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Researches on the sound level induced by operation of valve train components

The surface vibrations generated during operation of valvetrain components in combustion engine are transmitted as sound waves to the vehicle occupants. They can be measured using various techniques, and in particular the matrix sonometers. The most often it is measured the total sound level generated in the engine. Obtaining the data on the sound level generated by a single element of the valve train requires the use of a specific methodology, for example, experimental studies on the engine model. The paper contains a review of measurement techniques of the sound level in combustion engine and the different models used for studies on the sound level of engine valves. Model of the research stand was developed using FEM and presented in the article. The obtained sound levels resulting from the modeled signal introduced in selected locations of the engine valve train model have been presented in the article. There was a non-linear increase in the sound level with an increase in frequency of extortion.

Key words: *combustion engine, valve train, sound level, finite elements method*

Badania poziomu hałasu wywołanego pracą elementów rozrządu

Drgania powierzchniowe generowane podczas pracy elementów rozrządu silnika spalinowego są przekazywane jako fale akustyczne do użytkowników pojazdu. Mogą być mierzone przy użyciu różnych technik, w szczególności matrycy sonometrów. Najczęściej jest mierzony ogólny poziom hałasu generowany w silniku. Uzyskanie danych dotyczacych poziomu hałasu generowanego przez pojedynczy element rozrządu wymaga użycia specjalnej metodologii, na przykład, badań eksperymentalnych na modelu silnika. W pracy dokonano przeglądu technik pomiaru hałasu silnika spalinowego i różnych modeli silników wykorzystywanych do badań hałasu zaworów. Model wybranego stanowiska badawczego został opracowany przy wykorzystaniu MES i przedstawiony w artykule. Uzyskane poziomy hałasu wynikajace z modelowanego sygnału wprowadzonego w wybranych miejscach modelu rozrządusilnika zostały przedstawione w artykule. Zaobserwowano nieliniowy wzrost poziomu hałasu ze wzrostem częstotliwosci wymuszenia.

Słowa kluczowe: *silnik spalinowy, rozrząd silnika, poziom hałasu,metoda elementów skończonych*

1. Introduction

Nowadays in Poland the researches of total sound level of th vehicle should be carried out according to the norm PN-92/S-04051 [1].

As the sound level caused by combustion process in the engine usually exceeds the sound level of valve train, the results of this measurement rarely provide valuable information on the state of the valve train components. For more information about the sound level generated by the valve train can be achieved during the motor test or out-of-motor models of valve train.

According to the [2] the accelerating and closlier the valve settling speed has the greatest impact on the sound the valve train. During the opening of the valve it is important the time for erasing the valve clearance. The larger the valve clearance the higher sound level occurs. In the case of mechanical control the clearance, its value is limited due to the thermal expansion of the valve train components. However, during the on-heating of the engine there is higher level of sound level of valve train.

Currently, cam valve trains in internal combustion engines are equipped with hydraulic compensators that keep the valve clearance with a zero value. Worn or misstatement-rectly supported timing components may affect the operation of rockers and cause noise timing or improper performance of the engine. equipped with a hydraulic valve clearance compensators that maintain zero valve clearance value. Worn or improperly handled valve train components may affect the operation of rockers and cause sound of valve train or improper performance of the engine.

If the value of the clearance between the end of the rocker and the valve stem is greater than such specified by the manufacturer, it may mean that the lever axis or pushers are worn out, and this can cause the sound clicks when idling and low speed values [3].

Valuable results of comparative tests of sound the valve trains driven by gears: chain and belt drive, are presented in [4]. The sound level in the case of toothed belt transmissions was smaller with up to 5 dB than in the case of chain transmission. In both cases, an approximately linear increase of the sound level is observed with the increase of engine speed.

You can now also find cars with camless valve train, in which the problem of soung generated while valve operation is still important.

During the tests described in [5], valve movement cyclically forced by single-acting hydraulic actuator, after reaching a stroke of 8 mm, has ended up with settling valve into its insert with the speed in the range 1 ÷ 1.3 m/s. Occasionally, the obtained valuesof settling speed of about 1.4 to 1.5 m/s, were accompanied by a clear sound during testing. Only at smaller strokes, the settling velocity of the valve into its insert was of less than 1 m/s: for example, for stroke approx. 3 mm - valve settling velocity has been of 0.8 m/s and for the stroke of 1.2 mm - 0.4 m/s. The high values of settling speed increased the sound of valve train. It is therefore necessary the braking system of the valve, allowing the reduction of the mentioned settling speed to one of less than 0.1 m/s.

