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The Advancement of the Methods of Vibro-Acoustic Control of the ICE Gas Distribution Mechanism

A modern vehicle is a system of high-precision systems. In particular, over the past 20 years, the gas distribution mechanism (GRM) has been structurally released from any adjustments. However, alongside with that, the elimination of the adjustments did not entail a decrease in the number of GDM failures. The analysis of a number of works shows that the GDM mechanism accounts for more than 30% of ICE failures. The problem lies in the high sensitivity of modern engine systems to the oil drain interval, fuel quality, timeliness of maintenance, etc., i.e. to carrying out of routine activities. According to the analysis of the operating conditions of modern vehicles, the regulations are significantly violated in more than 50% of cases, which results is a considerable reduction in the service life of vehicles. In this connection, it is relevant to develop diagnostic methods allowing us to determine the degree of wear of engine components at all vehicle operation stages. One of such methods is the vibro-acoustic method. The research aims to develop a methodology and method for diagnosing GRM by analyzing vibration parameters together with a pressure signal in the cylinder, the angle of the ICE crankshaft. As a result, the mechanism for recognizing the fault conditions of individual GRM valves is determined. A set of diagnostic tools for the selective diagnosis of GRM elements has been developed. A methodology for determining the GRM phases and thermal clearances in the valve drive using the nondemountable method has been created.

Keywords:*engine; gas distribution mechanism; diagnostics; amplitude; frequency; phase; efficiency.*

1. INTRODUCTION

Modern engineering factories producing motor vehicles create developments on the principle of the minimum number of possible adjustments [1, 2]. In particular, the gas distribution mechanism is actually released from any adjustments, which used to be difficult and time-consuming [2, 3]. However, the precision of manufacturing these systems makes it much more serious to approach maintenance periods and frequency [4-6]. Thus, the presence of hydraulic couplings, hydraulic compensators, hydraulic tensioners, oil seals imposes essential requirements on the quality of engine oil and the timeliness of its replacement [7-9]. In case the oil drain terms and intervals are met, the engine completes its nominal service life [10-12]. In case the oil drain terms and intervals are violated, a vast chain of undesirable phenomena and processes occurs: accelerated wear of bearings and related increase in clearances [7, 13, 14]; wear of thecrankshaft main and connecting rod journals; decrease in the pressure in the main oil line and in the line supplying turbo compressor bearings [13, 15]; scuffing of pistons, rings and cylinders, etc. [16, 17]. In this connection, it is very important to comply with the maintenance and repair regulations, and prior to these procedures, to use in-place diagnostic methods [18]. Therefore, the task of developing highly sensitive and accurate methods for diagnosing the gas distribution mechanism is very relevant [19, 20]. This will allow us to determine the emerging failures at the initial stages [21- 24]. Thus, according to some data, more than 30 % of engine failures fall at the GDM (Figure 1).

The GDM is a complex of elements with different wear characteristics. Thus, according to [25], the engine GRM failures are distributed as follows Figure 2.

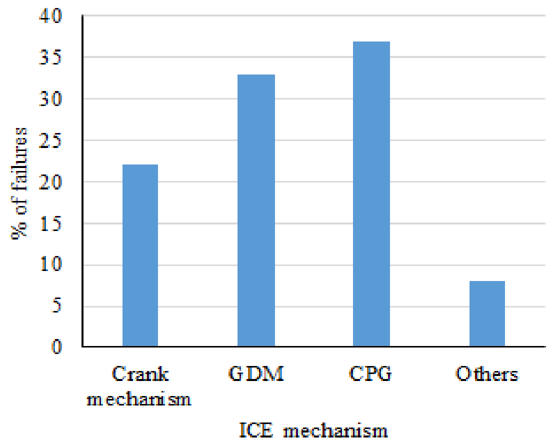

Figure 1. The distribution of engine mechanism failures

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It follows from Figure 2 that the prevailing number of failures falls at changes in the expansion clearance.

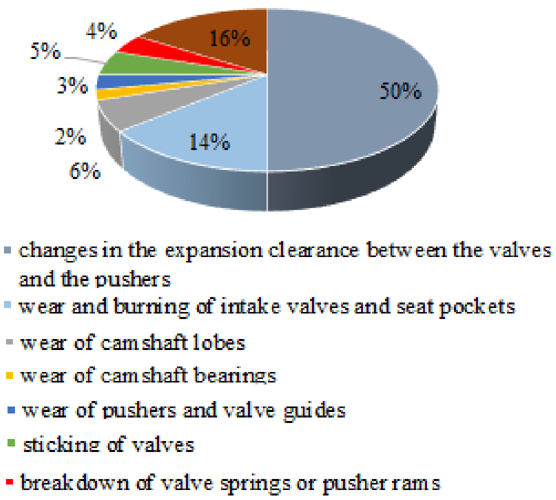

Figure 2. The distribution of failures in the gas distribution mechanism

So, Figure 3 presents a graphical illustration of an expansion clearance.

