Hydrodynamic Analysis of the Flow in an Axial Rotor and Impeller for Large Storage Pump

A I Bosioc¹, S Muntean², I Draghici³ and L E Anton¹

¹ Department of Hydraulic Machinery, University Politehnica Timisoara, Bv. Mihai Viteazu, No. 1, Ro-300222, Timisoara, Romania
² Center for Advanced Research in Engineering Sciences, Romanian Academy – Timisoara Branch, Bv. Mihai Viteazu, No. 24, Ro-300223, Timisoara, Romania
³ S.C. AQUATIM S.A., Str. Gheorghe Lazar, No. 11A, Ro-300081, Timisoara, Romania

E-mail: alin.bosioc@upt.ro

Abstract. In hydropower systems among hydropower plants there are integrated pumping stations (PS). In order to ensure higher flow rate, the pumps have constructive differences besides regular. Consequently, the complex shape of the suction-elbow with symmetric inlet generates an unsteady flow which is ingested by impeller. These phenomena's also generate stronger unsteady flow conditions, such as stall, wakes, turbulence and pressure fluctuations, which affect the overall mechanical behaviour of the pump with vibration, noise and radial and axial forces on the rotor. Alternatively, an axial rotor can be installed in front of the impeller. In this case, the flow non-uniformity will be decreased and the static pressure will be increased at the impeller inlet. Consequently, the efficiency behaviour practically remains unchanged while the cavitational behaviour is improved. From the assembly between axial rotor and centrifugal impeller, the axial rotor usually works in cavitation and is often replaced. The paper investigates experimentally and numerically the comparison between pump impeller without and with axial rotor hydrodynamics taking into account the flow given by the symmetrical suction elbow. Full three-dimensional turbulent numerical investigation of the symmetrical suction elbow, with axial rotor and without, pump impeller and volute are performed. The hydrodynamic analysis confirms that once the axial rotor is mounted in front of the pump impeller increase the static pressure and the incidence angle is improved at the inlet of the pump impeller.

1. Introduction

In hydropower systems among hydropower plants there are integrated pumping stations (PS). The main purpose is to consume the energy excess produced by the fluctuant wind or solar power in order to store energy. Also the PS can be found in water supplies for cities, pipelines for gas and petrol, cooling units for the energy, chemical and metallurgical industry. The PS is expensive and cannot be replaced completely, preferably being refurbished.

Usually, these pumps can reach high values of flow rate at high efficiency with an acceptable development of cavitation phenomena. The pumps have constructive differences besides regular pumps in order to insure higher flow rate. Consequently, the complex shape of the suction-elbow with symmetric inlet generates an non-uniform flow field which is ingested by impeller, [1]. Consequently, the incidence angle on the impeller blades is significantly modified during one complete rotation. These mismatches induce additional losses that are referred to incidence losses which affect the hydraulic performances, pump efficiency, total head, power input and required net positive suction head (NPSH) if cavitation is taken into account. These phenomena's also generate stronger unsteady flow conditions, such as stall, wakes, turbulence and pressure fluctuations, which affect the overall mechanical behaviour of the pump with vibration, noise and radial and axial forces on the rotor, [2].

Several investigations of the flow upstream the impeller have underlined the unsteady flow generated by the suction elbow and shaft. Ludtke [3] performed a detailed experimental investigation of the flow in radial and tangential suction chambers of a compressor in order to identify the flow structure. Three
flow separation regions are identified based on these investigations: the first region is located where the pipe diverges, the second one is displaced on the convex side of the inlet bend where the flow turns from radial to axial and the third zone is behind the shaft. As a result, the hydrodynamic performances of the pump are deteriorated as well as unsteady phenomena are generated. Different engineering solution can be evaluated in order to diminish the non-uniformity generated by the suction elbow. The straight engineering solution is to change the suction elbow but it is unfeasible from economical point of view. Alternatively, an inducer can be installed in front of the impeller, Anton [4]. According with Anton, the inducer is an economically feasible solution, which could be mounted upstream the pump impeller in order to increase the pressure at the pump inlet. In this case, the flow non-uniformity will be decreased and the static pressure will be increased at the impeller inlet. Consequently, the efficiency remains practically unchanged while the cavitation behaviour is improved. The inducer is an important part of many industrial pumps and rocket turbo pumps for which very low inlet pressure is available [5]. Although the use of inducers is frequent today, several aspects of their operation still remain difficult to model, especially in unsteady-state two-phase flow cavitating regimes. From the assembly between inducer and centrifugal impeller, the inducer usually works in cavitation and is often replaced. Schilling et al. [6], introduce a new concept of an axial rotor combined pump impeller. The axial rotor is an axial bladed rotor, the main task is to generate a sufficiently high pressure level in front of the pump impeller in order to avoid any kind of cavitation of the pump inlet, [7].

