Numerical Modeling of Discrete Components of Pressure Pulsation Spectra in Bladed Pumps

Sergey F. Timushev, Dmitry V. Klimenko, and Sergey Yu. Fedoseev
Department "Rocket Engines", Moscow Aviation Institute (National Research University) Moscow, Russia
Email: irico.harmony@gmail.com, srmitriy@gmail.com, it202mai@gmail.com

Hassan Karimii
Aerospace Department, K. N. Toosi University of Technology, Tehran, Iran
Email: karimi@kntu.ac.ir

Babak Afzali
Iran Iranian Space Research Center (ISRC), Tehran, Iran
Email: b_afzali@dena.kntu.ac.ir

***Abstract*— Study of improving the reliability and lifetime of bladed pumps is now of great importance. In this context, the key objective is to reduce the hydrodynamic vibration of screw-centrifugal pumps caused by pressure pulsations in the pump outlet casing. Due to the flow stepwise nonuniformity at the impeller outlet, pressure pulsations emerge at the rotor blades passing frequency and its harmonics. These vibrations cause a dynamic load on the components of pump body causing its vibration, so the calculation of the pressure pulsations amplitude in the screw-centrifugal pump at the early stage of the design is a relevant task. When defining pressure pulsations generated by the three-dimensional vortex flow of the screw-centrifugal pump their dual nature should be considered. The heterogeneous distribution of the flow parameters at the outlet of the centrifugal impeller generates acoustic disturbances that are propagated at the speed of sound in the operating fluid. At the same time, there are vortex disturbances that are convected by the main flow. Vortex oscillations of the main flow parameters is called "pseudosound" or the vortex mode. This paper develops a three-dimensional acoustic-vortex method of calculating pressure pulsation, which provides the ability to determine the amplitude of acoustic mode. It shows the derivation of the acoustic and vortex equations and the calculation example of the pressure pulsations amplitude at the screw-centrifugal pump outlet with different guide channel design. It shows the ability of modeling the combination components in the spectrum of pressure pulsations.**

***Index Terms*— pressure pulsations, centrifugal pump, discrete component of blade passing frequency (BPF), mixed harmonic, complex acoustic impedance first term, second term, third term, fourth term, fifth term, sixth term**

I. Introduction

Study of improving the reliability and lifetime of bladed pumps is now of great importance. For example, the screw-centrifugal pump is the main source of noise in hydraulic systems, the key source of hydrodynamic vibrations of the feed system in the modern turbo-pump units. The hydrodynamic vibration of the centrifugal pump is a serious problem on the way of increasing its reliability and lifetime over a long period of time. We first heard of this problem in the 1960s of the last centuries in connection with the destruction of large pumps [1,2]. The hydrodynamic vibration excited by the pressure pulsations occurring in the pump outlet casing due to the different natures of hydrodynamic reasons [3,4] that include vortex formation, flow recirculation, cavitation, stepwise distortion of the flow parameters at the centrifugal impeller outlet. The latter factor causes the pressure pulsations generation on the so-called blade passing frequency (BPF) and its highest harmonics and combination frequencies. These pressure fluctuations are the integral part of the centrifugal pump's working process [5]. In centrifugal pumps, they are of high amplitude due to the features of the flow stepwise inhomogeneity formation in the centrifugal blade row. It is well known that the physical nature of pressure pulsations in the centrifugal pump is a summary manifestation of pseudosound and acoustic oscillations. Pseudosound [6,7] or the vortex mode [8,9,10] attenuate quickly downstream from the rotor [11], leaving in the pressure pipe only the acoustic mode of pressure pulsations. The determination of the pressure pulsations amplitude in the screw-centrifugal pump at the early stage of design is a relevant objective. In determining the pressure pulsations generated by the three-dimensional vortex-like flow in the screw-centrifugal pump, it must be taken into account that the heterogeneous distribution of the flow parameters at the outlet of the inducer creates conditions for generating pressure pulsation at combination frequencies. At the same time, there are vortex disturbances that are convected by the main flow. The dual nature of pressure pulsations in centrifugal pumps is considered by applying the decomposition [12] in the two-dimensional method of calculation. The flow

Manuscript received August 9, 2018; revised July 18, 2019.

