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# ANALYSIS OF POSSIBILITIES IN USING CRANKSHAFT CASING PRESSURE MEASUREMENTS FOR DIAGNOSING TECHNICAL CONDITION OF THE PRC SYSTEM

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***Summary***

*In the paper is presented statistical analysis of the effect of compression-ignition engine working time on the phenomenon of exhaust gas pressure increase in crankcase and cylinder liner, pistons and rings wear. Characteristics of the crankcase pressure variations were made for the start-up speed and for selected rotational speeds of the 359 compression-ignition engine crankshaft as well as micrometric measurements of the cylinder liner and pistons and rings wear. The engine was operated under normal traction conditions while pressure analyses and cylinder liner micrometric measurements were made, respectively, after an operational run of: 5378 km, 100128 km, and 204864 km. Crankcase exhaust gas pressure measurements were made both with air vent open and closed for a cold (lubricating oil temperature of 285 K) and a warm engine (333 K). Basing on the analysis of obtained examination results, it was showed that crankcase exhaust gas pressure measurement (also during the start-up) may be used for determination of engine run and life.*

***Keywords:*** *scavenging, wear, piston, rings, cylinder, crankcase*

## 1. Introduction

Construction of the piston-cylinder system, into which such elements as piston, piston rings and cylinder liner (PRC) can be included, should ensure a slide fit under variable conditions of mechanical and thermal loads as well as preserve as large leak-tightness as possible [4]. Penetration of exhaust gases into crankcase in the form of scavenging between a piston, piston rings and a cylinder liner results in development of a specific pressure in crankcase. Moreover, a growth in scavenging intensity brings about increase of crankcase pressure, drop of engine power, increase of fuel and lubricating oil consumption, difficulties in engine start-up, in particular at low ambient temperatures, and excessive environmental pollution with combustion products. Too high excess pressure developing in crankcase is unfavourable for turbo-compressor lubrication system [7]. Every turbo-compressor has dynamic seals of the labyrinth type which are very sensitive to an increased level of excess pressure in crankcase. At that time, a free outflow of the oil lubricating turbo-compressor's bearings into oil sump is much hampered or even obstructed. Therefore, evaluation of the technical condition of the PRC system is very important, which can be a measurement of exhaust gas pressure in crankcase. This is because a pressure increase occurs together with a rise in wear.

## 2. Evaluation criteria for measurement method

When estimating the usefulness of crankcase exhaust gas pressure measurements for evaluation of cylinder liner wear degree, two criteria were adopted. First of all, results of crankcase exhaust gas pressure examination should be considerably correlated with cylinder liner wear and secondly, dynamics of determined signal change should be as large as possible. It results from literature [2, 3] that considerably correlated variables should be characterised by the correlation coefficient $r \geq 0.51$. It is known from experience that in case of statistical diagnostic analyses it is rather difficult to obtain such a considerable value of correlation coefficient and it requires large repeatability of measurement conditions. In this case, growth of exhaust gas pressure in crankcase induced by a rise in exhaust gas scavenging intensity in result of the increase of wear in the PRC group, in particular of cylinder liner wear, is regarded as a determined signal change. The following value was adopted as a dynamics index $d_p$:

$$d_p = \frac{X_m - X_o}{X_o}, \tag{1}$$

where:

$X_m$ - is a boundary value of signal indicating the necessity of performing a repair or taking an object out of service,
$X_o$ - is an initial value of signal characterising a new object after termination of the running-in period.

## 3. Formulation of research problem

The wear processes in the PRC assembly are unavoidable. Nevertheless, the most intensive is wear induced by friction phenomena and processes, which is reflected in cylinder liner wear. Material losses in co-operating parts bring about development of greater and greater clearances between PRC elements [5]. This favours increase in the intensity of exhaust gas scavenging into crankcase, destruction of oil film layer, development of more intensive erosion processes and increase of oil pressure in crankcase [6]. Therefore, pressure increase in crankcase [crankshaft casing] pressure can be used for forecasting the degree of engine wear, in particular that of cylinder liner.

