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Energy Procedia 148 (2018) 170–177

Energy Procedia

www.elsevier.com/locate/procedia

73rd Conference of the Italian Thermal Machines Engineering Association, ATI2018, 12-14 September 2018, Pisa, Italy

# Effects of viscosity on the performance of Hydraulic Power Recovery Turbines (HPRTs) by the means of Computational Fluid Dynamics (CFD) simulations

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**Abstract**

Centrifugal pumps are used for increasing the energy content of a liquid: this technology is used in chemical processes with liquids having specific chemical and physical characteristics. Most of the processes are closed-loop, meaning that the liquid is reused after a proper physical or chemical washing treatment is performed. Therefore, the pressure of the liquid has to be decreased by means of a lamination valve or a Hydraulic Power Recovery Turbine (HPRT) with the advantage of recovering energy. HPRTs are generally tested in both pump and turbine modes using water as working fluid. The technical report ISO/TR 17766 indicates the procedure to evaluate the performance of centrifugal pumps handling viscous liquids by supplying correction factors with respect to water, but no indications are given in turbine mode. This work provides correction factors able to evaluate also the performance of HPRTs handling viscous fluids in turbine mode by varying the proposed formulae in the technical report. Computational Fluid Dynamics (CFD) simulations of two tested HPRTs are performed using, at first, water as working fluid for validating the experimental results and, subsequently, the SELEXOL® solvent. Results show that the original correction factors are still valid for the HPRT B that has a parameter B, which is the main one to be involved in the evaluation of the correction factors, lower than 1. A better accuracy, instead, is achieved by modifying the correction factors of the HPRT A, having a value of B higher than 1.



*Keywords:* Chemical process; Oil & Gas; Hydraulic Power Recovery Turbine; Viscosity; CFD simulations

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1876-6102 
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Selection and peer-review under responsibility of the scientific committee of the 73rd Conference of the Italian Thermal Machines Engineering Association (ATI 2018).
10.1016/j.egypro.2018.08.046

## 1. Introduction

Energy recovery interventions are taking the field in different engineering fields to improve the performance of determined systems on both environmental and economic point of views [1]. In the small-scale hydropower sector there are several technologies used for achieving this goal and, among the most widespread, Pump-as-Turbine (PaT) [2] is currently the most used for several aspects, like cheaper cost and larger availability in the market due to its mass production in different sizes [3]. This technology is used for producing electric energy in remote and rural zones that are close to water sources (e.g. rivers and irrigation systems) [4], for energy recovery purposes in Water Distribution Networks (WDNs) [5, 6] and chemical plants, like oil refineries. In this latter case, PaT is named Hydraulic Power Recovery Turbine (HPRT) and its task is to supply the energy recovered from the process to a feed pump, operating on the same process, using a direct mechanical coupling. The main processes in which a HPRT is used are reverse osmosis, petroleum cracking and $H_2S$ removal ones [7, 8]. In the last one, liquids with a high viscosity that are able to maintain a liquid phase at high pressures and/or high temperatures are used. One of the main issues of HPRTs regards the forecast of the performance in turbine mode: for this reason, different prediction models have been developed taking into account data of centrifugal pumps in both direct and reverse modes [9-11]. However, these models supply data with water as working fluid and not with other liquids. Li [12-14] developed correction models for predicting PaTs' performance in both pump and turbine modes. To achieve this goal, Computational Fluid Dynamics (CFD) simulations on a PaT are carried out considering five different viscosities. Results showed that, in pump mode, the higher the viscosity, the lower both characteristic and efficiency curves, while, in turbine mode, the higher the viscosity, the higher the characteristic curve and the lower the hydraulic efficiency. In the Oil & Gas sector, the technical report ISO/TR 17766 [15] allows to evaluate the performance of centrifugal pumps handling viscous liquids by supplying correction factors with respect to water, but no indications are given in turbine mode. Starting from the results obtained by Li and the developed correlations, the goal of this work is to provide correction factors able to evaluate also the performance of HPRTs in turbine mode handling viscous fluids by changing some parameters of the proposed formulas in the technical report ISO/TR 17766. To this purpose, CFD simulations of two tested HPRTs have been carried out with water as working fluid for validating the experimental results [5] and, subsequently, other simulations are run with the SELEXOL® solvent: this solvent is used in a $H_2S$ removal process of the raw Syngas produced in the gasification located in an Italian oil refinery. The paper is organized as follows: Section 2 explains the main formulas used in the technical report ISO/TR 17766, the CFD models and set-ups taking into account the properties of the SELEXOL® solvent. Section 3 deals with the results obtained by CFD simulations considering two HPRTs operating in both pump and turbine modes, showing the different behaviours recorded by the two hydraulic machines and the proposed correction factors. Finally, Section 4 reports the conclusion of the work.