doi: 10.18178/ijmerr.8.6.947-953

in modern screw-centrifugal pumps have a significant three-dimensional nature, so there is a need to adapt acoustic-vortex equations for a three-dimensional case. Similar problems with high level of noise emission on the acoustical components that are multiple to the rotor speed are characteristic for the fan acoustics tasks of modern by-pass aircraft engines [13], train fans [14], computer and air-conditioning systems fans [15]. Fan noise consists of acoustical component and a wide-band noise [16]. The acoustical components on frequencies that are multiple to blades passing frequencies (BPF) usually dominate in the spectrum and determine the overall sound power level. Following the aero-acoustical analogy introduced by Lighthill [17], Curl [18], Floux, Williams and Hawkins [19] formulated the theoretical basis for analytical description of the sound generation process by pressure forces acting on the rotor and stator blades from the gas flow side. Currently, computer fluid dynamics and acoustics methods have developed widely allowing this to be used for determining the fans acoustic radiation [20, 21]. They are currently based mainly on the application of the Lighthill equation and aero-acoustic analogies as in the FWH-equation or on the application of Kirchhoff's theorem [22]. To accurately determine the sound power of the acoustic source in pumps and fans it is necessary to apply decomposition, i.e. splitting the acoustic and vortex (pseudosound) mode in the area of the source [23] where it is represented as derivative with respect of time from the vortex mode pressure [24]. This defines the source, the pressure pulsations and acoustic mode propagation in near field as a direct result of the numerical modeling.

II. Derivation of Basic Equations and Boundary Conditions

A. Acoustic-vortex Decomposition

In the isentropic flow, the increment enthalpy, pressure and density are related by thermodynamic relations as follow:

$$dh = \frac{dp}{p},\ dp = a^2 dp \tag{1}$$

where a is the sound velocity in the operating environment.

$$\frac{\partial \mathbf{V}}{\partial t} + \nabla \frac{\mathrm{V}^2}{2} - \mathbf{V} \times (\nabla \times \mathbf{V}) = -\nabla h + \nu\, \Delta \mathbf{V} \tag{2}$$

Let's conduct the decomposition of the velocity field of the compressible environment which is described by the following equations:

$$\frac{1}{a^2}\left(\frac{\partial h}{\partial t} + \mathbf{V}\nabla h\right) + \nabla \mathbf{V} = 0 \tag{3}$$

where ν is the kinematic coefficient of viscosity

$$\frac{d\mathbf{U}}{dt} = -\nabla H + \nu \Delta \mathbf{U} + \nabla \varphi \times \zeta \tag{4}$$

Here, the acoustic mode is introduced by the acoustic potential φ so that

$$\mathbf{V} = \mathbf{U} + \nabla \varphi \tag{5}$$

$$\nabla \times \mathbf{V} = \nabla \times \mathbf{U} = \zeta \tag{6}$$

$$H = h + \frac{d\varphi}{dt} + \frac{1}{2}(\nabla \varphi)^2 - \nu\, \Delta \varphi \tag{7}$$

Where

$$\frac{d}{dt} = \frac{\partial}{\partial t} + \mathbf{U}\nabla \tag{8}$$

The equation for acoustic mode of vibrations can be obtained by substituting i from the ratio (7) into the equation (3).

$$\frac{1}{a^2}\frac{d}{dt}\left(\frac{d\varphi}{dt} + \frac{(\nabla\varphi)^2}{2} - \nu\, \Delta \varphi\right) - \Delta \varphi = \frac{1}{a^2}\frac{dH}{dt} + \nabla \mathbf{U} \tag{9}$$

The right side of the equation (9) is the source of acoustic vibrations which is defined by the nonstationary part of the function (10)

$$S = \frac{1}{a^2}\frac{dH}{dt} + \nabla \mathbf{U} \tag{10}$$

The change in the divergence of the vortex mode velocity vector $\nabla \mathbf{U}$ is due to changes in density in the flow and does not depend on time. In the case of low average velocity of flow, the environment can be considered incompressible and

$$\nabla \mathbf{U} = 0 \tag{11}$$

Taking into consideration that the amplitude of acoustic oscillations is significantly less than pseudosound and, while ignoring the viscous dissipation, we will write the acoustic-vortex wave equation for the case when the vortex mode motion can be considered as the incompressible flow in the form

$$\frac{1}{a^2}\frac{d^2\varphi}{dt^2} - \Delta \varphi = \frac{1}{a^2}\frac{dH}{dt} \tag{12}$$

Taking into account linearization according to, the ratio (7) may be written as follows:

$$-\frac{d\varphi}{dt} = h - H \tag{13}$$

Using the expression (13) the equation(12) can provide (14)

$$\frac{1}{a^2}\frac{d^2 h}{dt^2} - \Delta h = S' \tag{14}$$

The disturbing function in the right side of the equation (14) can be expressed as the unsteady part of the expression(15):

$$-\Delta H=\nabla\left(\nabla\left(U^2/2\right)-\mathbf{U}\times\zeta\right) \tag{15}$$

Using the local complex specific acoustic impedance Z, the boundary condition can be represented in the form (16). Thus, decomposition is also ensured at the boundary conditions.

$$\frac{\partial(h-H)}{\partial n}=-\frac{1}{aZ}\frac{\partial(h-H)}{\partial t} \tag{16}$$

where n is the boundary normal, H is the oscillations of the enthalpy obtained during computation of vortex mode.

