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**NUMERICAL MODEL FOR ANALYSIS OF AXIAL-FLOW PUMPS
DESIGNED USING THE MINIMUM PRESSURE COEFFICIENT
CRITERION**

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Abstract. *In order to design better performing hydraulic machines, it is important to have a geometry that is capable of guaranteeing the flow characteristics required in the initial project and at the same time reduce the energy loss as much as possible. Computer simulation technologies are an effective form of achieving this goal, as they provide a complete picture of machine performance without the need to manufacture prototypes. Studies have proved that the minimum pressure coefficient criterion is suitable for the design of efficient cascades for both power consumer and power supplier turbomachinery. In order to create an efficient methodology for the design of axial-flow pumps, a numerical study of the minimum pressure coefficient on the suction side of the blade hub is presented, with the objective to complement the previous approaches that present only experimental studies of the cascade of turbomachinery designed using this criterion. Thus, a three-dimensional computational fluid dynamic (CFD) model will be built by applying the finite volume method using the Ansys CFX tool. To solve the RANS (Reynolds Averaged Navier Stokes) equations, the $k-\omega$ SST turbulence model was used. The validation of this computational model will be carried out with experimental data from axial-flow pumps designed using the classic wing theory and free vortex methodologies. It is intended to obtain an interpretation of the flow through the blade runner operating on design and off design conditions. The characteristic curves of the pump operation were obtained, meeting the criteria of convergence and mesh independence. The operating characteristics of the pumps were developed considering the convergence and mesh independence. The results showed a relation between the pressure field in a blade designed by the criterion of the minimum pressure coefficient and its performance. Comparisons between the curves obtained by numerical and experimental studies validated the methodology and confirmed that the criterion is adequate for the selection of an efficient cascade geometry.*

Keywords: Axial-Flow Pump, Minimum Pressure Coefficient, $k-\omega$ SST

1. INTRODUCTION

For turbomachinery design, it is essential to establish a methodology that guarantees high performance while maintaining the desired hydraulic characteristics. Due to the difficulties in treating the tridimensional and complex flow in turbomachinery, numerical simulation techniques have been developed to solve the equations that describe the fluid flow (Pinto *et al.*, 2017). Studies on the flow behavior and analysis of axial flow pumps have been proposed. Shi *et al.*, (2017) in his work investigated the flow in an axial-flow pump numerically and experimentally. The hydraulic performance and pressure fluctuation of a tank-style axial-flow pump device was analyzed, and an optimized scheme of the inlet and outlet passages was obtained. Zhang and Tang (2022) examined experimentally an axial-flow pump system in extreme conditions and proposed a numerical model for predicting the flow characteristics obtaining good results.

Even with the available computational methods for flow analysis in axial-flow pumps, it is necessary to adopt some design criteria to guarantee the required performances (Cruz *et al.*, 2008). In his pioneering work, Fernandes (1973), tested the applicability of the minimum pressure coefficient criterium for the selection of blade cascade geometries of axial-flow pumps with minimum losses, through a series of experiments utilizing Göttingen profiles. By using the

potential theory proposed by Mellor (1959), Fernandes (1973) demonstrated that the minimum pressure coefficient criterium if used as a blade load parameter has great influence in the determination of the behavior of axial-flow pumps. The minimum pressure coefficient criterium (C_{pmin}) was also tested in axial flow fans as a blade aerodynamic loading measure (Amarante Mesquita et al., 1996; 1999), and for low head hydraulic turbine design (Cruz et al., 2008). Recent works (Sutikno and Adam, 2011; Singh and Singh, 2017) confirmed the use of the minimum pressure coefficient as a measure of performance and design criterium for very low head axial flow turbines cascades. The results obtained by Muis et al. (2015) also showed very good performance of very low head turbines designed using the C_{pmin} , achieving a maximum efficiency of over 91% in the design operation conditions. Sotoude Haghghi et al., (2019) also verified the influence of the pressure coefficient in the performance of very low head axial turbines and obtained good efficiency. More recently, Zhang et al., (2022) studied the efficiency of turbines, and concluded that the pressure coefficient is one of the most important parameters in the machine performance.

