MODEL INVESTIGATION OF PISTON RING-CYLINDER LINER COLLABORATION ON A HIGH POWER ENGINE

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Abstract

Failures caused by improper collaboration between piston ring and cylinder liner still happen on engines of high output, marine ones in particular. Among causes of this phenomenon one can mention higher mechanical and thermal loads of crank mechanism as well as difficulties in supply and distribution of lubricating oil over the entire surface of cylinder liner. It would be extremely difficult and because of regulations in most cases impossible to perform tests on correct collaboration between rings and liner of real running engine and on the other hand such tests on a test stand would be quite expensive. Use of mathematical models and tests on simulation model stands offers a way to reduce these costs.

Basic dependences connecting piston ring geometry with oil film parameters and introductory results of computations performed with the use of analytical model of test stand will be presented in this paper. The tests are to be carried out for ring geometry corresponding to that of marine engine rings. Due to that, using presented earlier formulas and computer program constructed by the authors the phenomena accompanying operation of different versions of rings were analyzed from the point of oil film thickness and generated friction losses. These ring versions which revealed the most advantageous properties will be made and tested on a test stand.

Keywords: *marine combustion engine, piston ring, oil film, model test stand*

1. Introduction

Modern marine propulsion engines perform power of several megawatts per cylinder at the rotational speed of about 100 rpm (basic technical data of selected low speed propulsion engines made by MAN Diesel are in Tab. 1).

Such high power makes that engine functional subassemblies are subjected to high mechanical and thermal loads. Various constructional modifications are introduced in order to avoid hazard and ensure the correct operation. The crosshead is an example of a mechanism introduced to the ship engine crank set protecting the cylinder liner surface against the effect of a piston excessive pressure. Nowadays, practically all ship propulsion engines put into service are constructed as the crosshead ones. Modifications concern the construction of piston rings as well. The proper sealing of combustion chamber has been achieved using the ring pack consisting of different rings dedicated to the operation under conditions of high load and insufficient lubrication. It should be added that one of the characteristic features of lubricating oil is the Total Base Number necessary for neutralization of sulphur contained in diesel fuel of very poor quality.

Tab. 1. Principal particulars of low-speed marine engines [1]

Engine Model	L60MC-C	L70MC-C	K80MC-C	K98MC-C
Cylinder Bore D [mm]	600	700	800	980
Piston Stroke S [mm]	2022	2360	2592	2400
k factor (S/D)	3.37	3.37	3.24	2.45
Engine Speed [rpm]	105-123	91-68	93-70	97-104
Cylinder Output [kW/cyl]	2340	3270	3640	3020

Despite thorough tests various failures of this subassembly still happen (most often those connected with improper collaboration of rings and bore) which lead to the deterioration of engine indices and in extreme cases – to engine stop.

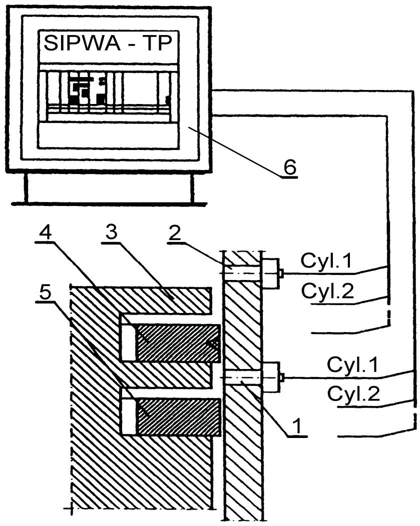

Fig. 1. Schematic of the SIPWA-TP diagnostic system; 1 – inductive sensor, 2 – temperature sensor, 3 – piston, 4 – SIPWA system ring, 5 – second piston ring, 6 – monitoring set [4]

Piston rings belong to these engine parts which inform about correctness of piston-liner operation. Numerous tests prove that improper wear of ring faces is the result of engine run failures. Due to that many diagnostic devices monitor the condition of rings. The SIPWA-TP system is an example of such system (see Fig. 1).

