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Complex approach to use the splitter blades impellers to improve the centrifugal pump performance

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Summary

This work presents the results of complex research of changes in the hydraulic, stress-strain state and dynamic characteristics of the double-entry single stage pump (specific speed nq=55) in the case of introduction of splitter blades in the basic design are presented in this paper. Since the splitter blades modify flow within the blade-to-blade channel, the energy loss will depend on their shape. Determination of the maximum possible increasing of impeller head while maintaining or increasing the efficiency is connected with multiparametric optimization of the splitter blades ge-ometry. Geometrical dimensions determining the shape and position of the splitter blades were used as variable parameters. Head and efficiency are the objective functions. Determination of hydrodynamic flow parameters was carried out by using the CFD methods. In this paper some thoughts regarding the influence of the shape and location of the splitter blade on the distribution of flow parameters are discussed. Comparative analysis of strength and vibration characteristics of the pump with the original impeller and splitter blade impellers carried out on the basis of FEM-analysis. The results of the this study highlighted aspects allow us to conclude that to ensure reliable operation of the pump approach to splitter blade impeller design should be more consistent and comprehensive.

Introduction

In a contrast to compressors the splitter blade impellers are not widely used in the centrifugal pumps with average specific speed. This type of impellers is mostly applicable in the high speed pumps and to low the NPSH by reducing inlet blade blockage. This work became a result of several reasons:

- A series of implemented projects with application of the splitter blades impellers for retro-fit of existing centrifugal pumps in order to change the individual pump performance.
- There are the results of research in the open sources considering the certain aspects of the splitters application: increasing the impeller's pressure head in case of limitation the pump flow path dimensions or improving suction performance or the positive effect on the pressure fluctuations at the impeller periphery.
- The intention to illustrate how to change the single pump parameters (energy, cavitation, strength, vibration) by applying a splitter because in some cases focusing on one of these parameters may overshadow the equally important aspects of pump reliability and operability.

This paper presents the results of a complex research describing the changes of hydraulic and dynamic characteristics of the single-stage double-suction pump (with specific speed n_q=55) in the case of the splitter blades introduction in the basic design. Since the splitter blades modify the flow within the blade-to-blade channel, the energy loss will depend on the blades' shape. In this paper some thoughts regarding the influence of the splitter blade shape and location on the distribution of the flow parameters are discussed.

Pump geometry and description of the experiment

As the object an axially split single-stage double-suction pump with shrouded impeller (diameter = 520 mm, 7 blades, specific speed = 55) has been selected (Figure 1).

Q_{rated}, m^3/h	9150
H, m	238
n, rpm	2905
NPSHR, m	48.5

Figure 1. Original pump

During the study the following methods have been used:

- Multiparametric optimization
-
- Interactive impeller design (performed in Cfturbo software pack)
-
- CFD (performed in ANSYS CFX and PumpLinx software pack)
-
- FEM (implemented in ANSYS Mechanical software pack).

At the first stage it was considered the influence of the location and shapes of splitters on the energy characteristics of the pump. Basing on the original impeller, 28 new impellers were designed with different splitter blades parameters. The base impeller was modified by adding the splitter blades preserving the same blade outlet angle β_2. The circumferential position of the splitter blades outlet has been chosen in the middle. In order to determine the maximum possible increase of the impeller pressure head with minimum reduction of efficiency it is necessary to optimize the geometry of the splitter blades. In this connection, two-criteria of five parametric optimization problem was solved. The combination of independent parameters defining the geometry of additional blades has been obtained based on a table with pseudorandom numbers, using LPτ – sequence to obtain the combinations of geometric parameters of additional blades. LPτ - sequence is the most evenly distributed among the currently known sequences points in Kn (n-dimensional space of pseudo-random points) [1].

As the independent variable parameters 5 geometric parameters were selected to describe the splitter blade (Figure 2):

- Blade inlet radius at the hub and shroud streamlines
-
- Relative position of the inlet blade in the blade-to-blade passage
-
- Blade inlet angles at the hub and shroud streamlines
-
- Inlet blade thickness
-

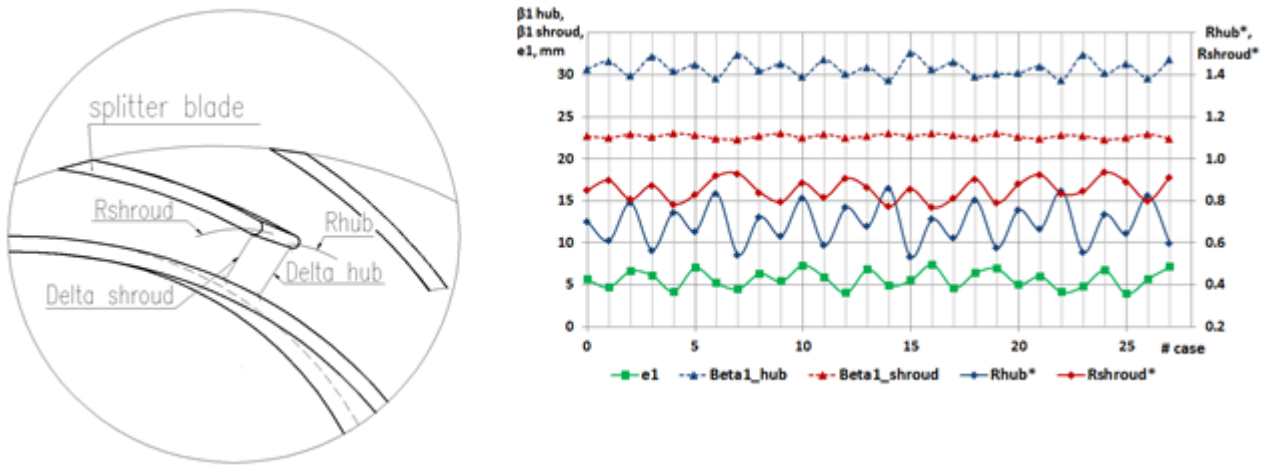

Figure 2. Varying the geometric parameters of a splitter blade

The interval of these geometrical parameters has been chosen within the range from 20 to 80%. To minimize the loss at the volute inlet the outlet of the impeller shall have maximally uniform distribution of pressure and velocity fields. In this regard, the output of the splitter blade is located in the middle of impeller channel and repeats geometry of the main blades. A series of calculations included 28 numerical experiments.

