ASSESSMENT OF POSSIBILITIES OF TESTING THE HIGH POWER ENGINE PISTON RING-CYLINDER LINER COLLABORATION ON A MODEL TEST STAND

Wojciech Serdecki

Institute of Combustion Engines and Transport
Poznań University of Technology
Piotrowo Street 3, 60-965 Poznań
tel.:+48 665 2243
e-mail: wojciech.serdecki@put.poznan.pl

Piotr Krzymień

Institute of Combustion Engines and Transport
Poznań University of Technology
Piotrowo Street 3, 60-965 Poznań
tel.:+48 665 2239
e-mail: piotr.krzymien@put.poznan.pl

Abstract

The design of piston rings for modern engines of low and medium power guarantees long reliable operation, thanks to proper selection of materials, suitable geometry of mating surfaces as well as the use of lubricating oils of appropriate quality. At present, the efforts of constructors focus on other aspects of operation of this kinematic pair. The aim is to reduce costs of engine run as well as improvement of ecologic indices (eg. by the reduction in lube oil consumption, reduction of friction losses through the reduction in ring number, and so on).

There still occur failures caused by incorrect collaboration of piston rings and cylinder liners on engines of high output, especially the marine ones. Higher mechanical and thermal loads of crank mechanism elements as well as difficulties with supply and correct distribution of lubricating oil over the liner entire surface are among the causes of these phenomena.

Because of limited measurement possibilities the research on correct collaboration of piston ring and cylinder liner could be quite difficult and expensive on a real marine engine. Such research could be carried out far easier on a model simulation test rig.

The presented paper will deal with the question if the test rig purposed for the investigation of kinematic pair collaboration - property of the Combustion Engine Chair - is suitable for evaluation of collaboration of marine engine piston ring with cylinder liner in the presence of lubricating oil.

Keywords: *marine engine, piston ring, oil film*

1. Introduction

High power two-stroke engines are at present (and as it can be forecast, in the near future as well) the basic propulsion units of great merchandise ships. They are low speed engines (rotational speed of about 100 rpm) of specific power of order of few hundred kW to few MW per cylinder (some technical data of MAN B&W propulsion engines have been collected in Tab. 1). There is a tendency of ship engine construction with higher S/D ratio. Quite recently the value of this ratio was about 2, whereas the S/D ratio of to-day long-stroke diesels excesses 4 (for comparison reasons, S/D = 0.7-1.1 is characteristic for contemporary gasoline engines and 0.8-1.15 for small diesels including engines for generating sets). One of the advantages of long-stroke engine is a high value of piston average speed that facilitate to maintain a continuous oil film (which eventually allows for reduction in friction losses).

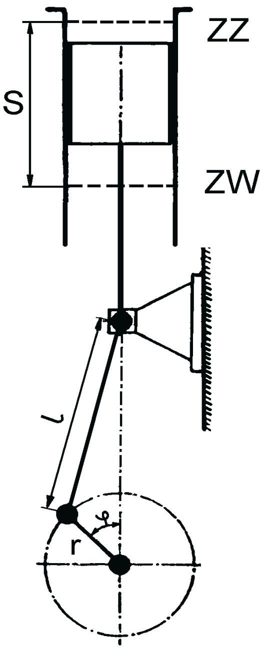


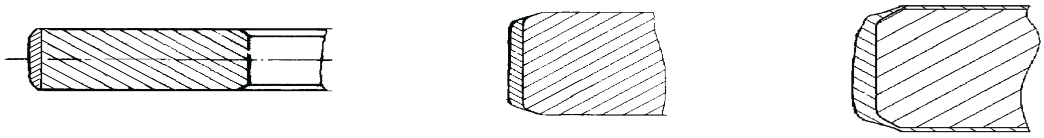
Fig. 1. Schematic of marine engine crank mechanism

Popularity of two stroke high power engines results from their high efficiency (exceeding 50%) as well as possibility of consumption of inexpensive, poor quality fuel. Most important disadvantages of such fuels are: high viscosity (fuel requires special treatment before injection) and high contamination with various impurities including sulphur.

