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Article

A Performance Prediction Method for Pumps as Turbines (PAT) Using a CFD Modeling Approach

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Abstract: Small and micro hydropower represents an attractive solution for electricity generation, with low cost and low environmental impact. The pump-as-turbine (PAT) approach has promise in this application owing to its low purchase and maintenance costs. In this paper, a new method to predict the inverse characteristic of industrial centrifugal pumps is presented. This method is based on results of simulations performed with commercial three-dimensional CFD software. Model results have been first validated in pumping mode using data supplied by pump manufacturers. Then, results have been compared to experimental data for a pump running in reverse condition. Experimentation has been performed on a dedicated test bench installed in the Department of Civil Construction and Environmental Engineering of the University of Naples Federico II. Three different pumps, with different specific speeds, have been analyzed. Using the model results, the inverse characteristic and the best efficiency point have been evaluated. Finally, results of this methodology have been compared to prediction methods available in the literature.

Keywords: energy saving; PAT; Urban Hydraulic Network; numerical modeling

Nomenclature

PRV	Pressure Reducing Valve
PAT	Pump as Turbine
CFD	Computational Fluid Dynamic
BEP	Best Efficiency Point
H_p	Pump head
Q_p	Pump flow rate
P_p	Pump power
η_p	Pump overall efficiency
H_t	Turbine head
Q_t	Turbine flow rate
P_t	Turbine power
η_t	Turbine overall efficiency
ψ	Specific head
φ	Specific capacity
π	Specific power
N_s	Pump specific speed
N_{st}	Turbine specific speed

1. Introduction

Electricity generation presents many issues and is studied with different techniques in order to reduce its production cost and environmental impact. Conventional production with fossil

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fuels presents problems associated with the high cost, rapid depletion and detrimental environmental effects of these fuels. Renewable energy is probably the best solution for the environmental issues, and many solutions have been developed since the last century, such as hydropower, hydrogen, fuel cells, biofuels, and solar power generation.

Among the renewable sources, small hydropower represents a very attractive source of energy generation. In many countries, small and micro hydropower is an important means of electricity generation. An efficient solution, from the point of view of energy efficiency, is the adoption of a turbine, but the purchase and maintenance costs of turbines render their application economically unattractive, especially for small hydropower [1-6].

Reverse-running centrifugal pumps (also called PAT, pumps as turbines) are a solution for generating and recovering power in small and micro hydropower situations. Pumps are relatively simple machines, inexpensive (compared to a hydraulic turbine), and readily available worldwide. It is estimated that the capital payback period of a reverse-running pump in the 5-50 kW range is less than two years [7, 8]. Moreover, the use of PAT could be suitable because manufacturers of turbines worldwide are less numerous than pump producers, the market for turbines is small compared to that for pumps, and pumps are mechanically simple and require less maintenance. Moreover, an integral pump and electric motor can be purchased for use as a turbine and generator set; pumps are available in a wide range of heads and flows and in a large number of standard sizes. Generally, pumps have short delivery times, spare parts (such as seals and bearings) are easily available and the installation can be done using standard pipes and fittings.

The use of a pump running in reverse mode to generate electricity is not new; the first applications started almost 80 or 90 years ago and many theoretical and experimental studies have been done [2-6, 9, 10]. Much research is still being conducted, especially to predict the operating conditions and the efficiency of centrifugal pumps running in reverse [11].

The selection of a proper PAT for an existing site represents a critical issue because pump manufacturers do not supply the characteristic of the pump running in reverse. Many methods have been used to predict the inverse characteristic of a pump, based on numerical models, experiments, or theoretical procedures [5-6].

Several studies based on a modeling approach with CFD code are available and generally showed good agreement with the available experimental data [4, 10]. A study [4] carried out with a computational model of a PAT is based on the concept called “flow zone”. The flow regime within a PAT is divided into four major flow regions (volute casing, impeller, casing outlet and draft tube). A comparison has been made between the experimental and numerical results of a single stage end suction centrifugal pump that was operated in turbine mode at a speed of 800 rpm. CFD predictions of the hydraulic parameters were in good agreement with the experimental results, but deviations (within 5% to 10%) have been found at certain load regions. Nautiyal et al. [5, 6] carried out a study on the application of CFD and its limitations for PAT using cases reported by previous researchers [4, 9, 10]. The study reported that CFD analysis was an effective design tool for predicting the performance of centrifugal pumps in turbine mode and for identifying the losses in turbo-machinery components such as the draft tube, impeller and casing, but there was some deviation between the experimental results and the CFD modeling results. Barrio et al. [11] carried out a numerical investigation on the unsteady flow in commercial centrifugal pumps operating in direct and reverse mode with the help of CFD code. The results of their simulation were in good agreement with the experimental results. The study revealed that in the reverse mode, the flow only matched the geometry of the impeller at nominal conditions; re-circulating fluid regions developed at low flow rates (near the discharge side of the blades) and high flow rates (near the suction side).

Many correlations based on theoretical approaches are available to predict the performance of a PAT. Several researchers (Stepanoff, Childs, Sharma, Wong, Williams, Alatorre-Frenk, and others) have presented correlations for predicting the performance of a pump-as-turbine

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[5]. These correlations were based upon either pump efficiency or specific speed. However, the deviation between experimental and predicted reverse operation of standard pumps has been found to be more than 20% [12]. The objective of these correlations is to calculate the best efficiency point (BEP) of pumps for operation in turbine mode by using the pump operation data provided by the manufacturer. In 1962, Childs [13] presented a PAT prediction method based on the efficiency of the pump. A similar approach was then presented by McClaskey & Lundquist [14] and Lueneburg & Nelson [15] in 1976 and 1985, respectively. Hancock [16] stated that for most pumps the turbine BEP lies within 2% of the pump mode BEP. Grover and Hergt [17, 18] proposed a PAT prediction method based on specific speed for the turbine mode (obtained similar to the specific speed for a pump). Grover's method is applicable for the turbine mode specific speed range 10-50 [17]. A comparison between experimental results and the methods proposed by the above researchers show relatively large deviations; therefore, the use of these formulae has been confined to approximate selection of PATs.

Finally, a large number of experimental studies can be used to evaluate the inverse characteristic. These are often limited to the specific pumps tested, so that they cannot serve as a valid tool for pump selection, but are very useful for tuning and validating theoretical and modeling analyses.

In this paper, authors show a methodology for obtaining the reverse characteristic of a pump, starting from the results of three-dimensional CFD models. The results are compared with data available in the literature.

