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EXPERIMENTAL EXAMINATIONS OF SYNTHETIC FLOW IMPACT ON AXISYMMETRIC NOZZLE OUTPUT JET

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Summary: *The impact of synthetic jet on the properties of the main jet flowing from an axisymmetric nozzle has been investigated. The examinations were preceded by a theoretical analysis of the resonance chamber and then followed by the investigation of the synthetic jet with an anemometric probe. The aim was to analyze the main jet properties in order to increase its efficiency of heat/mass transfer from the output jet onto the wall exposed to the jet.*

1. Introduction

The aim of the present research was a preliminary investigation whether it is possible to generate a jet which would control axisymmetric output jet. The output jet is generated in the axisymmetric nozzle with an inner core. The synthetic jet is generated by loudspeaker membrane (Fig.1). It is assumed that identifying the resonance frequency would allow for controlling the main output jet without an additional control flow which has been used so far.

The examinations were preceded by a theoretical analysis of the resonance chamber. At that stage there was determined a theoretical chamber resonance frequency. The essential experimental investigations followed the experimental analysis of the impact of the control signal shape on generating the synthetic flow from the resonance chamber. The following signals were tested: sinusoidal, rectangular, trapezoidal, and triangular. The rectangular signal (of input voltage) was selected for further research.

The second stage involved a experimental verification of the calculations. For the membrane vibration frequency range from 100 to 630 Hz with the step of 10 the airflow velocity measurements were made on central core outlet (Fig.3). The membrane control signal was made up by the 3.7 V rectangular signal. The signal was generated by MULTIFUNCTION GENERATOR KZ1406. The signal was amplified with PZL-1 laboratory set. The image of the signal sent to the loudspeaker membrane was observed on MP-6II display.

Both the quantitative (airflow velocity) and qualitative (flow character) analyses of the results were made. According to the authors, it is optimal to generate a flow of high velocity which

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results from the resonance frequency obtained. All that makes an extensive range of experiments justifiable.

Theoretical analysis

Synthetic jets are generated by pushing and pulling of fluid from an actuator. From point of view of energy transformation, potential energy of the pair of diaphragms, U, is transformed into the kinetic energy of the fluid in the orifice, K, during each period (the kinetic energy of the fluid in relatively large a cavity is negligible against the kinetic energy in the annular gap). The total energy of fluid in the actuator, E, is the sum:

$$E = K + U \tag{1}$$

As already mentioned, the experimental research used the rectangular signal defined with equation (2) and given in Fig 1.

$$y = \sum_{n=1}^{\infty} \left(A\left(t - 2n\frac{T}{2} \right) - A\left(t - (2n+1)\frac{T}{2} \right) \right) \tag{2}$$

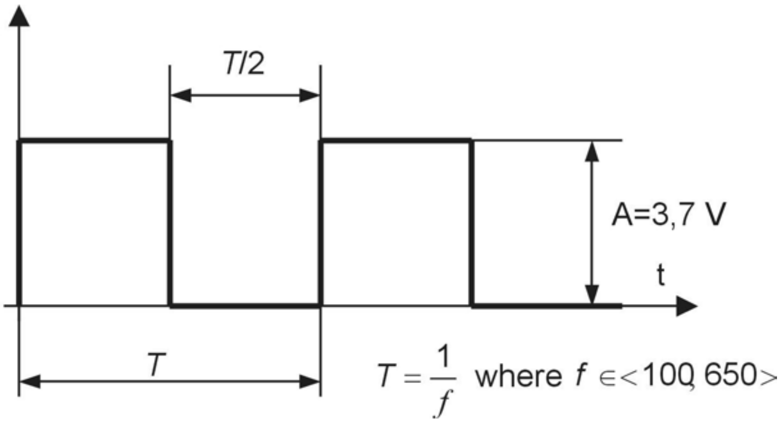

Fig. 1. Membrane control signal waveform diagram

Since equation (2) is rather complicated for analytical calculations a simplified equation was used, see Fig. 2, where $S_{\sin} = S_{pul}$ are given in equation (3).

