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DEVELOPMENT OF A MEASUREMENT SYSTEM FOR DYNAMIC CALIBRATION OF PRESSURE SENSORS

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Abstract – The aim of this paper is to present the developed measurement system for dynamic calibration of the pressure sensors at different average pressures. The dynamic pressure generator under discussion consists of the pneumatic cylinder, which is divided by the piston into two pressurized gas chambers. The piston-rod assembly is driven by an electrodynamic shaker. Experimental analyses and mathematical modelling were employed to demonstrate its capabilities and limitations.

Keywords: dynamic calibration, pressure pulsations, pneumatic cylinder, natural frequency

1. INTRODUCTION

Accurate dynamic pressure measurements are important in many application areas [1,2]. In automotive industry the pressure in the combustion chamber and in exhaust system is used as a control parameter to determine the amount of fuel to inject in the cylinders to achieve low emissions and low fuel economy. The measurements of dynamic pressure are vital in steam and gas turbines found in power plants and in jet engines used in aviation to determine the efficiency of turbomachines. In the engineering of buildings as well as for spacecraft, aircraft and other vehicles, aerodynamic time-dependent pressure field is measured in order to determine a distribution of aerodynamic forces. In medicine the cardiac dynamic blood pressure value provides an important tool for diagnosis, treatment, and prognosis both of critically ill patients and also of those recovering from surgical procedures. The increasing use of the pressure sensors to monitor rapidly varying physical quantities requires the use of sensors of suitable dynamic characteristics. This caused growing needs for dynamic testing and calibration of such sensors.

For dynamic calibration of pressure sensors an appropriate dynamic pressure generator with defined dynamic characteristics is required. Due to the fact that the amplitude and frequency pressure calibration requirements vary widely (e.g., from relatively small amplitudes of few Pa at frequency of few Hz added to high static bias pressure for turbomachinery applications to few hundreds of MPa at frequencies up to 100 kHz for studies in automotive applications), there is a variety of operating principles and configurations of the dynamic pressure generators. In general, dynamic pressure generators are divided into two classes: aperiodic and periodic [3]. The former are used for calibration methods in the time domain and the latter for calibration methods in the frequency domain. Aperiodic dynamic pressure generators create pressure steps or single pressure pulses resembling a half-sine wave and include the shock tube, quick-opening valve devices, the drop hammer and explosive devices. The range for use of aperiodic pressure generators is quite wide as they are able to generate high amplitudes and cover practically all the areas of industrial use of pressure sensors (quick-opening valve devices cover the lowest frequency ranges, whereas the shock tubes cover the highest frequencies). The frequency response of the pressure sensors can be directly obtained using periodic pressure generators. The pressure generated by a periodic dynamic pressure generator is a periodic function such as a harmonic-wave, a square-wave or any other periodic function. Periodic pressure generators include loudspeakers, sirens, rotating valves, shaker-based inertial loading systems, shaker-based direct force loading system and piston-in-cylinder steady-state generators (pistonphones). The main limitation of the periodic dynamic pressure generators is difficulty to achieve harmonic pulsations at higher amplitudes and frequencies in a gaseous medium due to nonlinearities that result from physical gas dynamics [4].

The purpose of this paper is to present our developed measurement system for dynamic calibration of the pressure sensors and its dynamic characteristics at different average pressures. The pressure pulsations in the measurement system are generated by the periodic dynamic pressure generator, which comprises the pneumatic cylinder, in which piston is driven by an electrodynamic shaker. In order to minimize the initial preload of the shaker due to the pressure acting on the piston, the cylinder chambers above and below the piston are pressurized with the same initial static pressure. In order to demonstrate its capabilities and limitations, the theoretical and experimental analyses were performed.

The mechanical implementation of the pressure generator was developed with the help of mathematical modelling. A mathematical modelling using the lumped physical-mathematical model is described in Section 2. The realization of the developed measurement system is presented in Section 3. In Section 4 the main characteristics of the pressure generator are discussed.