Tab. 1. Some vital technical data of low speed marine engines [3]

Engine type	**L35MC**	**S50MC-C**	**S70MC-C**	**K80MC-C**
Cylinder diameter D [mm]	350	500	700	800
Stroke S [mm]	1050	2000	2800	2592
k ratio (S/D)	3.0	3.82	4.00	4.00
Rotational speed [rpm]	210-178	127-95	91-68	93-70
Specific power [kW/cyl]	650	1580	3110	3640
λ ratio	0.42	0.46	0.44	0.44
Piston average speed [m/s]	7.35	8.46	8.49	8.10

Adjustment of engine functional subassemblies, first of all the crank-piston set, for high mechanical and thermal loads is necessary for achievement of required reliability. The efforts of R&D centers are focused on proper design, manufacturing and diagnostics of parts operation, rings in particular, that play the important role in a correct run of the crank mechanism. As the tests have revealed, piston rings react quickest to any incorrectness. In order to improve the piston durability its face is covered with protective layer (in the case of the first compression ring it is usually chrome or chrome ceramic) that counteracts the excessive wear. Also the chrome plating of ring grooves is being applied [2]. Beside the compression-oil rings the bronze leading rings installed in bottom part of piston (2 or 4 rings dependent on piston height) can be encountered on marine engines. Despite the reduction in ring number noticeable in recent years, the remaining rings guarantee a required tightness of combustion chamber thanks to the use of advanced materials. Similar tendencies in ring design can be observed in four stroke generator engines (see Tab. 2). Their face is provisionally formed and covered with protecting layer (see Fig. 2) and their number has been diminished substantially over the period of last few decades. In the nearest future one should expect a reduction in ring height, which assures better contact to cylinder and better tightness of combustion chamber.



a) b) c)

Fig. 2. Profiles of gen-set engines piston rings: a) AL20, b) L23, c) L21

Tab. 2. Basic technical data of diesels purposed for generating sets [3]

Engine type	L16	L23	L21
Cylinder diameter D [mm]	160	230	210
Stroke S [mm]	240	300	310
k ratio (S/D)	1.5	1.30	1.47
Rotational speed [rpm]	1000	720	900
Specific power [kW/cyl]	90	130	190
λ ratio	0.25	0.25	0.23
Piston average speed [m/s]	8.0	7.2	9.3

Quite a separate problem is an evaluation of cylinder liner lubricating correctness. In the case of marine engines separate lubricating systems supplied with oil of particular properties are being applied for cylinder lubrication. Modern marine engines are equipped with the Alfa electronic lubricating system which assures the oil dose supplied to the cylinder surface relative to the actual operating conditions. Use of this system allowed to reduce the oil consumption to the level of 0.7-0.8 g/kWh.

In the case of generator engines the control of oil dose supplied to cylinder surface is far more complicated, because oil mist lubrication is used in such engines. This makes the monitoring of oil film formation between collaborating parts of piston-crank mechanism extremely important.

However, research on phenomena accompanying formation of oil film on cylinder surface during piston movement encounters a number of obstacles, mainly of the measurement nature. The most important are: installation of different type sensors (temperature, oil film layer thickness, pressure) directly in parts of piston-cylinder assembly (in rings or piston), and still not fully solved problem of signal transmission to the measuring devices. Because of that it is much easier to carry out such research on a model or on individual elements. There are mathematical and material models of piston-cylinder set alike. Anyway, it should be stressed that the representation of phenomena occurred in the assembly is merely the approximation which should be verified in a course of experiments carried out on a real object, e.g. marine engine or its model (alternatively on a model representing operation of selected engine subassembly). Such a stand has been constructed in the Chair of Internal Combustion Engines of PUT, which allows to model the phenomena occur in engine piston-cylinder assembly and eventual evaluation of collaboration conditions like contact geometry, kinematics and dynamics of load as well as the events accompanying lubrication. Further part of this paper concerns the stand design and assessment if tests on piston ring and cylinder surface collaboration could be successfully carried out.

