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# HARMONY SOFTWARE FOR LOW NOISE DESIGN OF CENTRIFUGAL VENTILATORS OF TRANSPORT MACHINES

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## ABSTRACT
There is a close link between the pressure pulsation in the centrifugal ventilator working cavity and its noise level. The noise and pressure pulsation spectra of centrifugal ventilators are represented by a broadband noise to which clearly discernible discrete components are superimposed. Generally it is blade-passing frequencies (BPF). The level of these tone components mainly determines the ventilator noise characteristics, as the tonal noise is much uncomfortable for men's ear. Studies of centrifugal machines show that, as a rule, the discrete BPF component and its harmonics have maximum spectral amplitudes of pressure pulsations and noise under the rated operation mode. The current version of Harmony software package is developed for numerical modeling of pressure pulsations at blade passing frequencies.

## 1 - INTRODUCTION
BPF pressure pulsation tones are generated at multiple frequencies of the rotation speed. They are defined by the formula

$$f = kzf_r \tag{1}$$

where

- $f_r$ - Frequency of rotation, Hz
- $z$ - Number of impeller blades (number of "bars")
- $k$ - Harmonic order.

Experience shows that the amplitudes of BPF pressure pulsations depend on several design factors − form, number and disposition of blades of the impeller, configuration of the volute, radial gap between the impeller and volute cutwater. The amplification of pressure pulsations can happen due to matching of frequencies of oscillations with acoustic resonance frequencies of both the outlet duct and the volute i.e. the working cavity. In ventilators the length of acoustic waves can be comparable to the size of the casing and outlet volume. Therefore the variation of rotation speed, number of impeller blades can substantially modify amplitudes of pressure pulsations due to the resonance into the ventilator cavity.
Some works were published [1, 2] where different methods of prediction of BPF pressure pulsation have been proposed. Developed during 5 years by CETIM and IRI Ltd. Harmony software package [3−6] became a useful tool for designers and researchers in the field of vibration and noise problem in centrifugal pumps and ventilators.

## 2 - MAIN EQUATIONS
The numerical procedure is based on a representation of unsteady compressible fluid velocity **V** as a combination of vortex and acoustic modes,

$$\mathbf{V} = \mathbf{U} + \nabla\varphi = \mathbf{U} + \mathbf{V_a} \tag{4}$$

where:

- $\mathbf{U}$ - Velocity of transitional and rotational motion of absolutely incompressible liquid (vortex mode),
- $\mathbf{V_a}$ - Velocity of pure deformation (acoustic mode),
- $\varphi$ - Acoustic potential.

By assuming the subsonic isentropic pump flow and neglecting viscous effects, the last formula leads to acoustic-vortex equation relative to enthalpy oscillations. The latter is proportional to pressure pulsation that is the sum of acoustic (waves) and vortex ("pseudo sound") modes:

$$\Lambda^2 \frac{\partial^2 h}{\partial \tau^2} - \nabla^2 h = s \tag{5}$$

where:

- $\Lambda = \dfrac{u_2 z_1}{2\pi a}$ - Non-dimensional criterion,
- $h = (H + H_{\mathbf{a}}) \approx \dfrac{p - p_0}{\rho u_2^2}$ - Enthalpy (pressure) oscillation;
- $H$ - Pressure oscillation in incompressible liquid (vortex mode or "pseudo sound"),
- $H_{\mathbf{a}}$ - Pressure oscillation due to compressibility (acoustic mode or acoustic wave),
- $s$ - Disturbance function $-$ oscillatory part of $S$ function,
- $S = \nabla \left[ \nabla \left( \frac{U^2}{2} \right) - \nabla \times (\nabla \times \mathbf{U}) \right]$
- $\mathbf{U}$ - Vortex mode velocity field,
- $\tau = \dfrac{t}{T}$ - Non-dimensional time,
- $T$ - Main period of blade passing oscillation,
- $p$ - Pressure,
- $p_0$ - Mean pressure,
- $\rho$ - Mean density,
- $a$ - Mean sound velocity,
- $u_2$ - Impeller blade tip velocity,
- $z_1$ - Number of impeller blades.