Diameter measurements were taken with an inside micrometer calliper in horizontal plane being distant cylinder liner end face by 20 mm, which corresponds to the piston position in the upper dead centre (UDC). It is well-known that cylinder circularity becomes deteriorated in result of wear and resembles an oval. Its larger diameter (defined as $D_B$) corresponds to a plane perpendicular to the axis of engine crankshaft (it results from the dynamics of crankshaft-pistons-connecting rods system), while a smaller one (defined as $D_A$) occurs in a plane parallel to the axis of engine crankshaft. A clearance that develops then between a piston with rings and a cylinder is the main reason of both scavenging and pressure increase in crankcase. The character of clearance was determined by the following measures:

- circularity deviation measure:

$$\Delta D'_A = D_A - D_{avg}\ [\mu m] \tag{2}$$

where:

$$D_{avg} = \frac{D_A + D_B}{2} \tag{3}$$

$$\Delta D'_A = -\Delta D'_B \tag{4}$$

- cylinder liner deformation measure after insertion into engine body; it was assumed here that free cylinder liner diameters did not depart significantly from a $\phi$ size of $110^{+0.011}$ mm since production process in the aspect of precision is set out to that value; the character of clearance was expressed by a ratio of oval diameter deviations ($\frac{\Delta D_A}{\Delta D_B}$) from a cylinder nominal diameter (D) equal to 110.011 mm ($\Delta D_A = D_A - D$ and $\Delta D_B = D_B - D$).

## 4. Test results

Figure 1 presents a dependence of crankcase exhaust gas pressure in the function of rotational speed for the 359 engine respectively after a run of: 5378 km, 100128 km, and 204864 km. It can be seen that pressure in crankcase grows together with increase of operational run. For the rotational speed of crankshaft of 1500 $min^{-1}$ and that of 2200 $min^{-1}$, local pressure extremes occur. This is caused by the phenomenon of packing ring movements reported in literature [1]. Therefore, these speeds should be avoided when diagnosing technical condition of the PRC assembly. They affect a measurement error connected with the phenomenon of ring movement. However, it is evident that increase of operational run clearly affects pressure increase in crankcase.

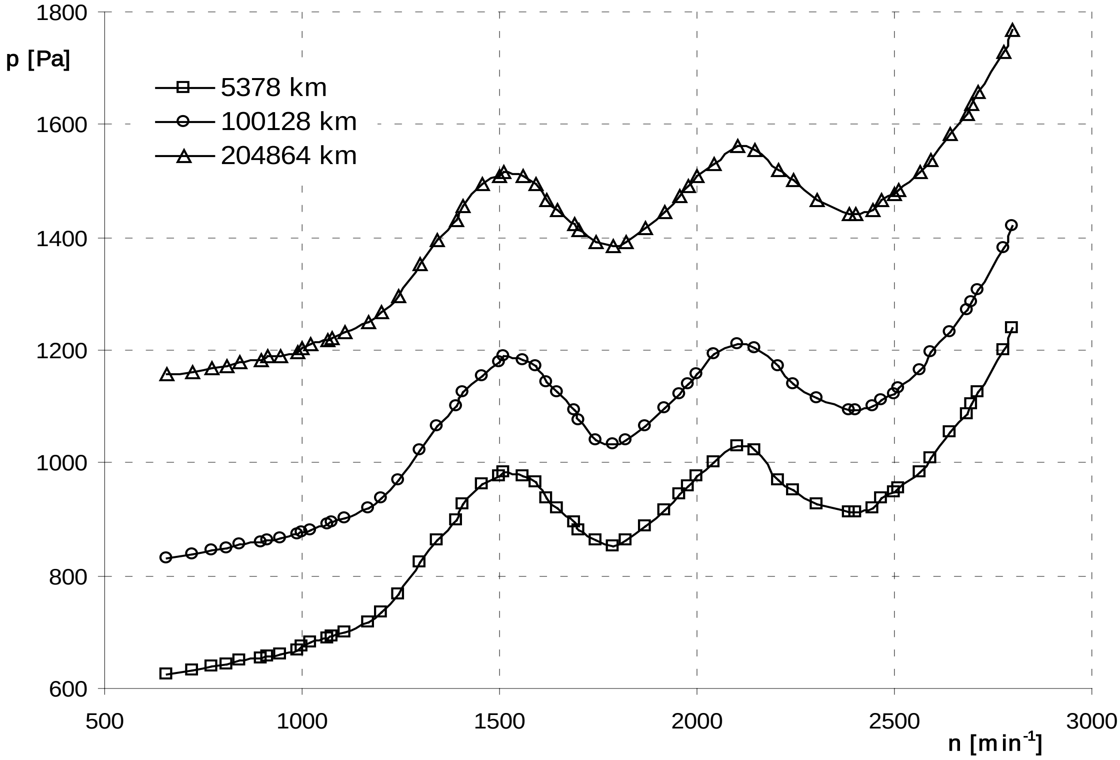


*Fig. 1. Dependence of the course of exhaust gas pressure in crankcase of the 359 engine in the function of rotational speed for different operational runs*

Figure 2 presents a dependence of crankcase exhaust gas pressure increase for a warm 359 engine (lubricating oil temperature of 333K) in the function of operational run expressed in kilometres. It can be seen that exhaust gas pressure in crankcase grows with increase of run.

