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Efficient 2D numerical method for reduction of BPF pressure pulsation and noise by optimization design of centrifugal pumps and ventilators

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Abstract

The efficient method for numerical modeling of pressure pulsation on blade passing frequencies (BPF) is developed. It is based on splitting the equations of compressible fluid dynamics into two modes - vortex and acoustic. As a result, the whole processor time for both modes of oscillations is reduced. It enables designers and researchers rapidly obtain computational results and to optimize geometry with the main goal of reduction of pressure pulsation as the source of hydraulic noise and vibration of the pump. Geometry may include arbitrary impeller blade profiles and arbitrary volute-vaned-diffuser geometry with one outlet pipe. Recently the computational procedure is built on 2D numerical methods It has been used for optimization of different centrifugal machines, including slurry, condensate, and diagonal pumps with simple volute, feed and high-speed rocket pumps with vaned diffuser, centrifugal ventilator for a car ventilation and for house heater, for high-speed centrifugal grinder and for lawn-mover. Number of impeller blades was from 2 to 48 and Helmholtz number changes from 0.01 to 0.5. A few results regarding high-speed centrifugal pump stage with vaned diffuser and impeller with splitters is outlined.

1. Introduction

The subject of this paper is prediction of blade-passing-frequency (BPF) pressure pulsation in centrifugal pumps. The cause of BPF pressure pulsation is hydrodynamic interaction between the impeller flow and the volute casing (or vaned diffuser). BPF pressure pulsation tones are generated at multiple frequencies of the rotation speed. They are defined by the formula

$$f = k\,z\,f_r \qquad (1)$$

Where

f_r - Frequency of rotation, Hz

z - Number of impeller blades

k - Harmonic order.

Amplitudes of BPF pressure pulsation depend on several design factors - shape, number and disposition of blades of the impeller and diffuser vanes, configuration of the volute, radial gap between the impeller and volute tongue or diffuser vanes, impedance boundary conditions imposed by a circuit. Pressure pulsations can amplify due to matching of frequencies of oscillations with acoustic resonance frequencies of both the outlet duct and the volute. In high-speed pumps and 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 resonance phenomena.

Some works were published, in which the methods of prediction of pressure pulsations were developed by a direct computation of non-stationary two-dimensional flow in a centrifugal impeller and volute with solution of averaged Navier-Stokes equations and k-ε model of turbulence [1, 2]. Other approaches [3, 4] used solutions of hydrodynamic equations accompanied with laser anemometric measurements. Analysis of rotor-stator interaction in a pump with vaned diffuser using 2D discrete vortex method is outlined recently as well [5].

In the flow part of the pump casing, there are two modes even two zones of perturbations, which differ in the physical nature of oscillations and equations describing their behavior. The first mode is pseudo-sound oscillations caused by unsteady vortex motion of liquid as an incompressible fluid. These oscillations occur only near the impeller. The velocity of propagation of the disturbances is equal to the main flow velocity; they are described by the non-linear equations of parabolic- elliptical type. The second mode is the acoustic oscillations, which extend throughout the entire zone of flow with the speed of sound; they are governed by a linear hyperbolic equation. This entire set of conditions leads to the fact that modern 3D N-S CFD codes for compressible fluid prove to be too ineffective for solving the problem of optimizing the design regarding pressure pulsation and noise.

For the solution of this problem the method is proposed, which is based on splitting the equations of compressible fluid dynamics into two modes - vortex and acoustic. In this case non-linear equations for unsteady vortex motion of an incompressible liquid are solved with a bigger time step. Wave equation relative to the pressure pulsation that takes into account acoustic impedances on the boundaries of computational domain is solved by an explicit method. As a result, the whole processor time for both modes of oscillations is reduced. Additional information can be retrieved from publications [5 – 10] and the web-site www.pump-harmony.ru.

2. Main Acoustic - Vortex Equation

The velocity in some points of the fluid can be determined as the sum of a velocity **U** transitional and rotary motion of a fluid as absolutely incompressible and velocity of a pure strain $\mathbf{V}_a$ (Cauchy - Helmholtz theorem).

