Original Paper

CFD Analysis of Cavitation Phenomena in Mixed-Flow Pump

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Abstract

This paper deals with the CFD analysis of cavitating flow in the mixed-flow pump with the specific speed of 1.64 which suffers from a high level of noise and vibrations close to the optimal flow coefficient. The ANSYS CFX package has been used to solve URANS equations together with the Rayleigh-Plesset model and the SST-SAS turbulence model has been employed to capture highly unsteady phenomena inside the pump. The CFD analysis has provided a good picture of the cavitation structures inside the pump and their dynamics for a wide range of flow coefficients and NPSH values. Cavitation instabilities were detected at 70% of the optimal flow coefficient close to the NPSH3 value (NPSH3 is the net positive suction head required for the 3% drop of the total head of the pump).

Keywords: mixed-flow pump, cavitation, CFD, unsteady flow phenomena

1. Introduction

The mixed-flow pumps often suffer from a very high level of noise and vibrations close to the optimal flow coefficient (ϕ_{BEP}). This negative property can be usually suppressed, but at the cost of decreasing the pump efficiency or a very robust and heavy construction. Of course, both these methods are hardly acceptable for the high performance pumps these days and that is why the detailed knowledge of the noise and vibration sources can be very useful for the pump optimisation.

Recently, many numerical and experimental studies concerning cavitation phenomena in the mixed-flow pumps have appeared, e.g. [1-7], but they are mostly concerned with the steady cavitation. Kobayashi and Chiba [8] modelled unsteady hydraulic forces acting on the impeller of a mixed-flow pump operated under cavitation using the Large Eddy Simulations (LES) while Yamamoto and Tsujimoto [9] studied extensively the cavitation surge in a centrifugal pump. Nevertheless there is still lack of information in the literature concerning cavitation instabilities in the mixed-flow or radial-flow pumps. Practically all studies dealing with these instabilities are devoted to inducers, which represent a special type of the axial-flow pumps. When operating under cavitation, inducers can suffer from a wide range of cavitation instabilities (Kamijo et al. [10] or Tsujimoto et al. [11]), even at the design flow rate. These instabilities generate oscillations dangerous for the machine operation, especially at high rotational speeds. One of the diagnostics which is used to determinate the type of cavitation instability is the frequency analysis. We have to examine the dominant frequencies and compare them with the rotational frequency f_Ω as well as the fundamental frequency of the

Table 1 Cavitation instabilities in inducers

Instability	Frequency Range
rotating cavitation	$1.1f_\Omega - 1.3 f_\Omega$
rotating cavitation, backward travelling mode	$0.9 f_\Omega$
partial cavitation oscillation	$< f_\Omega$
cavitation in backflow vortices	$0.16n f_\Omega - 0.21n f_\Omega$, n being number of vortices
attached uneven cavitation	f_Ω

inducer blade passage excitation. Table 1 comes from the works of Brennen [12] and Tsujimoto et al. [11], [13-14] and shows inducer typical frequencies of some cavitation instabilities in the inducers. To examine the unsteady cavitating performance, experimental methods have been widely used for a long time. Recently the progress in development of CFD tools has encouraged authors to examine unsteady cavitation phenomena in the numerical way (e.g. Berntsen et al. [15], Bouziad [16] or Tsujimoto et al. [17-18]). These numerical tests indicate that there are limitations in modelling of cavitation instabilities using URANS (Unsteady Reynolds-Averaged Navier-Stokes) based codes and that the future work will tend to the Large Eddy Simulations (LES), though they are still far from current engineering practice.

