Numerical simulation of steam flow inside the superheater section of an industrial boiler using a real gas model

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ABSTRACT

Flow mal-distribution inside manifolds hampers the overall efficiency of processes in industries. In supercritical boilers, improper flow of steam inside the superheater section leads to thermal accidents. However, carrying out numerical simulation of supercritical fluids flowing inside manifolds is challenging as the ideal gas law does not describe the behavior of these fluids properly. In the present work, numerical simulation of the flow of supercritical steam inside the superheater section of an industrial boiler has been performed using a real gas model. The proposed real gas model is first validated with experimental data associated with the steam properties. Subsequently, the effect of different inlet and outlet arrangements on the flow mal-distribution of steam in the superheater section of the boiler is investigated numerically using the real gas model. A modified inlet and outlet arrangement of the superheater header is proposed which reduces the maximum value of flow mal-distribution in the header by 19.7% and total pressure drop in the domain by 17%. The effect of the Reynolds number on flow mal-distribution in the header arrangement is found to be negligible. The absolute value of the heat absorption by the superheater tubes increases with increase in the value of the Reynolds number.

Keywords: CFD; supercritical; boiler; real gas; EOS; manifold

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1.0 INTRODUCTION

Manifolds are typically used to subdivide a large stream of fluid into small parallel streams and again to collect the processed fluid from the parallel streams into a large stream. Achieving proper flow distribution of the fluid inside manifolds is extremely important for the process industry. Non-uniform flow distribution inside manifolds and parallel channels drastically reduces the overall efficiency of the process [1]. Manifolds are broadly categorized into four types, namely dividing, combining, parallel and reverse manifolds. The superheater (SH) section of a typical industrial boiler is a combination of dividing and combining manifolds. It has an inlet header which acts like a dividing manifold, an outlet header which acts like a combining manifold, and the parallel U-shaped tubes which connect both the headers.

Extensive research work has been carried out in the field of manifold flow and many corrective steps have been proposed to achieve uniformity in flow distribution. Gandhi et al. [2] performed a numerical study of various header and tube arrangements to minimize the flow mal-distribution. Jiao et al. [3] carried out CFD simulations of a plate-fin heat exchanger with 13 parallel channels and they observed that generally the central channels witness the maximum value of fluid velocity in all conventional headers. Mohan et al [4] carried out parametric analysis of flow patterns in fuel cells having multiple channels. They remarked that the number of parallel channels, fluid flow rate and channel size are the primary factors influencing flow mal-distribution. Wen and Li [5] performed experimental as well as numerical study to examine the role of perforated plates on minimizing flow mal-distribution. However, in most of the research works the flow distribution has been studied hydrodynamically, without considering the energy effect. Moreover, in none of the
works studied by us the working fluid is supercritical steam. The heat transfer characteristics becomes completely different when the fluid flowing inside the manifold is supercritical steam.

Transfer of heat to supercritical fluids is a complex phenomenon which makes carrying out computational fluid dynamics (CFD) simulations of these fluids quite challenging [6]. The property change of fluids near critical point is so abrupt that ideal gas law fails to describe the behavior of fluids accurately. Therefore, considering the real gas effects becomes essential while carrying out CFD simulations involving such fluids. Moreover, due to the advancements in computational power and the need for accurate results, complex scenarios such as hydrogen under expanded jets and shock disturbance interactions have been studied by taking real gas effects into account [7-8]. In CFD simulations, the real gas effects can be accounted for either by constructing a look-up table of gas properties or by using real gas models based on the real gas equations of state (EOS). Though the use of look-up table leads to considerable reduction in computational time, on most occasions the thermodynamic consistency is not satisfactory as compared to the EOS approach [9]. Many researchers have carried out modeling and simulation of complex processes by considering the real gas effects through EOSs [10-17]. An EOS is a set of functionalities among the primary variables, such as pressure, volume and temperature, resembling the ideal gas law, with added features of real gas effects. The real gas model is developed on the platform of a real gas EOS in the basic framework of the concerned flow solver.

The EOS that describes the co-existence of vapor and liquid was first predicted by van der Waals, commonly known as van der Waals EOS.

\[
(P + \frac{a_0}{\nu^2})(V - b_0) = RT, \tag{1}
\]
where \( a_0 = 0.42747 \frac{R^2T_c^2}{P_c} \), \( b_0 = 0.08664 \frac{RT_c}{P_c} \).