2. Model stand for tests on ring and cylinder collaboration

The stand has been designed purposely to test ring-cylinder collaboration, including precise measurement of oil film parameters between mating surfaces.

Construction of the stand quite substantially differs from a typical piston-cylinder set of the engine. On the presented stand the oil film is generated between a motionless slat representing the section of straight ring and a plate representing the flat cylinder surface installed on a movable cart. The hydraulic oil is supplied to the governing valve. The pressure accumulator installed between pump and valve maintains a constant pressure in the system. Depending on the position of governing valve slide oil is pressed to the left or right side of actuator's piston which eventually is transformed to the cart's motion.

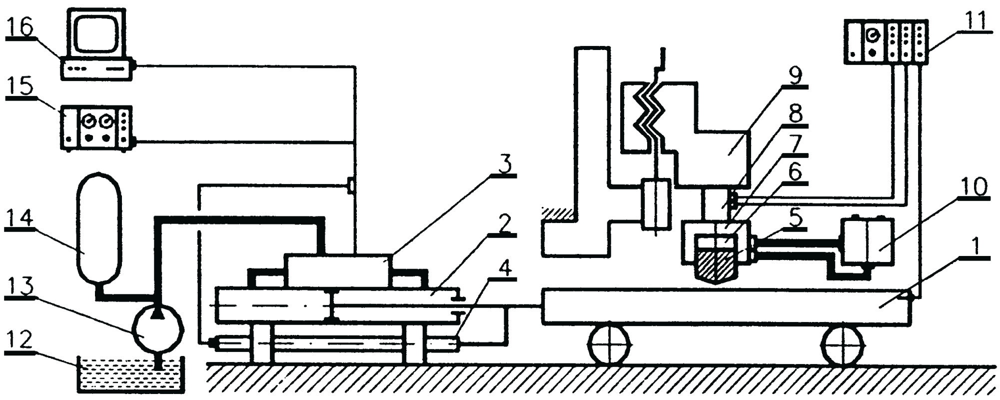


Fig.3 . Schematic of test stand; 1 - plate, 2 - actuator, 3 - control valve, 4 - position resistor, 5 - slat, 6 - flat spring, 7 - model of piston, 8 – force transducer, 9 - support, 10 - slat load, 11 - measuring-control system, 12 - reservoir of hydraulic oil, 13 - oil pump, 14 - pressure accumulator, 15 - analogue control circuit, 16 - digital control

A flat slat representing the piston ring slides over the plate covered with lubricating oil. Unlikely in real engine, the oil layer is not renewable, i.e. fresh oil does not come in consecutive strokes. The slat is mounted in a groove cut in a flat model of piston, hanged on a support and is pressed against collaborating surface with a spring (or a hydraulic servomotor controlled with a special electronic device). A number of slats of various face profiles corresponding to the real ring can be installed in a grip at the same time.

An analogue or digital circuit is used for plate movement control. The analogue one guarantees yield of reciprocating movement for stepwise parameter values (speed, stroke, λ ratio). The digital control uses a computer program which allows plate to move arbitrary, including the reciprocating move characteristic for combustion engines.

As mentioned before, the test stand allows a continuous measurement of oil film parameters generated between steady slat representing piston ring and a movable plate representing cylinder surface as well as other features of this system. Thanks to the location of sensors in slat and in grip as well other measurements are possible:

- thickness of oil film layer between slat face and plate surface at the position of sensor; thanks to two sensor sets (active and passive ones) on slat both ends it is possible to monitor the slat transversal movement,
- friction force accompanying the slat movement,
- slat pressure against plate surface.