Stage 1. Energy parameters

At the first stage the energy parameters were analyzed at the rated flow rate. 28 numerical experiments (CFD) were conducted to determine the changes in the pump head and efficiency. The computational model is a model of complete pump (Figure 3). For numerical stability reasons and to minimize the edge effects the computational domain is extended upstream and downstream. As a tool for decisions in the field of sampling points PumpLinx software pack was used, allowing perform the computer modeling of the fluid flow based on the solution of the Navier-Stokes system of differential equations for the circuit which uses the k-ε model of turbulence, with standard boundary function.

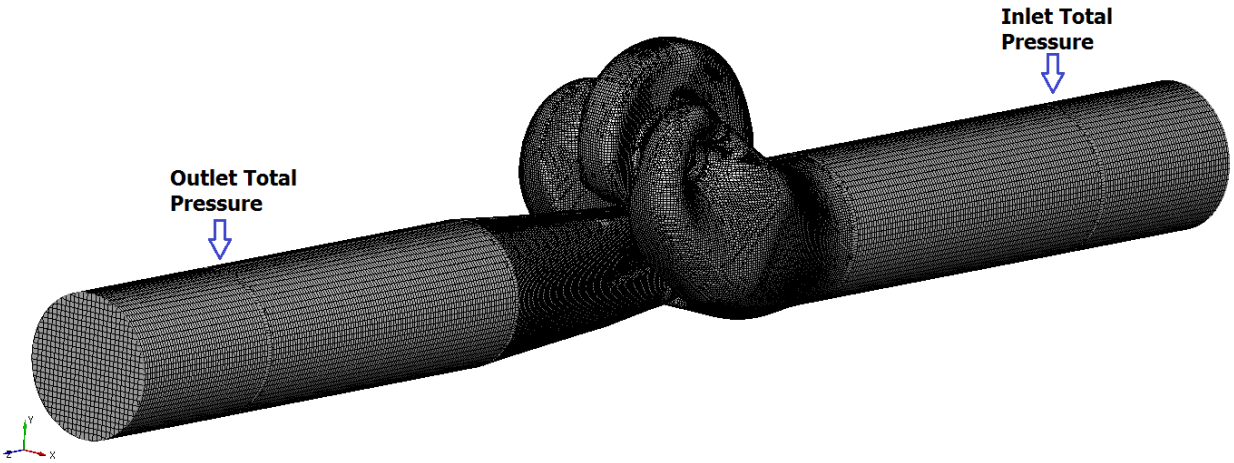

Figure 3. Computational fluid model

The pump head was defined as the difference between total pressures (dynamic and static pressure) in the cross sections by a distance 2.5* (pipe diameters) from the inlet of the semi-volute/outlet of the spiral volute. Figure 4 shows the pump head increasing in the dimensionless form versus hydraulic efficiency.

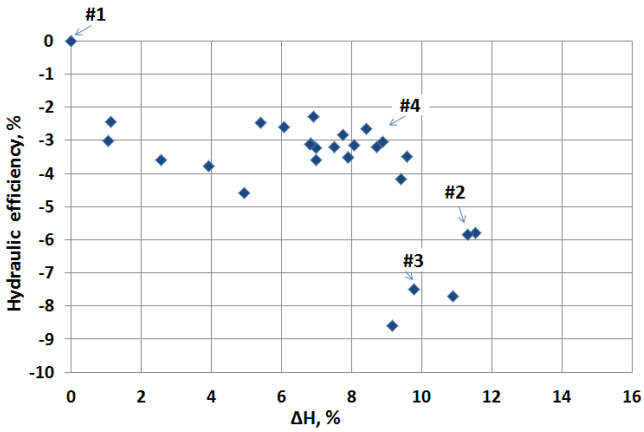

Figure 4. Solutions field

The data analysis showed that head increased within the range from 1.0 to 11.5% while the hydraulic efficiency decreased from 2.3 to 8.6% (maximum efficiency drop is observed in the variant with the highest head increase).
Then a number of response surfaces were built to assess the pump head and efficiency dependence on geometric parameters of the splitter blade. The response surfaces analysis (Figure 5) showed the following:

- The maximum increase of pressure with minimum reduction of efficiency is possible if the radius of the splitter blade on the hub is reduced while at the same time the radius on the shroud is increased.

- The minimum efficiency drop at relatively high increase of the pressure head has been observed when the angular position of the splitter blade inlet in the blade-to-blade channel is offset from the pressure side by 30 ... 40% on hub and shroud. It may be explained by redistribution of the main flow within the blade-to-blade channel from the pressure side to the suction side of the blades.