## 2. Research and methods

### *2.1. Technical report ISO/TR 17766*

Centrifugal pumps are used in several chemical processes with the task of increasing the pressure energy content of a determined working fluid. The performance of these machines are generally evaluated with water even though they can operate with other liquids having higher viscosity: in this case, it is not straightforward to evaluate their operating performance. The technical report ISO/TR 17766 solves this problem by providing a procedure to calculate the operating performance of centrifugal pumps for fluids that are different from water. Firstly, the evaluation of the parameter B, as equation (1) shows, must be carried out. In this equation, the viscosity of the operating liquid ($V_{VIS}$), expressed in cSt, flow rate and head at the Best Efficiency Point (BEP) obtained with water ($Q_{BEP-W}$, $H_{BEP-W}$), expressed in $m^3/h$ and m, respectively, and the rotating speed (N), expressed in rpm, are involved.

$$B = 16.5 \cdot \left( {V_{VIS}}^{0.5} \cdot {H_{BEP-W}}^{0.0625} / {Q_{BEP-W}}^{0.375} \cdot N^{0.25} \right) \quad (1)$$

B can assume different values and, depending on them, the technical report suggests three solutions: if B≥40, the formula shouldn't be used due to the high uncertainty of the results; if 1<B<40, corrections on both characteristic and efficiency performance curves of the pump have to be introduced (equations 2 and 3); if B≤1, only modifications on the efficiency curve have to be performed (equation 4, neglecting both equations 2 and 3). The correction factors for the flow rate ($C_Q$) and head ($C_H$) are obtained by coupling equation (2) with equation (3) and equation (5) with equation (6), respectively.

$$1 < B < 40 \rightarrow \begin{cases} C_Q = (2.71)^{-0.165\cdot(logB)^{3.15}} & (2) \\ Q_{VIS} = C_Q \cdot Q_W & (3) \end{cases}$$

$$B \leq 1 \quad \rightarrow \quad Q_{BEP-H} = C_Q \quad (4)$$

$$C_H = 1 - \left[(1 - C_Q) \cdot (Q_W/Q_{BEP-W})^{0.75}\right] \quad (5)$$

$$H_{VIS} = C_H \cdot H_W \quad (6)$$

The correction for the efficiency, as anticipated, regards the cases of 1<B<40 and B≤1: for the first case, the correction factor of the efficiency $C_\eta$ is evaluated by means of equation (7), while, for the last one, it is calculated by means of equation (8). Both equations have to be coupled with equation (9) for evaluating the final efficiency value.

$$C_\eta = B^{-(0.0547\cdot B^{0.69})} \quad (7)$$

$$C_\eta = 1 - [(1 - \eta_{BEP-W}) \cdot (V_{VIS}/V_W)^{0.07}]/\eta_{BEP-W} \quad (8)$$

