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NUMERICAL STUDY OF THE FLOW OF SUPERCRITICAL CO_2 IN A RADIAL CENTRIFUGAL PUMP

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Abstract. *The present work aims to numerically evaluate the performance of a multistage radial centrifugal pump operating with supercritical CO_2. Numerical simulations are carried out using the multiple reference frame (MRF) technique as implemented in the ANSYS® CFX® software. To calculate the enthalpy and temperature variation through the pump, the total energy equation is solved together with the mass and momentum equations. At the same time, the flow turbulence is modeled through the κ–ω SST model. The thermo-physical properties of the fluid are calculated through interpolations using a real gas property (RGP) thermodynamic table obtained from the NIST® REFPROP® coupled to the CFD code. The numerical model is validated with experimental data obtained from the pump operating with water. The results from the pump simulations operating with supercritical CO_2, at different rotating speed, flow rate, pressure and temperature at the pump inlet, allowed us to build the pump performance and temperature increment curves. With that, it was verified whether pumps operating with supercritical fluids could be characterized in a similar way as when they operate with water. This is a basic but important first step to evaluate the feasibility of using centrifugal pumps to boost supercritical fluids in the present context, especially due to the scarce literature available about this subject.*

Keywords: *Centrifugal pump, Performance curves, Supercritical fluid, CFD simulation.*

1. INTRODUCTION

The discovery of oil fields with high CO_2 concentration, such as in the Brazilian “Pre-Salt” regions and in the Southeast Asian oil and gas fields, has brought new challenges for their production. In the Southeast Asian fields, CO_2 molar concentration can reach up to 40% in the gaseous fraction (JPT, 2022), while in the Brazilian pre-salt fields the CO_2 molar concentration can reach up to 79% (Almeida, 2016). Besides environmental concerns, carbon dioxide is considered a contaminant that reduces the calorific value of natural gas, also representing a negative factor on flow assurance and others technical aspects.

Currently, during the production of these oil fields, the CO_2 undergoes a process of separation and compression until its supercritical state, and then it is reinjected or transported to other depleted fields in order to mitigate its release to the atmosphere. However, the available technologies used in this process are expensive, energy-intensive and occupy a significant area on the production units; in some cases, the area occupation can be as high as 60% (Passarelli et al., 2019). The latter can make oil and gas production unfeasible, mainly in offshore scenarios, where the high concentration of CO_2 requires large capacity installations. But the bigger challenge, more that the technology, is likely to be the economics of these projects and how operators justify the additional development cost with the implementation of these technologies (JPT, 2022). Likewise, at a supercritical CO_2 transport pipeline network, the high-energy requirement of compressors and booster pumps makes their operation expensive (Okezue and Kuvshinov, 2018).

To overcome the mentioned difficulties, new technologies are proposed, such as the high-pressure dense phase subsea separator (HISPTM) presented in the work by Passarelli et al. (2019). This technology allows the subsea separation of at least two phases: a liquid, rich in hydrocarbons, and a dense, supercritical phase rich in CO_2 (Passarelli et. al., 2019; De Souza et al. 2019). The liquid phase is sent to the production units, while a centrifugal pump could be used to inject the CO_2–rich dense phase into the oil reservoir.

However, besides other mechanical aspects, it is not clear whether a centrifugal pump, which is usually designed to work with liquids, will work under similar performance when handling a supercritical fluid in comparison with its baseline operation with water. Regardless of their high density, supercritical fluids have greater compressibility than liquids (Cunico and Turner, 2017). Thus, the temperature rise of the fluid through the pump, which is usually not a concern for operation with liquids, with the exception of very viscous ones, becomes important when it comes to pumping compressible fluids such as supercritical CO_2, because the compression work, the density variation and the other thermophysical properties directly depend on this parameter.

Although the capture, pipeline transportation and reinjection of supercritical CO_2 is not a new topic, research in the open literature on centrifugal pumps operating with supercritical fluids are quite scarce, especially numerical works. Moreover, no formal methodology was found to characterize centrifugal pumps performance operating with supercritical fluids, which makes the problem a knowledge gap.

In this context, the present work aims to evaluate the performance of a multistage centrifugal pump operating with supercritical CO_2, through a numerical approach using the ANSYS® CFX® CFD package. The variation of the thermophysical properties of the fluid is simulated through the implementation of thermodynamic tables in the numerical model, while the performance and temperature increase of the pump are evaluated through a polytropic approach, a borrowed concept from compressor theory. In addition, an algebraic model to calculate the temperature increase of the fluid along the pump is proposed, as an additional tool to help in the performance evaluation of supercritical fluid pumping. From the outcomes of the present study, we intend to explore the problem not just to evaluate the feasibility of using a centrifugal pump for pumping supercritical fluids in terms of its performance, but also to discuss other aspects that should contribute to the scarce literature on this subject.

