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NOISE CONTROL FOR A BETTER ENVIRONMENT

Centrifugal fan pressure pulsations and noise prediction with a novel CFD-CAA procedure

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ABSTRACT

This work proposes a new high-performance numerical simulation method of the three-dimensional acoustic field of a tonal component on blade passing frequency and its higher and combined harmonics produced by fans in computer devices, air conditioning systems and in aircraft and spaceship cabins. This method is based on the direct solution by a finite volume method of a Fourier-transformed convective wave equation that describes the propagation of sound in adiabatic and thermodynamically uniform irrotational stationary flow with respect to pressure perturbations in the form of a Fourier transformation. Boundary conditions are defined on sound absorbing boundaries in the complex impedance form, considering both the active and reactive component of the boundary impedance. Noise reduction is achieved by spatial redistribution of the tonal sound, which ensures its essential reduction in areas where people are located. The most effective approach is use of multi-layered sound-absorbing cellular structures (SAS), which can be installed on the inner surfaces of noise sources, and on the walls, ceilings and partitions of buildings. Optimal parameters of SAS and their location can be determined by multi-parametric computations of spatial sound fields for each tonal component of interest using the developed method.

Keywords: Noise, Tonal sound, BPF component
I-INCE Classification of Subject Number: 13

1. INTRODUCTION

Generation of hydrodynamic noise in the flow section of centrifugal fans occurs

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due to various non-stationary hydrodynamic phenomena which can be conditionally subdivided into the following two types:

1) those occurring due to hydrodynamic interaction between the flow leaving the impeller and the fan casing;

2) vortex phenomena;

The first type of non-stationary processes is a natural characteristic of centrifugal fans and of all bladed machines: it is due to stepwise non-uniformity of the flow at the impeller outlet. The non-stationary hydrodynamic interaction between the non-uniform flow (rotating together with the impeller) and the fan case causes vibrations at frequencies that are multiples of the impeller rotation speed.

The second type is non-stationary flow caused by the vortex nature of the fluid flow, which is expressed as small-scale turbulence in the boundary layer, formation of turbulent wakes during streamlining of housing elements, and generation of large-scale vortex structures and separation flows.

Study of the vibroacoustic characteristics of radial fans has shown that the intensity of vibration from hydrodynamic sources is proportional to the 4-6th degree of rotor rotation speed.

Small-scale vortices generate turbulent noise, which produces a low-intensity broadband component in pressure and vibration pulsation spectra.

Large-scale vortex structures generate high-amplitude weakly correlated pressure pulsations, expressed as an increase of the broadband component of the spectrum in low and medium frequency ranges.

Studies of centrifugal fans indicate that, as a rule, maximum amplitudes in the spectra of pressure and vibration pulsations in design conditions have discrete components at the frequencies of the impeller blades (BPF).

Studies of the flow in centrifugal fans with various blade geometries have been described in the papers [1,2]. Detailed studies of flow parameters in centrifugal compressors [3,4] and the flow in absolute and relative motion at the centrifugal pump impeller outlet [5] studies of flow in rectangular channels [6,7] which confirm that flow in the blade channel and at the outlet of the centrifugal impeller can be subdivided into two areas – a high-energy jet-flow and a low-energy wake-flow. This type of flow pattern entails substantial non-uniformity of relative and absolute velocities and angles of the flow across the impeller cascade spacing, since the low-energy area adjoins the trailing side of the blade. The distribution of static pressure across the blade cascade spacing at the impeller outlet, is close to uniform, so the difference in total fluid energy is primarily related to the fact that the velocities and angles of the flow are higher at the front side of the blade. Due to the non-uniform flow discussed above, the passing of the impeller blades entails periodic change in pressure in the casing correlated with frequency of the passing blade. Particularly drastic changes of flow parameters occur near the leading edges of the guide blades and at the volute tongue [8], which explains the attention paid to selection of an optimal gap between the impeller and casing. The present paper addresses numerical investigation of the noise generated by a centrifugal fan unit and offers some results of the study. The fan unit is used to pump air from the living module through absorbing cartridges connected to outlets of the unit. The fan unit is installed on a partition. It entrains air from the living zone of the command module, pumps it through the absorbing cartridges and discharges it into the instrument area. The operating mode of the unit is continuous (only one fan operates, and the outlet of the second fan is closed with a flap).

