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3D NUMERICAL ANALYSIS OF THE IMPELLER – STATOR INTERACTION INTO A STORAGE PUMP

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Lucrarea prezintă studiul numeric al interacţiunii rotor-stator într-o pompă dublu-etajată şi cu dublu flux. Mişcarea tridimensională a apei s-a considerat absolută şi permanentă în părţile fixe ale pompei (de exemplu, în stator), respectiv s-a considerat relativă şi permanentă în rotoare. Ecuaţiile de mişcare s-au rezolvat printr-o mediere de tip Reynolds (solver RANS), iar turbulenţa a fost luată în considerare printr-un model k–ω SST. Rezultatele au arătat că distribuţia presiunii pe palele statorice este influenţată de prezenţa palelor din amonte şi s-a propus o soluţie de diminuare a acestei interacţiuni.

The paper presents the impeller-stator interaction into a storage pump with two stages and double entry using numerical investigation. Three-dimensional steady absolute flow is assumed in pump's fixed parts (like stator) and for the impellers we considered three-dimensional steady relative flow. The numerical solution of the equations is obtained using a Reynolds averaged Navier-Stokes (RANS) solver and the turbulence viscosity is computed using k–ω SST model. The pressure distribution on the stator blade is perturbed in spanwise direction due to an inappropriate flow angle on the stator leading edge.

Keywords: CFD, impeller – stator interaction, mixing interface, storage pump

1. Introduction

The turbomachinery flow is unsteady due to the relative motion between different components of the machine, for example the impeller blade passing in front of the stator vanes or in front of the tongue of the volute. Furthermore, in hydraulic machines the flow is fully turbulent and three-dimensional [1, 2].

Computing the entire real flow (unsteady and turbulent) through the whole pump requires a large computer memory and computation time even for the most performing computers. Thus, a simplified simulation technique must be used in order to obtain useful results in a storage pump [3].

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Up to now CFD methods were applied to single components of storage pumps with one or two impellers [4, 5]. Also, the new developments offer the possibility to compute the flow in components. The perturbation generated by the suction elbow due to the complex geometry is ingested by the impeller [6, 7].

The paper presents our evaluation of the technical solution implemented into the storage pump with double entry and two stages installed in Romania in order to mitigate the strong impeller–stator interaction. Consequently, the 3D flow into a storage pump is performed then impeller–stator interaction is analyzed. The conclusions and perspectives of the research are outlined in last section. Our main goal is to improve technical solution in order to extend the storage pump life and reduce the maintenance and operation.

2. The storage pump

In the present work, the selected machinery is a centrifugal storage pump with two stages and double entry (see Figure 1) [8]. Both suction parts include an elbow with a complex three-dimensional geometrical shape. All impellers of the storage pump are identical from geometrical point of view having 7 blades each. The hydraulic passages of the storage pump between first impeller on each side and the double entry impeller comprises one stator with two rows of blades (stator and returning stay vanes), each having 9 blades. The flow from the outlet of the double entry impeller is ingested by the volute with 4 stay vanes (3 blades and the tongue).

Fig. 1. Double stage and double flux storage pump

3. Computational domains

The storage pump has a symmetry plane therefore the numerical investigation is performed through half of the pump. The computational domains are presented in Fig. 2. It consists in the entire suction elbow, one interblade channel for each of the two identical impellers and for the stator, and half of the volute. The 3D interblade channel is selected as computational domain based on periodic flow hypothesis taking into account that the numerical investigation is performed at best efficiency point (BEP) [1, 2].