Instead of the original ring, a ring 4 of special design has been put into the piston upper grove 3. Its characteristic feature is a V shape insert made of diamagnetic material (bronze), spread over its face with the plasma method. Changes in insert geometry, which correspond to ring wear are monitored by a contactless inductive sensor 1 mounted into cylinder liner and serving for evaluation of ring-cylinder collaboration.

A considerable effect of rings, on engine operation, makes that tests on their improvement are still continued. However, an evaluation of the proposed modifications is very difficult because of obstacles in carrying out tests on running engine (such tests on marine engines are forbidden by the rules of classification societies).

All this makes that many researchers carry out their tests using mathematical and real models of piston-cylinder assembly. The model stands offer wide research possibilities because the range of operational parameters, distribution of sensors, completion of measuring devices can be determined at the stage of design. The test stand constructed at the Poznan University of Technology allows modeling of phenomena occurring on engine piston-cylinder set. An analysis of possible use of this stand to marine engine research has been carried out in previous study [4]. The result was that the conditions of ring operation on real engine could be simulated to the restrained extend because of stand constructional limitations.

2. Definition of oil film parameters

In order to develop the research possibilities of the model test stand a mathematical model of its operation (expressed in programming language) has been constructed which allows carrying out tests also within the range of parameters impossible to achieve on the test stand. The flow chart of the model is presented in Fig. 2.

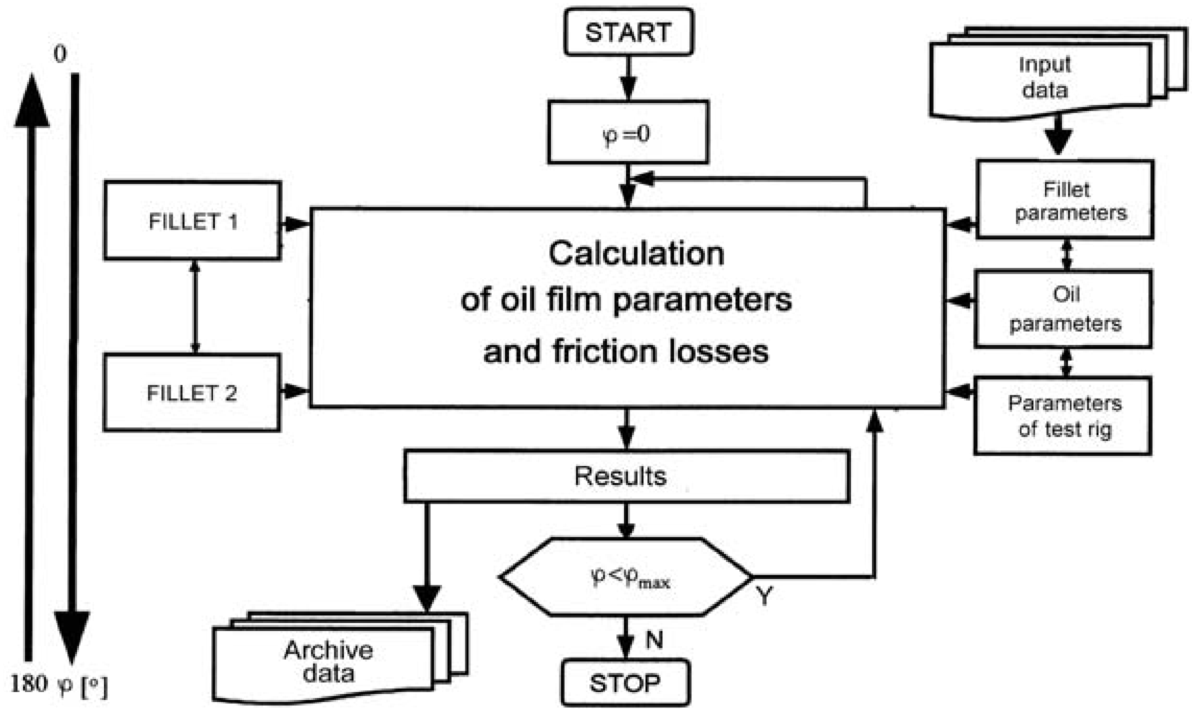

Fig. 2. The flow chart of test stand mathematical model [3]

