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Numerical simulation to analyze the physical behavior of centrifugal pumps as a turbine

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Abstract. In this research, a methodology based on the development of numerical simulations is proposed to analyze the physical behavior of centrifugal pumps such as a turbine. Numerical simulations were carried out using OpenFOAM software. For the validation of the numerical model, the construction of an experimental test bench was carried out. The analysis carried out involves the evaluation of performance parameters of the pump as a turbine, such as head, power, and efficiency. Additionally, the effect of the rotation speed on the previous parameters is evaluated. From the results obtained, it was shown that the maximum relative error was 4%, 3.4%, and 3.8% for the head, power, and efficiency parameters, respectively. In general, it was evidenced that the proposed numerical simulation has the ability to describe the real trends of the pump as a turbine for different flow conditions. In addition, an 11% increase in rotational speed was shown to cause a 12%, 1.9%, and 3% increase in head, power, and maximum efficiency. The proposed methodology is considered an adequate tool to analyze performance and identify the best efficiency point of pump systems such as a turbine. In this way, greater energy use is guaranteed.

1. Introduction
Centrifugal pumps are among the main equipment used in industrial processes, which are frequently used to transport different fluids [1-4]. Additionally, centrifugal pumps have the ability to function as a turbine, which is a viable solution to solve the problem of high economic costs associated with classic turbines, as is the case of small hydroelectric plants. Another advantage of centrifugal pumps is their easy construction, availability, low maintenance cost, and ease of operation. Pump as turbine (PAT) systems can be implemented for irrigation channels, sewage systems, reverse osmosis systems, and water distribution networks [5]. Despite the advantages, the efficiency of PAT systems tends to be significantly reduced when an adequate operating condition is not established [6]. In the last decade, a large number of investigations have been proposed to analyze experimentally, theoretically, and through numerical models the use of centrifugal pumps as turbines [7]. This particular type of application requires precise knowledge of the pump performance for the identification of the best efficiency point (BEP). Because this has a decisive influence on the economic cost and operating time of the associated energy process. Therefore, it is necessary to have a methodology that allows the prediction of the performance of the pump as a turbine.

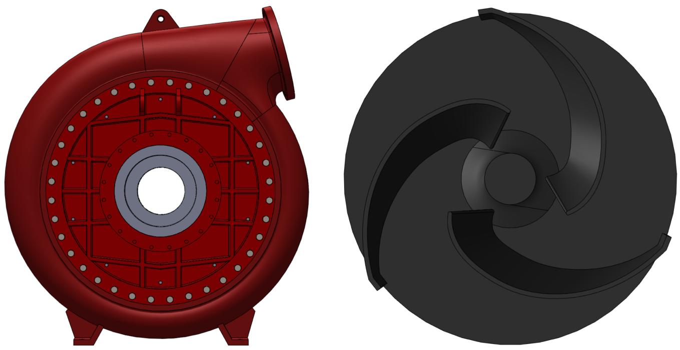
Some of the research available in the literature shows the use of empirical or experimental methods for the development of formulas that allow identifying the best efficiency point of the pump as a turbine [8,9]. However, the formulations of these equations are based on geometric designs and specific operating conditions. Naeimi, *et al.* [10] showed that the prediction method proposed by Stepanoff was able to achieve adequate precision for a one-stage centrifugal pump such as a turbine. Stefanizzi, *et al.* [11] and Novara, *et al.* [12] propose the use of statistical methods for the development of models that allow predicting the performance of pumps such as turbines. However, these models cannot be used universally and rely heavily on statistical databases.

In recent years, researchers have focused on the use of theoretical prediction methods to determine the BEP of pumps such as turbines. Liu, *et al.* [13] developed a formula for calculating the Euler head of a turbine and a pump, considering the slip coefficient. The above was used for the prediction of the BEP. Shi, *et al.* [14] constructed the Euler equation for the analysis of PAT systems, taking into consideration amplification factors. Wang, *et al.* [15] and Carpuso, *et al.* [16] developed formulations for the prediction of BEP in PAT systems, which were based on the blade inlet slip factor. Despite the advantages of theoretical models, the predictability is not high enough to describe the behavior of PAT systems.

The objective of this research is the development of a methodology for the analysis of the performance parameters of centrifugal pump systems such as a turbine. The above, by means of numerical simulations by means of the OpenFOAM software [17]. The analysis carried out involves performance parameters, such as head, power, and efficiency. In this way, it seeks to contribute to a better understanding of the physical behavior of pumps as a turbine.