The paper numerically investigates the comparison between pump impeller without and with axial rotor hydrodynamics taking into account the flow given by the symmetrical suction elbow. Section 2 presents the experimental test rig, while the section 3 describes the three-dimensional computational domains, boundary conditions and numerical setup. First, in section 4, the experimental pump characteristics are validated against numerical simulations. Next, the pump hydrodynamics are analysed for the case of the pump impeller with and without axial rotor. The last chapter draws the conclusions.

2. Experimental test rig.

An experimental test rig was developed at Politehnica University of Timisoara for assessing global performances (efficiency and cavitation) of centrifugal pumps. The test rig is completely manufactured from stainless steel. It is equipped with two tanks (each with the capacity of 1 m³), pipes, DN80 electromagnetic flow meter, valves which allow isolation of test section for quick mounting of different pump impellers which will be tested. The inlet and outlet pipes diameters are 0.1 m and 0.08 m, respectively. The test rig has mounted a symmetrical suction elbow scaled model (which is manufactured by Plexiglas), used in general for the pumping stations units [8].

At the inlet of the pump impeller (pipe diameter 0.1 m), the test rig is equipped with a gauge transducer able to measure the pressure range between -1 ÷ +2.5 bar, while a manometer registers the pressure range between 0 ÷ 6 bar at the outlet of the pump. All the electrical signals from different sensors are connected to a dedicated acquisition data system, [9]. The main hydraulic pump parameters are presented in below table:

| Parameter                        | Value   | Unit  |
|----------------------------------|---------|-------|
| Nominal pumping head $H_n$       | 44      | [m]   |
| Nominal discharge $Q_n$          | 0.0335  | [m³/s]|
| Nominal speed $n$                | 2900    | [rpm] |
| Power at nominal discharge $P_n$ | 20      | [kW]  |
| Nominal efficiency               | 72      | [%]   |
| Characteristic speed $n_c$ [10]  | 30      | [-]   |

The experimental investigations have been performed in order to assess the global performances (efficiency and cavitation) for both impellers with and without axial rotor [10].
3. Numerical simulation setup

In order to investigate the pump hydrodynamics with and without the axial rotor in front of the pump impeller, 3D numerical investigation is performed. The analysis will focus on the flow field generated by the symmetrical suction elbow and ingested by the pump impeller. The numerical domain corresponds to the pump assembly for two investigated cases. For the first case (case A), the numerical domain corresponds to the cylindrical inlet pipe, symmetrical suction elbow, a cylindrical part (between the suction elbow and the pump impeller), the pump impeller, the volute and the cylindrical outlet pipe, see Figure 2 left. The impeller model with 5 blades corresponds to a prototype pump [12] installed in a pumped storage power plant from Romania [11]. The 3D numerical domain includes a cylindrical inlet pipe, a symmetrical suction elbow, an axial rotor with three blades, a pump impeller, a volute together with cylindrical outlet pipe for the second investigated case (labelled case B). The axial rotor was designed to increase the static pressure at the pump impeller inlet improving the cavitation behavior [13]. Additionally, a more uniform flow field at the inlet of the pump impeller is expected to be obtained downstream to axial rotor. Therefore, a surface (identical with the pump impeller inlet) in order to analyze the flow non uniformities for both cases was defined.

![Figure 1](image1.png)

**Figure 1.** The experimental test rig from UPT. The dimensions are given in mm (left). The investigated pump impeller together with the axial rotor (right).

![Figure 2](image2.png)

**Figure 2.** Tridimensional computational domains include: elbow, without (left) / with (right) axial rotor, pump impeller and volute prepared for numerical simulation.