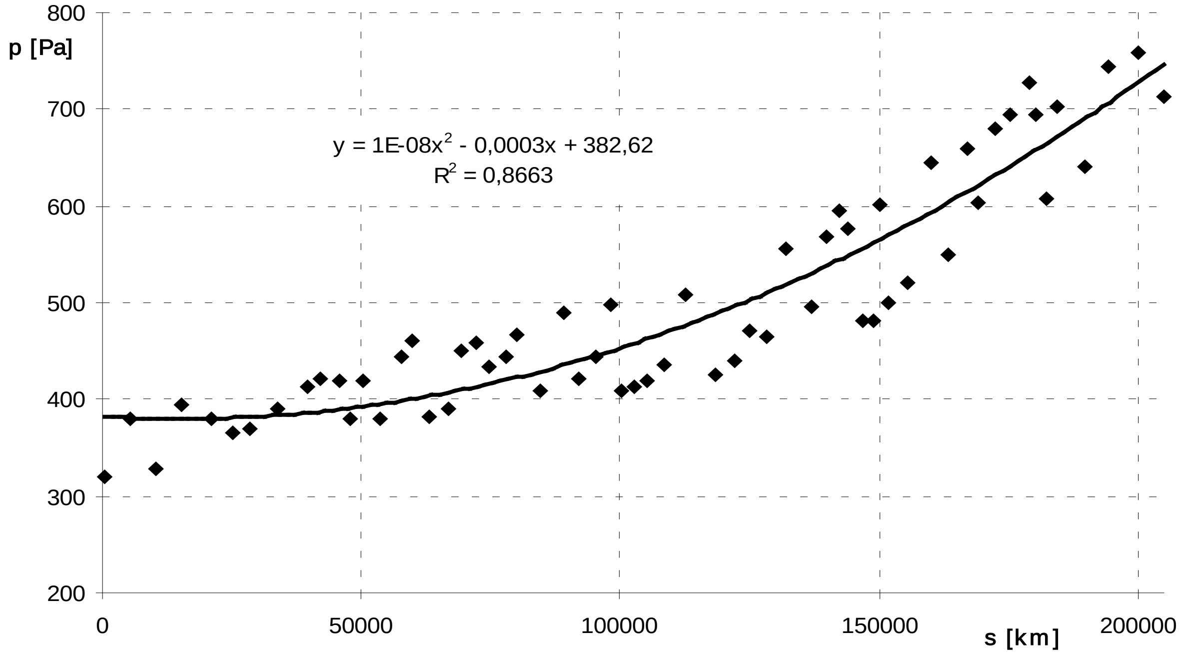


*Fig. 2. Increase of exhaust gas pressure in crankcase for a warm 359 engine in the function of operational run*

Figure 3 presents a dependence of exhaust gas pressure variations in crankcase in the function of 359 engine run for lubricating oil temperature of 285K.