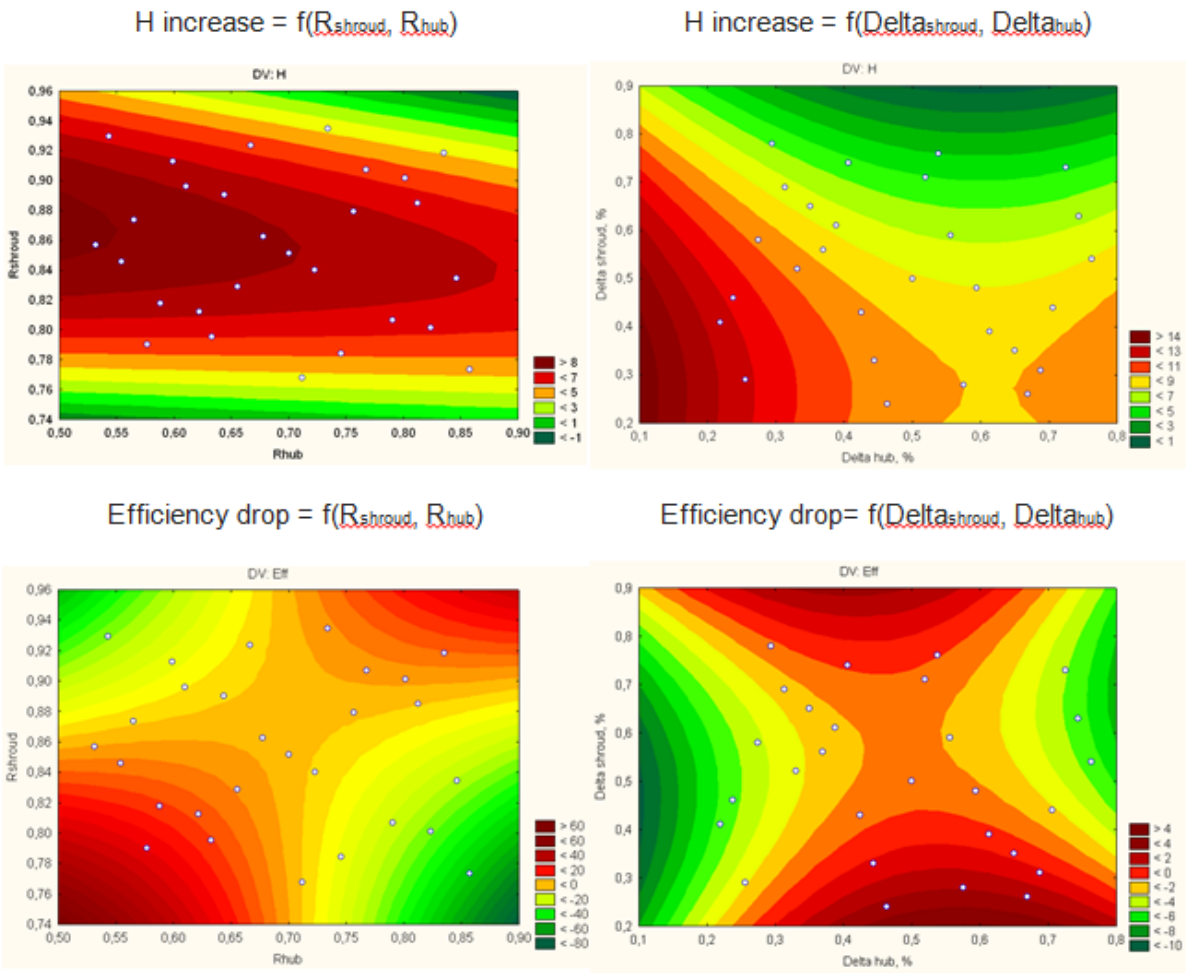

Figure 5. Response surfaces

For the subsequent comprehensive comparative analysis of the splitter blades characteristics in comparison with the base impeller the following impellers options have been chosen (Figure 6):

- Impeller #1 – base impeller
- Impeller #2 – splitter blades impeller which provides head increase up to 11.3% and efficiency drop by 5.8%.
- Impeller #3 – splitter blades impeller which provides head increase up to 9.8% and efficiency drop by 7.5%.
- Impeller #4 – splitter blades impeller which provides head increase up to 8.9% and efficiency drop by 3.0%.
- Impeller #5 that has 5 main and 5 splitter blades.

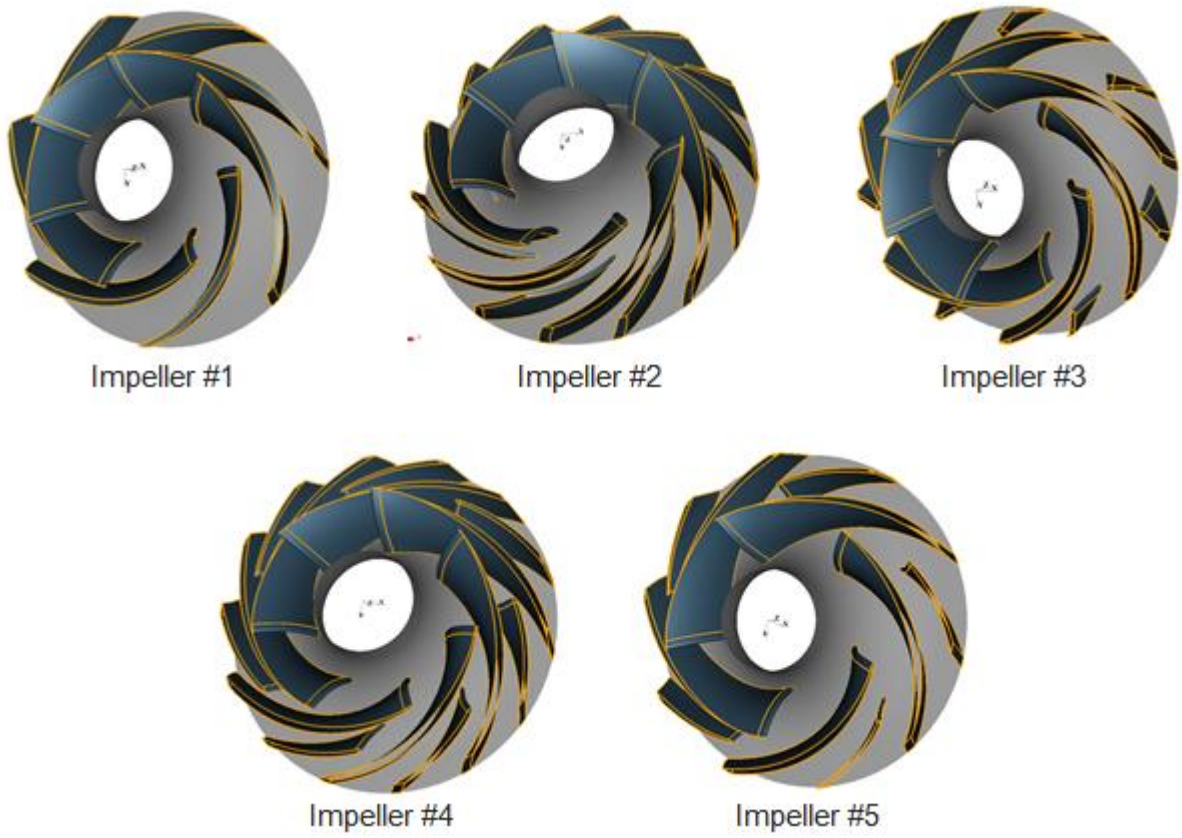

Figure 6. Impeller cases for further research

These options were chosen since they provide approximately the same head increase at rated flow rate, but have different splitter shapes.

Simulation for the calculation of the energy performance were carried out in the range of flow rates [0.7…1.1] Q_{rated} (Figure 7). The results are shown in Figure 7 in a dimensionless form. The accuracy of the calculation model is confirmed by the correlation of the CFD calculation and

experimental data for the base Impeller #1. It is clearly seen that the head and efficiency curves of the case with the splitter blades impellers are almost equidistant compared to the original (base) impeller.

