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MEASUREMENTS OF UNSTEADY FLOW PARAMETERS IN PIPE-RECEIVER AND PIPE-TURBOCHARGER SYSTEMS

In the paper, the characteristic features of an unsteady flow in two systems: a high-volume pipe-receiver and a pipe-turbine are described. The tests were made under the conditions of a pulsating pipe supply in the range of frequencies 20-100 Hz. The following measurements of instantaneous values of unsteady flow parameters were made: pressure (piezoresistive transducers), temperature (Constant Current Thermometer) and velocity (Constant Temperature Anemometer). The applied measurement methods - owing to good dynamic properties of transducers - permitted us to gather relatively rich and complete information on flow phenomena in such systems. It was shown that the pipe-receiver system behaved as a resonator with a marked range of amplitude amplification and a distinct relationship between the resonance frequency and the pipe length. In this system, an effect of high pressure signal attenuation along the pipe and a reverse flow were observed as well. On the other hand, the pipe-turbine system is free of characteristic resonance frequencies and the pressure signal is transmitted along the pipe up to the turbine inlet section practically without any changes.

Keywords: pulsating flow, pressure, temperature and velocity measurements, resonance

1. INTRODUCTION

At present, practically all newly-designed diesel engines are equipped with turbochargers (popularly referred to as turbocompressors). They are very sophisticated in structural and technological aspects, but one of a few elements where a significant improvement of the turbocharger operation can be still achieved are the systems of pipes connecting it to an engine. It concerns mainly the turbine, which follows from the flow unsteadiness in the turbine supply channel (caused by the engine's cyclic operation). Depending on the engine actual operating point, various pulse frequencies (that depend on the engine rotational frequency) and their different amplitudes (that depend on the engine load) are met in the pipe. Consequently, these two factors exert an influence on the shape of pulsations (shape of the pressure wave in the turbine supply channel under unsteady flow conditions).

An especially wide variability of parameters that describe the pulsating flow is characteristic of automotive engines for which the operating range is broad and a change in the operating point frequent.

Therefore, the investigations which would allow one to identify the characteristic phenomena occurring in the turbine supply pipe, were considered significant.

2. TEST RIG CONFIGURATION

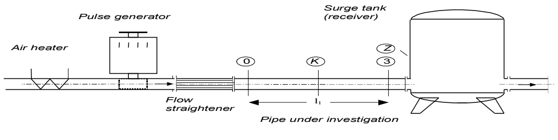
The basic condition that should be fulfilled by a test rig for testing the flow in the turbine supply channel is the possibility to generate a pulsating flow in the range corresponding to a typical traction diesel engine, while maintaining control of both pulse parameters (frequency and amplitude) and flow parameters (pressures, temperatures and mass flow rate). It was assumed in the first step that a straight pipe of circular section would be subjected to the investigations. A one-dimensional model of the pulsating flow (”x-t” model - [1]) used for the determination of instantaneous flow parameters at an arbitrary time instant and in an arbitrary pipe section was developed for such pipe geometry. That model was based on the fundamental equations of fluid mechanics (conservation equations of mass, momentum and energy), modelling respectively the friction losses on the pipe wall and the heat transfer through pipe walls. The equations of that model and its description, along with the formulation of boundary conditions can be found in [2].

As the model was initially used in calculations of pneumatic signal transmission in thin transmission pipes, closed with a pressure transducer at one end [3], it had to be adapted to new boundary conditions (non-zero velocity of the working medium at the beginning and the end of the pipe). It was decided to conduct an initial verification of the model for simpler boundary conditions, that is to say, to impose a condition of constant pressure at the pipe outlet. To achieve this, it was necessary to close the pipe by means of a high-volume receiver, whose task was to attenuate pulsations in the pipe outlet section (Fig. 1).

Thus, a test rig with two variants was built, namely: with a high-volume receiver (Fig. 1) and with a turbocharger turbine (Fig. 2), characterized by two different kinds of boundary conditions, i.e., constant and time-variable pressure at the pipe outlet.

