Heat transfer in supercritical steam flowing inside spiral tubes

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ABSTRACT

We investigate heat transfer in supercritical steam flowing in a spiral tube by conducting three-dimensional numerical simulations. The current numerical solver has been validated with the existing experimental results, and simulations are performed by varying different geometric parameters of a spiral tube. The flow dynamics and heat transfer in a spiral tube are compared against those in a straight tube. For the parameters range considered in the present study, it is found that the heat transfer coefficient (HTC) in the spiral tube is 29\% higher than that in the case of a straight tube for the same flow and thermal conditions. Our results indicate that the tangential velocity component resulting due to the spiralling effect of the steam is the primary reason for the enhancement of the HTC value. It is observed that while the HTC in a spiral tube is inversely related to the spiral diameter, it does not exhibit a strong relationship with the spiral pitch. Moreover, three existing heat transfer correlations are evaluated under the spiral flow condition and it is observed that none of them can calculate the HTC value accurately in spiral tubes. Using the Buckingham $\pi$-theorem, three modified correlations are proposed for the low, moderate and high heat flux regimes, which accurately predict the wall temperature and HTC of supercritical steam in spiral tubes in all the heat flux regimes. The correlations have an error band of less than +/-20\%.

Keywords: CFD; supercritical; spiral; Dean number; diameter ratio; HTC

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1.0 INTRODUCTION

Due to increased cycle efficiency and environmental benefits, the use of supercritical steam in thermal power plants is becoming more common. However, the heat transfer process involving supercritical fluids is a complex phenomenon and it requires to consider real gas effects while calculating property change of supercritical fluids [1]. During heat transfer to such fluids, heat transfer enhancement (hte) or heat transfer deterioration (htd) may happen, depending on several factors such as mass flux, wall heat flux, inlet temperature, etc. [2-3]. The heat transfer coefficient (HTC) experiences a steep jump during hte and the wall temperature of the tube shoots up during htd. While the deteriorated HTC is an unwanted phenomenon, there is continuous effort from the designer’s side to prolong the process of hte. There are several active and passive methods to achieve hte. Active methods are those which involve external forces such as electric or acoustic and passive methods use specially designed surface geometries [4]. The use of helically or spirally shaped tubes is one increasingly popular passive way of achieving hte, and in some cases avoiding the htd [5]. Even, in the case of subcritical fluids, there are evidences of improved heat transfer due to the use of spiral tubes [6].

The spiral tubes derive their superior heat transfer characteristics from their ability to pack more surface area in limited space [7-10]. There is also evidence of improving the overall heat transfer by just inserting a small spiral tube inside a regular straight or spiral tube, mainly because of the reduced cross-sectional area and the added turbulence due to spiralling inside tube [5, 11]. In processes involving radiative heat transfer also, spiral tubes offer better heat transfer efficiency, though the wall emissivity influences the overall HTC to a great extent [12]. Zhao et al. [13] performed numerical studies on heat transfer to supercritical water and investigated the centrifugal force acting on the fluid in spiral tubes. In addition to the centrifugal force, there are other factors such as tube orientation, helical diameter and pitch etc. which affect the heat transfer in spiral tubes as well [6, 7, 14, and 15]. There is a significant amount of research work going on in this field of passive heat transfer enhancement technique, especially in exploring the exact role of different flow and geometrical conditions on HTC.

Xu et al. [16] performed experimental studies on heat transfer to spiral tubes with varied flow conditions and they stated that the HTC improves with an increase in mass flux. Li et al. [7] investigated the role of tube orientation on heat transfer to supercritical CO₂. Mirgolbabaei [6] performed a numerical study on the impact of helical pitch and tube diameter on the overall effectiveness of heat exchangers. They stated that the effectiveness of heat exchangers is
inversely related to the helical pitch and there is no definite dependence of effectiveness on tube diameter. However, in none of the studies discussed above, the working fluid was supercritical steam. There are few studies in the literature that suggest that the HTC may increase with a decrease in helical diameter, but there is no direct evidence of any impact of helical pitch on HTC, especially with supercritical steam as working fluid.

As discussed above, the heat transfer process to supercritical fluids is a complex phenomenon. Added to that the complexity of spiral tubes, it makes the heat transfer to supercritical fluids flowing inside spiral tubes an under-explored subject in literature. There are several correlations in literature to predict the HTC for straight and spiral tubes, but each correlation is specific to the geometry and flow condition. The Dittus-Boelter correlation is the most widely used correlation, which is given by

\[ Nu_b = 0.023 \times Re_b^{0.8} \times Pr_b^{0.4}. \]  

Though this correlation is more suitable for sub-critical conditions, many researchers use this as the basis to develop newer ones. Yamagata et al [2] proposed a correlation to estimate the Nusselt’s number at bulk fluid temperature (\( Nu_b \)) for supercritical steam with an error band of +/- 20%. Bishop’s [17] correlation is one correlation which was developed under supercritical conditions of steam. There is a constant debate in the literature whether to calculate the thermophysical properties of the fluid at the bulk fluid temperature or at wall temperature while using in the correlation. Swenson’s [18] correlation is one such correlation that uses the wall temperature as a basis for calculating the thermophysical properties of fluid. The Bishop’s and Swenson’s correlations are given by

\[ Nu_b = 0.0069 \times Re_b^{0.9} \times Pr_b^{0.66} \left( \frac{\rho_w}{\rho_b} \right)^{0.43} \left( 1 + 2.3 \frac{D}{x} \right), \]  

\[ Nu_w = 0.00459 \times Re_w^{0.923} \times Pr_w^{0.613} \left( \frac{\rho_w}{\rho_b} \right)^{0.231}, \]

respectively. In the above equations, the subscripts ‘b’ and ‘w’ denote the properties calculated at the bulk fluid temperature and wall temperature, respectively. All the correlations discussed above are developed and validated for straight tubes.

