Experimental Study of a Pump as Turbine to Drive Submersible Pumps in Artificial Lift Applications

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Abstract. *The method of artificial lift through a conventional electrical submersible pump (ESP) is well known in petroleum production. The alternative hydraulic submersible pump technology (HSP) consists of a submersible pump coupled to the same shaft of a hydraulic turbine, which is driven by high pressure water from a surface booster pump. The advantages of the HSP system over the conventional ESP for offshore applications have been discussed, mainly due to its characteristics of wider operating range, higher rotation speeds and higher mean time to failure (MTTF). In view of this, the present study is aimed at the performance evaluation of a centrifugal pump working as turbine (PAT), i.e., operating in reverse mode, to drive the pump. The impeller of a conventional submersible pump operating as turbine is investigated. A specific apparatus was built for measurement of performance parameters such as torque, rotation speed, flow rate and pressure drop through the turbine, collected according to a standard procedure. Performance curves such as head, brake power and efficiency are presented and a comparison with the performance of the same impeller operating as pump is discussed. A method available in the literature to derive the turbine performance parameters from the correspondent pump performance is used for comparison with data.*

Keywords*: Petroleum Engineering, Artificial Lift, Hydraulic Submersible Pump, Pump as Turbine, Centrifugal Pump.*

1. Introduction

Artificial lift method called Hydraulic Submersible Pump (HSP) is an alternative to the widely used Electrical Submersible Pump (ESP) in offshore oil production. The HSP method consists of a submerged centrifugal pump coupled to the same shaft of a hydraulic turbine, which is driven by a booster pump located at surface. The hydraulic turbine provides the mechanical energy on the pump shaft for lifting oil.

Laboratory studies and field applications demonstrate the attractiveness of the method. In situations of oil flowing with high void fractions of gas and high viscosity, the HSP exhibited a wide operating range without gas locking condition and increased mean time to failure (Harden and Downie, 2001; Mali *et al*., 2010). Usually, the ESPs require gas handlers and/or gas separation systems to increase the gas tolerance at pump inlet (Bagci, 2010). Such devices require higher capacity to supply electrical energy, space in production string for installation and reduce the power of the set. In turn, the HSP presents smaller restrictions of space inside the casing due the absence of limitations found in electrical motors and cables, usually responsible for failures in marine environment.

References of the so-called Hydraulic Power Recovery Turbines (HPRT) are often found in the literature. Such turbines are used for energy recovery in valves or other throttling devices (Gülich, 2008). There are also several studies on centrifugal pumps operating in reverse mode as turbines (Alatorre-Frenk, 1994; Derakhshan and Nourbakhsh, 2008; Fernandez *et al*., 2004; Singh and Nestmann, 2010), here denominated Pumps as Turbines (PAT). The PATs present efficiencies comparable to operation as pumps, ease of maintenance and inspection due to their characteristics of simplified construction and geometry, and market availability. Some applications are in small hydroelectric power plants, transportation systems of water, and in processing plants as an alternative to throttling devices.

Turbines for driving submersible pumps must obviously meet the requirement of space limitations of the wells, which suggests use of turbines similar to ESPs in diameter. In view of this, this paper presents an experimental study of a PAT consisting of a conventional ESP impeller operating in reverse mode. Thus, an experimental apparatus was built, allowing the measurement of torque and power shaft, water flow rate, differential pressure and rotating speed, with their maximum uncertainties involved. The study focuses only the PAT, with a modified Prony brake coupled to its shaft used as load, instead of a pump.

Performance curves of head, power and efficiency are presented. The model proposed by Alatorre-Frenk, mostly based on volute pumps operating in reverse mode, was used to characterize the PAT under investigation from the performance curves of the same impeller operating in the pump mode.

2. Experimental setup

The prototype built in the laboratory consists of a HSP driven by a PAT, Fig. 1. For the present study, the PAT was decoupled from the pump.

The PAT has only one ESP impeller, illustrated in Fig. 2. It is single suction, closed and radial type and presents specific speed of 0.34 at BEP (best efficiency point) in pump mode.

An outlet diffuser was not used in the PAT but its cylindrical casing was modified. Two tangential inlets and a single outlet were designed for the flow of the power fluid. A booster pump was used to inject high pressure water in the PAT.