The program basic computational block carries out calculations of oil film parameters including its thickness and friction losses generated by the ring motion. Computations are based on formulas of hydrodynamical theory of lubrication expressed numerically. Their application facilitates analysis of operation of practically any piston ring. However, it is worth mentioning that besides numerical ones there are also analytical methods of ring operation evaluation [2]. In most cases the dependences obtained using analytical methods are very complicated ones and their application is limited only to the basic versions of ring design. However, these dependences can be used at the initial stage of ring design which allows quicker and cheaper definition of optimal solution.

When considerations resulting from the hydrodynamical theory of lubrication are limited only to the wedge effect a possibility of definition of relations between the oil film minimum thickness h_m and ring face geometry occurs and ring run parameters can be found according to the formula:

$$h_m = \sqrt{\frac{\eta \cdot u \cdot b_f}{p_s} W_u} \,. \tag{1}$$

where:
u - axial speed of ring movement,
η - viscosity of lubricating oil,
b_f - axial height of ring face wetted with oil film (aggregate of b_n and b_s),
h_m - oil film minimum thickness,
p_s - ring elastic pressure,
W_u - ring face geometry effect on oil film mean pressure factors.

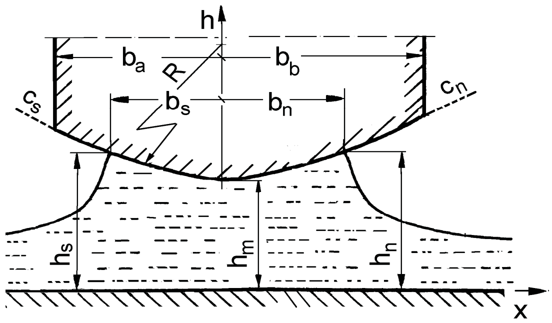

Fig. 3. Sketch of a barrel piston ring

The W_u factor present in Eq. (1) which facilitates consideration of ring face geometry is connected with oil film minimal thickness by the following complex relations (for the case of double barrel ring – notation as in Fig. 3):

$$W_u = \frac{3\cdot A_n^2}{\left(1+A_n^2\right)\cdot\left(A_n + C\cdot\sqrt{H_o - 1}\right)^2}\left[1 - \frac{H_o\cdot\left(2+A_n^2\right)}{2\cdot\left(1+A_n^2\right)} + C\,\frac{\left(H_o-1\right)^2\cdot\left(1+A_n^2\right)}{2\cdot H_o\cdot A_n^2}\right], \tag{2}$$

where

$$A_n = b_n\sqrt{\frac{c_n}{h_m}},\quad A_s = -b_s\sqrt{\frac{c_s}{h_m}},\qquad C = c_s / c_n, \tag{3}$$

while the unknown value of relative thickness H_o can be calculated with following equation:

$$\left(1-0{,}75\cdot H_o\right)\cdot\left[\frac{1}{1+A_n^2} - \frac{\operatorname{arc\,tg} A_n}{A_n} + C\cdot\frac{\sqrt{H_o-1}}{A_n}\left(\frac{1}{H_o} + \frac{\operatorname{arc}tg\sqrt{H_o-1}}{\sqrt{H_o-1}}\right)\right] - \\ -0{,}5\cdot H_o\left[\frac{1}{\left(1+A_n^2\right)^2} + \frac{C\cdot\sqrt{H_o-1}}{A_n\cdot H_o{}^2}\right] = 0\,. \tag{4}$$

The t value of friction force can be calculated according to the formula

$$t = \pi\cdot d\cdot\frac{\eta\cdot u\cdot b_f}{h_m}T_u\,, \tag{5}$$

where T_u is the factor taking into consideration ring face geometry when calculating the friction force.

