Numerical Prediction Of Torque On Guide Vanes In A Reversible Pump-Turbine

Turbine and pump mode operation

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***Abstract*—The paper presents the numerical flow analysis of the torque on guide vanes pivots in reversible pump-turbine (PT) in turbine and pump mode operation. Flow conditions in a typical PT often point out the importance of the unsteadiness of the flow, first of all because of its instability caused by the viscous effects – the boundary layer separation, vortex spreading and the influence between rotating and non-rotating parts. The detailed analysis of the time dependent distribution of torque on guide vanes is presented. The numerical analysis was performed for complete pump-turbine with various turbulence models. The numerical results were compared with the experimental results.**

Keywords—water turbine, reversible pump-turbine, CFD, guide vanes, dynamic chyracteristics

I. Introduction

For the numerical flow analysis in the complete reversible pump turbine (Fig. 1) the ANSYS-CFX15 [1] was used. The computational grids were prepared with ANSYS-ICEM Hexa and Turbo Grid mesh generator. The computational grid of the complete PT consists of around two million elements.

The quality of computational grid is an important condition for accurate numerical flow analysis results, particularly in case of dominant unsteady flow [2]. Grid refinement is very important near the walls. Beside on the grid refinement the special attention has been done also on the grid quality (y+), on mesh orthogonality and on expansion and aspect ratio. In the case of model size PT, the y+ near the guide vanes is usually 20 - 50.

First part of the paper presented the steady state analysis of the flow in turbine and pump mode and comparison with experimental results [3] obtained with model test. This part is important to define the accuracy of the numerical method.

Second part deals with unsteady calculation of rotor-stator interaction in pump mode to find out the time dependent distribution of the torque on guide vanes. The time step has been adequate to two degrees of the runner revolution. The computation is running for three complete runner revolutions – rotation angle F=1080^{o}.

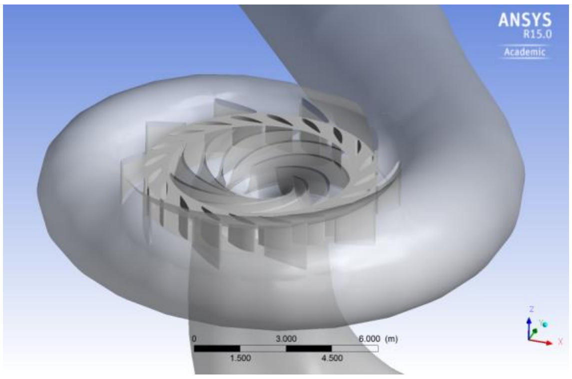

Fig. 1 Computational domain for pump turbine

The results of numerical and experimental method are compared for both regimes – turbine and pump mode.

II. Numerical Method

Turbulence models are used to predict the effects of turbulence in fluid flow without resolving all scales of the smallest turbulent fluctuations. A number of models have been developed that can be used to approximate turbulence based on the Reynolds Averaged Navier-Stokes (RANS) equations. Some have very specific applications, while others can be applied to a wider class of flows with a reasonable degree of confidence. The models can be classified as either eddy-viscosity or Reynolds stress models.

One of the most prominent turbulence models, the k-ε (k-epsilon) model, has been implemented in most general purpose CFD codes and is considered the industry standard model. It has proven to be stable and numerically robust and has a well established regime of predictive capability. For general purpose simulations, the k-ε model offers a good compromise in terms of accuracy and robustness.

While standard two-equation models, such as the k-ε model, provide good predictions for many flows of engineering interest, there are applications for which these models may not be suitable. Among these are: flows with boundary layer separation, flows with sudden changes in the mean strain rate, flows in rotating fluids and flows over curved surfaces. A Reynolds Stress model may be more appropriate for flows with sudden changes in strain rate or rotating flows, while the SST model may be more appropriate for separated flows, such as in pump mode operation. The SST model is recommended for high accuracy boundary layer simulations. The convergence behavior of the k-ω model is often similar to that of the k-ε model.