The tests of sound level for valves driven electromagnetically on the test bench were shown in [6, 7].

In the current paper it has been analyzed vibration levels generated during strikes of valves into inserts on the test bench. Valve drive has held through the camshaft driven by the electric motor.

A model of the test bench has been developed using Finite Element Method. In this model, the acoustic pressure distributions is calculated for different excitation frequencies identified with the frequency of the valve strokes into insert.

The aim of the analysis is to obtain the relationships between sound pressure/level and the excitation frequency, and to compare it with the dependence obtained during measurements of the total sound level on the test bench as a function of the camshaft speed.

2. Testers for studies on the sound level

The noise tests described in [2] was performed on a physical model of a single section of the valve, separated from the real four-stroke engine valve train. The valve was driven by the camshaft, using the flywheel without causing extra sound. The sound level was measured for ever smaller valve clearance values in the range from 1 mm to 0 mm. In the latter case (zero valve clearance), while closing the valve, there was no contact between the valve head and its insert. This state is not allowed during the real operation of the engine, as it would lead to its destruction.

During the studies it was reported two local maxima in valve sound course during registered time. The first was due to the prevalence and the reset of valve clearance when opening the valve. The second was due the influence of the valve settling velocity into its insert during closing the valve. The differences between the two local maxima resulted from the fact that during opening period the cam has excited some of valve train elements, and during closing period the valve has hit into the system insert - cylinder head.

At FEV [8] a virtual model of the entire drive, containing sub-models of individual components of the drive is used to determine the natural frequencies, vibration and noise propagation in the vehicle already in the design phase and during further checking in the development process of the drive system. Researches on the virtual bench simulations rely on a combination of Multi-Body Simulations and Finite Element Method. This allows accurate calculation of the excitation mechanisms, as well as the transfer of structural behaviours. Calculation of mechanisms generating vibrations is carried out using models of FEM, including rigid bodies connected by means of rigid or flexible joints. Rigid body reflect masses and inertia the individual elements of the system, and the joints reflect load-bearing capacity of the bearings. Dynamic effects of elastic structures are computed using the Finite Element Method and reduced using special computational procedures to a few degrees of freedom (usually energy-equivalent) before performing a full multiple simulations. Finite element calculations made in the time domain allow the execution of multi-element simulation for one cycle and obtaining as a result the audible noise. Thus, various acoustic systems can be tested for frequency, the total sound level and quality. In addition to the evaluation of the surface speed it can be calculated sound propagation in the air. The calculations of the sound in the air can be done with varying degrees of detail - from simple solutions with empirically set degrees of propagation to advanced calculations using the Boundary Element Method. This calculation methodology developed by FEV has been used and proven successful for many engines.

In [9] it was studied the mechanism of valve settlement as a source of vibration and the vibration transmission through the cylinder head. The aim was to determine the spectral characteristics of the excitation and to reference it to the structural and mechanical properties of the camshaft in the cylinder head. Researches were carried out on the cylinder head of the DOHC 1.5 litre diesel engine at the speed of 1810 rpm.

Vibrations of the cylinder head generated by one operating valve train were measured in a convenient location on the structure. Vibrations caused by hitting of the settling valve were extracted from the total vibration signal and used to recover the indirectly measured impact force. The recovered force was determined by inverse filtering the vibrations of the cylinder head using the transfer function of the cylinder head. Transfer function of the cylinder head was measured between the place of observation of the cylinder head and the valve/insert contact zone.

It was found that the transfer of vibrations comprises two transmission paths: the settlement force transfers energy to the cylinder head through the insert, and through the valve train and camshaft. The path through the valve train is the main path, because resonances of valve train can increase the transmission of vibrations.

During studies of vibrations and engine sound as described in [10] it has been used a four-cylinder SI engine. On the second cylinder it was mounted, via a special adapter, the sensor of gas pressure in the cylinder. In the middle of the outlet side of the cylinder head it was mounted the acceleration sensor for vibration measurement. At a distance of 10 cm from the centre of the upper surface of the cylinder head it was placed the microphone to the measurement of sound level. The studies were carried out at two fixed engine speeds: idle one of 800 rpm and one of 3500 rpm. The signals were recorded and processed using the Short-Time Fourier Transform, Wiegner-Ville Distribution and Wavelet Transform.