## 3 - METHOD

The problem being solved is two-dimensional with using vorticity and streamline function.
The task of pressure oscillation field determination splits into three main steps. Having the pump geometry defined, the first step with impeller flow analysis gives an unsteady boundary condition for the solution of vortex mode equations. At the second step the unsteady direct procedure provides a converging oscillatory solution for the incompressible liquid flow (so called "pseudo-sound" oscillations). In the third step the unsteady pressure field is obtained as a function of time, satisfying the complex specific impedance for acoustic mode.

## 4 - APPLICATION DOMAIN

The code is applicable to centrifugal pumps or ventilators with specific speed $n_s$<150 ( $n_s = 193.3\omega \sqrt{Q} H^{-3/4}$, SI units are applied) under the normal operation mode. Normal operation mode guarantees the accuracy of computation 1–3 dB based on the following conditions:

- Subsonic flow.
- Homogeneous fluid.
- No cavitation (before the first critical mode).

- Delivery range is 0.8 – 1.3 of the rated value.

## 5 - COMPUTATIONAL RESULTS

Pulsation of pressure essentially depends on acoustic characteristics of the working cavity of centrifugal machine. Acoustical behavior is defined by the main non-dimensional criterion $\Lambda$ of equation (5). It represents relation between impeller tip radius and acoustic wavelength of the main BPF tone. Fig. 1 shows pulsation field for low frequency air pump model ( $\Lambda = 0.04$). The characteristic feature of unsteady pressure in the volute is lower pressure zones linking with blade exit edges and rotating with impeller. They form pseudo-sound oscillations near the impeller exit. Acoustic perturbations dominate at the volute exit, in conical diffuser.

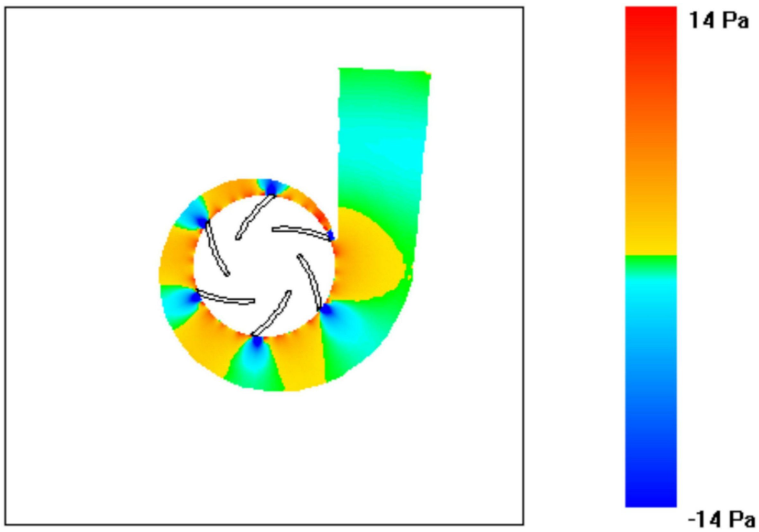


**Figure 1:** Pulsation of pressure in the air pump model.

Computation of ventilator with relatively high frequency ($\Lambda$=0.37) is presented on Fig. 2. In this case pseudo-sound oscillations take place closer to the impeller exit while acoustic waves occupy the most part of the volute casing.