$$T_u = \frac{4}{A_n + C\cdot\sqrt{H_o-1}}\left\{\begin{aligned}&\operatorname{arc}tg\,A_n - \frac{3\cdot H_o}{8}\left(\frac{A_n}{1+A_n^2} + \operatorname{arc\,tg} A_n\right) - \\ &-C\cdot\left[\operatorname{arc\ tg}\sqrt{H_o-1} + \frac{3\cdot H_o}{8}\left(\frac{\sqrt{H_o-1}}{H_o} + \operatorname{arc\,tg}\sqrt{H_o-1}\right)\right]\end{aligned}\right\}. \tag{6}$$

Diagrams in Fig. 4 are the graphic presentation of equations (2) and (6).

3. Tests on models of piston ring selected designs

Asymmetric profile of ring face covered with protecting layer is a typical design of a modern marine engine compression ring. However, literature does not provide detailed information on characteristic parameters of this surface.

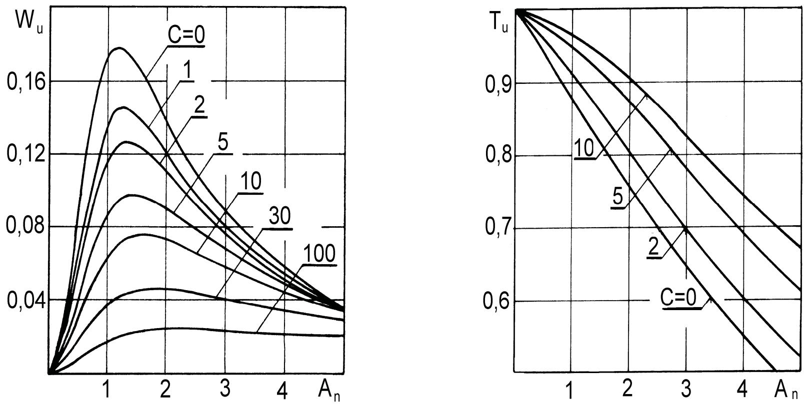

Fig. 4. Course of the W_u and T_u factor vs. the A_n coefficient for selected values of the C factor characteristic for double barrel ring

Due to that, using presented earlier formulas and computer program constructed by the authors the phenomena accompanying operation of different versions of rings were analyzed from the point of oil film thickness and generated friction losses. These ring versions which revealed the most advantageous properties will be made and tested on a test stand.

Model tests program consist of five constructional versions of symmetrical barrel ring (see Tab. 2).

Tab. 2. Data of barrel ring face profile and parameters of calculations

Version	$c_n = c_s$ [1/m]	R [m]	$f_a = f_b$ [um]	Parameters of calculations
1	5.00	0.1	281	viscosity η = 0.05; 0.15; 0.30 [Ns/m^2]
2	1.65	0.3	93	angular speed ω =5; 10; 15 [rad/s]
3	1.00	0.5	56	pressure p_s = 0.05; 0.10; 0.15 [MPa]
4	0.71	0.7	40	
5	0.55	0.9	31	$b_a = b_b$ = 15 mm (symmetrical ring)

It has been assumed that the ring face profile could be described both with a parabola ($y = c_i \cdot x^2$) or a section of circle ($y = \sqrt{R^2 - x^2}$), because for so small values of curvature the results obtained are convergent.

In a course of simulation tests 27 possible combinations of input data sets were tested for all constructional versions of ring (Fig. 5 shows exemplary courses of selected data characteristic for the oil film obtained for a single version).