2. Prediction Methods

In this paper, a methodology to calculate the inverse characteristic of a commercial pump is presented. This methodology is based on the results of CFD modeling using a commercial code developed to simulate centrifugal machines. In this paragraph, a short description is given of several prediction methods available in literature. These methods will be used to analyze the proposed methodology and to discuss the results.

The following equations summarize different methods to predict the pump inverse characteristic. They are based on theoretical or experimental analyses [4, 12].

$$\frac{H_t}{H_p} = \frac{1}{\eta_P} \qquad \frac{Q_t}{Q_p} = \frac{1}{\sqrt{\eta_P}} \qquad \eta_T = \eta_P \qquad N_{st} = N_s\eta_p \tag{1}$$ Stepanoff

$$\frac{H_t}{H_p} = \frac{1}{0.85{\eta_p}^5+0.385} \qquad \frac{Q_t}{Q_p} = \frac{0.85{\eta_p}^5+0.385}{2{\eta_p}^{9.5}+0.205} \qquad \eta_t = \eta_p - 0.03 \tag{2}$$ Alatorre and Frenk

$$\frac{H_t}{H_p} = \frac{1}{{\eta_p}^{1.2}} \qquad \frac{Q_t}{Q_p} = \frac{1}{{\eta_p}^{0.8}} \qquad P_t = P_p \qquad \eta_t = \eta_p \tag{3}$$ Sharma

$$\frac{H_t}{H_p} = -1.4 + \frac{2.5}{\eta_p} \qquad \frac{Q_t}{Q_p} = -1.5 + \frac{2.4}{{\eta_p}^2} \qquad \frac{\eta_t}{p} = 1.158 - 0.265N_{st} \tag{4}$$ Schmiedl

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$$\begin{cases} \frac{H_t}{H_p} = 2.693 - 0.0229N_{st} \\ \frac{Q_t}{Q_p} = 2.379 - 0.0264N_{st} \\ \frac{\eta_T}{\eta_P} = 0.893 - 0.0466N_{st} \end{cases} \quad \text{Grover} \tag{5}$$

$$\frac{H_t}{H_p} = 1.3 - \frac{6}{N_{st}-3} \qquad \frac{Q_t}{Q_p} = 1.3 - \frac{1.6}{N_{st}-5} \quad \text{Hergt} \tag{6}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p} \qquad \frac{Q_t}{Q_p} = \frac{1}{\eta_p} \qquad \eta_t = \eta_p \quad \text{Childs} \tag{7}$$

Derakhshan and Nourbakhsh [19] introduce a method based on theoretical analysis to evaluate the BEP of an industrial centrifugal pump. This method is based on the geometrical and hydraulic characteristics of the pump in direct mode. The final formula to evaluate the turbine's maximum efficiency is:

$$\eta_t = \frac{P_{nt}}{\gamma \cdot Q_t \cdot H_t} = \frac{\gamma \cdot Q_t \cdot H_t - P_{vt} - P_{lt} - P_{et} - P_{it} - P_{mt}}{\gamma \cdot Q_t \cdot H_t} \tag{8}$$

Presented methods are very different and based on different hypotheses. Each method can be used only for a limited set of pumps and none of them allows the reverse running conditions to be accurately predicted for all geometries and over a wide range of pump specific speeds.

3. Simulation Model

Three different centrifugal pumps have been modeled in order to obtain the data necessary to predict the performance of the pumps by the described procedures. The analyzed pumps are commercial ones and have three different specific speeds. The main characteristics are summarized in table 1.

Table 1. Pumps characteristics

	Impeller Diameter [mm]	**Delivery Outlet Diameter [mm]**	**H_{bep} [m]**	**Q_{bep} [m^3/s]**
(N_s 37.6)	190	80	39	148
(Ns 20.5)	200	70	60	45.4
(Ns 64.0)	120	80	3.9	54

It was decided to use pumps with different heads (from 3.9 to 60 m) and flow rates (from 45.4 to 148 m^3/s) to have different geometries and operating conditions to better test the prediction method.

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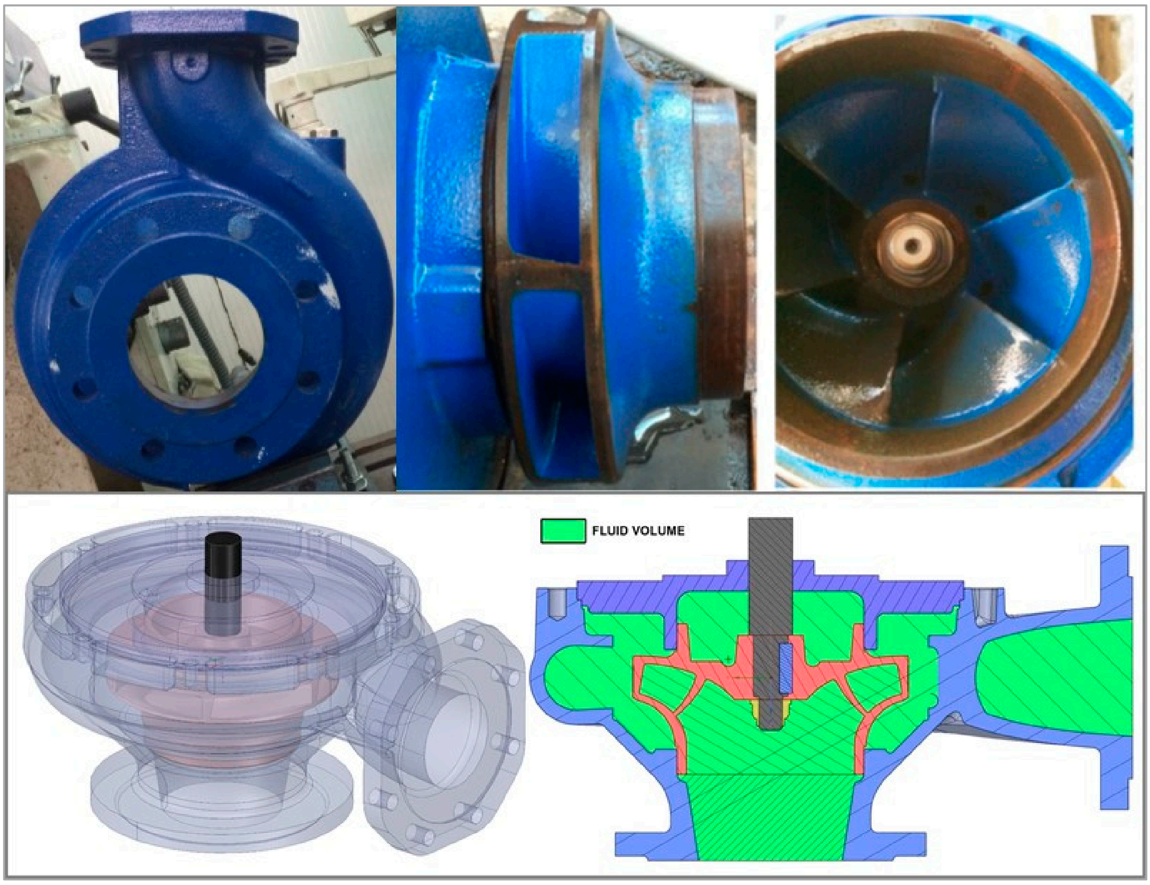