The features of the instrumentation used are summarized in Tab. 1. Measurement of the rotor rotation speed was performed by means of a tachometer. To measure the torque a modified Prony's brake system was used, consisting of a hydraulic brake set, an aluminum bar and a load cell, Fig. 3. The brake torque was calculated as the product of the force measured at the load cell and the normal distance of its line of action to the shaft center. All variables measured were integrated into a data acquisition system, except the shaft speed that was manually inserted through the user interface of this system.

The tests were performed by varying the braking torques along specific ranges of flow rates. The braking torques varied between minimum and maximum loads applied to the PAT shaft. Minimum load corresponds to the sufficient load to obtain a minimum measurable value of torque and the maximum load corresponds to the maximum load that can be applied to the PAT shaft without blocking it. Flow rate ranges were defined by the variation of the booster pump speed, through its variable frequency drive. For every braking torque applied, the respective measured variables were collected and stored through the data acquisition system. The uncertainty analysis of measurements was performed using the procedure presented by Holman (1994). Considering the uncertainties provided by the manufacturers of the several instruments summarized in Tab. 1, the uncertainties for the efficiency and coefficients of power, head and flow rate defined in Eq. (1) are estimated as 3.3, 1.8, 3.2 and 0.6%, respectively.

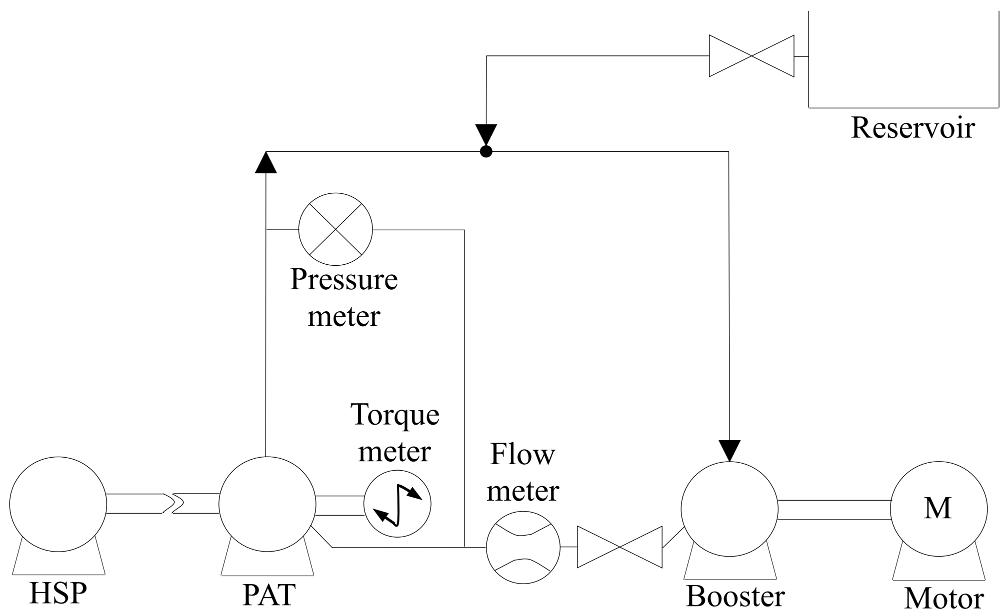

Figure 1. Schematics of the experimental test facility.

Table 1. Features of the instrumentation used in the experimental test facility.

Variable	Device	Measurement Principle	Range	Accuracy
Speed	Speed sensor	Optical reflection	2.5 – 100000 rpm	± 0.05% of reading + 1 digit
Force	Load cell	Wheatstone bridge	0 – 5 kg	± 0.02% of full scale
Pressure	Differential pressure transducer	Capacitive	0 – 20 bar	± 0.125% of full scale
Flow rate	Coriolis flow meter	Coriolis	0 – 400 L/min	± 0.30% of flow rate
Density	Coriolis density meter	Coriolis	800 – 1200 kg/m^3	± 2 kg/m^3

Figure 2. PAT impeller under investigation.

