COMPUTING OF THE LEVAVASSEUR TYPE AIR-JET FLAT GENERATOR WITH TWO RESONATORS

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Abstract. The paper proposes a method of dimensional and parametric calculation of the air-jet Levavasseur flat generator with the two resonators. Acoustic research results of the generator with two cylindrical resonators designed after the developed method showed the two working frequencies: the one is from the sound field and other - from the ultrasound field. Comparison of the working frequencies obtained by calculation and the ones from acoustic measurements showed a good frequency coincidence.

Key words: calculation, air-jet generator, flat, Levavasseur, resonator, frequency.

1. INTRODUCTION

The existing sound and ultrasound air-jet generators, considering the manner of sound field producing, can be classified in [1]:

– Hartmann generators, which use non-stationary processes in supersonic gas jets on its interaction with solid obstacles and cavity;

– Helmholtz generators (or "aerodynamic whistles"), which use non-stationary processes on the interaction with subsonic jet with a sharp solid edge;

– vortex generators, which use the instability phenomenon of the rotational gas flow.

Levavasseur generator, which is composed of two tore cavities – one of the generator and other of the resonator [2, 3], after the means of operation can be brought both to "aerodynamic whistles", and to the vortex generators, to which the flow instability is due to swirl systems developed within the jet. The resonance cavity is in opposite phases with the generator cavity bringing the generator operational efficiencies to increase.

There are different operating regimes of the generator with two resonators:

– operation at low supply pressure (below 0.14 MPa) and low gas speed – the usual behavior of the Levavasseur generator at which the sound is pure, free of harmonic frequency noise;

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– operating at high supply pressure, but at subsonic speeds of the jet, where in addition to fundamental frequency and higher harmonics are produced, which is the favorable situation for industrial applications;

– operating at high supply pressure and at jet's supersonic speeds, when there are high frequencies generated by the swirl produced in the input and downstream of the cavity resonators edge.

This paper is an attempt to dress these complex processes in mathematical form and develop a method for computing design generators of this type.

2. TEORETICAL BACKGROUND AND CONSTITUENT EQUATIONS

We analyze the dynamics of one-dimensional gas flow in a flat jet *Levavasseur* generator (Fig. 1).

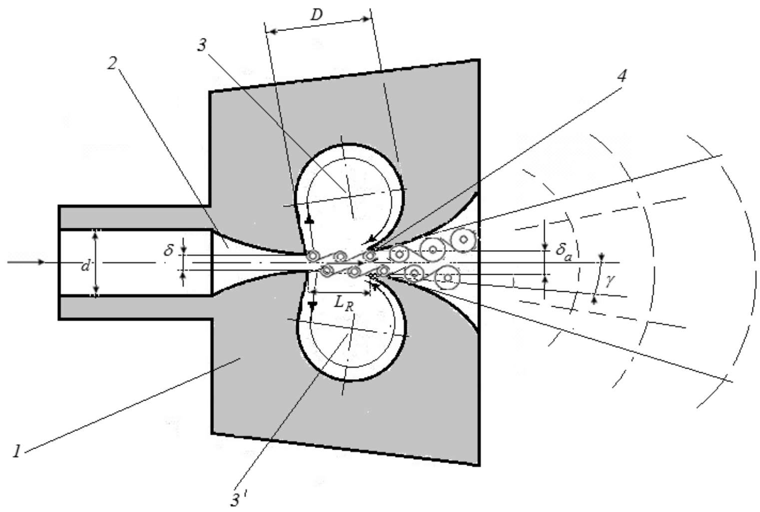

Fig. 1 – Gas dynamic flow scheme into the air-jet generator with two resonators: 1 – base, 2 – convergent nozzle, 3, 3' – resonance cavity, 4 – edge resonator, D – diameter of the resonance cavity; δ – convergent nozzle slot; δ_a – generator slot; L_R – length of the resonator throat; d – input channel diameter.

