PUMPS USED AS TURBINES
Power Recovery, Energy Efficiency, CFD Analysis

by

Jasmina B. BOGDANOVIĆ-JOVANOVIĆ [a]*, Dragica R. MILENKOVIĆ [a], Dragan M. SVRKOTA [b], Božidar BOGDANOVIĆ [a], and Živan T. SPASIĆ [a]

[a] Faculty of Mechanical Engineering, University of Nis, Nis, Serbia,
[b] Jaroslav Cerni Institute for the Development of Water Resources, Belgrade, Serbia

Original scientific paper
DOI: 10.2298/TSCI1403029B

As the global demand for energy grows, numerous studies in the field of energy efficiency are stimulated, and one of them is certainly the use of pumps in turbine operating mode. In order to reduce time necessary to determine pump characteristic in turbine operating mode problem was studied by computational fluid dynamics approach. The paper describes various problems faced during modeling (pump and turbine mode) and the approaches used to resolve the problems. Since in the majority of applications, the turbine is a pump running in reverse, many attempts have been made to predict the turbine performance from the known pump performance, but only for best efficiency point. This approach does not provide reliable data for the design of the system with maximum energy efficiency and does not allow the determination of the head for a wide range of flow rates. This paper presents an example of centrifugal norm pump operating in both (pump and turbine) regime and comparison of experimentally obtained results and computational fluid dynamics simulations.

Key words: *pump used as a turbine, computational fluid dynamics analysis, BUTU method, energy efficiency*

Introduction

There are many instances in the water supply systems and processing industry where is required to reduce pressure of fluid. This pressure reduction is usually accomplished trough the use of a throttling valve. In this method, the energy of fluid stream is lost. Currently, emphasis is being placed on more effective energy usage in mentioned cases. Therefore, areas in which energy is wasted are being closely monitored and methods for energy recovery are being investigated.

A pump used as a turbine (PAT) can deploy the hydropower potential extremely efficiently and economically with straightforward technical means (there are many examples [1] realized by KSB company, which is world leader in this area). PAT are suitable for applications where pressure differences are to be reduced or where the head and flow rate of an installation can be exploited. The power generated can be used either for internal purposes or to fed into the public grid. Thanks to low investment costs, PAT solutions pay for themselves after a very short time.

* Corresponding author; e-mail: bminja@masfak.ni.ac.rs

As previously stated, the paper analyzes the possibility of computational fluid dynamics (CFD) application in order to determine the best PAT for the existing system.

The potential for power recovery in a water distribution system exists whenever water flows from a high pressure to a low pressure in such a manner that throttling or pressure break chambers occurs. Nis County Water Authority currently imports about 1000 liters per second of water through a gravity flow distribution system consisting of about 20 kilometers of pipeline. As a result, this system requires several inline pressure control facilities designed to regulate pressure and flow. These in-line facilities are potential sites for the installation of PAT. One possible design procedure for the selected location is presented in the paper, using the part of water distribution system of Nis as an illustrative example. In the last chapter, the analysis of PAT aggregate and the analysis of possible energy efficiency increasing have been carried out.

Pumps are relatively simple and easy to maintain and they also have a competitive maximum efficiency when compared to conventional turbines. Perhaps the major benefit is that mass production of pumps means that they are comparatively much more cost-effective than conventional turbines. PAT systems are cost-efficient and widely available. In addition, they have simple design and easy maintenance compared to conventional turbines. At the other hand, such systems do not have variable guide vanes for the purpose of turbine regulation. PAT systems have lower efficiency compared to conventional turbines.

A PAT system generally uses the pump's induction motor as an AC generator. For grid-tied installations, induction motors are usually the easiest rotating generation to interconnect directly. For stand-alone installations, capacitors are required to provide reactive power that allows the pump's induction motor to generate AC electricity. With the use of variable speed technology, by use of asynchronous motor-generator or synchronous motor-generator with frequency converter, the rotational speed of the pump-turbine can be varied. Thus, the turbine operating range can be extended; the pump capacity can be adjusted to using just the currently available amount of energy. This technology stabilizes the grid efficiently.

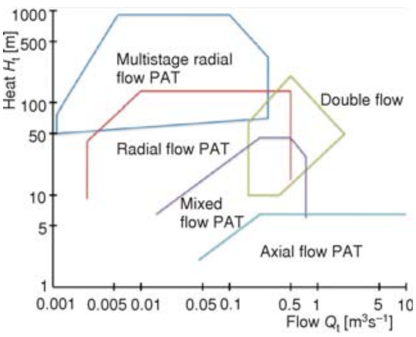

Figure 1. Operating area of PAT [2]

As there are many different types of pumps that can be used as a turbine Chapallaz *et al.* [2] gives the rough guide in fig. 1. how to select the right PAT. Multistage pumps are only typically used in cases where the head is very high, and when the flow rate is high either multi-flow pumps or a system of single flow pumps in parallel is used. During the years, some empirical formulas are presented in literature [3, 4]. These formulas are used for estimation of PAT operating parameters, knowing the best efficiency parameters of the pump regime.