Test stand analogue or digital control allows selection of:
- angular speed ω, in a range from 0 to 12 rad/s,
- piston stroke S, in a range from several millimeters to 0.8 m,
- the λ ratio, in a range from 0 to 0.3 (0.5),
- slat pressure against plate surface; this pressure can be constant (its value depends only on characteristics of spring) or adjustable by the hydraulic actuator.

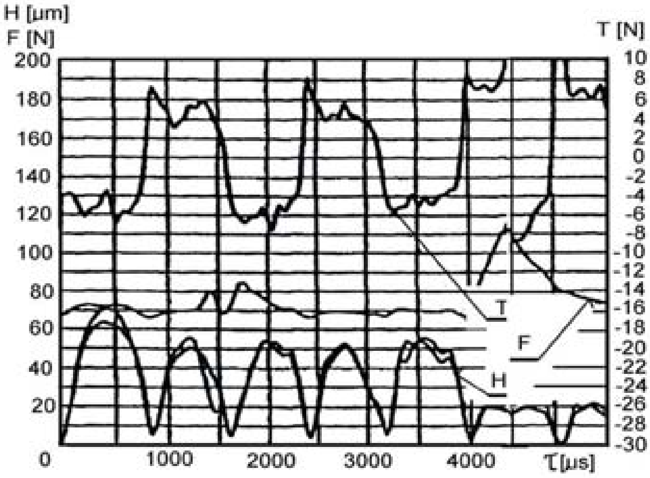


Fig. 4. Exemplary courses of oil film thickness H, friction force T and slat load F measured during operation of test stand [5]

The described test stand has been used already for research on phenomena accompanying the collaboration between rings and cylinder in engines of low power [4]. In order to carry out similar research for high power engines (marine and generator ones) one should know the parameters characteristic for kinematics of such engines and should check if these parameters can be performed on a test stand.

3. Analysis if simulation of high power engine crank-piston system operation is possible on a test stand

Following requirements should be satisfied if simulation of ring movement was possible on an earlier described stand maintaining the parameters of a real engine:
- an actuation of plate cyclic movement within a range corresponding to the real stroke,
- an actuation of plate movement with the speed of real piston,
- an actuation of plate movement relative to the λ ratio.

An introductory analysis of stand operational parameters presented in chapter 2 has already proved that full reproduction of marine engine operational conditions is not possible. One of the engine operation characteristic parameters is a piston average speed defined as:

$$c_s = \frac{S \cdot n}{30} = \frac{S \cdot \omega}{\pi}, \tag{1}$$

where:

S - stroke (test stand plate) [m],

n (ω) - simulated rotational [rpm] or angular speed [rad/s].

For conditions possible to simulation on the test stand a plate average speed calculated according to Eq. (1) is about 3 m/s, that is almost three times less than for engines presented in Tab. 1-2. Changes in piston displacement x, speed v and acceleration a usually are presented as a function of crank angle φ, according to following equations [1]:

$$x = 0{,}5 \cdot S\,(1 - \cos\varphi + 0{,}5 \cdot \lambda \cdot \sin^2\varphi), \tag{2}$$

$$v = 0{,}5 \cdot S \cdot \omega (\sin\varphi + 0{,}5 \cdot \lambda \cdot \sin 2\varphi)\,, \tag{3}$$

$$a = 0{,}5 \cdot S \cdot \omega^2 (\cos\varphi + \lambda \cdot \cos 2\varphi)\,, \tag{4}$$

where:
λ - is a crank radius r to connecting rod length l ratio (see Fig. 1).

Such form of presentation allows to carry out an analysis of changes in speed corresponding to the changes in crank angle (starting at TDC - see Fig. 5) but it does not help to evaluate the speed at selected points of its way. Because of that special graphs were constructed (see Fig. 6-7) in which one can find changes in piston speed at selected points. Only such way of presentation illustrates measuring possibilities of the simulation stand.