In the literature, though few correlations are proposed for spiral tubes, each one has its own stipulation in terms of parameter range, flow condition, etc. Dravid et al. [19] proposed a correlation for estimating the \( Nu \) in helical flows, but it is valid only for laminar regimes.
Cengiz et al. [20] proposed two correlations for two regimes of Dean number ($De$), but their study consisted of a rotating helical pipe, which is significantly different from studies with stationary pipes. Rahul et al. [21] developed a correlation for estimating $Nu$ in a cross-flow scenario. However, their working fluid was air. Kalb et al. [22] proposed a correlation to estimate the $Nu$ in curved tubes, other than spiral or helical ones. As discussed above, each correlation is specific to the geometry and flow conditions.

In the present work, the heat transfer of the supercritical steam flowing inside a spiral tube is numerically studied with the help of commercially available CFD software, Ansys Fluent 18.1. The numerical model is first validated with the experimental data of Morky et al. [23-24]. The advantage of spiral tubes over straight tubes in terms of heat transfer characteristics is first established. The reason for enhanced heat transfer in spiral tubes is probed numerically. The role of geometrical parameters such as diameter ratio $D_r$ (ratio of spiral diameter to tube internal diameter) and pitch ratio $b_r$ (ratio of spiral pitch to tube internal diameter) in influencing the HTC is investigated. Few existing correlations for calculating $Nu$ in tube flows are evaluated under spiral conditions. Finally, three new correlations are proposed for low, moderate and high heat flux regimes and they are validated with the data generated from simulation.

2.0 GEOMETRY AND NUMERICAL SCHEME

To validate the numerical model, a straight tube of 4 meters length and 10 mm internal diameter is considered. The straight tube is similar to the one used by Morky et al. [23] in their experiments. Supercritical steam enters the tube through the inlet, receives heat from the heated cylindrical wall and exits the tube through the outlet. After validation, the numerical model is used to study the flow and heat transfer to supercritical steam in spiral tubes. The spiral tube has the same internal diameter ($d$) and length ($L$) as that of the straight tube. A schematic of the spiral tube is shown in Fig.1. The flow model for numerical simulation is discretized in Ansys mesher with predominantly hexahedral mesh elements. Adequate number of prism layers is generated near the tube wall to ensure that the $y^+$ value in numerical simulation remains near unity. The helical diameter ($D$) and helical pitch ($b$) are varied to create different flow models, keeping $L$ constant.
2.1 Solver

The flow of supercritical steam is simulated using Ansys Fluent 18.1, which is a finite-volume based commercial CFD solver. The governing equations are the conservation of mass, conservation of momentum (Navier-Stoke equations) and conservation of energy. The pressure-based segregated algorithm SIMPLEC (Semi-Implicit Method for Pressure Linked Equations-Consistent), which is based on a predictor-corrector approach, is used for solving the equations. A second-order central differencing scheme is used for pressure, diffusion terms, while a second-order upwind scheme is used for the convective terms in the governing equations. The SST k-ω turbulence model with standard wall functions has been chosen as the turbulence model to simulate the steam flow. The convergence criteria for iterative calculations used in the CFD simulations are as follows. The minimum root mean square residual is 1e-04 for flow and the turbulence parameters and 1e-06 for energy parameters.

2.2 Materials and Boundary conditions

The supercritical state of steam is simulated by using the real gas model based on the National Institute of Standards and Technology (NIST) material library. The real gas model uses the NIST thermodynamic and transport Properties of Refrigerants and Refrigerant Mixtures Database Version 9.1 (REFPROP v9.1) to evaluate thermodynamic and transport properties. The REFPROP v9.1 database is a shared library that is dynamically loaded into the solver [25]. The experimental conditions of Morky et al. [23-24] are prescribed as boundary conditions for the simulations. For the two validation cases, the inlet of the straight tube is prescribed with a mass flux value of 500 kg/m²s and 1002 kg/m²s and the outlet is prescribed
with a pressure value of 242 bar and 239 bar respectively. The wall heat fluxes for these cases are 335 kW/m$^2$ and 681 kW/m$^2$ respectively. The inlet temperature of steam in both the cases is 350 °C. For simulation with spiral tubes, a combination of steam mass flux, wall heat flux, inlet temperature and steam pressure is chosen from Morky et al. [23-24] set of data and prescribed as boundary conditions. The range of parameters considered in simulations is depicted in Table-1. None of the experimental conditions chosen for simulation falls under deteriorated heat transfer regime.