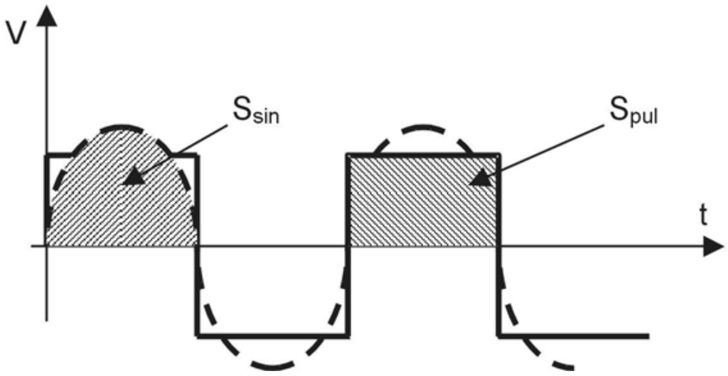

Fig. 2. Replacing the rectangular signal with sinusoidal signal

$$x_D = x_{D\max} \sin(\omega t) \tag{3}$$

where $x_{D\max}$ is the maximum displacement of diaphragm from neutral position, and $\omega = 2\pi f$.

Because the velocity of diaphragm movement is,

$$dx_D / dt = \omega x_{D\max} \cos(\omega t) \tag{4}$$

the velocity in actuator annular gap of diameter D can be described from continuity equation as

$$u = (A_D / A)\omega x_{D\max} \cos(\omega t) \tag{5}$$

the cross-section areas $A_D = \pi D_D{}^2 / 4$ and $A = \pi Da$ are considered for diaphragms and the annular gap, respectively. Let us assume that the kinetic energy of the fluid can be evaluated as

$$K = \rho L_e A u^2 / 2 \tag{6}$$

where ρ is density. The L_e is 'equivalent fluid column length' in annular gap, which is commonly assumed as

$$L_e = L + 8D/(3\pi) \tag{7}$$

where L is geometric length of the annular gap.

Eqs. (5, 6) give the kinetic energy as

$$K = \rho L_e A[(A_D / A)\omega x_{D\max} \cos(\omega t)]^2 / 2 \tag{8}$$

The potential energy of the diaphragm can be written as for linear springs

$$U = PA_D x_D / 2 = K_P A_D x_D 2, \tag{9}$$

where P is the static pressure, which causes diaphragm displacement x_D, and K_P is the diaphragm constant, which is defined as

$$K_P = P / x_D \tag{10}$$

The resonance frequency is determined from the maximums of the kinetic and potential energy in the top/bottom dead centers of diaphragms. If the kinetic energy of the fluid in the annular gap is zero, then the potential energy is maximum, $U_{\max}$. On the other hand, in a neutral position of diaphragms, the kinetic energy of the fluid in the annular gap is maximum, $K_{\max}$, and then the potential energy is zero

$$E = K_{\max} = U_{\max} \tag{11}$$

The maximums of kinetic and potential energy are from Eqs. (8, 9)

$$K_{\max} = \rho L_e A[(A_D / A)\omega x_{D\max}]^2 / 2 \tag{12}$$

$$U_{\max} = K_P A_D x_{D\max} 2 \tag{13}$$

Taking into account that $\omega = 2\pi f$, the natural frequency can be derived from Eqs. (12, 13) as

$$f = \left(\frac{1}{2\pi}\right)\left(\frac{D}{D_D}\right)\sqrt{\frac{K_P}{\rho L_e}} \tag{14}$$