B. Acoustic-vortex equation and source function

While ignoring the convective members, the acoustic vortex wave equation can be written in the Cartesian coordinates as

$$\frac{1}{a^2}\frac{\partial^2 h}{\partial t^2}-\frac{\partial^2 h}{\partial x^2}-\frac{\partial^2 h}{\partial y^2}-\frac{\partial^2 h}{\partial z^2}=S' \tag{17}$$

For the case of mean flow low speed, the source function in the equation (17) can be obtained from taking into account (11) in a Cartesian coordinate system, as the unsteady part S of the expression (18).

$$S=2\begin{pmatrix}\frac{\partial U_y}{\partial x}\cdot\frac{\partial U_x}{\partial y}+\frac{\partial U_z}{\partial x}\cdot\frac{\partial U_x}{\partial z}+\frac{\partial U_z}{\partial y}\cdot\frac{\partial U_y}{\partial z}-\\-\frac{\partial U_x}{\partial x}\cdot\frac{\partial U_y}{\partial y}-\frac{\partial U_x}{\partial x}\cdot\frac{\partial U_z}{\partial z}-\frac{\partial U_y}{\partial y}\cdot\frac{\partial U_z}{\partial z}\end{pmatrix} \tag{18}$$

C. Method of acoustic-vortex equation solution

Solution to the equation (17) is divided into two tasks - the calculation of the non-stationary flow for the incompressible fluid model, which defines the disturbing function, and the solution of the inhomogeneous wave equation relative to the pressure pulsations h. A similar approach is used, for example, in the work [25].

For the vortex mode, the unsteady-state Navier-Stokes equations are solved by applying the standard k-ε model of turbulence. Experience shows that such approach produces successful results in simulating BPF pressure pulsations for the pump's steady-state mode of operation. The iterative procedure develops from zero initial conditions to obtaining convergence to periodic oscillating solution, followed by the definition of the source function.

The calculation of the non-stationary flow of the vortex mode using the Navier-Stokes equations (19) and the equations of continuity (20) is taken for the first step of the acoustic-vortex method:

$$\frac{\partial\mathbf{V}}{\partial t}+(\mathbf{V}\nabla)\mathbf{V}=-\frac{\nabla P}{\rho}+\frac{1}{\rho}\nabla((\mu+\mu_t)(\nabla\mathbf{V}) \tag{19}$$

$$\nabla\cdot\mathbf{V}=0 \tag{20}$$

These equations are supplemented by the equations *k-ε (turbulent energy - dissipation rate)* of the turbulence model. In this model of turbulence [26] the turbulent viscosity μ_t is expressed through values k and ε as follows

$$\mu_t=C_\mu\rho\frac{k^2}{\varepsilon} \tag{21}$$

$$\frac{\partial k}{\partial t}+\nabla(\mathbf{V}k)=\frac{1}{\rho}\nabla\left(\left(\mu+\frac{\mu_t}{\sigma_k}\right)\nabla k\right)+\frac{G}{\rho}-(\varepsilon-\varepsilon_{ini}) \tag{22}$$

$$\frac{\partial\varepsilon}{\partial t}+\nabla(\mathbf{V}\varepsilon)=\frac{1}{\rho}\nabla\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\nabla\varepsilon\right)+\frac{\varepsilon}{k}\left(C_1\frac{G}{\rho}-C_2(\varepsilon-\varepsilon_{ini})\right) \tag{23}$$

where ε_{ini} is the initial value of the turbulent dissipation. Via G the expression is designated

$$G=\mu_{eff}\frac{\partial V_i}{\partial x_j}\left(\frac{\partial V_i}{\partial x_j}+\frac{\partial V_j}{\partial x_i}\right) \tag{24}$$

Model k-ε parameter values are equal to:

$$\sigma_k=1;\ \sigma_\varepsilon=1.3;\ C_\mu=0.09;\ C_1=1.44;\ C_2=1.92 \tag{25}$$

The boundary condition for fluid velocity in turbulence flow on the wall is set using a numerical approximation of the logarithmic law for the tangential components of velocity on the wall [27]. The numerical method has been implemented on a rectangular mesh with local adaptation and submesh resolution of complex geometry. This approach ensures the computational accuracy in the high gradients zone of unsteady flow parameters (in rotor-stator interaction zone), as well as the effective solution of the wave acoustic-vortex equation with minimal computational resources.

The vortex mode computations have been carried out using sliding meshes techniques (to transmit data from the rotor to the pump inlet and outlet casing) with three sub-areas shown in the Fig. 1. It also shows the rectangular adaptive computational mesh. The mesh contains more than 250,000 cells. The time step of the non-stationary computation is 0.00001 second of the physical time. The processor time for computation of the rotor's one full turn is about 2 hours on the computer processor i5.