2. PHYSICAL-MATHEMATICAL MODEL

The developed dynamic pressure generator consists of the pneumatic cylinder, which is divided by the piston into two pressurized gas chambers. The piston-rod assembly is driven by an electrodynamic shaker. The geometrical parameters of the physical-mathematical model are schematically presented in Fig. 1. The main properties of the mathematical model are presented in this paper, but see, for instance, [5–7] for details of the derivation of similar lumped-element models.

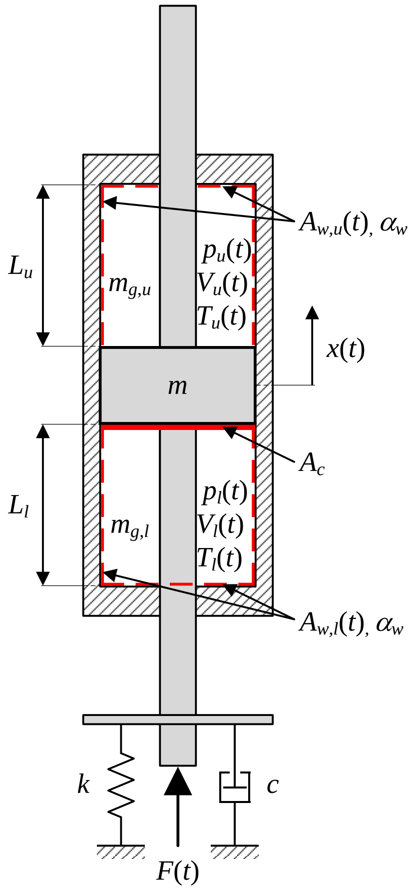

Fig. 1. Model of the dynamic pressure generator with two pressurized gas chambers.

The piston-rod assembly is modelled as a solid body with one degree of freedom that is defined by its position $x(t)$ in an upward direction along the cylinder (starting position is in the middle of the cylinder). The motion equation for the piston-rod assembly can be written as:

$$m\frac{\mathrm{d}^2x(t)}{\mathrm{d}t^2} = F(t) - kx(t) - \left(p_u(t) - p_l(t)\right)A_c - c\frac{\mathrm{d}x(t)}{\mathrm{d}t} - mg, \quad (1)$$

where m is the mass of the piston-rod assembly, $F(t)$ is the external shaker's force applied to the piston-rod assembly, k is the spring constant of the piston-rod assembly (considering the effects of the piston seal, the shaker's suspension stiffness …), c is the damping coefficient, $p_l(t)$ and $p_u(t)$ are the absolute pressures in lower and upper gas chambers of the pneumatic cylinder, respectively, and $A_c = \pi(D_c^2 - D_{rod}^2)/4$ is the piston effective area.

Taking into account the relations $\gamma = c_p/c_v$ and $c_p = c_v + R$, where γ is the adiabatic index, c_p is the specific heat at constant pressure and c_v is the specific heat at constant volume, considering the pressures and temperatures in the upper and lower gas chambers of the cylinder to be spatially homogeneous (which holds true for the wavelengths of the pressure pulsations, which are large compared with the linear dimensions of the chambers), ideal gas and neglecting the kinetic and gravitational energies, the energy equation for the closed system gives:

$$\frac{1}{\gamma-1}\left(\dot{p}_i(t)V_i(t) + p_i(t)\dot{V}_i(t)\right) = -p_i(t)\dot{V}_i(t) - \dot{Q}_i(t), \quad (2)$$

where the subscript $i = u$ or l (u denotes the upper and l the lower gas chamber). The volumes of the gas in each chamber of the pneumatic cylinder are $V_u(t) = A_c(L_u - x(t))$ and $V_l(t) = A_c(L_l + x(t))$, respectively, where L_i is an initial height of the chamber. In these equations, the heat exchange with the surroundings $\dot{Q}_i(t)$ is modelled as the convective heat transfer between the gas with the spatially averaged temperature $T_i(t)$ and the cylinder wall with the temperature T_w:

$$\dot{Q}_i(t) = \alpha_w A_{w,i}(t)\left(T_i(t) - T_w\right), \quad (3)$$

where α_w is the convective heat transfer coefficient and the surface areas for the upper and lower chamber are $A_{w,u}(t) = A_c + \pi D_c(L_u - x(t))$ and $A_{w,l}(t) = A_c + \pi D_c(L_l + x(t))$, respectively.