### Table-1: Range of steam parameters considered in our simulations.

<table>
<thead>
<tr>
<th>Mass flux (kg/m$^2$s)</th>
<th>Heat flux (kW/m$^2$)</th>
<th>Inlet temperature (°C)</th>
<th>Inlet pressure (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>499-1499</td>
<td>236-681</td>
<td>341-350</td>
<td>239-242</td>
</tr>
</tbody>
</table>

The turbulence parameters at the inlet and outlet in the simulation are specified in terms of turbulent intensity (the ratio of root-mean-square of the velocity fluctuations to the mean flow velocity) and hydraulic diameter. The following formula is used to have a gross estimation of the turbulent intensity for flow inside ducts [25],

$$I = \frac{U'}{U_{avg}} = 0.16\left(Re_{D_H}\right)^{-\frac{1}{6}},$$

where $I$ is the turbulence intensity, $U'$ is the root-mean-square of the velocity fluctuations, $U_{avg}$ is the mean flow velocity and $Re_{D_H}$ is the Reynolds number of the flow based on hydraulic diameter. For a flow with a Reynolds number value of 50,000, the turbulent intensity value is close to 4% [25]. Therefore, in the present simulation, a turbulent intensity of 5% has been assigned at the boundaries. For all the simulation cases, the wall of the tube is treated as a rough wall with average wall roughness value as 63 μm.

### 2.3 Grid convergence study

Simulations are performed with three different grids, namely 1.89 million, 2.12 million and 2.67 million cells, to rule out the dependency of results on grid sizes. The spiral tube with $D$ as 0.084 m and $b$ as 0.033 m is chosen as the flow model. Mass flux of 1002 kg/m$^2$s and wall heat flux of 681 kW/m$^2$ are prescribed as the boundary conditions. Steam at a temperature of 350°C and a pressure of 239 bar is taken as the working fluid. The area-averaged value of steam temperature at tube outlet ($T_o$) and that of the last section of the spiral wall ($T_{w-o}$) are extracted in each case and are compared in Table-2. The changes in the outlet temperature of steam and
wall temperature are 0.04 % and 0.08 %, respectively when the number of grids changes from 2.12 million to 2.67 million. Therefore, the grid with 2.12 million number of cells is selected as the optimum grid and in the subsequent simulation cases this discretization scheme is followed.

Table-2: Comparison of outlet temperature values predicted for different grids.

<table>
<thead>
<tr>
<th>Grid size (million)</th>
<th>T_o (° C)</th>
<th>% Change</th>
<th>T_w-o (° C)</th>
<th>% Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.89</td>
<td>409.93</td>
<td>-</td>
<td>476.54</td>
<td>-</td>
</tr>
<tr>
<td>2.12</td>
<td>409.22</td>
<td>-0.17</td>
<td>474.81</td>
<td>-0.36</td>
</tr>
<tr>
<td>2.67</td>
<td>409.41</td>
<td>0.04</td>
<td>475.2</td>
<td>0.08</td>
</tr>
</tbody>
</table>

2.4 Estimates of uncertainty

To estimate the uncertainty associated with the numerical analysis, additional simulations are performed with different pressure-based segregation algorithms and discretization schemes. The straight tube with $d$ as 10 mm and $L$ as 4 meters is chosen as the flow model. Mass flux of 1002 kg/m$^2$s and wall heat flux of 681 kW/m$^2$ are prescribed as the boundary conditions. Steam at a temperature of 350 °C and a pressure of 239 bar is taken as the working fluid. The SST k-ω turbulence model with standard wall functions has been retained as the turbulence model. As depicted in Table-3, three different combinations of pressure-based segregation algorithms and discretization schemes are implemented in simulation. The QUICK scheme is a weighted average of second-order upwind and central interpolations discretization schemes for convective terms. The area-averaged value of steam temperature at tube outlet ($T_o$) and that of the last section of tube wall ($T_{w-o}$) are extracted in each case and are compared in Table-3. As it can be seen in table-3, the deviations are -0.009% and 0.004% in $T_o$ and $T_{w-o}$, when the algorithms and discretization schemes changed from the SIMPLECT and second-order upwind to the SIMPLE and QUICK. Therefore, the uncertainty in property prediction by the simulations is not significant in the present simulations.
Table-3: Comparison of temperature values predicted by different segregation algorithms and discretization schemes.

<table>
<thead>
<tr>
<th>Algorithms and discretization schemes</th>
<th>Outlet parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_o$ (°C)</td>
</tr>
<tr>
<td>SIMPLEC, second order upwind</td>
<td>409.029</td>
</tr>
<tr>
<td>SIMPLEC, QUICK</td>
<td>409.025</td>
</tr>
<tr>
<td>SIMPLE, QUICK</td>
<td>409.020</td>
</tr>
</tbody>
</table>

3.0 VALIDATION OF NUMERICAL MODEL

To assess the accuracy of the numerical model, the predicted parameters must be compared with the existing data set, preferably experimental. However, a reliable source of experimental data set pertaining to heat transfer to supercritical steam in spiral tubes is not openly available in literature. Therefore, the experimental data set of Morky et al. [23-24] pertaining to heat transfer to supercritical steam in straight tubes is used as the reference data for model validation. The test geometry of Morky et al. [23] is used as the flow model. The variation of bulk fluid temperature ($T_b$), wall temperature ($T_w$) and HTC along the length of the tube, as predicted by the numerical model is compared with experimental data in Fig.2. In both cases, the predicted trend of $T_b$ exactly matches with that of the experiment. The trend of $T_w$ and HTC as predicted by the numerical model also matches very closely with that of the experiment.