In flows where the turbulent transport or non-equilibrium effects are important, the eddy-viscosity assumption is no longer valid and results of eddy-viscosity models might be inaccurate. Reynolds Stress models naturally include the effects of streamline curvature, sudden changes in the strain rate, secondary flows or buoyancy compared to turbulence models using the eddy-viscosity approximation. Theoretically, Reynolds Stress models are more suited to complex flows, however, practice shows that they are often not superior to two-equation models. If convergence is difficult, it is recommended that a k-ε or k-ω based model solution be obtained first and then a Reynolds stress model solution can be attempted from the converged two-equation solution. It is frequently observed that Reynolds Stress models produce unsteady results, where two-equation models give steady state solutions. This can be correct from a physical standpoint, but requires the solution of the equations in transient mode.

Transient Rotor-Stator model should be used any time when it is important to account for transient interaction effects at a sliding (frame change) interface. It predicts the true transient interaction of the flow between a stator and rotor passage. In this approach the transient relative motion between the components on each side of the GGI connection is simulated. It ultimately accounts for all interaction effects between components that are in relative motion to each other. The interface position is updated each time step, as the relative position of the grids on each side of the interface changes. The principle disadvantage of this method is that the computer resources required are large, in terms of simulation time, disk space and quantitative post processing of the data. It is possible to start a transient rotor/stator computation from a simple initial guess, or from an existing prediction. If you are interested in the start-up transient of the machine, then start from the appropriate physical initial conditions. If you are interested in simulating a periodic-in-time quasi-steady state, then it may be helpful to first obtain a steady state sliding interface solution using Frozen Rotor interfaces between components.

The simulations presented in the paper were carried out with the commercial solver Ansys CFX, using two-equation RANS turbulence models: the standard k-ε turbulence model, the shear-stress-transport (SST) model, as well as the ε-based Speziale-Sarkar-Gatski Reynolds stress model (SSG RSM). Additionally, one advanced model was used: the scale-adaptive-simulation SST turbulence model (SAS SST).

The SAS SST turbulence model [4] is a so-called second generation URANS model, according to the classification. The model is essentially the SST turbulence model with an additional source term in the ω transport equation. The term can detect the unsteadiness of the solution through the comparison of the RANS length scale to the von Karman length scale. The result of the unsteadiness is a decreased turbulent kinematic viscosity. Thus, the SAS SST develops LES-like solution in unsteady regions. If the time step size is too large the unsteady structures can't be resolved and the model obtains a URANS or RANS solution.

III. Pump Turbine Operation

In pump turbine rotor-stator interaction is often the main cause for flow instabilities and unsteady flow. In pump mode the flow has diffuser effects and is highly turbulent.

In the first part is presented the calculation of losses in the flow domain of the PT in turbine mode. A detailed numerical analysis of the losses in each part of the PT: spiral casing and stay vanes, guide vanes, runner blades and draft tube is very important for further investigation. The accuracy of the numerical results is confirmed by comparison of numerical and experimental results of efficiency distribution (Fig. 2).

Harmful phenomena in the pump mode are usually vibrations which dependents on the discharge and the inlet angles of the guide vanes. Vibrations in general occur as a consequence to the rotor-stator interaction.

To obtain reliable cause of the vibrations, the pressure pulsations were analyzed with unsteady flow calculation methods. This can be done using different two-equation turbulence models; such as the k-ω SST (Shear Stress Transport) turbulence model, which well describes the flow around the blades and ever important flow separations, and accounts for the transport of the turbulent shear stress. The time step at such computations should be 1 to 3 degrees of runner rotating angle. There should be enough time steps, so the runner turns several times.

The experimental results necessary to make comparison between numerical and experimental results [3] was obtained on the model tests which comply with all IEC 60193 requirements for PT. All hydraulic and energy parameters, cavitation, pressure pulsations, momentums on the guide vane axis, as well as axial force, were tested.

The numerical analysis in turbine mode was performed for six different relative guide vane openings. The comparison of efficiency distribution is presented on the Fig. 2.

The numerical analysis in turbine mode was done to test the quality of the computational grid and the accuracy of the numerical method. The comparison of experimental and numerical results shows that the difference is close to the average difference for such type of turbine.