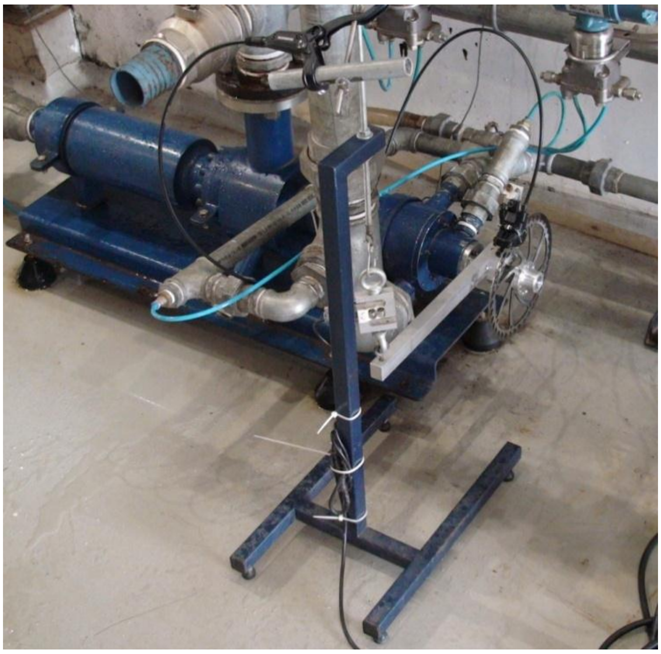

Figure 3. Prony brake system for torque measurement of the tested PAT.

3. Experimental results

The variation of rotation speed of the booster pump is a convenient way to vary the flow of water injected into the PAT. Higher is the booster speed higher is the flow rate and speed in the PAT. In Fig. 4 are presented the speed and flow rate in the PAT as functions of the booster pump speed, without brake load applied. There is a minimum speed observed in the booster pump, 1200 rpm, that represents a minimum flow rate in the PAT needed to start its operation, around 122.4 L/min. From 1500 rpm are observed increments of speed in the PAT higher than the booster pump. For 2900 rpm in the booster pump, the PAT speed was 3703 rpm for a flow rate of 269.1 L/min.

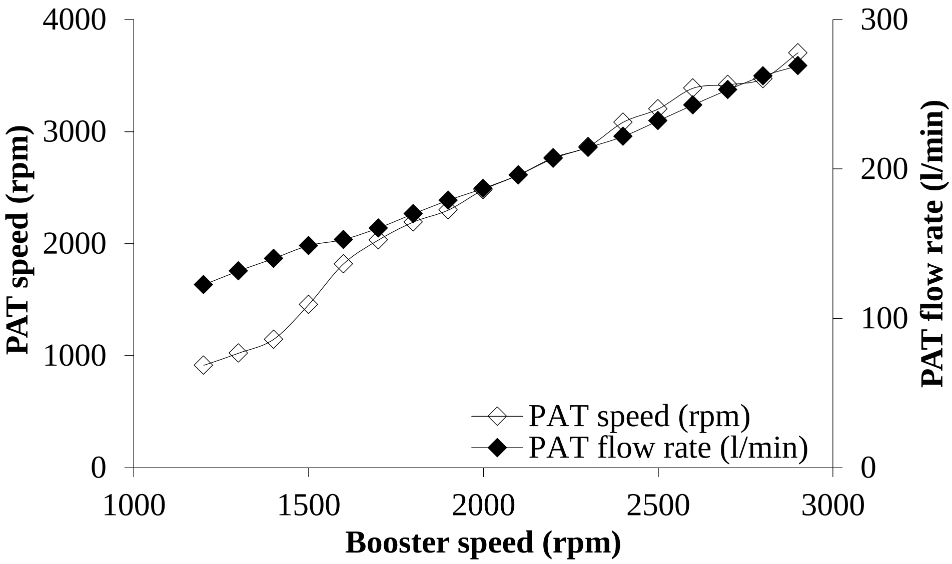

Figure 4. Speed variations and flow rate through the tested PAT.

Through the proposed experimental procedure are presented in Fig. 5 the dimensionless performance curves of PAT. The dimensionless coefficients of flow (C_q), head (C_h) and power (C_p) are respectively defined as:

$$C_q = \frac{Q}{\omega D^3} \quad ; \quad C_h = \frac{gH}{\omega^2 D^2} \quad ; \quad C_p = \frac{P}{\rho \omega^3 D^5} \tag{1}$$

where Q (m^3/s), H (m) and $P = T\omega$ (W) are the flow rate, head and shaft power, respectively; T (N.m) is the shaft torque, $\omega=2\pi N/60$ (rad/s) is the rotation speed, N is the frequency of rotation (rpm), D (m) is the diameter of impeller, ρ (kg/m^3) is the density of the fluid and g (m/s^2) is the gravity acceleration.