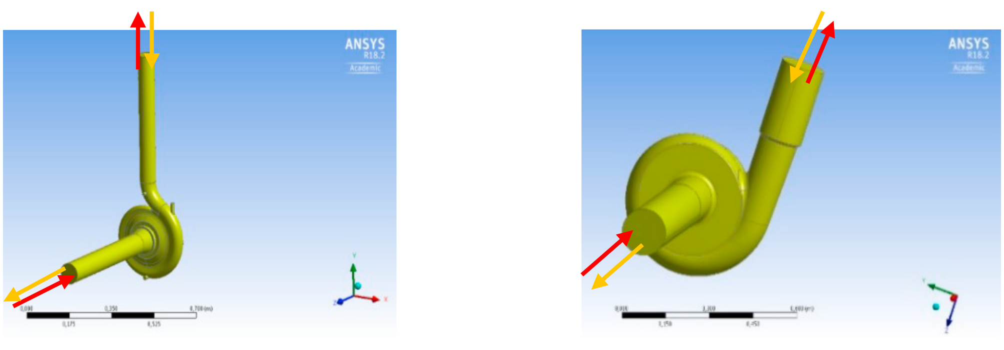
$$\eta_{VIS} = C_\eta \cdot \eta_W \quad (9)$$

### *2.2. CFD models, mesh and set-up*

CFD simulations are carried out with the ANSYS® CFX solver and used to validate the experimental results of two HPRTs tested with water [9]. Table 1 lists the main geometrical characteristics of the HPRTs, while Fig. 1a and Fig. 1b show the computational domain of the HPRT A and HPRT B, respectively.

*Table 1. Main HPRTs characteristics*

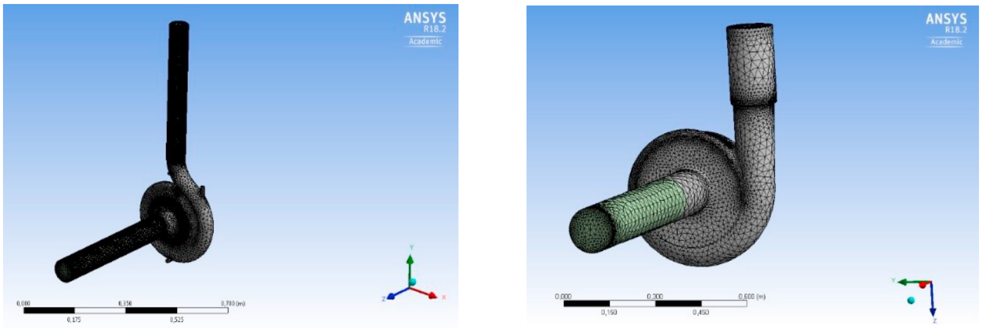
| HPRT | A | B |
|---|---|---|
| Type-Impeller | Centrifugal – Double disk | Centrifugal – Single disk |
| Number of blades | 7 | 10 |
| Flow rate [$m^3/h$] at BEP in pump mode | 50 | 432 |
| Head [m] at BEP in pump mode | 10 | 32 |
| Efficiency at BEP at BEP in pump mode | 0.76 | 0.66 |
| Mechanical power [kW] at BEP in pump mode | 1.79 | 57.08 |
| Rotating speed | 1450 [rpm] | 1500 [rpm] |
| Impeller diameter | 0.193 [m] | 0.340 [m] |




a) b)

*Fig. 1. Computational fluid domain of HPRTs A (a) and B (b) running in pump mode (red) and turbine mode (orange)*

Besides the impeller and the volute, two pipes are added upstream and downstream the machines. The length of the axial pipe is five times the diameter of the impeller and it allows to have a fully developed flow at both inlet (pump mode) and outlet section (turbine mode) of the two hydraulic machines. The meshes of the fluid domains of HPRT A and B are shown in Fig. 2a and Fig. 2b, respectively.




*a)* *b)*

*Fig. 2. Meshes of HPRTs A (a) and B (b) computational fluid domains*

Table 2 lists the number and the type of elements used to discretize the computational fluid domains, while the grid independence has been ensured by successive mesh refinements.

*Table 2. Spatial discretization of HPRTs A and B*

| Geometry | # of elements (A) | # of elements (B) | Elements type (A) | Elements type (B) |
|---|---|---|---|---|
| Impeller | 1,398,917 | 1,514,004 | Tetrahedra | Tetrahedral, Pyramids & Wedges |
| Pipes | 195,342 | 107,752 | Tetrahedra & Wedges | Tetrahedra & Wedges |
| Volute | 431,669 | 182,814 | Tetrahedral, Pyramids & Wedges | Tetrahedral, Pyramids & Wedges |
| Total | 2,025,928 | 1,804,670 | Tetrahedral, Pyramids & Wedges | Tetrahedral, Pyramids & Wedges |

After the CFD models are generated, proper Boundary Conditions (BCs) are set. Regarding both inlet and outlet BCs, both of them are applied to the extreme sections of the two pipes. A reference of 0 bar and a liquid temperature of 35.5 °C are imposed. Table 3 lists both fluid dynamic and turbulence parameters used for setting up the fluid domain.