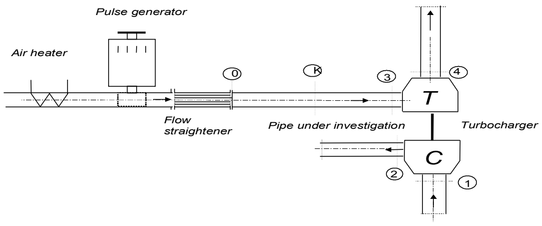
In both cases, the pipe is supplied in a pulsating way with a pulsation generator - a rotating valve [2]. A set of electric air heaters situated in the flow system enables one to control the working medium temperature and to make the turbocharger operation conditions close to the real ones.

In the system with a receiver, a tank of the volume equal to 0.14m3 operating in an open system (outflow from the tank to the atmosphere) was used. In the system with a turbine, a Garrett T2 turbocompressor was applied.



(0) – inlet section;
(3) – outlet section;
(K) – control section;
(Z) – surge tank (receiver).

Fig. 1. Pipe-receiver system test rig.



T – turbine;
C – compressor;
(0) – inlet to the pipe;
(1) – inlet to C;
(2) – outlet from C;
(3) – outlet from the pipe/inlet to T;
(4) – outlet from T.

Fig. 2. Pipe-turbocharger system test rig.

In both cases, a complete set of flow parameters (pressure, temperature, flow velocity) at the pipe inlet (section (0)) and its outlet (section (3)) was measured, whereas in section (K) situated at the midpoint of the pipe length, only the control pressure measurement was made. In the system with a receiver, pressure in the receiver (Z) was recorded as well.

Endevco 8530C piezoresistive pressure transducers, screwed into the side wall of the channel and recording a static pressure signal from it, were used in the pressure measurements. As far as measurements of instantaneous temperatures and velocities are concerned, two twin systems based on the technology of thin wires were applied. The sensors were made in a two-wire system, with two separate wires: the first one operates in the constant current system as a constant current thermometer (CCT), and the second one in the constant temperature system as a constant temperature anemometer (CTA).

The limit frequencies for the transducers applied exceed the value of several tens of kHz, (100 Hz in case of the CCT thermometer [4]) which allows for their application in the range of frequencies of interest (up to 100 Hz), without knowledge of their dynamic characteristics, as they are treated as inertialess sensors in this range of frequencies. On the other hand, their static calibration was necessary. The pressure transducers were calibrated by comparison of the signal from the transducer, recorded at the outlet of the acquisition card, with the value of pressure (coming from the same measurement section) measured with a water column manometer with a reading accuracy equal to 1mm H2O. The CCT was calibrated by comparison of its recordings to E-type miniature thermocouple (wire diameter ∅ 0.05 mm) recordings, whereas the CTA was calibrated using the mean flow mass rate measurement (made upstream of the pulse generator) with a multi-hole annubar probe [5].

3. SHORT CHARACTERISTICS OF THE MEASUREMENT RESULTS

Owing to very extensive results obtained, only the most important conclusions will be presented here. A more detailed description of the measurement results can be found in [6, 7].

3.1. Processing of the recorded signals

Signals from the transducers were recorded with a Keithley DAS 1801ST acquisition card, and then recalculated with static characteristics equations. Due to the periodic character of the signals, approximations of the characteristics were made with the Fourier trigonometric series expressed as follows:

$$f(t) = a_0 + a_1\cos(2\pi ft) + b_1\sin(2\pi ft) + \ldots + a_i\cos(i2\pi ft) + \\ + b_i\sin(i2\pi ft) + \ldots + a_n\cos(n2\pi ft) + b_n\sin(n2\pi ft) \quad , \tag{1}$$

where: a_0 - constant component (corresponding to the mean value of the signal), a_i, b_i - coefficients of the series for the *i*-th harmonics (i = 1, …, n), taking into account the values of samples included in the time range corresponding to the integer multiple of the pulsation period $T = 1/f$.

In further analysis, the amplitude of the *i*-th harmonics in the following form:

$$c_i = \sqrt{(a_i)^2 + (b_i)^2} \tag{2}$$

and the relative amplitude (related to the mean value):

$$c_{wi} = c_i / a_0 \ , \tag{3}$$

were used.