The equation of state for air in each gas chamber can be written as:

$$p_i(t)V_i(t) = m_{g,i}RT_i(t), \quad (4)$$

where $m_{g,i}$ is the mass of the gas and R is the gas constant.

Considering (1)–(4), the physical-mathematical model can be derived as a system of six, nonlinear, first-order differential equations for the piston position $x(t)$, the piston velocity $v(t)$, the gas-chambers pressures $p_u(t)$ and $p_l(t)$, and the gas-chambers temperatures $T_u(t)$ and $T_l(t)$ as a function of time t:

$$\begin{aligned}
&\dot{x}(t) = v(t),\\
&\dot{v}(t) = \frac{1}{m}\left[F(t) - kx(t) - \left(p_u(t) - p_l(t)\right)A_c - cv(t) - mg\right],\\
&\dot{p}_u(t) = \frac{1}{A_c(L_u - x(t))}\left[\gamma p_u(t)A_c v(t) - \right.\\
&\qquad \left. -\left(\gamma-1\right)\alpha_w\left(A_c + \pi D_c(L_u - x(t))\right)\left(T_u(t) - T_w\right)\right],\\
&\dot{T}_u(t) = \frac{T_u(t)}{p_u(t)A_c(L_u - x(t))}\left[\left(\gamma-1\right)p_u(t)A_c v(t) - \right.\\
&\qquad \left. -\left(\gamma-1\right)\alpha_w\left(A_c + \pi D_c(L_u - x(t))\right)\left(T_u(t) - T_w\right)\right],\\
&\dot{p}_l(t) = \frac{1}{A_c(L_l + x(t))}\left[-\gamma p_l(t)A_c v(t) - \right.\\
&\qquad \left. -\left(\gamma-1\right)\alpha_w\left(A_c + \pi D_c(L_l + x(t))\right)\left(T_l(t) - T_w\right)\right],\\
&\dot{T}_l(t) = \frac{T_l(t)}{p_l(t)A_c(L_l + x(t))}\left[-\left(\gamma-1\right)p_l(t)A_c v(t) - \right.\\
&\qquad \left. -\left(\gamma-1\right)\alpha_w\left(A_c + \pi D_c(L_l + x(t))\right)\left(T_l(t) - T_w\right)\right].
\end{aligned} \quad (5)$$

The system of differential equations is solved with the fourth-order, Runge-Kutta, fixed-step method. In the simulations within this paper, we considered the initial conditions $p_u(0) = p_l(0) = p_0$, $T_u(0) = T_l(0) = T_w$, $x(0) = 0$ and $v(0) = v_0$, where such value of v_0 is selected that the periodic steady-state response is quickly achieved. The dimensions of the pneumatic cylinder and the mass of the piston-rod assembly refer to the actual configuration of the dynamic pressure generator, as discussed in Section 3, see Table 1. The effective spring constant k and the effective damping coefficient c were estimated based on the experimental results at the average absolute pressure of 600 kPa in the vicinity of the resonance frequency. The convective heat transfer coefficient α_w was assumed to be a constant value. The simulations were performed for different average pressures by considering the harmonic external shaker's force $F(t) = F \sin(2\pi f t)$, where the amplitude of the force F was set to the value that results in the generated relative pressure pulsation of approximately 1% at the frequency $f = 5$ Hz.

Table 1. Dimensions and effective lumped parameters referring to the actual dynamic pressure generator.