A structured mesh of 3.3 M cells is used for the case A while 4.1 M cells for the case B, respectively. The computational domains for both cases are presented in Figure 2, while Table 2 includes the detailed numerical simulation conditions.
Table 2. Mesh data and numerical conditions for the investigated cases.

| Case | A                  | B                  |
|------|--------------------|--------------------|
| Type of the mesh | hexahedral | hexahedral |
| Total number of mesh cells | 3.382.675 | 4.110.925 |

Numerical setup

| Case | A                  | B                  |
|------|--------------------|--------------------|
| pump impeller speed | 3000 rpm | 3000 rpm |
| axial rotor speed | [ ] | 3000 rpm |
| Discharge | 0.0335 m$^3$/s | 0.0335 m$^3$/s |
| Turbulence model | k-ω SST | k-ω SST |
| Inlet conditions | normal velocity 3.525 m/s | normal velocity 3.525 m/s |
| Outlet conditions | static pressure 4 bar | static pressure 4 bar |

FLUENT [14] software was used for the flow computation considering the k-ω SST turbulence model. The second order schemes and SIMPLE algorithm for coupling velocity-pressure fields are selected. The threshold value for numerical solution convergence is imposed below to 1e-6. The simulation time step, dt= 1 ms is considered in this computation with 20 inner iterations on each time step. Moreover, the mean static pressure measured experimentally at the outlet of the pump, is imposed at the outlet section of the numerical domain in both cases.

4. Results and analysis

From experimental investigation and numerical simulation, a first analysis consisted to evaluate and validate the pumping head without and with axial rotor. Both cases have been compared taking into consideration the similar discharge and same pump impeller speed. For the case A, the numerical simulation estimates the pumping head with 6.6% more, see Table 3. The numerical simulation estimates the pumping head with 6% more than measured value in the case of pump impeller coupled with axial rotor (case B).

Table 3. Comparison the pumping head between experimental and numerical cases.

| Cases            | Discharge [m$^3$/s] | Head [m] | Error [%] |
|-----------------|---------------------|----------|-----------|
| without axial rotor (case A) | Experimental investigation | 45.2 |          |
| with axial rotor (case B) | 3D numerical simulation | 0.0335 | 47.9 |
|                  |                      |          | 6%        |

One can observe in Table 3 that the pumping head remains practically unchanged when the axial rotor is installed in front of the pump impeller, [15]. However, the axial rotor is expected to increase the static pressure and flow incidence on the impeller blades. Accordingly, our analysis focus on following two aspects: (i) the distribution of the minimum static pressure and (ii) the distribution of the flow at the inlet of the pump impeller.
The location of the minimum pressure in both cases is evidenced in Figure 3. As it is expected, the minimum pressure region (light blue spot) is identified for case A (pump impeller only) on the suction side of each pump impeller blade near to the leading edge. Moreover, the minimum static pressure region is significantly different from one blade to another due to the non-uniform flow field generated by the suction elbow [10].

The minimum pressure region is passed from the pump impeller blades to the inducer blades when the axial rotor is mounted in front of the pump impeller (case B) [16]. Moreover, the extension of the regions with minimum static pressure is diminished. As a result, the cavitational behaviour of the pump is improved. This statement is supported by the experimental investigations performed on the test rig [10].

The tangential velocity component has an non-uniform distribution provided by vortices generated by three-dimensional complex geometry of the symmetrical suction elbow mounted upstream to the pump impeller [17],[18]. This provides a circumferential distribution of the relative flow angle $\beta$ at the leading edge of the blades of the pump impeller, which causes unsteady loadings on its, [17]. A cross section located at the inlet of the pump impeller is defined (see Figure 2 - analysis surface) in order to be investigated the distribution of the relative flow angle $\beta$ for both cases. The relative flow angle $\beta$ distribution on this cross section is only generated by the symmetrical suction elbow for the case A while it distribution for case B is mainly influenced by the axial rotor and less by the symmetrical suction elbow.

The relative flow angle $\beta$ is defined according to the following formula:

$$\beta = \arctan \left( \frac{v_m}{u - v} \right)$$  \hspace{1cm} (1)

$$v_m = \sqrt{v^2 + v^2}$$  \hspace{1cm} (2)

where:

$$u = r \cdot \omega$$  \hspace{1cm} (3)
The relative flow angle $\beta$ takes into consideration the meridian velocity $v_m$ (composed by in axial velocity $v_a$ and radial velocity $v_r$), the transport velocity $u$ and the tangential velocity $v_t$.

According with Figure 4 for the case with pump impeller only (case A), the relative flow angle $\beta$ has a consistent variation near the sleeve and casing. The maximum non-uniformity of the relative flow angle is concentrated in two opposite regions. For this case the variation is between $\pm 10^\circ$, while in the middle of the surface the relative flow angle varies between $\pm 5^\circ$.