For the tester it was used the cylinder head of the 1.6 l engine with twin overhead cam timing which drove 16 valves. The inlet camshaft was driven by an electric motor with speed and torque varied by the controller. The valve train elements were lubricated with oil pressurized to 0.2 MPa, which was supplied from the test bench lubrication system. On the cylinder head it was mounted several acceleration sensors, of which one was always positioned in the middle of the outlet side of the cylinder head. One accelerometer was placed on the inlet valve head. It was also measured the angle and the speed of the electric motor. The study was carried out at the speed of the camshaft equal 1800 rpm. The signals were recorded and processed using the FFT analyzer.

It was noted that there are two main sources of vibrations. One of them was interaction between the camshaft and the tappet caused by the dynamic forces. The other was caused by hitting of the settling valve. Considering the transfer characteristic for each source and path corresponding to it, it was found that the strength of cam interactions and the impact force were the dominant source of vibrations up to 6 kHz, while the impact force was dominant only for the frequency range 10 - 20 kHz.

3. Models describing sound level of valves in the valve train

In the paper [11] it was described the 1-order model of the system sound, which is used in the design of an engine system, and which is a semi-empirical model of quasi-constant characteristics (not including the crank angle). Coefficients and constants of such a model are characteristic for the engine group, for which they are designated and may not be transferred to the generalized formulas for the absolute magnitude of sound level. This type of model is useful for understanding basic parametric dependences of sound. It can be used for coarse determination of the relative influence or trends.

The level of engine total sound can be expressed by the formula (1),

$$p_{SPL,E} = \sum_i p_{SPL,i} \tag{1}$$

where: $p_{SPL,E}$ is the total sound pressure level of the engine measured at a distance of one meter from the surface of the engine, $p_{SPL,i}$ reflects the share of combustion, strokes of piston, valve train, fuel injection, and accessories.

In [12] it has been noted that engine sound induced by the valve train can be expressed by the formula (2),

$$p_{SPL,Valvetrain} = f\left(N_E^{5.8}, f_{VT}, c_{VT}\right) \tag{2}$$

where: N_E - engine speed, f_{VT} - structural factor of valve train, c_{VT} - valve clearance in valve train.

As it was described in [10], in order to understand the dynamic behaviour of the OHC-type valve train in SI engine, it was developed a simple harmonic oscillator with four degrees of freedom and a model of the reduced masses, which was verified, in terms of their usefulness, with experimental results. That model was used in the design process to make the modifications and to obtain structures with improved sound quality.

Vibration sources were identified through the analysis of dynamics obtained from the simulations of final experiment on a mounted engine and on the test bench.

4. Tester used to study the sound the valves driven by camshaft

The scheme of the test bench for studies on wear the valves, their inserts and guides and on a sound of cam-driven valves in valve train is shown in the Figure 1 [7]. For the construction of the test bench it was used two-cylinder, inline injection pump of diesel engine. On the test bench there was a possibility of adjusting the valve lift and simultaneously, but in an indirect way, the speed of the valve, by changing the clearance between the tappet and valve using the adjusting screw. On the test bench, the control of relationship between valve lift and speed was realized indirectly, through simultaneous measurement of valve lift and acceleration.

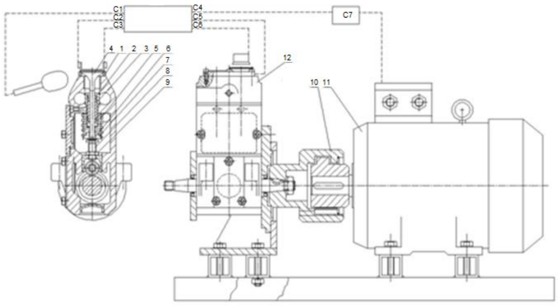

Fig. 1. The test bench for studies on wear the valves, their inserts and guides and on a sound of valves driven by camshaft [7]; C1 - microphone, C2 - valve displacement sensor, C3 - valve acceleration sensor, C4 - the engine speed sensor, C5 – seat insert temperature sensor, 6 - heater, C7 - control cassette, 1 – seat insert, 2 - sleeve, 3 - guide, 4 - valve, 5 – valve spring, 6 - locks, 7 - retainer, 8 - adjustment screw 9 - lock nut, 10 - clutch, 11 - the electric motor, 12 - pump