The percentage deviation in the average value of predicted parameters from that of experimental parameters is reported in Table-4. The maximum deviation in predicted value of parameters from experimental value, in both the cases is under 5%. Therefore, the numerical model is quite accurate and the same can be used to simulate the flow of supercritical steam in spiral tubes.
Fig. 2: Comparison of variations in $T_b$, $T_w$ and HTC with respect to $L$ as predicted by CFD simulation with experimental data. (a) $T_b$ and $T_w$ for 500 kg/m$^2$s and 335 kW/m$^2$, (b) $T_b$ and $T_w$ for 1002 kg/m$^2$s and 681 kW/m$^2$, (c) HTC for 500 kg/m$^2$s and 335 kW/m$^2$, and (d) HTC for 1002 kg/m$^2$s and 681 kW/m$^2$.

Table-4 Comparison of predicted value of steam parameters with experimental values.

<table>
<thead>
<tr>
<th></th>
<th>500 kg/m²s 335 kW/m²</th>
<th>1002 kg/m²s 681 kW/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average value</td>
<td>Experiment</td>
<td>Simulation</td>
</tr>
<tr>
<td>$T_b$ (°C)</td>
<td>379.46</td>
<td>381.94</td>
</tr>
<tr>
<td>$T_w$ (°C)</td>
<td>437.05</td>
<td>438.52</td>
</tr>
<tr>
<td>HTC (kW/m²K)</td>
<td>6.32</td>
<td>6.01</td>
</tr>
</tbody>
</table>

The numerical model predicts the values of $T_b$, $T_w$ and HTC satisfactorily. However, the variation of thermophysical properties of steam when the bulk fluid temperature approaches the pseudo-critical value is extremely abrupt. Another study has been carried out to assess the capability of the numerical model, especially the in-built NIST real gas model to capture the
large change in thermophysical properties near pseudo-critical point. The test geometry of Yamagata et al. [2] has been considered as the flow model and the experimental conditions of Lemon et al. [26] has been prescribed as the boundary condition for this study. A mass flux of 1260 kg/m²s is prescribed at the inlet of the test cylinder and a heat flux of 565 kw/m² is used at the wall. The inlet temperature of steam is fixed at such a value that the pseudo-critical state of steam is achieved inside the test cylinder. The values of density (ρ), isobaric specific heat (Cₚ) dynamic viscosity (µ) and thermal conductivity (k) predicted from our CFD study and previous experiment (Lemon et al. [26]) are compared in Fig. 3. It can be seen that the results obtained from the present CFD study agree well with the experimental results.

![Figure 3: Comparison of variations in thermophysical properties of steam with bulk fluid temperature, as predicted by the real gas model with that of experiments (a) ρ, (b) Cₚ, (c) µ, and (d) k.](image)

### 4.0 DATA GENERATION

To propose new correlation for spiral tubes, a sufficiently large pool of data pertaining to heat transfer to supercritical steam in spiral tubes is essential. For this purpose, nineteen numerical simulations are performed with various flow and geometry conditions. To begin with the mass flux, wall heat flux, inlet temperature and steam pressure are varied and six distinct cases are formulated with a fixed spiral geometry. Further, the D and b of the spiral tube are varied and seven distinct cases are formulated with a fixed set of flow conditions. Finally, six
different flow conditions are applied to a straight tube of same $d$ and $L$ to compare the heat transfer pattern with spiral tubes.

### 4.1 Advantages of spiral tubes over straight tubes

As discussed above, six different flow conditions are applied to the fixed geometries of spiral and straight tubes and numerical simulation of steam flow performed. The variation of $T_b$, $T_w$ and HTC with respect to the length of spiral and straight tubes, for two simulation cases are produced in Fig.4. Though there is no significant difference in $T_b$ between a straight tube to spiral tube, the values of $T_w$ are observed to be less in the case of a spiral tube. However, the biggest difference is observed in the variation of HTC. The HTC in the spiral tube is significantly higher as compared to the straight tube in both the cases considered.

The average values of temperature and HTC in both straight and spiral tube is reported in Table-5. As can be seen that there is around 29 % improvement in average HTC value from straight to spiral tubes in both the cases simulated. The primary reason for increased heat transfer co-efficient in spiral tubes is the tangential velocity arising out of spiraling effect. The vector plots of tangential velocity drawn on two planes along the flow path in the spiral tube are produced in Fig.5. It can be seen that the presence of tangential velocity on the cross-section promotes secondary flow or cross-flow. Secondary flow brings in better mixing across the cross-section and consequently better heat transfer. Therefore, the HTC in case of spiral tubes is substantially higher when compared with straight tubes.