2. METHOD

2.1 Governing Equations

The model describing an isoentropic non-viscous flow is based on a concept suggested by Crow. S. [9] and Artamonov K. I. [10]. This model is used as an example to analyze the method of acoustic-vortex decomposition and the source of acoustic radiation. The equations that are normally used in aeroacoustic computation models were derived using the terms of hydrodynamic flow and, as mentioned in papers by Fedorchenko A. T. [11], Doak P.[12] and Goldstein M. [13], cannot be treated as purely acoustic. The right-hand sides of these equations, or the source terms, describe the generation of disturbances of flow parameters without distinguishing the acoustic component, and the left-hand sides describe the spatial-temporal propagation of wave-type acoustic and vortex disturbances taking into account the convective transfer and spatial non-uniformity of sound velocity.

Boundary conditions of the impedance type are recorded in terms of disturbances of the flow parameters compared with their average value. The original aeroacoustic equations are linearized with respect to their disturbed values. Such linearization enables Fourier transformation of the obtained equations and impedance boundary conditions and formulation of the respective boundary problem. Papers [14,15,16] contain validation and demonstrative examples of an efficient numerical solution of a boundary problem formulated using the proposed method for the modified Crow-Artamonov model using the finite volume method on a Cartesian grid adapted to the computational space boundary.

Definitions

Angular brackets $\langle\ \rangle$ imply than the time average of the formula inside the brackets is used. A stroke implies its oscillatory component with respect to the average value.

We express enthalpy as a sum $h=\langle h\rangle+h'$, where enthalpy is $h=\int_{p_{ref}}^{p}\frac{dp}{\rho}$, and its pulsation component is $h'=\int_{\langle p\rangle}^{\langle p\rangle+p'}\frac{dp}{\rho}$, where $p'=p-\langle p\rangle$ are pressure pulsations with respect to its time-averaged value.

The adiabatic sound velocity is used, determined using time-averaged parameters of the medium: $c=c(\langle p\rangle,\langle S\rangle)$.

The Fourier transformation $\Phi(\omega)$ from the time function $f(t)$ is as follows:

$$\Phi(\omega)\cdot f(t)=\frac{1}{\sqrt{2\pi}}\int_{-\infty}^{\infty}e^{-i\omega\cdot t}f(t)\cdot dt$$

where $\omega>0$ is frequency.

2.2 Modified Crow-Artamonov model (isoentropic non-viscous flow)

Decomposition of the velocity field and derivation of the wave equation with respect to enthalpy

We express velocity $\mathbf{v}$ as the sum of main flow velocity $\mathbf{u}$ and velocity of acoustic $\nabla\varphi$ motion.

$$\mathbf{v}=\mathbf{u}+\nabla\varphi \tag{1}$$

We express the law of conservation of momentum and mass as follows:

$$\frac{\partial\mathbf{v}}{\partial t}+\nabla\frac{\mathrm{v}^2}{2}-\mathbf{v}\times\left(\nabla\times\mathbf{v}\right)=-\nabla h+\upsilon\,\Delta\mathbf{v} \tag{2}$$

$$\frac{\partial\rho}{\partial t}+\nabla\rho\mathbf{v}=0 \tag{3}$$

where υ is the kinematic viscosity coefficient.

For an adiabatic flow a small disturbance of pressure and density is associated with the disturbance of enthalpy, as follows:

$$\delta h=\frac{\delta p}{\rho}=c^2\frac{\delta\rho}{\rho} \tag{4}$$

We express Equation (3) in a non-divergence form:

$$\frac{1}{\rho}\frac{D\rho}{Dt}+\nabla\mathbf{v}=0 \tag{5}$$

Hence, taking account of Equation (4), we obtain an equation in respect of enthalpy:

$$\frac{1}{c^2}\frac{Dh}{Dt}+\nabla\mathbf{v}=0 \tag{6}$$

Now let us suppose that the acoustic component of velocity is much less than the main flow velocity:

$$\left|\nabla\varphi\right|<<\left|\mathbf{u}\right| \tag{7}$$

Then from Equation (2) we derive:

$$\frac{\partial(\mathbf{u}+\nabla\varphi)}{\partial t}+\nabla\frac{(\mathbf{u}+\nabla\varphi)^2}{2}-(\mathbf{u}+\nabla\varphi\times\left(\nabla\times(\mathbf{u}+\nabla\varphi)\right)=-\nabla h+\upsilon\Delta(\mathbf{u}+\nabla\varphi) \tag{8}$$

