Assessment of acoustic and pulsation characteristics of centrifugal pumps

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Abstract. Numerical investigations of unsteady viscous flow in centrifugal pumps were conducted in this paper. Verification of numerical approach was done. Hydraulic components of the pump were modernized with the purpose to improve their acoustic characteristics. Modernization was held on the base of isotropic turbulence noise measuring techniques. Results were confirmed by experiment. Pressure pulsations were studied in a flow between rotating impeller and stator for initial and improved flowing parts. As a result of numerical research, various impellers with different blade outlet angles were designed. During optimization, the optimal blade outlet angle was found, which provides increased vibroacoustic characteristics while maintaining a high level of efficiency.

1 Introduction

One of the most important requirements for pumping equipment along with efficiency is low noise level. High level of noise could be caused by design of the pump, unsteady flow, vortex flow fields, interaction of flow with flow part elements [1-6]. It is important to have an instrument for assessment of the reasons and sources where noise begins. Nowadays, software packages exist, which are recommended for computation of viscous fluid dynamics in turbomachines. It will be useful to create methods for assessment of noise parameters of pumps. These methods could integrate three-dimensional calculation of viscous flow and semi-empirical calculations of hydrodynamic noise on the base of turbulent flow parameters in flowing part.

On the example of a centrifugal pump, an approach is considered in this paper, which allows identifying sources of noise in flow part of the pump and evaluating their contribution in acoustic power qualitatively and quantitatively. This approach enables the engineer on the first stage of designing process to improve acoustic quantities of the pump. Experimental measurements of noise level are made integrally in certain points of flow part and don’t allow evaluating the contribution of certain elements of flow part (impeller’s blades, return vanes of the continuous diffuser, flow turning and other) in noise generation. Such analysis could be held through numerical calculation.

One of the conditions of normal pump functioning is lack of vibration during its operation. In most cases, vibrations have hydrodynamic causes [7], such as interaction of flow with
return vanes, when the flow in the stator acts back on the velocity field in the impeller. These phenomena give rise to pressure pulsations, mechanical vibrations and alternating stresses in various pump components. Consequently, the most important steps to reduce the vibration is improving of flow part geometry, choosing of appropriate combinations of impeller blade and diffuser vane numbers, the radial gap between the impeller blades and the diffuser, blade inclination at outlet. A study on the impact of two last factors have been done in this work. Previously, such studies were made experimentally [8, 9], but now it is possible to hold investigations with the help of numerical flow modelling in transient analysis [10, 11].

2 Methods

2.1 Verification of mathematical model of viscous flow

Before performing the calculations, it is necessary to work out a method for numerical modelling of viscous flow in multistage centrifugal pumps. A distinctive feature of such pumps is the small sidewall gaps between the impeller and annular collector (less than 1%). The method should give qualitatively and quantitatively correct prediction of the characteristics of such pumps. Verification of mathematical model was conducted on the CNS 240-1525 pump. It is a 12-stage centrifugal pump with next nominal operation parameters: \( n = 3000 \) rpm; \( Q = 240 \) m\(^3\)/h; \( H = 1525 \) m; \( \eta = 79\% \); \( N = 1.2 \) MW; \( n_s = 72 \). The liquid model of the flow part of this pump included two intermediate stages. The energy parameters of the pump were calculated from the second stage. Grid generation was held in ANSYS Meshing [12]. The grid is unconstructed, made up from tetrahedral elements. To simulate the boundary layer, prismatic layers with parameters that satisfy the requirements of the k-e turbulence model were generated. The total number of elements that made up an entire flow part was 16 million. The calculation of viscous flow in the flow part of the pump was carried out in the ANSYS CFX software module. Both steady and transient approaches were performed. The calculated and experimental characteristics are presented in Fig. 1.