On the test bench there was the possibility of free valve rotation, change the valve lift and camshaft speed up to 2800 rpm. The temperature of inserts heated by the hot air stream was controlled by thermocouples and could be varied in the range of 293-793 K. During series of measurement it could be measured the volume wear and the mass wear of the valve, its insert and guide, and the level of total sound using the sonometer.

5. Models of the acoustic wave propagation and of the tester for studies on sound level the valves

It has been made the following assumptions:

- air occurring in the modelled area is the compressible, non-viscous fluid, there is no specific flow, the average density and pressure are uniform throughout the area of air.

- the acoustic wave in the air has a form described by the equation (3) [6]

$$\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} - \nabla^2 p = 0 \qquad (3),$$

where: p - sound pressure, $c = \sqrt{E/\rho_o}$ - the speed of sound in air, E – bulk modulus of fluid, ρ_0 - air density, t - time.

- displacements in nodes for the structure of the metallic elements are calculated from the equation (4) [6]:

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F\} \qquad (4)$$

where: $[M]$ = structure mass matrix, $[C]$ = structure damping matrix, $[K]$ = structure stiffness matrix, $\{\ddot{u}\}$ - at nodes acceleration vector, $\{\dot{u}\}$ - at nodes velocity vector, $\{u\}$ - at nodes displacement vector, $\{F\}$ - forces vector.

- for harmonically varying excitation of the structure, acoustic pressure oscillations caused by such excitation are described by the equation (5) [6]:

$$\frac{\omega^2}{c^2}\overline{p} - \nabla^2 \overline{p} = 0 \qquad (5)$$

where: $\omega = 2\pi f$, and f - frequency of excitation.

- for the contact between elements of the air and the elements of the structure, it is used the equation (6) [6]:

$$\{n\}\{\nabla p\} = -\rho_0 \{n\}^T \left\{\frac{\partial^2}{\partial t^2}\{u\}\right\} \qquad (6)$$

where: $\{n\}$ - unit vector normal to the surface of the air, ρ_0 - air density, $\{u\}$ - vector of displacements in nodes of the structure being in contact with air.

- on the border of fluid it has been assumed the full absorption of sound (7) [6]

$$\int_{vol} \delta P \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} d(vol) - \int_{vol} \delta p \nabla^2 p d(vol) + \int_S \delta p \left(\frac{r}{\rho_0 c}\right)\frac{1}{c}\nabla^2 p \frac{\partial p}{\partial t} d(S) = 0 \qquad (7)$$

where: r - absorption coefficient for air border.

The calculation of acoustic pressure distribution, on the test bench for measurement of wear the valves, has been made in the model of such test bench elaborated using the Finite Element Method. The geometry of this model is shown in the Figure 2.

The mentioned model includes only a simplified geometry of the pump body and of valve heads, because their outer surfaces are a direct source of the acoustic wave propagating through the air. The bottom surface of the body has been fixed during calculation. It has been assumed that the source with the highest signal strength is hitting of valves into their inserts. Other sources of sounds have been omitted.

The body has been surrounded by a layer of air. On the interface of the air and aluminium elements it has been introduced suitable boundary conditions. The whole volume has been surrounded by an air sphere with the radius of 0.5 m. At the border it has been placed finite elements mapping the sound absorption effect in the extending to infinity area of air. As the excitation, it has been introduced the harmonically varying displacement of the valve surface with an amplitude of 0.0001 m and a fixed frequency which has been changed for each case of the calculation, assuming a value between 1 - 30 Hz. The mentioned excitation has been to map vibrations that occur during hitting the valves into their inserts during experimental investigations.

To simplify the calculation it has been assumed that all modelled solid structures are homogenous solids.