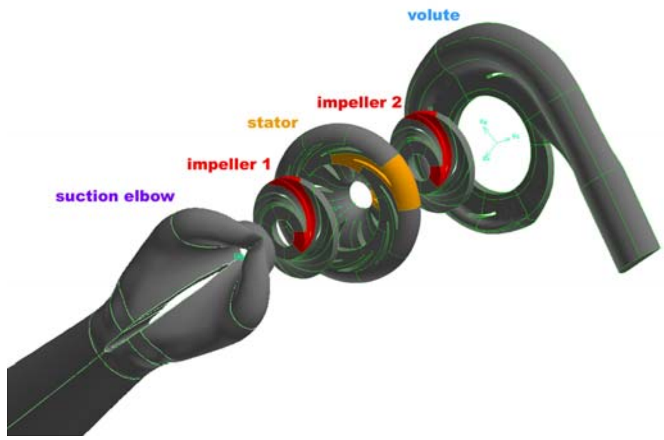

Fig. 2. The three-dimensional view of the half hydraulic passages of the storage pump: suction elbow, first impeller, stator, second impeller and the volute

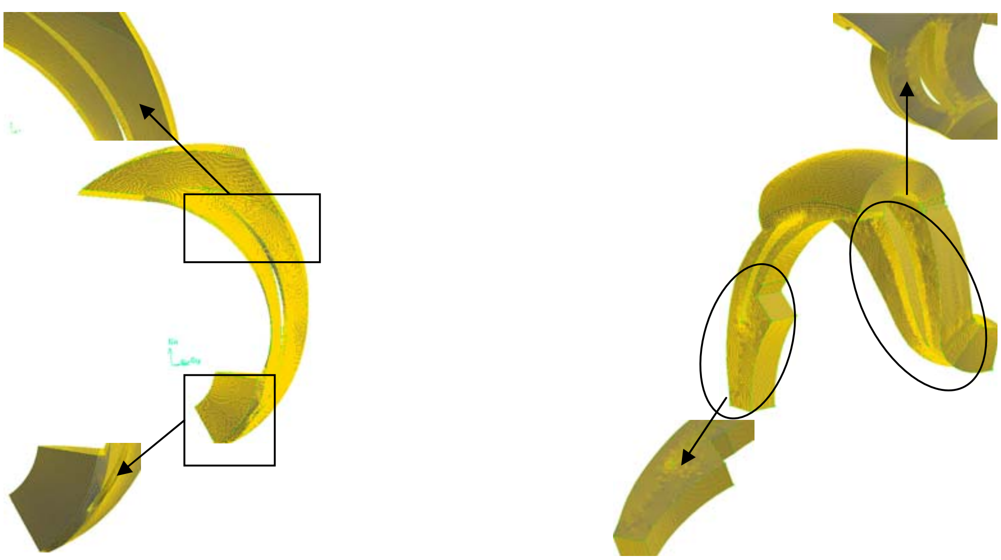

Fig. 3. 3D computational domains with its structured mesh: impeller (left) and stator (right).

A structured mesh with 475680 cells is generated on the computational domain of the first impeller (see Figure 3), while a structured mesh with 859380 cells is used on the stator computation domain using Gambit. The Mixing Interface Technique is used in order to take into account the impeller–stator interaction. This method implies an iterative process with passing the information from computational domain to another through the boundary conditions. The mixing algorithm infers a circumferentially average procedure that means the wakes are mixed [9].

4. Equations and boundary conditions

The hydrodynamic quantities calculus is based on the governing equations for the incompressible fluids flow (continuity and movement equations):

$$\nabla \cdot \vec{v} = 0 \tag{1}$$

$$\frac{\partial \vec{v}}{\partial t} + (\vec{v} \cdot \nabla)\vec{v} = \vec{f} - \frac{\nabla p}{\rho} + \nu \Delta \vec{v} \tag{2}$$

In the present investigation a *full 3D, stationary, viscous and incompressible fluid flow* is considered. Thus the equation (2) can be written as:

$$(\vec{v} \cdot \nabla)\vec{v} = \vec{f} - \frac{\nabla p}{\rho} + \nu \Delta \vec{v} \tag{3}$$

The viscosity coefficient is written as a sum between the molecular viscosity coefficient η and the turbulent viscosity coefficient η_T [3]: $(\eta + \eta_T)$. These equations are applied in this form *in the stator* analysis, where *steady absolute flow* is assumed. For *the impeller*'s analysis, *steady relative flow* is assumed. In a rotating frame of reference with angular speed $\omega = \omega\, k$, k being the unit vector of the pump axis direction. By introducing the relative velocity

$$w = v - \omega \times r\,, \tag{4}$$

with r the position vector, the left hand side of (3) becomes:

$$\nabla(\rho w w) + 2\rho\omega \times v + \rho\omega \times (\omega \times r). \tag{5}$$

The numerical solution of these equations is obtained with the expert code FLUENT 6.3, using a *Reynolds averaged Navier-Stokes (RANS) solver* and the turbulence viscosity is computed using *k–ω SST model*.