The number of harmonics necessary to reproduce correctly the acquired signal variation was different for various pulse frequencies and fell in the range 2 - 4. In Figure 3, an exemplary pressure diagram and its approximation with only the first harmonics (Fig. 3a) and four first harmonics (Fig. 3b) is shown.

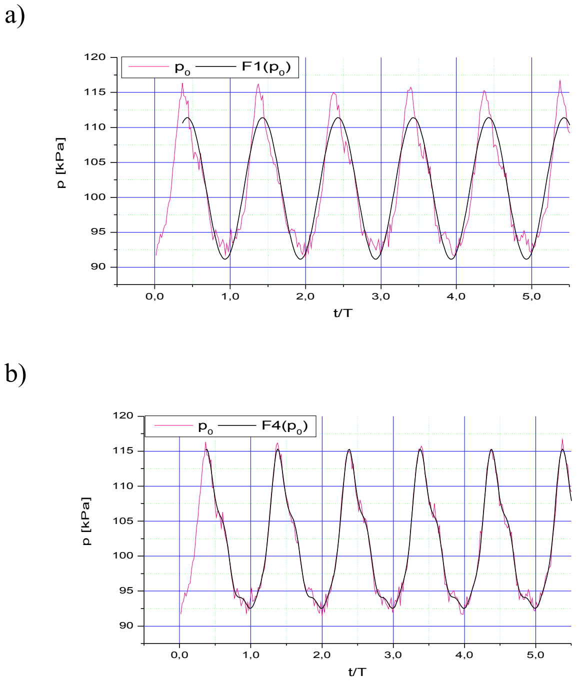


Fig. 3. Recorded signal of the pressure p0 and its approximation by means of the Fourier series with one (a) and four (b) first harmonics, respectively. The measurements were made for a pulse frequency of 40Hz.

3.2. Results of the measurements

The measurements were conducted for two different lengths of the pipe, namely: lR1 = 0.545 m (pipe R1) and lR2 = 1.247 m (pipe R2) and two various levels of the mean mass flow rate: m_1 = 0.02 kg/s and m_2 = 0.07 kg/s, measured at the test rig inlet with an annubar probe.

In Figure 4, exemplary diagrams of pressure recorded in sections (0),(K),(3) and (Z) (left column), and of temperature and specific mass flow rate[1] in sections (0) and (3) (mid and right column, correspondingly) for the system with a receiver are presented. The thicker lines that are an approximation of the plots by means of the respective number of harmonics of the Fourier series are imposed on the thin lines corresponding to the signals from transducers.