The compressed air is allowed in the central channel and is accelerated in the convergent nozzle formed of two resonators. The air flow reaching the resonators input is divided into three jets: a central jet and two lateral jets. The lateral jets undergoing cylindrical cavities contour and intersects the central outlet jet diverting it. Thus occurs regularly air remove from the resonator cavities, producing pressure pulses with frequency determined by the attachement rate of the central jet on the outer edge of the resonator. For its part, the rate of the attachment jet on edges is determined by the number of vortex produce by peripheral parts of the central jet. Because the cavities diameters are equal, the interference of oscillations which ensure the increase of amplitude occures, which allows generating powerful waves with high frequency including the ultrasounds.

A basic criterion that describes the operation of air-jet sonic generators is *Strouhal* number:

$$\mathrm{Sh} = \frac{f \cdot \delta}{\upsilon}, \tag{1}$$

where: f [Hz] is the working frequency; δ [m] – slot nozzle; υ [m/s] – gas jet velocity.

The maximum enhance of the oscillating process at the acoustic disruption of the jet gas flat flow is held within the field Sh = 0.2 to 0.3 [4]. The value near to optimal *Strouhal* number Sh = 0.34 is recommended in the paper [5].

The gas flow velocity depending on the regim flow of gas in the convergent nozzle outlet section, being that subsonic or critical.

In the subsonic regime the gas flow nozzle velocity is determined by applying the *Saint-Venant* formula [1]:

$$\upsilon = \sqrt{\frac{2k}{k-1} \cdot \frac{P_0}{\rho_0}\left[1-\left(\frac{P_{at}}{P_0}\right)^{\frac{k-1}{k}}\right]} \ \text{[m/s]}, \tag{2}$$

where: $P_0 = p + P_{at}$, Pa is the supply absolute air pressure; p – supply manometer air pressure; P_{at} – environmental pressure; $\rho_0 = \dfrac{P_0}{R \cdot T_0}$ = 3.57 kg/m^3 – reservoir air density; R = 287 J/(kg·K) – gas constant for air; T_0 = 273.12+ t_0, K – reservoir air temperature; t_o [°C] – air temperature in the reservoir; k = 1,4 – adiabatic exponent (air).

If the total gas pressure P_0 is greater at $\left(\dfrac{2}{k+1}\right)^{\frac{k}{k-1}}$ = 1.72 times than the environmental pressure P_{an} in which the flow take place, in the convergent nozzle exit section is installed the critical regime.

Critical velocity is given by relation:

$$\upsilon = a_* = \sqrt{\frac{P_0}{\rho_0} \cdot \frac{2k}{k+1}} \ \text{[m/s]}. \tag{3}$$

Convergent nozzle slot is obtained from the necessary value f of frequency, number Sh and calculated gas flow velocity υ:

$$\delta = \frac{\mathrm{Sh} \cdot \upsilon}{f}. \tag{4}$$

Slot nozzle width parameter h is not decisive. Also recommended for flat jets ratio:

$$\delta / h > 6. \tag{5}$$

Then the convergent nozzle cross section is:

$$S = \delta \cdot h, \tag{6}$$

where: δ [m] is the slot nozzle; h [m] – the width of the slot nozzle.

Air mass flow rate through the orifices and nozzles using the *Saint-Venant* for subsonic gas flow will be:

$$\dot{m} = \rho_0 \cdot \upsilon \cdot S \ [\text{ kg/s}], \tag{7}$$

where S [m^2] is the area of the convergent nozzle exit section (Fig. 1).

Air mass flow rate for regime flow with the thermodynamic critical parameters are determined by the formula:

$$\dot{m} = \rho_* \cdot a_* \cdot S, \tag{8}$$

here ρ_*, a_* are the critical parameters determined from the initial parameters P_0, ρ_0 and adiabatic exponent k of air. Air critical density is:

$$\rho_* = \rho_0 \left(\frac{2}{k+1} \right)^{\frac{1}{k-1}} \ [\text{ kg/m}^3]. \tag{9}$$

The vortex from the resonance cavities 3 and 3' transfer the pressure perturbations in the long-distance L_R at the time period equal to L_R / a, where a is the sound speed.

Analyzing the dynamics of the jet flow, the long-distance L_R can be determined from [6] with the equation:

$$L_R = n \cdot l_v - \frac{L_R}{a} f \cdot l_v, \tag{10}$$

where: n is an integer number ($n = 1$ corresponds to symmetric perturbation systems, $n = 2$ – for a asymmetric perturbation systems – Fig. 1), and l_v is a distance between mini vortex from perturbation systems (it is considered that $f\ l_v = \upsilon$).