2. METHODOLOGY

2.1 Numerical modeling

The single-phase flow of a supercritical fluid within a rotating subdomain is governed by the continuity, Eq. (1), momentum, Eq. (2), and total energy transport equations, Eq. (3). In this work, they are numerically solved assuming Reynolds averaging for turbulence modeling (RANS) in a non-inertial reference system, through the Element-based Finite Volume Method and the Frozen-Rotor model as implemented in the ANSYS® CFX® computational fluid dynamics package (2022). Details about the numerical discretization, convergence criteria, the Frozen-Rotor model and other technical numerical aspects can be found in Stel et al. (2015) and ANSYS (2022).

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left(\rho \vec{V} \right) = 0 \tag{1}$$

$$\frac{\partial \rho \vec{V}}{\partial t} + \nabla \cdot \left(\rho \vec{V} \cdot \vec{V} \right) = -\nabla P + \nabla \cdot \left(\tau_{eff} \right) - 2\rho \cdot (\vec{\Omega} \times \vec{V}) - \rho \cdot \vec{\Omega} \times (\vec{\Omega} \times \vec{r}) \tag{2}$$

$$\frac{\partial \rho h_{tot}}{\partial t} - \frac{\partial p}{\partial t} + \nabla \cdot \left(\rho \vec{V} \cdot h_{tot} \right) = \nabla \cdot \left(k \nabla T \right) + \nabla \cdot \left(\vec{V} \cdot \tau_{eff} \right) + S_E \tag{3}$$

Above, P, T, ρ, k and h_{tot} represent the pressure, temperature, density, thermal conductivity and total enthalpy of the working fluid, respectively; $\vec{r}$, $\vec{V}$, $\vec{\Omega}$ are the position vector of a fluid particle, the velocity vector in the rotating coordinate system (non-inertial) and the angular velocity vector. In turn, S_E is a source term that includes the internal heat generation and others energy sources such as viscous work; τ_{eff} is the effective stress tensor for RANS equations, which includes the laminar and turbulent effects of the flow. The last two terms of Eq. (2) correspond to the Coriolis and centrifugal effects, respectively. They appear as a consequence of adopting a rotating (non-inertial) frame of reference. However, these equations could be assumed generically for multiple reference frame simulations since, for a static frame of reference, these two terms naturally vanish as $\vec{\Omega} \equiv 0$.

The flow turbulence was modeled using the $\kappa - \omega$ SST model (Menter, 1994). This model adequately predicts turbulent flows in regions close to and away from walls. In addition, it is considered one of the most suitable for modeling turbomachinery operating with supercritical fluids, as discussed for example by (Kim et al., 2014) and (Ameli et al., 2018).

The domain of the numerical solution is based on the 9-stage centrifugal pump model CM1-9 from Grundfos®. Figure 1 shows a scheme of the numerical domain used in the simulations, while Figure 2 shows the numerical grid adopted for the simulations. The numerical model assumes three stages of the pump, since simulating all nine stages proved to be too computationally expensive. Also, the complicated suction and discharge sections of the actual pump were simplified by straight, circular pipes in the numerical model, which was made long enough to make the inlet and outlet boundaries far from the region of interest, i.e. the three pump stages.

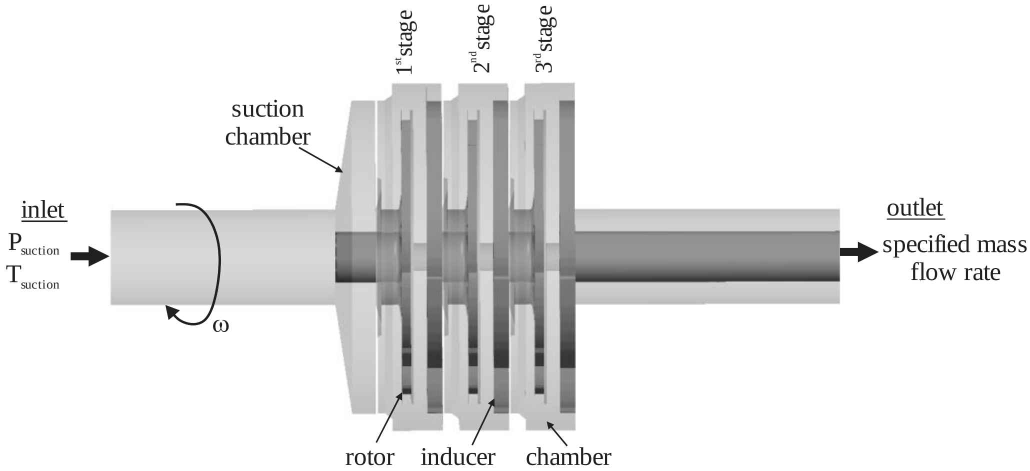

Figure 1. Numerical domain scheme used in the simulations.