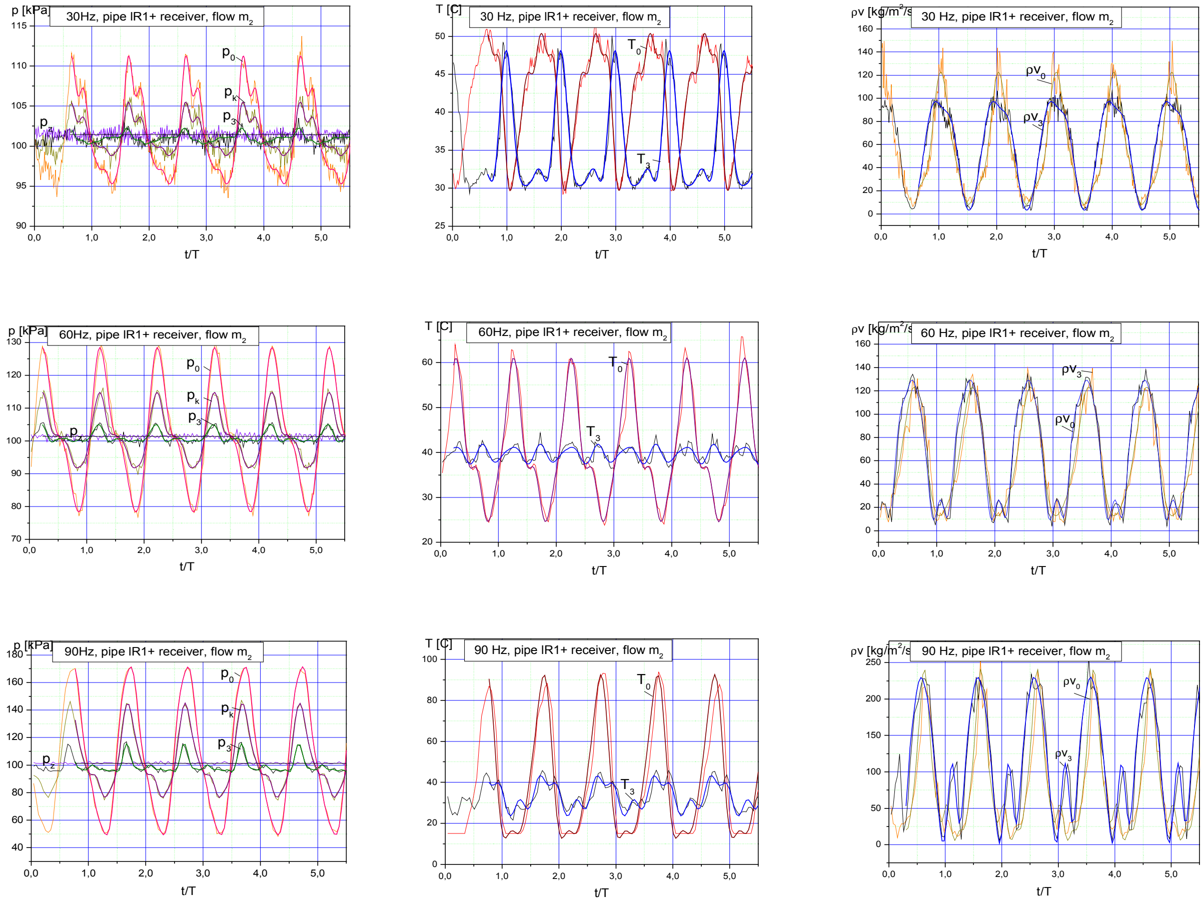


Fig. 4. Pressure (left column), temperature (mid column) and specific mass flow rate (right column) signals in pipe control sections recorded for different frequencies.

It can be stated on the basis of the analysis of pressure characteristics that the pulse amplitude is gradually attenuated along the pipe length until pulsations disappear completely in the receiver. At the same time, the shape of the pressure wave in subsequent sections does not change

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[1] Under real conditions, the specific mass flow rate $\rho v = \dot{m} / A$, which determines the instantaneous mass flow rate related to the pipe section area, and not only the velocity v itself, is the outlet signal from the anemometric probe.

practically. In the case of temperatures, pulsations are also attenuated between the pipe inlet and outlet, however the value of this attenuation depends to a larger extent on the pulse frequency. Also, the shape of characteristics for sections (0) and (3) differs much more significantly than for pressures. On the other hand, the plots of the specific mass flow rate do not exhibit an existence of its attenuation (pulse amplitude remains unaltered), whereas in the vicinity of resonance, a reverse flow from the receiver occurs, which can be seen on the diagrams in the form of a local increase in the specific mass flow rate[2] (cf. the diagram for ρv at 90 Hz - Fig. 4). A detailed analysis of this effect is shown in Fig. 8.

The analysis of pulse amplitudes as a function of their frequency is very interesting indeed.

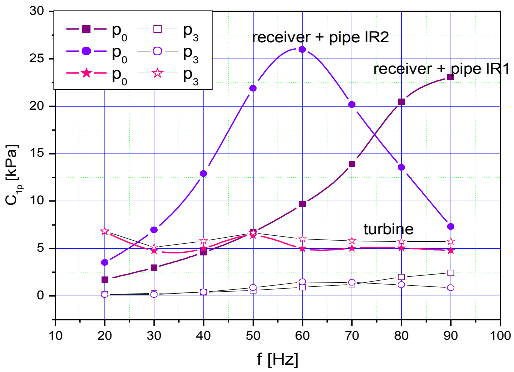
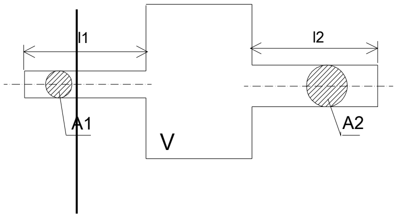


Fig. 5. Amplitudes of the first harmonics of pressure for the pipe-receiver system (pipes R1 and R2) and the pipe-turbine system.