The frequency f can be expressed by the *Strouhal* number Sh:

$$f = \frac{\mathrm{Sh} \cdot \upsilon}{\delta}. \tag{11}$$

Equation (10) may also be presented as:

$$\frac{L_R}{\delta} = \frac{n}{\left(1+\mathrm{M}\right)\cdot \mathrm{Sh}}, \tag{12}$$

where: $\mathrm{M} = \upsilon / a$ is the *Mach* number; $a = 331{+}0.59\cdot(t - t_0)$ [m/s] – the speed of sound in air; at the temperature t [°C]; $t_0 = 0$ °C.

The natural frequency of *Helmholtz* resonator type is determined by the relationship [8]:

$$f = \frac{a}{2\pi}\cdot\sqrt{\frac{S_R}{b\cdot V_R}}, \tag{13}$$

where: S_R [m^2] is the opening cross section area of resonance cavity; A [m/s] – the speed of sound; b [m] – opening cross section depth; V_R [m^3] – resonance cavity volume.

The volume or cylindrical resonance cavity with diameter D and the depth h (which constructive is equal convergent nozzle width) can be determined by the formula:

$$V_R = \frac{\pi\cdot D^2}{4}\cdot h\,. \tag{14}$$

Opening of rectangular cross-section resonance cavity will be:

$$S_R = L_R\, h\,, \tag{15}$$

where L_R is the length and h the width of opening.

From resonators geometry (Fig. 1) results the relationship for depth of the opening resonance cavity:

$$b = \frac{\delta_a - \delta}{2} = L_R\cdot \mathrm{tg}\gamma\,, \tag{16}$$

where $\mathrm{tg}\,\gamma = \mathrm{tg}\,(10\text{–}15)^0$.

From equation (13) and relationships (14–16) is obtained the resonance chambers diameter:

$$D = \frac{a}{f\cdot(\pi^2\cdot \mathrm{tg}\gamma)^{1/2}}\,, \tag{17}$$

where: $a = 331{+}0.59\cdot(t - t_0\,)$ [m/s] is the speed of sound in air at the; temperature t [°C]; $t_0 = 0$ °C; f [Hz] – generator frequency; γ [degree] – half cone angle generator output channel (Fig. 1).

Input channel diameter d is determined from the continuity equation of air mass flow after condition:

$$d \geq 2 \cdot \sqrt{\frac{\delta \cdot h}{\pi} \cdot 1.6} \tag{18}$$

Initial dates needed to calculate the sizing of the air-jet flat generator with two resonators are: f [kHz] – frequency of the generator; p [MPa] – working air manometer pressure of generator; P_{at} [MPa] – the ambient pressure at which the generator works; $\dot{m}$ [kg/s] – mass flow or volume air flow Q [m^3/h].

3. AIR-JET FLAT GENERATOR WITH TWO RESONATORS AND ACOUSTICAL PARAMETERS

Based on the proposed computing method, it was designed and constructed the two resonators flat air-jet generator as presented in Fig. 2. Geometric and functional parameters of the generator were determined by formulas (1)-(18), working pressure (the manometer) p = 0.05 MPa, the air flow Q = 10 m^3/h. For the calculation it was considered the generator f = 10.5 kHz. The results are given in Table 1.

Table 1

The air-jet two resonators generator's acoustic and size computing parameters

P Pa	Q m^3/h	P_{at} Pa	f kHz	Sh	δ mm	D mm	H mm	n	L_R mm	d mm
$1.6 \cdot 10^5$	10.0	$1.1 \cdot 10^5$	10.5	0.3	1,2	8,0	8.06	1	4.08	7.06

The data in Table 1 correspond to the case when in the generator is symmetric vortex system (n = 1). If asymmetric vortex (n = 2) is produced and for the dimensional parameters from Table 1 calculated frequency the increase and appear the second frequency of f = 21.2 kHz.

Generator have a flat form housing 3 with a rectangular channel within which are two flat resonators 1 and 2 of 8 mm thickness. Each resonator has one resonant cylindrical cavity with inside diameter of 8 mm. The upper resonator 2 has a rounded bottom and a rectangular opening of 4 mm to the generator axis symmetry. The down resonator 1 has a rounded top with a rectangular opening up. Their replacement with the rounded side face to face creating a central channel for passing the working fluid (compressed air).