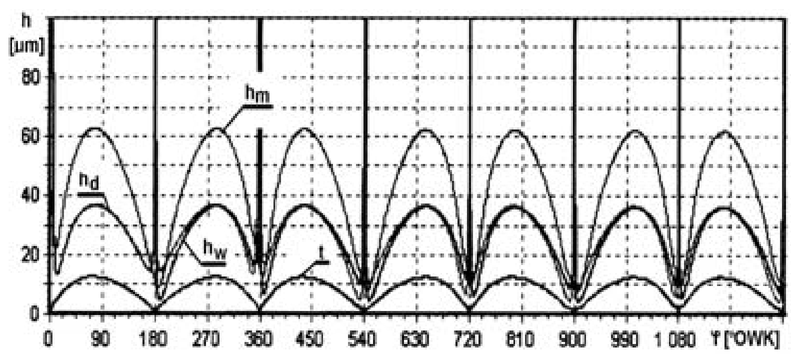

Fig. 5. Exemplary courses of the oil film minimal value (h_m), oil layer thickness before (h_d) and behind (h_w) the ring and friction force vs. crank angle

On the other hand, Figs. 6 and 7 show curves representing changes in the mean value of oil film thickness and friction force for ring versions tested (precisely, for a chosen radius of ring face profile R). Calculations of the mean value have been carried out for the third stroke of stand, i.e. the range of crank angle 360 to 540 degrees.

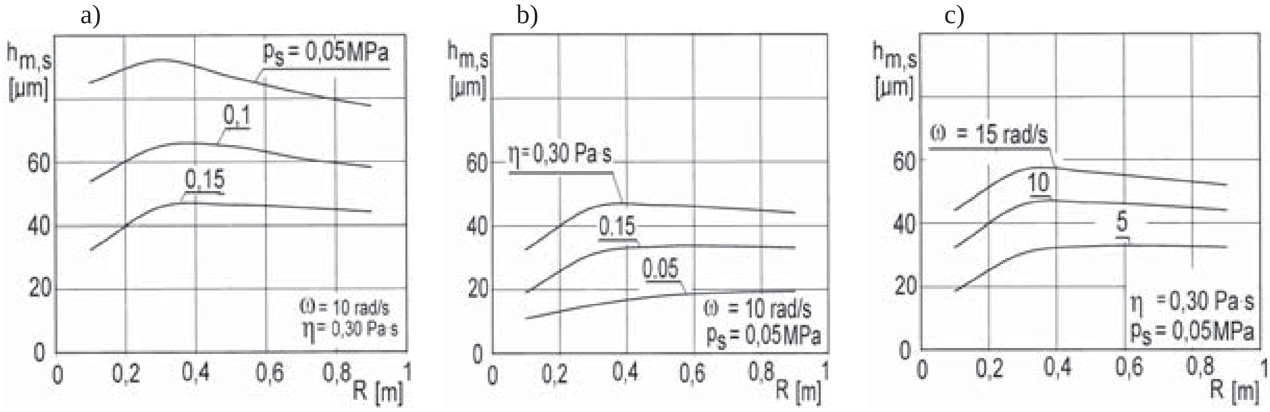

Fig. 6. Changes in the $h_{m,s}$ mean minimum oil film thickness vs. the R ring profile curvature radius for selected values of ring operation parameters (values shown in charts)

The courses presented in Fig. 6 show that there is a certain range of ring face profile radii (corresponding to the R radius of about 0.2 to 0.4 m) which gives the highest value of oil film mean thickness (which means that the conditions for oil film formation are the best within this range). Also the value of friction force accompanying the ring movement is relatively low for these ring radii (see Fig. 7).