An expansion clearance changing in size causes intense collisions between the valve and the seat [14, 18, 25]. Pulses from the collisions of elements can be recorded by vibro-acoustic methods synchronously with each valve. Besides, when diagnosing the GDM mechanism, much attention should be paid to in-place diagnostics and built-in ICE control means [26, 27, 28]. The vibro-acoustic diagnostic means meet the given criteria [29, 30, 31].

Currently, researchers use several methods to measure expansion clearances. They include:

1. Assessing the air-tightness of the cylinder-piston group and the gas distribution mechanism by the scavenging method and leak monitoring. This method allows us to detect leakages in the GDM valves and valve causing the leakage. This method is rather universal and widespread in the diagnosing practice. However, this method is characterized by significant disadvantages: it does not allow us to control expansion clearances of the intake and release valves in operation.

2. The method of compression measurement in the ICE cylinder. As for this method, its undoubted advantage is universality and usability. However, the disadvantages include low reliability. The application of this method does not allow to determine the size of the gaps and a non-working valve.

3. Micro-measurement of the cam and pusher gradient with an ICh-10 dial gauge. This method is widely used in repairs. Its advantage is direct controllability of geometry deviations. However, its significant disadvantages are: the need to disassemble the mechanism and considerable labor intensity of the process.

4. The method of clearance check in the GDM using a probe. When implementing the method, the costs are minimum and include just the purchase of the probe. However, the use of the method requires disassembly and special appliances. The method is rather laborious.

5. The method for monitoring the GDM phases with a pressure sensor in the ICE cylinder. The method is universal and does not require significant timing for measurements. However, the high rates of compression processes and the low sensitivity of the pressure sensors do not allow us to reach a high accuracy of fault recognition [32].

6. The method for monitoring the GDM clearances with a displacement sensor located on the valve. This method is typically used for research purposes. It did not find application in practical operation. The disadvantages include: significant complexity of installing the displacement sensor on the valve, complexity of signal release and control, considerable labor intensity.

7. The method for monitoring the GDM clearances by the vibration parameters using an accelerometer. This method is universal and applicable to various vehicle systems. It has high selectivity, accuracy and reliability of the fault recognition process. However, it requires comprehensive research and the development of schemes and algorithms for its application [33, 34, 35].

At the same time, obtaining of diagnostic signs is the main problem solved by a detailed study of the sound formation mechanisms and the resulting vibration processes from the collisions of the parts of the studied object [36].

The technical condition of the gas distribution mechanism is assessed by the following indicators: the size of expansion clearances in the valve drive, wear of the drive gears and camshaft lobes of the gas distribution mechanism, elasticity of the springs, size of the clearance between the guide bush and the valve rod, integrity of the valves and the seats, setting of the GDM phases. A partial or complete disassembly of the engine is needed to determine these values when direct control methods are used, while the accuracy of the results is high but not always complete, since some properties of the parts are most fully manifested during their interaction.

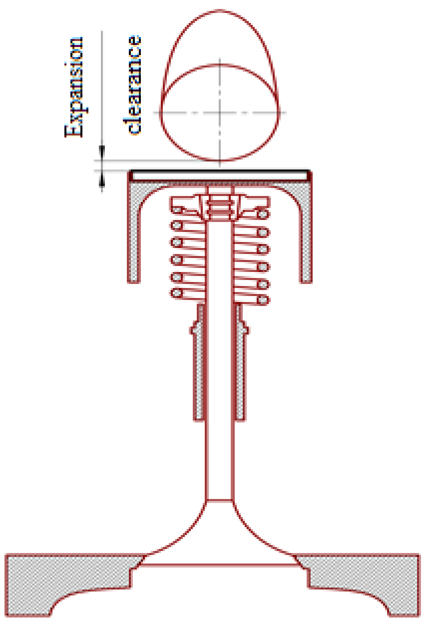

Figure 3. A graphical illustration of an expansion clearance

To this end, there are continuous developments and improvement of methods for diagnosing the technical condition of the GDM based on the indirect signs, which allow us to quickly and accurately identify a defect at an early stage, using an in-place method: 1. Power of mechanical losses on to the GDM drive; 2. Pressure/discharge in the inlet/outlet pipe; 3. The change in the pressure in the cylinder; 4. Air-tightness of the piston space; 5. Intracyclic change in the crankshaft angular speed; 6. Physical and chemical composition of used operational materials; 7. Vibration /noise level; 8. Heat generation rate, etc.

The purpose of the research: 1. To determine the GDM phases and the size of expansion clearances in the valve driveusing the in-place method by analyzing the structure of the vibration signals generated during the engine operation due to the collision of GDM parts. 2. To determine and analyze the factors influencing the parameters of the received vibration signal.

To use the vibro-acoustic method, we should consider some provisions of the theory and methodology of the process.

2. THEORETICAL RESEARCH

The quality of gas exchange is one of the key factors influencing the engine technical and economic performance. One of the main mechanisms responsible for controlling gas flows is the gas distribution mechanism, the main task of which is to create optimal engine operating conditions.