BUTU method and other empirical formulas for PAT performance prediction

An empirical method, based on curve fitting of experimental data, is presented in the BUTU method (referring to the acronym of "Pump as Turbine" in Spanish) which was developed in Mexico and completed in Great Britain. The method predicts turbine performance at both Best Efficiency Point (BEP) and values away from this point. This is very valuable as a selected PAT will typically not operate at exactly its BEP but somewhere close to it. The errors incurred in this method are reported to be around 10% and more.

Review of BUTU method is summarized by following formulas:

$$\frac{P_{\mathrm{rp}}}{P_{\mathrm{rt}}} = 2\eta_p^{9.5} + 0.205 \tag{1}$$

$$\frac{H_{\mathrm{rp}}}{H_{\mathrm{rt}}} = 0.85\eta_p^{5} + 0.385 \tag{2}$$

$$\eta_{\mathrm{rt}} = \eta_{\mathrm{rp}} - 0.03 \tag{3}$$

$$\frac{P_{\mathrm{t}}}{P_{\mathrm{rt}}} = (1-k)\left(\frac{Q_{\mathrm{t}}}{Q_{\mathrm{rt}}}\right)^2 + k\frac{Q_{\mathrm{t}}}{Q_{\mathrm{rt}}} \tag{4}$$

$$k = -\frac{1}{0.96(\omega_{\mathrm{st}} - 0.2)^{-0.92} + 0.13} \tag{5}$$

$$\omega_{\mathrm{st}} = \frac{2\pi n_{\mathrm{rt}}\sqrt{\dfrac{P_{\mathrm{rt}}}{\rho}}}{60\sqrt[4]{(gH_{\mathrm{rt}})^5}} \tag{6}$$

$$\frac{P_{\mathrm{t}}}{P_{\mathrm{rt}}} = \frac{\mathrm{e}^{\left(0.37\frac{P_{\mathrm{t}}}{P_{\mathrm{rt}}}-1\right)} - 1}{0.37} + 1 \tag{7}$$

All methods for PAT performance prediction, given by Williams [3, 4], Nepal micro hydropower (NMHP) [5], Sharma [6], Stepanoff [7], McClaskey and Lundqvist [8], Krivichenko [9] can be represented by formulas:

$$Q_{\mathrm{t}} = K_{\mathrm{Q}}Q_{\mathrm{p}}, \quad H_{\mathrm{t}} = K_{\mathrm{H}}H_{\mathrm{p}}, \quad \eta_{\mathrm{t}} = K_{\eta}\eta_{\mathrm{p}} \tag{8}$$

where coefficients K_{Q}, K_{H}, and K_{η} are given in tab. 1.

Table 1

	K_{Q}	K_{H}	K_{η}
NMHP	1.25	1.38	1
Williams	1.65	2	1
Sharma	$1/\eta_p^{0,8}$	$1/\eta_p^{1.2}$	1
Stepanoff	$1/\eta_p^{0.5}$	$1/\eta_{\mathrm{p}}$	1
McClaskey	$1/\eta_{\mathrm{p}}$	$1/\eta_{\mathrm{p}}$	1
BUTU	$\dfrac{0.85\eta_p^5 + 0.385}{2\eta_p^{9.5} + 0.205}$	$\dfrac{1}{0.85\eta_p^5 + 0.385}$	$1 - \dfrac{0.03}{\eta_{\mathrm{p}}}$
Krivichenko	0.9-1	1.56-1.78	0.75-0.8

A relatively good prediction of BEP point in turbine mode by BUTU and other methods (given by Sharma [6], Stepanoff [7] McClaskey and Lundqvist [8], Krivichenko [9] *etc.*)

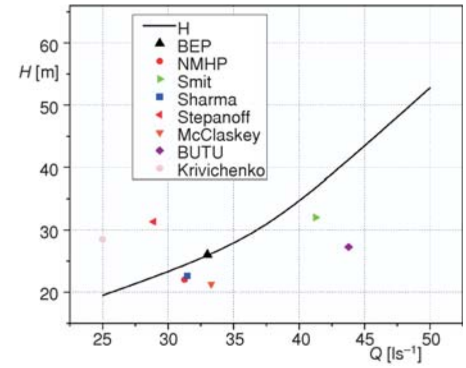

Figure 2. Performance prediction using empirical methods

should be obtained. However, when should provide a complete performance curve in turbine mode, large errors occur, making these methods quite unreliable.