The computational space reflects adequately the actual conditions of the operating fluid, given the hydrodynamic interaction of all components of the pump casing including guide vanes in the inlet, rotating blade system of the rotor with triple threaded screw inducer and centrifugal impeller with seven main and seven additional shortened blades, as well as a twelve-channel guide vane.

When solving the wave equation, the explicit method in complex variables for the BPF individual harmonic is used.

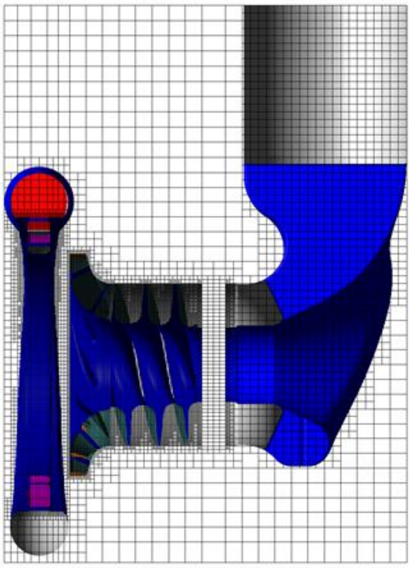

Figure 1. Computational area and mesh.

III. Comparison of Experimental and Computational Results

A. Experimental Screw-centrifugal Pump.

On Picture 2 you can see the pump casing sketch of one of the model screw-centrifugal pumps where the pressure pulsation measurements were performed.

To figure axis labels, use words rather than symbols. Do not label axes only with units. Do not label axes with a ratio of quantities and units. Figure labels should be legible, about 9-point type.

Color figures will be appearing only in online publication. All figures will be black and white graphs in print publication.

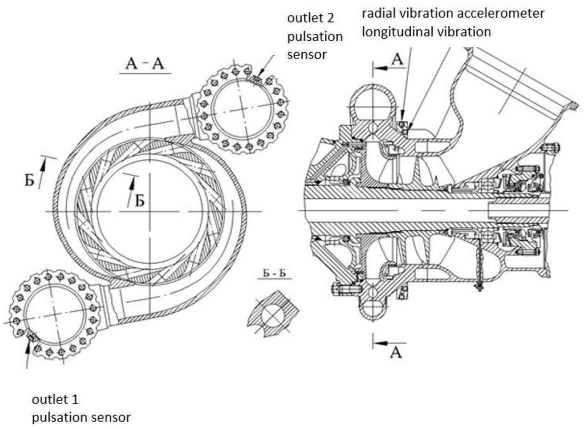

Figure 2. Sketch of the screw-centrifugal pump.

The pump is completed with pressure pulsation sensors in various points of the outlet casing. Pressure pulsation sensors are installed on the channel of guide vane (points 1, 2), along the volute collector (points 3, 4, 5) and at the pump outlet (point 6), as shown in Fig. 3.

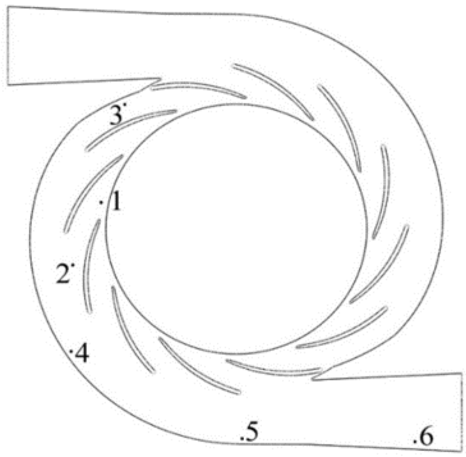

Figure 3. Measurement points.

The obtained pressure pulsation spectra in the outlet casing of the screw-centrifugal pump are characterized by the presence of the broadband noise-induced component and discrete components that are multiple to rotor frequency. It should be noted that, as a rule, the frequencies of all discrete components of the pressure pulsation spectra are multiple to rotor frequency (On Fig. 4, the spectra in measurement points 1, 2 are shown).

At design point of operation, the maximum amplitudes in the pressure pulsations spectra have discrete components at impeller's blade passing frequencies (BPF) and at their highest and mixed harmonics.

BPF harmonics are determined by the formula

$$f_b = k\ z_1\ f_p \tag{26}$$

where

f_p - rotor speed, Hz;

z_1 - number of the centrifugal impeller blades, screw;

k - harmonic number.