Air-jet generator acoustic parameters were determined at the Noise and Vibration Laboratory of the Polytechnic University of Bucharest. Acoustic intensity levels were measured and recorded the acoustic signal in real time. FFT spectral analysis of these signals in the frequency range 0–25 kHz with a program LABwiev [7] was performed. Spectrograms obtained for each measurement were presented with ZOOM method for determinate the resonant spectral values accurately.

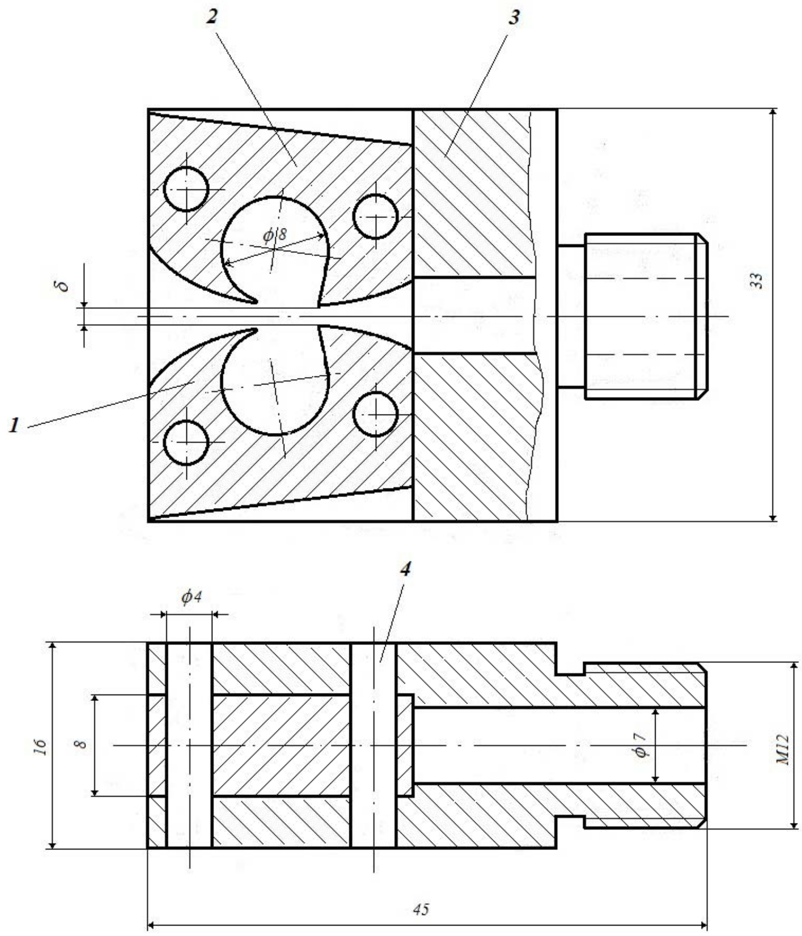

Fig. 2 – The axial air-jet flat generator with two resonators: 1 – housing; 2 – upper resonator; 3 – down resonator; 4 – pin fixation; δ – generator slot

They determined the acoustic parameters of the generator at supply pressure (manometric) of 0.05 MPa, at which the generator was calculated. Here in Fig. 3 the measured acoustic pressure pulses in real time, and in Fig. 4 is date FFT spectral analysis of this signal LABwiev program.

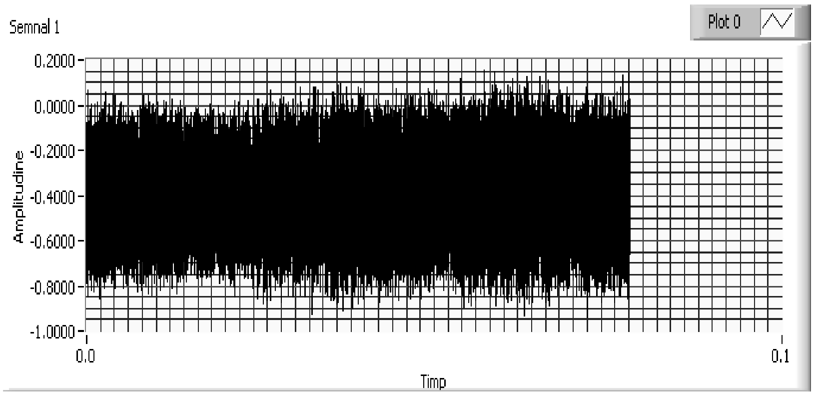

Fig. 3 – Sound pressure pulsations produced by the two resonators air-jet generator in real time (the generator working pressure p = 0.05 MPa).