2. Description of Pump and Numerical Procedure

The pump which is an object of this paper has the dimensionless specific speed $N = 1.64$ and is produced in two configurations: with a volute (Fig. 1) or with the axial diffuser (with nine vanes) and 90° elbow (Fig. 2). The mixed-flow impeller is open with the tip clearance of $0.002 R_{T2}$, where R_{T2} is the radius of the blade tip at the impeller outlet. It has four blades. A comprehensive CFD analysis of the pump has been done for both configurations. Because there is some difference between the optimal flow coefficients of both versions (about 7%), we refer in this article to the relative flow rate φ/φ_{BEP} , where φ_{BEP} is the flow coefficient at the best efficiency point.

Fig. 1 Model of pump with volute

Fig. 2 Model of pump with axial diffuser and elbow

The available measurements (Fig. 3) show the peak of the noise as well as vibrations at about 70% of φ_{BEP} , which is inside the typical allowable operating range of the pump. These vibrations and noise can be found at rather high NPSH values, but they are very strong especially close to the NPSH value equal to NPSH3. The minimum value of the vibrations and noise appears at the flow coefficient about 117% of φ_{BEP} . This corresponds well with the NPSHi(φ) curve (Fig. 4), which is very similar for both pump configurations and reaches its maximum at the flow coefficient of 70% of φ_{BEP} and the minimum value approximately at 108% of φ_{BEP} . Here NPSH3 is defined as the net positive suction head required for the 3% drop of the total head of the pump and NPSHi represents the net positive suction head required for the cavitation inception. The measurements of the noise and vibrations were performed with the volute pump at the NPSH value of 9 m. This is (with the exception of flow coefficients larger than 121% of φ_{BEP}) higher than NPSH3 (Fig. 5), but lower than the NPSHi value. Therefore it can be expected that the level of the noise and vibrations is closely connected to the cavitation phenomena, which can include (at least at the flow coefficients close to the noise level maximum) the cavitation instabilities with highly unsteady behaviour.

In this study the ANSYS CFX commercial CFD package has been used to solve URANS equations together with the Rayleigh-Plesset model, which is included in the ANSYS CFD software to describe the interphase mass transfer in the framework of the homogenous multiphase model. The two-phase system is solved for the vapour and the liquid with non-condensable gas. Though this cavitation model is based on a highly simplified Rayleigh-Plesset equation for the radius of a spherical cavitation bubble, it is very robust and effective for the analysis of the problems, where the void fraction is highly important in forming flow structures but where the details of the bubble dynamics (especially the collapses and rebounds) are not of primary interest [5].

The SST-SAS turbulence model [19] has been employed to capture the highly unsteady phenomena inside the pump. It represents a good compromise between widely used SST model and LES or DES (Detached Eddy Simulation) models, which are too costly for comprehensive turbomachinery applications. The SST-SAS model requires a sufficiently high near-wall resolution of the computational grid. It could not be guaranteed in the whole computational domain and for every flow coefficient. That is why the computational grid is highly refined especially close to the impeller walls and in front of the impeller. The automatic near-wall treatment implemented inside the ANSYS CFX code has been applied to switch from the low-Reynolds number wall formulation to the wall functions when the near-wall resolution become insufficient. The computational grids represent approximately 2 mil. grid nodes for both pump models.

Fully transient calculations have been applied with the time step chosen according to the angular speed Ω ; there are 180 time steps during one shaft revolution, which means that every time step the rotor rotates by 2 degrees. With such a time step three inner coefficient loops per step are sufficient to reach a convergent solution. Two transient rotor-stator interfaces have been used; one between the inlet piping and the impeller and one between the impeller and the diffuser or volute. To control the flow coefficient, the net positive suction head as well as the turbulence during calculation, we have prescribed the total pressure together with the flow direction, the 5% turbulence intensity and the zero void fraction at the inlet of the computational domain and the bulk mass flow rate at the outlet.