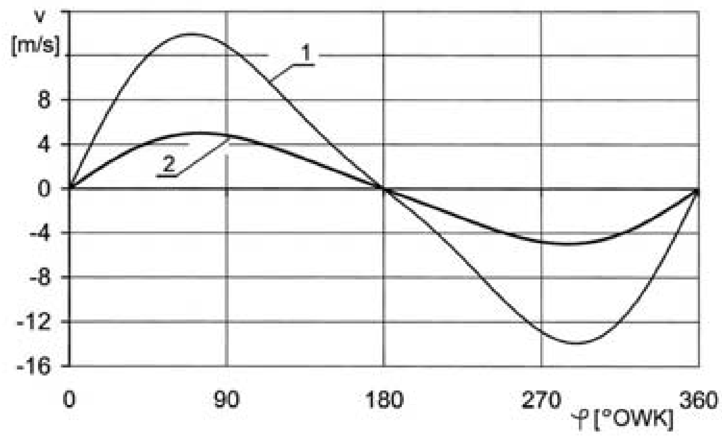


Fig. 5. Courses of speed vs. crankshaft rotation for: 1 - the S70MC-C engine, 2 - test stand plate

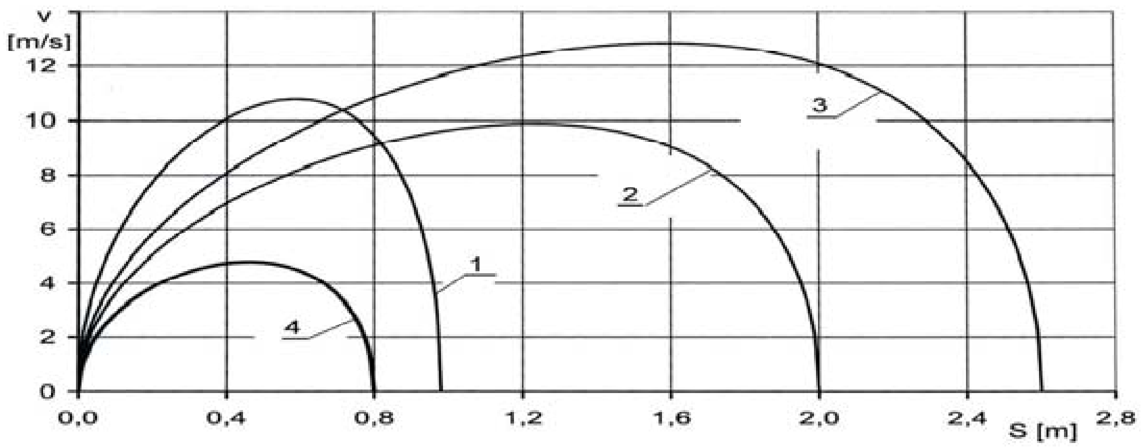


Fig. 6. Courses of speed vs piston displacement for following engines: 1 - L35MC, 2 - S50MC-C, 3 - K80MC-C, and 4 - test stand plate

As it can be observed in Fig. 6 the stroke of all marine engines taken into consideration (within the range from 1.0 to 2.6 m) is bigger than the displacement of test stand plate (0.8 m). Also maximum speed of these engines is almost twice the speed of plate (despite the fact that simulated rotational speed 12 rad/s (115 rpm) exceeds the speed of real marine engine.

Presented study proves that the full reproduction of piston movement is impossible on such test stand.

Similar analysis has been carried out for genset engines. In this case the plate maximum displacement was far bigger than piston stroke but on the other hand the speed which could be simulated was far lower (Fig. 7).

Exemplary courses of the L23 engine piston speed and the test stand plate speed (parameters S and λ are identical) are presented in Fig. 8. The test calculations showed that for an arbitrary selected point of piston stroke and plate displacement the speed ratio is constant and equals 7.57.

Because both the oil film layer thickness and friction force are relative to the piston speed [5] one can assume that these quantities on engine and on the test stand remain in similar proportion.