The gas exchange process can be qualitatively estimated by the charge ratio:

$$\eta_V = \frac{M_1}{M_0}, \tag{1}$$

where, M_1 is the actual amount of an incoming charge entered the engine cylinder during the intake process; M_0 is the amount of an incoming charge that could be placed in the working volume at the inlet air parameters (p_0, T_0)

So, the values of η_V for different ICE types are presented in Table 1.

Various factors influence cylinder charging: 1. To which harmonic resonance the intake manifold is tuned; 2. What injection system is used; 3. Air cleaner resistance; 4. The number of valves per one cylinder; 5. The temperature of the fuel-air mixture; 6. Inlet channel resistance, etc.

Table 1. The values of η_V for different ICE types

ICE type	η_V
Spark ignition engine	0.75...0.85
Gasoline injection engines	0.80...0.96
Naturally-aspirated diesels	0.80...0.90
Turbocharged diesels	0.80...0.95

Gas distribution phases significantly influenceη_V, violation of which leads to worsening of toxicity indicators, reduced engine power, increased fuel consumption, reduced life of individual parts and mechanisms, in some cases, increased noise, vibration and other negative manifestations.

Let us analyze the working phases of a gasoline ICE (Figure 4).

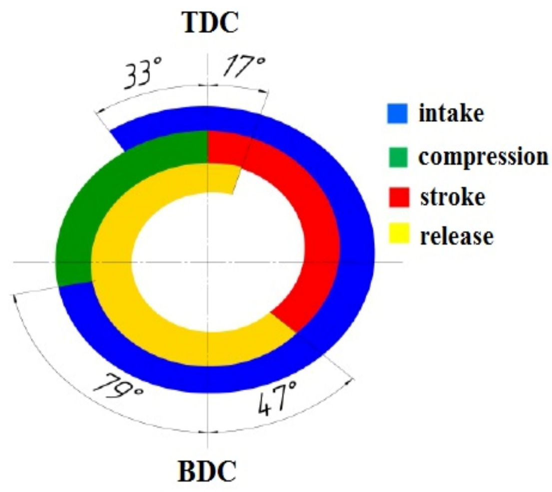

Figure 4. An analysis of gasoline ICE phases

So, it can be seen from Figure 4 that the intake phase lasts for 292^0. It starts 33^0 before the TDC and lasts for 79^0 after the BDC. This is accompanied by the opening and closure of the intake valve [18, 29, 31]. The release phase, which lasts for about 240^0, is also important for the theoretical study of the vibration process parameters. It starts 47^0 before the BDC and lasts for $10\text{-}13^0$ after the TDC. The operating procedure of the individual engine cylinders is presented in Table 2.

It is representative to consider the dynamics of changes in the defect (clearance size) during the GDM operation (Figure 5).

Table 2. The operating procedure of the individual engine cylinders

Crankshaft rotation angle (degrees)	Cylinder number			
	1	2	3	4
0-180	stroke	release	compres-sion	intake
180-360	release	intake	stroke	compres-sion
360-540	intake	compres-sion	release	stroke
540-720	compres-sion	stroke	intake	release

So, zone 1 is formed by the permissible vibration for new machines delivered from the factory. It forms a running-in zone, which is within 20 % of the time resource. Zone 2 is formed by a normal operation section, where the clearance size is stable and actually remains constant along the entire length (10-40 % of the time resource).

Zone 3 is characterized by an intensive wear section and, accordingly, the growth of the amplitude parameters of the vibration process (lasts for 50-80 % of the resource time). And the final section is 4, where the clearance size can significantly increase and exceed the threshold limit value (0.01-1 % of the resource time). In section 4, amplitude signals are often distinguishable without any amplifier equipment. However, it is characterized by a relatively small defect formation period [25, 37].

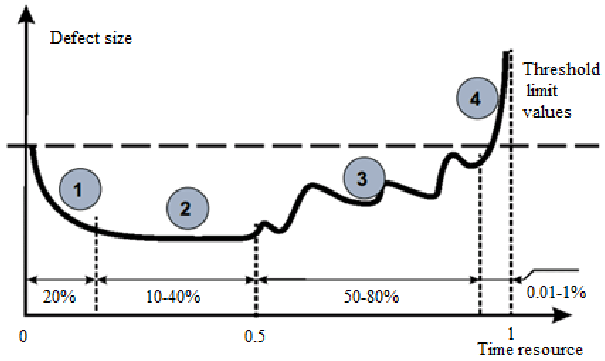

Figure 5. The dynamics of changes in the defect (clearance size) during the GDM operation

Currently, the method for the meansquare value of vibration velocity or vibration acceleration is widespread [18, 25, 38, 39]. Its use is justified by the interconnections of output parameters with the change in clearances during operation. The mean square values of the output parameters are calculated by the formula:

$$V_{ms} = \sqrt{\frac{1}{T}\int_{t_i}^{t_i+T} [V(t)]^2 dt}\,, \qquad (2)$$

where T is the analyzed time period, s; t_i is the initial time period, s; $V(t)$ is the analyzed vibration velocity values, mV/s.

