Study of BPF pressure pulsations reduction in centrifugal bladed machines using splitters

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Abstract. Presence of intensive pressure pulsations and noise on blade passing frequencies is a characteristic feature of all types of centrifugal bladed machines including pumps. Under definite conditions pressure pulsations can reduce the pump reliability and life cycle. High level of tonal noise can be inappropriate due to ecological requirements. The paper is devoted to the study of application of splitters for reduction of pressure pulsations in centrifugal pumps. 2D and 3D numerical methods are used in the study. Initial computational tests were performed with a two-dimensional formulation for a centrifugal pump with a simple volute using 2D discreet vortex method for the impeller flow and 2D acoustic-vortex approach for pressure pulsations prediction. 3D unsteady flow computations of an air model of the centrifugal pump are undertaken with the impeller having splitters. Calculations of pressure pulsations show that the use of additional blades reduces the amplitude of the first harmonic of blade passing frequency by more than three times compared to the design without splitters. It correlates with flow parameters distribution along the impeller outlet diameter that has less nonuniformity and a more balanced vorticity profile. Application of splitters reduces the amplitude of pressure pulsations by a factor of 3.

1. Introduction

Currently, more importance is attached to the study of problems of increasing the reliability and service life of centrifugal pumps that is the main source of noise in hydraulic systems, a main source of hydrodynamic vibration of the feed system in modern turbopump units. The first mentions of this problem appear in the 60s of the last century in connection with the destruction of large industrial pumps [1,2]. Hydrodynamic vibration is excited by pressure pulsations arising in the pump flow cavity due to hydrodynamic sources of different nature [3] including vortex formation, flow recirculation, cavitation, blade step nonuniformity of flow parameters at the outlet of the centrifugal impeller. The latter factor causes the generation of pressure pulsations at the so-called blade passing frequency (BPF) and its higher harmonics and combination frequencies that dominate in the spectra. These pressure fluctuations are an integral part of the centrifugal pump working process [4]. In centrifugal pumps, they have a high amplitude due to the peculiarities of the formation of step inhomogeneity of the flow in the centrifugal blade cascade. Detailed studies of the flow parameters in absolute and relative motion at the outlet of the impeller of centrifugal pumps, fans [5], compressors [6] confirm that the flow in the blade channel and at the outlet of the centrifugal impeller has significant nonuniformity due to the peculiarities of the formation of the boundary layer in the blade channel and
secondary flows [7,8]. It is well known that the physical nature of pressure pulsations in a centrifugal pump is a total manifestation of pseudosound and acoustic perturbations. The pseudosound or vortex mode [9] decays rapidly downstream of the rotor, leaving only the acoustic mode of pressure fluctuations in the pressure pipeline. The dual nature of pressure pulsations in centrifugal pumps can be considered by applying decomposition [10] with the introduction of acoustic-vortex equations. A particularly sharp change in the flow parameters occurs near the leading edges of the guide vanes and in the tongue of the volute; therefore, such a great attention is paid to the choice of the optimal gap between the rotor and the guide vane or the tongue of the volute [11], this is confirmed by experimental data from different sources on the amplitude of pressure pulsations and dynamic stresses in centrifugal pumps [12]. Computational fluid dynamics (CFD) modeling currently becomes a routine approach for studies in the problems of pressure pulsations and acoustics [13-15]. Studies of pressure pulsations in the working cavity of the pump provides information on unsteady loads [16-19] and the generation of vibration and noise in the pump. When measuring the dynamic stresses at the input edges of the guide vane [20] of a centrifugal pump by strain gauges, it was found that in the flow rate range 0.6-1.0 of the best efficiency point (BEP), the values of dynamic stresses are directly proportional to the amplitudes of pressure pulsations.

In the pump design the splitter blades widely used to improve cavitation and energetic parameters [21] while there are a few results regarding its effect on the pressure pulsations in turbomachines [22,23]. It is evident that application of splitters reduces nonuniformity of flow parameters at the centrifugal impeller tip radius and can reduce BPF pressure pulsations and noise. Thus it is useful to undertake a research of its optimal shape and position in the main blade channel.

2. Method and calculations

In the isentropic flow the enthalpy, pressure and density increments are related thermodynamically (1):

\[ di = \frac{dP}{\rho}, \quad dP = a^2 d\rho \quad (1) \]

where \( a \) is the sound velocity in the working fluid.