In order to have a well defined problem, the boundary conditions must be added. The boundary conditions are computed at best efficiency point with the parameters from Table 1.

Table 1

Operating point parameters of the pump

Parameter	**Symbol**	**Value**
Rotational speed	n [rot/min]	1000
Flow rate	Q [m^3/s]	3
Pumping head	H [m]	247
Hydraulic power	P_h [kW]	7269.219

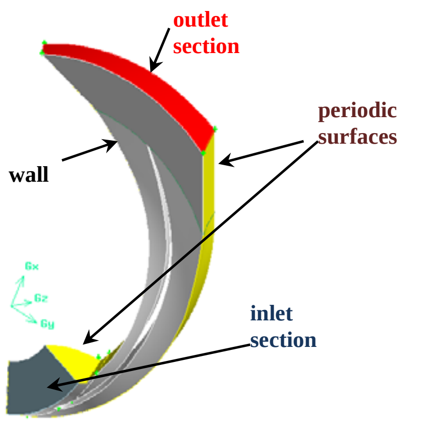

Fig. 4. Boundary conditions for the impeller

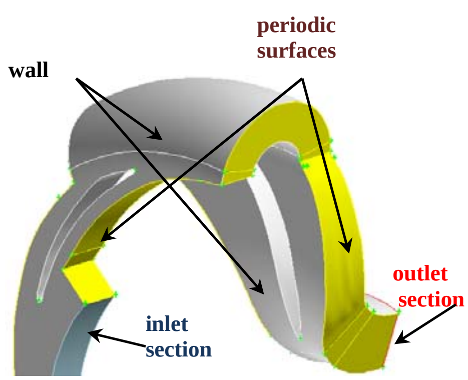

Fig. 5. Boundary conditions for the stator

In this case the following boundary conditions are taken into account for the impeller (see Figure 4) and for the stator (see Fig. 5):

– *inlet section*: the velocity components are imposed;
– *outlet section*: the pressure condition is imposed;
– *solid frontiers* are considered as impenetrable, which can be written $\vec{v} = 0$;
– *periodic surfaces*: on the surfaces of impeller and stator interblade channels, the periodicity of the velocity, pressure and turbulence parameters is imposed, as:

$$\vec{v}(r,\theta,z) = \vec{v}\left(r,\theta + \frac{2\pi}{N_R}, z\right), \tag{6}$$

$$p(r,\theta,z) = p\left(r,\theta + \frac{2\pi}{N_R}, z\right). \tag{7}$$

5. Numerical results

The pressure coefficient is defined according to the following equation:

$$c_p = \frac{p - p_{ref,I}}{\rho g H}. \qquad (8)$$

The color map of the pressure coefficient distribution on the impeller blade is shown in Fig. 6.

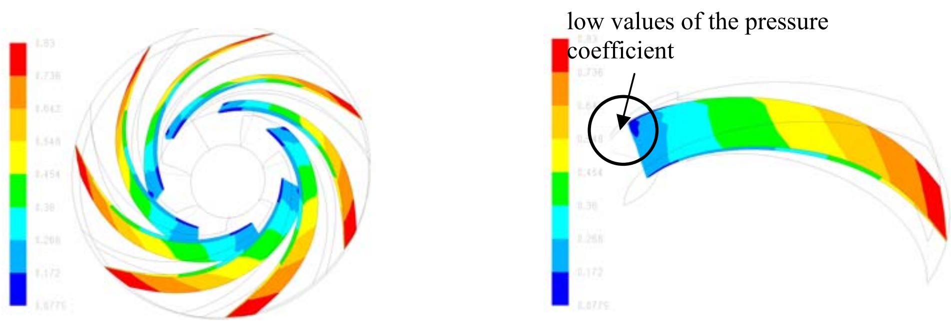

Fig. 6. Pressure coefficient distribution on the pressure side of the impeller blade and on the suction side of the impeller blade