![Fig. 1. Performance characteristic of CNS 240-1525 pump.](https://doi.org/10.1051/e3sconf/20199107006)
Steady state gave a divergence 6% between the calculated head and the experiment value on the nominal mode. The measurement error during the transient state was in permissible limit. CNS 240-1525 pump has small sidewall gap between the impeller and radial diffuser \((D_3/D_2 = 1.013)\), therefore, steady state, obviously, doesn’t allow qualitatively resolving the flow in the region between the impeller and diffuser. This fact leads to high error when calculating pump parameters. For this reason, a non-stationary (transient) approach was used in subsequent numerical investigations.

### 2.2 Evaluation of generated noise

A general assessment of hydrodynamic noise can be made using two different approaches:

1. Using a semi-empirical approach [13] based on the kinetic energy of turbulence and its dissipation rate. This approach makes it possible to estimate the magnitudes of broadband noise generated by isotropic turbulence in a flow.

2. An approach based on the results of the calculation of the flow in the non-stationary formulation of pressure pulsations at various points of the flow part.

To estimate the magnitude of the broadband noise according to the method [13], it is enough to have a converged solution in a stationary formulation using the RANS model of turbulence.

In [13], Prudmann, using Lighthill's acoustic analogy, derived a formula for calculating the acoustic power generated by isotropic turbulence:

\[
P_A = \alpha \rho_0 \left( \frac{u^3}{l} \right) \frac{u^5}{a_0^5},
\]

where \(u\) – velocity, \(l\) – characteristic size, \(a_0\) – sound velocity, \(\rho_0\) – medium density. Considering the kinetic energy of turbulence \(k\) and its dissipation rate \(\varepsilon\), equation (1) takes the form:

\[
P_A = \alpha \rho_0 \varepsilon M_t^5,
\]

where \(M_t = \sqrt{2k/a_0}\).

The calibration constant \(\alpha\) has a value of 0.1 which is obtained on the base of the calibration of Sarkar and Husaini [14] while developing this method of calculation.

Acoustic power can be obtained from the next formula:

\[
L_p = 10 \log \left( \frac{P_A}{P_{ref}} \right),
\]

where the value of the reference acoustic power \(P_{ref} = 10-12\) W/m³.

Formula (1) allows getting a rough estimate of the acoustic power generated by isotropic turbulence in the flow part of the pump.

The second approach to assessment of the generated noise in the flow section of the pump is based on the spectral analysis of pressure pulsations in the flow section. For this, a non-stationary calculation of the flow of a viscous fluid must be performed, then pressure fluctuations are measured at control points. From the literature it is known [16] that the main source of pressure pulsations results from the interaction of the impeller blades with the return vanes. Therefore, the control points were in the gap between the impeller and the guide vanes. Therefore, the control points were in the gap between the impeller and the diffusor on the radius \(R = 1.05(D_2/2)\), where \(D_2\) – diameter of the impeller. Also, to evaluate the vibroacoustic efficiency of the diffuser, control points were established in the outlet pipe of
the pump, behind the diffusor. Because of periodicity principle of pressure pulsations, it is useful to use fast Fourier transformation to evaluate the results.

2.3 Evaluation of the pulsations of forces, applied to the pump rotor

Because of the different number of impeller’s blades and diffusor’s vanes, a circular flow irregularity starts to emerge. This irregularity raises radial forces which act on the rotor. Due to the complex spatial flow through the pump, the amplitude-frequency characteristic is a convenient representation of the results for analyzing this effect. Amplitude-frequency characteristic of the radial force can be obtained by fast Fourier transform (FFT) from dependence of the projections of forces on the coordinate axes – $F_x$ and $F_y$. Using CFD methods, it is possible conduct a numerical investigation of this effect by performing a non-stationary calculation. In that case the time dependences of the radial force projections on the $x$ and $y$ axes can be obtained by integrating the pressure over the surface of the impeller.