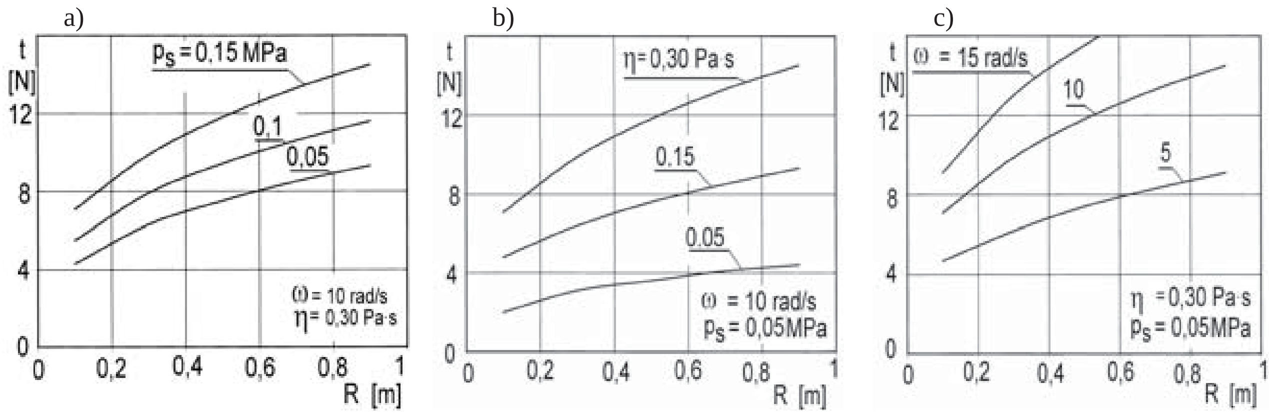

Fig. 7. Changes in the t mean friction force vs. R ring profile radius for selected values of ring operation parameters (given in Figs.)

However, one should remember that a continuous oil layer separating both surfaces (its thickness exceeds the combined asperity height of both surfaces) is above all the indispensable condition of their correct collaboration. This means that for an evaluation of the certain constructional version of ring one should analyze not only a course of oil film mean thickness but above all the changes in its minimum value. Relevant calculations of the thickness have been carried out in order to determine its dependence on the R radius and other selected parameters. The exemplary results obtained for chosen viscosities of lube oil can be found in Fig. 8.

The results obtained show that the drop in oil viscosity causes a parallel drop in minimum oil film thickness. Assuming a limit value of combined asperity height on the level of 0.5 μm (h_{gr}) one can evaluate a minimum admissible value of ring face profile curvature radius. Curves in Fig. 8 show that this value increases along with the deterioration of oil film formation (for example, for

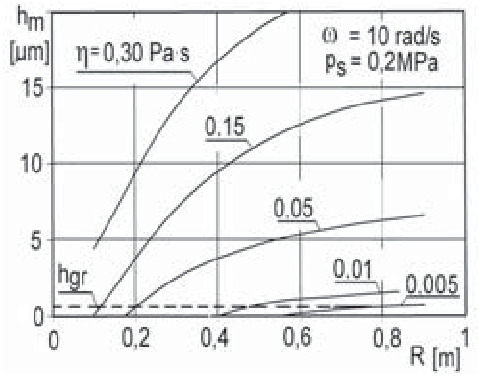

Fig. 8. Changes in minimum oil film thickness h_m vs. profile curvature radius R, for different oil viscosities

oil viscosity 0.01 Pas it is 0.45 m while for 0.005 Pas it is over 0.7 m). One can forecast that the admissible ring radius could be even higher when ring pressure against liner increases and ring speed lessens.

Comparing the achieved results with that published by the ring producers, one can observe a far going convergence. For example, in the case of the RT marine engines the R radius of the first compression ring lays within the range of ~0.60 to ~0.95 m (depending on engine type) when the total axial height is 15 to 30 mm (no substantial effect of the R_a = 0.3 mm radius of roundness – see Fig. 9).

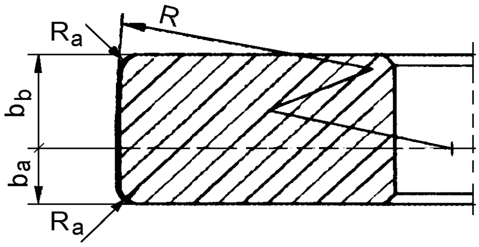

Fig. 9. Schematic of the RT marine engine piston ring

Ring versions taken into account in the simulation program were of the symmetrical barrel ones. It is necessary to repeat calculations for asymmetrical ring ($b_b > b_a$), because rings of such profile are most often used on marine engines.

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