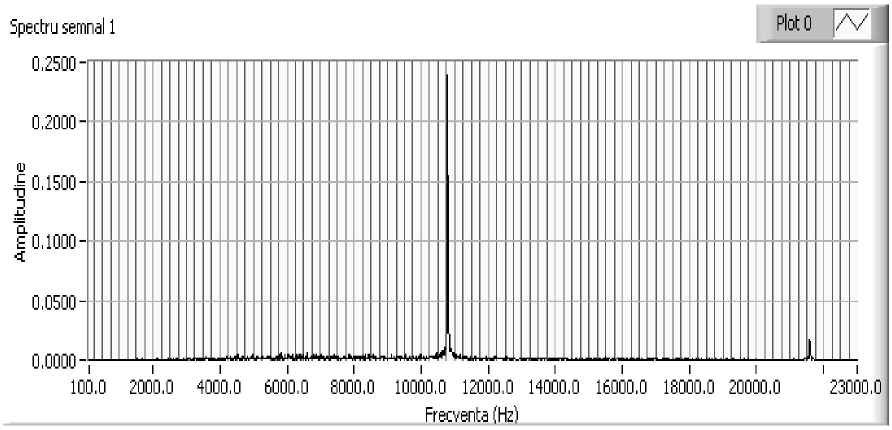

Fig. 4 – FFT spectral analysis of sound pressure pulsations with the program LABwiev (working pressure p = 0.05 MPa).

They found two working frequencies: sound frequency of 10.76 kHz with intensity level equal to 109.88 dB and ultrasound frequency of 21.52 kHz with intensity level equal to 108.09 dB. The overall level of acoustic intensity produced by the generator is 112.32 dB.

Comparing the measured values of frequencies with those obtained from the calculation (first of 10.5 kHz and second of 21.2 kHz), we confirm the positive validation of methodology conceived.

4. CONCLUSIONS

It was designed a method for dimensional parameters calculation of the air-jet generators with two *Levavasseur* type resonators, based on the pressure and volumetric flow rate of working air, fundamental frequency and environment technological pressure in which is generator operated.

Based on the conceived methodology it was designed and built an air-jet experimental generator with two resonators. The acoustic field parameters were studied – the level intensity and frequency noise produced by the generator.

Following acoustic frequencies research carried out were found twoworking frequencies – first is in the audible field, and the second within the ultrasound field. Comparing the working frequency obtained by calculation and the ones from acoustic measurements showed a good coincidence.

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REFERENCES

1. BĂLAN, G., *Aerogazodinamică*, Edit. Tehnica-Info, Chişinău, 2003.
2. GAVREAU, V., *Générateurs pneumatiques d'ultra-sons intenses*, Acustica, **8**, *3*, pp. 121-130, 1958.
3. POPA, B., ISCRULESCU, V., *Procese de ardere în camp sonor*, Edit. Academiei Române, Bucharest, 1973.
4. CROW, S.C., CHAMPAGNE, F.H., *Orderly structures in jet turbulence*, Journal Fluid Mechanics, **48**, *3*, pp. 547-591, 1971.
5. RAJAGOPALAN, S.,WONG, K.T., *Development of the mixing layer of a plane jet under acoustic excitation*, Australian Fluid Mechanics Conference, University of Tasmania, Hobart, Australia, December 14-18, 1992.
6. MA, R, SLABOCH, P.E., MORRIS, S.C., *Fluid mechanics of the flow-excited Helmholtz resonator*, Journal of Fluid Mechanics, **623**, pp. 1-26, 2009.
7. MAGHEȚI, I., SAVU, M., *Teoria şi practica vibrațiilor*, Edit. Didactică şi Pedagogică, Bucharest, 2007.