Figure 1. Geometry and fluid volume

The simulation models were built using the commercial code PumpLinx®, CFD software developed by Simerics Inc. [20 – 24]. It numerically solves the fundamental conservation equations of mass, momentum and energy and includes robust models of turbulence and cavitation. PumpLinx® has its own grid generator and uses a body-fitted binary tree approach. Moving and stationary fluid volumes are treated simultaneously, and each volume connects to the other via an implicit interface.
Meshes of each pump model are shown in 2 where the rotors are green while all other fluid volumes are transparent.

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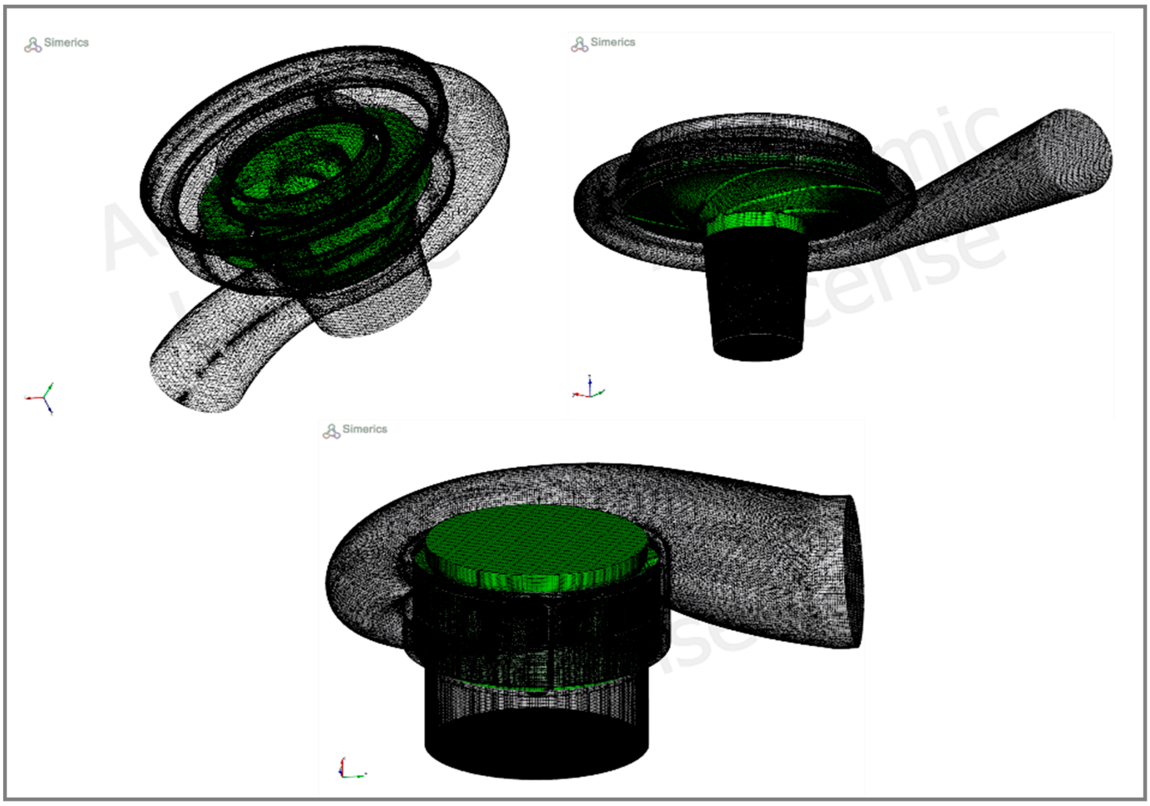

Figure 2. Binary tree mesh for three pumps

The details of the three-dimensional grids for each pump are shown below. Simulations have been run with an Intel(R) Xeon(R) CPU 2.66GHz (two processors).

PUMP 1 (N_S=37.6):

- Total number of cells : 851.673
- Total number of faces : 3.383.745
- Total simulation time: 8.9 h as Pump, 9 h as PAT

PUMP 2 (N_S=20.5):

- Total number of cells: 1.039.450
- Total number of faces: 3.926.412
- Total simulation time: 4.2 h as Pump, 4.8 h as PAT

PUMP 3 (N_S= 64):

- Total number of cells: 324.596
- Total number of faces: 3.348.318
- Total simulation time: 4.5 h as Pump, 4.8 h as PAT

As said, using the presented models it is possible to investigate the internal fluid dynamics in the direct (as pump) and reverse (as turbine) modes. Model results for the Pump 1 are shown in figure 3; in figure 3a the pressure distribution at the BEP is presented for both working modes in the pressure range (0 ÷ 9) bar.
Figure 3b shows the velocity vectors in the fluid volume; the velocity range (0 ÷ 32) m/s is the same for the direct and reverse modes. In this picture (figure 3b), it is possible to visualize the flow evolution inside the machine and the acceleration/deceleration of the fluid.
Both figures confirm that the velocity is higher in the reverse mode.