Velocity $\mathbf{V}_a$ represents the acoustic perturbations enabled by the compressibility of fluid.

Let us enter scalar function - acoustic potential φ. Then the acoustic velocity

$$V_a = \nabla\varphi \tag{2}$$

Thus for the fluid velocity the following expression is obtained:

$$\mathbf{V} = \mathbf{U} + \nabla\varphi = \mathbf{U} + \mathbf{V_a} \tag{3}$$

The velocity of incompressible fluid flow determines the vorticity. Let us substitute now the relation (3) in the main Euler equations of a compressible fluid. For the space scale and the characteristic velocity we shall take the impeller radius R2 and impeller tip velocity u2. Then the dimensionless quantities of radius (r), velocity (U), time (t) and enthalpy (i) will be as follows:

$$\widetilde{r} = \frac{r}{R_2}; \widetilde{\mathbf{U}} = \frac{\mathbf{U}}{u_2}; \tau = \frac{t}{(2\pi R_2)/(u_2 z_1)}; \widetilde{i} = \frac{i}{u_2^2} \tag{4}$$

Here z_1 – number of main impeller blades. Under assumption of subsonic isentropic flow of a non-viscous fluid one obtains an equation of pressure oscillations (due to acoustical and vortex motion)

$$\Lambda^2 \frac{\partial^2 h}{\partial \tau^2} - \widetilde{\Delta} h = -\widetilde{\Delta} g \tag{5}$$

where Λ is Helmholtz similarity criterion of the given problem. It is simple to show that the parameter Λ represents relation of the impeller tip radius R_2 to the main BPF wavelength λ.

The amplitude of pressure pulsation in an hydraulic machine is by an order of magnitude lower than the mean undisturbed pressure, thus for enthalpy oscillations (as a sum of vortex and acoustic perturbations) it is possible to write approximately

$$h = \widetilde{i} - \widetilde{i}_0 \approx \frac{(P - P_0)}{\rho_0 u_2^2} = \frac{P'}{\rho_0 u_2^2} \tag{6}$$

where P – pressure of compressible fluid, i_0, P_0 and ρ_0 – mean enthalpy, pressure and density. Similarly for oscillations of the function g we obtain pressure pulsation (P_v-P_0) in "vortex-mode motion"

$$g \approx \frac{(P_v - P_0)}{\rho_0 u_2^2} = \frac{P_v'}{\rho_0 u_2^2} \tag{7}$$

produced by non-stationary vortex motion of the medium as an incompressible – so called "pseudo-sound".

Right part of the wave equation (5) is determined from (7, 8) -- unsteady velocity field of the vortex mode (incompressible fluid) flow.

$$-\Delta P_v = \nabla[\nabla(\frac{U^2}{2}) - \nabla \times (\nabla \times \mathbf{U})] \tag{8}$$

By using a local specific acoustic impedance Z (complex value), the boundary condition at the impeller outlet and at the pump casing exit section can be put in the form (9)

$$\frac{\partial\,(h_k - g_k)}{\partial\, n} = -\frac{\Lambda k}{Z_k}\frac{\partial\,(h_k - g_k)}{\partial\, \tau} \tag{9}$$

Where k is a number of BPF harmonic, n – normal direction to the boundary.

Volute casing walls are assumed rigid. Nevertheless, there is a possibility to define a local specific impedance of the pump housing wall that will be interesting to study the effect of, for example, damping coating.

3. Solution Method

The problem of pressure oscillation field determination splits into three main steps. The first one is the incompressible liquid flow analysis in the impeller to obtain unsteady boundary condition of the vortex mode flow using Discrete Vortex Method (DVM). This boundary condition is represented in the form of rotating velocity distribution "attached " to the impeller exit diameter The second step is the unsteady vortex mode flow computation into the pump casing with consequential determination of the disturbance function (7 – 8), and the third is the solution of the inhomogeneous wave equation (5) relatively to pressure oscillations. The last satisfies the complex specific impedance for acoustic mode and unsteady boundary condition for the pseudo-sound oscillations by relations (9). Recently the computational procedure is built on 2D numerical methods.