For the considered case of centrifugal norm pump operates as a turbine, the best performance prediction is obtained using the methods given by Sharma, NMHP, and McClaskey, which is presented in fig. 2.

Numerical simulation

It has been mentioned in the introductory section that CFD, besides the experimental testing, is another possible mean of solution regarding the determination of turbine mode characteristic and optimization of the internal hydraulics within a PAT control volume.

During the design process of pump or turbine there are many issues that should be taken in consideration. On the other hand, tendency of modern industrial practice is to shorten the time required for the manufacture of a prototype, and the developments of methods that can provide in short term a reliable guidance for product development. In this way it is possible to significantly reduce the number of experimental tests, then to increase the energy efficiency of the existing system or to optimize a new plant in the design process.

The objective of experimental and numerical comparisons is to calibrate the CFD model to make quick and reliable predictions of hydraulic conditions in PAT under different optimization stages. This eventually helps in making recommendation for the different geometric modifications on the PAT to get required performance alleviation.

An accurate CFD model has many advantages over the experimental means. The model not only evaluates the global variables, but also determines the pressure losses in various flow zones of the PAT control volume. The internal zone losses have a great significance in understanding the hydraulic phenomenon and also more importantly give direction to the performance optimization. In case of pumps, a large number of papers and the experimental results are available [10], while for the pumps in turbine mode there is no so extensive research. A number of authors gave certain results, but they refer to specific examples of individual pumps [11-15]. CFD computations have been commonly used for approximately twenty years to predict the hydraulic performance of pumps and other turbo-machines [16-18].

First of all, CFD computations made it possible to study and improve the design of blades. For that purpose, the use of periodical conditions became a means to reduce the size of the computational domain, to improve the mesh and reduce the CPU time. Furthermore, it was possible to use a steady state approach to study operating points located close to the best efficiency point. It is important to note that the unsteady regimes, as well as interactions with other elements (spiral case, vaned diffusers) must be considered as the entire flow domain.

The approach that considers only impeller geometry with periodic boundary conditions allows quickly obtaining preliminary results for the case of the pump, and the question is whether it is possible for the case of the pump in turbine operating regime.

In this study, first characteristic of radial impeller in pump mode is determined and compared with experimentally obtained results, and then the possibility of determining the characteristics of the impeller in turbine operating mode is discussed. As the norm centrifugal pumps very frequently use impellers of different diameters for the same volute design, simple geometry of the volute is designed which have large tongue gap (20% of the impeller radius).

The flow through the pump and PAT has been simulated through the commercial code ANSYS CFX, which incorporates the 3-D incompressible Reynolds-averaged Navier-Stokes equations. The standard k-ε turbulence model along with log-law distribution for wall is commonly used for all test specimens. However there are few differences with respect to the schemes incorporated for convergence of computational results.

The computational mesh is unstructured, consists of 270373 nodes and 1116956 elements (934200 tetrahedral, 695 pyramids and 182061 wedges), as represented in fig. 3.

The meshing is refined in the vicinity of the walls and around the blade in order to correctly obtain the peripheral velocity gradients and the friction effects: the average value of y+ is equal to 50; specific cell thickness progression laws in the meridian, hub-to-shroud, and blade-to-blade directions are applied to ensure good grid quality. The number of elements used in the numerical simulation is fixed after the grid independence study (fig. 4). The global vari-