![Figure 4. Distribution of the relative flow angle \( \beta \) on the cross section located at the inlet of the pump impeller for both cases: case A (left) and case B (right)](image)

When the axial rotor is installed in front of the pump impeller (case B), a maximum variation is observed near the casing of $\pm 10^\circ$, while in the middle of the surface and close to the sleeve the maximum variation is between $\pm 3^\circ$. From the quantitative evaluation of the relative flow angle, one can observe a large non-uniformity for case A. Once the axial rotor is mounted in front of the pump impeller, the relative flow angle $\beta$ became more uniform on cross section located upstream to the pump impeller. However, it can be seen a minimum area near the casing due to the non-uniformity created by the symmetrical suction elbow and ingested by the axial rotor.

Particularly, the relative flow angle $\beta$ is determined on the horizontal lines located downstream to both ribs (Rib 1 and Rib 2) in order to be quantified the flow field deviation from axial flow, Figure 5.

![Figure 5. The radial distribution of the relative flow angle $\beta$ on the horizontal lines located downstream to both ribs: case A (left) and case B (right)](image)
As a result, this distribution reveals an excess near the sleeve and a deficit near the casing for case A (pump impeller only - left). When the axial rotor is mounted in front of the pump impeller (case B) the distribution can be considered practically constant from sleeve to casing. From hydrodynamic point of view, a constant radial distribution of the relative flow angle from sleeve to casing is preferred [19].

The deviation angle ($i$) from axial flow is computed extracting the radial distribution of the relative flow angle associated to the axial flow inlet (Figure 5) and from the relative flow angle at the inlet of the pump impeller plotted in Figure 4. For case A the deviation angle is directly affected by the flow from the symmetrical suction elbow, while for case B the deviation angle takes into consideration the flow from the symmetrical suction elbow and the influence of the axial rotor mounted in front of the pump impeller. Accordingly the incidence angle was calculated with the following formula:

$$i = \beta - \beta_{axial}$$  \hspace{1cm} (5)

Consequently, a non-uniform deviation angle can be observed for case A, Figure 6. This non-uniform flow from the impeller inlet induces an unsteady loading on the pump impeller blades. For case B the deviation angle tends to have a constant distribution, leading to a more uniform blade loading.

![Figure 6](image)

Figure 6. The deviation angle map on the cross section located at the inlet of the pump impeller: case A (left) and case B (right)

On the other hand, in order to quantitatively evaluate the deviation angle three circles (near to hub labelled, $r=15\%$ near the sleeve, in the middle $r=50\%$, and near to casing $r=85\%$) are figured out in Figure 6. The distribution of the deviation angle for both cases is shown in Figure 7.

![Figure 7](image)

Figure 7. The deviation angle plotted on three circles located on the cross section located at the pump impeller inlet: case A (left) and case B (right)
For pump impeller only (case A), the deviation angle has a variation of ±10° near the sleeve and casing. When the axial rotor is used (case B), the deviation angle varies near the casing, while in the middle and near the sleeve is approximately obtained a constant distribution on all section. In our case, a more uniform deviation angle is obtained for case B using the axial rotor in front of the pump impeller. In this case the appropriate deviation angle cover 85% on the inlet cross section, while for case A the appropriate deviation angle cover only 40% on the inlet cross section of the pump impeller, respectively. Clearly, a more uniform flow field is ingested by the pump impeller when an axial rotor is installed upstream to it in order to “mix” the non-uniformity generated by the suction elbow.

5. Conclusions
The paper presents the experimental and full 3D numerical investigations performed into a centrifugal pump in order to evaluate the hydrodynamic behaviour at the pump impeller inlet. Two cases are investigated: first case with pump impeller only and the second case with the axial rotor mounted in front of the same pump impeller. The axial rotor was designed in order to increase the static pressure at the pump inlet in order to improve the cavitational behaviour [13]. Firstly, the pumping head is experimentally obtained. The numerical results for both cases are validated against experimental data to assess the accuracy of the numerical procedure.

Secondly, the minimum static pressure was identified based on numerical investigation. The analysis showed that once the axial rotor is mounted, the minimum pressure increase moreover is moved from the leading edge of the impeller blades to the inducer domain. As a result, when the axial rotor is mounted in front of the pump impeller improves the cavitational behaviour.