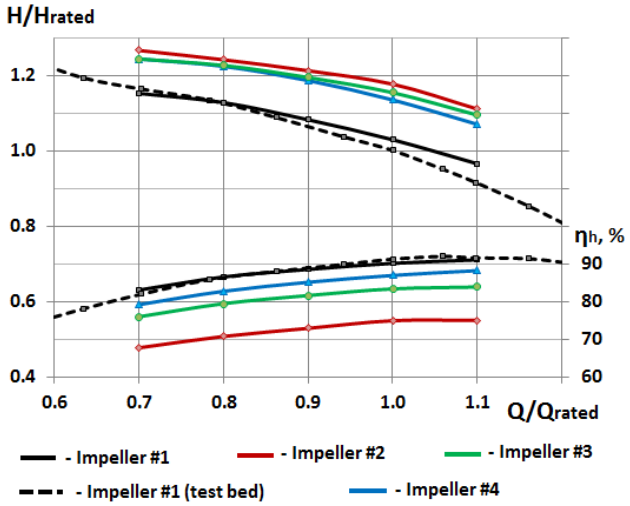

Figure 7. Results of numerical calculation (CFD)

Stage 2. Cavitation qualities

Cavitation in centrifugal pumps leads to erosion of the channel walls, drop of head/power/efficiency. Cavitation of centrifugal pumps often causes vibration. The pump operation in a cavitation mode may cause serious accidents and therefore is not permissible, especially when pumping flammable and toxic media. Therefore it is necessary to consider the question whether the splitter blade impeller can change the cavitation characteristics of the pump. At first sight nothing shall change since the pump entry area is not changed. On the other hand, there is a dependence of NPSH on the relative and absolute velocity parameters at the blade's entry.

$$NPSH = m\frac{V_0^2}{2g} + n\frac{W_1^2}{2g} \tag{1}$$

where

V_0 - velocity at the entry of impeller;
W_1 – the relative velocity at the entry of the blade;
m, n – experienced coefficients characterizing the rate of increase vs. its average value.

That is, for an identical impeller entry geometry the relative distribution speed W_1 and the absolute value of V_0 will affect the change in NPSH3. The numerical calculation of cavitation characteristics was performed using PumpLinx [3].

The suction specific speed of the base impeller is equal nss = 178. For further study the Impeller #5 (see Figure 6) was also considered. CFD calculations showed that Impeller #5 (4 main blades and 4 splitter blades) provides 0.95% less pressure head and 1.95% less efficiency compared to Impeller #1 (base impeller).
The calculation of series characteristics NPSH curves (with constant speed by successive reduction of the inlet pressure (each curve is measured at Q = constant)) were carried out. The

obtained results in the range of [0.7 ... 1.1] Qrated are shown in Figure 8. As we can see the splitter blades didn't change significantly the NPSH characteristics of the base Impeller #1.

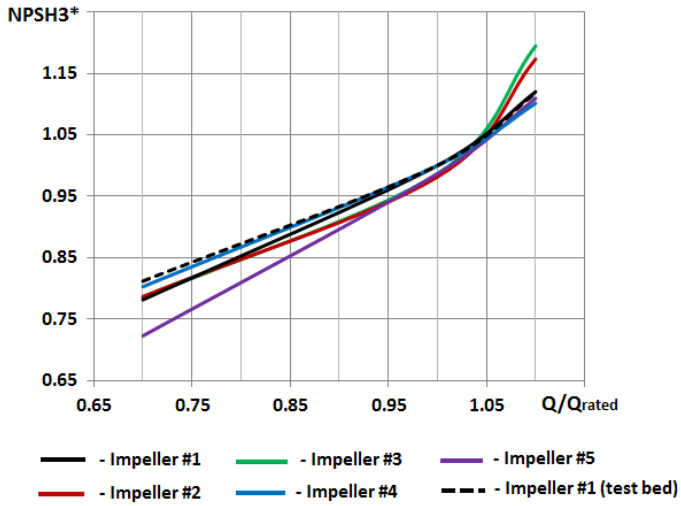

Figure 8. NPSH3 curves for different cases

Let's consider how to change the kinematic parameters included in the Formula (1) for the studied cases. As it can be seen in Figure 8 for cases Impeller#1 and Impeller#2 the introduction of the splitter blade does not bring notable changes in the relative velocity downstream since the blade-to-blade channel is wide enough. If we consider the cases Impeller#1 and Impeller#5, we can say that inlet blade blockage (calculated using [2]) differs by 3%. It has little effect on the change of the meridional velocity. But there are more important changes in relative velocity in the inlet field (Figure 9) especially at the 0.7 Q_{rated} mode. For the case Impeller#5 there is a clear correspondence between reducing NPSH3 and a low relative velocity on the suction side at the blade entry.

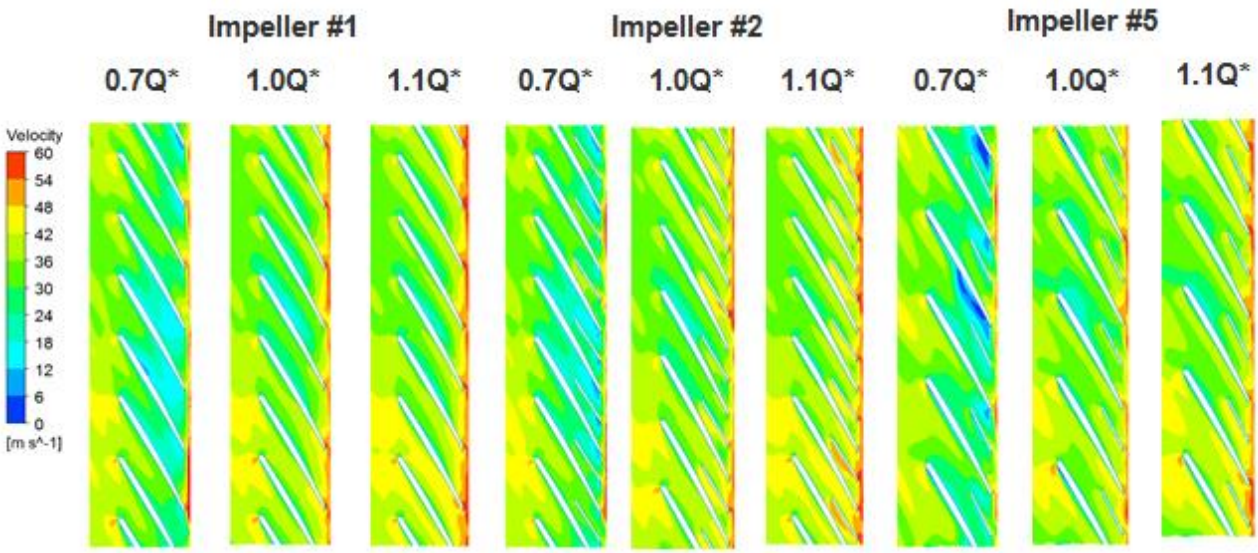

Figure 9. Pictures relative velocity distribution (mean surface)

The reasons given above have confirmed the distribution pictures of steam cavern for different pressure conditions at the pump inlet (Figure 10). Also it clearly shows that the cavity primarily appears in the main blades.