The efficiencies of the PAT in turbine mode (η_T) and pump mode (η_P) are respectively defined as:

$$\eta_T = \frac{P}{\rho QgH} = \frac{C_p}{C_q C_h} \quad ; \quad \eta_P = \frac{\rho QgH}{P} = \frac{C_q C_h}{C_p} \tag{2}$$

The experimental points were collected at different speed. At highlighted points were observed the minimum or maximum speed, 830 rpm and 3942 rpm, respectively. The points corresponding to higher rotational speeds are located in the lower regions of the curves around 0.01 of flow coefficient.

As characteristic of turbine performance curves, the head increases with increasing flow rate, Fig. 5(a). Similarly to that observed by Fernandez *et al*. (2004) in his study, the curves do not pass through the origin of the axes, indicating that there is, at low flows, a recirculation region with head dissipation and null shaft torque, Fig. 5(b).

The maximum efficiency found, Fig. 5(c), was approximately 20% for a flow coefficient of 0.016. The low efficiency must be analyzed considering the hydraulic and mechanical losses of the system used. The high hydraulic power provided by booster in comparison to the shaft power obtained at PAT indicates possible significant losses by shock of the fluid with the blades at inlet of impeller (incidence not tangential to the blades), recirculation inside the impeller and leaks through gaps between the impeller and casing. In addition, there are mechanical losses by friction in bearings and sealing elements. The absence of a diffuser at the outlet of PAT may contribute to significant losses of kinetic energy, resulting in a lower pressure drop over the turbine and reduced efficiency. All these aspects should be considered in an optimized design of PAT with an ESP impeller.

It is also observed in Fig. 5(c) the dispersion of points in the region of improved efficiency, possibly due to the difficulty of adjusting brake system for maximum loads without locking the PAT shaft. Possibly significant uncertainties related to the procedure and method for torque measurement (Prony's brake) were minimized by taking three readings of the PAT rotation speed.