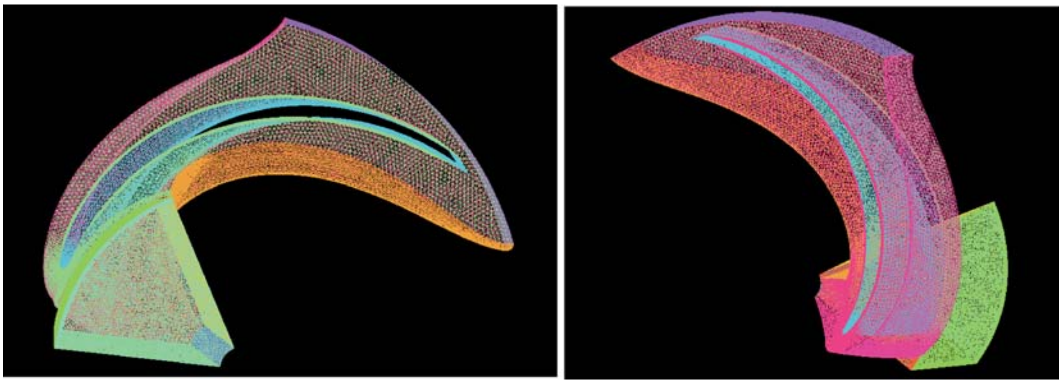

Figure 3. Numerical (discretization) mesh of the pump

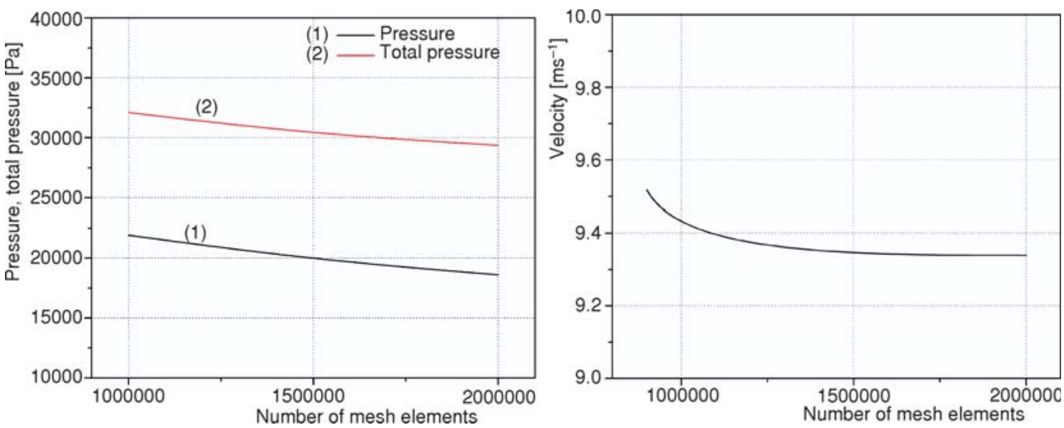

Figure 4. Grid independence test

ables that are simulated from the CFD model, namely the net head (H) and the hydraulic output torque (T_{hyd}) along with the given variables of discharge (Q) and rotational speed (n) are used to evaluate the overall pump characteristic.

It is to be noted that the CFD evaluates the hydraulic torque; hence, the efficiency is defined as the hydraulic efficiency, with the equation:

$$\eta_{\text{hyd}} = \frac{2\pi n T_{\text{hyd}}}{60\rho g Q H} \tag{9}$$

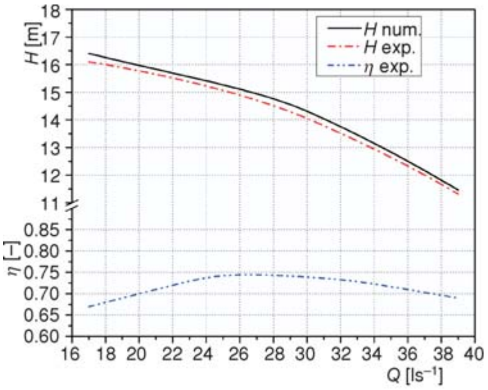

Figure 5. Pump operating curve (obtained numerically and experimentally)

Figure 5 presents pump-operating characteristic obtained numerically and experimentally, and pump efficiency, as well. Red dash-dot line represent experimentally obtained results, while solid line presents numerically obtained Q-H pump curve. It is obvious that numerically obtained results gives slightly higher values of pump head, which is probably due to the lack of spiral casing in the numerical simulations. However, it is obvious that the obtained pump curves have good agreement and error does not exceed 3%.

After the simulation of pump mode the same geometry and mesh is used for simulation of turbine working regime. The comparison between CFD results and the calculated results, according to available previously mentioned methods for best efficiency point (BEP), shows the big difference that must be explained. As shown at fig. 6

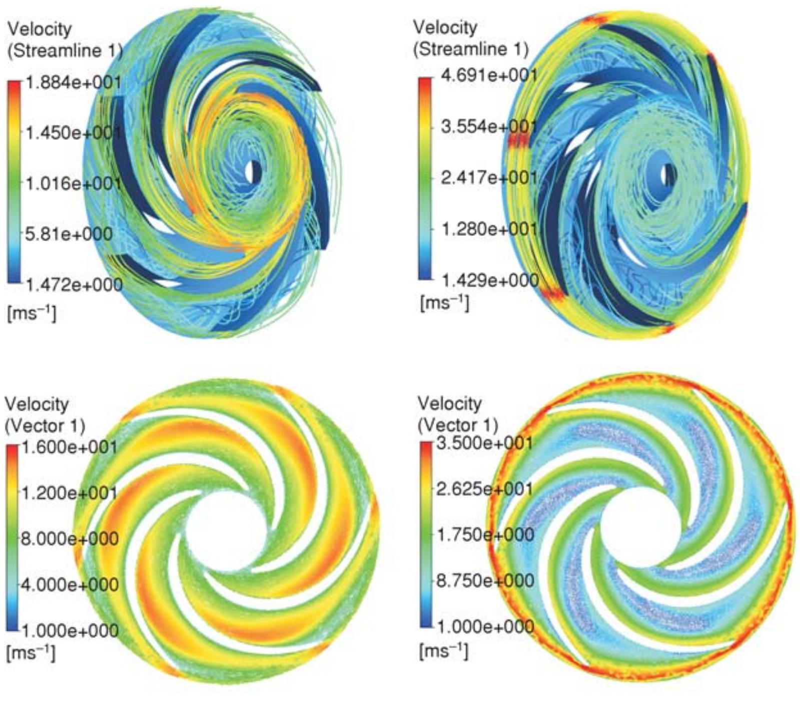

Figure 6. Streamlines and velocities for pump (left) and turbine (right) operating regime (BEP)