To explain this phenomenon and its computational support, the study of the flow nature at the outlet of the centrifugal impeller is of fundamental importance.The issues of experimental and calculation studies of flow in centrifugal impeller machines have been recently given the increased attention in our country and abroad. The studies of flow in centrifugal fans with different geometry of blades are covered in the work [28]. The detailed study of the flow parameters in centrifugal compressors [29] and flow in the absolute and relative motion at the impeller outlet [30,31] of the centrifugal pumps confirm that the flow in the blade channel and at the outlet of the centrifugal impeller can be divided into two areas, namely, high-energy jet and low-energy wake area. Such nature of flow determines the significant nonuniformity of relative and absolute speeds and angles along the impeller's cascade pitch since the low energy zone is adjacent to the blade's trailing side. Due to the above discussed heterogeneity of the flow, when passing the impeller's blades, the pressure change in the pump's stator takes place periodically with frequencies of rotating blades passing. Particularly sharp change of flow parameters takes place near the blade leading edge of the guide vanes and on the volute tongue, in the area of hydrodynamic interaction of rotor-stator. That is why so much attention is paid to choice of the optimum gap between impeller and the guide vane or volute tongue [32,33,34].

Such factors as the impeller geometry deviation in the circumferential direction and asymmetrical position of the impeller's blade leading edges relative to the blades of the superposed axial screw inducer result in circumferential distortion of flow parameters distribution at the impeller outlet with the order of circular symmetry equal to 1. Rotating together with the impeller, this nonuniformity excites pulsations of pressure in the pump outlet with rotor frequency.

In this centrifugal pump with double-row centrifugal impeller with seven main and seven additional shortened blades, and three-blade screw on Fig. 4, in the pressure pulsation spectra, the BPF harmonic components with rotor frequencies of 3, 7, 14 and mixed harmonics that are multiple to 4, 10, 11 from the rotor frequency were registered.

The first phase computation of the vortex mode is performed in the model of incompressible fluid. This allows defining the pressure pulsations near the stator and rotor interaction zone rather precisely, in particular in the guide vane channels and at the beginning of volute where the vortex mode of oscillations is dominated.

The computational analysis of the pressure pulsations in the screw-centrifugal pump with three-blade screw, seven main and seven additional blades of the centrifugal impeller identifies the dominant discrete components at channel input of the guide vane at frequencies 4, 7 and 14 f_p.

Considering the main discrete components of pressure pulsations spectrum at the input of the guide vane channel, it can be noted that a significant change in the amplitude of the discrete component takes place when the flow rate and design flow rate ratio is 0.65-0.8 (see Fig. 5 – 6). During the flow rate reduction, the BPF amplitude according to the number of main blades and mixed harmonic 4 f_p increase essentially The amplitudes of the blade passing frequencies and the mixed harmonic decrease with the flow rate further reduction and rising in the intensity of back flows.

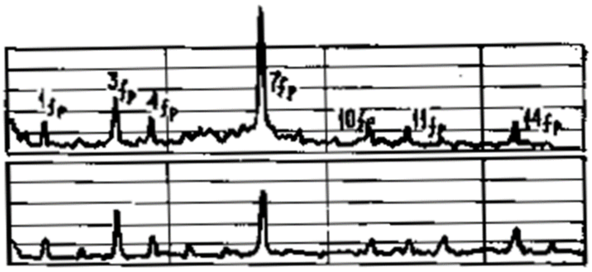

Figure 4. Amplitude-frequency spectra of pressure pulsations in the guide vane channel.

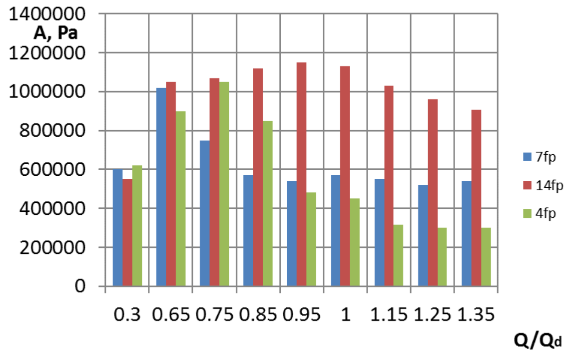

Figure 5. Amplitudes change of discrete components of the pressure pulsations spectrum in a guide vane channel.

These results confirm the conclusion that the occurrence of back flows in the pump input is accompanied by a disruption of the flow circular symmetry in rotor and symmetry of the spiral flow in the pump's volute outlet contributing to the pressure oscillations generation on rotor and sub-rotor frequencies.

The corresponding computation data on the total signal of pressure pulsations and spectrum is shown for the flow rate mode 0.85 on Fig. 5.

The pressure pulsations amplitude of the spectrum discrete components significantly reduces due to the rapid attenuation of the vortex mode oscillations (pseudosound). For the computational analysis of this effect, the pressure oscillation modeling in the experimental pump with a double volute was carried out by solving the acoustic-vortex equation (17).

The comparison of the design and measured data is shown on Fig. 7. BPF amplitude reduction is found to be more than 20 dB along the pump's outlet path downstream from the outlet of the centrifugal impeller to the pump outlet.