It can be seen the blue spot on the blade pressure side, near the leading edge with minimum value of the pressure coefficient. This indicates that the flow has an impropriate incidence angle at the impeller inlet section, being orientated more on the suction side. Consequently, the leading edge shape is not aligned to the flow angle which means the discharge taken into account is smaller than discharge value. Also, it can be seen that there are only positive values in the pressure coefficient distribution, which indicates that there is no cavitation risk during the pump's operation for the analyzed point.

Fig. 7 shows the absolute velocity coefficient distribution on the mixing interface situated at dimensionless radius $r/R_2 = 1.001$, near to the outlet section of the first impeller displaced at $R_2 = D_2/2$. The absolute velocity coefficient was defined according to the equation: $c_v = v\big/\sqrt{2gH}$.

A view of absolute velocity coefficient distribution is shown on the cross section of the interblade channel, in Figure 7(left). It can be seen that is a pronounced non uniformity of the velocity field at the impeller outlet corresponding to the wake – jet phenomenon, as Prof. Brennen has shown [1].

The distribution of this velocity coefficient is plotted in terms of the dimensionless axial extension of the impeller on the impeller outlet ($b = B/D_2$). One can observe that the non uniformity is more significant near the shroud due to the meridian shape of the channel, in Figure 7(right). However, these circumferential non-uniformities are not passed forward in the stator due to mixing interface technique.

Fig. 7. Absolute velocity coefficient distribution on the impeller outlet

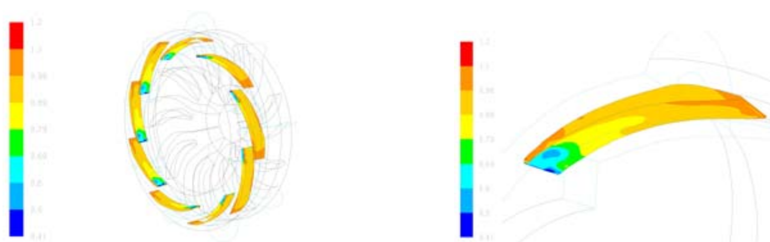

Fig. 8. Pressure coefficient distribution on the first blade of the stator

The pressure coefficient distribution on the first blade of stator is presented in Fig. 8. It can be observed the blue region on the suction side of the stator blade near to the leading edge. It indicates that the incidence angle on the stator inlet section is orientated on the blade pressure side. It is important to mention, the stator blade is straight while the trailing edge of the blade impeller is leaned. Consequently, the lean flow from impeller outlet impacts on the straight blade of the stator producing spanwise variation of the pressure distribution, see Figure 8. One solution to mitigate the impeller–stator interaction consists in increasing the clearance between impeller and stator.

6. Conclusions

The paper presents numerical analysis of the impeller–stator interaction between the first impeller and the first stator of a storage pump with two stages and double entry. Analyzing the flow parameters inside the first impeller of the pump it could be observed that the pressure coefficient distribution on the blades presents a non uniformity region, near the blade leading edge. This non uniformity was considered to be due to an impropriate incidence of the flow on the impeller blade. It was observed that the absolute velocity field on the impeller outlet section is strongly influenced by the presence of the blade, the wake-jet phenomenon being remarked. Also, the pressure coefficient distribution on the stator blade is perturbed in spanwise direction due to an impropriate flow angle on the stator leading edge. One solution to mitigate the impeller–stator interaction consists in increases the clearance between impeller and stator. Further, the flow field inside the entire pump will be analyzed in order to evaluate the energetic performances on the entire storage pump.

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