$$\frac{d\mathbf{u}}{dt}=-\nabla H+\upsilon\Delta\mathbf{u}+\nabla\varphi\times\nabla\times\mathbf{u} \tag{9}$$

where:

$$H=h+\frac{d\varphi}{dt}+\frac{1}{2}\left(\nabla\varphi\right)^2-\upsilon\,\Delta\varphi \tag{10}$$

$$\frac{d}{dt}=\frac{\partial}{\partial t}+\mathbf{u}\nabla \tag{11}$$

Neglecting in Equation (9) and Equation (10) the viscous terms and squared velocity of acoustic motion $\left|\nabla\varphi\right|^2$, as well as the effects of interaction between the acoustic and vortex modes (the last term in Equation (9)), we obtain:

$$h=H-\frac{d\varphi}{dt} \tag{12}$$

$$\frac{d\mathbf{u}}{dt}=-\nabla H \tag{13}$$

Inserting Equation into Equation (5), we obtain:

$$\frac{1}{c^2}\frac{d}{dt}\left(H-\frac{d\varphi}{dt}\right)+\Delta\varphi+\nabla\mathbf{u}=0 \tag{14}$$

Or, taking account of the adopted linearization pattern

$$\frac{1}{c^2}\frac{d^2\varphi}{dt^2}-\Delta\varphi=\frac{1}{c^2}\frac{dH}{dt}+\nabla\mathbf{u} \qquad (15)$$

Applying a gradient operator to Equation (13), we obtain:

$$\nabla\frac{d\mathbf{u}}{dt}=-\Delta H \qquad (16)$$

Applying operator $\frac{d}{dt}$ to Equation (15) and, neglecting the difference $(\mathbf{u}\nabla)\Delta\varphi-\Delta(\mathbf{u}\nabla\varphi)$, we obtain:

$$\frac{1}{c^2}\frac{d^2(H-h)}{dt^2}-\Delta(H-h)=\frac{1}{c^2}\frac{d^2H}{dt^2}+\frac{d}{dt}\nabla\mathbf{u} \qquad (17)$$

Hence,

$$\frac{1}{c^2}\frac{d^2h}{dt^2}-\Delta h=\nabla\frac{d\mathbf{u}}{dt}-\frac{d}{dt}\nabla\mathbf{u} \qquad (18)$$

Source function and Fourier transformation

Inserting enthalpy into Equation (18) in the form of $h=\langle h\rangle+h'$, we obtain the following equation:

$$\frac{1}{c^2}\frac{d^2h'}{dt^2}-\Delta h'=f \qquad (19)$$

with a source term

$$f=\nabla(\mathbf{u}\nabla)\mathbf{u}-(\mathbf{u}\nabla)\nabla\mathbf{u}-\frac{1}{c^2}\frac{d^2\langle h\rangle}{dt^2}+\Delta\langle h\rangle \qquad (20)$$

Taking into account Equation (7) allowing substitution of the vortex mode velocity by full velocity in Equation (20), applying the Fourier transformation $\Phi(\omega)$ to Equation (19) and Equation (20), and changing over to tensor form, we express Equation (19) as:

$$\frac{1}{c^2}\left(i\omega+\langle \mathrm{v}_i\rangle\frac{\partial}{\partial x_i}\right)\left(i\omega+\langle \mathrm{v}_j\rangle\frac{\partial}{\partial x_j}\right)h'-\frac{\partial^2 h'}{\partial x_i^2}=\Phi(\omega)(f) \qquad (21)$$

$$\Phi(\omega)(f)=\Phi(\omega)\left(\frac{\partial \mathrm{v}_i}{\partial x_k}\frac{\partial \mathrm{v}_k}{\partial x_i}\right) \qquad (22)$$

Boundary conditions

A projection of the linearized and Fourier transformed equation of the impulse to wall normal, obtained on the assumption that the value of the normal Mach number is small, is as follows:

$$i\omega\cdot \mathrm{v}_n'+\frac{\partial h'}{\partial n}=0 \qquad (23)$$

where v_n' is the component of pulsating velocity normal with respect to the wall.