Finite element grid was made automatically by the commercial programme [13] and shown in the Figure 3. It also presents the boundary conditions. For metallic structures it was used the spatial 8-nodes finite elements SOLID45 [13] with the degrees of freedom being displacements in the direction of the OX, OY and OZ axis. For the air area, in which the sound pressure distribution was calculated, it was used the spatial 8-nodes element FLUID30 [13], in which the degree of freedom has been the pressure. For a one-element layer being in direct contact with the metallic structure parts it was used the spatial 8-nodes finite elements FLUID30 [13], in which degrees of freedom were pressure and displacements in the direction of the axis OX, OY and OZ. At the border of the air volume it was introduced 4-nodes surface finite elements FLUID130 [13] representing the sound absorption effect of air through the area extending to infinity, outside the area containing finite elements FLUID30 [13]. It could be the degenerated form of finite elements: tetrahedral one for the SOLID45 and the FLUID30 and triangular one for the FLUID130 [13]. In the nodes on the outer surfaces of the valve heads the harmonically varying displacements UY, with the set amplitude and frequency, were introduced as the excitation.

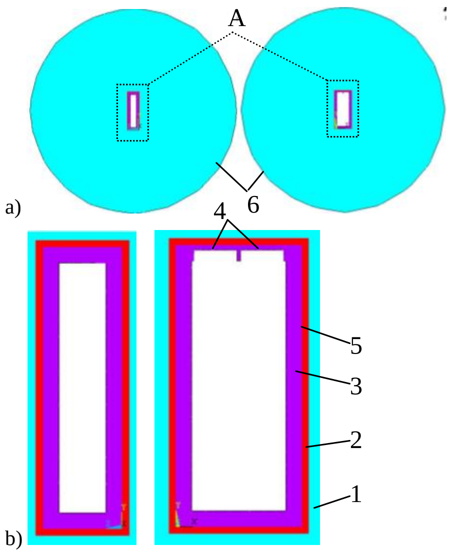

Fig. 2. The scheme for the model geometry. a) cross-sections of the model by planes of symmetry in two perpendicular views, b) zoomed fragments A of cross-sections from the Figure 1a; 1 - air area, 2 - intermediate layer of air in contact with metallic elements, 3 - body, 4 - valves, 5 - the interface between the metallic elements and air, 6 - air border surface area

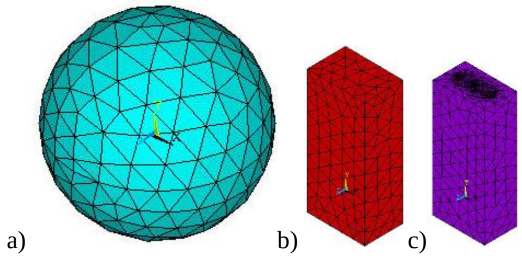

Fig. 3. The finite element grid and boundary conditions. a) inside: FLUID30 [13] with pressure as the degree of freedom, on the outer surface: FLUID130 [13], b) FLUID30 [13] with pressure and displacements UX, UY, UZ as degrees of freedom, c) SOLID45 [13] – with displacements UX, UY, UZ as degrees of freedom, in the nodes on the outer surfaces of the valve heads it has been introduced the harmonically varying displacement UY with the set amplitude and frequency.

It has been assumed the value of the reference pressure to be equal p_{ref} = 2 * 10-5 Pa [6].

It allowed determination of the sound level in decibels according to the formula (8) [6].

$$L_p(t) = 20\log_{10}\frac{p_t}{p_{ref}} \tag{8}$$

6. Results of calculations

The measured sound level as a function of the rotational speed of the camshaft for different valve strokes was shown in the Figure 4 [7].

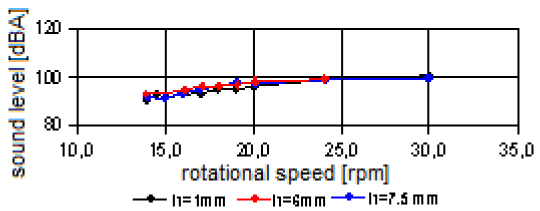

Fig. 4. The course of the average sound level against speed of the camshaft; black line - 1 mm valve stroke, the environmental sound level of 40 dBA, the red line - 6 mm valve stroke, the environmental sound level of 50 dBA, blue line - 7.5 mm valve stroke, the environmental sound level of 65 dBA [7]

The measured sound level grew slowly with the increase in camshaft speed, stabilizing further. Changes in the sound level were practically independent of valve stroke.