Fig. 3 Measured noise emissions. Pump with volute

Fig. 4 NPSHi (defined clearly in chapter 3) at different flow coefficients. Pump with volute. CFD

Fig. 5 Curves of NPSH required for cavitation inception, 3% drop of pump total head and cavitation breakdown. Pump with volute. CFD

3. CFD Analysis of the Overall Cavitation Performance of Pumps

As it was mentioned in the previous chapter, we can expect, that the level of the noise and vibrations is closely connected to the cavitation phenomena. Moreover, these phenomena can depend not only on the flow rate ϕ/ϕ_{BEP} , but also on the NPSH value. For better understanding, Fig. 5 shows three important cavitation curves: $NPSHi(\phi)$, $NPSH3(\phi)$ and $NPSHb(\phi)$, where $NPSHb$ is the net positive suction head required for the cavitation breakdown. It is important to define how the $NPSHi$ curve was obtained. In practice, there are different criteria when the cavitation inception occurs. They are based either on the static pressure, which has to drop below the saturated vapour pressure on an agreed minimum area, or on the calculated void fraction. We used the second criterion (which seems to be more correct) and required to reach the value of void fraction $\alpha = 0.1$, no matter what the area is. In Fig. 5 we can see, that close to ϕ_{BEP} , $NPSH3(\phi)$ and $NPSHb(\phi)$ curves approach each other, but going to the right (or especially to the left) the difference $NPSH3 - NPSHb$ increases. This is caused by a different character of the head-drop curves. Close to ϕ_{BEP} the head-drop curves are very steep at $NPSH = NPSH3$ while at suboptimal flow coefficients the gradient decreases, as we can see in Fig. 6 - 7 which show the comparison of the head-drop curves for $\phi/\phi_{BEP} = 0.64$ and $\phi/\phi_{BEP} = 1.03$ obtained both with the CFD

analysis and the experiment. In Fig. 5 we can also see, that for the flow coefficients $\varphi \geq \varphi_{BEP}$ the distance between NPSHi and NPSH3 does not change too much; it is in the range from 14 to 17 m. However for suboptimal flow coefficients the distance rapidly grows and reaches its maximum at about 70% of φ_{BEP} .

Figures 8 - 13 describe the cavitation regions inside the impeller for different flow rates and different values of NPSH. We used figures from the pump with the axial diffuser. Nevertheless the pictures of cavitation regions inside the impeller of the volute pump do not differ considerably. In fact changes of these regions in time are more visible, so we tried to find some "average" pictures for each flow condition. Figure 8 shows the places on the blades where the cavitation starts under $NPSH = NPSHi$. To make the cavitation regions more visible, we visualized the isosurfaces, on which the void fraction α reaches 7% (though $\alpha_{max} = 0.1$ on a very small area). In Fig. 9 - 13 the cavitation regions in the impeller can be seen for NPSH values of NPSH3 and NPSHb. For every value of NPSH and φ/φ_{BEP} there are two pictures. The first one describes the isosurfaces, on which the void fraction α reaches 10%. This is a numerical equivalent of the experimental visualization of the cavitation clouds. The second one shows the isosurfaces, on which the void fraction reaches 90%. In other words, it shows the cavitation caverns filled with the void fraction only. These caverns behave as the obstacles changing the shape of the blade-to-blade passages of the impeller. As the flow coefficient increases, they are moving from the suction side to the pressure side of the blades. For NPSH equal to NPSH3, the cavitation regions are increasing when the flow rate φ/φ_{BEP} rises from the value of 0.55 and reach the maximum size at $\varphi/\varphi_{BEP} = 0.93$. Then with following increase of the flow rate, the cavitation regions decrease and move to the pressure side of the blades. For NPSH equal to NPSHb, the cavitation regions are practically the same size for $\varphi/\varphi_{BEP} = 0.55$ and $\varphi/\varphi_{BEP} = 0.74$, then with increasing flow rate they decrease and reach the minimum size at $\varphi/\varphi_{BEP} = 1.1$. With following increase of the flow rate the cavitation regions move to the pressure side of the blades and start to increase their size. At the flow rate of 1.1 there is practically no difference between cavitation regions for NPSH values of NPSH3 and NPSHb.