at design point, the internal flow or velocity vector is very smooth along the curvature of the blades for pump regime. In turbine regime, flow separation is developed near the leading edge. The single and double vortical flow structures are observed in the impeller, and the flow pattern has changed significantly from the pump operation mode.

Based on a review of the results two conclusions can be drawn:

– Net head obtained for turbine operating regime based on the recommendations of various authors differ considerably.
– Lack of spiral case in the simulations for turbine mode affects the flow pattern, low hydraulic efficiency and net head.

In order to obtain better results it is necessary to consider the whole geometry of the impeller and volute casing.

The geometry of the pumps is usually designed to get an optimum coupling between the impeller and the volute at the so-called nominal flow rate. In this situation, the flow matches suitably the geometry of the machine, and thus the flow instabilities are reduced to a minimum. In contrast, these optimum-coupling conditions cannot be kept when the pump operates in reverse mode at the same flow rate. For this reason, for the simulation of the pump in turbine mode, the geometry of the volute casing is formed (fig. 7), which have a very simple form and minimum radial gap between the impeller and volute tongue is 20% of outer radius.

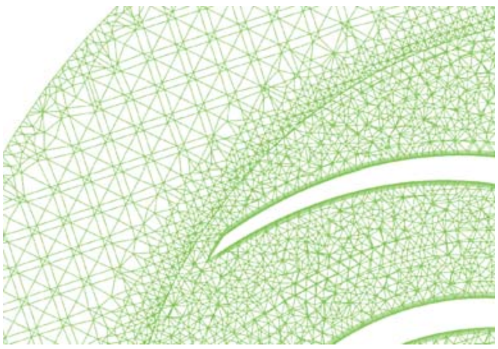

Figure 7. Numerical (discretization) mesh of the volute and impeller

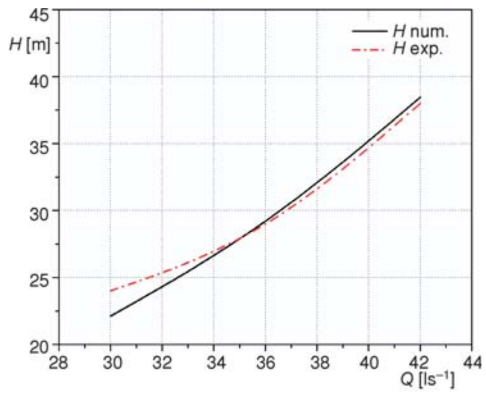

Figure 8. PAT operating curve

The calculations were performed for five flow rates in reverse mode of operation. The predictions from the numerical model were subsequently validated with experimental data collected at laboratory.

The experimental flow-head characteristic in direct (pump) and reverse (turbine) mode of operation and the predictions obtained from the numerical model are shown in figs. 5 and 8. As observed, the general trend of the curves is suitably predicted by the model, showing a negative slope with flow rate for direct mode and a positive slope for reverse mode. It is also seen that the numerical predictions are in very good agreement with the experimental values, showing maximum (relative) errors less than 4%.

Figure 9 presents results for different flow rates when pump operates in reverse mode (PAT). As noted, the pressure diminishes from the outer to the inner section of the impeller due to the continuous transmission of energy from fluid to blades.

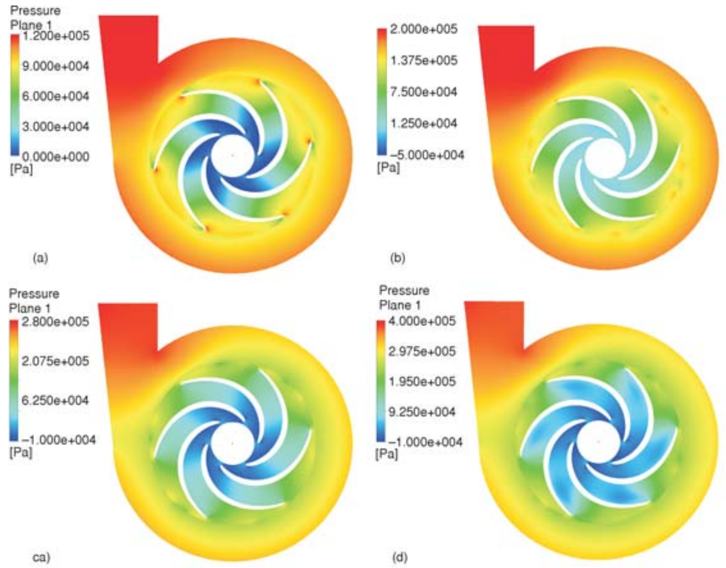

Figure 9. Instantaneous map of static pressure (a) Q/Q_{bep}= 0.75, (b) Q/Q_{bep}= 1, (c) Q/Q_{bep} = 1.1, (d) Q/Q_{bep} = 1.3