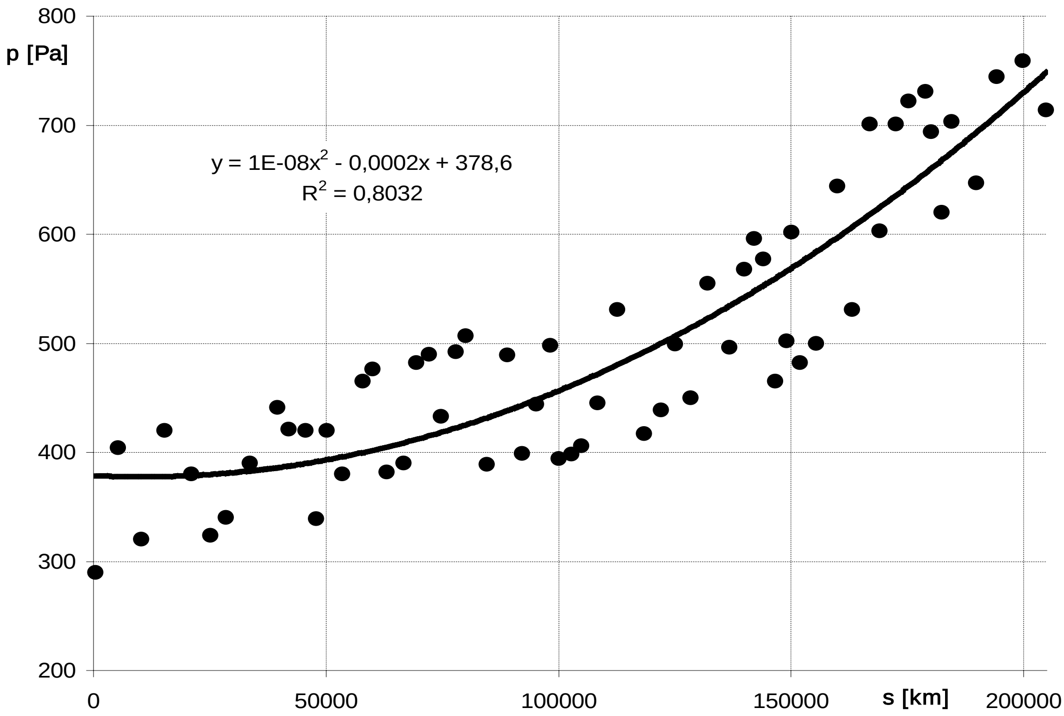


*Fig. 3. Increase of exhaust gas pressure in crankcase for a cold 359 engine in the function of operational run*

For a closed crankcase air vent, an increase of exhaust gas pressure presented in Fig. 4 was obtained.

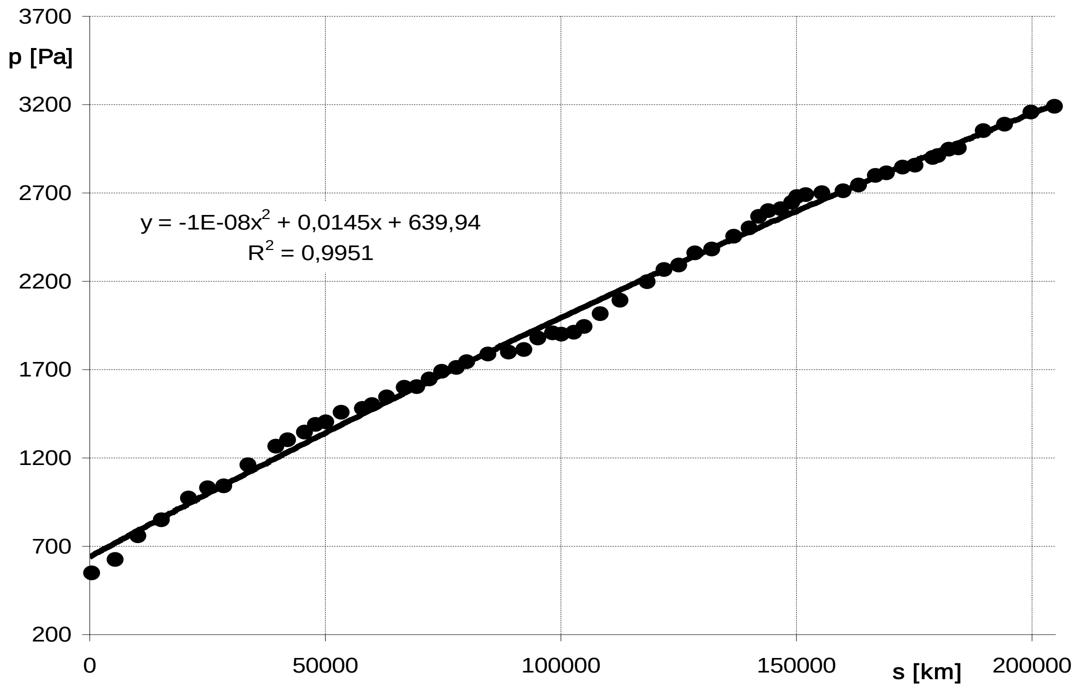


*Fig. 4. Increase of exhaust gas pressure in crankcase of the 369 engine for closed air vent in the function of operational run*

On the other hand, average tear values for respective cylinders of the 359 engine after a run of 204864 km amounted to:

- for first cylinder – 0,032 mm,
- for second cylinder – 0,041 mm,
- for third cylinder – 0,066 mm,
- for fourth cylinder – 0,081 mm,
- for fifth cylinder – 0,074 mm,
- for sixth cylinder – 0,065 mm,

whereas maxmimum wear values were as follows:

- for first cylinder – 0,120 mm,
- for second cylinder – 0,105 mm,
- for third cylinder – 0,118 mm,
- for fourth cylinder – 0,124 mm,
- for fifth cylinder – 0,122 mm,
- for sixth cylinder – 0,129 mm.