Fig. 6 Head-drop curves for $\varphi/\varphi_{BEP} = 0.64$. Comparison of CFD analysis and experiment. Pump with volute

Fig. 7 Head-drop curves for $\varphi/\varphi_{BEP} = 1.03$. Comparison of CFD analysis and experiment. Pump with volute

Fig. 8 Cavitation inception ($\alpha_{max} = 0.1$). Regions in the impeller with void fraction over 7%

4. CFD Analysis of Dynamic Behaviour of Pumps Due to Cavitation

Based on the results presented in the previous chapter we decided to perform detailed analysis of the cavitation dynamics at the flow rate $\varphi/\varphi_{BEP} = 0.7$ for both pump variants. As the measurements of the pressure pulsations and vibrations are available only for the volute pump, most results in this paper are devoted to this one. There are two ways of presenting the highly unsteady cavitation phenomena in the pump: prepared videos and frequency analysis. Unfortunately videos are not suitable for documents

Fig. 9 Cavitation regions in the impeller. Pump with axial diffuser, $\phi/\phi_{BEP} = 0.55$

Fig. 10 Cavitation regions in the impeller. Pump with axial diffuser, $\phi/\phi_{BEP} = 0.74$

Fig. 11 Cavitation regions in the impeller. Pump with axial diffuser, $\phi/\phi_{BEP} = 0.93$

Fig. 12 Cavitation regions in the impeller. Pump with axial diffuser, $\phi/\phi_{BEP} = 1.1$

Fig. 13 Cavitation regions in the impeller. Pump with axial diffuser, $\phi/\phi_{BEP} = 1.3$

Fig. 14 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.7$, $NPSH > NPSHi$. CFD analysis

Fig. 15 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.7$, $NPSH = 9$ m. CFD analysis

Fig. 16 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.7$, $NPSH = NPSH3$. CFD analysis

Fig. 17 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.7$, $NPSH = NPSHb$. CFD analysis

distributed usually in PDF format, so we have to replace them with only several figures. As far as the frequency analysis is concerned, it is based on the Fast Fourier Transformation (FFT). To be able to link different frequencies with cavitation phenomena, we first analysed the spectrum of the discharge pressure for the regime with NPSH value greater than NPSHi. The results are shown in Fig. 14. Here we can see, that the dominant frequency is $4f_{\Omega}$ (impeller blade frequency). It can be linked to the pressure pulsations created when the trailing edges of the blades pass the volute tongue. We can also find increased amplitudes close to the frequency of $3f_{\Omega}$ and in the vicinity of the shaft frequency. As far as the amplitudes in the vicinity of f_{Ω} are concerned, they reflect the fact that the calculated flow inside the impeller passages is not perfectly periodic. The frequency of $3f_{\Omega}$ corresponds to the periodic changes in the impeller inlet recirculation, which is located close to the blade tips. Figure 15 shows the frequency spectrum of the discharge pressure for the regime with NPSH equal to 9 m, which is the same value as used during pump measurements. The result does not differ much from Fig. 14, only the amplitudes close to the shaft frequency start to separate into two peaks.