The net head across the impeller increases with increasing flow rates, as expected for a turbine mode (see characteristics of fig. 8). While the pressure seems quite uniform at the tongue region for the 75% of optimal flow rate (also called nominal flow rate or flow rate at best efficiency point Q_{bep}), it can be seen that there is a low pressure region between the impeller and the tongue for the higher flow rates.

The analysis of the relative velocity vectors shows that the guidance of the flow across the passageways is not as adequate as it was previously in pump mode. Thus, it can be noted that the predicted nominal flow rate presents the best guidance among the results shown in fig. 10 but, nonetheless, a small re-circulating region can be observed near the trailing edge of the blades (pressure side). This region increases in magnitude and it extends further down the inner section of the impeller at 75% of nominal flow rate, thus causing the incoming fluid to move

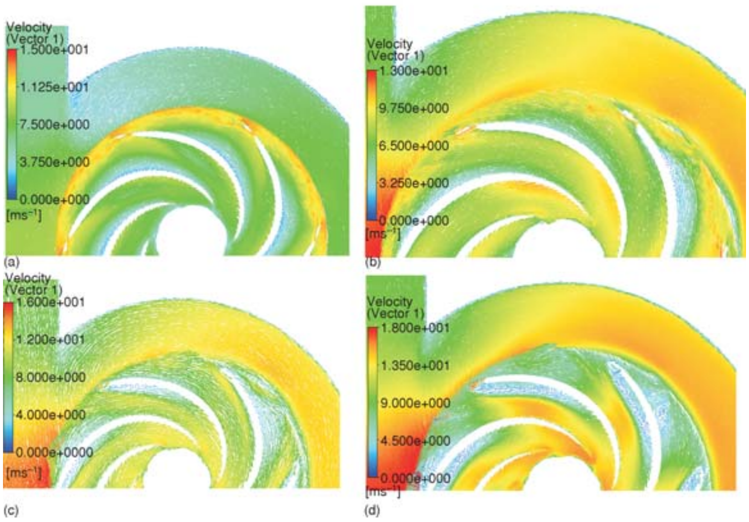

Figure 10. Instantaneous map of relative velocity vectors for (a) Q/Q_{bep} = 0.75, (b) Q/Q_{bep} = 1, (c) Q/Q_{bep} = 1.1, (d) Q/Q_{bep} = 1.3

closer to the suction side of the blades, as observed in the top image of fig. 10. At Q/Q_{bep}= 1.3, the increasing angle of the incoming fluid causes the re-circulating region to move closer to the suction side of the blades, as noted in the bottom image of fig. 10. This region extends almost to the middle of the passageway length and it shifts the fluid stream towards the pressure side of the blades.

In pump mode case, simple geometry of spiral case and increased mesh size gives results lower then experimentally obtained. The flow progress is not so smooth as in case when only impeller geometry is considered with high quality mesh. Figure 11 shows the relative velocity vectors for two different ratio, Q/Q_{bep}= 0.6 and Q/Q_{bep}= 1,4. It is observed that the static pressure increases continuously along the impeller passageways from the inner to the outer region as energy is transmitted due to the impulse of the blades.

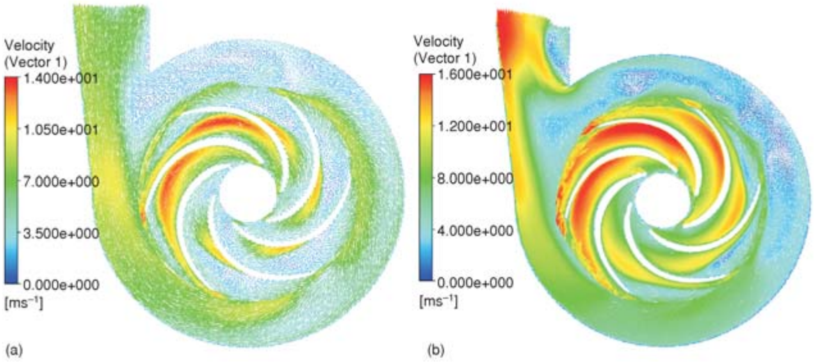

Figure 11. Instantaneous map of relative velocity vectors for (a) Q/Q_{bep} = 0.6, (b) Q/Q_{bep} = 1.4