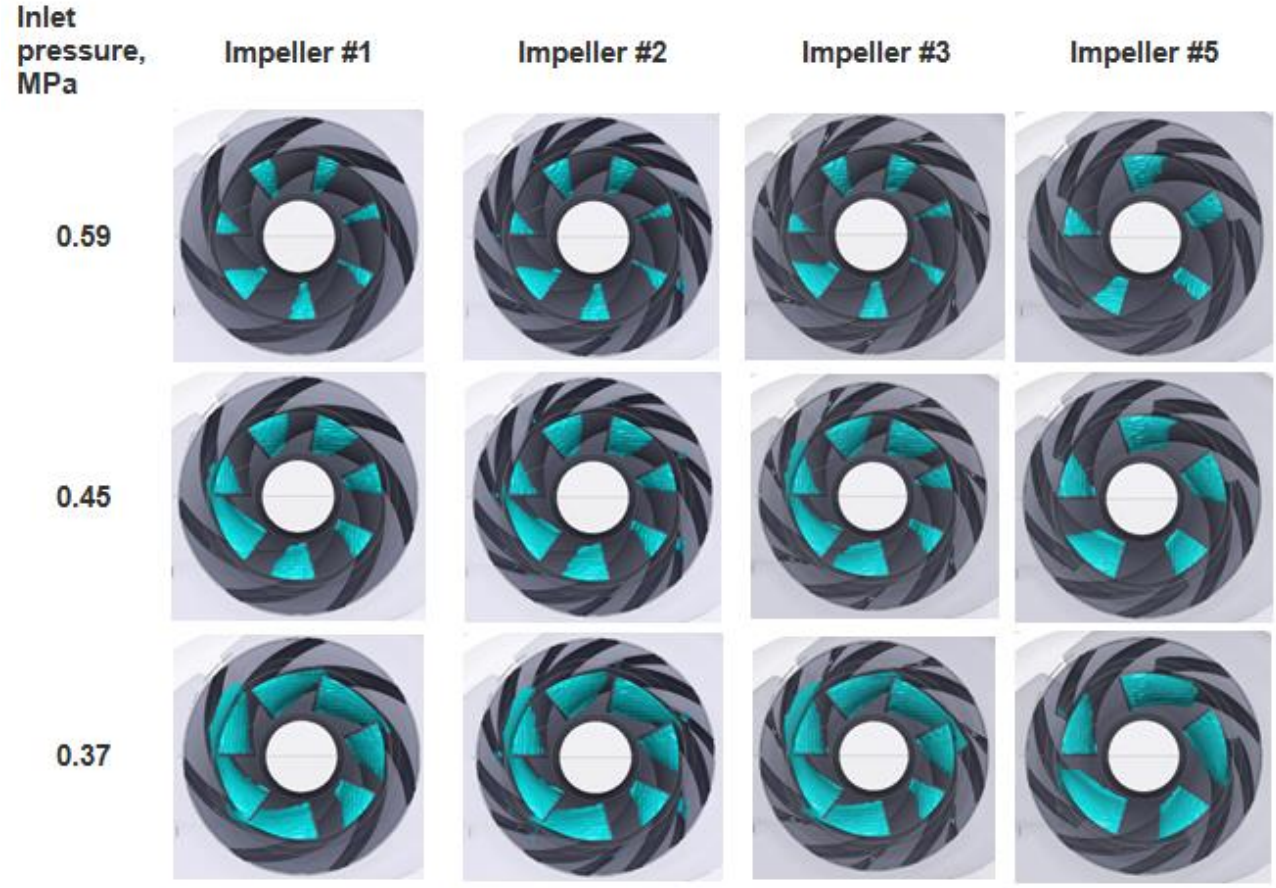

Figure 10. Pictures of cavity in different modes

Stage 3. Vibration analysis

The range of a centrifugal pump vibration within the range of audible frequencies (20-20 000 Hz) is a continuous one with strong discrete components, which are the most dangerous due to their harmful effects on the human body, so their research and development of measures to address them are of the greatest practical interest.

The most common excitation force is mechanical unbalance of the pump. It originates from the residual unbalance of all parts fitted to the rotor and from residual shaft runout. Unbalancing forces and their distribution and phase angle along the rotor are not predictable, but are held within acceptable limits due to balancing procedures and preset limits for residual shaft runout. Sometimes the coupling is a source of radial forces, especially if the pump and motor shafts are not properly aligned.

It is more difficult to handle the hydraulic dynamic forces. They may be more or less random, especially at minimum flow conditions, or they may be periodic forces characterized by their frequency. Being known as subsynchronous forces due to rotating stall at low flows, while synchronous forces, often called “hydraulic unbalance”, are due mainly by the following:

- Uneven velocity distribution caused by input, rib, vortices
- Deviation of flow inside the impeller
- Asymmetry of the discharging element
- Inaccuracy in the blade-to-blade channel after casting

Hydraulic dynamic forces may have a strong effect on forced shaft response. A good prediction of these forces as a function of impeller and volute/diffuser geometrical parameters and tolerance is not generally possible. Radial dynamics excitation forces at higher harmonics of the rotational frequency depend on blade numbers of impeller and volute/diffuser. The unsteady impeller outlet flow generates a pressure field that rotates with a 'blade' frequency $f_{blade}=z_{La}\times n/60$. Harmonic vibration of the blade frequency has its own amplitude. The fb is a constructive feature of the equipment, and isn't a unique sign of trouble in the pump. Particular attention they deserve to begin in certain defect condition and operation, when the amplitude of the blade frequency harmonics begins to increase.

