Diagnostics of piston space tightness by pressure change in engine casing under dynamic loading

A V Safonov¹, M L Vertey¹, A P Syrbakov², G V Shnitkov¹, S N Krivtsov³ and
N N Berezhnov³

¹Novosibirsk State Agrarian University, 160, Dobrolyubova Street, 630039, Novosibirsk, Russia
²Irkutsk National Research Technical University, Department of Road Transport, 83, Lermont-tov Street, 664074, Irkutsk, Russia
³Kemerovo State Agricultural Institute, 5, Markovtseva Street, 650003, Kemerovo, Russia

E-mail: sirbakovap@yandex.ru

Abstract: One of the shortcomings of the method of determination of piston space tightness state by crankcase gases flow is high error. Primarily, it relates to the fact that when crankcase gas flow changes, such negative effect as the pulsation of crankcase gases appears. The value (magnitude) of pulsation of different engines varies greatly, and so does the error of the method. The diagnostics method of combustion engines CPG by the change of engine crankcase gases state parameter is discussed in the work. Theoretical substantiation and experimental testing of the proposed method are carried out.

1. Introduction

There is a tendency to reduce piston stroke and depth and, respectively, piston rings width in the construction of reciprocating engines. First of all, these changes are connected with engine uprating and reduction of frictional losses. Unfortunately, indicated changes in dimensional features of engine design parameters create conditions for spread of such defect as piston rings gumming. This defect leads to deterioration of all technical and economic indices of engine: excess oil and fuel consumption, etc. Toxicity of exhaust gases also increases. Availability of counterfeit engine oil aggravates the situation. Therefore, piston rings gumming appears more frequently in operation practices.

When operating time of a car of tractor grows, technical state of cylinders gets worse too. Increased complexity of design makes engines less appropriate for diagnostics using traditional methods. According to practice, existing methods of piston space tightness diagnostics [9]-[14] are not able to detect above mentioned defect, because their diagnostic variables have low resolving capacity.

On the one hand, as piston rings are gummed, piston space tightness declines. On the other hand, compression increases due to the sealing of leakage by engine oil. In this case, the engine is often dismantled and repaired improperly. After cleaning parts, it is found out that the value of structural diagnostic parameters does not attain its maximum.

Nowadays, methods, allowing carrying out in-place cleaning of gummed piston rings, are being developed. Control of cleaning efficiency is obstructed. Lack of knowledge about functioning of the engine with defective, in particular, gummed CPG, as well as methods of operational and reliable in-
place control of piston space tightness, impedes wide introduction of methods with necessary informative value when diagnosing CPG with gummed piston rings. Scientific research is topical owing to this fact.

The aim of the research is to develop the method of piston space tightness diagnostics with necessary informative value for the assessment of CPG state in case when piston rings are gummed, in particular.

2. Subject and Methods of Research
Informative assessment of the state of gummed CPG is possible at tests of engine under load. As in operation conditions, diagnostics of engines in quiescent load mode is difficult, methods of diagnostics of engine in dynamic load mode are in priority.

It is proposed to use parameter, characterizing the state of crankcase gases inside engine casing, as a diagnostic parameter.

The method of assessment of CPG state by pressure and consumption of crankcase gases is used as a prototype.

Crankcase gases consumption is a diagnostic parameter in well-known method of diagnostics of CPG by crankcase gas consumption. It is known that this method has low resolving capacity for small engines and is carried out in static mode.