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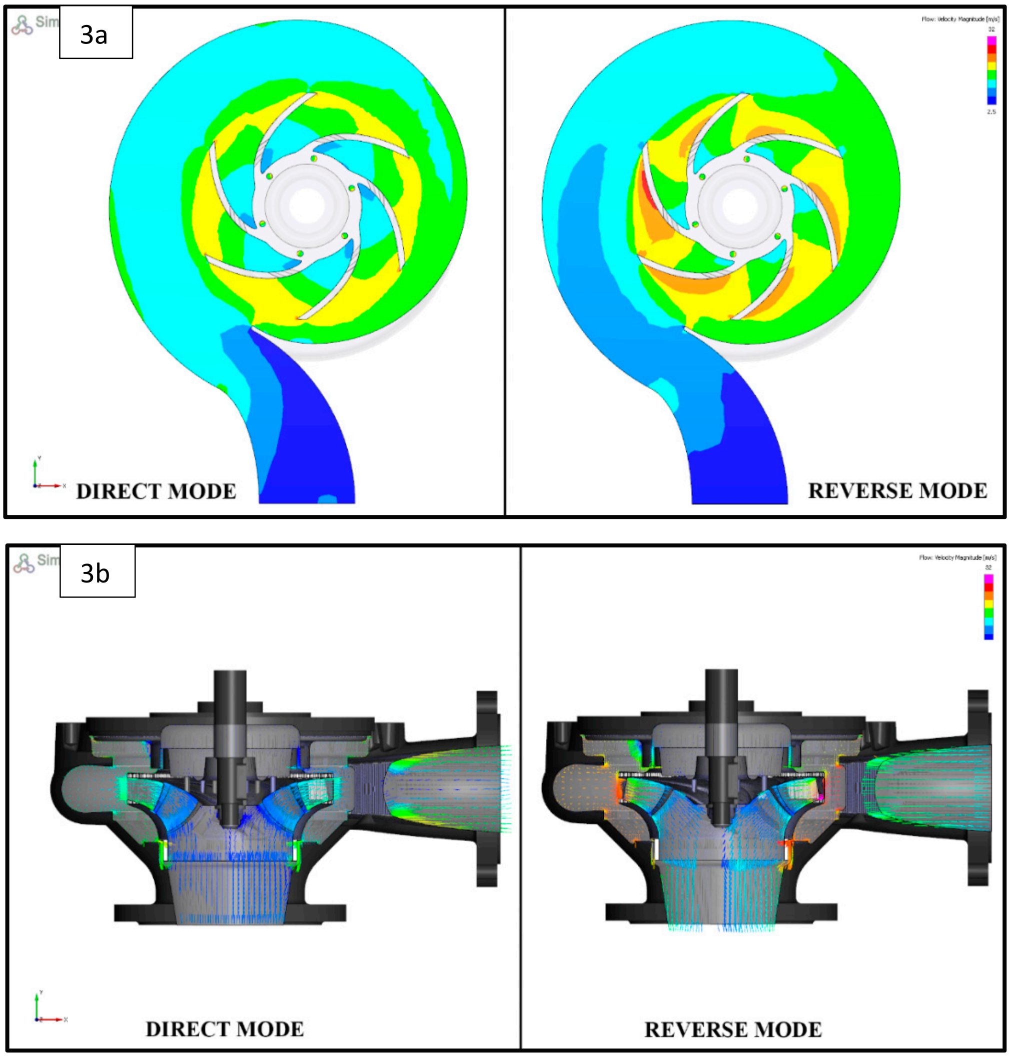
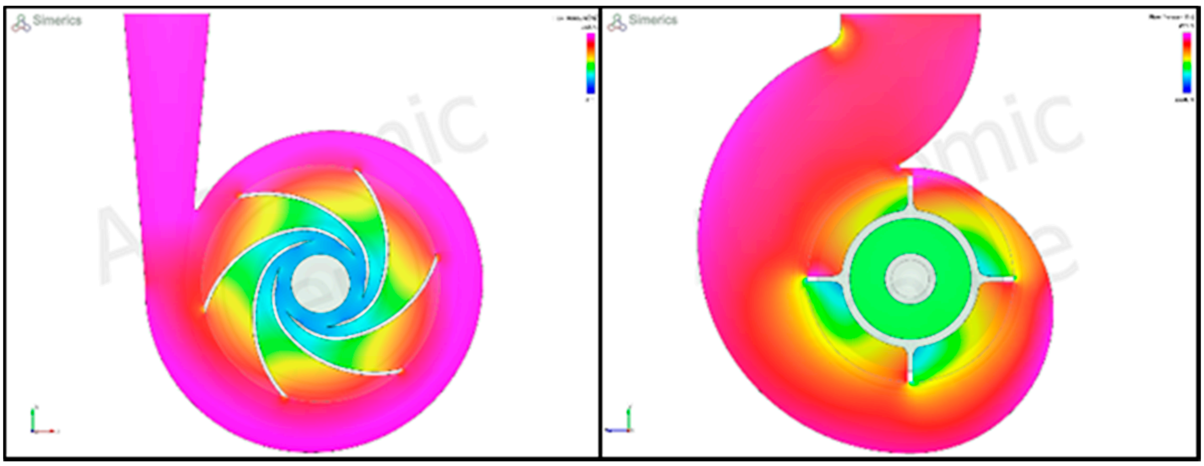


Figure 3. Model results pump 1 (N_s = 37.6)

Similarly, figure 4 shows the pressure distributions for pump 2 (N_S= 20.5) and pump 3 (N_S= 64).



N_s= 20.5 N_s= 64

Figure 4. Pressure distribution in the fluid volume of pumps 2 and 3

For the direct mode, CFD models have been validated using the data supplied by the pump

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manufacturers. In Figure 5, the head vs. flow rate plot (as the blue curves) are shown. Across the range of flow rates (30 – 207 m^3/h), the head varies from 47 to 3 m. Plots in Figure 4, the model results is in red.