Pressure pulsation is the typical characteristics of hydrodynamics, which can indirectly reflects the dynamic information of inner flow characteristics. In order to study the characteristics of pressure fluctuation for our study cases the static pressure for the period was taken on the periphery of the Impeller#1, Impeller#2 and Impeller#4 (transient mode). Prior to the CFD analysis there was speculation that Impeller#2/Impeller#4 shall have a more uniform for at the outlet by more than the number of blades at the exit. But further calculations have shown that this assumption was premature in these cases. In Figure 11 it is seen that amplitude of static pressure in the cases Impeller#2/Impeller#4 increasse as compared to Impeller#1. This increasing the static pressure fluctuation level may be connected with the fact that splitter blade divides main blade-to-blade channel into two short channels. In the case of Impeller#2/Impeller#4 one the two channels has large opening angle that affects the output parameters.

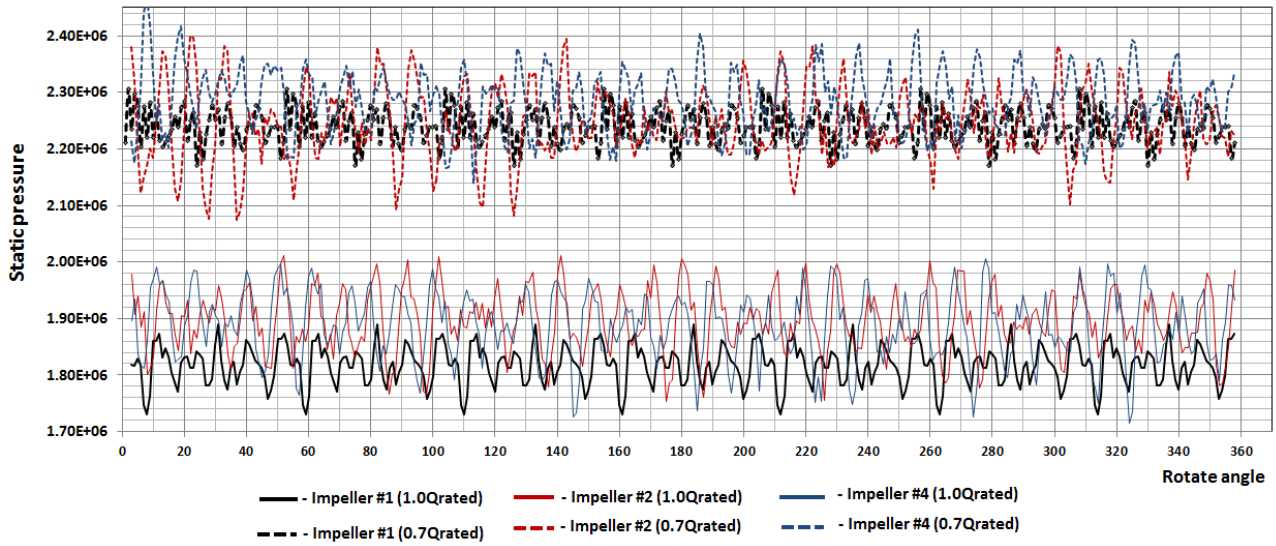

Figure 11. Distribution of static pressure around the impeller periphery (Impeller#1, Impeller#2) in the mode $1.0Q_{rated}$, $0.7Q_{rated}$.

The next step was the calculation of vibration associated with the hydrodynamic processes in the flow. The calculation method was transient structural dynamics analysis. Model for FEM vibration analysis is shown in Figure 12. The vibration is usually measured on the bearing housing in the horizontal, vertical and axial directions.

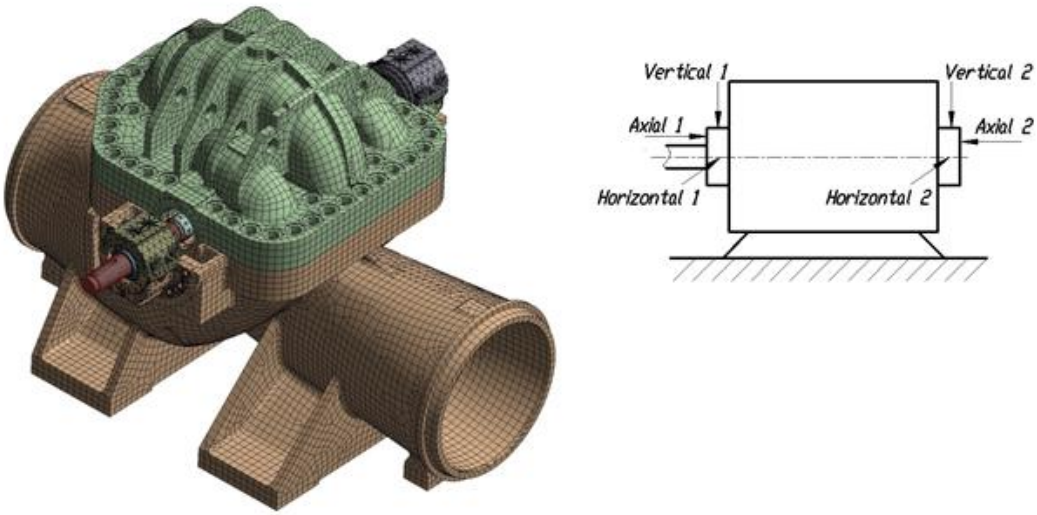

Figure 12. Model for FEM analysis and points to determine the vibration

The boundary conditions were the fixed support at pump casing feet and the force vector applied to mass center of the impeller. This force as a function of rotation angle was obtained by the calculation transient CFD–analysis (for example, in Figure 13).