Parameter	Description	Value
m	Mass of the piston-rod assembly	0.839 kg
k	Effective piston-rod spring constant	$650 \cdot 10^3$ Nm^{-1}
c	Effective damping coefficient	550 Nsm^{-1}
g	Acceleration of gravity	9.81 ms^{-2}
D_c	Inner diameter of the cylinder	0.08 m
D_{rod}	Diameter of the piston rod	0.012 m
L_u	Height of the upper cylinder chamber	0.01075 m
L_l	Height of the lower cylinder chamber	0.01075 m
T_w	Cylinder wall temperature	293 K
α_w	Convective heat transfer coefficient	500 Wm^{-2}K^{-1}

3. MEASUREMENT SYSTEM

The measurement system is schematically presented in Fig. 2. A forced-air-cooled electrodynamic shaker (LDS, V406), drives a piston to create harmonic pressure pulsations in the pneumatic cylinder, see Fig. 3. The amplitudes and frequencies of the pressure pulsations can be generated within the limits imposed by the maximum in the permanent-magnet shaker's sine peak force of 196 N, a shaker frequency range of 5 Hz to 9 kHz and a shaker's maximum peak-to-peak displacement of 17.6 mm. The initial piston location inside the pneumatic cylinder is measured with the Hall-effect position sensor (Festo, SMAT-8M, sensitivity 0.32 Vmm^{-1}, measuring range 21.5 mm, accuracy ±0.06 mm) inserted into the T-slot of the cylinder. The operation of the shaker is monitored with the aid of a signal from an IEPE accelerometer (Dytran, 3097A2T, sensitivity 9.975 mVm^{-1}s^2, measuring range –500 to +500 ms^{-2} peak, accuracy 5% in the frequency range 0.3 Hz to 10 kHz), which senses the motion of the piston-rod assembly. In the upper chamber of the pneumatic cylinder the ports for connecting the reference pressure sensor and the sensors under test are made. The pressure pulsations in the upper chamber of the pneumatic cylinder are measured with a piezoelectric transducer (Kistler, 7261, sensitivity 2193 pCbar^{-1}, cal. measuring range –1 to +1 bar, linearity ≤0.3% of full scale output, internal volume 1.5 cm^{-3}). The pressure transducer is connected to the cylinder port with a short both-side-threaded screw in order to eliminate the effects of connecting tubes on the magnitude of dynamic errors (see e.g., [8]). In order to measure the vibrations transmitted to the pressure transducer through the cylinder, an IEPE accelerometer (Dytran, 3097A2T, sensitivity 10.121 mVm^{-1}s^2, measuring range –500 to +500 ms^{-2} peak, accuracy 5% in the frequency range 0.3 Hz to 10 kHz) is fastened to it.

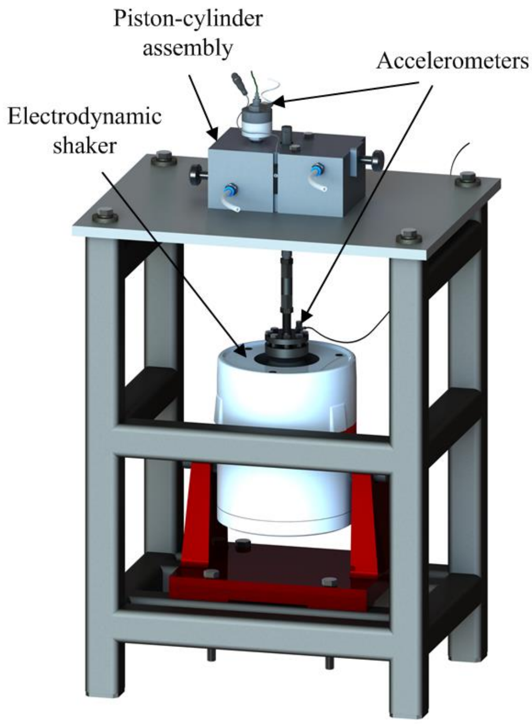

Fig. 2. Schematic view of the measurement system.