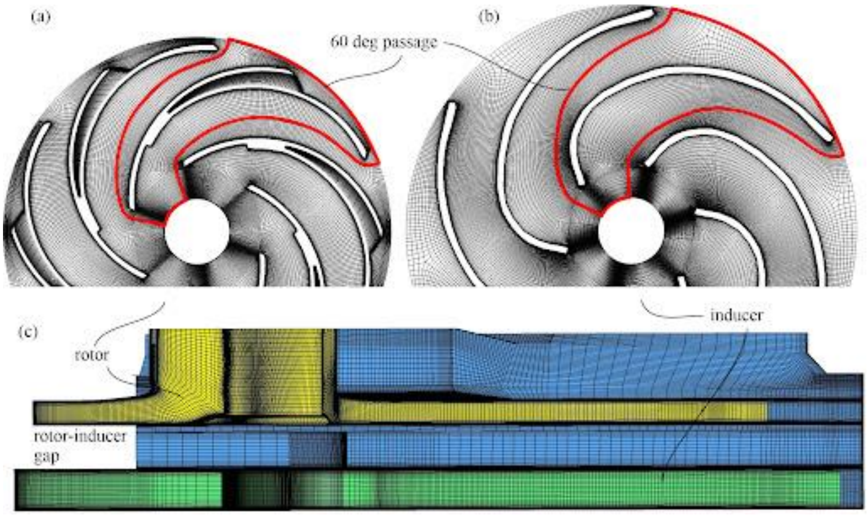

Figure 2. Numerical grid adopted for the simulations: (a) rotor, (b) inducer and (c) centrifugal pump one-stage.

As both the rotor and inducer of the pump stages have six blades, only 1/6 of the azimuthal extent (360º/6=60º) of the geometry was modeled, as indicated by the red line in (a) and (b) of Figure 2, to save computational time. At the inlet of the domain, the absolute suction pressure and temperature are specified, while the mass flow is specified at the outlet.

Variations of the thermophysical properties of the supercritical CO_2 were simulated through interpolations on a RGP (Real Gas Property) thermodynamic table, which is previously calculated using models from the NIST® REFPROP® program and then uploaded to the ANSYS® CFX® code. This allows obtaining stable numerical simulations with lower computational time compared to solving an actual equation of state during run-time. Also, the NIST® REFPROP® models are said to be well validated for supercritical CO_2, as described for instance in (Monge, 2014) and (Ameli et al., 2018).

2.2 Algebraic model to calculate the fluid temperature inscrease through the pump

Unlike traditional liquid pumping, except for very viscous liquids, the performance characterization of centrifugal pumps operating with supercritical fluids is strongly dependent on the knowledge of the discharge temperature, which is also important for design and control of the actual pumping system. However, in most cases, the discharge temperature is another unknown parameter to be calculated.

In the literature, there are several approaches for calculating the discharge temperature during a compression process. However, some of them depend on numerical integrations, while others are limited to single-stage turbomachinery, e.g., the isentropic model, or are a function of the same discharge temperature, such as the polytropic model. Because of this, a physically consistent and yet practical algebraic model has been developed. This model allows the fluid temperature rise to be calculated from known operating conditions, without need for numerical integration, discharge temperature iteration, table interpolation or other methods from literature that would require programming.

The first step in developing the model was to find a generic relation or equivalence between the turbomachinery operating conditions and the thermodynamic properties of the fluid involved in the process. For this, the problem was evaluated through the integral approach of the first law of thermodynamics, considering a control volume involving the centrifugal pump. Furthermore, it is assumed that the pumping process reaches the steady state, is adiabatic, that gravitational are negligible, and the shaft work is predominant over other works. From these assumptions, the integral form of the first law of thermodynamics reduces to Eq. (4), which indicates that the specific mechanical shaft work of the pump is completely transformed into total fluid energy in the form of total enthalpy increase.

$$\frac{\dot{W}_{sh}}{\dot{m}} = \left(h_{tot\_2} - h_{tot\_1}\right) \tag{4}$$

where $\dot{W}_{sh}$ is the shaft power consumed by the centrifugal pump, $\dot{m}$ is the mass flow rate, $h_{tot} = h + V^2/2$, is the total enthalpy, h is the static enthalpy and V is the fluid velocity. Subscripts 1 and 2 represent the pump's suction and discharge conditions, respectively.