Summarizing the previous considerations one can state that the presented test stand allows to:

- evaluate the phenomena accompanying the collaboration of piston ring and cylinder surface at presence of lubricating oil, including oil film thickness and friction force for a single ring and ring pack,
- only a partial representation of marine engine piston displacement, because it covers just a section of stroke at far higher speed,
- just a partial representation of genset engine piston displacement, because it covers full stroke but at far lower speed.

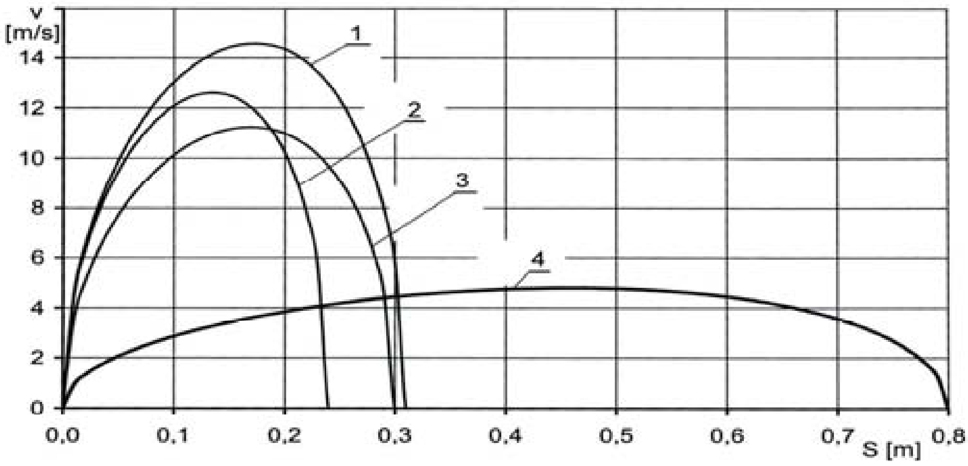


Fig. 7. Courses of speed vs piston displacement for following engines: 1 - L21, 2 - L16, 3 - L23 and 4 - test stand plate

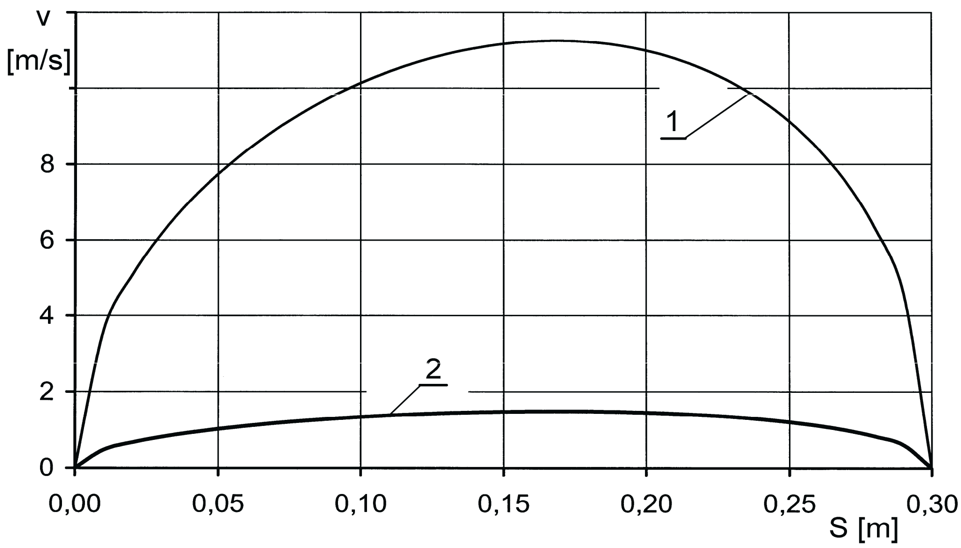


Fig. 8. Comparison of the L23 engine piston speed and plate speed courses

Differences in a way of lubrication of real engine piston-cylinder system and its model on test stand constitute another problem worth studying but not analyzed in this paper. Therefore, the effect of encountered differences in phenomena analyzed will be the subject of next paper.

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