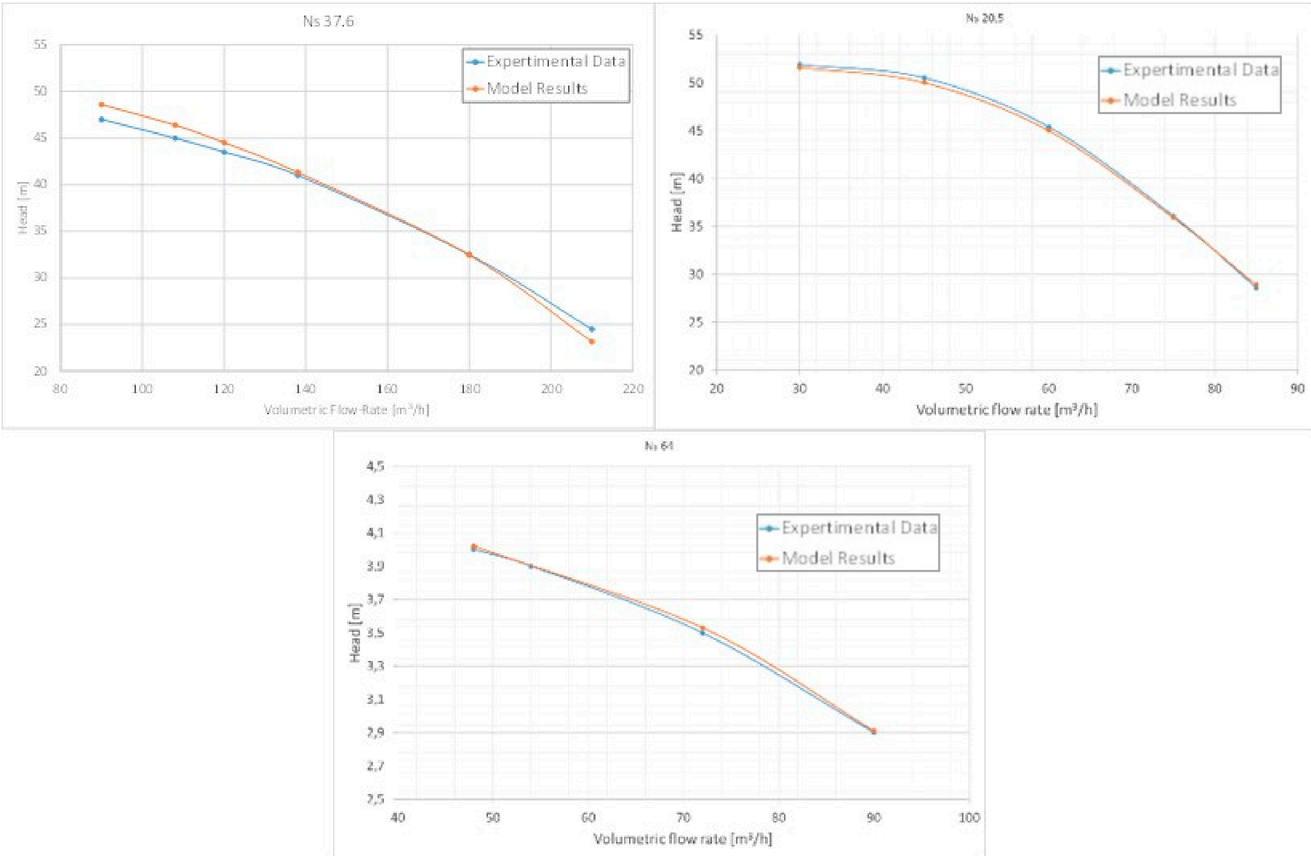

Figure 5. Model validation: Comparison between manufacturer data and model results

The comparison in figure 5 demonstrates the accuracy of the adopted methodology; in fact, the percentage error is always less than 4% while for many points the error is near zero.

4. N_s 37.6 Pump Model Validation with Experimental Data

Once the model had been validated in the pumping mode, it was decided to validate it also in the reverse mode, to assess whether the model reproduces the turbine mode well. Because the proposed methodology is based only on the results of the CFD model in reverse conditions, the validation in reverse condition was necessary to confirm the entire methodology.

The model of centrifugal Pump 1 has been validated with data from an experiment performed on a dedicated test bench of the Department of Civil Construction and Environmental Engineering of the University of Naples Federico II. The bench is enables testing a centrifugal pump running in reverse mode. The aim of this activity was to further validate the simulation model in reverse conditions.

The test bench reproduces a full-scale hydraulic network, made up of four nodes (figure 6). An external pump increases the water pressure to simulate the behavior of a real urban network while an air chamber stabilizes the flow rate. The tested pump has been installed in one node where two pressure-reducing valves (PRV) regulate the water flow rate and the pressure at the inlet and outlet of the pump.

The electric motor of the pump is linked to an inverter and the produced electrical power is connected to the urban power grid. In the node, two pressure transducers, P_1 and P_2 (Burkert®

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model 8314), and a flow meter Q (Siemens® mag 500) have been installed. All test bench data have been acquired by a home-made acquisition system. Furthermore, a 360-tooth encoder has been installed on the electrical motor to acquire the shaft speed.

Experiments have been performed only in steady-state conditions, varying the water flow rate and the pressure at the inlet of the pump, for different shaft speeds. In particular, the flow rate has been varied between (8 - 21) l/s, and the shaft speed between 300 and 2200 rpm. During the test, pressure at the inlet and outlet of the pump have been acquired at a sample frequency of 1 Hz.

As stated above, the tests have been done in steady-state conditions, running the pump in reverse mode. The flow rate, the pressures at the inlet and outlet of the pump and the shaft rpm were measured. In figure 7, all results of the experimental campaign are shown.

To examine the PAT performance, the total head [m] versus shaft rpm is reported, varying the water flow rate for all the examined conditions. Results confirm what is known from the literature: the PAT head increases with the rpm and with the flow - rate, and it can be easily noted that, for the tested conditions, the head varies between 0.1 and 1.8 meters.