2. Methodology

For the development of the numerical simulations, a 3D model corresponding to the centrifugal pump was built, which includes the geometry of the volute and the impeller. The geometric models were made using SolidWorks software, as shown in Figure 1.



(a) (b)

Figure 1. 3D model of the centrifugal pump; (a) casing, (b) impeller.

For the pump operating fluid, water was selected under a condition of ambient temperature (27 °C). The behavior of the fluid was defined from the average Reynolds flow equations, which describe the behavior of incomprehensible three-dimensional fluids as shown in Equation (1) and Equation (2) [18].

$$\frac{\partial \overline{u_i}}{\partial x_i} = 0, \quad (1)$$

$$\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right], \quad (2)$$

where u_i is the mean axial velocity component, u_j is the mean radial velocity component, x_i is the axial axis, x_j is the radial axis, ρ is the density, p is the time-averaged pressure, μ is the dynamic viscosity, and μ_t is the turbulent viscosity, respectively. To close Equation (1) and Equation (2) the shear stress transport (SST) $k - \omega$ model proposed by Menter, *et al.* [18] because it is widely used for problem-solving in rotating machines since it maintains the robustness of the $k-\varepsilon$ model (where k is the turbulent kinetic energy and ε is the turbulent energy dissipation) and the high prediction capacity in the regions close to the wall such as the $k-\omega$ model. The turbulent kinetic energy transport equations and the specific turbulence dissipation rate (ω) of this model are shown in Equation (3) and Equation (4) [18].

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}\left(\mu_{eff,k}\frac{\partial k}{\partial x_i}\right) + \widetilde{P}_k - \rho\beta' k\omega, \tag{3}$$

$$\frac{\partial}{\partial x_i}(\rho\omega u_i) = \frac{\partial}{\partial x_i}\left(\mu_{eff,\omega}\frac{\partial \omega}{\partial x_i}\right) + \frac{\alpha}{v_t}\widetilde{P}_k - \rho\beta\omega^2 + 2(1-F_1)\rho\frac{\sigma_{\omega,2}}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_i}, \tag{4}$$

where k is the turbulent kinetic energy, ω is the turbulence frequency, $\mu_{eff,k}$ is the effective diffusivities for the turbulent kinetic energy, $\mu_{eff,\omega}$ is the effective diffusivities for specific dissipation rate, α is the correlation constant for the production of the specific dissipation rate, v_t is the kinematics viscosity, $\widetilde{P}_k$ is a production limiter to prevent turbulence buildup in stagnation regions, $\beta' = 0.09$, $\beta = 0.075$ and $\sigma_{\omega,2} = 0.856$ are model constants, and F_1 is a blending function, respectively.

The OpenFOAM software [17] is used to solve the numerical model since it is freely accessible. A tetrahedral mesh was defined for the entire computational domain. To evaluate the sensitivity of the meshing, the behavior of the height of the centrifugal pump with meshes of different numbers of nodes was analyzed; the results obtained are shown in Figure 2.

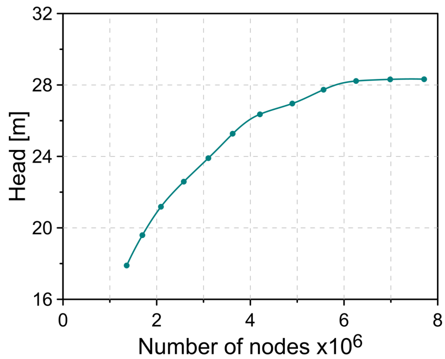


Figure 2. Mesh sensitivity analysis.

In Figure 2, it was observed that the height of the pump did not vary considerably for a number of nodes greater than 6 million. From this number of nodes, it was evidenced that the change in the pump head was less than 0.4%. Therefore, a number of nodes of 6.5 million were selected since it complies with the numerical precision and the computational economic cost is not high.

To verify the reliability of the results obtained through the numerical model, it is necessary to carry out a validation process using experimental data. Due to the above, a test bench was built for the centrifugal pump operating as a turbine, as shown in Figure 3.

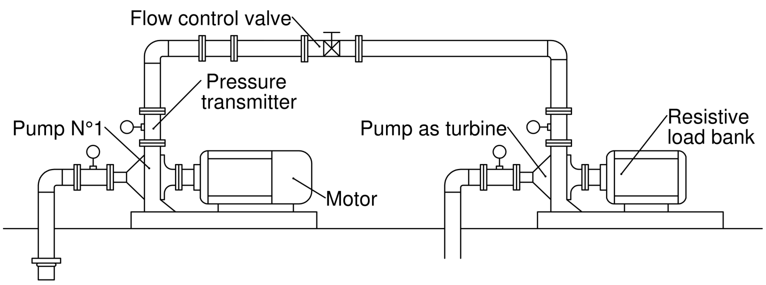


Figure 3. Test bench diagram of the pump as a turbine.