Acoustic pressure distributions obtained from the calculation for different excitation frequencies were shown in the:

Figure 5 - for the excitation frequency of 1 Hz,

Figure 6 - for the excitation frequency of 10 Hz,

Figure 7 - for the excitation frequency of 13Hz,
Figure 8 - for the excitation frequency of 15 Hz,
Figure 9 - for the excitation frequency of 20 Hz,
Figure 10 - for the excitation frequency of 25 Hz,
Figure 11 - for the excitation frequency of 30 Hz.

Figure 12 contains a zoomed part of the Figure 11.

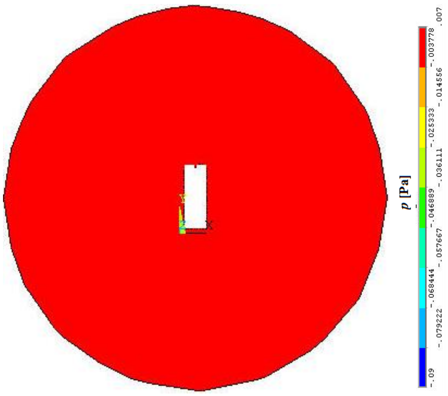

Fig. 5. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 1 Hz

Fig. 6. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 10 Hz

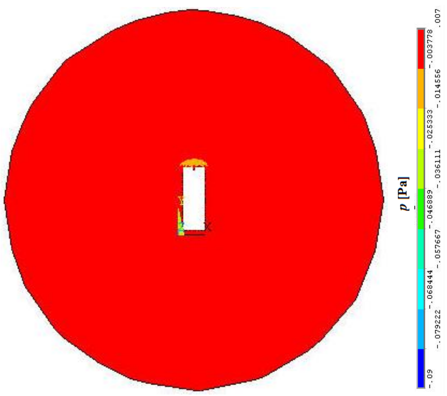

Fig. 7. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 13 Hz

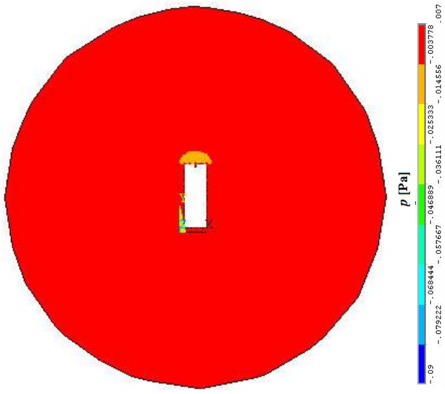

Fig. 8. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 15 Hz

The Figure 13 shows a graph of acoustic pressure as a function of excitation frequency at a point far from the valve by 0.01 m, what corresponds to the placing of boundary surface of sonometer during experimental investigations.

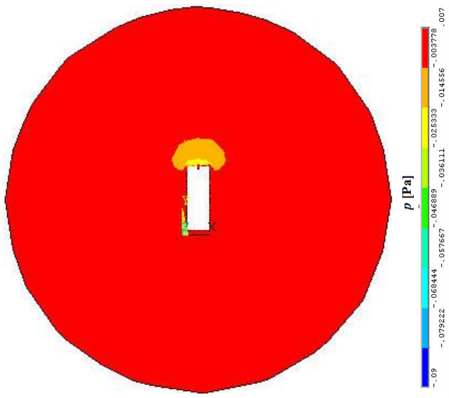

Fig. 9. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 20 Hz

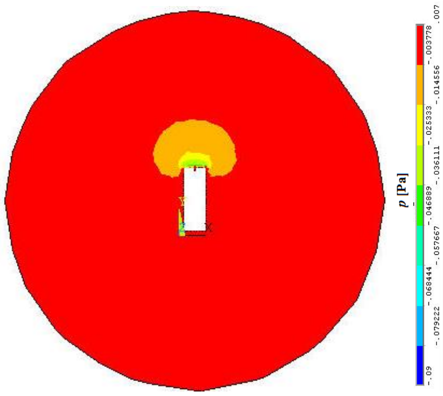

Fig. 10. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 25 Hz

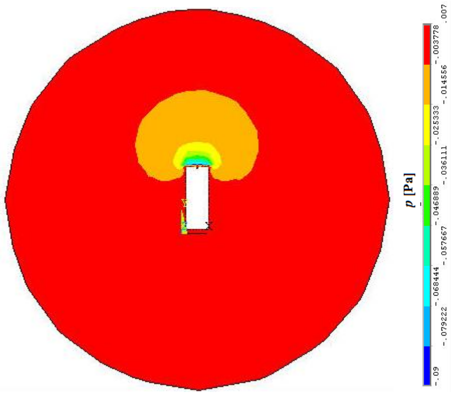

Fig. 11. The distribution of acoustic pressure *p* for the harmonic excitation with the amplitude of 0.0001 and the frequency of 30 Hz