*Table 3. Fluid dynamic and turbulence parameters set-up*

| Type of boundary | Fluid dynamic parameters | Turbulence parameters |
|---|---|---|
| Inflow | Mass Flow Rate [kg/s] normal to boundary condition | Intensity 5% |
| Outflow | Average Static Pressure 1 [bar] with Radial Equilibrium | - |
| Wall | Adiabatic and no-slip condition | - |
| Interface Impeller-Volute | Conservative Interface Flux with Pitch Angle of 360 [degree] | Conservative Interface Flux |
| Interfaces Pipes-Volute | Conservative Interface Flux with Pitch Angle of 360 [degree] | Conservative Interface Flux |

The interfaces allow to link two or more elements for assembling the overall fluid domain. Three interfaces are present: two of them, related to the connection between the two pipes and the volute, are stationary, while the last one, regarding the connection between the impeller and the volute, is rotating. The Frozen Rotor Model (FRM) is used for simulating the rotating domain: it is a steady state method that uses rotating reference frame to reduce the computational efforts. Two positions of the impeller are simulated per each HPRT operating at BEP: the angular sectors between the nail of the volute and the closest blade's tip are equal to 17.8˚, 45.5˚ and -5.4˚, 12.6˚ for the HPRTs A and B in a counterclockwise direction, respectively. The highest percentage difference was equal to 3.88%, which is recorded in the evaluation of the head of the HPRT A. Reynolds Average Navier Stokes (RANS) equations and k-ω (two-equations) turbulence model [16] are used and discretized using the High-Resolution scheme, while for the near wall-treatment an automatic wall function is applied. The automatic wall function allows to switch from wall-functions to a low-Reynolds near wall formulation as the mesh is refined. The obtained $y^+$ values range between 0 and 200, being acceptable since it is suggested to have values lower than 300 [17, 18]. Only one outlier, equal to 850, is obtained for the HPRT B: however, this value is achieved in the cavity between the impeller and the rear disk, close to the shaft, due to the low flow velocity. Since this zone does not provide a significant torque for the impeller, it can be considered negligible. The convergence residuals criteria is set to $10^{-4}$, taking into account the Root Mean Square (RMS) values.

### *2.3. SELEXOL® properties*

In this paper, the SELEXOL® solvent is used for simulating the behaviour of the HPRTs when they handle liquids with higher viscosity. This solvent is generally used in a closed-loop process for removing the $H_2S$ from the synthesis gas produced in gasification plants. Table 4 lists the properties of this solvent that is involved in the CFD simulations.

*Table 4. SELEXOL® properties [19, 20]*

| Property | State-Value | Property | State-Value |
|---|---|---|---|
| Thermodynamic state | Liquid | Molecular weight (kg/kmol) | 148.22 |
| Viscosity at 35.5 °C (cP) | 4 | Specific Heat (J/gK) | 2.05 |
| Density at 35.5 °C (kg/m$^3$) | 1012 | | |

## 3. Results and comments

### *3.1. Pump mode performance curves and correction factors*

CFD simulations are performed taking into account two different liquids: water and SELEXOL®. Firstly, they are carried out with water in order to validate the results obtained from laboratory tests [5]; considering the head, the efficiency and the mechanical power as main reference magnitudes, a maximum error of 6.48% is obtained for both HPRTs A and B. Subsequently, other simulations are performed with the SELEXOL® having the properties listed in Table 4. Fig. 3a and Fig. 3b show the performance curves of the HPRT A obtained with CFD simulations in pump mode with SELEXOL® and their comparison with the experimental tests and the trend obtained using the correction factors of the technical report ISO/TR 17766. It is worth to notice that the BEP is achieved at the same flow rate value, independently of the liquid typology.