3.1 Application Domain

Generally the code is applicable to centrifugal pumps or ventilators with specific speed n_s<150 ($n_s = 193.3\omega\sqrt{Q}H^{-3/4}$ if SI units are applied), n_s<2120 using rpm, US gpm, ft under the designed operation mode but recently there is an encouraging results in n_s level 250 – 350 (SI units).
Geometry may include arbitrary impeller blade profiles and arbitrary volute-vaned-diffuser geometry with one outlet pipe. The method has been used for optimization of different centrifugal machines, including slurry, condensate, and diagonal pumps with simple volute, feed and high-speed rocket pumps with vaned diffuser, centrifugal ventilator for a car ventilation and for house heating, for high-speed centrifugal grinder and for lawn-mover. Number of impeller blades was from 2 to 48 and Helmholtz number changes from $\Lambda = 0.01$ to 0.5.

3.2 Software Package and Computation Process

Numerical algorithms are realized in five main modules written in C and C++. It provide an environment to work with 3 × 3 different cases simultaneously. A built-in interface for the determination of impedance boundary conditions gives a possibility to take into account the connected circuit. Once the computation procedure, that goes consequently through three main steps, finishes, it becomes possible to obtain the fluctuating pressure map in the casing at a selected time point as well as the pressure time history at any point within the pump casing with corresponding RMS value and spectrum data.

5. Computational Results

Computational results of this numerical procedure give a possibility to optimize geometry with the main goal of reduction of pressure pulsation as the source of hydraulic noise and vibration of the pump.
A few results regarding high-speed centrifugal pump stage with vaned diffuser and impeller with splitters are outlined in this section. In the Fig. 1 is presented a snap-shop of distribution of free vortexes emanating from blades. Solid circles indicate positive vorticity (contra-clock-wise

swirl). The resulting distribution of impeller flow parameters at the impeller exit along one blade channel between long blades is shown on the right of Fig. 1. One can see non-uniform dimensionless absolute radial Ur and tangential Ut velocity components and static pressure distribution that cause BPF pressure pulsation.