Let us analogously present the vibration acceleration formula:

$$a_{ms} = \sqrt{\frac{1}{T}\int_{t_i}^{t_i+T} [a(t)]^2 dt}\,, \qquad (3)$$

where $a(t)$ is the analyzed vibration acceleration values, mV/s^2.

The kurtosis study method is widespread:

$$R_k = \frac{\int_{t_i}^{t_i+T} (A - A_{av})^4 P(x) dx}{\sigma^4}\,, \qquad (4)$$

where$A$andA_{av} are the instantaneous and average values of the vibration signal amplitudes recorded from the vibration sensor during the diagnostics, mV; $P(x)$is a probability function of a random value; σis a mean square deviation of the diagnosed parameter,

The basisofthe theory of the vibration process analysis is the reduction of interference (noise) to the possible minimum. So, if the output signal, for example, of a vibration amplitude, is represented as a sum of the base function $a(t)$, the information function $i(t)$ and the interference (noise) $z(t)$, we can write:

$$Q(t) = a(t) + i(t) + z(t)\,, \qquad (5)$$

The ordinary practice of vibration data processing builds a new information function, which is written in the form of the equation:

$$R(t) = S[a(t)] + S[i(t)] + S[z(t)]\,, \qquad (6)$$

In this case, when analyzing the data, it is necessary to ensure the condition: $S[a(t)]$=0.

Besides, if the interference (noise) component $S[z(t)]$ is minimized, only the information component of the signal $S[i(t)]$ remains at the output, which bears a positive result without interferences and other vibration processes in a complex oscillogram of the vibration process.

Different values can be used to quantify mechanical vibration amplitudes. Figure 6 shows mutual deviations of the amplitude excursion of the peak value (range), the average value and the mean square value of vibrations.

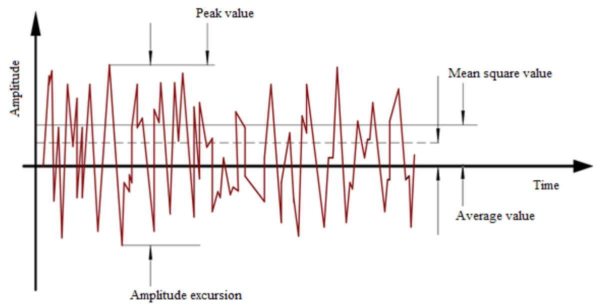

Figure 6. Mutual deviations of the amplitude excursion of the peak value (range), the average value and the mean square value of vibrations

The mean square value (MSV) is an important characteristic of the vibration amplitude. To calculate it, we should square the instantaneous values of the vibration amplitude and average the resulting values by the time. To obtain the correct value, the averaging interval should be at least one cycle of vibrations. After that, the square root is extracted, and the MSV is obtained. This vibration characteristic is widely used in all calculations relating to vibration power and energy. It does not depend on phase shifts between individual components of the measured vibration ranges.

The amplitude excursion (range) is used to quantify the movement of mechanical vibrations. The peak value is the maximum value of mechanical vibrations taken into account when quantifying short-term mechanical shocks. The average value is connected with the temporal development of mechanical vibrations, but its practical application is limited because it is not directly connected with any physical quantity of these vibrations.

3. RESEARCH METHODS

A research bench based on a 4-cylinder 8-valve in-line engine was used as an experimental setup. The choice is determined by the prevailing number of these engines in the domestic auto industry, the possibility of matching the vibration signal with the rotation angle of the crankshaft and the pressure sensor installed in the cylinder, and the ability to choose any possible combinations of clearances in the GDM.

The considered GDM with a direct drive and a rectilinear moving pusher is mounted in the ICE cylinder head and, in particular, contains intake and release valves, which rods move back and forth in the guide bushes, valve springs with their mounting parts and pushers, which move back and forth in the guide holes of the cylinder head. The pushers are equipped with adjusting washers, which are an intermediate element of the kinematic connection between the camshaft lobe and directly the pusher. The latter contact the lobes of the rotating camshaft, which is driven by an endless cogged belt kinematically connected with the drive drum of the ICE crankshaft.

Usingan ICH-10 dial indicator with a 0.01 mm scale, rigidly mounted on the ShM-IIH magnetic stand, we determined the GDM phases and built a circulardiagram with an expansion clearance of release valves of 0.35 ±0.05 mm, intake valves of 0.2±0.05 mm (Figure 7).

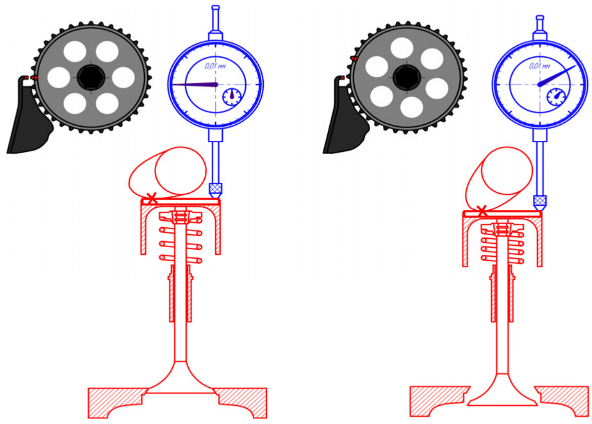

Figure 7. Control using an ICH-10 dial indicator with a 0.01 mm scale, rigidly mounted on the ShM-IIH magnetic stand

To register the vibration signal and its phase parameters in the function of the engine crankshaft rotation angle, we selected vibration instrumentation, a pressure sensor to control the pressure in the cylinder, and an accelerometer (Figure 8).