For a prediction of pressure pulsation and noise in the bladed machine, the mathematical model is based on a representation of the fluctuating flow velocity field \( \mathbf{V} \) as a combination of vortex and acoustic modes (2):

\[ \mathbf{V} = \mathbf{U} + \nabla \phi = \mathbf{U} + \mathbf{V}_a \quad (2) \]

where \( \mathbf{U} \) - Velocity of transitional and rotational motion of an incompressible fluid (vortex mode), \( \mathbf{V}_a \) - Velocity of pure deformation (the acoustic mode), \( \phi \) - Acoustic potential.

Taking assumptions of the subsonic flow and small acoustic velocities and, after linearization of Euler equations, one can obtain the following acoustic-vortex equation:

\[ \frac{1}{a^2} \frac{d^2i}{dt^2} - \Delta i = -\Delta j \quad (3) \]

where the source function in the right side of equation (3) can be expressed in terms of field of velocities of incompressible vortex-mode flow from equation (4):

\[ -\Delta j = \nabla (\nabla \cdot U^2) - \mathbf{U} \times (\nabla \times \mathbf{U}) \quad (4) \]

The vortex mode velocity field \( \mathbf{U} \) is determined from the solution of unsteady equations of incompressible fluid. The mathematical model of incompressible liquid flow bases on Navier-Stokes and continuity equation (5, 6):
\[
\frac{\partial \mathbf{U}}{\partial t} + \nabla (\mathbf{U} \otimes \mathbf{U}) = -\frac{\nabla P}{\rho} + \frac{1}{\rho} \nabla ((\mu + \mu_t)(\nabla \mathbf{U} + (\nabla \mathbf{U})^T))
\] (5)

\[
\nabla \cdot \mathbf{U} = 0
\] (6)

The \( k - \varepsilon \) model of turbulence is used to determine the turbulent viscosity by relation (7):

\[
\mu_t = 0.09 \cdot \rho \cdot \frac{k^2}{\varepsilon}
\] (7)

In 2D computations vortex-mode flow parameters in the volute computed using streamline function-vorticity equations while impeller flow parameters computed with the discreet vortex method [10].

3. Results and discussion

Firstly, the influence of intermediary shorten blades on the outlet impeller blade cascade flow parameter distribution and pressure pulsations is computationally studied on the base of a centrifugal pump with main parameters outlined in Table 1.

| Parameter                  | Value | Unit    |
|----------------------------|-------|---------|
| RPM                        | 1370  | Round/min |
| Flow rate                  | 34.7  | l/s     |
| Impeller inlet diameter    | 70    | mm      |
| Inlet blade angle          | 5     | deg     |
| Impeller tip diameter      | 265   | mm      |
| Outlet blade angle         | 25    | deg     |
| Simple volute casing       | -     | -       |
| Radial gap                 | 29.7  | %       |
| Fluid                      | water | -       |

All computations are completed for the pump with simple volute casing. “Infinite – pipe” impedance condition is defined at the pump exit. Then 3D CFD analysis is undertaken for an air model of the centrifugal pump with guide vane channels, spiral casing and the impeller having seven long blades and seven splitters.

3.1. 2D computations

Six options of the impeller geometry are shown in figure 1. They include 5 main blades (impeller 3), 5 main and 5 shorten blades, positioned axial-symmetrically at the impeller exit (impeller 4), the same number of long and short blades but positioned non-axial-symmetrically at the impeller exit (impeller 5-7 and 8). The displacement of splitters made so that the leading edge of the splitter is placed into the low-energy zone, and the splitter locates closer to the suction side of the main blade channel. Profile 3 of the long blade was unchanged for all cases computed. Analysis of flow parameters distribution is made by discrete-vortex impeller flow module of the software package for 2D pressure pulsation modeling [10].
3.2. 3D computations and experimental facility

Experimental air pump model is developing currently for validation the 3D acoustic-vortex method. There are no available experimental data so far but computational data regarding pressure pulsation signals and spectra are obtained in the points located in the air pump facility for further comparison. Main parameters of the facility outlined in table 2.

| Parameter               | Value | Unit |
|-------------------------|-------|------|
| Rotation speed          | 500   | rad/s|
| Mass flow rate          | 0.3   | kg/s |
| Main impeller blades    | 7     | -    |
| Diffuser blades         | 12    | -    |
| Radial gap              | 6.5   | %    |

Six high-frequency pressure pulsations sensors installed on the rear part of the stator. They located in three diffuser channels at the inlet and outlet section by radii 220 and 270 mm as shown in figure 2.

3.3. 2D computational results

Distribution of radial and tangential velocity components along the exit diameter between two long blades show a considerable non-uniformity of flow that reflects in the vorticity distribution (figure 3). It is evident that splitter blades give an additional peak in the vorticity distribution. Pressure side of the blade channel is on the left, and rotation goes to the right in figure 3.
Figure 3. Distribution of vorticity along the main blade channel step with options: only long blades (3) and with the splitter position 4, 5, 6, 7, 8 (figure 1).