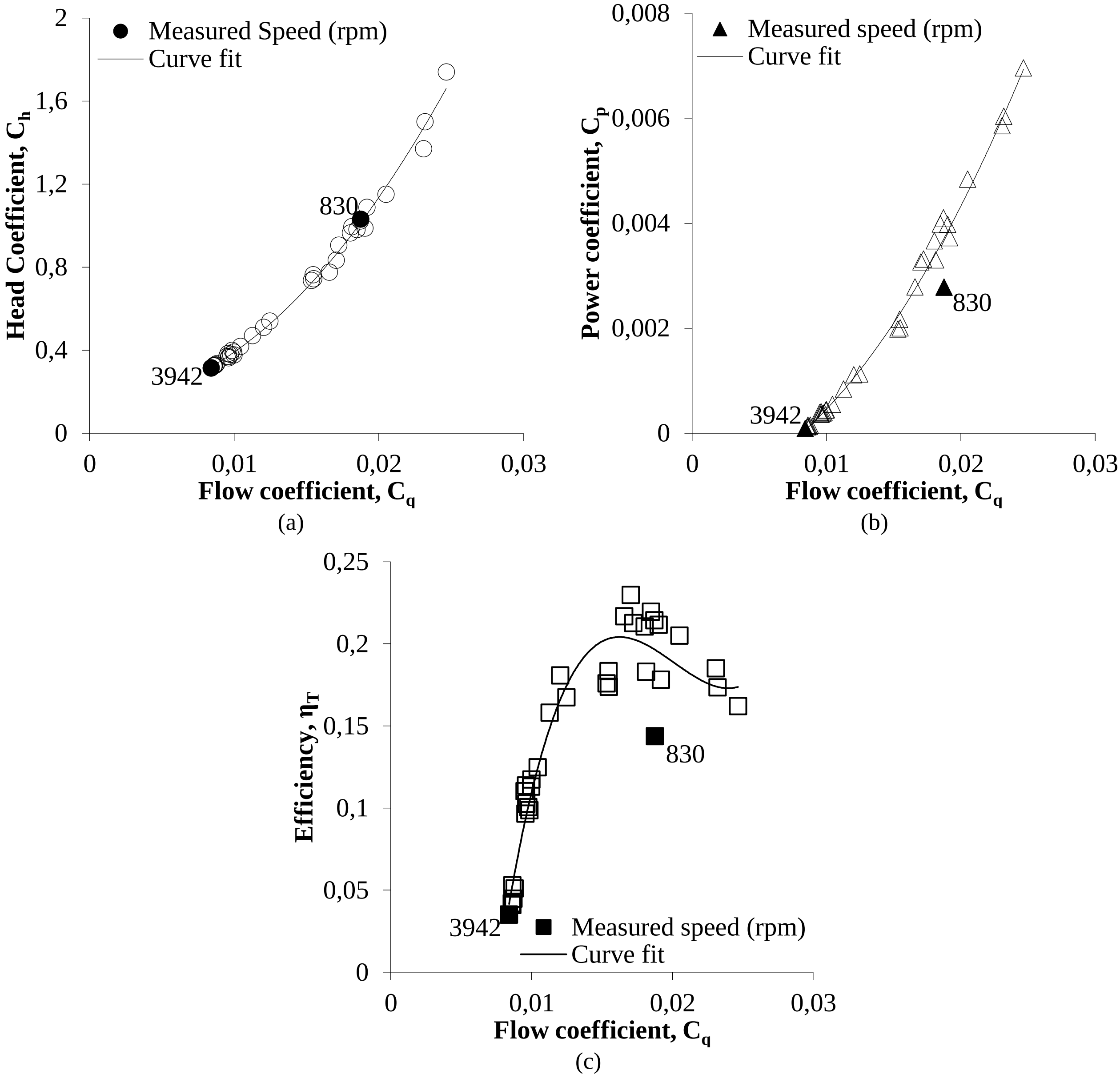

Figure 5. Dimensionless performance curves. (a) Head, (b) Power and (c) Efficiency.

3.1 Estimated performance curves of PAT

From the performance curves of the PAT operating in pump mode, its predicted performance in turbine mode can be calculated using the correlation proposed by Alatorre-Frenk (1994), which allows estimation of the performance parameters at BEP and also along the entire flow rate in turbine mode, based on performance parameters and geometry in pump mode.

In Eqs. (3-15) below, the subscripts $_{P}$, $_{T}$, $_{TE}$ refer to the pump mode, turbine mode and estimated turbine mode, respectively. The symbol ^ refers to the BEP and the definition of the dimensionless specific speed (Ω) is:

$$\Omega_P = \frac{\hat{\omega}_P \sqrt{\hat{Q}_P}}{\left(g\hat{H}_P\right)^{0.75}} \tag{3}$$

At BEP, the conversion factors for flow, head and efficiency between the two modes of operation are, respectively, given by:

$$\frac{\hat{Q}_{TE}}{\hat{Q}_P} = 1.21 \hat{\eta}_P^{-0.6} \tag{4}$$

$$\frac{\hat{H}_{TE}}{\hat{H}_P} = 1.21 \hat{\eta}_P^{-0.8} \left[1 + \left(0.6 + ln\,\Omega_P\right)^2\right]^{0.3} \tag{5}$$

$$\frac{\hat{\eta}_{TE}}{\hat{\eta}_P} = 0.95 \hat{\eta}_P^{-0.3} \left[1 + \left(0.5 + ln\,\Omega_P\right)^2\right]^{-0.25} \tag{6}$$

At all flow rates other than the BEP, the estimated characteristic curves of head (H_T) and power (P_T) are expressed by:

$$H_T \approx A_H Q_T^{\,2} + B_H Q_T \omega_T + C_H \omega_T^{\,2} \tag{7}$$

$$P_T \approx A_M Q_T^{\,2} \omega_T + B_M Q_T \omega_T^{\,2} \tag{8}$$

where the coefficients of head A_H (m^{-5}.s^2), B_H (m^{-2}.s^2), C_H (m.s^2) are expressed by:

$$A_H \approx \frac{E_{2T}}{2} \frac{\hat{H}_T}{\hat{Q}_T^{\,2}} \tag{9}$$

$$B_H \approx \left(E_T - E_{2T}\right) \frac{\hat{H}_T}{\hat{Q}_T \hat{\omega}_T} \tag{10}$$

$$C_H \approx (1 - E_T + \frac{E_{2T}}{2}) \frac{\hat{H}_T}{\hat{\omega}_T^{\,2}} \tag{11}$$