Next, the non-uniformity flow generated by suction elbow is deeply analysed because it is ingested by the impeller. The relative flow angle $\beta$ and the deviation angle $i$ have been calculated at the pump impeller inlet. When the axial rotor is used, the deviation angle is improved with 40%. As a result, a more uniform flow field is ingested by the pump impeller when an axial rotor is installed upstream to it in order to “mix” the non-uniformity generated by the suction elbow.

Acknowledgments
This work was supported by a grant of the Romanian National Authority for Scientific Research and Innovation, CNCS – UEFISCDI, project number PN-II-RU-TE-2014-4-1089.

References
[1] Gülich J F 2014 Centrifugal Pumps, 3rd ed., (Berlin: Springer Verlag)
[2] Bois G 2006 Introduction to design and analysis of high speed pumps. In Design and Analysis of High Speed Pumps, Educational Notes RTO-EN-AVT-143, Paper 1, Neuilly-sur-Seine, France: RTO. Available from: http://www.rto.nato.int/abstracts.asp, pp. 1-1 – 1-20.
[3] Ludtke A 1985 Centrifugal process compressors – radial vs. tangential suction nozzles, ASME Paper 85-GT-80.
[4] Anton L 1994 Îmbunătățirea caracteristicilor cavitaționale la pompele cu impulsor, PhD. Thesis, Timișoara.
[5] Japikse D 2001 Overview of industrial and rocket turbopump inducer design, CAV2001/sessionB7.001.
[6] Schilling R, Schber G, Hutter M and Thum S 2014 Development of a radial-axial pump-turbine for decentralized small pumped storage power plants, WasserWirtschaft Extra, 1, 43-47.
[7] Schober G, Hutter M and Schilling R 2014 Numerical analysis of a combined axial-radial pump-turbine, 18th International Seminar on Hydropower Plants, Innovations and Development Needs for Sustainable Growth of Hydropower, pp. ISHPP2014 ID 26, Viena.
[8] Draghici I, Bosioc A I, Muntean S and Anton L E 2014 Experimental investigation of the non-uniform inflow generated by the symmetrical section elbow of a large pump, U.P.B. Sci. Bull.,
Series D, **76**, 3, 207 – 214.

[9] Stanciu I R, Turcin I, Muntean S and Anton L E 2013 Cellular wind-power integration using remotely controlled pump hydro energy storage”, Proc. of the Romanian Academy, Series A, **14**, 3, 242-249.

[10] Muntean S, Bosioc A I, Draghici I and Anton L E 2016 Hydrodynamic analysis of the flow field induced by a symmetrical suction elbow at the pump inlet. 28th IAHR Symposium on Hydraulic Machinery and Systems, (Grenoble, France) (submitted)

[11] Cojocar M 2008 Hidroconstructia: Hydropower constructions, 2nd ed., 1, (Bucharest: Romania)

[12] Anton A 2010 In situ performance curves measurements of large pumps, *IOP Conf. Ser.: Earth Environ. Sci.*, **12**, 012090, 1-10.

[13] Moisa I, Susan-Resiga R, Muntean S 2013 Pump inducer optimisation based on cavitation criterion, Proceedings of the Romanian Academy Series A, 14(4), 317-325

[14] Fluent Inc., 2006) FLUENT User’s guide 6.3, Lebanon, USA

[15] Muntean S, Draghici I, Ginga G, Anton L E, Baya A 2015 Hydrodynamic design of a storage pump impeller using inverse method and experimental investigation of the global performances, WasserWirtschaft Extra, **1**, 28 – 32.

[16] Ginga G, Stanciu I R, Muntean S, Baya A and Anton L E 2012 3D Numerical Flow Analysis and Experimental Validation into a Model Impeller of a Storage Pump, in Proc. of the Conference Modelling Fluid Flow (CMFF’12), Budapest, Hungary, **2**, 804 – 811.

[17] Draghici I, Bosioc A I, Muntean S and Anton L E 2014 LDV measurements of the velocity field on the inlet section of a pumped storage equipped with a symmetrical suction elbow for variable discharge values, IOP Conf. Series: Earth and Environ. Sci., **22**, 032017, 1 – 9.

[18] Muntean S, Škerlavaj A, Draghici I and Anton L E 2015 Numerical analysis of the flow non-uniformity generated by symmetrical suction elbows of the large storage pumps, Proc. of the 6th IAHR Int. Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems, Ljubljana, Slovenia, 1-8.

[19] Ardizzone G and Pavesi G 1998 "Optimum incidence angle in centrifugal pumps and radial inflow turbines", Proc Instn Mech Engrs Part A, **212**, 97-107.