Favorable parameters along the blade span are obtained for the impeller option 7 where two vorticity peaks are approximately equal and give more balanced flow parameters and vorticity distribution at the impeller exit. Impeller options with intermediary blades increase the tangential velocity that gives a rise of impeller head by 20%.

This result confirms that the application of splitters can reduce flow parameters nonuniformity at the impeller tip radius and the main BPF harmonic amplitude of pressure pulsations and noise although the second BPF harmonic amplitude increases. For the impeller design 7 even near the impeller outlet the first BPF harmonic does not exceed 3000 Pa, whilst for the option 3, without short blades, this amplitude is higher than 10,000 Pa.

This result shows the importance of providing specific impeller geometry to achieve the desired pressure pulsation spectra. An increase of the second BPF harmonic amplitude is not essential as its level is considerably lower. Thus, the total level of pressure pulsation drops.

Computed pressure pulsation signals at the pump exit are shown in figure 4. There are shown three periods of main blade passing frequency. The amplitude is reduced by density and impeller tip velocity squared. One can see an essential change of amplitude and shape of signals due to presence of the second BPF harmonic. In field measurements the reduced amplitude of the first BPF harmonic of pressure pulsations at the pump exit was around 0.0015 that is well matched with 2D computation data (figure 4).

All impeller geometry options with intermediary shorten blades have an advantage in the impeller head and in the total amplitude of pressure pulsation. Regarding the reduction of pressure pulsation impeller #7 must be indicated as a good perspective to reduce pressure pulsations on the main BPF tone and total amplitude of pressure pulsation into the pump working cavity and in the outlet pipe. It gives a reduction of total amplitude of pressure pulsations by a factor of 3.
3.4. 3D computational results

Computational results with splitters can be compared with the data for the impeller design with long blades published in [24], where one can find details of the computational study. The reference (ambient) pressure is set 101000 Pa. The initial mesh was set with 10 mm cubic cells. After refinement of the mesh in the impeller, the outlet pressure changed by 3 percent so the refined mesh of 125000 cells was taken for the final computation. The time the step is set by CFL condition and equals approximately $1.8 \times 10^{-5}$ s. Boundary condition at the pump inlet is stagnation (ambient pressure). At the pump outlet the mass flow rate is set (table 2). Wall parietal functions are used on the pump and impeller surfaces.

Below in figure 5 the instantaneous static pressure field is outlined for the impeller with splitters.

**Figure 5.** Instantaneous static pressure field for the pump with splitters, Pa.

**Figure 6.** Sector for the velocity components distributions.

Distributions of instantaneous components of the absolute velocity are taken in the sector shown in figure 6.

Radial and tangential velocities for the impeller option with splitters have less non-uniformity as it is outlined in figures 7-8.
Figure 7. Radial absolute velocity with long blades (green) and splitters (red).

Point P1 is in the guide vane channel opposite the spiral casing tongue. It is taken for a comparison of results with long blades and with splitters (figure 9-10).

Figure 8. Tangential absolute velocity with long blades (green) and splitters (red).

Figure 9. Signal and pressure pulsation spectrum with long blades [24] at point P1.

Figure 10. Signal and pressure pulsation spectrum with splitters (current study) at point P1.

The maximum amplitude level is fixed at the first harmonic of the blade passing frequency (BPF) for the long blades – above 180 Pa. For the case with splitters, the main BPF amplitude is 3 times less and the second BPF harmonic has approximately the same amplitude.

4. Conclusion

3D CFD analysis of unsteady flow in the air centrifugal pump model having the impeller with seven main blades and splitters confirms the intensive BPF pressure pulsations in guide vane channels. For the case with splitters, the main BPF amplitude is three times less than for the impeller with single long blades. The second BPF harmonic slightly increases and has approximately the same amplitude as the first harmonic. This result proves the main findings using 2D computations of the centrifugal water pump with simple volute where the impeller with 5 long blades and splitters reduce the main BPF amplitude by a factor of three comparing the design with single long blades and it has been proved by field measurements. Pressure pulsations correlate with flow parameters distribution along the impeller outlet diameter that has less nonuniformity and a more balanced vorticity profile in 2D computational results. Favorable parameters along the blade channel outlet are obtained for the impeller option 7 where two vorticity peaks are approximately equal and give more balanced flow
parameters and vorticity distribution at the impeller exit. Impeller options with intermediary blades increase the tangential velocity that gives a rise of impeller head by 20%. 3D computations also show that application of splitters reduces non-uniformity of flow parameters at the impeller tip diameter.

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