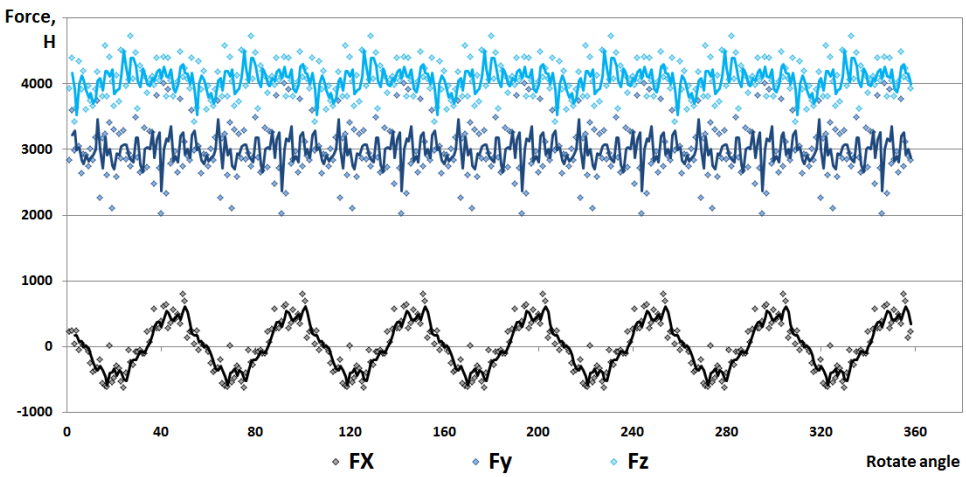

Figure 13. Forces for model Impeller#1 (1.0Q_{rated}), transient calculate mode

Pump rotor is supported by hydrodynamic bearings modeled by special type of finite element (spring) allowing to consider stiffness and damping of fluid film. It should be clarified that this vibration calculation did not considered the sources of vibration such as imbalance and misalignment of the shafts. As a result of the transient calculation is possible to get a vibration velocity graph per revolution (for example for bearing 2, Figure 14) and then to get out of these data root mean square velocity.

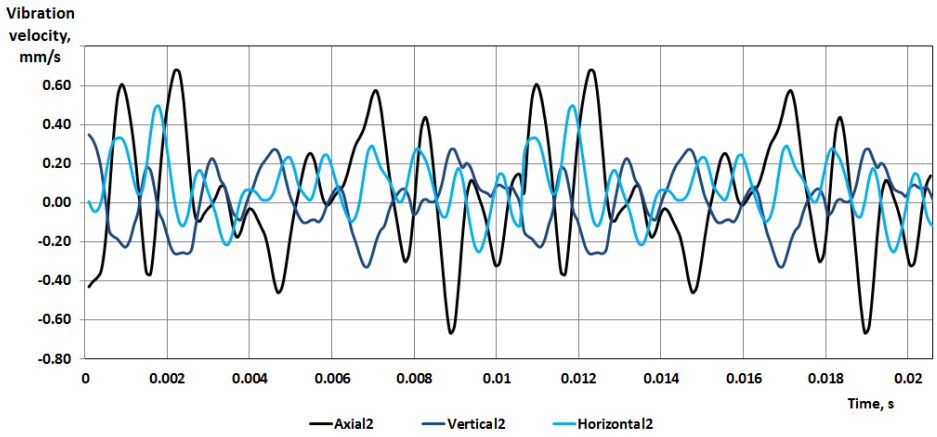

Figure 14. Vibration velocity on bearing housing for case Impeller#1 (1.0Q_{rated}) (transient calculate mode)

It is known that transient calculations whether hydrodynamics and mechanics, take a lot of time and computer resources. Since this calculation makes clear the contribution the hydrodynamics brings into vibration, it was decided to see what can be get if the forces of signal is decomposed into a Fourier series using the method of Fast Fourier Transform (FFT), and then to calculate with the boundary conditions in the form of a harmonic signal with 'blade' amplitude and frequency. Using the FFT applied to the components of forces on the rotor can be useful even before the vibration calculation stage. For example, for case Impeller#1 (Figure 15) FFT allows to conclude that among all the components of the force for case Impeller#1 explicit harmony is observed for the axial component of the force. This corresponds to the data from the experimental measurements (force vector is directed toward inlet flange). For case Impeller#2 harmony force begins to appear in all three components of the force, while not only the blade frequency, but also at subsynchronous frequencies.

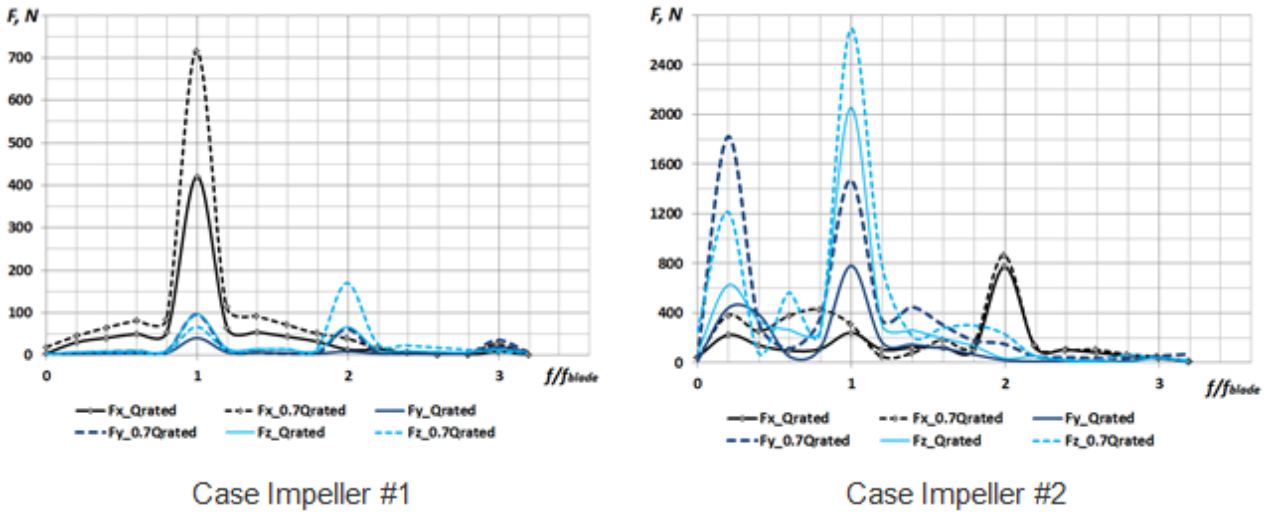

Figure 15. The range of forces resulted from FFT

Thus at the stage of calculating the forces acting on the rotor, it is possible to estimate the ratio of the hydrodynamic component vibration along the axes based on the spectrum analysis of the forces acting on the rotor.
Comparative results for these transient forces, for the harmonic forces and the experimental data (test bed) in the case Impeller#1 are presented in Figure 16. Comparison of the results of transient calculations and bench tests showed that there is good enough, but not excellent agreement in magnitude. This may be due to the fact that only the simulated effect on the rotor, while in reality there are also effects on the case. Also in this case, used as support sliding bearings that can make adjustments in the transmission of forces from the rotor to the case in comparison with rolling bearings. Therefore the matter of calculation model requires further clarification. But the results of transient calculation showed that proposed computational model is more accurate than the model where the harmonic signal is applied.