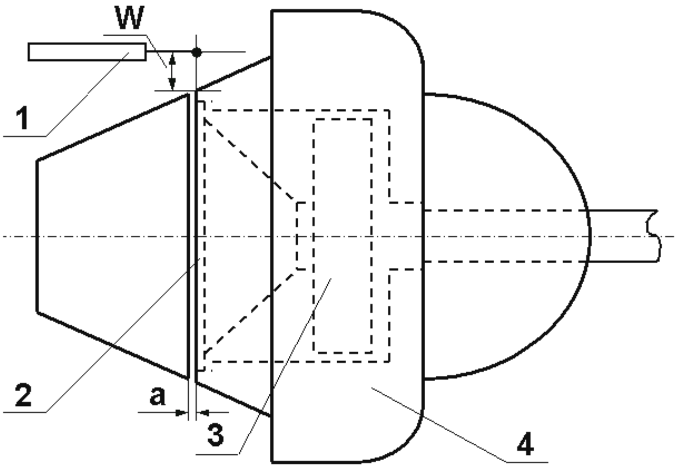

Fig.3. Diagram of inner core with loudspeaker: 1 – measurement probe, 2 – resonance chamber, 3 – loudspeaker membrane, 4 – inner core of the nozzle, w=3mm – distance between the probe and the inner cone, a=0.5mm – resonance chamber gap width

Having substituted the data in equation (14), the resonance frequency $f = 230\,\mathrm{Hz}$ was obtained.

Measurements results

The velocity was measured with the thermal anemometer, developed by Dantec, with a single measurement channel. The probe sampling frequency was: $f_s = 1000\,\mathrm{Hz}$, and the number of samples $n_s = 4096$. The measurement of velocity for each frequency consisted of three series separated from one another with a five-second interval.

The selected velocity results are given below. No satisfactory course of synthetic jet velocity of the resonance chamber outlet was obtained. The highest mean velocity was recorded for the membrane input signal of the frequency $f = 610\ Hz$.