In Figure 5, the relationship between the first pressure harmonics c1p in sections (0) and (3) and the pulse frequency f for the systems with a receiver (pipes R1 and R2) and for the system with a turbine has been shown. Both pipe-receiver systems are characterized by regions of a distinct increase in the pressure pulse amplitude. For the R1 pipe , this is a region in the vicinity of the frequency 90 Hz, for the R2 pipe - in the vicinity of 60 Hz. As an amplitude maximum was reached, these frequencies can be called the resonance ones. The increase in amplitude at resonance frequencies is very high - the amplitude increment is more than five times in comparison to the initial frequency of 20 Hz. The phenomenon of an increase in amplitude is accompanied by a significant increase in the noise emitted by the system. In the case of the system with a turbine, a resonance phenomenon does not occur. Amplitude oscillations are slight and they have a rather random nature.

a)



b)

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[2] The theromoenemometric probe does not differentiate the flow direction, therefore the reverse flow is recorded by it as a local increase in velocity. This phenomenon was analyzed in detail in [7].

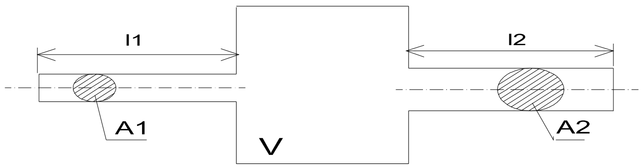


Fig. 6. Helmholtz resonator (a) and Lutz system (b) schemes.

It is interesting that the relationship between resonance frequencies and lengths of the pipes in the form:

$$\frac{f_{n1}}{f_{n2}} = \frac{\dot{f}_{n1}}{\dot{f}_{n2}} = \sqrt{\frac{l_{1R2}}{l_{1R1}}} = 1.5\,, \tag{4}$$

where: f_{n1}, f_{n2} - experimentally determined values of resonance frequencies for the pipes whose lengths are l_{1R1} and l_{1R}, correspondingly; $\dot{f}_{n1}, \dot{f}_{n2}$ - values of resonance frequencies determined for the system under analysis, modelled as a Helmholtz resonator [8] (Fig. 6a), is fulfilled.

The values $\dot{f}_{n1}, \dot{f}_{n2}$ are, as a matter of fact, divergent from the frequencies determined experimentally, however Eq. (4) is satisfied, which suggests that the relationship between the system natural frequency and the pipe length is of the following type:

$$f_n = k\sqrt{\frac{1}{l_1}}\,, \tag{5}$$

where k denotes a constant dependent on the system geometry and thermodynamic parameters that prevail in it.

On the other hand, the Lutz model provides a very good agreement with the experiments (Fig. 6b - [8, 9]). This model is more adequate for the pipe-turbine system under consideration, as it is - contrary to the Helmholtz resonator - an open system, including also an outlet pipe (taking off the working medium from the receiver). The resonance frequencies determined according to this model satisfy Eq. (6) - (notations as in Fig. 6b):

$$\frac{\omega_n}{a_0} = \frac{A_2}{V} ctg\left(\omega_n \frac{l_2}{a_0}\right) - \frac{A_1}{V} tg\left(\omega_n \frac{l_1}{a_0}\right), \tag{6}$$

where a_0 is the mean sound velocity in the system.

If we rewrite Eq. (6) as:

$$L = R_1 - R_2 = R \tag{7}$$

posing: $L = \frac{\omega_n}{a_0}$; $R_1 = \frac{A_2}{V} ctg\left(\omega_n \frac{l_2}{a_0}\right)$ and $R_2 = \frac{A_1}{V} tg\left(\omega_n \frac{l_1}{a_0}\right)$ the graphical solution can be applied - as shown in Fig.7.