The pressure reaches higher values at the lowest flow rate while decreasing with increasing flow rates, as expected for a centrifugal pump (see fig. 12). Nearly constant pressure field is noted for the flow rate Q/Q_{bep}= 0.6. In contrast, the highest flow rate (140% of rated) shows a low pressure region close to the exit of spiral case duct in contrast to higher pressure region between tongue and impeller. Obtained results quantitatively shows a little difference com-

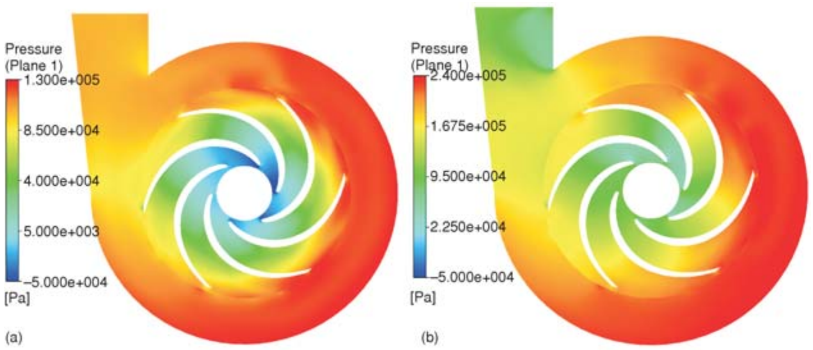

Figure 12. Instantaneous map of of static pressure for (a) Q/Q_{bep} = 0.6, (b) Q/Q_{bep} = 1.4 (for color image see journal web site)

pared to experimental but simple geometry of the volute do not show reliable flow pattern and gives results that deviate from reality.

Example of PAT application in the water distribution system

There are many potential places where the pumps as turbines can be used. Location Cukljenik in water distribution system of Nis is pressure braking chamber which is located at distance 2,5 km from the spring Studena. Spring Studena is located 17 km from the Nis at the elevation 400 m.a.s.l. Characteristic of the spring Studena is nearly constant flowrate during the year. Gravitational pipeline from Studena to Cukljenik is 2510 meters long and has a diameter of 400 mm.

A preliminary analysis of this location reveals that PAT aggregate should be operating with a flow rate of 205 l/s and net turbine head 67 m. It is required an adequate PAT aggregate for this purpose. Spiral centrifugal norm pumps of all manufacturers, shows standard operating curves range for defined pressure duct, impeller diameter and number of revolutions. Precisely for this reason, the possibility of such an aggregate application has been analyzed. Firstly, the best efficiency point in pump regime is determined, according to defined parameters related to turbine operating regime. Defining such a operating curve has been done due to recommendation given by Sharma and HMNP, considering the results obtained in the previous studied case.

Parameters used in simulations are Q =168 l/s and H = 50 m. These operating parameters correspond to the norm pumps 200-400, operating with 1450 rpm. In the pump regime, these pumps usually operate with 1480 rpm, and in the turbine mode with 1510 rpm, therefore one should take into account the difference of number of revolution in both operating regimes. The typical single stage norm pump impeller was designed and, according to previously mentioned procedure, the operating curve in turbine regime was determined (fig. 13).

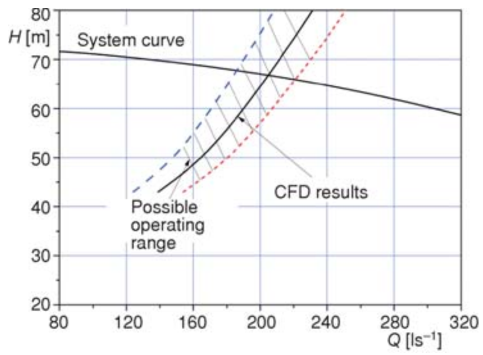

Figure 13. System *Q-H* characteristics and PAT operating curve for the water supply system Studena-Nis

In current operating regime, the pressure is reduced to the atmospheric pressure using a pressure break chamber, while in the case of PAT aggregate application the possible energy production could be 110 kW, and an estimated annual production is 800 MWh.

Conclusions

There are many potential locations for using PAT, such as: water supply systems, industrial applications, residual water utilization. Increasing the energy efficiency of existing system by installing PAT, instead of reducing high pressure in the system, provides a promising water management strategy. Two factors are important in designing such a scheme, namely: (1) a fairly accurate and quick prediction is required for the operating conditions, which takes into account the different performance characteristics of the pump when running as a turbine and pump, and (2) detail system characteristics in order to produce maximum available power. This paper presents that CFD analysis is an effective design tool for predicting the performance of centrifugal norm pumps in both regimes. In the case of pump mode the use of periodical conditions became a means to reduce the size of the computational domain, to improve the mesh and reduce the CPU time. In the case of turbine mode of operation, simple volute geometry gives fairly accurate results. These conclusions enable very quick evaluation of appropriate PAT aggregate, and then more detailed analysis using detail geometry of spiral case and finer mesh can be conducted. Future works in the field of computational analysis can further improve the prediction of pumps in reverse operation.