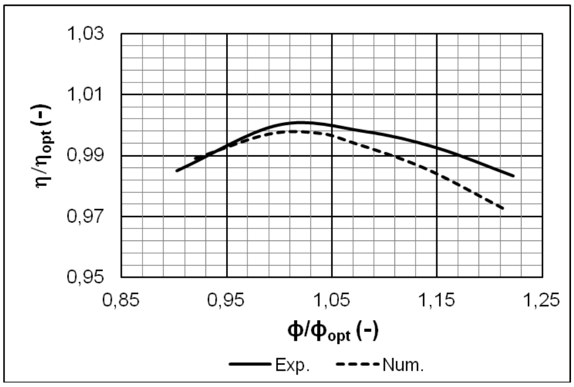

Fig. 2 Comparison between experimental and numerical results

In every day numerical analysis engineers use steady state methods to predict different characteristics as fast as possible. But such approach is not acceptable for all applications. Usually torque on guide vanes in turbune mode is analysed using steady state analysis and developers and researchers do not put enough attention to unsteady effects.

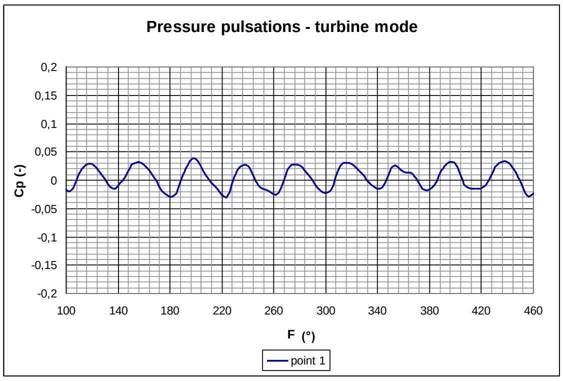

Fig. 3 Pressure pulsations for turbine mode

Pressure pulsations between runner blades [5] and guide vanes show difference in stability during time dependent observation of turbine and pump mode operation. Time is presented with runner revolution angle - F. Pump mode operation is much more unstable (Fig. 4) than turbine mode (Fig. 3) and consequently the reliable numerical results can be obtained only using unsteady analysis.

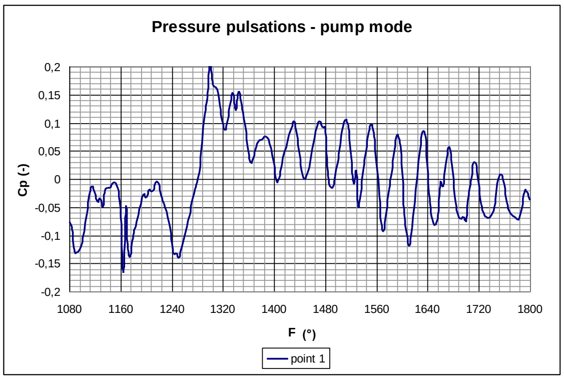

Fig. 4 Pressure pulsations for pump mode

On the following figures are presented time dependent torque on the shaft Fig. 5 during convergence process and oscillations of efficiency Fig. 6 in pump mode operation. Both results show unsteady behavior of the flow and necessity of minimal number of time steps.