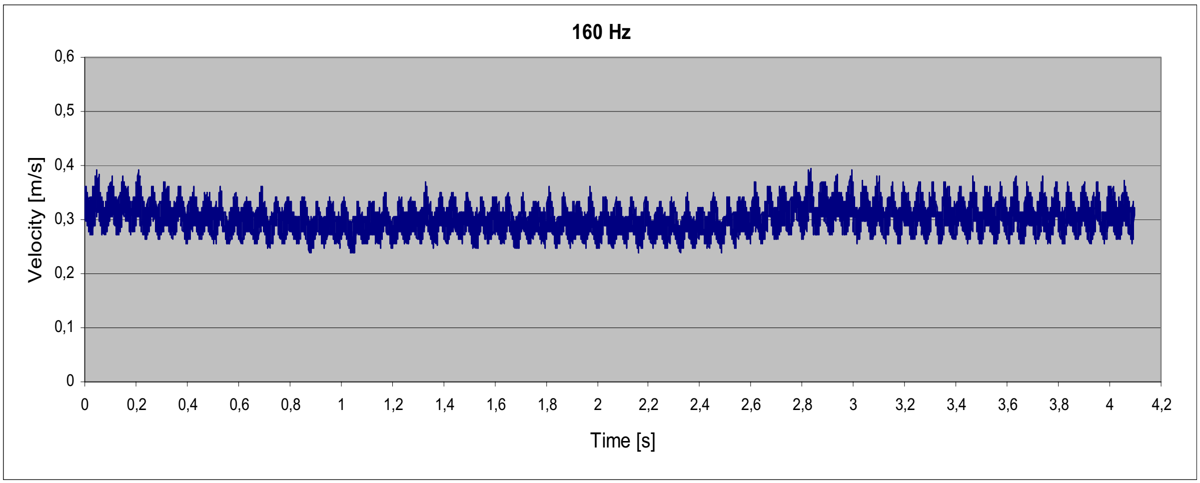

Fig. 4. Registered velocity of synthetic flow for the frequency of 160Hz

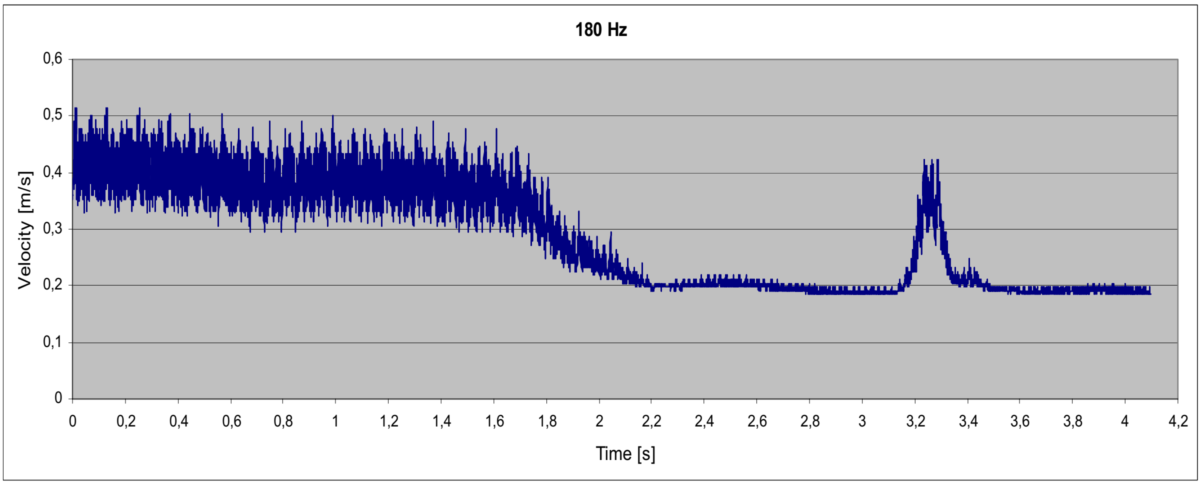

Fig. 5. Registered velocity of synthetic flow for the frequency of 180Hz

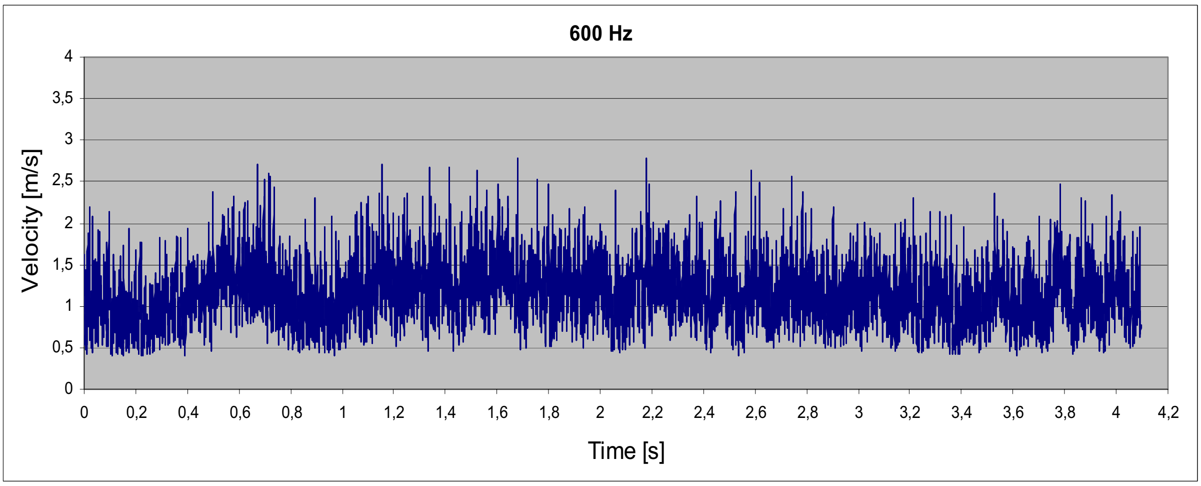

Fig. 6. Registered velocity of synthetic flow for the frequency of 600Hz

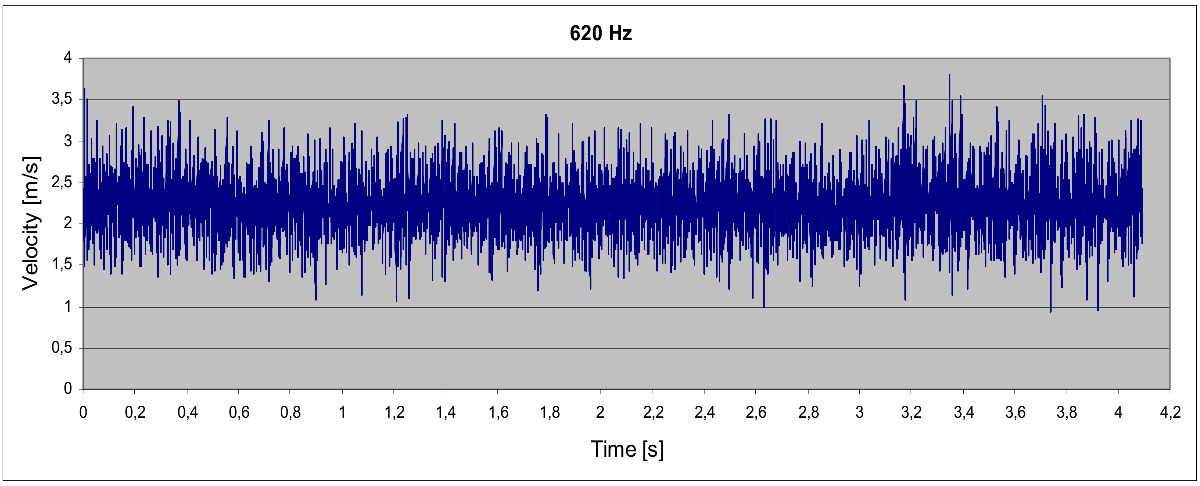

Fig. 7. Registered velocity of synthetic flow for the frequency of 620Hz

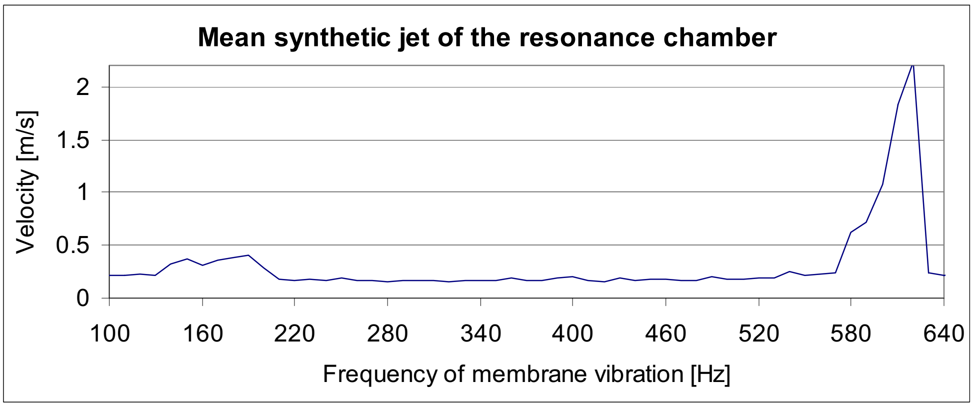

Fig. 8. Mean velocity depending on the membrane vibration frequency

Conclusions

Fig. 8, presenting mean velocity depending on the membrane vibration frequency, shows that the resonance frequency is $f = 610\,\mathrm{Hz}$. The value differs considerably from the theoretically determined value of $f = 230\,\mathrm{Hz}$, which requires further research. The difference is too big to be a result of the simplifications made during calculations. Therefore the results obtained can be only a qualitative confirmation of the occurrence of resonance.

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