The test bench is made up of a centrifugal pump, a variable frequency motor, and a measurement system. Pressure transmitters recorded the pressure at the inlet and outlet of the pumps. Pump N°1 provides high-pressure flow directed to the PAT system. The energy generated by the PAT system is consumed by a resistive load bank. The calculation of the parameters of head (H), power (P_t), and efficiency (η) is determined by Equation (5), Equation (6), and Equation (7) [19].

$$H = \frac{P_{inlet} - P_{oulet}}{\rho_f \times g}, \tag{5}$$

$$P_t = T \times \omega_r, \tag{6}$$

$$\eta = \frac{P_t \times 100\%}{\rho_f \times g \times Q \times H}, \tag{7}$$

where P_{inlet} and P_{oulet} are the pressure in the inlet and outlet of the pump as turbine, respectively, ρ_f is the density of the fluid, g is the gravity, Q is the volumetric flow, T is the torque, and ω_r is the angular speed of the shaft.

3. Results

To evaluate the predictive capacity of the numerical simulation, a comparison was made between the head, power, and efficiency parameters of the pump operating as a turbine obtained through simulation and experimental tests. The analysis was carried out between a flow range of 0.3 m^3/s-3 m^3/s. The results obtained are described below.

Figure 4 shows the comparison between the pump head operating as a turbine from the results of the numerical simulation and the experimental tests. In general, the curve predicted by the numerical simulation describes the same trend as the experimental results. With the increase in the flow level, an increase in the head of the pump was observed. The largest deviations between both curves occurred in high flow conditions, specifically between 1.92 m^3/s - 2.46 m^3/s. This deviation is attributed to the volumetric losses and the mechanical losses of the pump, which are not considered in the numerical simulation. Despite the above, the relative error between both curves was less than 4%, respectively.

Figure 5 shows the power of the pump operating as a turbine for a flow range of 0.3 m^3/s - 3 m^3/s. The experimental results obtained show that the pump describes a power between 97 kW - 179 kW. In the case of the numerical simulation, a power between a range of 100 kW - 180 kW was observed. This difference is directly associated with the larger pump head reported by the numerical simulation, as shown in Figure 4. The maximum relative error when comparing both curves was 3.4%, respectively.

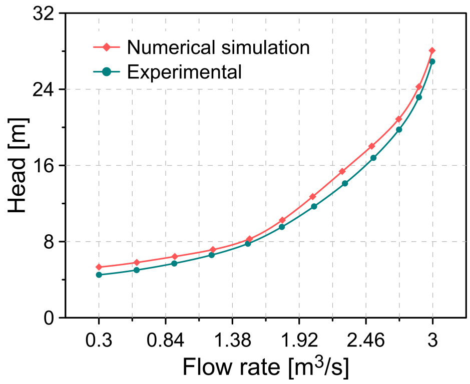


Figure 4. Pump head as turbine.

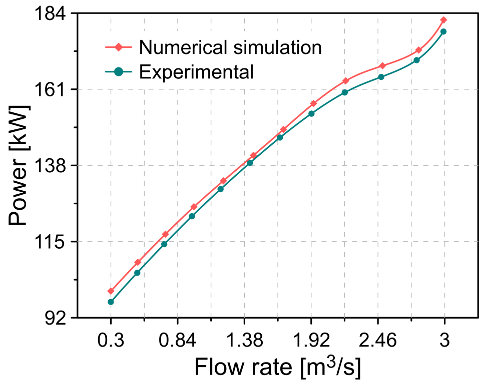


Figure 5. Pump power as turbine.

The analysis of the efficiency of the pump as a turbine is shown in Figure 6. Both the results of the simulation and the experimental data indicate that the maximum efficiency of the pump is obtained for a flow of 2.23 m^3/s. For this flow, an efficiency of 80.1% and 82.6% is obtained, corresponding to the experimental tests and the numerical simulation, respectively. On average, the deviation between the experimental data and the numerical prediction was 2.85%. Next, the influence of the rotation speed on the head, power, and efficiency parameters of the pump operating as a turbine is described. The analysis is performed for three-speed conditions, 2400 rpm, 2700 rpm, and 3000 rpm, respectively.