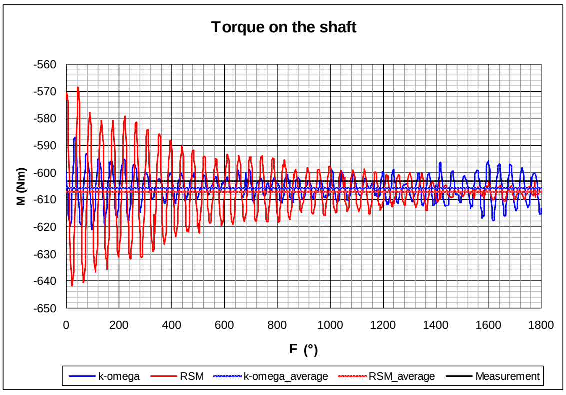

Fig. 5 Torque on the shaft

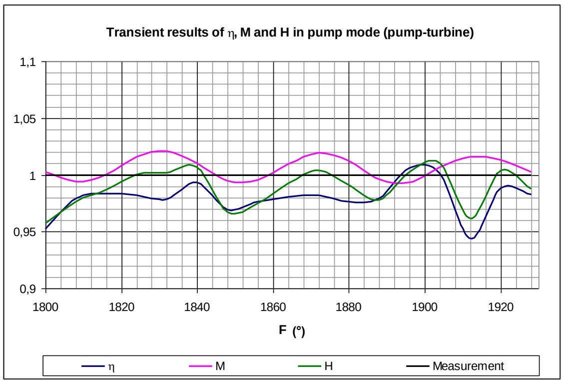

Fig. 6 Oscillations of efficiency in pump mode

Vortices and flow separations are well presented on Fig. 7, where the velocity distribution is shown on the middle cross sections inside the distributor and stay vanes for three different turbulence models: SST, SSG RSM and SAS SST.

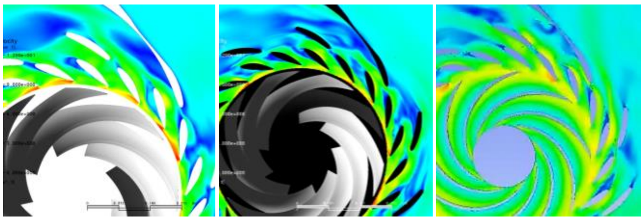

Fig. 7 Velocity distribution for different turbulence models

IV. Torque on Guide Vanes

The numerical analysis in pump mode was performed for two different runners at the same relative guide vane opening and with different turbulence models. From numerical analysis of the torque on guide vanes in reversible pump turbine (Fig. 8) in pump mode can be find out the big difference between steady and unsteady calculation. The first result of the torque on guide vanes for turbine mode operation is presented on Fig. 9. The comparison of numerical and experimental results shows the minimal and maximal value of the torque (two black lines) and amplitude of the torque oscillations obtained by measurement and oscillations of torque for all twenty guide vanes obtained by numerical analysis.

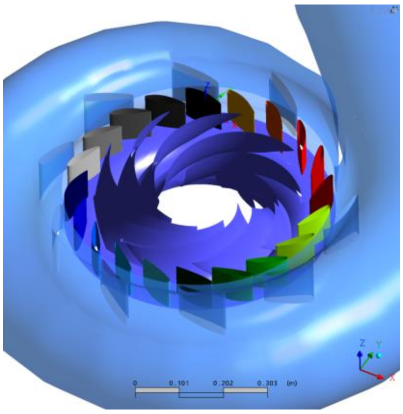

Fig. 8 Guide vanes positions

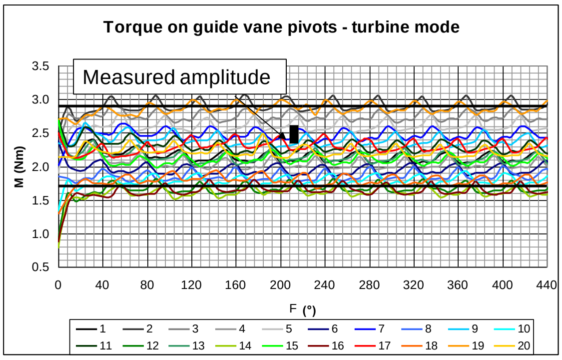

Fig. 9 Torque on guide vanes - turbine mode

The significant problem in pump mode operation is that the runner and guide vanes are exposed to the periodic pressure pulsation caused by the rotor-stator interaction. The numerical analysis in pump mode was performed for one relative guide vane opening with three turbulence models. From numerical analysis of reversible pump turbine in pump mode was found out the significant difference of the torque on guide vane pivots between steady and unsteady calculation. The main problem in steady state calculation is the convergence, mostly because of dominant unsteady flow behavior.