3 Results and discussion

A centrifugal pump with a specific speed of $n_s = 90$ was chosen as the object of study. The calculation of viscous fluid flow was performed in the ANSYS CFX software package [12]. Calculated area was consisted of flow at an inlet casing, impeller, axial continuous diffusor and collector. The flow was calculated in a steady state using the k-e turbulence model. The total pressure of 1 atm at the inlet boundary and the mass flow at the outlet were set as boundary conditions. Fig. 2 shows the hydrodynamic noise power field (dB), which is generated in the flow core in the channels of the continuous diffusor with return vanes. Places of high noise level in the core of the flow relate to eddy zones.

![Fig. 2. Hydrodynamic noise power in the flow core: a) an initial version of the continuous diffusor and return vanes; b) an improved version of the continuous diffusor and return vanes.](https://doi.org/10.1051/e3sconf/20199107006)

When designing the new continuous diffusor and return vanes, the eddy zones in the diffuser channels were significantly reduced. The maximum noise power in the channels of the modified diffusor was reduced from 35 to 5-10 dB (Fig. 2). Reducing the level of broadband noise was achieved using specific design of the return vanes. In the designed continuous diffusor, by increasing the thickness of the vanes, it became possible to reduce the diffusivity of the flow and eliminate the separation of the boundary layer.

After conducting a non-stationary calculation of the flow in ANSYS CFX, based on the analysis of the pressure fields, it was revealed, that the maximum amplitudes of pressure pulsations occur in the hydrodynamic trace behind the impeller (Fig. 3). Also, significant
values of pressure amplitude take place behind the return vanes of continuous diffusor and in the outlet pipe (Fig. 4-5).

Fig. 3. Amplitude-frequency characteristic at point behind the impeller (initial version).

Fig. 4. Amplitude-frequency characteristic at point behind the continuous diffusor (initial version).

Fig. 5. Amplitude-frequency characteristic at point in the outlet pipe (initial version).

The available experimental data indicate that the noise amplitude reaches the peaks in the outlet pipe at a frequency of 40 Hz and at a frequency of 300 Hz. The data obtained numerically show similar results.

A spectral analysis of the pressure pulsations in the stage with the new diffusor gave the results shown in Fig. 6-7.
Using an improved design of the continuous diffusor and return vanes, it became possible to reduce the amplitude of the pressure pulsations behind the vanes from 1100 Pa to 120 Pa, and in the outlet pipe from 1250 Pa to 116 Pa.

As a result of the work done it was revealed:
- on the base of semi-empirical methodic and calculated velocity fields, the value of broadband hydrodynamic noise, which is generated by the isotropic turbulence in the flow part, was evaluated quantitatively;
- as a result of transient flow calculation, the value of noise from pressure pulsations at the points with their greatest amplitude was evaluated; The adequacy of the results is confirmed by experimental data;
- using the developed design of continuous diffusor and return vanes, it became possible to reduce the level of broadband noise in the flow, as well as to reduce the pressure pulsations arising behind the return vanes vane and in the outlet pump.

3.1 Pulsations of forces on the rotor caused by hydrodynamic flow

Pulsations of forces arising on the rotor, and methods for reducing their amplitudes were investigated on the multistage pumps of the CNS type (multistage centrifugal pump).

The amplitude of force oscillations for the initial version of the pump was 1000 N, which can be explained by the proximity of the blade systems of the impeller and continuous diffusor ($D_2/D_1 = 1.0063$, gap - 0.63 %). The amplitude of the radial force pulsations of the improved stages with an increased radial clearance was significantly lower than the original
version with a small radial clearance between the impeller and the diffusor (Table 1). So, if for the initial version of the flow part the amplitude reaches 1000 N, for the improved variants 1 and 2 with an increased radial clearance – 300-400 N.

**Table 1.** Variants of CNS 240-2100 pump stages.

| Variant     | Initial | Variant 1 | Variant 2 |
|-------------|---------|-----------|-----------|
| Diffusor type | Channel-vaned | Channel-vaned | Vaned     |
| \(D_3/D_2\) | 1.0063  | 1.03      | 1.03      |
| \(D_4/D_2\) | 1.2     | 1.24      | 1.24      |
| \(H, m\)    | 2091    | 2110      | 2192      |
| \(\eta_h\)  | 0.855   | 0.877     | 0.897     |
| \(\eta\)    | 0.78    | 0.8       | 0.817     |
| \(F_x, F_y, N\) | 1000   | 400       | 300       |

An increasing the inclination of the trailing edge helps to reduce the radial force [15-18] (Fig. 8).