The coefficients A_M (kg.m^{-4}), B_M (kg.m^{-1}) are:

$$A_M \approx E_T \frac{\hat{T}_T}{\hat{Q}_T^{\,2}} \tag{12}$$

$$B_M \approx \left(1 - E_T\right) \frac{\hat{T}_T}{\hat{Q}_T \hat{\omega}_T} \tag{13}$$

and the dimensionless parameters, E_T and E_{2T} are expressed by:

$$E_T = 0.68 + 1.2\sqrt{\Omega_P} \tag{14}$$

$$E_{2T} = 0.76 + 2.1\sqrt{\Omega_P} \tag{15}$$

The tests of the PAT operating in the pump mode were performed by a pump manufacturer according to ISO 9906 and the corresponding test facilities are shown in Fig. 6.

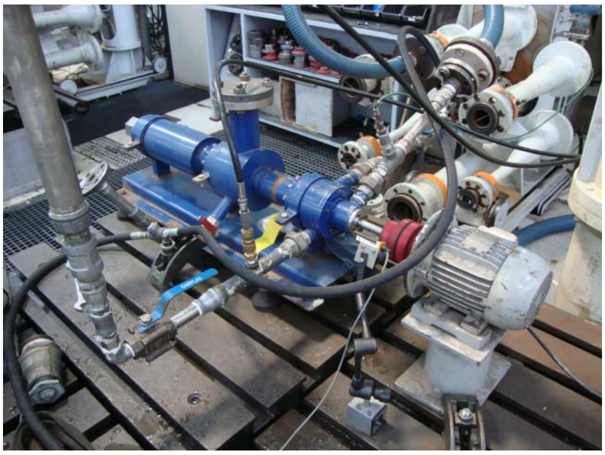

Figure 6. The PAT operating in the pump mode at test facilities of a pump manufacturer.

Figure 7 shows the performance curves of the PAT operating in the pump mode. The low values of efficiency and flow coefficient range indicate somewhat high hydraulic and mechanical losses, also detected in the turbine mode. In view of this, to determine the BEP on the estimated curves of the PAT, the flow coefficient range in the calculations was extrapolated until the maximum flow coefficient experimentally obtained in the turbine mode, approximately 0.025. In Fig. 8 are presented the performance curves of the PAT estimated through Alatorre-Frenks's correlation. At BEP, the predicted efficiency of the PAT is higher than the corresponding pump efficiency, i.e., η_T = 23.85% at C_q = 0.012 and η_P = 15.43% at C_q = 0.003.

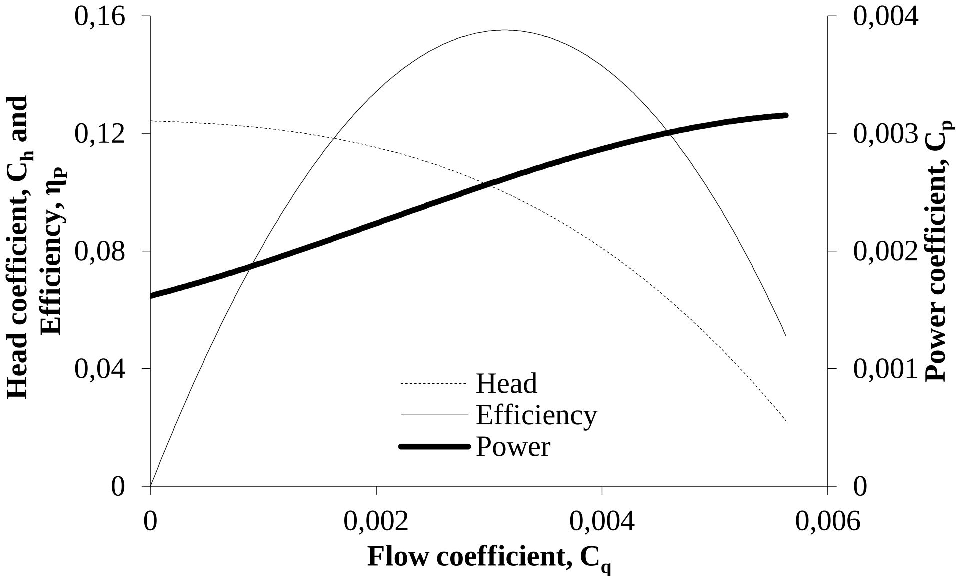

Figure 7. Performance curves of the tested PAT operating in pump mode.