In 2007 the Brazilian Government created the program “Caminho da Escola”, which aims to promote standardization of the scholar transport vehicles in Brazil (FNDE, 2015; Amarante Mesquita et al., 2015). In this scope, the project of a water tunnel for testing boat models was conceived. Considering the high cost and availability of large axial-flow pumps in Brazil, a project was started to design and build this type of pump, especially for the water tunnel. The objective of the present work is to apply the minimum pressure coefficient criterium to the design of this pump.

Ansys CFX and Ansys TurboGrid software package, specific to work with geometries of turbomachinery, were used to generate the computational mesh. The pump characteristics curves were obtained using the finite volume method to solve the RANS (*Reynolds Averaged Navier-Stokes*) equations and the $k-\omega$ SST turbulence model was used, since it is the most recommended when flow separation occurs (Silva, 2012). The CFD model was validated with the performance curves obtained experimentally by Fernandes (1973).

1.1 The minimum pressure coefficient criterion

The pressure coefficient is a parameter that gives information about the aerodynamic loading of a blade or airfoil. It is given by Eq. (1).

$$C_p = \frac{p - p_o}{\frac{1}{2} \rho w_\infty^2} \quad (1)$$

where p is the static pressure on the blade profile, p_o is the reference pressure and w_∞ is the average velocity on the blade profile. This coefficient can be used as a cascade selection criterion. Figure 1 illustrates this concept. There is a C_{pmin} interval limited by a lower value C_{psi} , and a higher value, C_{pss} . This interval corresponds to a cascade with minimum losses. For $C_{pmin} > C_{pss}$, a slight load is obtained but in return a large area of the blade is exposed to the flow, so higher friction is generated, although if the selected value for C_{pmin} is lower than the C_{psi} a high load is observed, increasing the chances of flow separation. The goal is to select the optimum interval, and the best way to achieve this task is by testing a series of turbomachines.

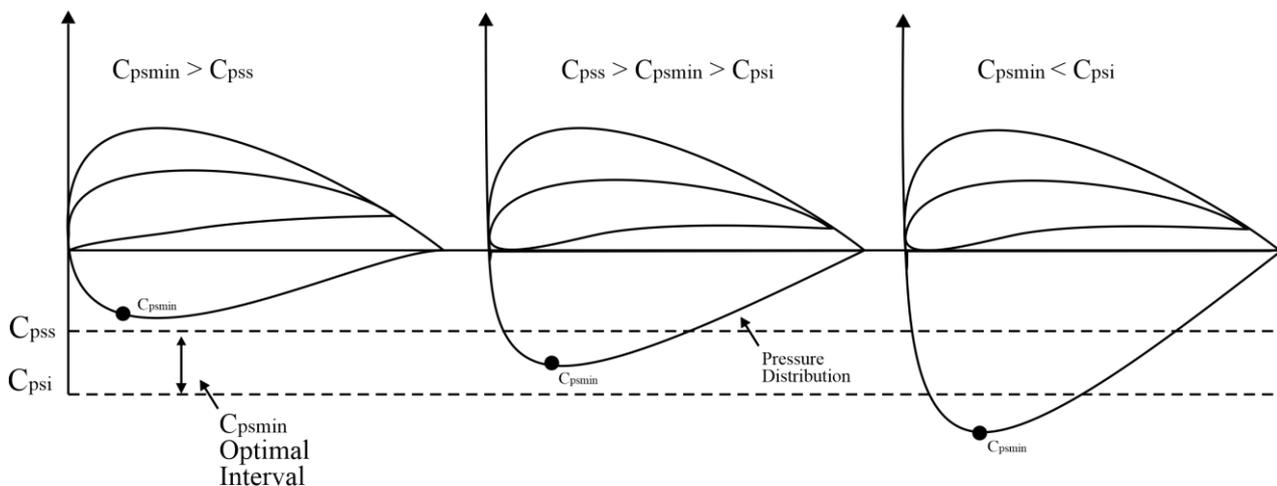


Figure 1. Minimum pressure criterion

For this specific case a C_{psmin} of -2.12 was selected as a project criterium for an axial-flow pump with $Q_o=0,040$ m³/s and $Y_o=50$ J/kg.