Several researchers have modified the attractive \((a_0)\) and repulsive \((b_0)\) terms in the van der Waals EOS and proposed new EOSs with better accuracy. One such modification is due to Redlich-Kwong [18], which improves the accuracy of property prediction, as compared to the original van der Waals EOS [19]. The Redlich-Kwong EOS (RKEOS) is given by

\[
P = \frac{RT}{V-b_0} - \frac{a_0}{V(V+b_0)^{0.5}}.
\]

(2)

Over the years, there have been several modifications to the original Redlich-Kwong EOS, to obtain better accuracy in the predictions. One more modification proposed to the original Redlich-Kwong EOS is by Soave [20], known as the Redlich-Kwong-Soave EOS (RKSEOS). This EOS is given by

\[
P = \frac{RT}{V-b_0} - \frac{a_0(1+(0.48+1.57\omega-0.176\omega^2)(1-T_R)^{0.5})^2}{V(V+b_0)}.
\]

(3)

Aungier [21] modified the original Redlich-Kwong EOS by introducing two additional parameters, namely the critical point compressibility factor ‘\( c_0 \)’ and the exponent ‘\( n \)’. He also established the optimum value of \( n \) for twelve different components over a range of pressure, temperature and acentric factor. The modification to the Redlich-Kwong EOS proposed by Aungier [21] (ARKEOS) is given by

\[
P = \frac{RT}{V+c_0-b_0} - \frac{a(T)}{V(V+b_0)},
\]

(4)

where \( a(T) = a_0T_R^{-n} \) and \( c_0 = \frac{RT_c}{a_0} + b_0 - V_c \).

For any real gas model, it’s very difficult and computationally intensive to capture the pseudo-critical phenomena accurately, mainly due to the abrupt property changes in the
vicinity of the critical point. However, there exist processes in the thermal industry where the behavior of steam only past its critical point is of importance, as the steam enters these processes only after achieving the critical stage somewhere in the upstream locations. The flow of steam in the SH section of supercritical boilers is one such process. Steam enters the SH section of such boilers at 220-260 bar, 420-480 °C [22], which is well past its critical parameters of 220 bar and 373.9 °C. Therefore, a real gas model which is capable of accurately describing property changes of the supercritical steam, without capturing the pseudo-critical phenomena is still useful to carry out numerical simulation of SH header arrangement.

In the present work, a real gas model has been developed in the basic framework of Ansys fluent 15.0, under the SH conditions prevailing in a supercritical boiler. The performance of the real gas model is compared with the experimental data of Lemmon et al. [23] to check the accuracy of the present model. Using our real gas model, the flow mal-distribution of supercritical steam inside the SH section of an industrial boiler of M/s Bharat Heavy Electricals Limited (BHEL) make is studied numerically. The effect of different geometrical arrangements and Reynolds number on flow mal-distribution of steam has been assessed. Three EOSs are considered to select the most suitable EOS to be used in the real gas model; they are the original Redlich-Kwong EOS (RKEOS) as described by Eq. (2), the modified Redlich-Kwong EOS by Aungier [21] (ARKEOS) as described by Eq. (4) and the Redlich-Kwong-Soave EOS (RKSEOS) as described by Eq. (3). Since in the present scenario, the working fluid is steam, the value of \( n \) used in Eq. (4) is 1.02 [21]. The value of the acentricity factor, \( \omega \) used for all the modification to Redlich-
Kwong equations in Eq. (3) and Eq. (4) is 0.3440, as suggested by Aungier [21] for the concerned range of temperature and pressure of steam.

2.0 METHOD

The CFD simulations are performed in two parts. In the first part, property variations of steam under the test condition of Yamagata et al. [24] are estimated using the real gas models based on the EOSs mentioned in section 1.0. Variations in steam properties for the bulk fluid temperature, as predicted by each model are compared with experimental results of Lemmon et al. [23] to identify the most accurate model. Each real gas model is a combination of one EOS to predict the specific volume along with other derived properties, and different relationships for predicting dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$). The real gas model consists of one EOS, in conjunction with the Sutherland formulation [25] for dynamic viscosity ($\mu$) and the Eucken formulation [26] for thermal conductivity ($\kappa$). Using the EOS, the CFD solver calculates the specific volume of fluid for a given pressure and temperature. After obtaining the values of pressure, temperature and specific volume of fluid, the solver calculates the derived properties using the expressions given in Appendix A. After obtaining the values of derived properties the solver then calculates dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$) using the Sutherland relationship [25] and the Eucken relationship [26], respectively as

$$\mu = \mu_0 \left(\frac{T}{T_0}\right)^{1.5} \frac{S + T_0}{S + T},$$

where $S$ is the Sutherland constant, $\mu_0$ and $T_0$ are the reference temperature and the reference viscosity, respectively, and

$$K = \mu \left(C_p + \frac{5}{4}R\right),$$
where R is the universal gas constant.