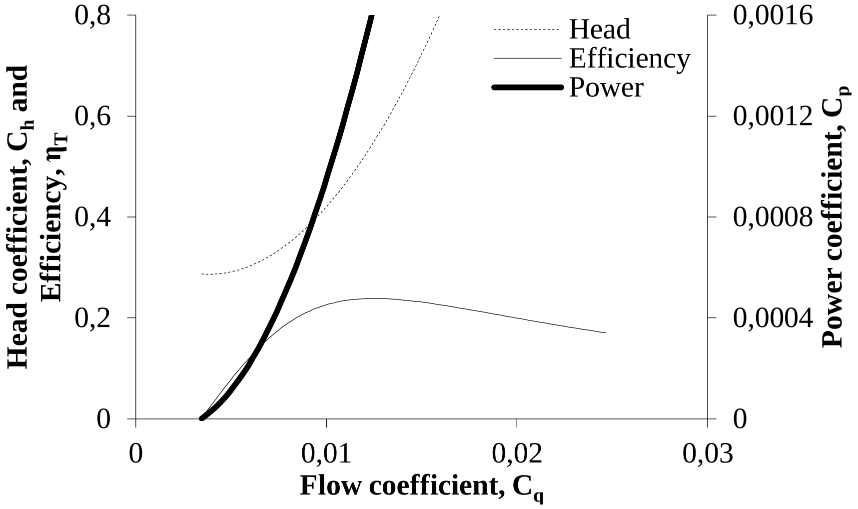

Figure 8. Estimated performance curves of PAT.