Patent [5] proposes to carry out diagnostics of piston space tightness using crankcase gas consumption in the mode of dynamic load of the engine, that, in its turn, allows to increase method sensitivity and informative value. The main shortcoming of the method is complexity of continuous measuring of instantaneous values of crankcase gas consumption at small time gap while racing. The problem is that it is impossible to track consumption of pulsing flow crankcase gases having multicomponent aggressive compound. These problems are partially solved, using thermal anemometry sensors to measure gas consumption, and also pulsation dampers to stabilize crankcase gas flow. The use of pulsation dampers results in distortion of real alteration of crankcase gas consumption (Figure 1), and with their absence, measurements are inaccurate, owing to pulsation. Figure 1a shows that while racing the value of crankcase gas consumption does not practically change, but due to big pulsations, the value of error is high. Sharply defined maximum is seen in figure 1b. When the pulsation is dampered out, measurement error decreases, but realistic picture of consumption change is lost, because pulsation dampers increase transient resistance, and it leads to the change of pressure in casing, and consumption graph is distorted.

Parameter of crankcase gases pressure acts as a diagnostic parameter in the method of CPG diagnostics by graph of pulsation of pressure in casing. This method has low accuracy and reliability, because pulsations of natural dynamic ground inside the case lay on gas pulsations from the break via CPG sealing [6].

Thus, it is necessary to determine the reasons of crankcase gases pulsations and their nature in different engine constructions, and to find out, which engines are more appropriate for the assessment of CPG state by crankcase gas pressure.

While operating, dynamic background is created in the core of engine casing [1]. The main reasons of this background are:
- natural dynamic background inside casing;
- breakthrough of gases via CPG leakage different tacts.
Natural dynamic background in engine casing is due to complicated hydraulic process in casing.

It has two phases:
- dynamic flow of medium inside casing (crankcase gas);
- periodic change of casing volume.
Figure 1. Change of crankcase gas consumption, conventional unit from time, msec:
(a) without pulsation damping; (b) with pulsation damping.

Dynamic flow of medium inside casing (crankcase gas) occurs from the volume, vented by the piston, into the volume, released by the piston (at their simultaneous movement in mutually opposite directions). It is provoked by moving mechanisms components inside engine casing, mainly, by its rotating crankshaft. It is the reason of constant complex vortex formations of gases inside the casing that produces minor pulsations, in basic, low amplitude hum.

Periodic change of casing volume is the main reason of big pulsations of crankcase gas (high amplitude hum), especially, at low frequency of engine drive shaft rotation. It occurs due to the movement of pistons displacing casing volume. Therefore, it depends on the following constructive parameters:
- number of cylinders;
- disposition of cylinders;
- jamming angle of crankshaft.

It is obvious that periodic change of volume of casing of inline-four engine should be reduced to zero, because, while operating, two pairs of pistons are moving synchronously in mutually opposite directions, by that compensating increase and decrease of casing volume (Figure 2).

Figure 2. Disposition of cylinders at extreme points.
It is necessary to pay attention to the fact that connecting rod of crank mechanism is in plane-parallel motion. And at 90 degree rotation of crankshaft piston, moving from top dead center, passes more than half of its way by the value called “Brix correction” (Figure 3) [7]:

\[ a = \frac{R^2}{2L} \]

where \( a \) – “Brix correction”; \( R \) – crank radius; \( L \) – crank length.

As a result, when pistons catch up, in total they displace the volume of casing by the value equal to:

\[ V = 4 \times a \times S \]

where \( S \) – piston area; \( V \) – change of volume of engine casing.

![Figure 3. Influence of «Brix correction» on cylinders disposition.](image)

Calculation of engine casing volume change \( V \) by the angle of rotation \( \phi \) of crankshaft by the number and disposition of cylinders for the main most popular constructions is held. To compare the values of pulsations in different constructions, constructive parameters of CPG and cranking mechanism are connected to one engine (ZMZ-406). Thus, the cylinder diameter is \( D=92 \) mm, the crank radius is \( R=43 \) mm, connecting rod length is \( L=134 \) mm [6].

The results are presented on the graph (Figure 4).

![Figure 4. Change of the casing volume by the angle of rotation of the crankshaft in different engine constructions.](image)

It is determined that a periodic change of the volume in engine ZMZ-406 per one rotation occurs 2 times by 0.2 liters.
In addition, inline-three, -five and -six engines and V6, V8 and V12 on the graph have straight line. It means that when the engines work, the volume of their casing does not change [6].