2. VALIDATION

2.1 Experimental

The first step in using commercial CFD programs is to obtain a geometric model that is compatible with the real physical phenomenon to be simulated. To create the geometries, it was first necessary to collect the coordinates of the profiles of each section of the blades. In order to design the pump, three profiles from the Göttingen series were used. The dimensions and properties of the impeller are described in Fig. 2. The global parameters of the pump are directly related to the specific speed value. The dimensionless coefficients define the operation characteristics of the pump and describe its behavior. Considering the pump head H , rotational speed N , external diameter De , flow rate Q , the operation point is defined by the following parameters Eq. (2) Eq. (3) and Eq. (4):

flow coefficient,

$$\Phi = \frac{Q}{(ND_e^3)} \quad (2)$$

head coefficient

$$\Psi = \frac{Hg}{N^2 D_e^2} \quad (3)$$

specific speed

$$n_q = N \frac{\sqrt{Q}}{H^{3/4}} \quad (4)$$

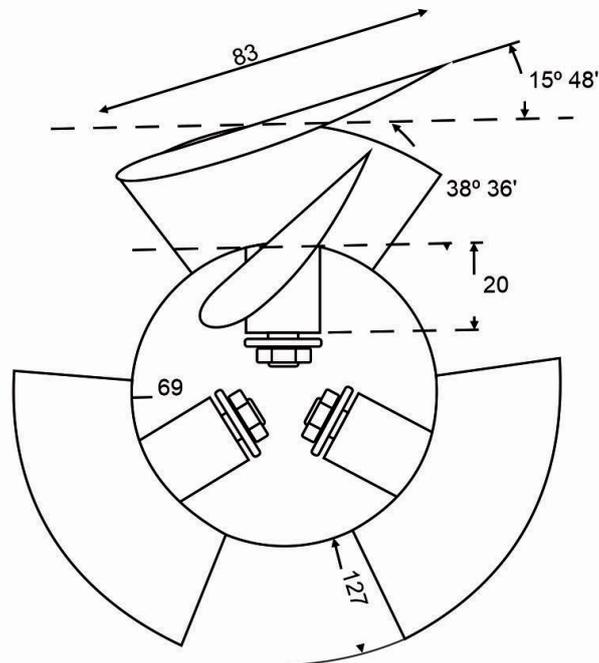


Figure 2. Axial-Flow Pump impeller dimensions in mm and dimensionless characteristics (Fernandes, 1973).

2.2 CFD Method

The group of cells forms the computational domain or numerical domain, which has an approximate geometry of the domain Fig. 3 that is studied. Most modern CFD compiler software has a computational domain generation package that handles most cases very well (Stolarski et al., 2018).