The contour of the tangential velocity at those two planes are plotted for both straight tube and spiral tube and are shown in Fig.6. It can be seen that for the case of straight tube the tangential velocity is almost zero throughout the cross-sections. On the other hand, the tangential velocity is significant in the case of a spiral tube and the absolute value of it is more towards the outer side of the spiral tube. The centrifugal force coming out of spiraling effect could be the primary reason for this effect.
Fig. 4: Comparison of variations in $T_b$, $T_w$, and HTC with respect to $L$ for straight and spiral tubes (a) $T_b$ and $T_w$ for the case of 484 kW/m$^2$ and 1002 kg/m$^2$s, (b) $T_b$ and $T_w$ for the case of 681 kW/m$^2$ and 1002 kg/m$^2$s, (c) HTC for the case of 484 kW/m$^2$ and 1002 kg/m$^2$s, and (d) HTC for the case of 681 kW/m$^2$ and 1002 kg/m$^2$s.

Table 5: Comparisons of predicted parameters for the straight tube with the spiral tube for (484 kW/m$^2$, 1002 kg/m$^2$s) and (681 kW/m$^2$, 1002 kg/m$^2$s).

<table>
<thead>
<tr>
<th></th>
<th>484 kW/m$^2$, 1002 kg/m$^2$s</th>
<th>681 kW/m$^2$, 1002 kg/m$^2$s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average value</td>
<td>Straight</td>
<td>Spiral</td>
</tr>
<tr>
<td>$T_b$ ($^\circ$ C)</td>
<td>376.02</td>
<td>376.18</td>
</tr>
<tr>
<td>$T_w$ ($^\circ$ C)</td>
<td>401.38</td>
<td>396.13</td>
</tr>
<tr>
<td>HTC (kW/m$^2$K)</td>
<td>19.48</td>
<td>25.13</td>
</tr>
</tbody>
</table>
Fig. 5: Vector plots of the tangential velocity at two planes for spiral tube. (a) Identification of internal planes in the spiral tube, (b) vector plot of tangential velocity on plane 9, and (c) vector plot of tangential velocity on plane 57.

Fig. 6: Contour plots of the tangential velocity at (a) plane number 9 of the straight tube, (b) plane number 57 of straight tube, (c) plane number 9 of the spiral tube, and (d) plane number 57 of the spiral tube.
The absolute values of velocity components at four different planes along the flow path, for both straight and spiral tubes are depicted in Table-6. In the straight tube, the share of radial and the tangential velocity on the planes is zero. On the other hand, in the spiral tube, the tangential component on each plane is dominant and the axial component is negligible. Clearly, the presence of radial and tangential velocity on the cross-section of spiral tube promotes better mixing and subsequently resulting in better HTC.

Table-6: Comparisons of velocity components on four planes for the straight and spiral tubes.

<table>
<thead>
<tr>
<th>Plane number</th>
<th>Velocity in straight tube (m/s)</th>
<th>Velocity in spiral tube (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Axial</td>
<td>Radial</td>
</tr>
<tr>
<td>9</td>
<td>1.8</td>
<td>0</td>
</tr>
<tr>
<td>39</td>
<td>3.3</td>
<td>0</td>
</tr>
<tr>
<td>57</td>
<td>4.9</td>
<td>0</td>
</tr>
<tr>
<td>87</td>
<td>7.1</td>
<td>0</td>
</tr>
</tbody>
</table>

4.2 Impact of geometrical parameters on heat transfer in spiral tubes

Since the primary reason for improved heat transfer in spiral tubes is the centrifugal force arising out of spiralling effect, it makes sense to probe the role of geometrical parameters such as \(D_r\) and \(b_r\) in heat transfer. Multiple cases are studied with different \(D_r\) and \(b_r\) to ascertain the role. Mass flux value of 1002 kg/m²s, heat flux value of 681 kW/m², 350 °C inlet temperature and 239 bar pressure are used as the boundary condition in these simulations. The variation of HTC along the tube length for different \(D_r\) and \(b_r\) are produced in Fig.7. It can be seen in Fig. 7(a) that the HTC increases with decrease in \(D_r\), whereas there is no definite pattern of HTC with changing \(b_r\), as depicted in Fig. 7(b). The change in average values of HTC with changing \(D_r\) and \(b_r\) are reported in Table-7. The HTC corresponding to a \(D_r\) value of 7.5 is 15.71 % higher than that with a \(D_r\) value of 10. However, with changing \(b_r\), there is no definite pattern for the HTC and the maximum change is only around 3%. 
Fig. 7: Comparison of HTC variation with respect to $L$ (a) for different $D_r$ and (b) for different $b_r$.

The main reason for the above trend is probably the closeness of spiral tubes having higher $D_r$ to straight tubes. For the tube with higher $D_r$, the flow is closer to that in a straight tube as the spiralling effect diminishes. Whereas for the tube with lower $D_r$, the spiralling effect is more, mainly because of the steeper directional changes, the fluid experiences continuously along the path. More spiralling effect gives rise to a higher share of tangential velocity across the tube cross-section and thereby promoting better mixing. Therefore, the HTC increases with a decrease in $D_r$ in the case of the spiral tube.

Table 7: Comparison of average value of HTC for different $D_r$ and $b_r$.