Pulsations of crankcase gas of the four-cylinder internal combustion engine limit diagnostics by the value of crankcase gas consumption. The diagnostic method by the value of crankcase gases pressure change at time gap of the engine dynamic load is proposed to be used.

It is supposed that crankcase gas consumption in the process of the engine dynamic load (free racing, cyclic racing-rundown) has the same instantaneous values as crankcase gas consumption at static load.

According to the principles of hydraulics, the value of crankcase gas consumption is determined by the average value of the pressure in cylinder (indicated mean pressure) and by the value of leakage using the following equation:

\[ Q_v = \varphi \cdot w \cdot \sqrt{2 \cdot g \cdot P_i} \]  

where \( \varphi \) – leakage coefficient; \( w \) – leakage area, m²; \( P_i \) – average value of final compression pressure, Pa.

As the values \( \varphi \), \( w \), and \( g \) of the engine are constant, functional correlation of two parameters is obtained:

\[ Q_v = f(P_i) \]  

As a result, the change of the value of crankcase gas consumption in the process of the engine free racing can be estimated by the change of indicated mean pressure in cylinders at engine racing.

In this case, the change of leakage, due to pressure increase, is not taken into account. When the pressure in cylinder grows, the pressing force of rings to cylinder walls increases, and the leakage decreases. This phenomenon is not typical for gummed and destroyed rings.

In the process of free racing, the engine briefly imitates full-load curve. Figure 5 shows experimental graphs of alteration of the indicated mean pressure value, taken in static mode and in free racing mode [2]. The analysis of the graphs shows that dynamic and speed characteristics in sustained mode are equidistant.

Thus, the graph of crankcase gas consumption at free racing of the engine represents the nature of the graph of indicated mean pressure at speed characteristic of the engine.

\[ \begin{align*}
\text{Figure 5. Speed characteristic of diesel engine by indicated mean pressure:} & \quad P_i^d \text{ – indicated mean pressure measured at dynamic mode (free racing);} \\
& \quad P_i^{cm} \text{ – indicated mean pressure measured at static mode (braking test).}
\end{align*} \]

As a result, the possibility of momentary measuring of crankcase gas consumption at all engine operation modes, including overload condition, appears. It allows increasing the accuracy of the method.

The growth of accuracy is provided by the fact that measuring is carried out at high values of indicated pressure.

Sample calculation is held by the example of engine D-243.

The value of indicated mean pressure [7] at idle speed \( P_i^{xx} \) is equal to the value of the average pressure of mechanical losses \( P_m^{320} \), which can be calculated by the following equation:
\[ P_{i}^{\mathrm{XX}} = P_{m}^{2320} = a + b \cdot W_{P,SR} = 0.105 + 0.012 \cdot 9.66 = 0.22 \text{MPa} \]  \hspace{1cm} (5)

where \( a, b \) – coefficients, depending on the type of the engine.

\[ W_{P,SR}^{2320} \] – average speed of the piston, corresponding to maximum speed, m/sec:

\[ W_{P,SR} = \frac{S \cdot n_{\text{max}}}{30 \cdot 1000} = \frac{125 \cdot 2320}{30 \cdot 1000} = 9.66 \text{m/sec} \]  \hspace{1cm} (6)

where \( S \) – piston stroke, mm; \( n_{\text{max}} \) – maximum crankshaft speed, rpm.