We express the Fourier transformed Meyers condition [17] on the outer surface of the SAS:

$$i\omega v'_n = \left(i\omega + \langle v_i \rangle \frac{\partial}{\partial x_i} - \frac{\partial \langle v_i \rangle}{\partial x_k} n_i n_k \right) \frac{p'}{\rho c Z} \quad (24)$$

Complex impedance Z is a function of ω ,the Mach number, pulsation amplitude and characteristics of the boundary layer.

The Sommerfeld radiation condition under non-reflecting boundary conditions is recorded taking into account the curvature correction of the wave front [18], proportional to the Laplacian operator in a tangent line to the plane surface.

Taking this into account, the boundary, or edge conditions are expressed as follows.

$$\frac{\partial h'}{\partial n} + \left(i\omega + \langle v_i \rangle \frac{\partial}{\partial x_i} - \frac{\partial \langle v_i \rangle}{\partial x_k} n_i n_k \right) \left(\frac{\partial p}{\partial h} \right)_s \frac{h'}{\rho c Z} = 0 \quad (25)$$

Non-reflecting condition at the outer boundary:

$$\frac{\partial h'}{\partial n} + \left(i\omega + \langle v_i \rangle \frac{\partial}{\partial x_i} - \frac{\partial \langle v_i \rangle}{\partial x_k} n_i n_k + \frac{ic^2}{2\omega} \Delta_n \right) \left(\frac{\partial p}{\partial h} \right)_s \frac{1}{\rho c} h' = 0 \quad (26)$$

Unsteady Flow Solution

The following equations [19] are solved when modelling air motion:

Continuity equations and impulse equations for modelling of fluid and gas motion:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \boldsymbol{V}) = 0 \quad (28)$$

$$\frac{\partial \rho \cdot \boldsymbol{V}}{\partial t} + \nabla(\rho \cdot \boldsymbol{V} \times \boldsymbol{V}) = -\nabla p + \nabla((\mu + \mu_t) \cdot (\nabla V + (\nabla V)^\tau)) + \rho \cdot g \quad (29)$$

KEFV turbulence model equations are used.

Turbulent viscosity is calculated on the basis of the following correlation:

$$\mu_t = C_\mu \cdot \rho \cdot \frac{k^2}{\varepsilon} \quad (30)$$

Equation for turbulent energy k:

$$\begin{aligned} \frac{\partial(\rho \cdot k)}{\partial t} &+ \nabla(\rho \cdot V \cdot k) \\ &= \nabla\left((\mu + f_t \frac{\mu_t}{\sigma_k}) \nabla k \right) + \rho \cdot \left(P_k + P_{k.gen} + G_k \right) \\ &- \rho\varepsilon(1 + \xi(\max(M_t^2, M_{t0}^2) - M_{t0}^2) - D_{pp} \end{aligned} \quad (31)$$

Equation for the turbulent energy dissipation rate ε:

$$\begin{aligned} \frac{\partial(\rho \cdot \varepsilon)}{\partial t} &+ \nabla(\rho \cdot V \cdot \varepsilon) \\ &= \nabla\left((\mu + \frac{\mu_t}{\sigma_\varepsilon}) \nabla \varepsilon \right) \\ &+ \frac{1}{\sqrt{1 + {}^{2}/_{Re_t}}} \frac{\varepsilon}{k} \rho (C_{e1} f_1 (P_k + G_k) - C_{e2} f_2 \varepsilon) \end{aligned} \quad (32)$$

Coefficients:

$$\sigma_k = 1;\ \sigma_\varepsilon = 1.3;\ C_\mu = 0.09; C_{e1} = 1.44; C_{e2} = 1.92 \quad (33)$$

Standard wall functions at the wall boundary are used.

2.3 Computational Domain and Grid

A computational grid divides the computational space into cells in a rectangular parallelepiped, which circumscribes the computational space.

Adaptation is local refinement of the initial computational grid. Subdivision of the grid cell is performed by means of bisection of each hexagonal cell edge in such a way as to obtain eight equal parallelepipeds of the cell of the next level of density. The cells of the initial computational grid are considered as zero level cells, the result of their single subdivision as first level cells, etc.