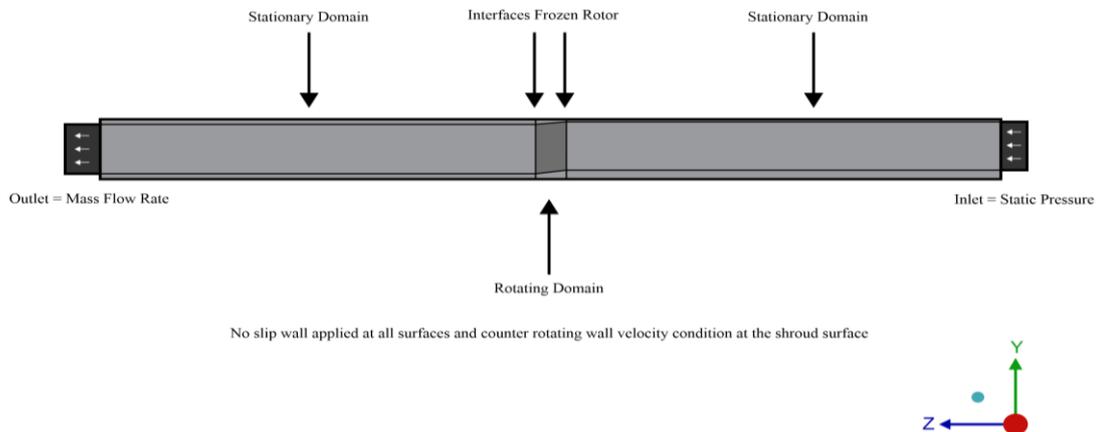


Figure 3. Tridimensional domain used to simulate the pump flow behavior

Within the domain there are the boundary conditions, which define how the flow behavior will be. This domain was selected in order to allow the flow conditions at the pump inlet to be closer to the real one, and thus the pump inlet region is far enough from the channel inlet region so that the flow is fully developed, and the outlet region is far enough away that there is no reverse flow.

The axial-flow pump impeller was generated importing the blade profiles into the CAD software Ansys Design Modeler. A fluid flow path is then created and exported to the commercial grid generator software Ansys Turbo Grid.

The numerical model consisted of three domains, that is, inlet, impeller, and outlet. Between the three domains there are two interfaces frozen rotor, which is recommended by the software in between a stationary and a rotating domain. No-slip boundary conditions are imposed at the solid walls, such as the inlet and outlet passage walls, the blade pressure and suction surfaces, the hub and the shroud. The outlet and inlet faces were configured as shown in Fig. 3. the mass flow rate for each point of operation in kg/s in the outlet and the inlet face was configured as static pressure. The grid independence test was conducted based on the pressure head across the pump as shown in Fig. 4. The grid number consists in all the elements of the three domains.

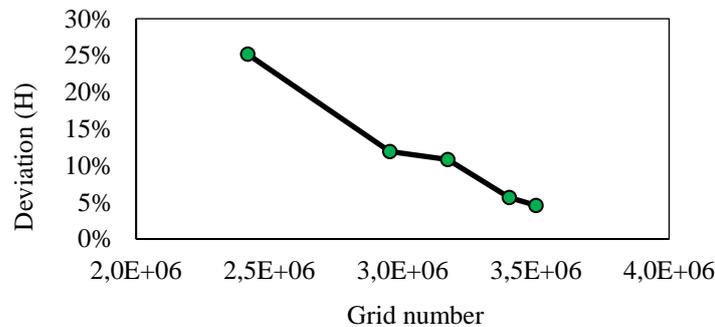


Figure 4. Grid independence test.

It can be found that the pressure head difference with respect to the finest model was less than 5% as the grid number of the numerical model increased to 3,5 million elements. Therefore, the model with 3,4 million grids was selected as a compromise between the accuracy and computational resource usage.

A periodicity condition was not used, that is, the rotor was discretized in its entirety, because the number of pump blades is small and, in this way, an optimal approximation of the flow conditions in the region of the blades can be obtained.

In between the clearance gap between the pump casing and the impeller shroud, 10 layers of grid cells were generated in the selected model. The skew angles of these grids were maintained between 15 and 170° with the y^+ value of grids at near-wall region maintained around 1, in order to ensure the quality of grids and to suffice the turbulence model requirements. Figure 5 shows the grid of the blade leading edge and the refinement of the tip and shroud gap.

The numerical simulation was conducted using the finite-volume CFD code Ansys CFX. The fluid was assumed to be Newtonian with a viscosity of $0,001 Pa \cdot s$ and a density of $997 kg/m^3$. At the rotating speed of $3450 rpm$, the Reynolds number based on the impeller tip was about $1,7E06$, so the internal flow was turbulent. The shear stress transport (SST)

$k-\omega$ turbulence model developed by Menter (1994) was selected because it combines the advantage of $k-\omega$ and $k-\varepsilon$ models in the near-wall regions and outside the boundary layers, respectively. In addition, it predicts better results in adverse pressure gradients and separating flow (Silva, 2012).