From de above equation, the discharge temperature, T_2, could be calculated indirectly as a function of the discharge enthalpy and discharge pressure, $T_2 = T(P_2, h_{tot\_2})$, an approach which can be performed in many commercial thermodynamic softwares. The discharge enthalpy, h_{tot_2}, is calculated from Eq. (4), considering that P_2, $\dot{W}_{eix}$, $\dot{m}$ are known performance parameters and h_{tot_1} is a function of the centrifugal pumps suction conditions, assumed to be also known. At first, this calculation process seems relatively simple. However, it has the disadvantage that interpolation is required, since there is no thermodynamic relation or equation of state to directly obtain the temperature from pressure and enthalpy. Also, this process must be repeated for each operating point of a performance curve for pumping scenarios, reasons that make this process less practical.

Thus, a second step is carried out to evaluate the enthalpy increment, because it represents the energy gained by the fluid through the pump. For a pumping or compressor perspective, it is expected that this enthalpy increment is always accompanied by a simultaneous increase in pressure and temperature through the device. This behavior indicates the existence of an interdependent relationship between these properties (h, P, T). According to the phase rule of thermodynamics for a pure substance, from a set of thermodynamic properties, specifying any two of these properties will allow obtaining all the remaining ones. In our case, pressure and temperature were chosen to specify enthalpy, $h=h(T,P)$, since they are properties that can be measured experimentally.

Since the differential of a property is exact and enthalpy is a state function that depends only on the initial and final thermodynamic state of the process, a differential relation for enthalpy with respect to temperature and pressure can be developed through the mathematical definition of the exact differential (Elliott and Lira, 2012), as follows:

$$dh = \left(\frac{\partial h}{\partial T}\right)_p dT + \left(\frac{\partial h}{\partial P}\right)_T dP \tag{5}$$

Equation (5) indicates that the enthalpy gain of the fluid can be estimated by the sum of two processes: (i) an isobaric process and (ii) an isothermal process. The final version of the proposed algebraic model starts from the solutions of this equation, applied to our problem. For this, the partial derivatives of this equation were replaced by their respective thermodynamic properties, $\left(\partial h/\partial T\right)_P = c_p$ and $\left(\partial h/\partial P\right)_T = v(1 - T\beta)$, where c_p is the isobaric specific heat, v is the specific volume and β is the coefficient of volumetric expansion.

After that, the equation was integrated as follows: (i) the first term was integrated over de discharge pressure isobaric line, P_2; and (ii) the second term was integrated over the inlet temperature isotherm, T_1. Here, T_1 and P_2 are assumed to be available parameters, the former because it is part of the operating condition at which one wants to evaluate performance, the latter because it is considered that at least the pump pressure rise curve is known from pump catalog, which can be re-evaluated for fluids other than the one used in the datasheet tests (usually water) using ordinary affinity laws. As a result, the proposed model expression for calculating the fluid temperature gain through the centrifugal pump results in:

$$T_2 - T_1 = \Delta T = \frac{\dot{W}_{sh}}{\dot{m} \cdot c_{p1}} + \bar{\mu}_{JT}\left(\Delta P\right) \rightarrow \bar{\mu}_{JT} = \frac{\mu_{JT}(T_1, P_1) + \mu_{JT}(T_1, P_2)}{2}$$
$$\mu_{JT} = -\frac{v \cdot \left(1 - T\beta\right)}{c_p} \tag{6}$$

where P, T, c_p, μ_{JT} and $\dot{m}$ represent the pressure, temperature, isobaric specific heat, Joule-Thomson coefficient and the mass flow rate, respectively; $\dot{W}_{sh}$ and ΔP are the pump's power consumption and pump pressure increase. Subscripts 1 and 2 represent pump's suction and discharge conditions, respectively.