The initial computational grid (subdivision of the computational space into cells in a rectangular parallelepiped), which circumscribes this computational space, is selected as uniform along all coordinate axes and cubical (with a 15 mm side). Dispersion (adaptation) of the computational grid is carried out in volume - within the entire sliding subarea - and along the walls of the housing and rotor (ref. Fig. 1 - 2). The computational grid was selected when studying the mesh convergence.

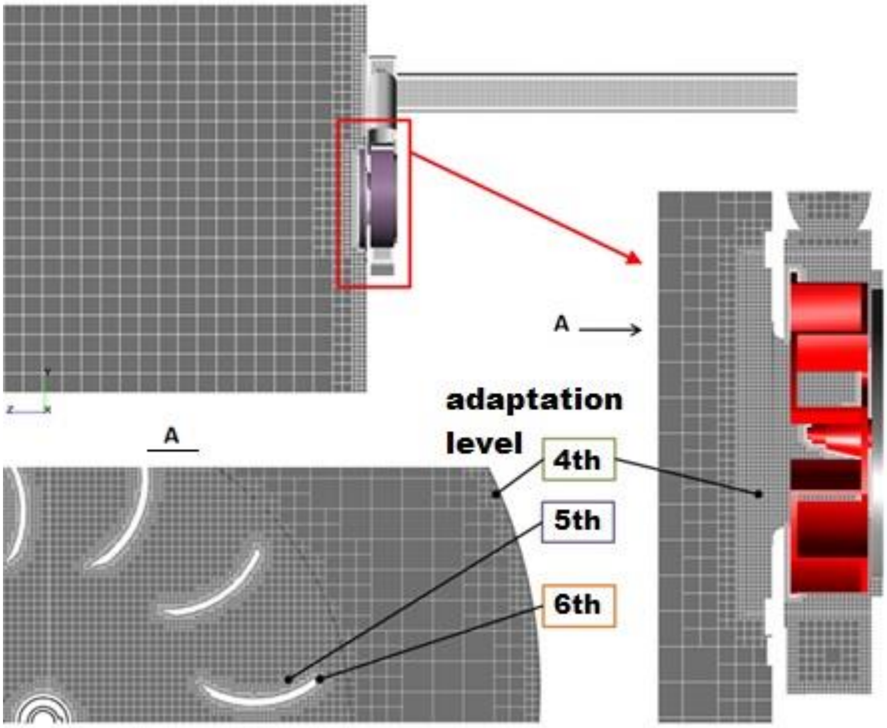

Figure 1. Computational grid for determination of fan characteristics

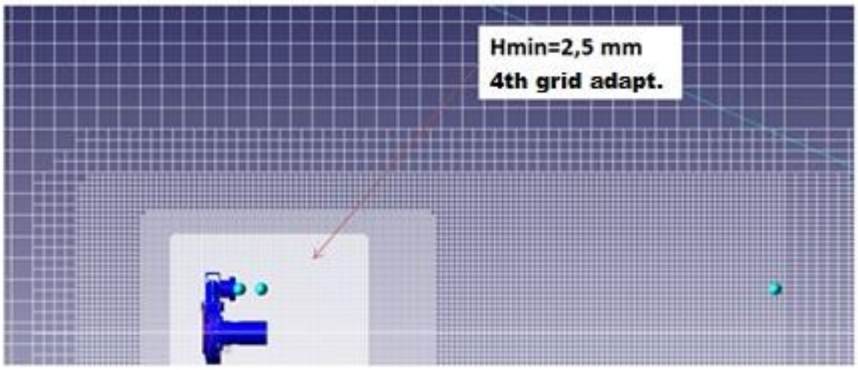

Figure 2. Computational grid for determination of acoustic field

3. RESULTS

3.1 Velocity field

The flow pattern is represented by velocity vectors (Fig. 3 - 4) and streamlines (Fig.3) in the specified planes.