Figure 7 shows the behavior of the pump head for the different rotation speeds obtained through the numerical simulation. It was evidenced that the increase in the rotation speed causes an increase in the pump head. Overall, it was observed that an 11% increase in speed caused a 12% increase in the pump head. Similarly, it was observed that the increase in speed tends to increase the power of the pump, as shown in Figure 8. This increase is more evident for high flow conditions. In the particular case of the present study, a significant increase in pump power was observed for flows greater than 1.83 m^3/s.

The analysis of the pump efficiency for the speeds of 2400 rpm, 2700 rpm, and 3000 rpm are shown in Figure 9. The maximum efficiency for each of the previous speeds was 78.0%, 80.5%, and 82.6%, respectively. In general, it was observed that the reduction of the rotation speed causes a decrease in the high efficiency operating zone ($\eta > 75\%$) of the pump as a turbine. It was evidenced that the flow range of the high-efficiency zone was 0.96 m^3/s-2.20 m^3/s, 1.11 m^3/s-2.60 m^3/s, and 1.28 m^3/s-2.86 m^3/s for a speed of 2400 rpm, 2700 rpm, and 3000 rpm, respectively.

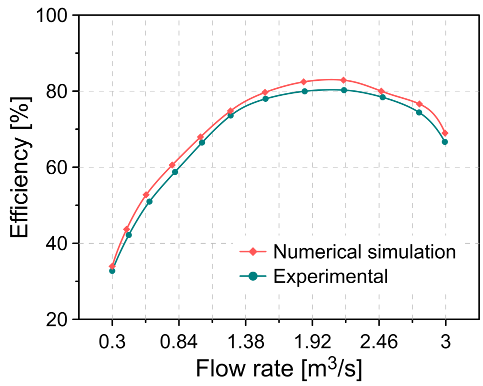


Figure 6. Pump efficiency as turbine.

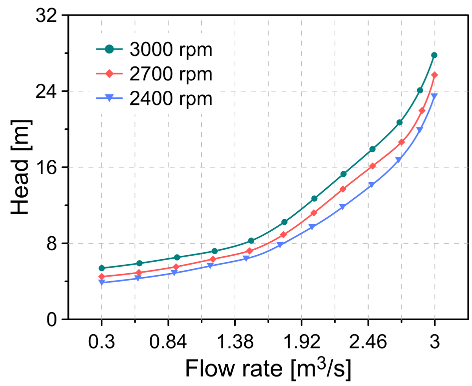


Figure 7. Pump head for different rotation speeds.

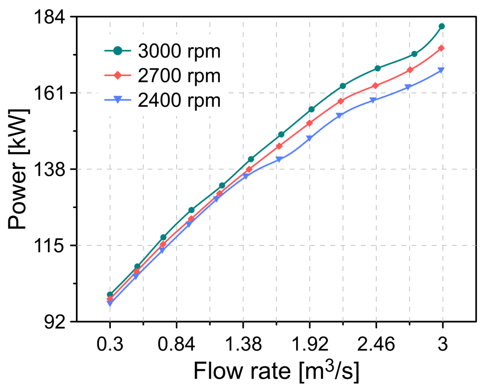


Figure 8. Pump power for different rotation speeds.

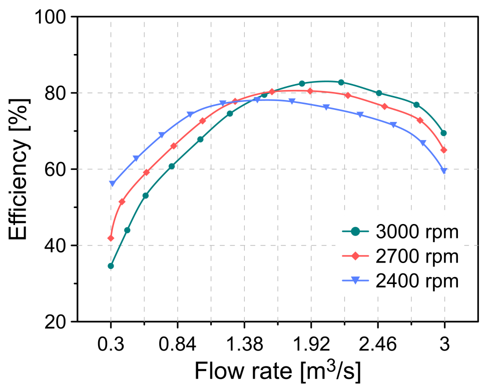


Figure 9. Pump efficiency for different rotation speeds.

4. Conclusions

In the present investigation, a methodology is proposed to analyze the performance of a centrifugal pump operating as a turbine by means of numerical simulation. To validate the simulation results, experimental tests were carried out on a test bench. Additionally, the influence of the rotation speed on the head, power, and efficiency parameters of the pump is evaluated.

The comparison between the performance parameters of the pump obtained by means of experimental data and the numerical simulation shows that the latter has the ability to describe the real trends of the pump as a turbine for the different flow conditions. The analysis of the maximum relative error was 4%, 3.4%, and 3.8% for the head, power, and efficiency parameters. This deviation is associated with volumetric losses and mechanical losses not considered in the numerical model. Despite the above, the error in the prediction of the parameters of the pump as a turbine remained less than 5%.

