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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 0.13g 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.

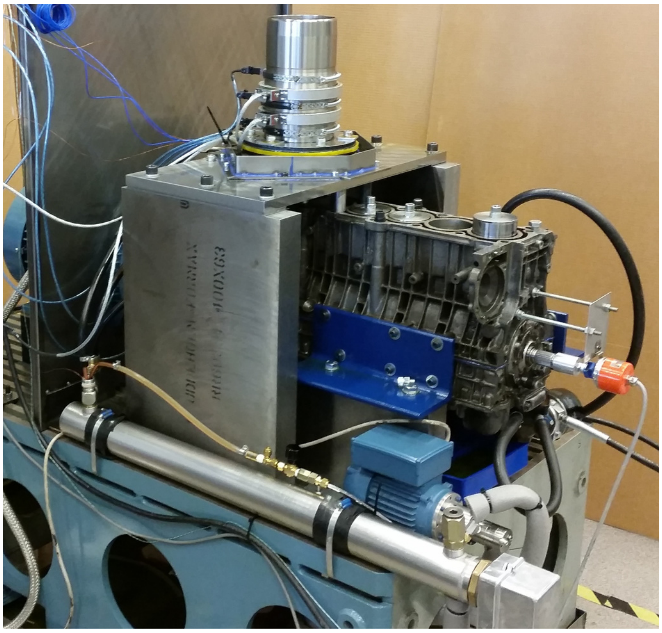

Fig. 1. The test rig.

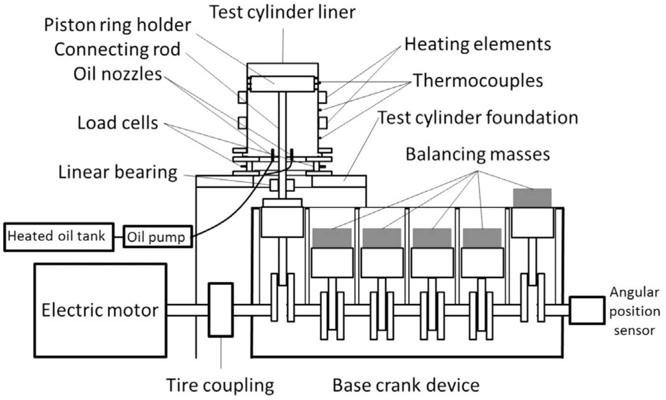

Fig. 2. Schematic view of the test rig.

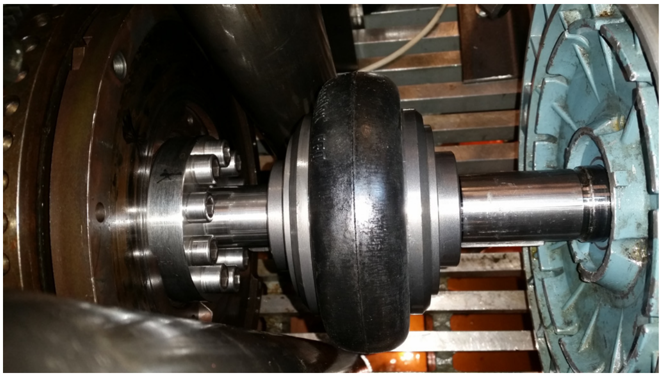

Fig. 3. Coupling of crank device to electric motor.

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