![Fig. 8. The inclination of the trailing edge.](image)

An influence of inclination of the trailing edge on the radial force pulsations and hydraulic efficiency have been investigated numerically. An inclination of the trailing edge was carried out on one and opposite directions relative to its vertical position. Calculations have shown that the direction of the edge inclination has a little effect on the amplitude function of the radial force pulsations from the angle of inclination. Amplitude-frequency characteristics at different inclination angle of trailing edge are shown in Fig. 9. Table 2 presents the values of the hydraulic efficiency of the investigated variants and the maximum amplitudes of the force pulsation.

![Fig. 9. Amplitude-frequency characteristics at different inclination angle of trailing edge: a) \(\gamma = 0^\circ\), b) \(\gamma = 42^\circ\).](image)
Table 2. Simulation results at different angles of inclination of the trailing edge.

| Inclination angle, deg | Hydraulic efficiency, % | Frequency of the main harmonic, Hz | The amplitude of the radial force pulsation, N |
|------------------------|-------------------------|------------------------------------|-----------------------------------------------|
| 0                      | 96,6                    | 1330                               | 360                                           |
| 14                     | 96,7                    | 1330                               | 342                                           |
| 42                     | 96,6                    | 1330                               | 193                                           |
| 54                     | 96,6                    | 1330                               | 64                                            |
| 66                     | 83,9                    | 1400                               | 27                                            |
| The backward inclination of the trailing edge |
| -18                    | 96,7                    | 1310                               | 312                                           |
| -47                    | 96,6                    | 1310                               | 142                                           |
| -56                    | 96,7                    | 1310                               | 60                                            |
| -67                    | 88,1                    | 1340                               | 26                                            |

The rotational frequency of investigated pump is $f_r = 50$ Hz, the blade frequency is $f_b = Z_{imp} \times f_r = 450$ Hz, the vane frequency is $f_{vane} = Z_{diff} \times Z_{imp} \times f_r = 5850$ Hz. The result frequency of the main harmonic of the pulsation force is in the range between the blade and the vane frequency. Fig. 10 shows the graphical dependence of the amplitude of the radial force pulsation on the angle of inclination of the trailing edge at positive angles of inclination.

![Fig. 10. Dependence of the amplitude of the radial force ripple on the angle of inclination of the output edge of the RC.](https://doi.org/10.1051/e3sconf/20199107006)

According to the results of the conducted numerical analysis, the following conclusions can be made:

- to reduce the amplitude of the pulsation of the radial force acting on the rotor, it is recommended to incline the trailing edge at an angle $\gamma$ not more than 50°, which allows reducing the amplitude of the radial force without decreasing of the hydraulic qualities of the impeller;
- the inclination of the trailing edge of the impeller can be made in both the backward and forward directions of impeller’s rotation. The conducted numerical studies show that the direction of inclination does not have a significant effect on the value of the radial force amplitude;
- if an increasing of inclination angle is more than 50°, the efficiency of the impeller decreases, the process of additional treatment of the blades surface in this case could be more difficult.
4 Conclusions

1. The methodology of numerical evaluation of the acoustic characteristics of a pump has been worked out. As a result of the application of described methodology for estimating broadband noise, as well as pressure pulsations in the flow, an upgraded flow part with improved noise characteristics was developed, which was later confirmed by experimental investigations.

2. Using numerical methods for calculating the flow of a viscous fluid, a study of a pump impeller with an inclined trailing edge was performed. The vibroacoustic characteristics of the pump were estimated by calculating the radial force pulsations on the rotor. As a result of the conducted numerical study, the optimal value of the inclination angle of the trailing edge, which provides the smallest values of the force pulsation amplitude preserving a high level of efficiency, was found.

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