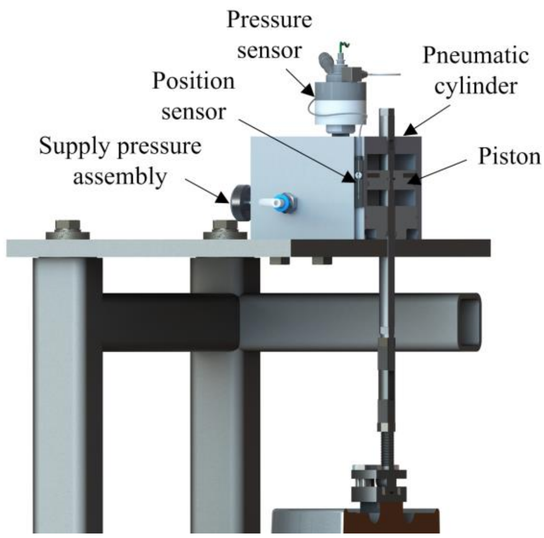

Fig. 3. Cross-sectional view of the piston-cylinder assembly.

In order to enable the dynamic calibration of pressure sensors at different average pressures and to minimize the initial preload of the shaker due to the pressure acting on the piston, the supply pressure ports are made in the upper and lower chamber of the pneumatic cylinder. The cylinder chambers above and below the piston are pressurized with the same initial static pressure using the gauge pressure controller (Druck, DPI 510, measuring range -100 to 1600 kPa, accuracy 0.1% of reading). In order to determine the absolute pressure, the barometric pressure was measured with the pressure transducer (ALMEMO, FD-A612-MA, measuring range 80 to 105 kPa abs, accuracy 300 Pa).

The signals from the accelerometers and the pressure transducer are amplified using an amplifier (Dewetron, DAQ-Charge, gain 1 to 1000, full scale output –5 to +5 V, accuracy 1% of full scale output, frequency range 0.3 Hz to 50 kHz (–3dB)) and charge amplifier (Dewetron, DAQ-Charge, sensitivity 0.01 to 0.05 V/pC, full scale output –5 to +5 V, accuracy 1% of full scale output, frequency range 0.07 Hz to 30 kHz (–3 dB)), respectively. The pressure, acceleration and position signals are connected to the data-acquisition (DAQ) board (National Instruments, NI USB-6251 BNC, resolution 16 bit, set sampling rate 100 kHz). The DAQ board also employs an analogue output to control the operation of the shaker. The control system's algorithms and the signal processing of all the measurement signals are realized in the LabVIEW programming environment.

4. RESULTS

Considering small pressure and volume changes and neglecting the heat transfer, the linear relation between the pressure change and the piston displacement can be written as [9]:

$$p_i'(t) = \frac{\gamma A_c p_0}{V_0} x'(t), \tag{6}$$

where p_0 is the initial absolute pressure, V_0 is the initial volume of the gas in each chamber, $p_i'(t)$ is the dynamic component of the pressure and $x'(t)$ is the dynamic component of the piston displacement. From (6) it is seen that the relative change in pressure $p_i'(t)/p_0$ is proportional to the relative change in volume $x'(t)/L_i$.

For small displacements the gas contained in the cylinder chambers can be modelled as one-dimensional spring. When piston is moved, due to the pressure change the internal gas acts on the piston as an added stiffness force. The effective spring constant of the piston-cylinder system is:

$$k_{ef} = k + k_g, \tag{7}$$

where k_g is the gas stiffness. The natural frequency of the piston-cylinder system can therefore be written as [10]:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_{ef}}{m}} = \frac{1}{2\pi}\sqrt{\frac{1}{m}\left(k + \frac{2\gamma A_c^2 p_0}{V_0}\right)}. \tag{8}$$

From (8) it is evident that due to higher gas stiffness the natural frequency of the moving piston in the cylinder with two pressurized gas chambers is expected to be higher as in the conventional configuration of the dynamic pressure generator with one pressurized chamber.