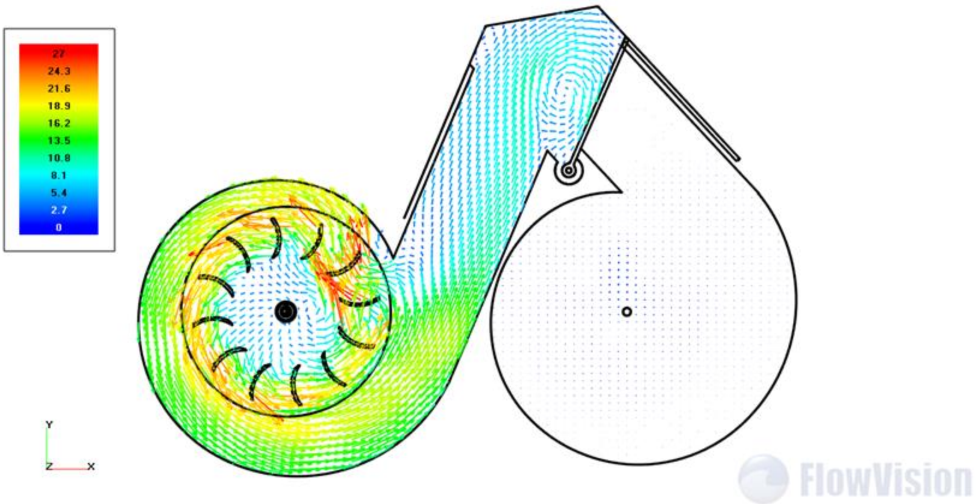

Figure 3. Velocity vectors, m/s, in the transverse plane of the fan unit. Shown at a pressure of 300 Pa at the outlet, fan 1 is in operation

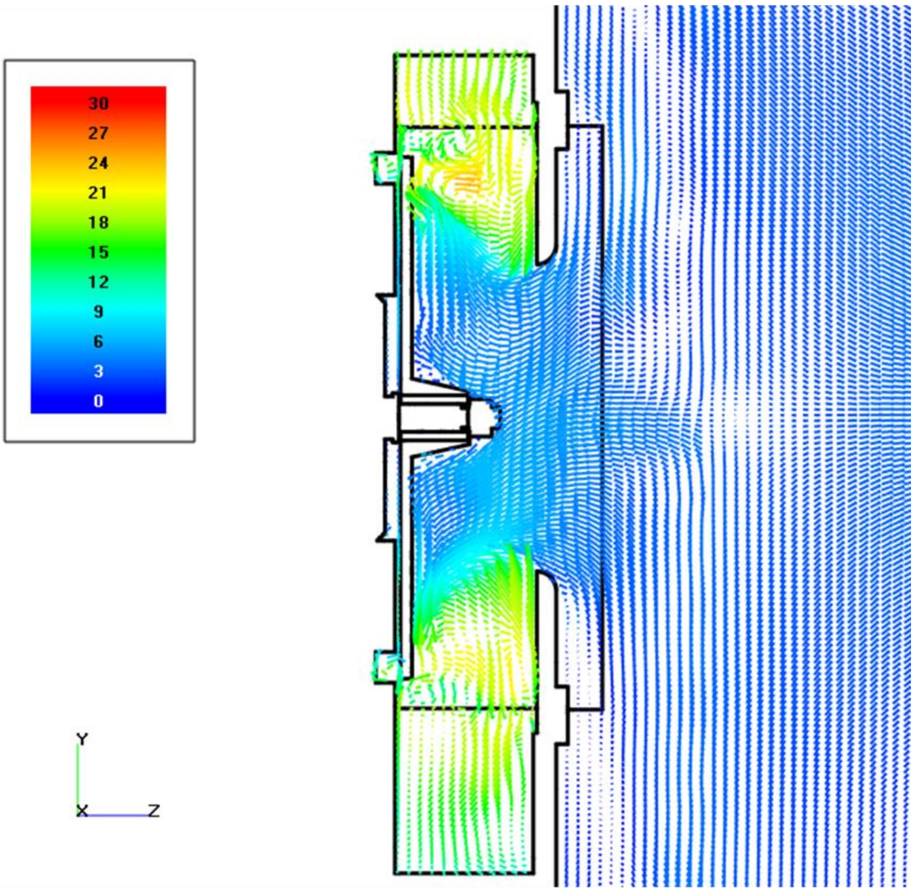

Figure 4. Velocity vectors, m/s, in the longitudinal meridional plane of the operating fan. Shown at a pressure of 300 Pa at the outlet, fan 1 is in operation