In order to determine a maximum oval, largest differences were calculated in planes A-A and B-B as well as C-C and D-D at one level $l_1$ = 20 mm from the top edge. The oval sizes were as follows:

| | | |
|---|---|---|
| - first cylinder | maximum 0.122 mm | minimum 0.012 mm |
| - second cylinder | maximum 0.140 mm | minimum 0.024 mm |
| - third cylinder | maximum 0.131 mm | minimum 0.016 mm |
| - fourth cylinder | maximum 0.099 mm | minimum 0.018 mm |
| - fifth cylinder | maximum 0.108 mm | minimum 0.014 mm |

| | | |
|---|---|---|
| - sixth cylinder | maximum 0.127 mm | minimum 0.022 mm |

In order to analyse the conicity, differences in dimensions were calculated in respective planes, i.e. A-A, B-B, C-C and D-D at levels $l_1$ = 20 mm and $l_7$ = 245 mm. The results point to negative taper (taper towards a lower part of cylinder should be considered to be a negative one). The conicity values were on average as follows:

| | | |
|---|---|---|
| - for first cylinder | 0.028 mm | negative |
| - for second cylinder | 0.025 mm | negative |
| - for third cylinder | 0.028 mm | negative |
| - for fourth cylinder | 0.023 mm | negative |
| - for fifth cylinder | 0.021 mm | negative |
| - for sixth cylinder | 0.019 mm | negative |

Maximum piston wear values in the plane perpendicular to piston pin axis were as follows:

| | |
|---|---|
| - for first cylinder | 0.12 mm |
| - for second cylinder | 0.13 mm |
| - for third cylinder | 0.09 mm |
| - for fourth cylinder | 0.11 mm |
| - for fifth cylinder | 0.12 mm |
| - for sixth cylinder | 0.11 mm |

Mean wear values for the height of respective rings were as follows:

- for first packing rings – 0.019 mm,
- for second packing rings – 0.012 mm, and
- for third packing rings – 0.009 mm.

Mean values from maximum wear values for the breadth of respective rings were as follows:

- for first packing rings – 0.189 mm,
- for second packing rings – 0.144 mm, and
- for third packing rings – 0.112 mm.

Mean wear values for the breadth of respective rings were as follows:

- for first packing rings – 0.151 mm,
- for second packing rings – 0.113 mm, and
- for third packing rings – 0.098 mm.

The size of piston-ring joint clearance in all packing rings after their insertion into a cylinder hole was larger that of entrance clearances by:

- 1.44 mm for first packing rings,
- 1.06 mm for second packing rings, and
- 0.93 mm for third packing rings.

In order to analyse a decrease in ring elasticity, crank pin effort values were measured at ring closure to piston-ring joint sizes measured after insertion into cylinder liner and comparison with initial values. The elasticity decrease was on average as follows:

- for first packing rings – 7.1 N,
- for second packing rings – 5.2 N, and
- for third packing  rings – 2.3 N.

## 5. Conclusions

When choosing a regression model, the following criterion was taken into consideration:

$$r^2 \rightarrow 1 \tag{5}$$

$$F_{cr} \leq F \rightarrow \text{maximum} \tag{6}$$

where F is a Snedecor's statistics testing significance of regression. The critical value $F_{cr}$ read from tables for a confidence interval $\beta = 0.95$ and a number of degrees of freedom $n_2 = 12$ and $n_1 = k - 1 = 1$ (k is a number of parameters of the regression equation) amounts to $F_{cr} = 4{,}75$. Correlation coefficient for the increase of pressure in crankcase in the function of operational run for a warm 359 engine amounts to $r^2 = 0{,}86$, whereas that for a cold engine is $r^2 = 0{,}80$. On the other hand, the best diagnostic measurement (in relation to pressure measurements) determining the engine run in kilometres is crankcase pressure measurement for closed air vent. Correlation coefficient in that case is $r^2 = 0{,}99$.

The determined signal change amounted for a warm engine to $d_p = (750\text{-}380)/380 = 0{,}97$, while for a cold engine it was $d_p = (740\text{-}370)/370 = 1{,}00$ at open air vent. On the other hand, the dynamics index $d_p$ at closed air vent amounted to $(3200\text{-}600)/600 = 4{,}33$ and was the largest. When evaluating the usefulness of maximum crankcase pressure measurement, it should be concluded that measurements ought to be made for closed air vent (largest correlation coefficients and largest signal dynamics change were obtained) and it is possible to clearly state that measurement of exhaust gas pressure in crankcase is closely correlated with operational wear in the PRC group.

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