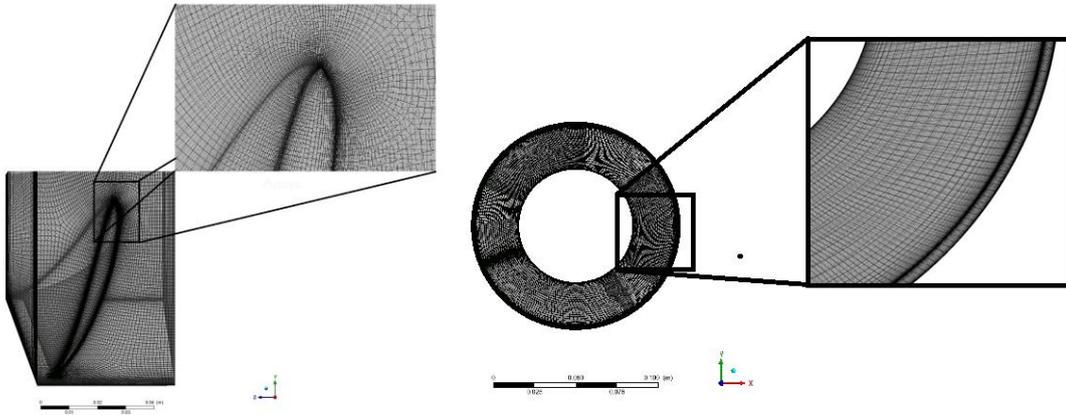


Figure 5. Volume grid of the axial flow pump

The pump head and efficiency Eq. (5) and Eq. (6) respectively of the numerical model were used as the macro parameters to monitor the convergence, and the iteration was stopped when the convergence criterion of variation of value for efficiency is less than 1% on the average of the last 20 iterations is met.

$$H = \frac{\Delta p}{\rho g} \quad (5)$$

$$n = \frac{\rho Q g H}{T \omega} \quad (6)$$

where Δp is the pressure difference between pump outlet and inlet, ρ is the water density, g is the acceleration of gravity, Q is the water flow rate and H is the head of the pump T is the torque consumed by the pump blade and ω is the angular velocity of the pump.

3. RESULTS AND DISCUSSION

Figure 6 shows efficiency and gives the pump head curve, comparing the performance curves of both CFD and experimental results. Note that each point corresponds to a specific flow rate under steady flow condition. Both the experimental and CFD simulation results show that the pressure head decreases with the increasing flow rate. It can be found that CFD simulations overestimate the performance. However, they follow the trends of the experimental results.

A qualitative agreement between the numerical and experimental results can be observed, especially near the project flow rate condition.

Pump efficiency peaks at 90% of the design goal in the numerical simulation when operating at the design point, as expected, then it drops as the flow rate goes down. What is also observed is a fast decrease in efficiency as the flow rate becomes higher than the design point. For the pump specific energy, it is possible to affirm that near the design flow rate it has less deviation in relation to the experimental results.

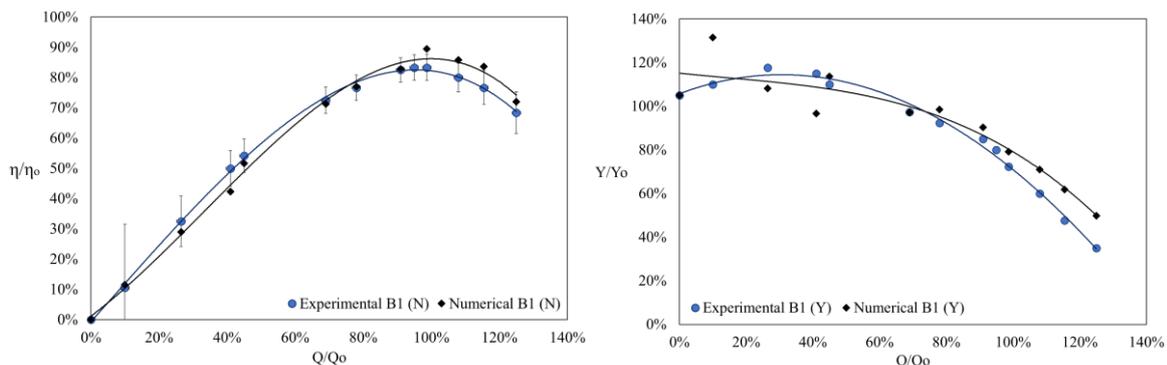


Figure 6. Numerical validation

The efficiency results are obviously overestimated because the value calculated is only the effective hydraulic power for the impeller but not for the pump. To calculate the effective power of the pump, it has to be taken into account the power losses due to the pump shaft and recirculation in the outlet caused by the flow recirculation losses.