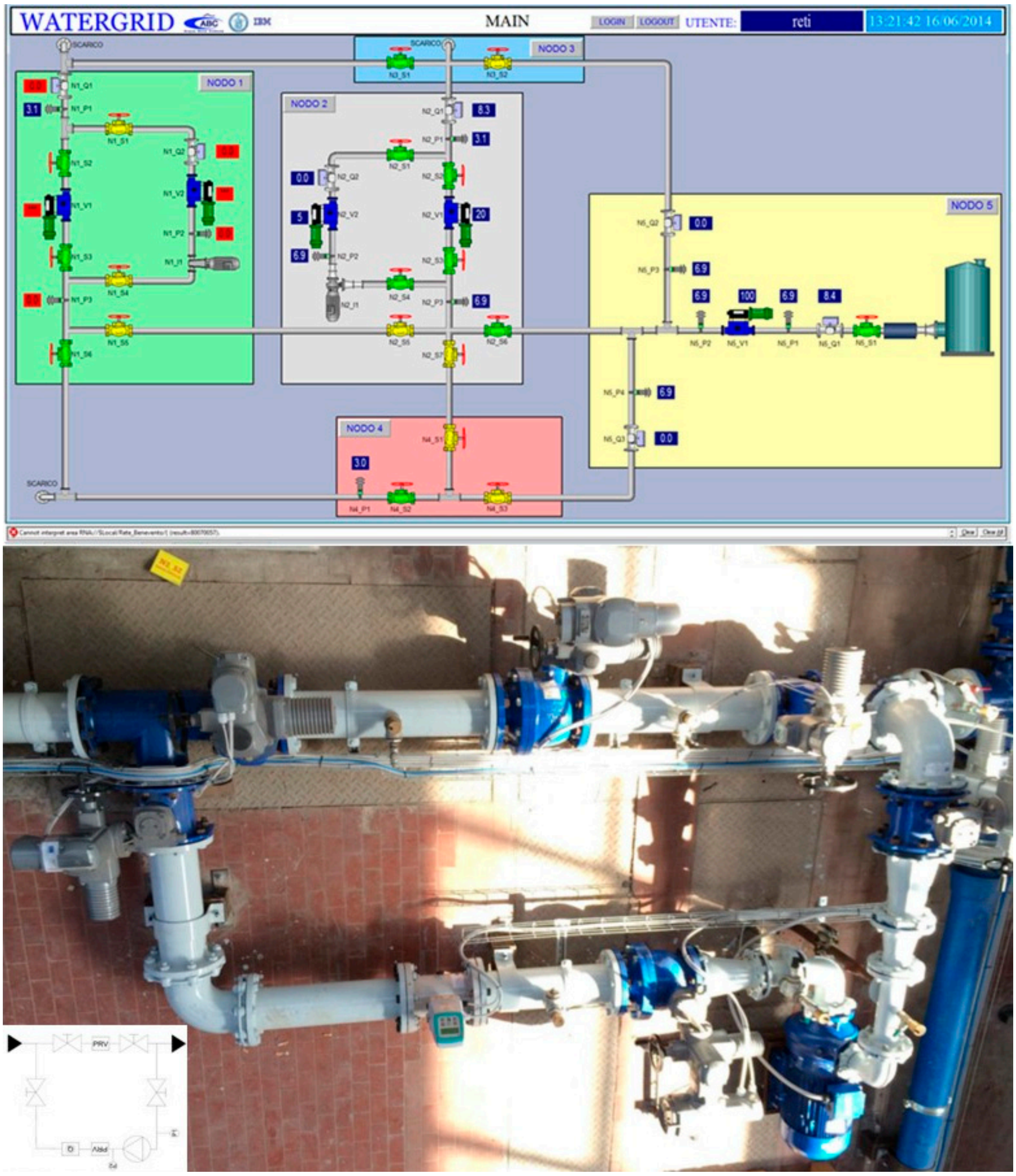

Figure 6. Test bench - water grid of the Department of Civil Construction and Environmental Engineering of the University of Naples Federico II

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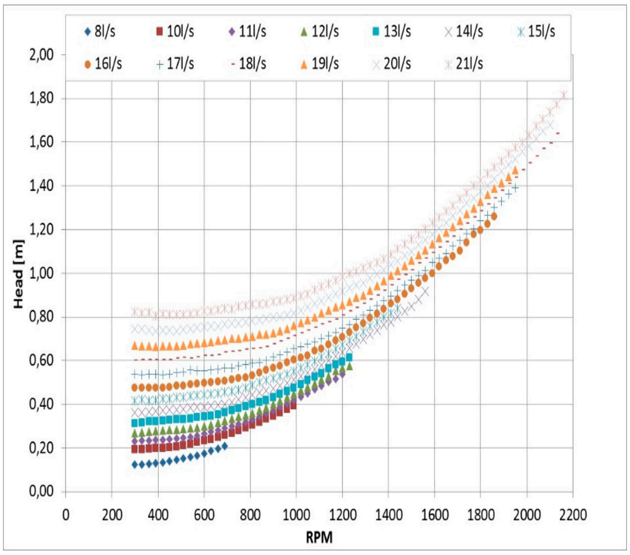

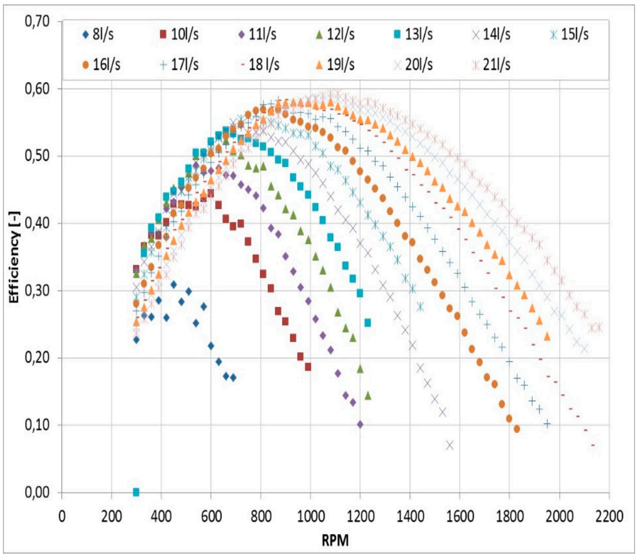

Figure 7. Experimental results

In figure 8, the whole validation of the simulation model is presented: it shows that the model reproduces the experimental data well with very small differences between the experimental and the model results for all the running conditions that were analyzed.

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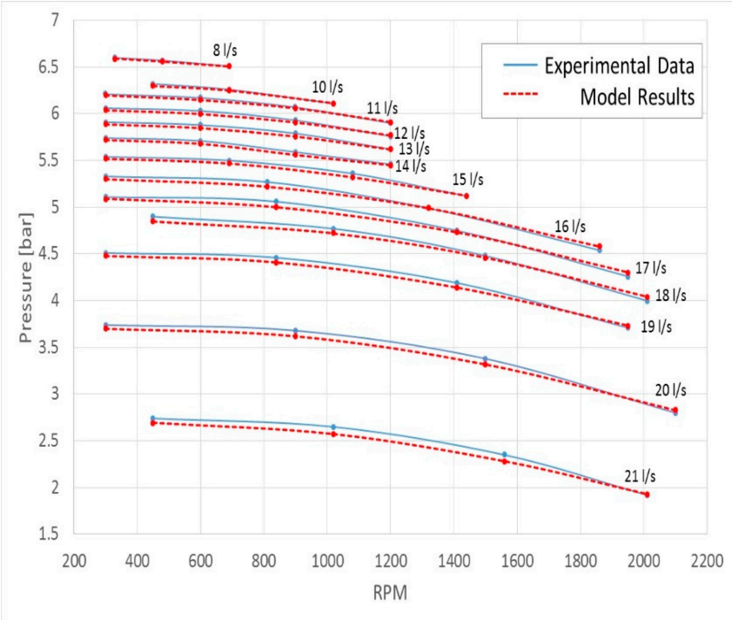