Equation (6) indicates that the proposed model can be viewed as composed of two parcels, both of which have operating parameters available from manufacturer catalogs: (i) the energy transferred to the fluid as shaft work per unit mass; and (ii) the centrifugal pump compression capacity, ΔP. Therefore, the greater the power consumed and the greater the pressure gain produced by the pump, the greater will tend to be the temperature increase. Furthermore, it is noted that the thermodynamic properties involved in the model can be specified from known information inherent to the process (T_1, P_1 and P_2).

It is important to note that, in the proposed model, the isobaric specific heat, c_{p1}, is considered constant for each pumping scenario, since the consideration of an average c_p did not show significant advantages for the cases evaluated and, for convenience it is specified as a function of the pump inlet conditions (T_1, P_1),. In turn, the average Joule-Thomson coefficient, $\bar{\mu}_{JT}$, will vary as a function of the different pressure gains produced by the pump.

As can be seen, the algebraic model is much more practical than the previously described enthalpy interpolation method. Its application depends only on the knowledge of the thermodynamic conditions at the inlet (P_1,T_1) and discharge pressure (P_2) of the pump for the specification of the properties involved in the model, as well as operational parameters, specifically the power consumed by the pump, the pressure gain and the associated flow rate for a given operating point. The latter are assumed to be available in catalog curves. Nevertheless, in practice, the pump catalog curves are usually obtained for water. However, as the results will show, they can be used as references to simulate and obtain pump performance curves operating with supercritical CO_2 by using dimensionless pump coefficients or affinity laws.

Detailed comparisons and discussions on the results of the proposed temperature gain model will be presented in the results section, where such results will be compared with numerical results, with those of Eq. (4) and with those obtained from the polytropic model, the latter being frequently used in the literature and considered the most widely used model in industry (Lüdke, 2004). Eq. (7) shows how the polytropic discharge temperature is calculated:

$$T_2 = \left(\frac{P_2}{P_1}\right)^m \rightarrow m = \frac{P}{c_p}\left[\frac{v}{T}\left(\frac{1}{\eta_p} - 1\right) + \left(\frac{\partial v}{\partial T}\right)_p\right] \tag{7}$$

where P, T, c_p, v and η_p represent the pressure, temperature, isobaric specific heat, specific volume and pump polytropic efficiency, respectively. Subscripts 1 and 2 represent the pump's suction and discharge conditions, respectively.

In the polytropic model application, it is necessary to calculate an average temperature polytropic exponent, $\bar{m}$. For this purpose, the thermodynamic properties at the pump inlet and outlet must be known, including the discharge temperature, making the model an iterative problem. Furthermore, in most applications of this model, the polytropic efficiency is considered unique and constant. However, in practice, the efficiency of a pump or compressor varies with operating conditions or molar concentrations if the working fluid is a mixture (Okezue and Kuvshinov, 2018).

3. RESULTS AND DISCUSSIONS

To verify the validity of the numerical model, the numerical results are first compared with experimental data from the pump operating with liquid water at 4500 rpm, as indicated in Figure 3, which compares numerical and experimental data of pressure rise, power and efficiency as a function of flow rate. A good agreement between the two cases is observed.

Although the numerical model slightly underestimates the experimental data at high flow rates, it agrees generally well with the experimental data through almost the whole operational curve of the pump, especially close to the best efficiency flow rate, 2.97 m³/h. In this sense, it is believed that the model is well calibrated and can be used as a reference in the analysis of simulations with supercritical CO_2.

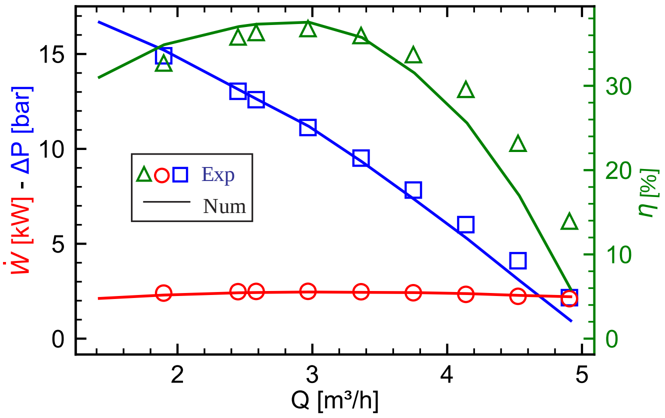

Figure 3. Comparison between numerical and experimental data of pressure, power and efficiency, as a function of flow rate, for the pump operating with water at 4500 rpm.