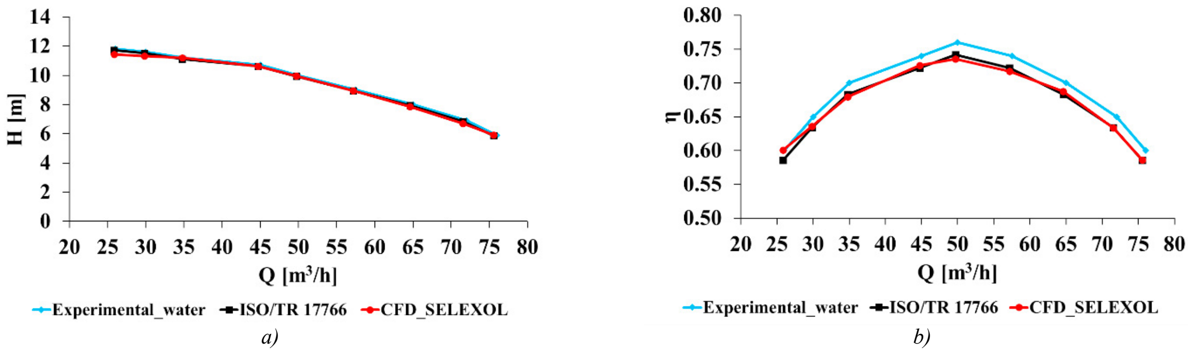
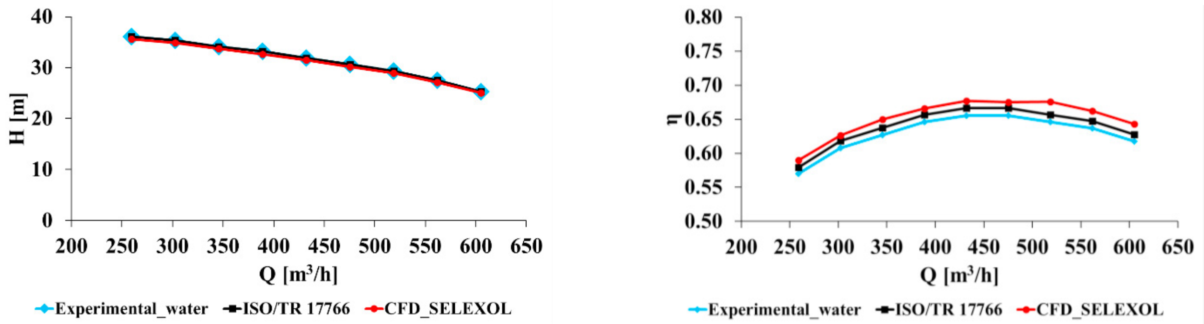



*Fig. 3. Comparison of both characteristic (a) and efficiency (b) curves of the HPRT A operating in pump mode with two liquids*

Fig. 4a and Fig. 4b show the results obtained from the CFD simulations of the HPRT B in pump mode with SELEXOL®. In addition, also the forecasted curves according to correction factors of the technical report ISO/TR 17766 are pointed out. Both Fig. 3 and Fig. 4 show that the correction factors of the technical report ISO/TR 17766 allow to correctly adjust the performance curves of the analysed HPRTs in an accurate way. Indeed, the maximum error for the HPRT A, taking into account the head, the efficiency and the power, was equal to 2.53%, while for the HPRT B it was equal to 4.26%.