Figure 8. Model Validation

5. Model Results

After the validation phase in pump and turbine mode, simulation models have been used to predict the efficiency curves of the three analyzed pumps. Then, all the simulations have been performed to obtain the data necessary to evaluate the inverse characteristics. In the following, specific head, specific capacity, specific power and efficiency are defined as:

$$\psi = \frac{gH}{n^2 D^2} \quad (9)$$ Specific head

$$\varphi = \frac{Q}{nD^3} \quad (10)$$ Specific capacity

$$\pi = \frac{P}{\rho n^3 D^5} \quad (11)$$ Specific power

$$\eta = \frac{P}{\rho Q H} \quad (12)$$ Efficiency

where H [m], Q [m^3/s], and P [W] are the head, flow rate and power, respectively. The rotational speed is n [RPS] and D [m] is the impeller diameter. In the reverse mode simulation, the boundary conditions in pump and PAT mode were the same (declared data in pump mode). The boundary conditions in reverse mode are summarized in table 2.

Table 2. Boundary conditions

Boundary Conditions	Pump 1	Pump 2	Pump 3
Outlet pressure	1.9 bar	1.9 bar	1.9 bar
Inlet Volumetric Flow	90 ÷ 210 m^3/h	30 ÷ 85 m^3/h	48 ÷ 90 m^3/h
T_{in}	293.15 °K	293.15 °K	293.15 °K
P_{sat}	2886 Pa	2886 Pa	2886 Pa

The specific head, the specific power and the efficiency have been evaluated for both pump and turbine mode and for all the studied pumps. These are plotted versus specific capacity in figure 9.

It is clear that at high capacity the specific head in reverse mode is always higher than in direct mode. In reverse mode, pumps have a larger power range than in direct mode. The trends are quite different and the curves arising at low flow rate and at higher capacities have a higher power value than in direct mode. In pump mode, the efficiency has a typical "bell shape" while in PAT mode its profile resembles that of a Francis turbine: it increases at low flow rates and reaches a maximum value at high flow rates.

Moreover, the pump with the low specific speed works with low flow rates but high heads in direct mode. In reverse mode at high flow rates, the head is higher than the direct-mode value. The pump with the high specific speed works at high flow rates but low heads. In reverse mode, working with the same flow rates, the maximum head is also higher than the direct-mode value.

For all pumps, the maximum value is lower than the direct-mode value. For low – specific speed pumps, this maximum value is approximately equal to the direct - mode value, while for high – specific speed pumps, it is lower.

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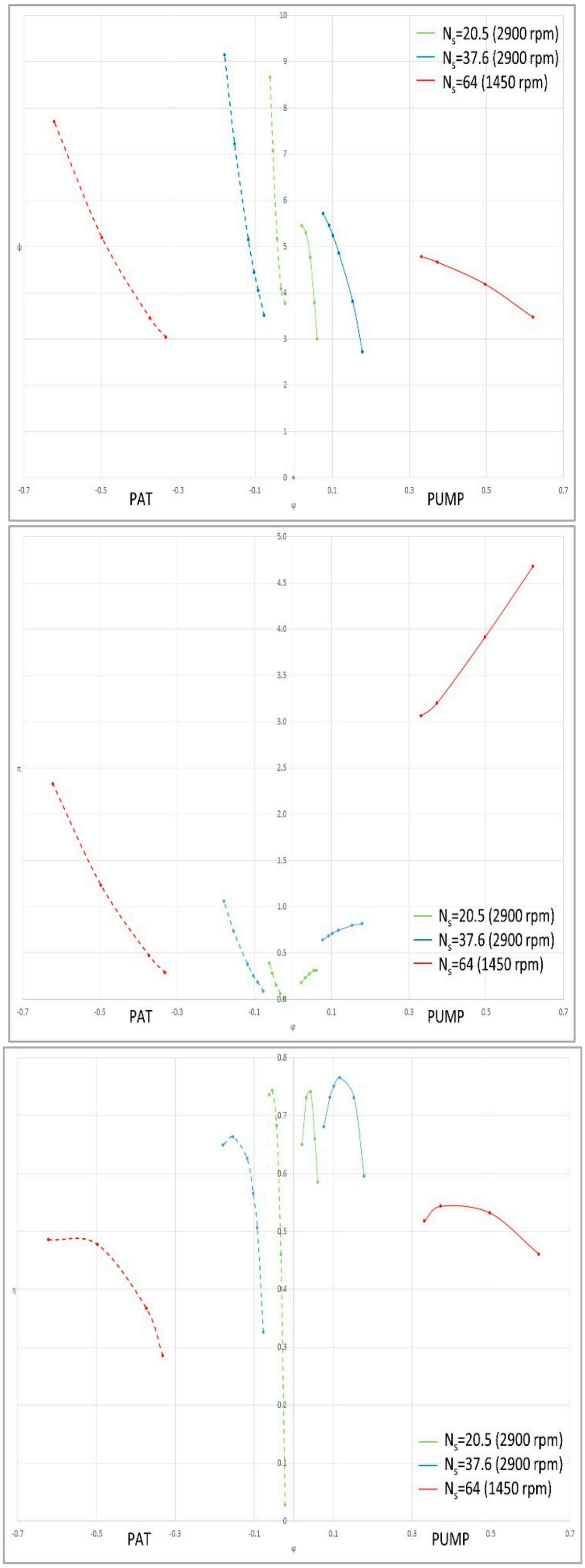

Figure 9. Specific head, specific power and efficiency

In conclusion, in table 2.5, the BEP values are summarized:

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Table 3. Boundary conditions

Boundary Conditions	Pump 1		Pump 2		Pump 3	
	Direct mode	**Reverse mode**	**Direct mode**	**Reverse mode**	**Direct mode**	**Reverse mode**
Head [m]	39	61	45.4	67	3.9	4.6
Capacity [m^3/s]	0.041	0.05	0.017	0.021	0.015	0.022
Power [kW]	20.5	19.98	10.01	10.24	1.05	0.75
Efficiency	0.787	0.663	0.743	0.741	0.543	0.487

6. Comparison of Prediction Methods

After the evaluation of the inverse characteristics, the results of the proposed methodology have been compared to the prediction methods available in literature. To this end, all the previously discussed methods have been applied to the three analyzed pumps. Some methods predict only the head and flow rate, while others also predict power and efficiency. In table 2.6, all the results are shown for pump 1.