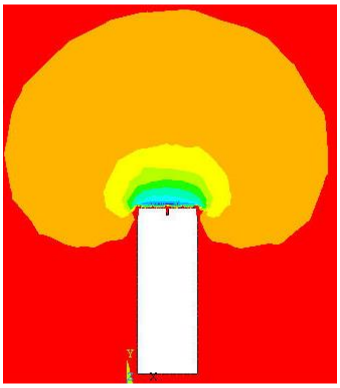
Fig. 12. The zoomed part of the Figure 11

As it is apparent from the observation of the Figures 5 - 12 the clear increase in acoustic pressure *p* has begun from the excitation frequency of 13 Hz.

The Figure 14 shows linear change of the phase Shift in acoustic pressure as a function of the excitation frequency.

The Figure 15 shows the logarithmic increase of sound level as a function of the excitation frequency what corresponds to the exponential increase of the acoustic pressure from the Figure 13.

For the excitation frequency equal 30 Hz the calculated sound level corresponding to the maximum acoustic pressure *p* exceeded slightly the value of 93 dB.

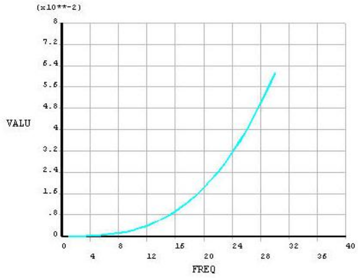

Fig. 13. The graph of acoustic pressure amplitude as a function of the excitation frequency at the point far from the valve by 0.01 m.

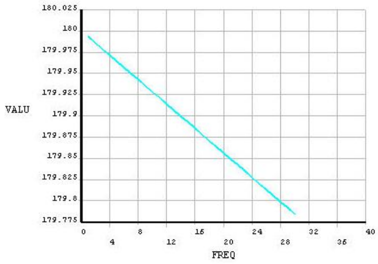

Fig. 14. The graph of the phase shift for the acoustic pressure as a function of the excitation frequency at the point far from the valve by 0.01 m

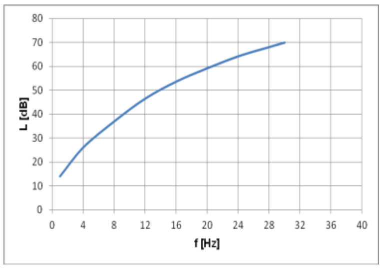

Fig. 15. The graph of the sound level as a function of the excitation frequency at the point far from the valve by 0.01 m

8. Summary

Calculated values of acoustic pressure increased exponentially with the increase of the excitation frequency. It corresponded to the logarithmic increase of the sound as the function of the excitation frequency, similarly to the increase of the total sound level, measured in the test bench, as a function of excitation frequency.

The calculated sound level was lower than measured one. It was resulted from that, the measured sound level was influenced, beyond the impact of valve strokes into insert, by the other sound sources, i.e. cam impact on the tappets.

Nomenclature/*Skróty i oznaczenia*

FEM, MES Finite Element Analysis / *Metoda Elementów Skończonych,*

N_E Engine Rotating Speed / *Prędkość obrotowa silnika,*

L_p Sound Level / *Poziom hałasu,*

t Time/*Czas,*

ρ_0 Density of Air / *Gęstość powietrza,*

c_{VT} Valve clearance / *Luz zaworowy,*

u Displacement / *Przemieszczenie,*

r Absorption coefficient / *Współczynnik absorpcji,*

p Acoustic Pressure / *Ciśnienie akustyczne,*

f Frequency / Częstotliwość,

c Acoustic velocity in Air / *Prędkość dźwięku powietrza,*

E Bulk Modulus of Air / *Moduł sprężystości powietrza*

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