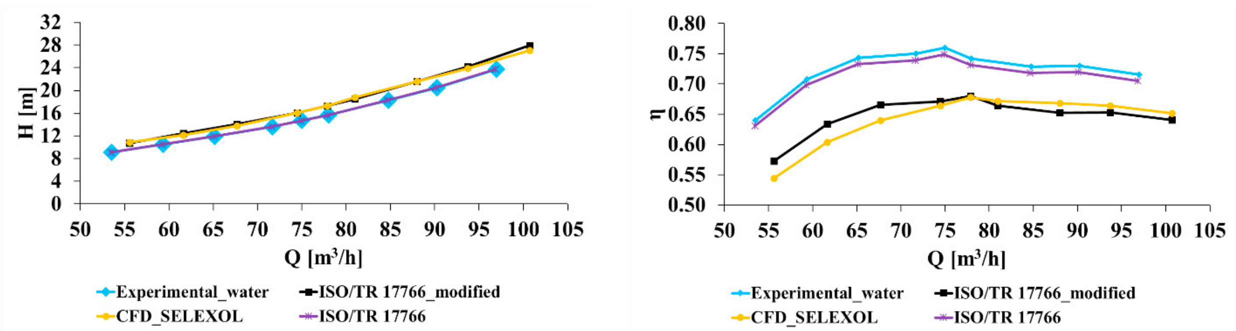

a) b)

*Fig. 4. Comparison of both characteristic (a) and efficiency (b) curves of the HPRT B operating in pump mode with two liquids*

Also in this case, the BEP is found at the same flow rate value. It is worth to notice that the values of the correction factors, depending on the parameter B, allow to correct accurately the performance curves of both hydraulic machines. However, two different efficiency trends are detected for the HPRTs A and B operating in pump mode with water and SELEXOL®. HPRT A shows that the efficiency's values obtained by handling SELEXOL® are lower than the ones with water, while the HPRT B shows an opposite behaviour. In the first case, the maximum difference is detected in correspondence of the BEP (3.28%), while lower ones are obtained at the extremities of the curve (0.11% at 26 $m^3$/h and 2.33% at 76 $m^3$/h). In the second one, the differences are similar per each analysed flow rate, resulting in an average value of 3.58%.

### *3.2. Turbine mode performance curves and correction factors*

The same procedure is followed for assessing the performance in turbine mode. For the HPRT A, the BEP in turbine mode with SELEXOL® is recorded at a flow rate value (77.88 $m^3$/h) that is approximately 4% higher than the one obtained with water (75 $m^3$/h). The increase of the flow rate at BEP of HPRTs in turbine mode is also observed in other works related to the study of other liquids having different viscosity with respect to water [7, 8]. The results are shown in Fig. 5a and Fig. 5b. In this case, the parameter B was higher than 1 (1.42) and specific coefficients are implemented in the equations in order to fit the forecasted performance curves, using the procedure suggested by the technical report ISO/TR 17766. The new equations for the proposed correction are expressed in equation (10), equation (11) and equation (12). These equations are obtained by imposing BEP values (flow rate, head and efficiency) obtained from the CFD analysis, thus achieving a Minimum Square Error (MRE) equal to 0 for that point.




a) b)

*Fig. 5. Comparison of both characteristic (a) and efficiency (b) curves of the HPRT A operating in turbine mode with two liquids*

$$C_Q = (2.71)^{+4.115\cdot(logB)^{3.15}} \tag{10}$$

$$Q_{BEP-H} = C_Q \cdot 1.14 \tag{11}$$

$$C_\eta = B^{-\left(0.4201\cdot B^{0.69}\right)} \tag{12}$$

On the other hand, the BEP of the HPRT B in turbine mode with SELEXOL® does not change with respect to the one obtained with water: results are shown in Fig. 6a and Fig. 6b. In this case, the parameter B is lower than 1 (0.56); thus, the correction factors for forecasting both characteristic and efficiency curves in turbine mode remained unchanged.