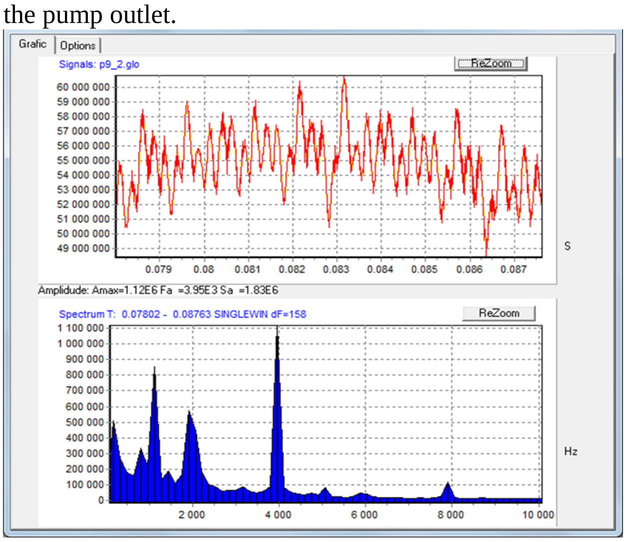

Figure 6. Total signal and the pressure pulsations spectrum in the guide vane channel at the relative flow rate of 0.85.

IV. Discussions

The explanation of mixed harmonics appearance in pressure pulsations spectrum is associated with the amplitude modulation of the flow vortex disturbance in the circumferential direction of the impeller cascade. The flow circular symmetry change causes the vortex oscillations of pressure in the absolute motion at the rotating impeller outlet.

Neglecting the initial phases, one can represent a signal with a frequency ω, modulated with the frequency Ω as follows:

$$s(t) = s_m(t)\cos(\omega t) = S[1 + M\cos(\Omega t)]\cos(\omega t) \quad (27)$$

The modulation depth M is defined as the ratio of the modulating disturbance amplitude to the amplitude of the main oscillation.

$$M = \frac{S_m}{S} \quad (28)$$

As a result of the amplitude modulation in the spectrum of pressure pulsations, combination (sideband) components with frequencies $(\omega - \Omega)$ and $(\omega + \Omega)$ emerge according to (29).

$$s(t) = S\cos(\omega t) + S\frac{M}{2}\cos[(\omega+\Omega)t] + S\frac{M}{2}\cos[(\omega-\Omega)t] \quad (29)$$

The power ratio of the lateral component and the main frequency can be $M^2/2$.

Thus, the initial nonuniformity of flow caused by the inducer modulates the uneven flow in the centrifugal impeller. With this, in the spectrum, the combination (sideband) components with frequencies emerge

$$f_m = f_p(mz_1 \pm z_a),\ m = 1, 2, 3... \quad (30)$$

where z_a is the number of blades of inducer or another order of circular symmetry in the relative motion.

Given this fact, it is possible to influence purposefully the spectral composition of pressure pulsations and the centrifugal pump vibration. Thus, the use of a centrifugal wheel with six main blades instead of seven in the same pump eliminates completely the discrete component 4 f_p .

The emergence of mixed harmonics [35] can also be associated with the global instability of the input flow or the non-stationary separation processes in the centrifugal impeller, such as the rotating stall. Mixed harmonics can significantly change the spectral composition of the pressure pulsations and mask the manifestation of pressure pulsations at the blades passing frequency.

V. Conclusion

The modeling of generation and propagation of pressure pulsation in screw-centrifugal pumps has been carried out using the acoustic-vortex decomposition of the compressible fluid pressure field.

The computations have proved that the reduction of BPF amplitude from the impeller exit downstream to the pump outlet section is more than 20 dB.

The appearance of discrete components in the pressure pulsations spectrum at (combined) frequencies is explained by the amplitude modulation of the flow stepwise inhomogeneity in the centrifugal impeller by the uneven flow at the screw inducer outlet. The similar phenomenon can be caused by another circular flow asymmetry in a relative coordinate system.

Acknowledgment

This research was partly funded by Russian Foundation for Basic Research (RFBR) under application No. 17-58-560004 and Iran National Science Foundation (INSF) under application No. 96003860.

Sergey Timushev was born in 1955, graduated from Moscow Aviation Institute (MAI) in 1978. Since 1980, he worked for Buran space program and from 1992, within Moscow Aviation Institute, for Arianne-Vulcain project. He made many experimental and computational studies in the field of pressure pulsation and vibration in centrifugal turbopump units of Liquid Propellant Rocket Engines (LPRE), including reduction of pressure pulsation and vibration caused by cavitation, reverse flows, acoustical amplification. In 1986, he proposed the original method for numerical modeling of pressure pulsation in centrifugal machines.