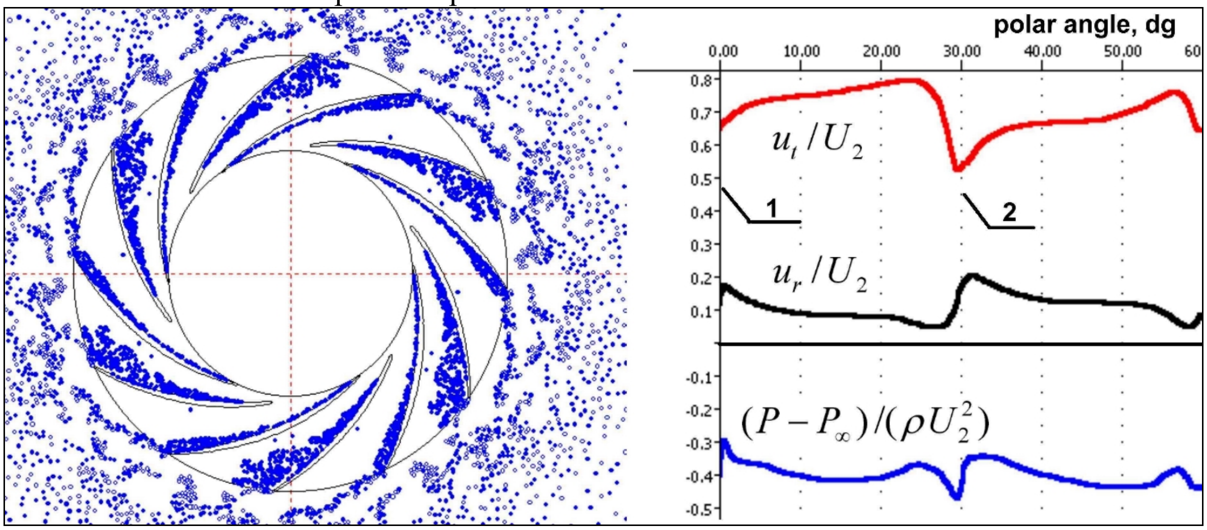

Fig. 1: Distribution of free discrete vortexes (left) and Pressure and velocity distribution along the impeller exit radius (right); 1 – pressure side of the long blade; 2 – pressure side of the short blade (rotation goes to the right), by the courtesy of Snecma Moteurs.

In Fig.2 results of the last two steps of computational procedure are presented. On the left one can see an instantaneous distribution of vorticity. This shows that negative vorticity perturbations penetrate and rapidly attenuate into diffuser channels so that the pressure pulsation downstream of diffuser (on the right) is represented by pure acoustical oscillations.

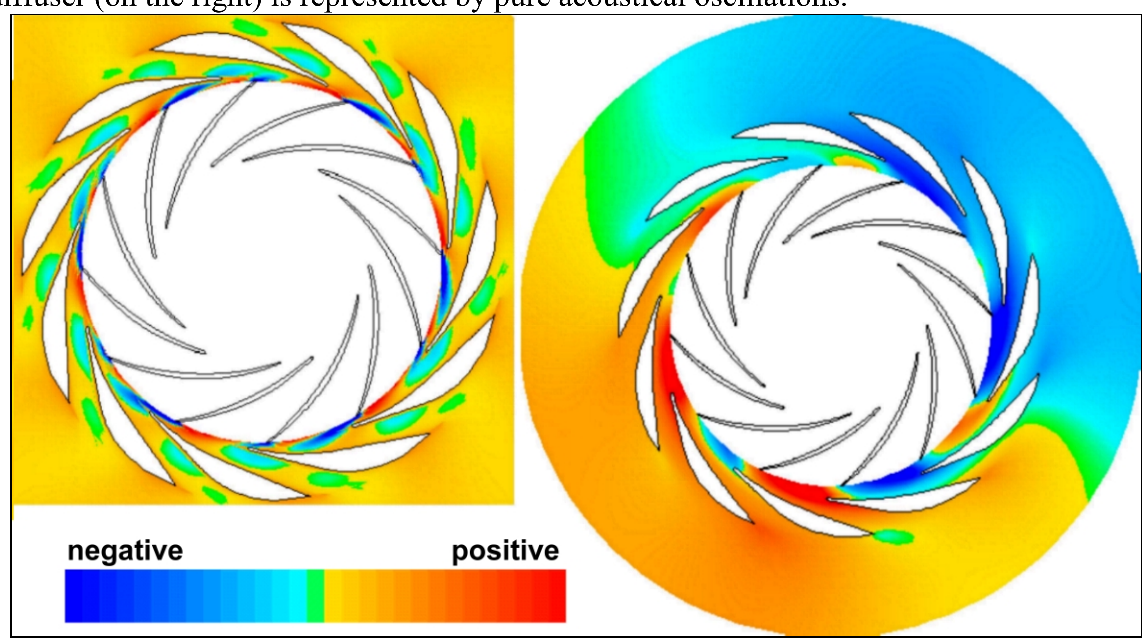

Fig. 2: Vorticity distribution snap-shot (left) and pressure pulsation field (right), by the courtesy of Snecma Moteurs.

It has been found that acoustical oscillations exist in the form of the backward wave (rotating contra impeller rotation). Rotation frequency of this wave coincides with the blade passing frequency.

5. Conclusion

Method of numerical modeling of BPF pressure pulsation shows its efficiency in optimizing centrifugal pump and ventilators' designs including high-speed pumps with vaned diffuser and double-row impeller. The main advantage of the method is based on splitting the equations of compressible fluid dynamics into two modes - vortex and acoustic and 3-steps computational procedure.

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