The performance of each real gas model is assessed and the most accurate model under supercritical conditions is identified.

Subsequently, the flow of the supercritical steam inside the SH header section of an industrial boiler is simulated, using the most accurate real gas model. The effect of different inlet and outlet arrangements on flow mal-distribution of steam is assessed. The effect of Reynolds number on flow mal-distribution is assessed by varying the incoming mass flow rate of steam. The design effectiveness of the header is studied based on the CFD analysis using the real gas model.

2.1 Computational domain:

To evaluate the performance of each real gas model, a cylindrical tube of length 1.5 m and an internal diameter 0.0075 m is considered. This configuration is similar to the one taken by Yamagata et al. [24]. An additional length of 0.25 meters, which corresponds to approximately 33 times the tube diameter, is considered at the inlet to allow the flow to be fully developed. The supercritical steam enters the tube through the inlet, receives heat from the wall and exits through the outlet of the tube. This configuration is only used to validate our model. Then using the appropriate model we simulate the steam flow in a real industrial configuration shown in Fig. 1.
Figure-1: Schematic diagram of the SH header arrangement with (a) original configuration, (b) the inlet and the outlet pipes at one end of header, and (c) the inlet and the outlet pipes at center of header.

Fig. 1 shows the schematic diagram of the SH header with three different inlet and outlet arrangements. In Fig. 1, the two branching pipes are the inlet and outlet pipes and the small U-shaped tubes are the SH tubes connecting the inlet and outlet headers. Supercritical steam enters the inlet header through the inlet pipe, travels along the length of the header while feeding the SH tubes simultaneously, reaches the outlet header and exits through the outlet pipe. The original header arrangement has the inlet and out pipes positioned diagonally opposite to each other, as shown in Fig. 1(a). Two modifications are proposed in this study wherein the inlet and outlet pipes are shifted to one end and center of the inlet header, as shown in Fig. 1(b) and 1(c) respectively. Henceforth in this paper, the original configuration is referred to as ‘type-1’, the modified configuration with inlet and outlet pipes at one end as ‘type-2’ and the configuration with inlet and outlet pipes at center as ‘type-3’. The internal diameter of both the headers and the pipes is 216 mm whereas that
of the SH tubes 40 mm. The SH section considered in the present study is a scaled-down version of the actual SH section of the industrial boiler.

2.2 Numerical approach

The flow of supercritical steam is simulated using Ansys fluent 15.0, which is a finite-volume based commercial CFD solver. The governing equations are the conservation of mass, conservation of momentum (Navier-Stoke equations), conservation of energy and the equations governing turbulence. The SIMPLEC algorithm, which is based on a predictor-corrector approach, is used for pressure velocity coupling. 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 standard k-ε turbulence model with standard wall functions has been chosen as the turbulence model to simulate the steam flow in the test section. The SST k-ω turbulence model with standard wall functions has been chosen as the turbulence model to simulate the steam flow in the SH header. 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. Additionally, before stopping the iterative calculations, it is ensured that the difference in mass flow rates between the inlet and the outlet of the domain is less than 0.5%.