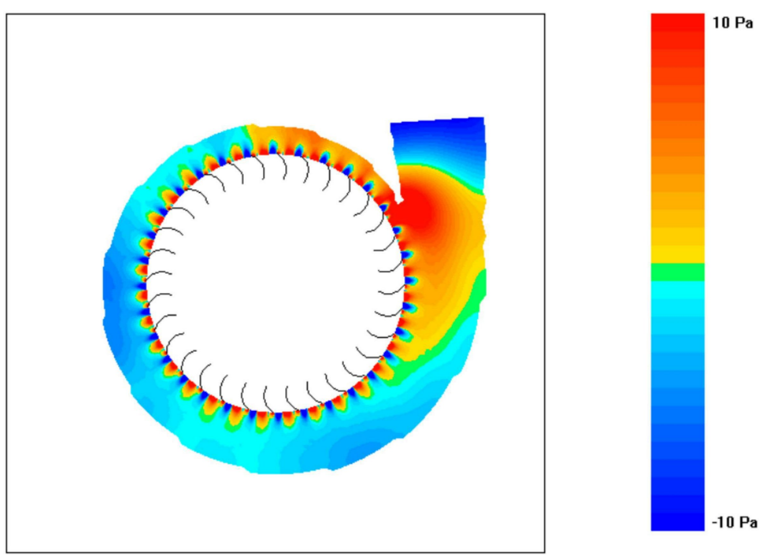


**Figure 2:** Pulsation of pressure in industrial ventilator.

Another high frequency case with $\Lambda$= 0.48 is shown on Fig. 3. The transverse dimension of the volute is comparable with impeller diameter. In that case transverse mode of acoustical pressure pulsation takes place in the volute as well.

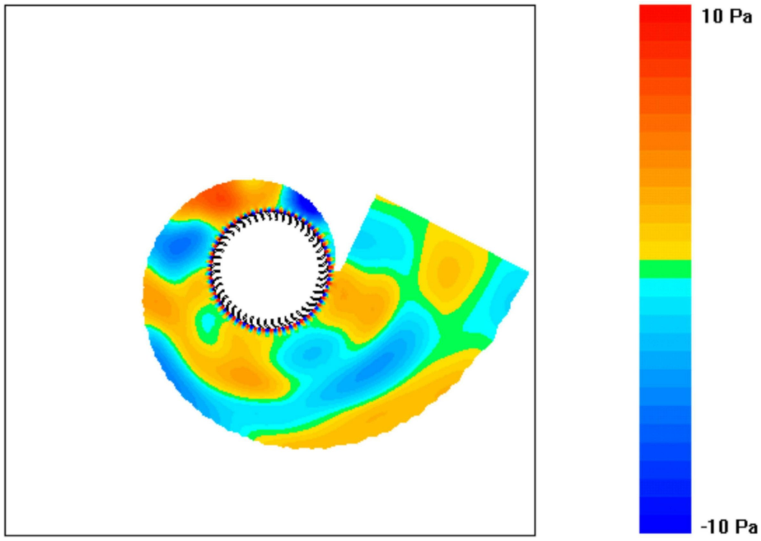


**Figure 3:** Pulsation of pressure in the ventilator model.

Computational study of pressure pulsation in the air pump model with bladed diffuser (Fig. 4) shows a great effect of relation of numbers of blades on pressure pulsation field. This phenomenon takes place due to interaction of acoustic waves outgoing from different diffuser channels. These examples enable to conclude that only with taking into account all essential acoustical characteristics of a ventilator it is possible to select the optimal design and number of blades.

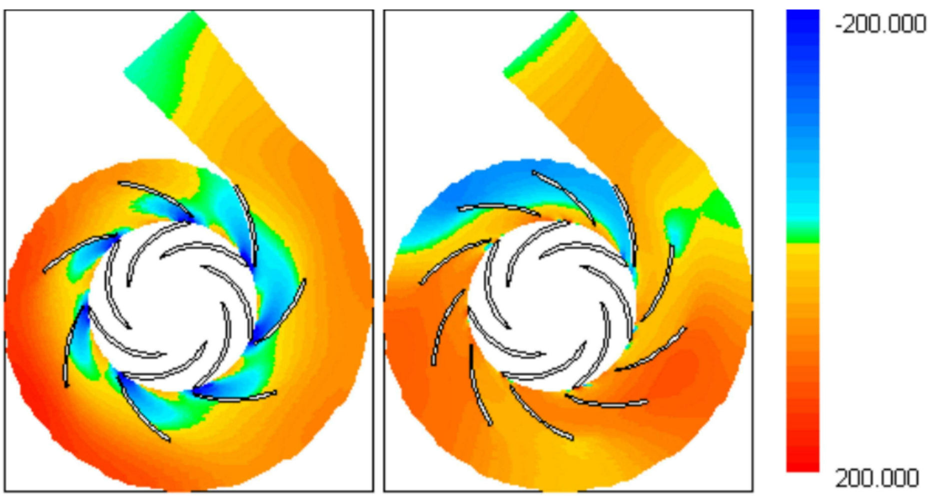


**Figure 4:** Pulsation of pressure in air pump model with 7 and 10 diffuser blades.

More detailed information about Harmony software is available on the Internet site http://www.pump-harmony.ru.

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