Fig. 4 shows the amplitude-frequency characteristics of the dynamic pressure generator obtained with the mathematical modelling described in Section 2. This theoretical study was carried out at three different initial absolute pressures (p_0 = 200, 400 and 600 kPa) and in the range of pulsation frequencies up to 500 Hz. The results of the simulations show that the natural frequency of the pressure generator increases from about 150 to 190 Hz by increasing the initial absolute pressure in the cylinder from 200 to 600 kPa due to higher stiffness of the gas in the cylinder at the higher average pressures.

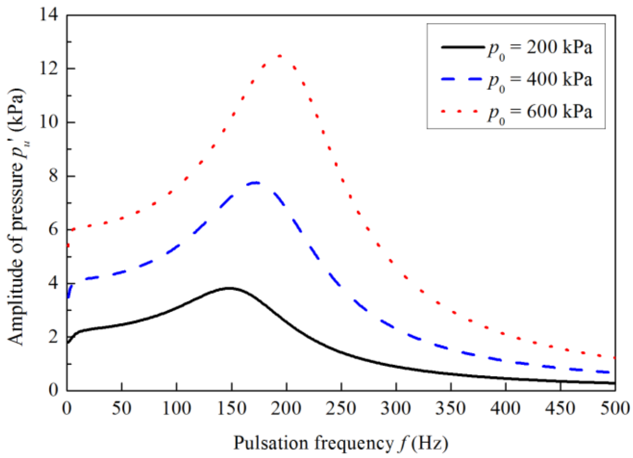

Fig. 4. Amplitude-frequency characteristics of the dynamic pressure generator obtained with the mathematical modelling.

Fig. 5 shows the ratio r between the amplitude of the generated relative change in pressure P_i'/p_0 to the amplitude of the relative change in volume X'/L_i :

$$r = \frac{P_i' L_i}{p_0 X'}. \tag{9}$$

From the results of simulations it is evident that the value of the ratio remains constant (r = 1.4) at higher pulsation frequencies, where the system can be considered as adiabatic. By decreasing the frequency of pressure pulsation, the ratio starts to decrease at certain frequency. This frequency is lower at higher average pressures in the cylinder. At the lowest pulsation frequencies, where due to the relatively slow volume changes over time with respect to heat transfer to the surroundings the system can be considered as isothermal, it is evident that the ratio approaches the value 1.

The experimental study of dynamic characteristics of the developed pressure generator was carried out by increasing the pressure pulsation frequency from 5 to 500 Hz by the step of 5 Hz at a constant harmonic voltage drive of the shaker, where the value of the voltage amplitude was set to the value that results in the generated relative pressure pulsation of approximately 1% at the frequency f = 5 Hz. Due to the fact that the piezoelectric measurement system does not measure the static pressure component and that the average gauge pressure meter used in the preliminary

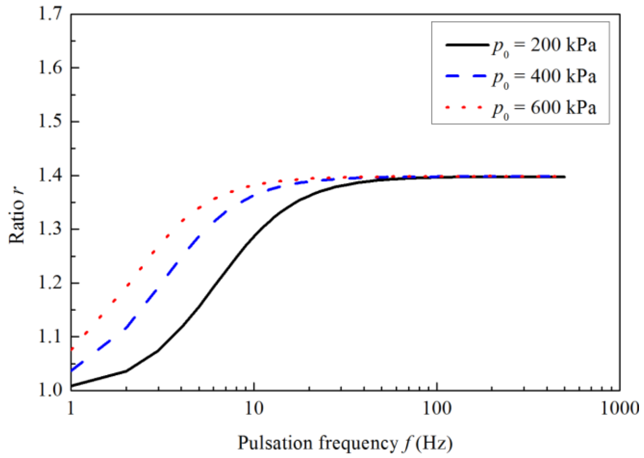

Fig. 5. Ratio between the generated relative pressure change to the relative volume change obtained with the mathematical modelling.

