kunkel Sebastian, Werner Martin, Wachtmeister Georg. Setting up a measuring device to determine the friction of the piston assembly. SAE Int J Mater Manuf 2011;4(2011-01-0227):340–51.
[17] Gore Michael, Theaker Michael, Howell-Smith Sebastian, Rahnejat Homer, King Paul D. Direct measurement of piston friction of internal-combustion engines using the floating-liner principle. Proc Inst Mech Eng Part D: J Automob Eng 2014;228(3):344–54.
[18] Law Theo, MacMillan David, Shayler Paul J, Kirk Geoff, Pegg Ian, Stark Roland. A new floating-liner test rig design to investigate factors influencing piston-liner friction. Technical report. SAE Technical Paper; 2012.
[19] Liao Kai, Chen Haijie, Tian Tian. The study of friction between piston ring and different cylinder liners using floating liner engine – part 1. SAE Technical Paper; April 2012.
[20] Chen Haijie, Liao Kai, Tian Tian. A numerical and experimental study of twin-land oil control ring friction in internal combustion engines part 2. SAE Technical Paper; April 2012.
[21] Sato Oki, Takiguchi Masaaki, Takayuki Aihara, Seki Yulchi, Fujimura Kazuhiro, Tateishi Yukio. Improvement of piston lubrication in a diesel engine by means of cylinder surface roughness. Technical report. SAE Technical Paper; 2004.
[22] Kirner C, Halbhuber J, Uhlig B, Oliva A, Graf S, Wachtmeister G. Experimental and simulative research advances in the piston assembly of an internal combustion engine. Tribol Int 2016;99:159–68.
[23] Larsson Tor, Bergstrom Kjell ac, Hinderman Terry, Hauptmann Lars-Gustaf, Wennstrom Lars, Denbratt Ingemar, et al. The volvo 3-litre 6-cylinder engine with 4-valve technology. Technical report. SAE Technical Paper; 1990.