Acknowledgments

This paper is result of technological project No. TR33040, Revitalization of existing and designing new micro and mini hydropower plants (from 100 to 1000 kW) in the territory of South and Southeast Serbia , supported by Ministry of Education, Science and Technological Development of the Republic of Serbia.

Nomenclature

g – gravitational const. $g = 9{,}81$ ms^{-2}
H_p – pump head, [m]
H_t – PAT head, [m]
n – rotational speed, [s^{-1}, min^{-1}= rpm]
P – power, [W]
Q – volume flow rate, [m^{3}s^{-1}], [ls^{-1}]
T_{hyd} – hydraulic torque, [Nm]

Greek symbols

η – efficiency, [–]
ρ – density, [kgm^{-3}]
τ – angular blade pitch, [–]

Subscripts

rp, bep – pump in best efficiency point
rt, bep – PAT in best efficiency point
hyd – hydraulic

References

[1] ***, KSB Company, Pumps Used as Turbines, Trend-Setters in Energy Generation and Recovery, 2012, www.ksb.com
[2] Chapallaz, J. M., *et al*., *Manual on Pumps Used as Turbines*, Vieweg, Braunschweig, Germany, 1992
[3] Williams, A., *et al.*, Pumps as Turbines and Induction Motors as Generators for Energy Recovery in Water Supply Systems, *Water and Environment Journal, 12* (1998), 3, pp. 175-178
[4] Williams, A. A., The Turbine Performance of Centrifugal Pumps: A Comparison of Prediction Methods, *Proc. IMechE, Part A., Journal of Power and Energy, 208* (1994), pp. 59-66
[5] ***, Nepal Micro Hydro Power, Pump-As-Turbine Technology, Intermediate Technology Development Group, 2005, www.microhydro.org.np
[6] Sharma, K., Small Hydroelectric Project-Use of Centrifugal Pumps as Turbines, Technical Report, Kirloskar, Electric Co. Bangalore, India, 1985
[7] Stepanoff, A. J., *Centrifugal and Axial Flow Pumps, Design and Applications*, John Wiley and Sons, Inc. New York, USA, 1957
[8] McClaskey, B. M., *et al*., Can You Justify Hydraulic Turbines? *Hydrocarbon Processing, 55* (1976), 10, pp. 163-169
[9] Krivchenko, G., *Hydraulic Machines: Turbines and Pumps*, Lewis, Boca Raton, Fla., USA, 1994
[10] Shaha., S. R., *et al.*, CFD for Centrifugal Pumps: A Review of the State-of-the-Art, *Procedia Engineering, 51* (2013), pp. 715-720
[11] Ramos, H., Borga, A., Pumps as Turbines: Unconventional Solution to Energy Production, *Urban Water, 1* (1999), 3, pp. 261-263
[12] Rawal, S., Kshirsagar, J. T., Numerical Simulation on a Pump Operating in a Turbine Mode, *Proceedings*, 23rd International Pump Users Symposium, Houston, Tex., USA, 2007, pp. 21-27
[13] Singh, P., Nestmann, F., Experimental Optimization of a Free Vortex Propeller Runner for Micro Hydro Application, *Experimental Thermal and Fluid Science, 33* (2009), 6, pp. 991-1002
[14] Yang, S., *et al*., Theoretical, Numerical and Experimental Prediction of Pump as Turbine Performance, *Renewable Energy, 48* (2012), Dec., pp. 507-513

[15] Singh, P., Nestmann, F., Internal Hydraulic Analysis of Impeller Rounding in Centrifugal Pumps as Turbines, *Experimental Thermal and Fluid Science, 35* (2011), 1, pp. 121-134
[16] Thakker, A., Elhemry, M. A., 3-D CFD Analysis on Effect of Hub-to-Tip Ratio on Performance of Impulse Turbine for Wave Energy Conversion, *Thermal Science, 11* (2007), 4, pp. 157-170
[17] Josse David Villegas Jimenez, Numerical Simulations on a Centrifugal Pump Operating in Turbine Mode, Eafit University, Engineering School, Mechanical Engineering Department, Medellin, Columbia, 2010
[18] Jain, S. V., Patel, R. N., Investigations on Pump Running in Turbine Mode: A Review of the State-of-the-Art, *Renewable and Sustainable Energy Reviews, 30* (2014), Feb., pp. 841-868

Paper submitted: February 12, 2014
Paper revised: May 31, 2014
Paper accepted: June 4, 2014