2.3 Boundary conditions

As our objective to consider the test geometry (cylindrical tube) is to validate the solver and our real gas models proposed for supercritical steam, we use the same geometry and boundary conditions as used by Yamagata et al. [24]. Steam at a mass flow rate of 1260 kg/m² s enters the flow domain at a pressure of 245 bar. A constant wall flux of 465 kW/m²
is applied at the wall. To ensure that steam enters the flow domain in supercritical conditions, the inlet temperature of steam is maintained at 400 °C. To simulate the flow of steam inside the SH header, the actual condition prevailing in the SH section of an industrial boiler is prescribed. Steam with a mass flow rate of 50 kg/s enters the SH inlet header at a temperature of 420 °C and pressure of 245 bar. The wall of the SH header is maintained at a temperature which is 50 °C higher than the inlet temperature of steam. The turbulence parameters at the inlet and outlet 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 to have a gross estimation of the turbulent intensity for flow inside ducts [27],

\[
I = \frac{U'}{U_{avg}} = 0.16(Re_{D_H})^{\frac{-1}{8}},
\]

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 of 50,000, the turbulent intensity value is close to 4% [27]. Therefore, in the present simulation, a turbulent intensity of 5% has been assigned at the boundaries. At the outlet, the pressure outlet boundary condition is used with a pressure value of 245 bar.

2.4 Grid convergence study

To rule out the dependency of results on grid size while selecting the best real gas model, simulations are performed with four different grids, namely, with 0.26 million, 0.90 million, 1.68 million and 3.33 million number of cells. The cross-section views of these grids are shown in Fig. 2. All the mesh elements are of hexahedral type with an exponential
growth in size from tube wall to center. The ARKEOS is used as the real gas model in grid convergence study. The results of grid independence study are presented in table-1.

![Cross-sectional view of different grids](image)

Figure-2: The cross-sectional view of different grids used. (a) 0.26 million cells, (b) 0.90 million cells, (c) 1.68 million cells and (d) 3.33 million cells.

Values of steam properties such as density ($\rho$), isobaric specific heat ($C_p$), dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$), along with temperature at the outlet of the tube, as predicted using the grids are reported in Table 1. From a grid size of 0.26 million cells to that of 3.33 million cells, the change in the outlet temperature of steam is <0.2%. The change in all the properties of steam is within 1%, when the grids vary from 0.26 million cells to 3.3 million cells. Therefore, the finest grid, i.e. the grid with 3.3 million cells has been chosen to generate the rest of the results presented in this study.
Table 1: Steam properties at the tube outlet obtained using different grids.

<table>
<thead>
<tr>
<th>Grid size (Millions)</th>
<th>T (°C)</th>
<th>ρ (kg/m³)</th>
<th>Cₚ (J/kg K)</th>
<th>µ (Pa S)</th>
<th>κ (W/m K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.26</td>
<td>456.51</td>
<td>100.59</td>
<td>4238</td>
<td>2.91600E-05</td>
<td>0.10009</td>
</tr>
<tr>
<td>0.90</td>
<td>456.79</td>
<td>100.81</td>
<td>4251</td>
<td>2.91404E-05</td>
<td>0.10015</td>
</tr>
<tr>
<td>1.68</td>
<td>456.89</td>
<td>100.85</td>
<td>4253</td>
<td>2.91372E-05</td>
<td>0.10016</td>
</tr>
<tr>
<td>3.33</td>
<td>457.47</td>
<td>100.97</td>
<td>4261</td>
<td>2.91280E-05</td>
<td>0.10020</td>
</tr>
</tbody>
</table>

3.0 RESULTS AND DISCUSSION

The results on the identification of the best real gas model are discussed first, followed by a discussion on the numerical simulations of the SH header section with the real gas model.

3.1 Identification of the best EOS

The variation in values of the density (ρ), isobaric specific heat (Cₚ), dynamic viscosity (µ) and thermal conductivity (κ) of steam with respect to the bulk fluid temperature, as predicted by each real gas model under the same conditions as that of Yamagata et al. [24] is depicted in Fig. 3. The experimental data of Lemmon et al. [23] in the respective temperature ranges is also plotted in Fig. 3. The properties predicted by ideal gas EOS, in conjunction with kinetic theory (KT) relationships for dynamic viscosity (µ) and thermal conductivity (κ), as described in section 3.2 are also presented in the Fig. 3 for the comparison purpose. As depicted in Fig. 3(a), the ORKEOS and the ARKEOS models predict steam density (ρ) quite accurately and the accuracy level of the RKSEOS model is
very poor. As depicted in Fig. 3(c) and 3(d), none of the models predict dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$) with a reasonable level of accuracy. It can be seen that all properties of steam in the supercritical condition predicted by the ideal gas law model diverge from experimental values. The ARKEOS and RKSEOS predict isobaric specific heat ($C_P$) values of steam quite close to experimental value, as depicted in Fig. 3(b).