Figure 7 shows the pressure distribution around blade hub span ($r/R=0.1$) at 3 different flow rates. Figures 8 and 9 show the same distribution but for mid span ($r/R=0.5$) and tip of the blade ($r/R=0.9$).

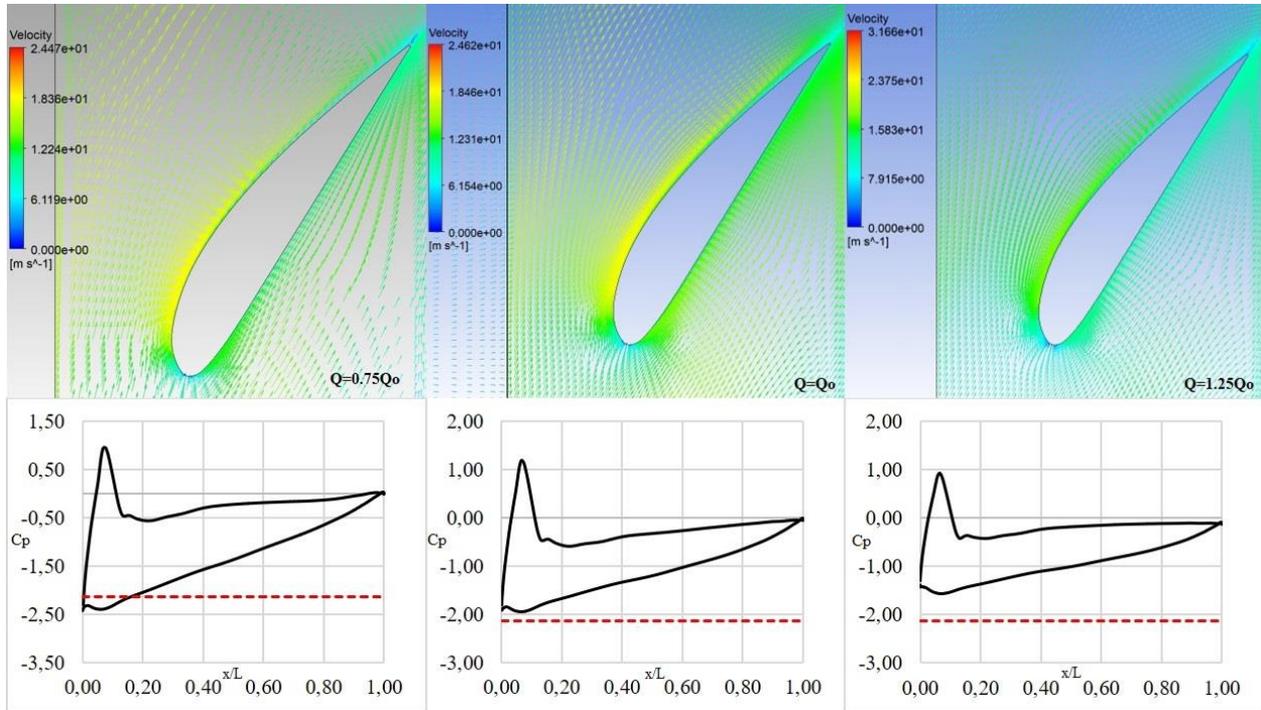


Figure 7. Pressure and Velocity distribution at hub span of the blade at different flow rates.