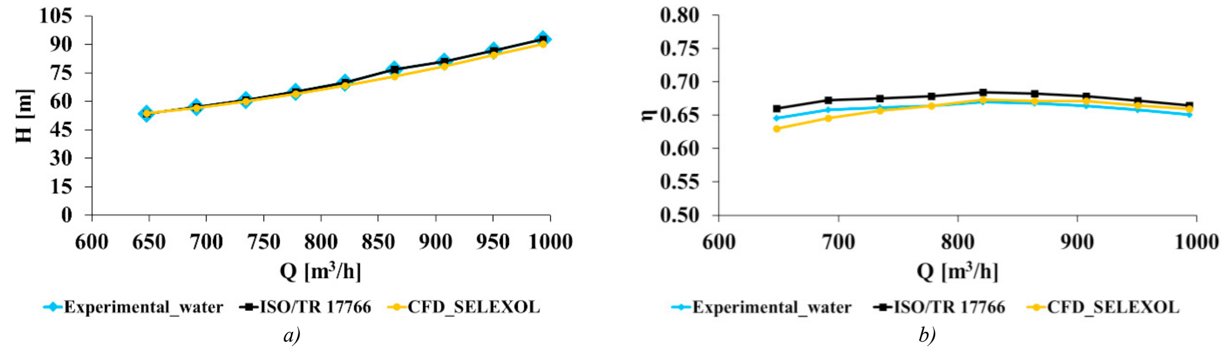



*Fig. 6. Comparison of both characteristic (a) and efficiency (b) curves of the HPRT B operating in turbine mode with two liquids*

Fig. 5 shows that, for the HPRT A, the correction factors of the technical report ISO/TR 17766 have to be modified for fitting both characteristic and efficiency curves in a quite accurate way, while, for the HPRT B, the same correction factors of the technical report ISO/TR 17766 allow to forecast both the performance curves without any modifications. This last result is important because it is possible to affirm that, if B>1, the correction factors have to be modified with the proposed equations, while, if B<1, the same correction factors used for evaluating the pump performance can be also used in turbine mode. Finally, regarding the errors, the highest one recorded for the HPRT A, taking into account the head, the efficiency and the power, was equal to 7.29%, while for the HPRT B it was equal to 6.53%. Regarding the general trend of both characteristic and efficiency curves with water and SELEXOL®, different behaviours are recorded in both turbine and pump modes. The characteristic curve of HPRT A showed higher values of head by handling SELEXOL® instead of water, while no differences are detected in the HPRT B. For the efficiency curves, the same behaviour in pump mode is present in turbine mode, even though the HPRT A showed lower differences (max. 2.50%) despite of the HPRT B (max. 14.99%). However, since discordant results are obtained by the analysis of the two HPRTs in terms of both characteristic and efficiency curves' evaluation, further studies regarding the influence of the viscosity have to be carried on by analysing other HPRTs.

## 4. Conclusions

In this paper, two centrifugal pumps used in both pump and turbine mode (HPRTs) handling a liquid with a higher viscosity than water are studied by the means of CFD simulations. Generally, pumps are tested only with water, which involves a lack of information when they operate with other liquids. For this reason, some studies are carried on to supply correction factors and adjust their performance curves when they handle other liquids. This issue can be encountered in several applications, like in oil refineries where there are several chemical processes that require high pressures and/or temperatures. The technical report ISO/TR 17766 supplies correction factors for modifying the performance curves of centrifugal pumps, obtained with water, handling liquids with higher viscosity than water. However, it does not provide any information related to the turbine mode. Thanks to CFD simulations of two machines, validated with the laboratory tests with water, the behaviour of these machines in both pump and turbine modes with SELEXOL® is studied. This fluid is a solvent that is generally used in some $H_2S$ removal processes from the synthesis gas produced in gasification plants, which presents a higher viscosity with respect to the water. Results showed that, in pump mode, the correction factors of the technical report ISO/TR 17766 allow to evaluate the performance curves of both hydraulic machines in an accurate way. On the other hand, the corrections in turbine mode presented two different behaviours. The HPRT A showed a B value higher than 1 (1.42) and, for its performance prediction in turbine mode, both correction factors related to the flow rate, head and efficiency are modified in order to fit the CFD results. The corrected coefficients are reported and the new performance curves are compared to the results of the CFD simulations. The maximum error, taking into account the head, the efficiency and the power, was equal to 2.53% and 7.29% in pump and in turbine mode, respectively. On the other hand, the HPRT B showed a B value lower than 1 (0.56) and, in this case, the same correction factors of the technical report ISO/TR 17766 can be used effectively. The maximum error, taking into account the same previous magnitudes, was equal to 4.26% and 6.53% in pump and in turbine mode, respectively. These encouraging preliminary results allow the authors to conclude that the parameter B could influence the correction factors in turbine mode. Other test cases have to be performed and studied for further validating this hypothesis.

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