The transient calculation was started from the initial value obtained with steady analysis. After the first revolution of the runner the results are still unstable, so the calculation proceeds for five complete runner revolutions. The time dependent torque on all twenty guide vanes between 4.5 and 5 revolutions is presented on the Fig. 10, Fig. 11 and Fig. 12 for the model size of pump turbine which is equal to 0.350 mm at pump inlet and for three turbulence models. Black lines on all three figures show the limits between minimum and maximum torque obtained from measurements.

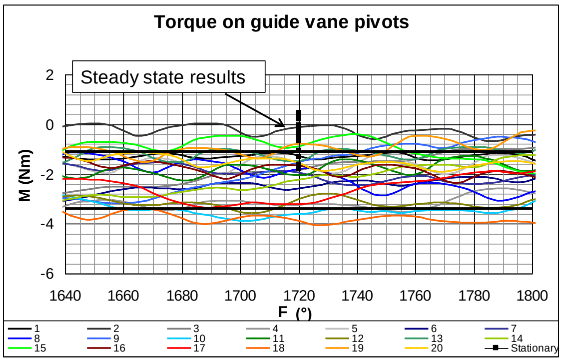

Fig. 10 Torque on guide vanes - pump mode (SST)

The absolute values of the torque obtained by unsteady calculations are different comparing to steady calculation. The results of steady calculation are presented by dots and for all turbulence models the results are almost completely out of the range, obtained by the measurements (black lines).

From all different analysis it was determined that the number of time steps for one runner revolution should be at least 180 – time steps should not be larger than two degrees of revolution.

The values of the torque obtained by unsteady calculations are much bigger in comparison with steady calculation. In our case the average value of the torque has been up to five times smaller with steady calculation. From all different analysis we determined that the number of time steps have to be at least equivalent to two or three complete runner revolutions. It means at least 740 time steps and consequently several thousand iterations.

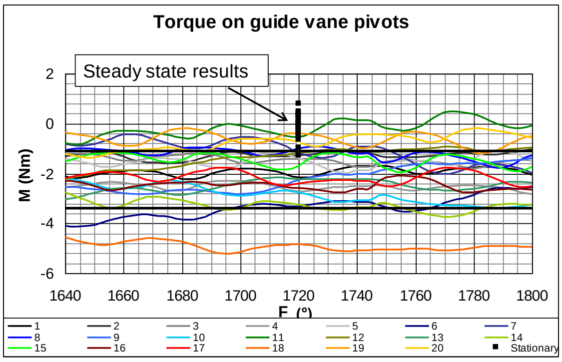

Fig. 11 Torque on guide vanes - pump mode (SSG RSM)

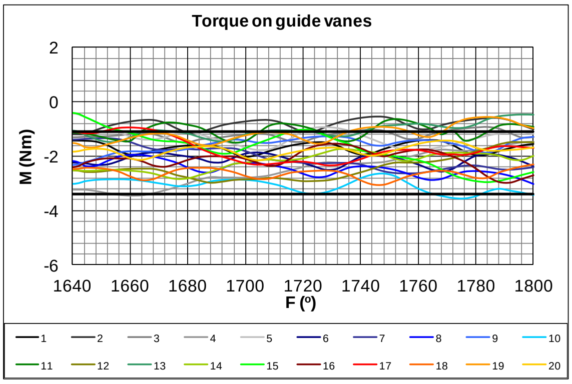

Fig. 12 Torque on guide vanes - pump mode (SAS SST)

The same calculation was done for the runner with changed outlet angle and the obtained results are very similar. Usually steady state analysis is done to obtain the energetic characteristics and also torque on the guide vanes pivots. In turbine mode steady state results are quite accurate. A lot of comparisons between experimental and numerical results for radial and axial turbines has been done in past years. But in the present research work is found out that for reversible pump turbine in pump mode the steady state results of torque on guide vanes pivots are completely wrong.

The flow in diverged geometries is dominantly unsteady and one of unsteady numerical method is necessary. Another important issue is the length of the time step and total number of time steps. Usually unsteady analysis is very time consuming and that is why the calculations is normally finished after executing too small number of time steps. If the time step is presented as a part of runner rotating time, for reliable results at least three complete runner revolutions is necessary to obtain stable converged and sufficiently accurate results of torque on guide vanes pivots.