Figure 4 shows the curves of dimensionless head coefficient, ψ, and power coefficient, Π, as a function of the flow coefficient, ϕ, for simulations of the centrifugal pump operating with water (baseline) and supercritical CO_2 at different rotating speed and thermodynamic suction conditions. For both coefficients, ψ and Π, a similar behavior is observed. Both performance curves for supercritical CO_2 present excellent agreement, which are also found to be close to the dimensionless water curve. This behavior suggests that the ordinary pump similarity laws traditionally used for liquids can be extended for supercritical fluids. However, a small displacement of the CO_2 curves in relation to the water curve is observed. This may be a consequence of the lower viscosity of supercritical CO_2, so the head coefficient is slightly higher, and the power consumption is lower than the water values, positively affecting the pump's efficiency.

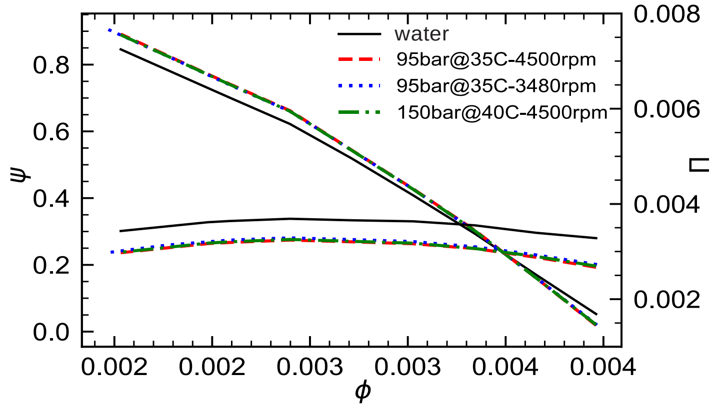

Figure 4. Head and power coefficients as a function of the flow coefficient for the centrifugal pump operating with supercritical CO_2 under different conditions and with water.

Figure 5 shows the temperature increase curves obtained through the CFD model, the polytropic model, Eq. (7), the first law of thermodynamics, Eq. (4), and our proposed model, Eq. (6), for the pump operating with supercritical CO_2 under two rotating speeds and at a suction condition of 95 bar and 35 ºC. It is observed that the proposed model has the ability to predict the temperature increase with an accuracy similar to the ones obtained with the other traditional models. In addition, all models show a good agreement with the temperature rise curve obtained numerically.

The results also show the significant influence of the rotation speed on the fluid temperature increase. This behavior is expected, since the consumed power and pressure gain of the centrifugal pump are proportional to its rotation. Then, as the proposed model indicates, the temperature increase is directly proportional to these two operational parameters, although the thermodynamic condition at the pump suction is also important on this respect. Also, it can be seen that, the higher the flow rate, the lower the temperature increase. Again, this behavior is related to the pressure increase of a centrifugal pump, which decreases as the flow rate increases.

Furthermore, it is observed that in the specific case of low flow coefficients, all models tend to slightly overestimate the numerical temperature. One explanation for this is related to the equivalence relationship between the enthalpy gain of the fluid and the specific power consumed by the pump, Eq. (4). In fact, the proposed model and the first law model are directly dependent on this equivalence, while the polytropic model will have an indirect dependence through efficiency. Figure 6 shows the ratio between fluid enthalpy gain and specific shaft power, calculated from numerical data, as a way to check the impact of the equivalence assumed at Eq. (4) on the prediction accuracy of the models.

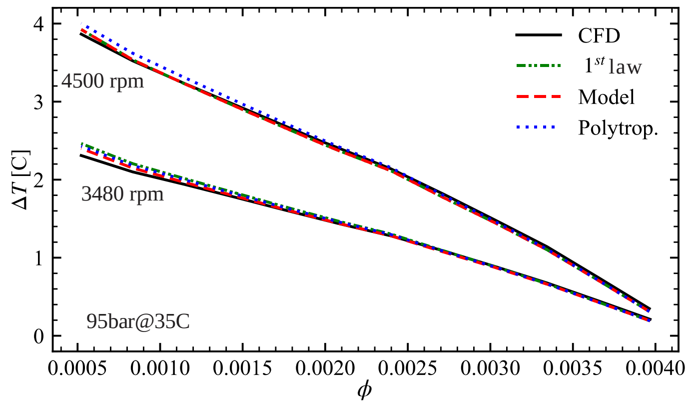

Figure 5. Temperature increase curves obtained through CFD, the proposed model and traditional models for the pump operating with supercritical CO_2 under different rotating speeds.