The situation is quite different in the case $NPSH = NPSH3$ (Fig. 16). The dominant frequencies can be found at the values of about $0.44f_{\Omega}$ and $3f_{\Omega}$ and we can also see the increased amplitude close to the value of $1.44f_{\Omega}$. The lowest frequency $0.44f_{\Omega}$ can be connected with the cavitation clouds, which separate periodically from the backflow region in front of the impeller. The second frequency which characterizes the periodic changes of the cavitation in the backflow region is $3f_{\Omega}$. Another periodic phenomenon which was detected is connected with the cavitation regions on the suction side of the impeller blades. They change their size and shape with the frequency of $1.44f_{\Omega}$. In the following step we analyzed the frequencies for NPSH equal to NPSHb (Fig. 17). It is interesting that the frequencies of $0.44f_{\Omega}$ and $3f_{\Omega}$ are suppressed in this case. We have found that in this regime the cavitation clouds remarkably block the impeller passages and so the backflow in front of the impeller practically vanishes (Fig. 18 - 22). We can therefore conclude that frequencies of $0.44f_{\Omega}$ and $3f_{\Omega}$ are closely linked to the interaction of cavitation and the backflow in front of the impeller. In Fig. 23 we can see the separation of the cavitation clouds in front of the impeller, which is adjusted in a fixed position. The time interval between two successive pictures represents the rotation of the impeller by 10° . The changes of the cavitation regions on the suction side of the impeller blades are shown in Fig. 24. The time interval between two successive pictures represents the rotation of the impeller by 65° .

We have already mentioned that we can compare calculations with the measurements of pressure pulsations and vibrations of the volute pump. Unfortunately the results of the measurements are available only for $NPSH = 9$ m. In Fig. 25 we can see the FFT analysis of the measured discharge pressure pulsations. There is a dominant frequency of $4f_{\Omega}$ (impeller blade frequency) and increased amplitudes can be found also close to the frequency of $3.3f_{\Omega}$, which is similar to the frequency of $3f_{\Omega}$ in the CFD analysis. Unlike the calculations, the measurements did not find increased amplitudes in the vicinity of the shaft frequency. Concerning the measured vibrations, they are shown in Fig. 26. There are two dominant frequencies: the shaft frequency and the impeller blade frequency. We can also find a small peak at the frequency of $1.2f_{\Omega}$ and increased amplitudes between the values of

Fig. 18 Backflow regions in the impeller. Pump with axial diffuser, $\phi/\phi_{\text{BEP}} = 0.55$

Fig. 19 Backflow regions in the impeller. Pump with axial diffuser, $\phi/\phi_{\text{BEP}} = 0.74$

Fig. 20 Backflow regions in the impeller. Pump with axial diffuser, $\phi/\phi_{\text{BEP}} = 0.93$

Fig. 21 Backflow regions in the impeller. Pump with axial diffuser, $\phi/\phi_{\text{BEP}} = 1.1$

Fig. 22 Backflow regions in the impeller. Pump with axial diffuser, $\phi/\phi_{\text{BEP}} = 1.3$

$3.3f_\Omega$ and $3.7f_\Omega$. As far as the shaft frequency is concerned, we suppose it rises from the mechanical reasons (impeller imbalance and coupling). To compare Fig. 26 with the CFD analysis we calculated the axial and radial forces acting on the impeller (Fig. 27) for $NPSH = 9$ m. The frequency analysis of the radial forces, which are the main source of vibrations, shows three dominant frequencies. The most important is the impeller blade frequency $4f_\Omega$, quite high amplitudes can be found at the frequencies of $1.4f_\Omega$ and $8f_\Omega$. Increased amplitudes can be also found close to the frequencies of $0.44f_\Omega$, f_Ω and $3f_\Omega$. When the value of $NPSH$ drops to $NPSH_3$, the FFT analysis of the calculated radial forces becomes practically unchanged (Fig. 28) while the amplitudes of the axial forces change dramatically, especially close to the frequencies of $0.44f_\Omega$ and $3f_\Omega$.

We also tried to compare the FFT analysis of the calculated discharge pressure pulsations as well as the forces acting on the impeller of the volute pump and the pump with the axial diffuser. The results are very similar with exception of the impeller blade frequencies $4f_\Omega$ and $8f_\Omega$, which practically disappear. It can be seen e.g. from Fig. 28 - 29 showing the FFT analysis of the forces acting on the impeller of both pump configurations at the same flow conditions.