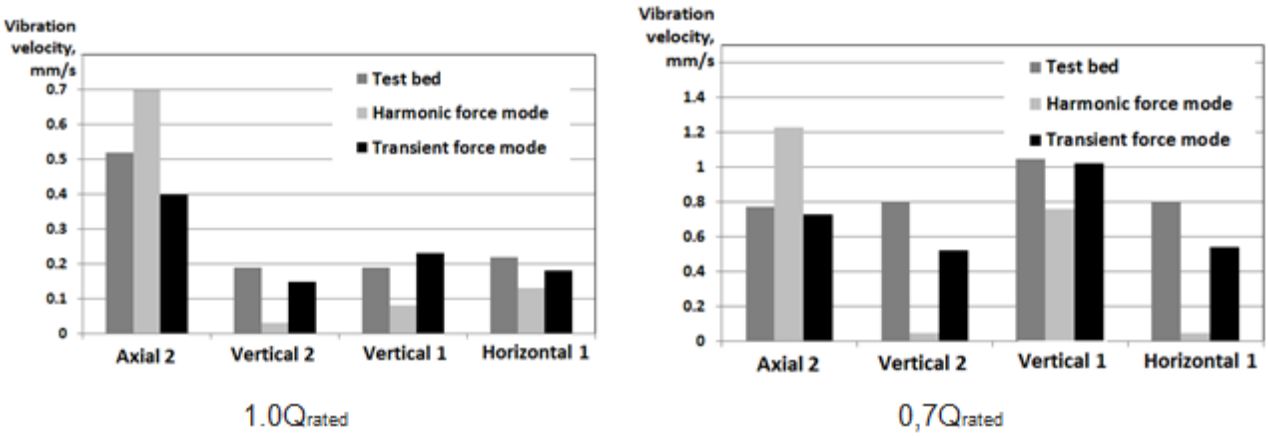

Figure 16. Results with conditions of harmonic forces for case Impeller#1.

Nevertheless the proposed transient calculate model allows to get a relative comparison between the vibration velocity parameters for the cases under consideration. The following is a comparison of the obtained data for the cases Impeller#1 and Impeller#2 (Figure 17).

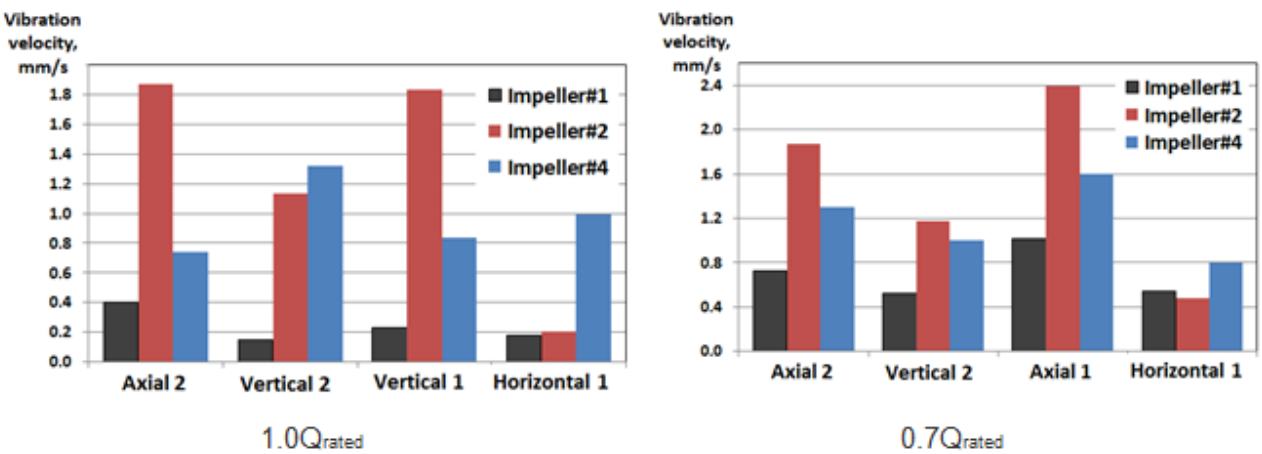

Figure 17. Comparison of the obtained data for Impeller#1, Impeller#2, Impeller#4 cases.

For this case the vibration levels of pump were much higher in comparison with the base case. An interesting point was seen while comparing the velocity levels for the two flow rate modes. At the $1.0Q_{rated}$ and $0,7Q_{rated}$ cases Impeller#2/Impeller#4 the difference between the vibration velocity is less than the difference between the vibration velocity in the cases Impeller#1. So, it shows that prior review for the structure of pump is able to prevent potential problems of the pump at the design stage. It will be effective to consider the case of splitter blades impeller with blade system '4 main blades/4 splitter blades' giving the same pressure head to analyze the possibilities for improving the vibrodynamic behavior of the pump.

Conclusion

According to this research it clearly seen that application of the splitter blades requires comprehensive analysis of changes in the pump performance.
The numerical results showed the following for existing double-entry single stage pump (specific speed = 55):

- Changing the shape of the splitter blade it is possible in order to increase the pump head, while maintaining the original impeller output diameter to an average of 10% and the efficiency drop to 3%. At the same time through the use of a special experimental design and subsequent analysis of response surfaces plan may gain an understanding of how to change the head and efficiency, depending on the geometrical parameters defining the position and shape of the splitter blade.

- Changes in cavitation qualities are not observed. If reduction of NPSH is necessary it should be considered as the option of reducing the number of blades at the entrance from 7 to 4 (with simultaneous introduction of the splitter blades)

- Adding the splitter blades is changing the pressure fluctuations at the interface of impeller and volute. Different design parameters of splitter blades have a certain effect on the pressure fluctuations at the interface of impeller and volute.

- The vibration analysis by FEM can examine the relative change in vibration of the pump casing when the usual impeller is replaced with a splitter impeller. In one of the cases the calculation showed that, contrary to expectations, vibration velocity has significantly grown compared to the base case. To obtain an accurate prediction of vibration velocity it is necessary to make additional research in order to clarify the boundary conditions responsible for stiffness and material damping to eventually get a verified calculation model.

It shall be noted that the study was conducted to demonstrate one of the complex analysis methods of the pump characteristics change when using the splitter blade impeller. Applying such criteria allows a more consistent approach to impeller design. Furthermore, there is a possibility of obtaining the analytical relations between head/efficiency and splitter blade geometry by using a regression analysis method. It will allow rapid and quite accurate prediction of the pump characteristics with splitter blades impeller.

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