Table 4. Comparison for pump 1

Methods	H [m]	Q [m^3/s]	P [kW]	η
Model results	**61.42**	**0.05**	**19.98**	**0.663**
Stepanoff	49.55	0.0463	22.5	0.787
Alatorre-Frenk	42.79	0.091	28.92	0.757
Sharma	51.99	0.0498	20.5	0.807
Schmiedl	43.94	0.0514	16.82	0.759
Grover	78.59	0.0657	36.92	0.729
Hergt	41.90	0.0508	-	-
Childs	49.55	0.0522	19.96	0.787
D&N	58.56	0.0411	18.17	0.769

The flow rates calculated using the methods of Sharma and of Hergt and Schmiedl are very close to those of the proposed CFD methodology. Grover's and Alatorre-Frenk's methods overestimate this value while the other methods underestimated the value by a margin of 10-15%. All methods underestimate the head value except for Grover's. These values diverge with errors of up to 30%.
Evaluating the percentage deviation as:

$$percentage\ deviation = \frac{predicted\ value - cfd\ value}{cfd\ value} \cdot 100 \qquad (13)$$

In figure 10 the deviations between the predictions of these methods and the simulated data

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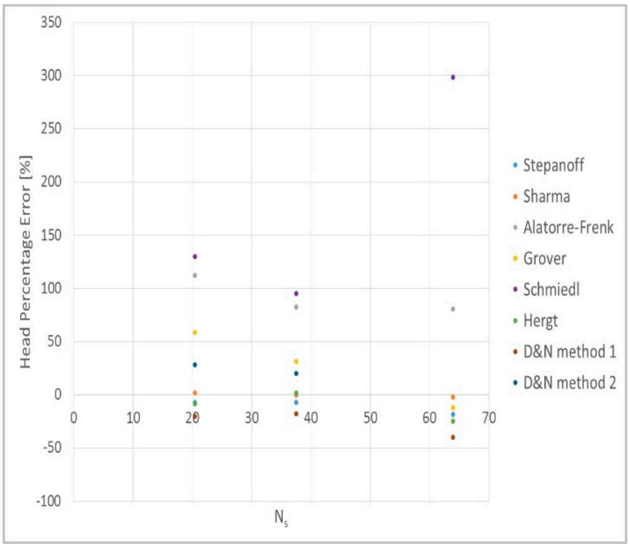
are plotted vs. the pump specific speeds for all three pumps.



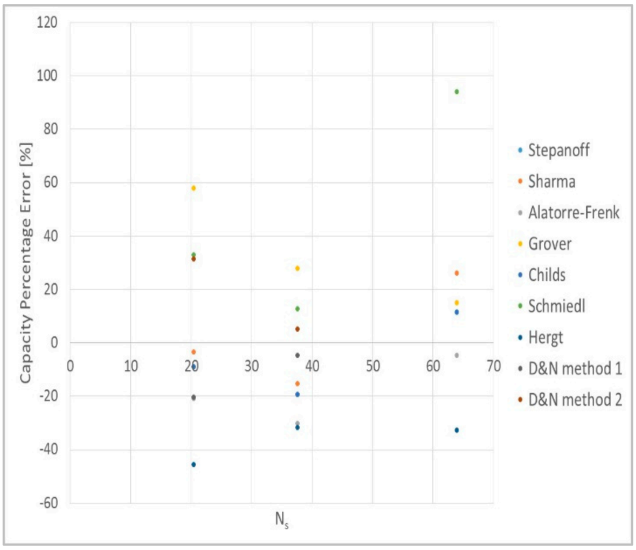


Figure 10. Prediction methods comparison

It easy to observe that in some cases the deviation is very high.
To evaluate the Derakhshan and Nourbakhsh efficiency [19], the CFD model results were used as shown in figure 11. In this figure, the relative velocity magnitude [m/s] is shown and the angle between the relative and absolute velocity is highlighted in red, at BEP conditions and in pump mode.

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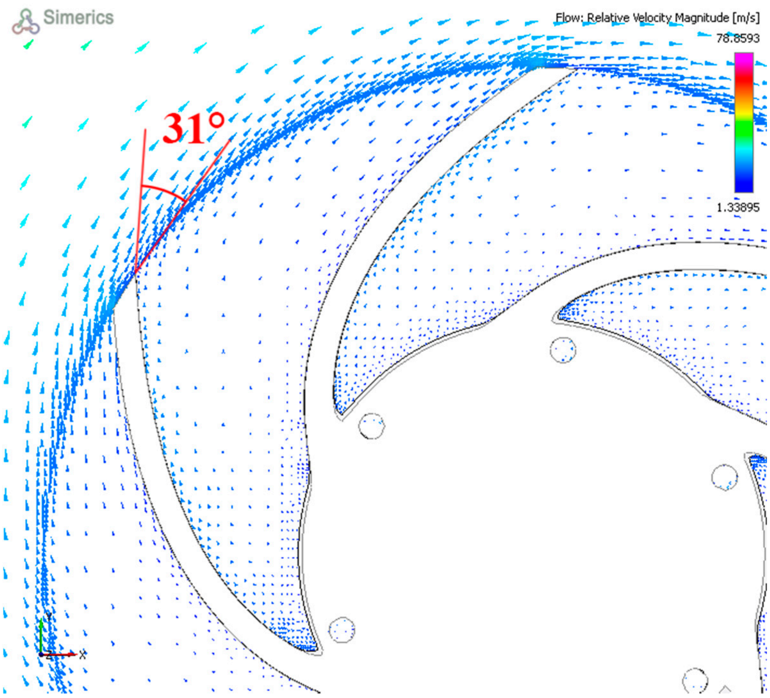

Figure 11. Relative velocity magnitude

7. Conclusions

In this paper a methodology to predict the inverse characteristic of a centrifugal pump has been presented. This methodology is based on the results of a three-dimensional simulation model build with a commercial CFD code. Three industrial pumps have been analyzed, with different specific speeds. First, the simulation models have been validated with data supplied by the pump manufacturers. Then the results of an experimental campaign have been used to validate a model simulating the pump working in reverse conditions.

Starting from the CFD model results, the specific head, capacity, power and efficiency have been evaluated and the best efficient point of all the analyzed pumps was found. Furthermore, several prediction methods have been applied to the tested pumps and their predicted values were compared with those of the proposed methodology. Some methods (e.g., Childs' method) are not in accord while others (e.g., Stepanoff's method) show small relative differences.

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