5. Discussion of cavitation instability in backflow vortices

The most pronouncing frequency that we have discovered ($0.44f_\Omega$) is connected with the values $NPSH = NPSH_3$ and $\phi/\phi_{BEP} = 0.7$. As we mentioned in the previous chapter, it is closely linked to the interaction of the cavitation and the backflow in front of the impeller. The mechanism of the cavitation in backflow vortices formed due to the shear layer between the swirling backflow and straight main flow is described in Fig. 30 – 31, which come from the works of Tsujimoto and his coauthors. Figure 32 shows the position of the calculated backflow vortices for the values $NPSH = NPSH_3$ and $\phi/\phi_{BEP} = 0.7$. It has been already mentioned, that in the regime $NPSH = NPSH_b$ and $\phi/\phi_{BEP} = 0.7$ the cavitation clouds remarkably block the impeller passages and so the backflow in front of the impeller practically vanishes. The frequencies of $0.44f_\Omega$ and $3f_\Omega$ are suppressed in this case.

In [20] we examined the dynamic behaviour of cavitation phenomena in the suction part of a radial-flow multistage pump with the three-blade inducer. Here the unsteady cavitation in the backflow vortices was obtained with the dominant frequency $f_1 = 0.166 nf_\Omega$, where $n = 3$ is the number of backflow vortices. This corresponded well with the observations by Tsujimoto [14] who measured the dominant frequency $f_1 = 0.16 nf_\Omega$, $n = 5$ in the three-blade inducer. We concluded that URANS based models of turbulence (including the SST-SAS model) were not able to predict the number of backflow vortices different from the number of inducer blades. The same is valid in this study concerning cavitating flow in the mixed-flow pump. We could detect four vortices separating periodically in the backflow region. So the dominant frequency of $0.44f_\Omega$ can be written this way: $f_1 = 0.11 Z_1 f_\Omega$, where $Z_1 = 4$ is the number of the impeller blades.

Fig. 23 Cavitation clouds created in backflow vortices.
 $\phi/\phi_{BEP} = 0.7$, $NPSH = NPSH_3$. CFD analysis.
Interval between pictures is 10° of impeller rotation.
Pictures arranged left to right, up to down

Fig. 24 Changes of cavitation regions on impeller blades.
 $\phi/\phi_{BEP} = 0.7$, $NPSH = NPSH_3$. CFD analysis.
Interval between pictures is 65° of impeller rotation.
Pictures arranged left to right, up to down

Fig. 25 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.7$, NPSH = 9 m. Experiment

Fig. 26 FFT analysis of vibrations. Pump with volute. $\phi/\phi_{BEP} = 0.7$, NPSH = 9 m. Experiment

Fig. 27 FFT analysis of forces acting on the impeller. Pump with volute. $\phi/\phi_{BEP} = 0.7$, NPSH = 9 m. CFD analysis

Fig. 28 FFT analysis of forces acting on the impeller. Pump with volute. $\phi/\phi_{BEP} = 0.7$, NPSH = NPSH3. CFD analysis

Fig. 29 FFT analysis of forces acting on the impeller. Pump with axial diffuser. $\phi/\phi_{BEP} = 0.7$, NPSH = NPSH3. CFD analysis

As it can be seen from Fig. 18 - 22 the backflow region in front of the impeller increases considerably with decreasing flow rate, except the regimes with the NPSH values close to NPSH_b. To find how this increase of the backflow region influences the cavitation instability in the backflow vortices, we performed the FFT analysis of the discharge pressure at the flow conditions NPSH = NPSH₃ and $\phi/\phi_{BEP} = 0.55$. The comparison of the FFT analysis done for two flow rates $\phi/\phi_{BEP} = 0.7$ and $\phi/\phi_{BEP} = 0.55$ and for the NPSH value of NPSH₃ is shown in Fig. 33 - 34. It can be seen that the amplitude of the frequency $f = 0.44f_\Omega$ considerably increased with the decreasing flow rate while the frequency of $3f_\Omega$ became weaker than the impeller blade frequency. As far as the frequency of $1.4f_\Omega$ (connected with the changes of the cavitation regions on the suction side of the impeller blades) is concerned, its amplitude is slightly lower than the one for $\phi/\phi_{BEP} = 0.7$. The reason is probably that at the flow conditions NPSH = NPSH₃, $\phi/\phi_{BEP} = 0.55$, the cavitation regions on the suction side of the impeller blades are smaller than those for the flow rate $\phi/\phi_{BEP} = 0.7$ (Fig. 9 – 10).