The value of indicated mean pressure \( P_{i}^{\text{max}} \) of the engine, corresponding to maximum power, can be calculated by the following equations:

\[ \begin{aligned}
N_{l} &= \frac{P_{i} \cdot V_{h} \cdot n}{30 \cdot \tau} \\
N_{l} &= \frac{M_{i} \cdot n}{9550} \\
\end{aligned} \]

\[ \begin{aligned}
P_{i}^{\text{max}} &= \frac{30 \cdot \tau (M_{e} + M_{m})}{9550 \cdot V_{h}} = \frac{30 \cdot 4 \cdot (258 + 68)}{9550 \cdot 4.75} = 0.86 \text{MPa} \]  \hspace{1cm} (7)

\[ M_{i} = M_{e} + M_{m} \]

where \( \tau \) – tach coefficient; \( M_{e} \) – maximum torque, \( (M_{e}=258 \text{ H\cdot m}) \); \( V_{h} \) – engine capacity, \( l \); \( M_{m} \) – moment of mechanical losses, \( \text{H\cdot m} \).

\[ M_{m} = \frac{P_{m}^{1600} \cdot V_{h}}{30 \cdot \tau \cdot 10^{3}} = \frac{0.18 \cdot 4.75}{30} \cdot 9550 = 68 \text{H\cdot m} \]  \hspace{1cm} (8)

where \( P_{m}^{1600} \) – pressure of mechanical losses at revs, corresponding to maximum torque, MPa:

\[ P_{m}^{1600} = a + b \cdot W_{P,SR}^{1600} = 0.105 + 0.012 \cdot 6.66 = 0.18 \text{MPa} \]  \hspace{1cm} (9)

where \( W_{P,SR}^{1600} \) – mean piston speed, corresponding to revs, at which the engine develops maximum torque, m/sec.

\[ W_{P,SR}^{1600} = \frac{S \cdot n_{m}}{30 \cdot 1000} = \frac{125 \cdot 1600}{30 \cdot 1000} = 6.66 \text{m/sec} \]  \hspace{1cm} (10)

where \( n_{m} \) – frequency of engine crankshaft rotation, at which the engine develops maximum torque rev/min.

Let us compare obtained data \( P_{i}^{\text{XX}} \) and \( P_{i}^{\text{max}} \):

\[ \frac{P_{i}^{\text{max}}}{P_{i}^{\text{XX}}} = \frac{0.86}{0.22} = 3.9 \]  \hspace{1cm} (11)

The calculations shows how much the value of maximum indicated pressure is higher than the value of indicated pressure at idle speed. It leads to the increase of the accuracy that is the main progress of the examined method.

In this method the authors decided to temporarily exclude the release of the engine casing from crankcase gases while loading. Free racing-rundown is used as dynamic loading. It is proposed to release the casing from accumulated crankcase gases via special discharge valve, opened for engine rundown period. It is also suggested to carry out tests with at least five cycles of racing-rundown in order to diminish error and to establish instabilities. Thus, tests present series of continuously repeating racings and rundowns, accompanied by periodic opening of discharge valve in the end of racing and closing in the beginning of racing. By preliminary hypothesis, the obtained value should have high diagnostic properties, characterizing the state of engine CPG.

The growth of pressure in casing at engine racing at temporary crankcase sealing is proportional to the amount of gases, erupting through CPG for examined period. It comes from Mendeleev-Clapeyron law (equation 11) (if crankcase gas is close to ideal conditions). As crankcase gases consumption is directly proportional to relative CPG leakage, and the engine works under load while racing (that increases resolving capacity of this parameter), the possibility to qualitatively assess technical state of CPG appears. It means that we obtain indirect parameter for the assessment of leakage of engine CPG.
\[
PV = \frac{m}{M} RT \quad \Rightarrow \quad P = m \cdot C \quad \Rightarrow \quad \frac{dP}{dt} = \frac{dm}{dt} \cdot C \quad \Rightarrow \quad \frac{dP}{dt} = Q \cdot C
\]

(12)

where \( P \) – crankcase gas pressure, MPa; \( V \) – casing volume, l; \( m \) – gas mass in casing, kg; \( R \) – molar gas constant; \( T \) – crankcase gas temperature, °K; \( C \) – constant; \( t \) – time, h; \( Q \) – crankcase gas mass consumption, kg/m³; \( \Delta \) – change of the parameter for the period of free racing; \( \bar{Q} \) – average gas consumption for the period of free racing, kg/m³.