Figure 8. Vibration instrumentation, a pressure sensor and an accelerometer

We used USB-AutoscopeIII and a laptop with a data analysis software as vibration instrumentation. After measuring the parameters,the oscillogram has a form presented in Figure 9.

Valve timings are accompanied by a shock action. We can accurately determine the structural parameters of the GDM elements only by a shock pulse when the valve is placed on the seat. Knowing the engine operating procedure and the GDM phase, we can mark timings (moments of openings) of all 8 engine valvesin the oscillogram of pressure in the cylinder: cylinder 1 - green; cylinder 2 - blue; cylinder 3 - red; cylinder 4 - yellow.

It can be seen in the oscillogram of Figure 9 that between:

- opening the release valve of cylinder 1 and opening the intake valve of cylinder 2
- opening the release valve of cylinder 3 and opening the intake valve of cylinder 1
- opening the release valve of cylinder 4 and opening the intake valve of cylinder 3
- opening the release valve of cylinder 2 and opening the intake valve of cylinder 4

thecrankshaftrotationangleis 14 degrees.

- opening the intake valve of cylinder 2 and opening the release valve of cylinder 3
- opening the intake valve of cylinder 1 and opening the release valve of cylinder 4
- opening the intake valve of cylinder 3 and opening the release valve of cylinder 2
- opening the intake valve of cylinder 4 and opening the release valve of cylinder 1

thecrankshaftrotationangleis 166 degrees.

Let us marktimings (moments of closures) of all 8 engine valves in the oscillogram of pressure in the cylinder (Figure 10).

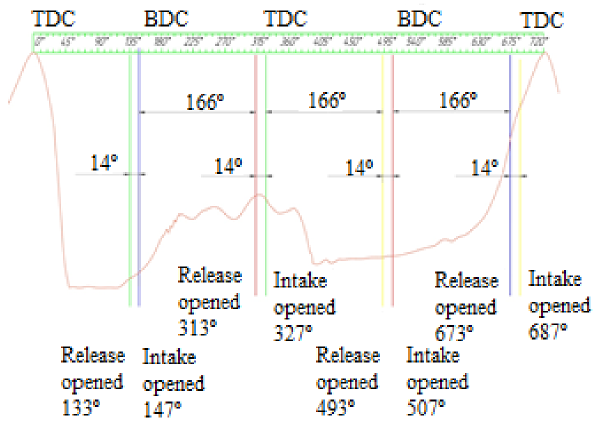

Figure 9. The oscillogram of changes in the pressure in the cylinder synchronized with the ignition pulses (valve timings)

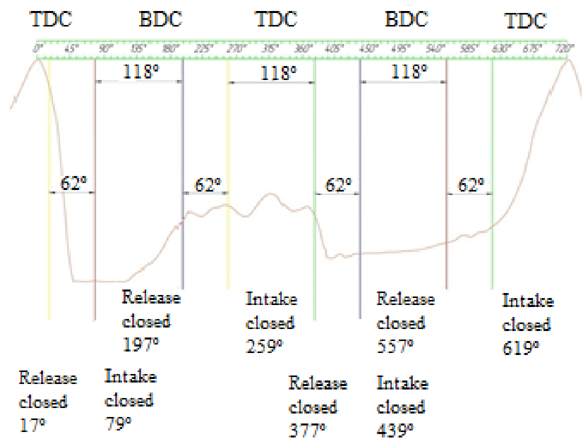

Figure 10. The oscillogram of changes in the pressure in the cylinder synchronized with the ignition pulses (valve timings)

It can be seen in the oscillogram of Figure 10 that between:

closure of the release valve of cylinder 4 and closure of the intake valve of cylinder 3

closure of the release valve of cylinder 2 and closure of the intake valve of cylinder 4 closure of the release valve of cylinder 1 and closure of the intake valve of cylinder 2

closure of the release valve of cylinder 3 and closure of the intake valve of cylinder 1

thecrankshaftrotationangleis62degrees.

Closure of the intake valve of cylinder 3 and closure of the release valve of cylinder 2

Closure of the intake valve of cylinder 4 and closure of the release valve of cylinder 1

Closure of the intake valve of cylinder 2 and closure of the release valve of cylinder 3

Closure of the intake valve of cylinder 1 and closure of the release valve of cylinder 4

the crankshaft rotation angle is 118 degrees.

These data determine the likelihood of whether there can be an overlap of vibrationswhen the valve disk hits the seat during the closure and the moment when the camshaft lobe touches the pusher when the valve of various cylinders begins to open.

When choosing a place of application, we made oscillograms in the absence of expansion clearances of the valves from the valve cover and the outer surface of the cylinder head (Figure 11).