6. Conclusion

The presented CFD analysis has provided a good picture of the cavitation structures inside the pump and their dynamics for a wide range of flow coefficients and NPSH values, which would be very difficult and expensive to obtain through the experimental research. Cavitation instabilities were detected at the flow rate of $0.7\phi_{BEP}$ close to the NPSH₃ value. The frequencies of $0.44f_\Omega$ and $3f_\Omega$ are closely linked to the interaction of the cavitation and the backflow in front of the impeller. The frequency of $1.4f_\Omega$ is connected with the dynamics of cavitation regions on the suction side of the impeller blades. They change their size and shape, as shown in Fig. 24. The calculated cavitation performance $\psi(\phi, NPSH)$ has been compared with the available measurements in a closed horizontal plane loop with a good agreement. The calculated results and their frequency analysis also correspond fairly well to the measured noise, pressure pulsations and vibrations.

Fig. 30 Cavitation in backflow vortices
(credit Y. Tsujimoto)

Fig. 31 Picture of backflow vortex cavitation
in inducer (credit Y. Tsujimoto)

Fig. 32 Cavitation in backflow vortices $\phi/\phi_{BEP} = 0.7$, NPSH = NPSH₃. CFD analysis.
Cavitation regions are blue coloured (picture on the left) or white coloured (picture on the right).
Backflow regions are red coloured

The FFT analysis of the calculated discharge pressures and forces is based on the dataset of 2048 timesteps, which represents about 11.4 impeller revolutions. About ten more impeller revolutions were necessary to obtain initial periodic solution. It requires quite large CPU times for every flow regime. It can be seen from the graphs of the FFT analysis that 2048 timesteps are not

enough to get a really fine resolution of the frequencies and corresponding amplitudes. Also those ten initial revolutions may be insufficient to provide a perfectly periodic solution and that is why the lowest calculated frequencies can be influenced by the computational reasons. Nevertheless the amplitudes corresponding to the frequencies of $0.44f_\Omega$, $1.4f_\Omega$ and $3f_\Omega$ (regime $\phi/\phi_{BEP}=0.7$, $NPSH = NPSH3$) are high enough to be linked to the cavitation and flow phenomena, not to the numerical errors. The experimental verification and visualization is nevertheless planned in the close future.

Fig. 33 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.7$, $NPSH = NPSH3$. CFD analysis

Fig. 34 FFT analysis of discharge pressure. Pump with volute. $\phi/\phi_{BEP} = 0.55$, $NPSH = NPSH3$. CFD analysis

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Nomenclature

f_Ω	Shaft rotational frequency [s^{-1}]
F	Force acting on impeller [N]
N	Specific speed [-]
$NPSH$	Net positive suction head [m]
$NPSH3$	Net positive suction head required for 3% drop of the pump total head [m]
P	Pressure [Pa]
R	Radius [m]
Z_I	Number of impeller blades [-]
α	Void fraction [-]
δ	Tip clearance [m]
ρ	Density of fluid [$kg \cdot m^{-3}$]
ϕ	Inlet flow coefficient [-]
ψ	Head coefficient [-]
Ω	Angular speed [$rad \cdot s^{-1}$]

Subscripts

ax	Axial
b	Breakdown
BEP	Best efficiency point
exp	Experimental
i	Inception
I	Impeller
out	Pump outlet
rad	Radial
T	Tip
1	Impeller inlet
2	Impeller outlet

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