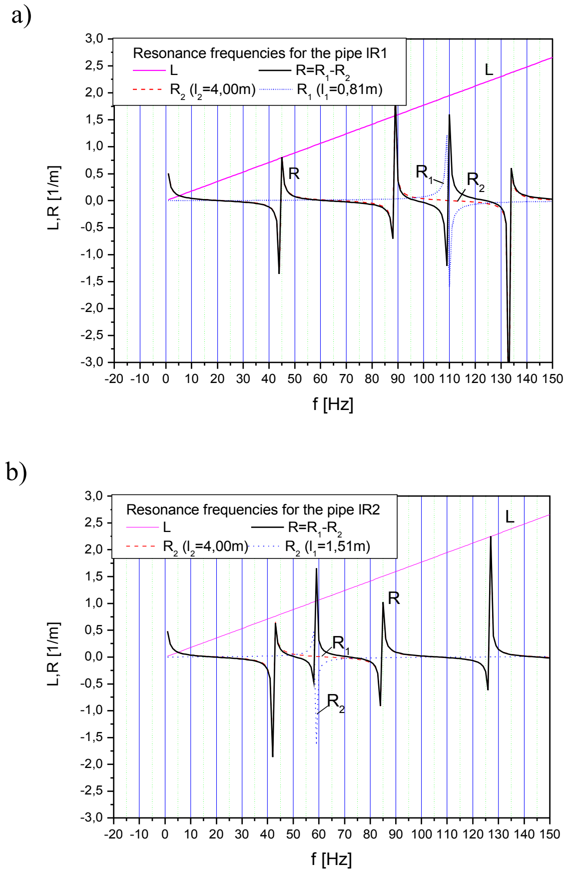
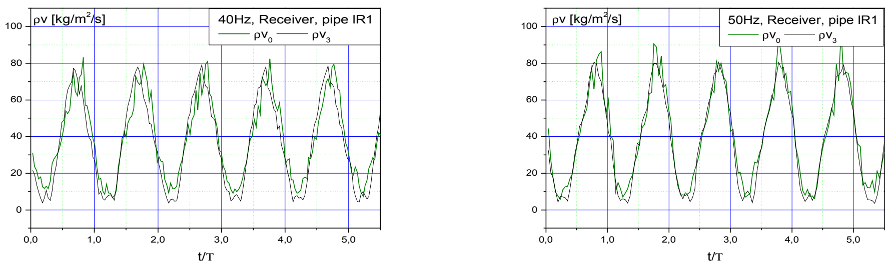


Fig. 7. Graphical determination of resonance frequencies in the pipes lR1 (Fig. 7a) and lR2 (Fig. 7b) applying the Lutz model. The intersection of curves L and R indicates the resonance frequency.

The accuracy of the Lutz model is very good indeed. Resonance frequencies determined by this method (Fig. 7 a and b) coincide exactly with those determined experimentally.

Another interesting phenomenon identified in the pipe-receiver system is the reverse flow occurring near resonance (this effect was mentioned while describing Fig. 4).

Figure 8 shows how the specific mass flow rate signal varies while approaching the resonance frequency (90 Hz in this case)