<table>
<thead>
<tr>
<th>$D_r$</th>
<th>Average HTC</th>
<th>% change</th>
<th>$b_r$</th>
<th>Average HTC</th>
<th>% change</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>9.10</td>
<td>-</td>
<td>2.5</td>
<td>9.7</td>
<td>-</td>
</tr>
<tr>
<td>9.4</td>
<td>9.41</td>
<td>3.4</td>
<td>3.3</td>
<td>9.93</td>
<td>2.3</td>
</tr>
<tr>
<td>8.4</td>
<td>9.93</td>
<td>9.1</td>
<td>3.8</td>
<td>9.93</td>
<td>3.7</td>
</tr>
<tr>
<td>7.5</td>
<td>10.53</td>
<td>15.71</td>
<td>4.2</td>
<td>10.06</td>
<td>3.7</td>
</tr>
<tr>
<td>-</td>
<td>-</td>
<td>-</td>
<td>5.1</td>
<td>10.04</td>
<td>3.5</td>
</tr>
</tbody>
</table>
5.0 CORRELATION

5.1 Evaluation of existing correlations

Each correlation existing in literature is specific to the geometry and flow condition. Though the Dittus-Boelter correlation as described in Eq. (1) is mainly valid for sub-critical conditions, it’s always better to benchmark any new correlation against this one. Bishop’s and Swenson’s correlations, as described in Eqs. (2) and (3), are the ones more suitable for supercritical steam. In the present study, these three correlations are evaluated by using the simulated values of steam parameters in the spiral tube. The last term of Eq. (2) is omitted in the evaluation exercise. The flow model considered in this numerical simulation has a $D$ value of 0.084 meters and $b$ value of 0.033 meters. The flow condition considered is 500 kg/m$^2$s mass flux, 335 kW/m$^2$ heat flux, 242 bar steam pressure and 350°C inlet temperature. The variation of HTC along the tube length, as calculated by each of these correlations is produced in Fig. 8, along with the values of HTC predicted by CFD solver. It can be seen that both the Dittus-Boelter’s and Bishop’s correlations predict a sharp peak in HTC which is not present in simulated values of HTC. Similarly, the Swenson’s correlation severely under-predicts the HTC, when compared with simulated value. Therefore, none of the correlations discussed predicts the HTC value close to simulated value and there exists scope to develop new correlations, suitable for supercritical steam flow in spiral tubes.

![Fig.8: Comparison of the HTC calculated by the existing correlations with the present simulation result.](image-url)
5.2 Formulation of new correlation

As a matter of fact, the HTC is a function of multiple factors such as the tube diameter, fluid velocity, thermophysical properties etc. In this part, a holistic approach is followed to include all possible factors affecting the HTC in spiral tubes and they are listed in Table-8. The Buckingham \( \pi \)-theorem is employed to carry out a dimensional analysis and to identify the individual \( \pi \)-terms.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>HTC</td>
<td>Heat Transfer Co-efficient</td>
<td>MT(^{-3})K(^{-1})</td>
</tr>
<tr>
<td>(d)</td>
<td>Internal diameter of tube</td>
<td>L</td>
</tr>
<tr>
<td>(D)</td>
<td>Helical diameter</td>
<td>L</td>
</tr>
<tr>
<td>(b)</td>
<td>Helical pitch</td>
<td>L</td>
</tr>
<tr>
<td>(k_w)</td>
<td>Thermal conductivity of fluid at (T_w)</td>
<td>MLT(^{-3})K(^{-1})</td>
</tr>
<tr>
<td>(k_b)</td>
<td>Thermal conductivity of fluid at (T_b)</td>
<td>MLT(^{-3})K(^{-1})</td>
</tr>
<tr>
<td>(\rho_w)</td>
<td>Density of fluid at (T_w)</td>
<td>ML(^3)</td>
</tr>
<tr>
<td>(\rho_b)</td>
<td>Density of fluid at (T_b)</td>
<td>ML(^3)</td>
</tr>
<tr>
<td>(\mu_w)</td>
<td>Viscosity of fluid at (T_w)</td>
<td>ML(^{-1})T(^{-1})</td>
</tr>
<tr>
<td>(\mu_b)</td>
<td>Viscosity of fluid at (T_b)</td>
<td>ML(^{-1})T(^{-1})</td>
</tr>
<tr>
<td>(C_p)</td>
<td>Specific heat capacity of fluid</td>
<td>L(^2)T(^{-2})K(^{-1})</td>
</tr>
<tr>
<td>(v)</td>
<td>Fluid velocity</td>
<td>LT(^{-1})</td>
</tr>
</tbody>
</table>

As listed in Table-8, there are 12 variables in total and 4 fundamental variables, which indicates that there have to be 8 dimensionless \( \pi \)-terms to describe the problem considered in the present study. The fundamental variables are mass (M), length (L), time (T) and temperature (K). The dimensionless \( \pi \)-terms are elaborated in table-9.
Table-9: The dimensionless terms associated with the problem considered.