The percentage deviation in mean value of properties as predicted by each model from that of Lemmon et al. [23], in respective temperature ranges is depicted in Table 2. The accuracy level of all models in predictions of dynamic viscosity ($\mu$) and thermal...
conductivity (κ) is extremely poor. Therefore, the relationships used in the model for these properties, i.e. the Sutherland relationship [25] and the Eucken relationship [26] need to be modified. However, the ARKEOS seems to predict the values of density (ρ) and isobaric specific heat (C_P) reasonably accurately with percentage deviation of -2.07 and -8.4, respectively as compared to the mean experimental values in the respective ranges. Therefore, the most accurate EOS to predict density (ρ) and isobaric specific heat (C_P) is the ARKEOS model.

Table 2: Comparison of accuracy level of each model with experimental value

| % Deviation in the mean value of predicted properties from mean experimental values |
|---------------------------------|-----|-----|-----|-----|
| Model  | ρ  | C_P | μ   | κ   |
| ORKEOS | 2.62 | -18.08 | 9.15 | 38.64 |
| ARKEOS | -2.07 | -8.4  | 8.66 | 66.09 |
| RKSEOS | -12.19 | 8.54  | 8.59 | 84.02 |
| Ideal Steam | -38.47 | -62.27 | 19.10 | -19.53 |

3.2 Identification of the best relationships for μ and κ:

Various relationships for dynamic viscosity (μ) and thermal conductivity (κ) are explored to improve the overall accuracy level of the model. For dynamic viscosity (μ), the Cheremisinoff relationship [28], the KT relationship [29] and the International Association for the Properties of Water and Steam (IAPWS) relationship [30] are explored. Similarly, for thermal conductivity (κ), the KT relationship [29] and IAPWS relationship [31] are explored.
The Cheremisinoff relationship [28] for the dynamic viscosity ($\mu$):

$$\mu(T) = 6.3 \times 10^7 \frac{M_W^{0.5} \rho_{C}^{0.6666}}{T_c^{0.1666}} \left( \frac{T_R^{1.5}}{T_R + 0.8} \right),$$  \hspace{1cm} (7)

where $M_W$ is the molecular weight of gas.

The KT relationship for the dynamics viscosity ($\mu$) [29]:

$$\mu = 2.67 \times 10^{-6} \sqrt{T \times M_W} \frac{\sigma^2 \Omega}{\mu_0}, \hspace{1cm} (8)$$

where $\sigma$ and $\Omega$ are kinetic theory parameters.

The IAPWS relationship for dynamic viscosity ($\mu$) [30]:

$$\overline{\mu} = \overline{\mu}_0 (T) \times \overline{\mu}_1 (\overline{T}, \overline{\rho}) \times \overline{\mu}_2 (\overline{T}, \overline{\rho}),$$  \hspace{1cm} (9)

where,

$$\overline{\mu}_0 (T) = \frac{10.0 \sqrt{\overline{T}}}{\sum_{i=0}^{\infty} \overline{T}^i},$$

$$\overline{\mu}_1 (\overline{T}, \overline{\rho}) = \exp \left[ \overline{\rho} \sum_{i=0}^{5} \left( \frac{1}{\overline{T}} - 1 \right)^i \sum_{j=0}^{6} H_{i,j} (\overline{\rho} - 1)^j \right],$$

$$\overline{\mu}_2 (\overline{T}, \overline{\rho}) = 1, \text{ for industrial use},$$

$$\overline{\mu} = \frac{\mu^*}{\mu^*}, \overline{T} = \frac{T^*}{T^*}, \overline{\rho} = \frac{\rho^*}{\rho^*}, \mu^*, T^* \text{ and } \rho^* \text{ are reference values. } H_i \text{ and } H_{i,j} \text{ are coefficients prescribed by the IAPWS.}$$

The KT relationship for the thermal conductivity ($\kappa$) [29]:

$$k = \frac{15}{4} \frac{R}{M_W} \mu \left[ 4 \frac{C_p M_W}{R} + \frac{1}{3} \right]$$  \hspace{1cm} (10)