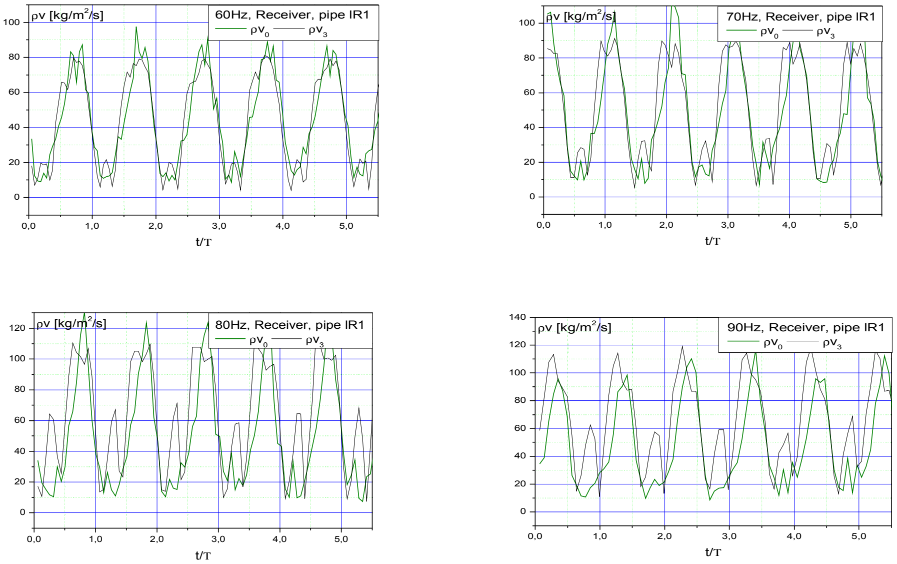


Fig. 8. The change of variations of specific mass flow rate ρv in the inlet (0) and outlet (3) control section while approaching resonance (achieved at 90 Hz).

When we are far from resonance, both mass flow rate signals (ρv_0 and ρv_3) are similar - no attenuation (observed for pressure signals) can be observed (diagrams for 40 and 50 Hz - Fig. 8). Starting from 60 Hz a slight change of the ρv_3 signal takes place - another local maximum occurs in the range of the lowest flows, and its value systematically increases while approaching 90 Hz. This effect is particularly strong for the outlet section, as it is closer to the receiver which seems to be cyclically loaded and discharged. The influence of the reverse flow can be also seen (but on a much smaller scale) on the ρv_0 diagrams.

Similar effects were observed for lR2 pipe in the neighbourhood of 60 Hz.

4. CONCLUSIONS

In the paper, the results of measurements of pulsating flow parameters in the turbocharger systems: a pipe-receiver and a pipe-turbine have been presented. The applied transducers and the methods of acquisition and processing of the signals that correspond to instantaneous values of flow parameters have been described. It has been shown that a system with a receiver behaves like a resonator, with a distinctly defined resonance frequency depending on the pipe length. The increase in the pulse amplitude in the resonance region is very considerable (in the case of pressure, it exceeds five times the reference amplitude values) and refers to all the flow parameters recorded. As far as pressure is concerned, we also observe a significant attenuation of pulsations along the pipe length, until they are attenuated completely in the receiver. In the case of velocity (or more precisely: specific mass flow rate), such attenuation does not occur - the amplitude remains constant, however a reverse flow, which is an effect of periodic unloading of the receiver along the backward direction, appears.

In the pipe-turbine system, the scope of phenomena is much less complex: resonance phenomena do not occur, pressure signal attenuation does not take place either - the signal is transmitted along the pipe without any changes practically.

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POMIARY NIESTACJONARNYCH PARAMETRÓW PRZEPŁYWU W UKŁADACH PRZEWÓD-ZBIORNIK ORAZ PRZEWÓD-TURBINA ZESPOŁU ŁADUJĄCEGO

Streszczenie

W artykule opisano zjawiska przepływowe zaobserwowane w układach przewód-zbiornik oraz przewód turbina zespołu ładującego przy pulsacyjnym charakterze zasilania tych układów. Wykazano, iż układ ze zbiornikiem zachowuje się jak rezonator, z wyraźnym obszarem wzmocnienia amplitud wszystkich badanych parametrów przepływu (ciśnienia, temperatury i prędkości). Przy stałej objętości zbiornika, położenie tego obszaru zależy od długości przewodu, co wskazuje na podobieństwo do rezonatora Helmholtza.

Jednak przyjęcie modelu układu przepływowego w tej postaci nie daje zbieżności wartości częstości rezonansowych wyznaczanych z modelu oraz z pomiaru. Zbieżność tę, daje dopiero zastosowanie jako modelu układu Lutza, uwzględniającego odcinek przewodu za zbiornikiem.

Charakterystyczną cechą układów ze zbiornikiem jest tłumienie sygnału ciśnienia wzdłuż długości przewodu, z kolei w układzie z turbiną, takie tłumienie jest pomijalne, sygnał w zasadzie bez zniekształceń jest przenoszony wzdłuż długości przewodu do przekroju wlotowego turbiny. W układzie tym nie występują również efekty rezonansowe.

Analiza opisanych zjawisk przepływowych była możliwa dzięki zastosowaniu bardzo szybkich przetworników mierzonych parametrów przepływu.