<table>
<thead>
<tr>
<th>Term</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Nu_b$</td>
<td>$\frac{htc \times d}{k_b}$</td>
</tr>
<tr>
<td>$D_r$</td>
<td>$\frac{D}{d}$</td>
</tr>
<tr>
<td>$b_r$</td>
<td>$\frac{b}{d}$</td>
</tr>
<tr>
<td>$\mu_r$</td>
<td>$\frac{k_w}{k_b}$</td>
</tr>
<tr>
<td>$\rho_r$</td>
<td>$\frac{\rho_w}{\rho_b}$</td>
</tr>
<tr>
<td>$Re_b$</td>
<td>$\frac{d \times v \times \rho_b}{\mu_b}$</td>
</tr>
<tr>
<td>$Re_w$</td>
<td>$\frac{d \times v \times \rho_b}{\mu_w}$</td>
</tr>
<tr>
<td>$Re_b^*Nu_b$</td>
<td>$\frac{C_P \times \rho_b \times d \times v}{k_b}$</td>
</tr>
</tbody>
</table>

Applying the Buckingham π-theorem, the initial form of correlation is obtained and the same is produced in Eq. (4). The constant ‘C’ and the exponents in Eq. (4) need to be fine-tuned to obtain the final form of correlation as

$$Nu_b = C \times Re_b^{a_1} \times Pr_b^{a_2} \times \rho_r^{a_3} \times \mu_r^{a_4} \times D_r^{a_5} \times b_r^{a_6}.$$  \hspace{1cm} (4)

For scenarios involving fluid flow in a spiral tube, $De_b$ is more meaningful than $Re_b$. The expression of $De_b$ also inherently contains the diameter ratio ($D_r$) as given by Eq. (5)

$$De_b = Re_b \frac{d}{\sqrt{D_r}} = \frac{Re_b}{\sqrt{D_r}}.$$  \hspace{1cm} (5)

Therefore, in Eq. (4), the $Re_b$ and $D_r$ are replaced with $De_b$ in the subsequent formation of correlation. By analysing the dependency of $Nu_b$ on other variables in the parameters range considered, a correlation matrix is formed and the same is reported in Table-10. As it can be seen, the correlation co-efficient of $b_r$ is -0.12, which indicates that there exists a very weak correlation between $Nu_b$ and $b_r$. In Fig.7 (b) also, it is demonstrated that the dependency of HTC or $Nu_b$ on $b_r$ is very weak. Therefore, the term $b_r$ may be omitted from the correlation. The reduced form of correlation, considering the correlation coefficients is described in Eq. (6)
\[ Nu_b = C \times De_b^{0.881} \times Pr_b^{0.192} \times \rho_r^{0.496} \times \mu_r^{0.402}. \] 

(6)

**Table-10: Correlation matrix listing different dimensionless variables and their coefficients.**

<table>
<thead>
<tr>
<th>Variables</th>
<th>( Nu_b )</th>
<th>( De_b )</th>
<th>( Pr_b )</th>
<th>( \rho_r )</th>
<th>( \mu_r )</th>
<th>( b_r )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Correlation coefficients</td>
<td>1.0</td>
<td>0.881</td>
<td>0.192</td>
<td>0.496</td>
<td>0.402</td>
<td>-0.12</td>
</tr>
</tbody>
</table>

A quantity named specific heat flux is defined as the ratio of wall heat flux to mass flux. The entire set of data is divided into three regimes, based on the specific heat flux value. The low heat flux regime is defined by a specific heat flux value lower than 0.4 kJ/kg. The moderate heat flux regime is defined by a specific heat flux value in the range of 0.4 kJ/kg to 0.6 kJ/kg and the high heat flux regime is defined by a specific heat flux value higher than 0.6 kJ/kg. By further analyzing the data set and minimizing the squared error between the \( Nu_b \) value calculated by the correlation and the same by simulation, three correlations are proposed for the three heat flux regimes. For low, moderate and high heat flux regimes, they are described in Eq. (7), (8) and (9) respectively

\[
Nu_b = 0.0207 \times De_b^{0.908} \times Pr_b^{-0.018} \times \rho_r^{0.457} \mu_r^{-0.599},
\]

(7)

\[
Nu_b = 0.0311 \times De_b^{0.8234} \times Pr_b^{-0.1572} \times \rho_r^{-3.396} \mu_r^{1.5851},
\]

(8)

\[
Nu_b = 0.0399 \times De_b^{0.755} \times Pr_b^{0.197} \times \rho_r^{-0.576} \mu_r^{-5.644}.
\]

(9)

### 5.3 Comparison of the proposed correlation with the simulations

Using the above three correlations, the \( Nu_b \) for each simulation case is calculated as per the heat flux regime the particular case fits into and the same is compared with simulated value in Fig.9. As it can be seen, both the calculated and simulated values of \( Nu_b \) are nicely placed around the 45° line. Out of all the data points, 95% fall within an error margin of +/-20%.
Further, using the developed correlations the $T_w$ and HTC values for all the cases are calculated and compared with simulated value. Fig.10 (a) shows the comparison of $T_w$ and HTC values for one case in high heat flux regime, Fig. 10 (b) shows the same for one case in low heat flux regime and Figs. 10 (c) & (d) show the same for two cases in moderate heat flux regime. It can be seen that the calculated values of $T_w$ and HTC match very closely with the simulated values in all the regimes.