The IAPWS relationship for the thermal conductivity ($\kappa$) [31]:

$$\overline{k} = \overline{k}_0 (\overline{T}) \times \overline{k}_1 (\overline{T}, \overline{\rho}) + \overline{k}_2 (\overline{T}, \overline{\rho}).$$  \hspace{1cm} (11)
where,

\[
\overline{k}_0 (\bar{T}) = \frac{\sqrt{T}}{\sum_{i=0}^{\infty} L_i}
\]

\[
\overline{k}_1 (\bar{T}, \bar{\rho}) = \exp \left[ \bar{\rho} \sum_{i=0}^{4} \left( \frac{1}{i} - 1 \right) \sum_{j=0}^{5} L_{ij}(\bar{\rho} - 1)^j \right].
\]

\[
\overline{k}_2 (\bar{T}, \bar{\rho}) = 0, \text{ assumed for regions outside critical region.}
\]

$L_i$ and $L_{i,j}$ are coefficients prescribed by IAPWS.

The dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$) values predicted by these modified models are compared with those of Lemmon et al. [23]. It is observed that the inaccuracy in thermal conductivity ($\kappa$) prediction by the relationship of Eucken [26] is improved when the Sutherland relationship [25] for dynamic viscosity ($\mu$) is replaced with Cheremisinoff [28] relationship in the model. However, this led to inaccuracy in dynamic viscosity ($\mu$) prediction. The IAPWS relationships for both dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$) seem to be predicting the properties with greater accuracy. The mean values of dynamic viscosity ($\mu$) and thermal conductivity ($\kappa$) predicted by the IAPWS relationship deviates from the mean values of Lemmon et al. [23] by 1.68 % and -9.56 % respectively.

### 3.3 The most accurate real gas model for supercritical steam:

The model with ARKEOS and IAPWS relationships is the most accurate one in terms of property prediction. However, in the initial phase i.e. for temperature up to 410 °C, the IAPWS relationship slightly under-predicts thermal conductivity ($\kappa$), when compared with experimental values. The $\overline{k}_2$ factor for IAPWS relationships in Eq. (11) represents the enhancement in thermal conductivity in the critical region. Since in the present application...
the critical region is excluded from the calculation, the factor $k_2$ is ignored. This is primarily to achieve better speed of calculation as the associate equations for $k_2$ are quite complex and will require significant time for solving. As a result, there is a discrepancy between the predicted and experimental values of thermal conductivity ($\kappa$), when the bulk fluid temperature is still near the critical point. However, the discrepancy in values is only for a short span and it starts to mitigate once the bulk fluid temperature moves away from the critical point. The comparison of the predicted values of steam properties by the model with IAPWS relationships in conjunction with ARKEOS, with the experimental values of Lemmon et al. [23] is presented in Fig. 4. In Fig. 4, after the bulk fluid temperature crosses a value of 410 °C, the predicted values of thermal conductivity ($\kappa$) come much closer to that of experimental value. This is to be noted that the pseudo-critical temperature of steam at a pressure of 245 bar is approximately 383 °C. The maximum percentage deviation in property predicted by this model from experimental values is of -9.56% and the same is observed in the prediction of thermal conductivity ($\kappa$).
3.4 Simulation of SH header arrangement with real gas model

Using the developed real gas model, simulation of supercritical steam inside the SH section of the industrial boiler is carried out. To assess the impact of inlet and outlet arrangement on flow mal-distribution, three different geometrical variations of the SH header section of the boiler, as discussed in Sec. 2.1 have been modelled. The extent of non-uniformity ($\%ENU$), as defined by Gandhi et al. [2] is considered as the parameter to assess the mal-distribution of steam mass flow in SH header arrangements. The $\%ENU$ is defined as

$$\%ENU = \frac{m_i - \overline{m}}{\overline{m}} \times 100,$$

where $m_i$ is the mass flow received by individual SH tube and $\overline{m}$ is the average value of mass flow received by all tubes. Further, to assess the impact of Reynolds number on flow mal-distribution, three different inlet mass flow rates have been considered in the simulations.