One of the research aims is the experiment to reveal the dependence of crankcase gas pressure change from different states of cylinder-piston group (CPG) (i.e. piston space tightness). The analysis of the methods of imitation of CPG leakage shows that they have an error, because at every test there is a necessity to dismantle and assemble internal combustion engine. It entails workload increase and decreases the economy of works. There is a method, when leakage is developed by consistent removing of compression rings. The second method is drilling holes in piston top and cylinder heads. The third method is to change thermal gap of valves (valve is clamped) – it cannot be always realized in modern internal combustion engines. That is why it is necessary to have a method of imitation of CPG leakage, which allows to diminish workload and to increase experiment efficiency, and will be used at laboratory work [8].

Possible options, considering the construction of the ZMZ4062.10 engine, are analyzed to imitate CPG leakage. As the engine chosen has constructive advantages (axis of spark plug and axis of piston are in parallel, and they have a small bias from each other), the method of CPG imitation, consisting in installation of replaceable jets with calibrated holes in a threaded hole in piston top, is proposed. CPG leakage is determined by using pneumotester (Figure 6).

**Figure 6.** Scheme of determination of CPG leakage: 1 – power unit; 2 – manometer; 3 – replaceable jet; 4 – piston.

The readings are taken by car diagnostic unit AMD-4A in the mode of self-recorder oscilloscope.

The scheme of experimental unit is presented in figure 7.

Electronic device 1, allowing carrying out engine racing-rundown at opened throttle valve in automatic mode, is designed to fulfill automatic racing-rundown (Figure 7). Realization concept of given function consists in periodic switching of engine injectors, depending on frequency of rotation of engine crankshaft. It means that racing-rundown of petrol engine in this case occurs by fuel shutoff. Discharge valve 4 is connected to this device and operated in automatic mode.
Figure 7. Scheme of experimental unit: 1 – electronic unit of automatic racing; 2 – button; 3 – inductive relay; 4 – discharge valve; 5 – pressure sensor; 6 – sensor of frequency of crankshaft rotation; 7 – engine injectors.

Figure 8. Exterior view of electronic device for operating electromagnetic nozzles.

The venting of engine casing is switched off before test. Intake fitting of venting of casing of idle run and intake fitting of the main circuit of casing ventilation are shut down. Discharge valve 4 is connected at the output of the main circuit of ventilation; pressure sensor 5 is installed at the output of ventilation union of casing of idle run.

Electropneumatic valve KAM19-01 with enlarged opening area is used as discharge valve. Sensor of pressure of boost air charging 47.3829, used in diesel engines GAZ-560, is applied to measure the pressure.

The valve starts working in automatic mode only when button 2 is pushed. In other cases it remains opened.

Exterior view of experimental unit is shown in Figure 5-9.
3. Study Results
Signals from pressure sensor and sensor of disposition of crankshaft are registered during the experiment.

CPG is tested in three different technical states. In the first case, caps are installed in pistons. In the second case, jets with a diameter of 0.5 mm are placed in pistons. And in the third case, jets with a diameter of 1 mm are used. The results of the tests are presented in graphics in Figure 10-12. Abscissa scale shows the time, vertical scale reflects pressure value in conventional units and the value of frequency of crankshaft rotation.

![Figure 9. Exterior view of experimental unit.](image)

![Figure 10. Change of pressure in engine casing at free racing with periodic insulation of casing from the environment without jets.](image)

![Figure 11. Change of pressure in engine casing with free racing using periodic insulation of casing from the environment by jets with a diameter of 0.5 mm.](image)
Figure 12. Change of pressure in casing of engine ZMZ-4062 at free racing with periodic insulation of casing from the environment by jets with a diameter of 1.0 mm.