The comparison between experimental and numerical analyses show good agreement between time dependent solution of torque amplitudes for guide vanes and maximal values of experimentally obtained torque amplitude (Fig. 13).

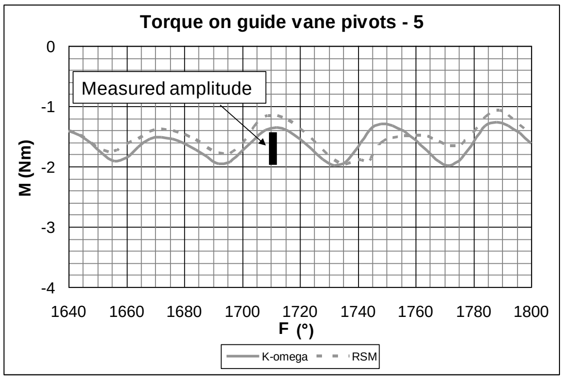

Fig. 13 Detailed torque distribution for guide vane nr. 5 – comparison between two turbulence models and between numerical and experimental amplitude

In some cases also three complete runner revolutions is not enough to obtain stable result curves. The reason is the strong unsteady flow dependent on the shape of the profiles of the guide and stay vanes, the shape of spiral casing and especially the shape of the spiral casing tongue. In all cases the torque characteristics for the guide vanes near the spiral casing tongue are very arbitrary. For all guide vanes the time dependent characteristic of the torque gives us the same frequency equal to the

$$\nu = nz \, ,$$

where n is the rotational speed of the runner and z is the number of runner blades. The average values of the torque for each guide vanes are much different, up to the three times. The amplitudes of the torque fluctuations are very similar for the most of the guide vanes. The difference between model and more than ten times bigger prototype shows some discrepancies at some guide vanes. Also the comparison between different turbulence models shows the similar differences. The difference between the smallest average value of the torque and the biggest is for all presented numerical analysis the same.

In the pump mode also the pressure fluctuations measurement for 98% peak-to-peak amplitude was done at the outlet of the spiral casing. The pressure fluctuation is presented as

$$\Delta p_E = \frac{\Delta p}{\rho E} \, .$$

For the measuring operating points the pressure fluctuations amplitudes were compared for experimental and numerical results. The value of experimental pressure fluctuation is near 2 %, in comparison to numerical results where the value of pressure fluctuation is about 1,8 %.

V. Conclusions

The numerical analysis of the torque on guide vanes in reversible pump-turbine in pump mode operation shows some difficulties in the reliability of numerical results.

In the paper a detailed numerical investigation of torque on guide vanes pivots in radial reversible pump turbine is presented. The first results were obtained by steady state analysis and discrepancy between numerical and experimental results was significant. Such results is not usable for industrial use and further research work proving our assumption that only unsteady numerical analysis can give reliable results. This is also known for accurate prediction of energetic characteristics of pump mode operation. Another important conclusion from presented work is the recommendation of the length of the single time step and also advice about sufficient number of total time steps of time dependent analysis.

In this work different turbulence models were used and the final conclusion from this part of the research is that Reynolds Stress turbulence model is not suitable, because of convergence problems and very unstable behavior. SST model gives more reliability in the numerical analysis process, but our recommendation is SAS SST turbulence model.

In the future research work more accurate results can be obtained with better and finer computational grids and perhaps usage of Large Eddy Simulation (LES) based turbulence models.

Because the analysis was done only for one type of reversible pump turbine in the near future is planned to analyze more different pump turbines for different geometries and different operating conditions.

The recommended parameters of numerical procedure helps industrial users to obtain the reliable and accurate results in the computational time suitable for commercial projects.

VI. Notations

Term	**Symbol**
Hydraulic energy	E [Jkg^{-1}]
Runner revolution angle	F [^{o}]
Torque on guide vanes	M [Nm]
Rotation speed	n [s^{-1}]
Pressure	p [Pa]
Number of runner blades	z [-]
Efficiency	η [-]
Frequency	ν [s^{-1}]
Discharge coefficient	ϕ[-]

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