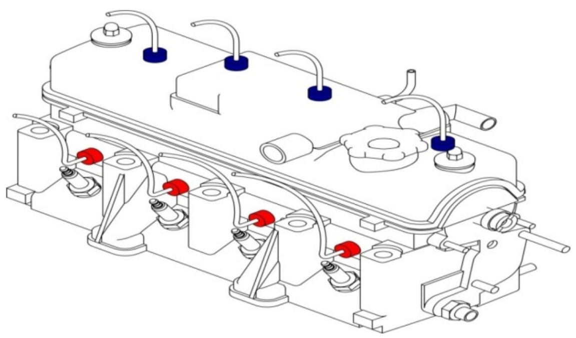

Figure 11. Places of accelerometer application in the absence of expansion clearances of the valves

After that, the data from the sensors were digitized and analyzed.

When comparing the oscillograms, it can be seen that the level (intensity) of noise (interference) has a smaller value when the sensor is installed on the outer surface of the cylinder head.

4. EXPERIMENTAL RESEARCH RESULTS

We measured the vibration pulses and analyzed the obtained data at expansion clearances from 1mm to 0.05 mm with a pitch of 0.05 mm, at the crankshaft speed of 1000min^{-1}, 1300 min^{-1}, 1500 min^{-1}(Figure 12).

As it can be seen from Figure 12, the dependence of the maximum amplitude of the vibration pulse takes on a clearly non-linear form with an increase in the magnitude of the expansion clearance. In the area of large clearances of 0.4-1.0 mm, amplitudes are easily distinguishable without any amplifier equipment and special signal filtering means.

The amplitude of the beginning of the valve opening also contains important information on the technical condition of the expansion clearance. The dependence of the amplitude of the beginning of the valve opening, mV on the magnitude of the expansion clearance in the valve mechanism is shown in Figure 13.

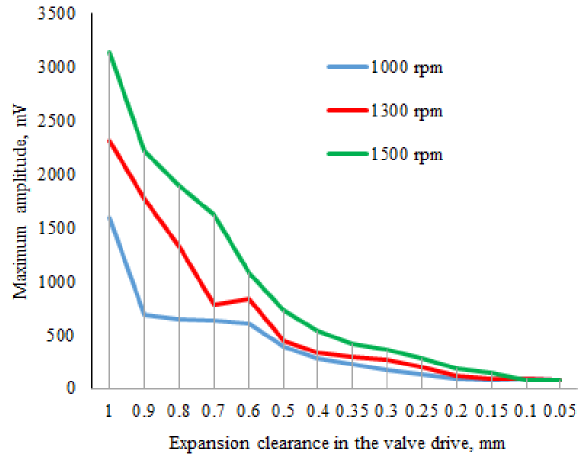

Figure 12. The dependence of the maximum amplitude of the vibration pulse, mV on the magnitude of the expansion clearance in the valve drive, mm

As it can be seen from Figure 13, the amplitude of the beginning of the valve opening is significantly lower than the maximum amplitude of the vibration pulse in Figure 12. The amplitude of the pulse initiation carries the information on the phase components of the signal of the moment when the valve begins to open. The larger it is, the easier it is to distinguish the valve timing.

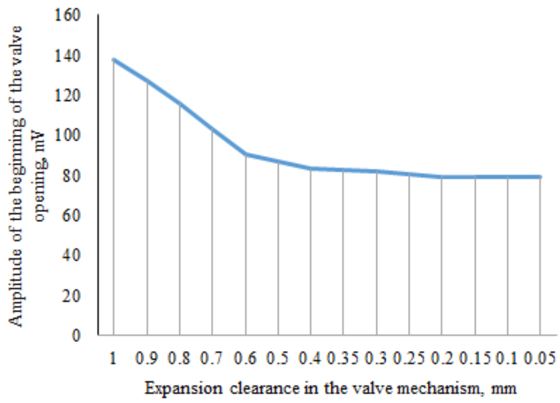

Figure 13. The dependence of the amplitude of the beginning of the valve opening, mV on the magnitude of the expansion clearance in the valve mechanism, mm

The frequency of the vibration pulse is also important. Thus, Figure 14 shows the dependence of the signal frequency, Hz on the magnitude of the expansion clearance in the valve mechanism, mm.

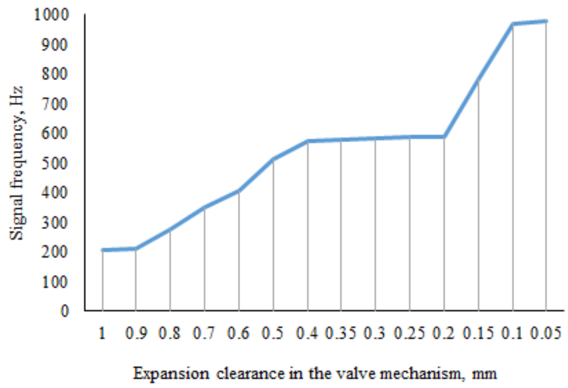

Figure 14. The dependence of the signal frequency, Hz on the magnitude of the expansion clearance in the valve mechanism, mm

From the dependence in Figure 14, we can see a downward tendency of the signal frequency with an increase in the expansion clearance in the valve mechanism. This is natural in terms of the vibro-analysis of small clearances in relation to large ones.