As expected, in cases where the equivalence is fulfilled, that is, when the ratio is close to unity, the results of the proposed methodologies show close agreement with the numerical ones. However, for low flow rates, this ratio starts to deviate from unity, indicating that the specific power consumed by the pump is greater than the enthalpy gain of the fluid, which causes the models to overestimate the temperature gain in relation to the numerical data. In general, it is observed that for the numerically evaluated scenarios, the equivalence between the enthalpy gain and the specific shaft power is fulfilled for a wide range of flow rates. Even in the low flow rate cases, the smallest value of the ratio was calculated as 0.88, so for practical purposes the assumptions leading to Eq. (4) work well considering its simplicity and accuracy. Therefore, a good estimate or measurement of shaft power will allow the models to calculate the temperature gain accurately.

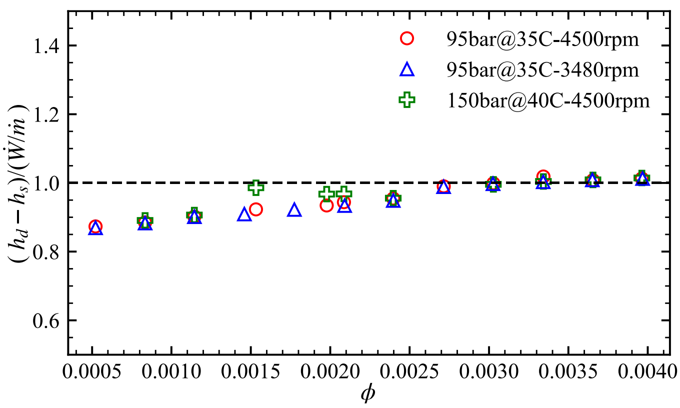

Figure 6. Ratio between the total enthalpy gain of the fluid and the power consumed by the centrifugal pump, from the numerical results for different operational conditions.

As mentioned, the supercritical CO_2 pumping characterization depends on the knowledge of the discharge temperature. Once the discharge temperature is estimated, it is possible to specify other thermodynamic properties used in calculating pump performance parameters, such as head and efficiency. To assess the differences between using a traditional pump approach or a polytropic process in calculating these performance parameters, Fig. 7(a) compares head curves as a function of fluid flow rate for the centrifugal pump operating with supercritical CO_2 at different rotating speeds and considers 95 bar and 35 ºC in the pump suction.

The pump head, adapted for the case of a compressible fluid (H_{pum}), Eq. (8), is compared against the polytropic heads, using the conventional model (H_p), Eq. (9) , and the model proposed by Mallen-Saville (1977) ($H_{p(M\text{-}S)}$), Eq. (10). The latter is estimated with the results of the proposed temperature rise model and those of the first law, since the discharge properties of the model are a function of the discharge temperature. Traditional models, on the other hand, are estimated with numerical results.

In turn, Figure 7(b) compares the conventional efficiency for pumps, $\eta = Q \cdot \Delta P / \dot{W}_{sh}$, against the polytropic efficiency, $\eta_p = H_p/(h_2 - h_1)$, for the different pump simulation scenarios operating with supercritical CO_2. In the results shown for the 95 bar and 35°C suction conditions, the results for 3480 rpm and 4500 rpm are plotted together.