experiments showed large effects of the dynamics of its connecting tubing on the dynamic characteristics of the developed pressure generator, which is to be determined in this paper, we assumed that the average absolute pressure in the cylinder is equal to the sum of the initial static gauge pressure set by the pressure controller before each measurement and the barometric pressure. The assumption of the average pressure being constant during the measurements is based on the results of preliminary experimental analyses, which showed that the changes of the average pressure inside the dynamic pressure generator during the measurements of the pressure pulsations with the frequencies up to 500 Hz are within ±1% of the initial static pressure value.

The dynamic pressure generator enables the generation of pulsating pressure at different average pressures. Fig. 6 presents the time variations of the absolute pressure measured in the dynamic pressure generator for the pulsation frequency of 100 Hz at two different average pressures. The presented signals were constructed by adding the dynamic pressure component measured with the piezoelectric transducer during the pulsations to the average absolute pressure in the cylinder.

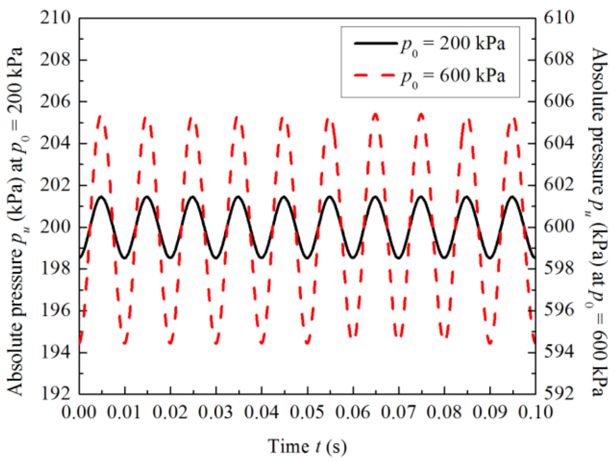

Fig. 6. Measured time variation of the pulsating absolute pressure (f = 100 Hz).

The results in Fig. 7 show the measured amplitude-frequency characteristics of the dynamic pressure generator at different initial pressures. The trend of the measured natural frequencies of the dynamic pressure generator is similar to the solution of the physical-mathematical model. The measured natural frequency increases from approximately 160 to 190 Hz by increasing the initial absolute pressure from 200 to 600 kPa. In comparison with the theoretical results, the measured amplitudes of the pressure pulsations are in general slightly lower. This results from the fact that the excitation force of the electromagnetic shaker is proportional to its electric current, whereas the measurements were performed at a constant electric voltage drive over the entire frequency range, which increases the damping of the actual system [11].

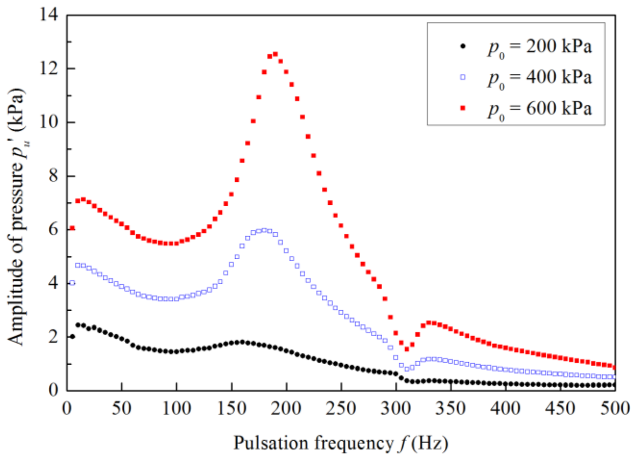

Fig. 7. Measured amplitude-frequency characteristics of the dynamic pressure generator.