The last decade he works on the problem of computational prediction of pressure pulsation in centrifugal pumps and ventilators and reduction of vibration and noise in such machines. In 2002-2004 he worked for EU scientific project NORMA on noise and vibration reduction in high speed ships.

He received his Dr.Tech.Sc. degree in 1996 on numerical modeling of pressure pulsations in centrifugal pumps and ventilators. He is author and co-author of 100 scientific papers and is an affiliate member of the INCE/Europe, member of IIAV and ASME.

Currently he works as a professor of turbomachinery in the Department 202 "Rocket Engines" of MAI.

Fedoseev S. Yu. was born in 1989 in Moscow, Russia. He graduated Moscow Aviation institute (National Research University), rocket engines department, in 2012. He received PhD in engineering in 2016. His main scientific interests are noise and vibration in turbomachines, unsteady phenomena in pump systems and CFD- analysis of the flow in turbomachines.

Klimenko D.V. was born in 1990 in Moscow, Russia. He graduated Moscow Aviation institute (National Research University), rocket engines department, in 2012. He received PhD in engineering in 2017. His main scientific interests concentrated on application of CFD analysis for optimization of turbomachinery flow path with the main goal to improve the cavitation quality, reduce noise and vibration.

Hassan Karimi was born in Iran in 1967. He received his B.Sc. in mechanical engineering from Isfahan University of Technology, Iran and his M.Sc. and Ph.D. in aerospace engineering from Moscow Aviation Institute, Russia. His main research interests are in the areas of modeling of reactive flows, space propulsion systems, heat transfer and system design. He is a faculty member of aerospace department of K. N. Toosi University of Technology as professor since 1999 and now he is the head of the department. He was president of Iranian Space Research Center from 2011 to 2015. He has more than 100 publications in national and international journals and conferences. He is also the author of 3 books in Persian. Professor Karimi has been selected as expert reviewer of Khwarizmi International Scientific Award in Iran for 5 times and is also the educational assistant and director of joint programs between Iran and Russia since 2009.

Babak Afzali was born in Iran in 1982. He received his B.Sc. from Amirkabir University of Technology, Iran and his M.Sc. and Ph.D. from K. N. Toosi University of Technology, Iran. He has been involved in a wide range of research projects spanning from industry-led technology development in the field of spacecraft and satellite systems to academic researches on active flow control using high frequency actuators, resonance ignition systems, CFD, simulation of fuel supply systems and simulation of many industrial processes. He has more than 20 papers in national and international journals and conferences. He is a faculty member of Iranian Space Research Center (ISRC) since 2018. Dr. Afzali received the best thesis award among Iranian PhD theses in the field of aerospace engineering in 2017.

REFERENCES

[1] Y. N. Chen Water, *Pressure Oscillations in the Volute Casings of Storage Pumps.-Sulzer Technical Review*, Research number, pp. 21-34, 1961.

[2] R. A. Shtrub, Pressure Oscillations and Fatigue Stresses in Pumps and Combined Pump-Turbine Units of the Hydroelectric Pumped Storage Power Plants. - Power Machines and Installations, 1964, vol. 86, no. 1, pp. 117-121.

[3] B. V. Pokrovsky, E. Y. Yudin, "The main features of the noise and vibration of centrifugal pumps," *Journal of Acoustics*, vol. XII, no. 3, pp. 355-364, 1966.

[4] J. F. Guelich, U. Bolleter, "Pressure pulsations in centrifugal pumps," *Transactions of the ASME. Journal of Vibration and Acoustics*, April 1992, vol. 114, pp. 272-279.

[5] O. Petit, H. Nilsson, "Numerical investigations of unsteady flow in a centrifugal pump with a vaned diffuser," *International Journal of Rotating Machinery*. vol. 2013, Article ID 961580, pp. 14.

[6] D. I. Blokhintzev, *Acoustics of Inhomogeneous Moving Medium*, M. Nauka, 1981, 206 p.

[7] L. D. Landau, E. M. Lifshitz, "Continuum mechanics," M.: State Publishing House of Technical-Theoretical Literature, 1954, pp. 795.

[8] K. I. Artamonov, *Thermohydroacoustic Stability*, M.: Mechanical Engineering, 1982, pp. 261.

[9] M. E. Goldstein, "Aeroacoustics: Transl. from English," M.: Mechanical Engineering, 1981, pp. 295.

[10] E. P. Syolyarov, "Sound excitation by small disturbances of entropy and vorticity in spatially-inhomogeneous flows of the impressing ideal gas," *The Acoustics of Turbulent Flows*, M.: Nauka, 1983, p. 3-14.

[11] T. Yuasa, T. Hinata, "Flow pulsation behind the centrifugal wheel," Jehara Dziho, 1980, no. 114 (translated from Japan No.G-39508.) M.: VTsP, 1981), 7 p.