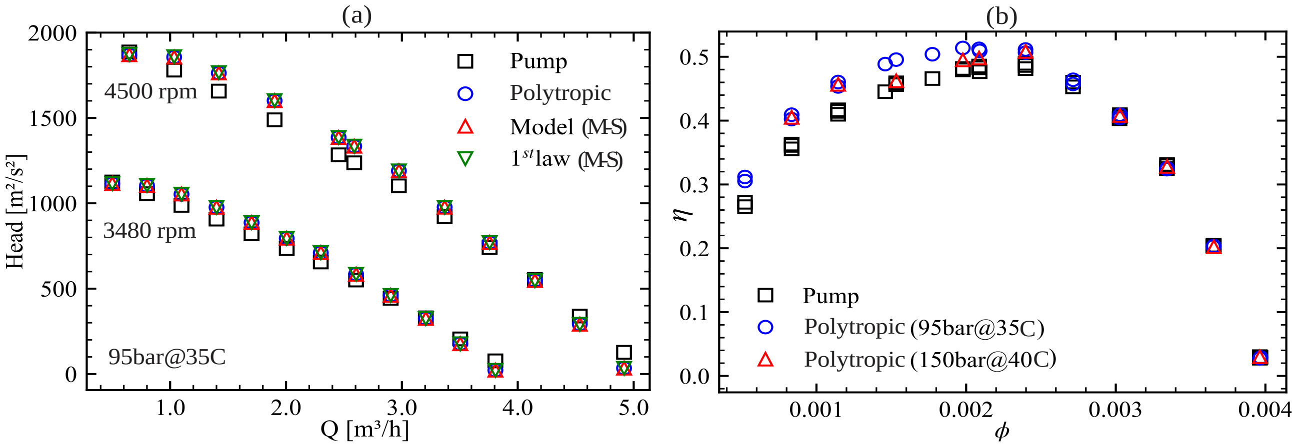

Figure 7. (a) Comparison between the curves of the pump head, traditional polytropic head, and Mallen–Saville polytropic head (using different temperature models), for different rotating speed and suction conditions of 95 bar and 35°C. (b) Comparison between the efficiency curves, as a function of the flow coefficient, calculated in the traditional form for pumps and a polytropic process, for different centrifugal pump operating conditions.

$$H_{pum} = P_2/\rho_2 - P_1/\rho_1 \tag{8}$$

$$H_p = \frac{P_1}{\rho_1} \cdot \frac{n}{n-1}\left[\left(P_2/P_1\right)^{(n-1)/n} - 1\right] \Rightarrow n = \frac{ln(P_1/P_2)}{ln(\rho_1/\rho_2)} \tag{9}$$

$$H_{p(M-S)} = \left(h_2 - h_1\right) - \frac{\left(T_2 - T_1\right)}{ln(T_2/T_1)}\left(s_2 - s_1\right) \tag{10}$$

where P, T, h, s and ρ represent the pressure, temperature, enthalpy, entropy and density of the fluid, respectively, and n is the polytropic exponent of head.

In the scenarios evaluated, it can be observed that for flow rates above the BEP, 2.3 m³/h for 3480 rpm and 2.97 m³/h for 4500 rpm, the values of the pump head and the polytropic heads are very close to each other. However, for flow rates below BEP, the polytropic heads tend to be larger than the pump head. This difference becomes more evident at the 4500 rpm rotating speed. Because of this behavior, it is believed that as the speed or pressure ratio increases, the difference between pump heads and polytropics will become more significant. On the other hand, the results of the polytropic heads, calculated with different models, resulted in good agreement with each other. This suggests that the Mallen-Saville model can be used for our problem and that the variations between the results of the temperature gain models had no significant influence on the calculation of the polytropic head for the simulated scenarios.

On the other hand, it is observed that there is a good agreement between the behavior of the pump and polytropic efficiency curves. Moreover, the values calculated by the two approaches were relatively close, especially for high flow rates. This behavior reveals that in the evaluated scenarios, the pump catalog efficiency can be used to estimate the polytropic temperature exponent, Eq. (7), when the polytropic efficiency is unavailable. It is also observed that under some operating conditions, the polytropic efficiency is somewhat higher than that of pumps. As already mentioned, one of the reasons for this behavior is the lower viscosity of supercritical CO_2 compared to water and the lower power consumption, as shown in Figure 4.

4. CONCLUSIONS

This paper numerically investigates the performance of a multistage radial centrifugal pump operating with supercritical CO_2. Before the simulations with supercritical CO_2, the numerical model was validated with experimental data from the pump operating with water, with a good agreement being obtained between the numerical and experimental data, especially near the best efficiency flow rate.

The analysis with dimensionless numbers of pumps, using the numerical results, shows that the common similarity laws of pumps used with liquids, like water, can be used to estimate the performance curves of a centrifugal pump operating with supercritical fluids.

An algebraic model has been proposed to estimate the temperature gain of the fluid through the centrifugal pump. This model was shown to be practical and only depends on known operational parameters, such as pump application conditions and manufacturer's catalog curves. Furthermore, the results of the proposed model showed good agreement with CFD data and other traditional models. For some operating points, it resulted in a superior agreement with the numerical results than the other models.

The comparison between the heads, calculated traditionally for centrifugal pumps and a polytropic process, showed that the polytropic head is more significant than that of pumps when the rotational speed or the pressure ratio increases. This latter increases with increasing rotational speed and decreasing flow rate. In turn, the polytropic heads, calculated with different models of temperature and head, agreed well one with each other. On the other hand, the small difference between the pump efficiency and the polytropic efficiency suggests that the latter can be estimated from the simpler to calculate, hydraulic efficiency of centrifugal pumps.

The results of this investigation could help bringing a greater understanding about the behavior of centrifugal pumps operating with supercritical CO_2 in terms of its performance and other aspects that should contribute to the scarce literature on this subject.

5. ACKNOWLEDGEMENTS

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