Additionally, the identification of the maximum efficiency flow condition through the numerical simulation was in agreement with the experimental results. The development of the numerical simulation made it possible to evaluate the influence of the rotation speed on the performance parameters of the pump as a turbine. An 11% increase in rotational speed was shown to cause a 12%, 1.9%, and 3% increase in the pump head, power, and maximum efficiency, respectively. In general, it was shown that the methodology proposed in this research has the ability to evaluate the behavior of centrifugal pumps such as turbines, which allows analyzing the performance of this type of system and identifying the maximum efficiency condition. In this way, greater energy use in the pump is guaranteed.

References

[1] Forero J D, Ochoa G V, Alvarado W P 2020 Study of the piston secondary movement on the tribological performance of a single cylinder low-displacement diesel engine *Lubricants* **8(11)** 97

[2] Consuegra F, Bula A, Guillín W, Sánchez J, Duarte Forero J 2019 Instantaneous in-cylinder volume considering deformation and clearance due to lubricating film in reciprocating internal combustion engines *Energies* **12(8)** 1437

[3] Orozco W, Acuña N, Forero J D 2019 Characterization of emissions in low displacement diesel engines using biodiesel and energy recovery system *International Review of Mechanical Engineering* **13(7)** 420

[4] Al-Yasiri A J, Ali M A, Ali R S, Bekheet H N 2020 Renewable energy sources in international energy markets: reality and prospects *International Journal of Innovation, Creativity and Change* **11(3)** 376

[5] Agarwal T 2012 Review of pump as turbine (PAT) for micro-hydropower *International Journal of Emerging Technology and Advanced Engineering* **2(11)** 163

[6] Yousefi H, Noorollahi Y, Tahani M, Fahimi R 2019 Modification of pump as turbine as a soft pressure reduction systems (SPRS) for utilization in municipal water network *Energy Equip. Sys.* **7(1)** 41

[7] Binama M, Su W-T, Li X-B, Li F-C, Wei X-Z, An S 2017 Investigation on pump as turbine (PAT) technical aspects for micro hydropower schemes: a state-of-the-art review *Renew. Sust. Energ. Rev.* **79** 148

[8] Barbarelli S, Amelio M, Florio G 2016 Predictive model estimating the performances of centrifugal pumps used as turbines *Energy* **107** 103

[9] Wang T, Wang C, Kong F, Gou Q, Yang S 2017 Theoretical, experimental, and numerical study of special impeller used in turbine mode of centrifugal pump as turbine *Energy* **130** 473

[10] Naeimi H, Shahabi M N, Mohammadi S 2017 A comparison of prediction methods for design of pump as turbine for small hydro plant: implemented plant *IOP Conf. Ser.: Earth Environ. Sci.* **83(1)** 12006:1

[11] Stefanizzi M, Torresi M, Fornarelli F, Fortunato B, Camporeale S M 2018 Performance prediction model of multistage centrifugal pumps used as turbines with two-phase flow *Energy Procedia* **148** 408

[12] Novara D, McNabola A 2018 A model for the extrapolation of the characteristic curves of pumps as turbines from a datum best efficiency point *Energy Conversion and Management* **174** 1

[13] Liu M, Tan L, Cao S 2019 Theoretical model of energy performance prediction and BEP determination for centrifugal pump as turbine *Energy* **172** 712

[14] Shi G, Liu X, Wang Z, Liu Y 2017 Conversion relation of centrifugal pumps as hydraulic turbines based on the amplification coefficient *Advances in Mechanical Engineering* **9(3)** 1

[15] Wang X, Yang J, Xia Z, Hao Y, Cheng X 2019 Effect of velocity slip on head prediction for centrifugal pumps as turbines *Mathematical Problems in Engineering* **2019** 1

[16] Capurso T, Stefanizzi M, Pascazio G, Ranaldo S, Camporeale S M, Fortunato B, Torresi M 2019 Slip factor correction in 1-D performance prediction model for PaTs *Water* **11(3)** 565

[17] Greenshields C J 2015 *OpenFOAM The Open Source CFD Toolbox Programmer's Guide Version 3.0.1* (London: OpenFOAM Foundation Ltd.)

[18] Menter F R 1994 Two-equation Eddy-viscosity turbulence models for engineering applications *AIAA Journal* **32(8)** 1598

[19] Derakhshan S, Nourbakhsh A 2008 Theoretical, numerical and experimental investigation of centrifugal pumps in reverse operation *Experimental Thermal and Fluid Science* **32(8)** 1620