[12] S. Timushev, J. Tourret, "Numerical simulation of BPF pressure pulsation field in centrifugal pumps," in *Proc. 19th International Pump Users Symposium*, February 2002, Houston, Texas

[13] Y. D. Khaletskii and A. S. Pochkin, "Fan noise reduction of an aircraft engine by inclining the outlet guide vanes," *Acoustical Phys.* vol. 61, no. 1, 2015, pp. 101-108.

[14] Y. Jiang, M. Åbom, L. P. Feng, S. Timouchev, C. Maliczak, "Investigation of sound generation from an axial fan for engine cooling," in *Proc. 3rd International symposium on Fan Noise*, 19-21 September, 2007, Lyon, France.

[15] J. Thimany, Attacking Noise Early. Mechanical Engineering, pp. 136, no. 8, pp. 44-49, 2014.

[16] A. Guédel, "Acoustique des ventilateurs," CETIAT. PYC LIVRES, 1999

[17] M. J. Lighthill, *Proceedings of the Royal Society*, London A 211,. On sound generated aerodynamically. Part I. General Theory, 1952, pp. 564-587.

[18] N. Curle, "The influence of solid boundaries upon aerodynamic sound," *Proc. Royal Soc*. A 231, p.505-514, 1955.

[19] J. E. Flowcs-Williams and D. L. Hawkings, "Philosophical transactions of the royal society A264," Sound generation by turbulence and surfaces in arbitrary motion , pp. 321-342, 1969.

[20] S. Caro, S. Moreau, "Aeroacoustic modelling of low pressure axial flow fans," in *Proc. 6th AIAA/CEAS Aeroacoustics Conference*, 2000.

[21] S. Caro, R. Sandboge, J. Iyer, Y. Nishio, "Presentation of a CAA formulation based on lighthill's analogy for fan noise," in *Proc. of 3rd International symposium on Fan Noise*, Lyon, France, pp. 19-21 September, 2007.

[22] F. Farassat and M. K. Myers, "Extension of Kirchhhoff's formula to radiation from moving surfaces," *Journal of Sound and Vibration*, vol. 123, pp. 451-461, 1988.

[23] A. A. Aksenov, V. N. Gavrilyuk, and S. F. Timushev, "Numerical simulation of tonal fan noise of computers and air conditioning systems," *Acoustical Physics*, vol. 62, no. 4, pp. 447–455, 2016.

[24] S. Timouchev, J. Tourret, G. Pavic, A. Aksenov, "Numerical 2-D and 3-D methods for computation of internal unsteady pressure field and near-field noise of fans," *Noise Control Engineering Journal*, vol. 54, no. 1, January 2006, pp. 15-20

[25] N. Witkowski, A. Hueppe, and M. Kaltenbacher, "Comparison of compressible and incompressible CFD methods for the acoustic analysis of flow induced noise in confined flows," NOVEM 2015, No.49310, 9 p.

[26] D. C. Wilcox, "Turbulence modeling for CFD" // DCW Industries, Inc. pp. 460, 1994.

[27] Software package for gas and fluid flow simulation FlowVision. Version 2.5.0. Manual CAPVIDIA, 1999-2007 Leuven, Belgium, 284 p.

[28] R. Sweem, "Measurements of the mean flow velocity and pulsations at the outlet of the centrifugal fan with forward curved blades," *Energy Machines and Installations*, 1981, no. 2.

[29] Eckardt, "Instantaneous measurements in the jet-wake discharge flow of a centrifugal compressor impeller," *Energy Machines and Installations*, 1975, no. 3.

[30] A. I. Timshin, "Experimental research of the flow structure at the outlet of the pump's impeller," *Hydraulic Machines*. Kharkiv: Publishing house KhGU, 1971, no. 4.

[31] V. S. Erdrich, R. N. Zhukovsky, "The study of unsteady flow in the model pump-turbine of the radiaxial type in pumping mode," in *Proc. VNII Gidromash. M.*, 1972, no. 44

[32] J. S. Bunjes, "Diffuser or volute pump? A comparison of perfomances: Fluid Machinery for the Oil," Petrochemical and Related Industries./ Eur. Congr. The Hague, 24-26 March, 1981. London: Mech. Eng. Publ, 1981.

[33] B. V. Pokrovsky, V. Y. Rubinov, "Effect of discharge on the vibroacoustic characteristics of the centrifugal pump," in *Proc. VNII Gidromash. M.*, 1970, no. 40.

[34] W. H. Fraser, I. J. Karassik, A. R. Bush, "Study of pump pulsation, surge and vibration throws light on reliability vs efficiency," *Power*, 1977 August.

[35] G. Pavesi, G. Cavazzini, and G. Ardizzon, "Time-frequency characterization of rotating instabilities at a centrifugal pump with a vaned diffuser," *International Journal of Rotating Machinery* vol. 2008, pp. 10.