Besides, the phase width of the valve signal is essential for analyzing clearance values. Figure 15 shows the dependence of the phase width of the valve signal, degrees of the crankshaft turn, on the magnitude of the expansion clearance in the valve mechanism, mm.

From Figure 15, we can see a clear downward tendency of the signal phase width with an increase in the expansion clearance in the valve mechanism. However, when analyzing this parameter, it is easy to make an error connected with a poor discernibility of the phase duration at small clearances. Therefore, we need amplifier equipment with noise suppression filters to measure the phases. This will significantly increase the measurement accuracy.

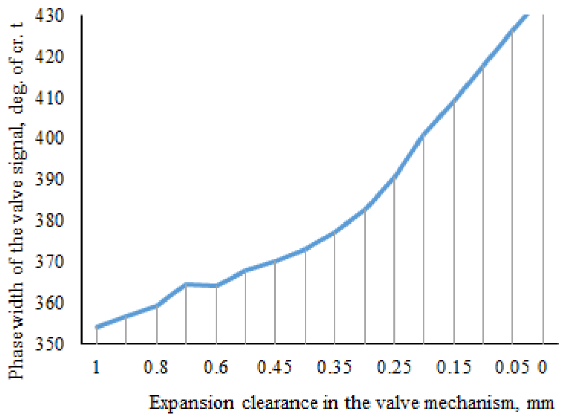

Figure 15. The dependence of the phase width of the valve signal, degrees of the crankshaft turn, on the magnitude of the expansion clearance in the valve mechanism, mm.

In the main part of the experimental research, we studied the amplitude and frequency vibration parameters of the valve mechanisms with varying expansion clearances of 0.9 mm, 0.6 mm, 0.3 mm, 0.2 mm, 0.15 mm, 0.1 mm and 0.05. mm (at the crankshaft speed of 1000 min^{-1}, 1300 min^{-1}, 1500 min^{-1}). The experimental research will be presented in the overlap(Figure 16).

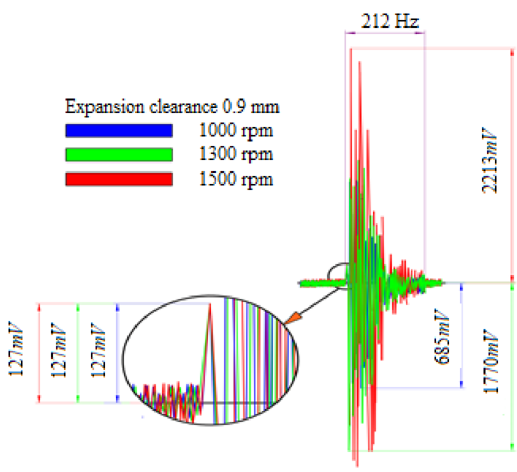

Figure 16. The results of experimental studies of the amplitude and frequency vibration parameters of the valve mechanisms at the expansion clearance of 0.9 mm (at the crankshaft speeds of: 1000 min^{-1}, 1300 min^{-1}, 1500 min^{-1})

An analysis of the data in Figure 16 shows that at the valve clearance of 0.9 mm, the vibration phase was 212 Hz. The initial amplitude of the vibration signal for all the three variants was 127 mV. The maximum amplitude of the vibration signal was: at 1000 min^{-1} - 685 mV; at 1300 min^{-1} - 1770 mV; at 1500 min^{-1} - 2213 mV.

Let us present the results of experimental studies of the amplitude and frequency vibration parameters of the valve mechanisms at the expansion clearance of 0.2 mm (at the crankshaft speed of: 1000 min^{-1}, 1300 min^{-1}, 1500 min^{-1}) (Figure 17).

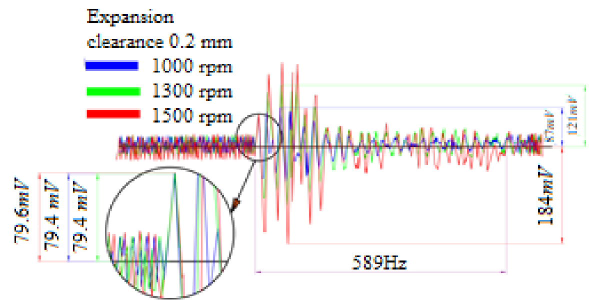

Figure 17. Title The results of experimental studies of the amplitude and frequency vibration parameters of the valve mechanisms at the expansion clearance of 0.2 mm (at the crankshaft speed of: 1000 min^{-1}, 1300 min^{-1}, 1500 min^{-1})

An analysis of the data in Figure 17 shows that at the valve clearance of 0.2 mm, the vibration phase was 589 Hz. The initial amplitude of the vibration signal for all the three variants was 79.4-79.6 mV. The maximum amplitude of the vibration signal was: at 1000 min^{-1} - 87 mV; at 1300 min^{-1} - 121 mV; at 1500 min^{-1} - 184 mV.