From the presented characteristics in Fig. 7 it is evident that at the pulsation frequency around 300 Hz some deviations from the typical trend for the amplitude-frequency characteristics appear. The performed experimental study of this effect showed that it results from the external vibrations of the pressure generator and their influence on the piezoelectric transducer. Fig. 8 shows amplitudes of the accelerations measured on the pressure

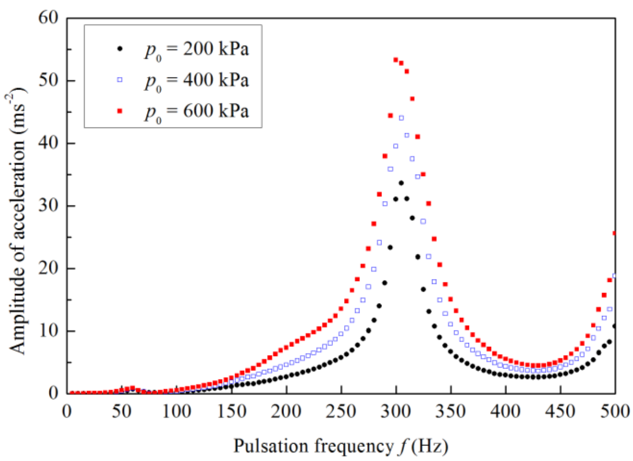

Fig. 8. Amplitude-frequency characteristics of the measured pressure transducer accelerations.

transducer. It is seen that the largest accelerations occur around the frequency of 300 Hz. At this frequency the largest deviation from the expected trend of the amplitude characteristic is about 0.5 kPa at $p_0 = 600$ kPa (see Fig. 7), which confirms that the pressure transducer has the acceleration sensitivity of approximately 0.01 kPam^{-1}s^2 as stated by its manufacturer.

5. CONCLUSIONS

The ability of the dynamic pressure calibrations of the pressure sensors at different average pressures is very important, because other researchers have indicated that the dynamic properties of the pressure sensors are also dependent on the value of the average pressure during the pressure pulsations [12]. The dynamic pressure generator under discussion, which is realized with two equally pressurized gas chambers, has some advantages over a conventional configuration with one pressurized chamber. Due to the fact that the developed pressure generator enables pressurization of both gas chambers of the pneumatic cylinder with the same initial pressure, it minimizes the initial preload of the shaker due to the pressure acting on the piston. As the result, the electromagnetic shaker enables the generation of larger displacements of the piston and therefore larger pressure pulsations with the same excitation force.

The results presented in this paper show that the developed dynamic pressure generator is applicable in the frequency range of hundreds of Hz. The mathematical modelling and experimental analyses presented in this paper offer an insight into the capabilities and limitations of the developed dynamic pressure generator. Its dynamic characteristics are defined by the properties of the piston, the both gas chambers and the connection of the pressure sensor to the pressure generator. The results show that the natural frequency of the pressure generator increases by increasing the initial pressure in the cylinder due to higher stiffness of the gas in the cylinder at the higher average pressures.

From the results of mathematical simulations it is evident that the heat transfer to the surroundings also influences the characteristics of the dynamic pressure generator. At higher pulsation frequencies the system can be considered as adiabatic and the ratio between the amplitude of the generated relative change in pressure to the amplitude of the relative change in volume of the cylinder chamber remains constant at value 1.4. At the lowest pulsation frequencies, where due to the relatively slow volume changes over time with respect to heat transfer to the surroundings the system can be considered as isothermal, the ratio approaches the value 1.

In the future work, we will improve the characteristics of the developed dynamic pressure generator by increasing the rigidity of its mounting in order to prevent the transmission of the external vibrations through the cylinder to the pressure sensors. Because the measurements of both dynamic and average pressure components are preferred to be measured without the influence of the dynamics of the pressure meter connecting tubing, the flush mounted reference pressure sensor, which will enable the measurements of both pressure components, will be applied.

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