The analysis of graphs shows that with the increase of leakage, the pressure of crankcase gas grows proportionally (when CPG functions properly). This method allows determining dynamic power of engine by acceleration of free racing and pressure in engine casing at the same time. Resolving capacity of proposed method is determined in comparison to the method of the pressure of the end of compression stroke. Initial data for the assessment of two methods are presented in the Table 1.

|                  | $P_{c}/P_{s}$, kg/cm$^2$ | $P_{i}/P_{s}$, con.units | $S_{i}$, mm$^2$ |
|------------------|--------------------------|--------------------------|----------------|
|                  |                          |                          |                |
|                  | 1                        | 1.05                     | 1.16           |
|                  | 1                        | 0.66                     | 0.54           |
|                  | 0.27                     | 0.78                     | 1.6            |

4. Conclusion

Pulsations of crankcase gas of four-cylinder internal combustion engines limit the diagnostic method by the value of crankcase gas consumption. Also, aggressive compound of exhaust gases leads to the failure of flow sensor. The use of crankcase gas pressure as a diagnostic parameter to determine CPG state at full insulation of casing from the environment at the operation of internal combustion engine in free racing-rundown mode with 0.78 mm$^2$ leakage has resolving capacity 5 times higher than the method of the pressure of the end of compression, when the starter scrolls. Workload at insulation of the crankcase gas ventilation system can be reduced, using special clamps, and pressure sensor can be installed once and employed at periodic diagnostics. The tests held acknowledge possibility of further development of proposed method for in-place diagnostics of CPG of internal combustion engines and allow marking out factors, which should be taken into account at future research: ignition dwell angle and the amount of fuel delivered to cylinders.

References

[1] Ventsel S V, Korovyaniskiy I A 1982 Gas dynamic background in engine casing Engines Construction 1 32–36
[2] Klein A T 1972 Indicator indices of motor and tractor engines in dynamic self-loading mode. Mechanization of agricultural production processes Scientific work (Novosibirsk: Novosibirsk Agricultural Institute)
[3] Stanislavskiy L V, Ulanovskiy E A, Ignatov O R 1983 Diagnostics of cylinder-piston group of diesel engine by crankcase gas consumption Engines Construction 11 37–38
[4] Safonov A V, Voronin D M, Vertey M L, Ponizovskiy A Y, RU Patent No. 2486486 (2016)
[5] Voronin D M, Ponizovskiy A Y, Malyshko A A, Vertey M L, RU Patent No. 2343445 (2009)
[6] Voronin D M, Guskov Y A, Vertey M L, Safonov A V 2016 Influence of constructive parameters of the engine on the value of pulsation of crankcase gas flow KGAU Bulletin 12 112–117
[7] Nikolaenko A V 1984 Theory, construction and calculation of motor and tractor engines Manual for agricultural institutions of higher education (Kolos) 335 p
[8] Vertey M L 2010 Influence of leakage of cylinder-piston group on the characteristics of free racing of petrol engine NSAU Bulletin 2(14) 77–79
[9] Kao M, Moskwa J J 1995 Nonlinear diesel engine control and cylinder pressure observation J. of Dynamic Systems Measurements and Control (Transactions of the ASM) 117(6) 183–192
[10] Gallacher A M, Gmbh A, Krebl W H, Gmbh A 2015 Dynamic engine testing: why? No. 952301
[11] Charchalis A, Dereszewski M 2014 Analysis of rotational speed fluctuation of diesel engine shaft for evaluation of combustion process J. of Vibration Eng. and Technol. 2(5) 415–421
[12] Brown T S, Canada N R C o, Neill W S, and Defence D o N 2015 Determination of engine cylinder pressures from crankshaft speed fluctuations No. 920463
[13] Ponti F, Rinaldi M, Asme 2008 Analysis of the relationship between mean indicated torque and its waveform for modern common rail diesel and gasoline engines Proc. of the Spring Technical Conf. of the ASME Internal Combustion Engine Division pp 149–159
[14] Volvo Trucks Deutschland: Wersunterlangen zur Kompressionsprüfung (Dietzenbach) 1998