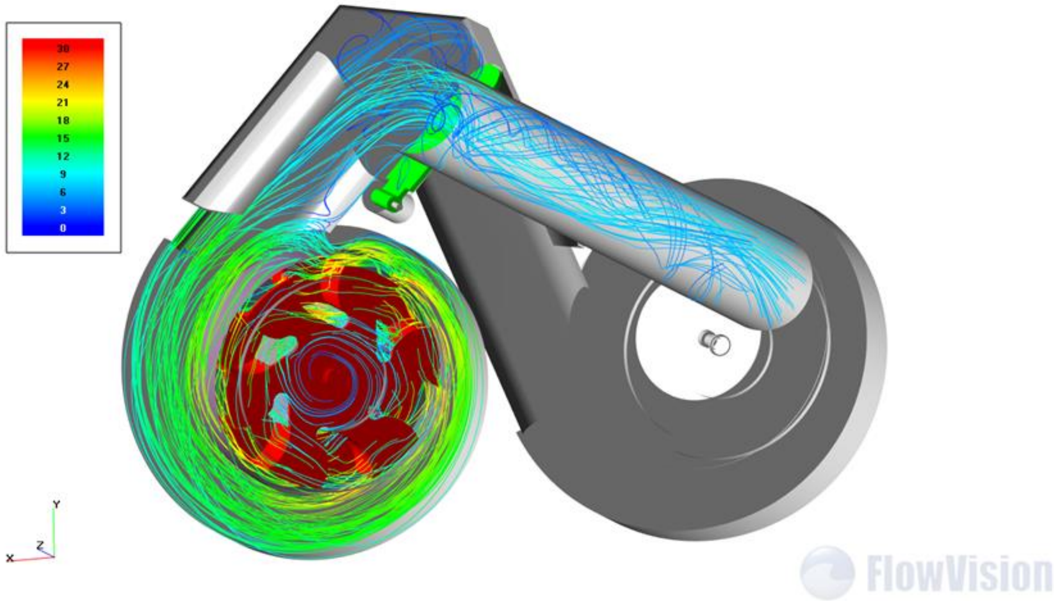

Figure 5. Streamlines (colors indicate velocity level, m/s). Shown at a pressure of 300 Pa at the outlet, fan 1 is in operation

3.2 Pressure Field

Static pressure distribution in the transverse plane is presented in Fig. 6.

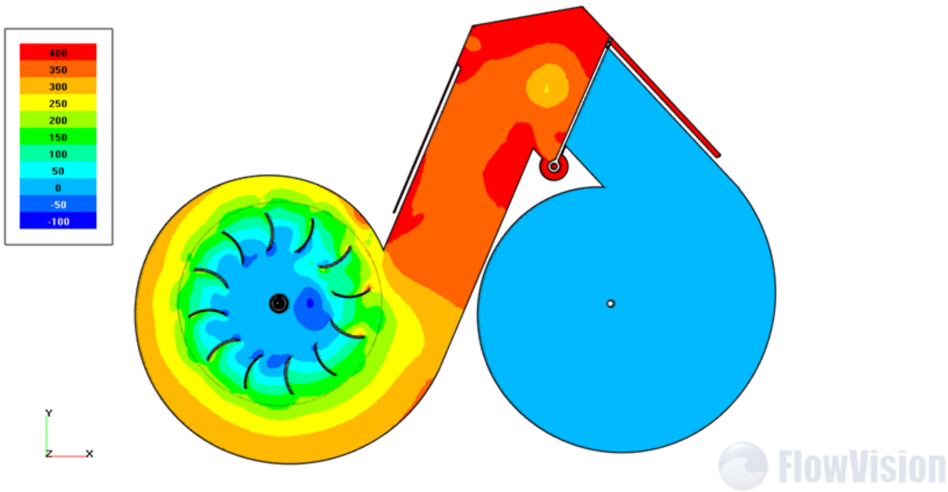

Figure 6. Static pressure, Pa, in the transverse plane of the fan unit. Shown at a pressure of 300 Pa at the outlet, fan 1 is in operation

The obtained data indicate a significant non-uniformity of pressure and velocity distribution in the centrifugal impeller channels by steps of the blade cascade, suggesting dominance of tonal BPF sound generation.

4. CONCLUSIONS

The paper offers a formulation and justification of the boundary problem of centrifugal fan aeroacoustics in relation to enthalpy pulsations with boundary conditions in respect of the Fourier transform of enthalpy disturbance and with impedance walls in the modified Crow-Artamonov model.

The non-stationary flow in a centrifugal fan was calculated, and it was found that the mechanism of BPF tonal sound generation dominates.

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