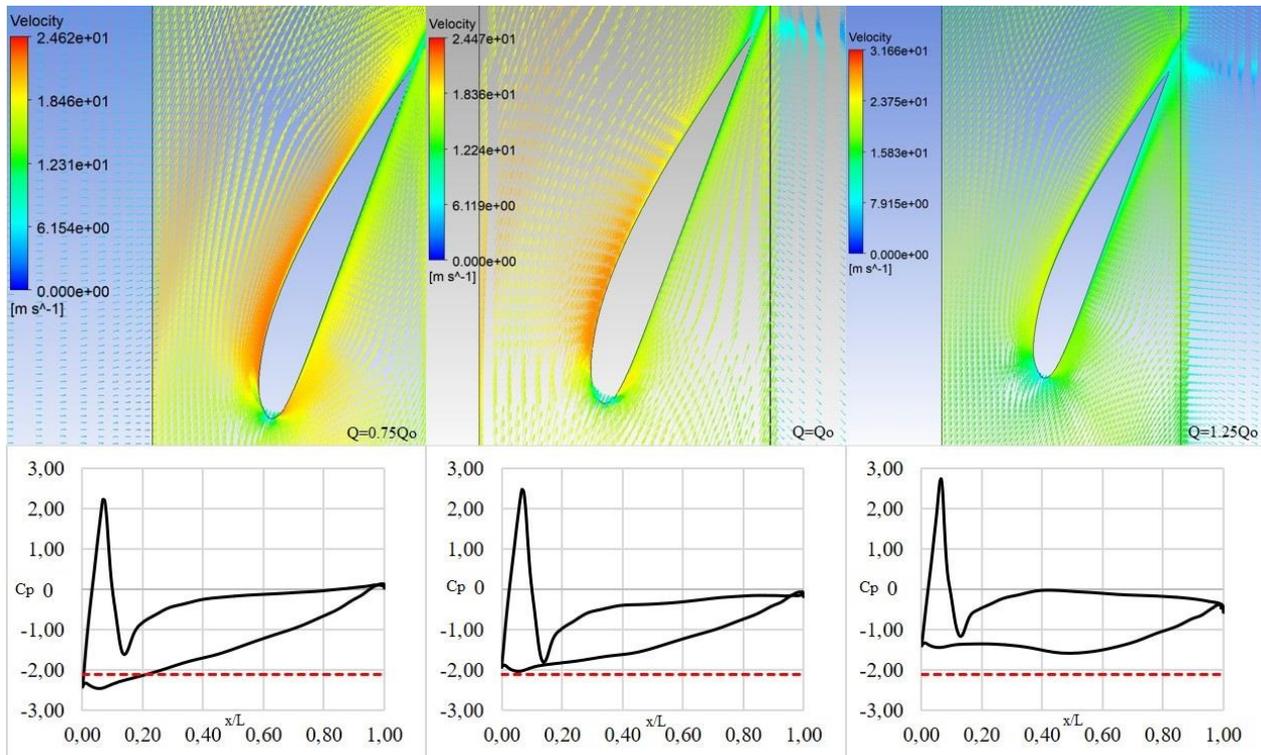


Figure 8. Pressure and velocity distribution at mid span of the blade at different flow rates.