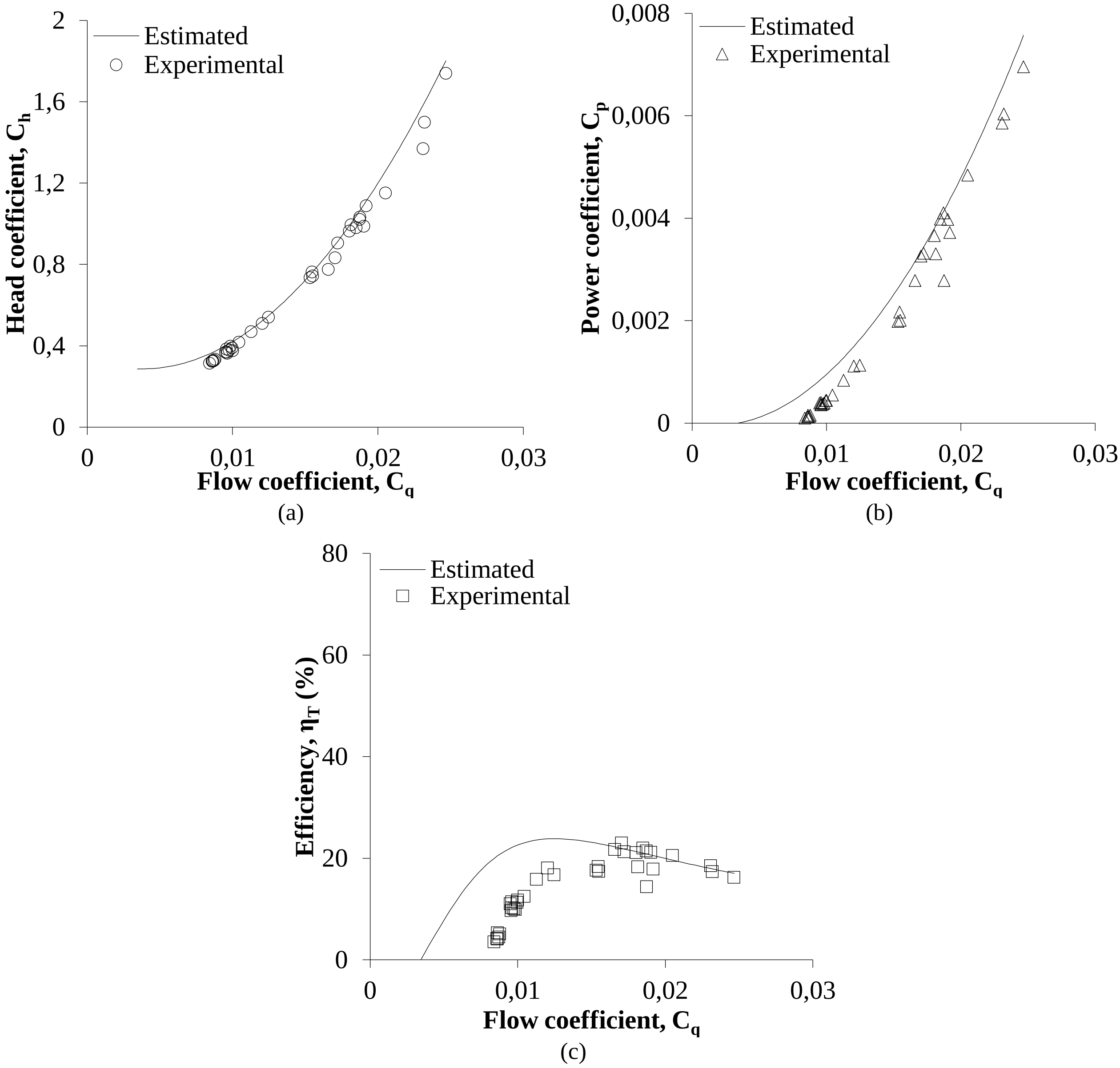

Figure 9. Estimated and experimental performance curves of PAT. (a) Head, (b) Power and (c) Efficiency.

The estimated and experimental performance curves of PAT are presented in Fig. 9. It can be observed that the beginning of the operation was estimated in the theoretical curves for a flow coefficient lower than the experimental curves, 0.003 and 0.008, respectively. Deviations at BEP, for efficiency and for coefficients of power, head and flow rate are, -3.8, 48.9, 33.4 e 26.2%, respectively. The deviation at the power coefficient is mainly related to the different methods used for power measurements: in the pump mode, wattmeters coupled at the electrical motor that drives the pump were used, whereas in the turbine mode the combination of the Prony brake and tachometer was used.

Due to modifications in the casing, the deviations are related to higher variations of hydraulic and mechanical losses, between the modes of operation, when compared to the same variations in a conventional centrifugal pump. Another relevant issue is that such modifications may have influenced the relationships between the specific speed and efficiency that are parameters used in the conversion factors between the pump mode and turbine mode. Also, usually the method that relates the BEP conditions between these two modes of operation, including the correlation of Alatorre-Frank (1994), have approximately 20% of deviation from the experimental data (Derakhshan and Nourbakhsh, 2008; Gülich, 2008; Singh and Nestmann, 2010).

A better agreement can be observed between the efficiency curves for flow rates above the BEP, from a flow coefficient of 0.017.The method represents satisfactorily the type of PAT under investigation, considering the agreement observed on the curves of head and power until the flow coefficient of 0.025 that could be reached in the pump mode experiments.

4. Conclusion

In this study were obtained the performance curves of a PAT that operates with an ESP impeller and uses a modified Prony brake system as load on the shaft. A specific measurement apparatus was built according to standard procedures, of performance parameters such as flow rate, head, rotation speed, and torque through the Prony’s brake. The brake system demonstrated to be capable of working at high rotational speeds in the PAT and withstand high temperatures. Despite of its simplicity, this method of torque measurement is related to specific uncertainties, as observed in the dispersion of experimental points in the region of BEP, which are difficult to quantify and evaluate for improved measurement purposes.

The experimental curves were compared with an available correlation for the prediction of performance curves of the PAT. The agreement between the head and power curves can be considered satisfactory until the flow coefficient of 0.025 that could be reached in the pump mode experiments. For the efficiency curves, a better agreement was observed for flow rates higher than the flow at BEP, from a flow coefficient of 0.017.

Deviations can be attributed to the modifications in the casing, specifically at fluid inlet/outlet and absence of an outlet diffuser. These probably resulted in relatively high hydraulic and mechanical losses, which may have influenced the relationships between the specific speed and efficiency that are parameters used in the conversion factors between the modes. Deviations in the shaft power curves can be also related to the different methods used for power measurement: wattmeter for the pump mode test and Prony’s brake and tachometer for turbine mode test.

However, the general conclusion that a PAT exhibits nearly the same maximum efficiency, than for the operation as pump was confirmed. The efficiencies at BEP are about 20% and 15%, respectively, for turbine mode and pump mode.

Further studies are needed on ESPs that operate in reverse mode, with the objective of validating a specific method for predicting performance curves and for improving the overall efficiency.

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