Fig.11 shows the comparison of $T_w$ and HTC values for varying $D_r$. In section 4.2, it was discussed that the HTC value is inversely related to the $D_r$ value. In Fig.11, the trend of HTC variation with respect to $L$, as estimated by the proposed correlations also confirms the same as the peak in the HTC curve in Fig.11 (a) through Fig.11 (d) keeps reducing. More importantly, the trend of HTC and $T_w$ variation, as estimated by the correlations matches very closely with that by simulation.
Fig. 10: Comparisons of the wall temperature and HTC values calculated by the proposed correlation with simulation results for different heat flux regimes. (a) Specific enthalpy = 0.67 kJ/kg, (b) specific enthalpy = 0.39 kJ/kg, (c) specific enthalpy = 0.48 kJ/kg, and (d) specific enthalpy = 0.58 kJ/kg.
Fig.11: Comparison of the wall temperature and HTC calculated by the proposed correlation with simulation results for different values of \(D_r\). (a) \(D_r=7.5\), (b) \(D_r=8.4\), (c) \(D_r=9.4\) and (d) \(D_r=10\).

6.0 CONCLUSIONS

In the present work, several numerical simulations are performed to study the process of heat transfer to steam at supercritical conditions in spiral tube configurations. The numerical model is first validated with previous experimental data on heat transfer to supercritical steam. The numerical model is able to reproduce the reported experimental data with a deviation of under 5%. Using the model, the advantage of spiral tubes over straight tubes, in terms of heat transfer improvement is established. Over the parameters range considered in this study, the HTC of a spiral tube is 29 % higher compared with that of a straight tube, for similar flow and thermal conditions. The contour plots of velocity at different cross-sections along the flow path show the significant presence of the tangential velocity in the case of the spiral tube. The absolute value of the tangential velocity on these cross-sections is more towards the outer side of the spiral tube, which indicates that the centrifugal force acting on the fluid in outward directions is the primary reason for the enhanced heat transfer. The role of geometrical
parameters, such as $D_r$ and $b_r$ in improving the HTC in the spiral tube is also studied numerically. The value of $D_r$ is seen to be inversely related to HTC in the case of the spiral tube, while there is no definite relationship between HTC and $b_r$. The value of the HTC is improved by 15.71% when the $D_r$ is changed from 10 to 7.5, while keeping L, d and flow conditions the same.

Further, the performance of three standard correlations for the heat transfer has been evaluated for spiral flow conditions. The Dittus-Boelter correlation and Bishop’s correlation predict a sharp peak in the HTC curve, which is absent in the curve with the simulated HTC values. The Swenson’s correlation severely under-predicts the HTC value. Finally, using the Buckingham $\pi$-theorem, three correlations are proposed for the low, moderate and high heat flux regimes. The value of $Nu_b$ calculated using these correlations has an error band of +/-20% when compared with simulated values. The correlations are able to calculate the values of the HTC and $T_w$ with a reasonable level of accuracy in all heat flux regimes. The developed correlations can be helpful in the heat exchangers design in terms of the sizing of the heat exchangers. This is to be noted that the developed correlations can even be refined by broadening the range of data set considered for simulation. The detailed interpretation of the centrifugal force impacting on the thermophysical properties of supercritical steam through numerical analysis constitutes a major portion of the future work in this field of study.

ACKNOWLEDGMENTS:

The authors thank the management of BHEL for encouraging us to take up this project. The authors acknowledge the constant support provided by the technical team of ANSYS Inc.

FUNDING:

This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors.
NOMENCLATURE

\( D \)  
Spiral diameter

HTC  
Heat Transfer Co-efficient

NIST  
National institute of standards and technology

\( b \)  
Spiral pitch

\( d \)  
Internal diameter

\( htd \)  
Heat transfer deterioration

\( hte \)  
Heat transfer enhancement

\( v \)  
Fluid velocity

\( z \)  
Axial position along the length of tube

\( C_p \)  
Specific heat capacity at constant pressure

\( D_r \)  
Diameter ratio

\( De_b \)  
Dean number at bulk fluid temperature

\( T_o \)  
Outlet temperature

\( T_{w-o} \)  
Average temperature of last section of wall

\( T_b \)  
Bulk fluid temperature

\( T_w \)  
Wall temperature

\( k_b \)  
Thermal conductivity at bulk fluid temperature

\( k_w \)  
Thermal conductivity at wall temperature

\( Nu_b \)  
Nusselt’s number at bulk fluid temperature

\( Nu_w \)  
Nusselt’s number at wall temperature

\( Pr_b \)  
Prandtl number at bulk fluid temperature

\( Pr_w \)  
Prandtl number at wall temperature

\( Re_b \)  
Reynolds number at bulk fluid temperature

\( \mu_b \)  
Dynamic viscosity at bulk fluid temperature

\( \mu_w \)  
Dynamic viscosity at wall temperature

\( \mu_r \)  
Ratio of Dynamic viscosity at wall temperature to that at bulk fluid temperature

\( \rho_b \)  
Density at bulk fluid temperature

\( \rho_w \)  
Density at wall temperature

\( \rho_r \)  
Ratio of density at wall temperature to that at bulk fluid temperature
REFERENCES