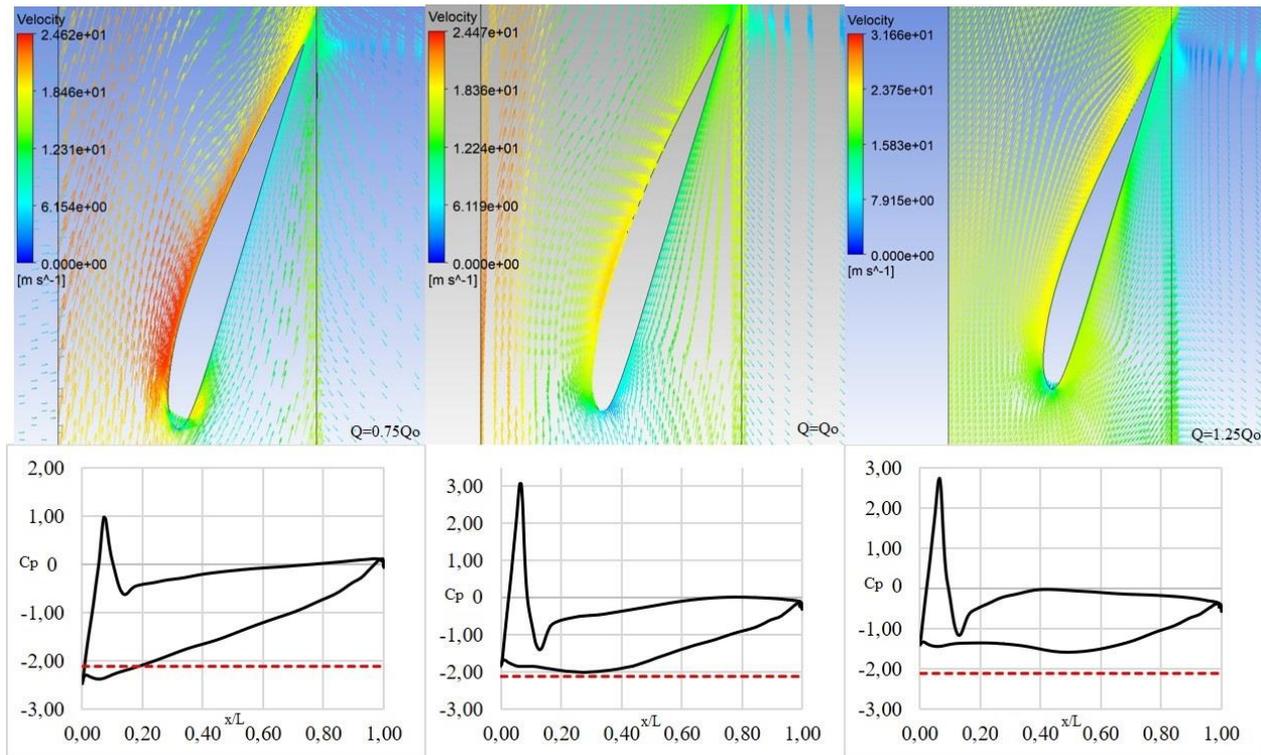


Figure 9. Pressure and velocity distribution at the tip of the blade at different flow rates.

As can be seen in the Fig. (7), Fig. (8) and Fig. (9), the minimum pressure coefficient selected for the machine is indicated in the red line (-2.12) and comes very close to the project value at the design flow rate.

It is also possible to perceive that the change in flow rate causes the pressure coefficient to diverge from the project value, which can cause friction in the upper flow rates and flow separation in the lower flow rates as well as losses in the pump performance due to these effects.

These pressure distribution deviations due to the change in flow rate can cause negative pressure in the tip of the blade so a few adjustments on the pressure coefficient project value in these strategic locations of the blade have to be made.

4. CONCLUSION

Computational fluid dynamics was used to study the internal flow field of an axial-flow pump. The available experimental tests validated the numerical simulation to predict the pump characteristics and performance curves with similar trends. The comparison between simulation and experimental results for the efficiency yields a maximum differential of around 8%. Most of this difference is due to losses caused by the behavior of the flow in the outlet of the pump due to the influence of the pendulum engine and shaft, which is not taken in to account in the simulation.

This good agreement of the simulations and measurements can be observed, and it has reasonably validated the CFD results. Therefore, SST $k-\omega$ turbulence model is able to predict the flow phenomenon in the axial flow pump. The visualization of the flow field, static pressure provided a qualitative validation of the influence of the pressure distribution in the pump performance.

The results showed the flow behavior in strategic regions of the analyzed blade. From these results, it is confirmed that the application of minimum pressure coefficient criterium can be employed on axial-flow pumps design.

The results attest to the validity of the minimum pressure coefficient criterion for the selection of cascade geometry for the best efficiency point. However, the value does not remain constant throughout the span of the blade. For flow points above design and low flow, the value of the minimum pressure coefficient is changed, which can lead to losses due to friction and flow separation, as well as cavitation in certain situations.

5. ACKNOWLEDGEMENTS

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