Let us present the results of experimental studies of the amplitude and frequency vibration parameters of the valve mechanisms at the expansion clearance of 0.1 mm (at the crankshaft speeds of: 1000 min^{-1}, 1300 min^{-1}, 1500 min^{-1}) (Figure 18).

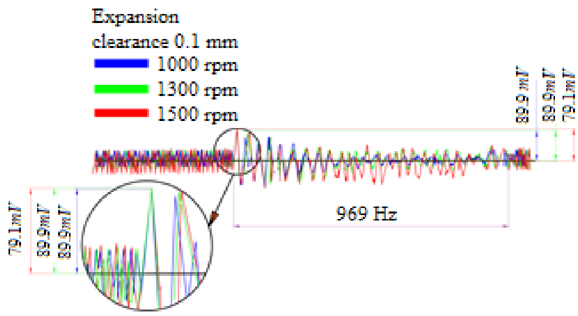

Figure 18. Title The results of experimental studies of the amplitude and frequency vibration parameters of the valve mechanisms at the expansion clearance of 0.1 mm (at the crankshaft speeds of: 1000 min^{-1}, 1300 min^{-1}, 1500 min^{-1})

An analysis of the data in Figure 18 shows that at the valve clearance of 0.1 mm, the vibration phase was 969 Hz. The initial amplitude of the vibration signal for all the three variants was 79.1-89.9 mV. The maximum amplitude of the vibration signal was: at 1000 min^{-1} - 89.9 mV; at 1300 min^{-1} - 89.9 mV; at 1500 min^{-1} - 79.1 mV.

5. CONCLUSION

This work resulted in the creation of a vibro-acoustic system for diagnosing the technical condition of the

GDM valves. The developed methodological techniques for identifying the maximum amplitudes of the vibration signal, initial amplitudes, and phase parameters of the vibration signal allow us to determine the vibration burst of any valve with a high accuracy.

We determined the boundary values of the rise rates of the amplitude values of vibration-shock pulses from individual valves at the ICE crankshaft speed of 1000, 1300 and 1500 min^{-1}.

We established that a vibration pulse could be selectively recognized with a clearance of 0.1 mm at the ICE crankshaft speeds of 1000, 1300 and 1500 min^{-1}, with 0.90 confidence. Clearances of 0.2-0.9 mm are recognized with a higher accuracy of 0.95-0.99. The control accuracy degree grows with an increase in the clearance.

In turn, the rates of vibration occurrence are strictly tied to the time frame and values of the ICE crankshaft speed. However, the lower distinguishability boundary at the noise level does not allow us to fix timely the beginning of the vibro-amplitude formation. In this case, the error can be 3-15 % for the clearance values of 0.1 mm. With an increase in the clearance, the error of recognizing the beginning of the vibration pulse decreases sharply, and with the clearances of 0.3-0.9 mm it is less than 1 %.

We recommend to use the developed schemes (Figures 9 and 10) for recognizing the moments of the vibration pulse formation from various GDM valves to automotive service enterprises to increase the diagnostic accuracy.

As compared to other works, we chose the most informative installation sites for vibration sensors and accelerometers.

During the study, errors from thermal changes in gaps in GDM were determined. Also the time frames, after which the thermal error is minimal was founded. For this engine, these intervals were 8-15 seconds.

The study simultaneously took into account the following parameters and their interconnections: amplitude, phase, frequency, temperature. Moreover, we assessed the amplitudes of the beginning of vibrations from the valve collision. We analyzed the maximum amplitude values. We identified the boundaries along the width of vibration pulses for various clearances in the GDM valves of 0.1-1.0 mm.

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УНАПРЕЂЕЊЕ МЕТОДА ВИБРО-АКУСТИЧНЕ КОНТРОЛЕ МЕХАНИЗМА ЗА ДИСТРИБУЦИЈУ ГАСА КОД СУС МОТОРА

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Савремено возило представља скуп високо прецизних система. Последњих 20 година нису рађена никаква побољшања механизма за дистрибуцију гаса (МДГ), а није се ни смањио број отказа рада МДГ-а. Анализом је утврђено да 30% свих отказа рада СУС мотора отпада на МДГ. Проблем представља велика осетљивост система савремених мотора на одливање уља, квалитет горива, правовременост одржавања, тј. обављање рутинских активности. Анализа радних услова савремених мотора је показала непоштовање закона у више од 50% случајева, што скраћује радни век возила.

Зато је од значаја развијање дијагностичких метода којима се дефинише степен хабања делова мотора у свим фазама рада. Један од таквих је вибро-акустички метод. Циљ рада је развијање методологије и методе за дијагностиковање отказа МДГ-а применом анализе параметара вибрације заједно са сигналом за притисак у цилиндру, нагибом радилице. Анализом је идентификовано стања отказа рада вентила МДГ-а. Развијен је низ алата за селективну дијагнозу елемената МДГ-а. Такође је развијена методологија за одређивање фаза и термичког зазора код вентила применом метода без демонтаже.