3.4.1 Effect of inlet and outlet arrangement

The $\%ENU$ predicted by CFD analysis of 3 types of arrangements is plotted in Fig. 5. The $\%ENU$ for the ‘type-1’ arrangement varies from -21.9% to +20.1%, indicating that there is a huge mal-distribution among the SH tubes. For the ‘type-2’ arrangement, the $\%ENU$ varies from -5.9% to +5.9%. However, for the ‘type-3’ arrangement the $\%ENU$ variation is the least, it varies from -2.2% to +1.3%. There is a reduction of 19.7% in the maximum value of $\%ENU$ from ‘type-1’ to ‘type-3’ header arrangement. The primary
factor influencing the %ENU in all the arrangements is the resistance to fluid flow. In the ‘type-3’ and ‘type-2’ arrangements, the inlet and outlet pipes face each other thereby providing the least resistance path to steam. The majority of steam tends to take the least resistance path and reach the oppositely positioned outlet pipe by traveling through the SH tubes. Therefore the mass flow rates and %ENU are high in centrally and end positioned SH tubes in ‘type-3’ and ‘type-2’ arrangements respectively. On the other hand, in the case of ‘type-1’ arrangement, the outlet pipe is positioned diagonally opposite to the inlet pipe. There is no least resistance path available to steam near entry point and the tangential velocity inside inlet header makes the steam to travel till the end of the inlet header, where it gets the least resistance path and reaches the outlet pipe by travelling through the SH tubes. Therefore, the SH tubes positioned at the end of the inlet header get the maximum steam mass flow and hence the maximum value of %ENU.

![Figure-5: Comparison of flow mal-distribution for three different types of the SH header arrangements.](image-url)
The contour of steam temperature on a middle plane inside the inlet header is depicted in Fig. 6. The minimum temperature of steam is observed at the inlet for all the arrangements. For ‘type-1’ and ‘type-2’ arrangements, the inlet is at one end of inlet header whereas for ‘type-3’ arrangement, the inlet is at the center of the inlet header. As steam travels inside the header arrangement, it keeps receiving heat from the hot walls and the steam temperature keeps increasing.

Figure-6: Comparison of steam temperature contour on a middle plane inside inlet header for three different types of SH header arrangements namely (a) ‘type-1’, (b) ‘type-2’ and (c) ‘type-3’.
The contours of wall heat flux on the surface of SH tubes and headers, as predicted by CFD analysis for each arrangement is depicted in Fig. 7. For the ‘type-2’ arrangement, the maximum heat flux is absorbed by surfaces of those SH tubes which are present near the steam entry i.e. at inlet end of inlet header. For the ‘type-3’ arrangement, the maximum heat flux is absorbed by surfaces of those SH tubes which are present in the central location. However, in the case of the ‘type-1’ arrangement, the surface heat flux distribution is relatively uniform. The surface heat flux is a direct function of steam mass flow rate and the temperature difference between steam and wall. As depicted in Fig. 5, in case of ‘type-2’ and ‘type-3’ arrangements, the maximum amount of steam mass flows through the SH tubes located at the inlet end and central portion respectively. Additionally, the steam temperature is lowest at the entry point, which is the inlet end and central portion for ‘type-2’ and ‘type-3’ arrangements respectively, as depicted in Fig. 6. Minimum steam temperature corresponds to maximum temperature difference. Therefore, the maximum amount of heat flux is received by SH tubes present at the inlet end and central portion for these cases. On the other hand, for the case of ‘type-1’ arrangement, the maximum amount of steam mass flow is carried by those SH tubes which are present at the outlet end (Fig. 5), whereas the coldest steam is at the inlet end (Fig. 6). Therefore, the surface heat flux distribution is relatively uniform. The total pressure drop of steam for the arrangement of ‘type-1’ is 84251 Pa and that for ‘type-2’ and ‘type-3’ is 83769 Pa and 69869 Pa respectively. There a reduction of 17% in total pressure drop from ‘type-1’ to ‘type-3’ header arrangement.
Figure-7: Comparison of the temperature contour on the SH tube walls for three different types of the SH header arrangements, namely (a) ‘type-1’, (b) ‘type-2’ and (c) ‘type-3’.

The contours of steam density (ρ), isobaric specific heat (Cp), dynamic viscosity (μ) and thermal conductivity (κ) on the middle plane inside the inlet header of ‘type-3’ arrangement, as predicted by the real gas model are depicted in Fig. 8. As shown in Fig. 6(c), the steam temperature reaches its maximum value near both the ends of the inlet header for ‘type-3’ arrangement. For supercritical steam, with increase in bulk temperature value, the values of density (ρ), isobaric specific heat (Cp) and thermal conductivity (κ) keep reducing and the value of dynamic viscosity (μ) keeps increasing (Fig. 3). In Fig.8
also, both the ends of the header witness the minimum values of density ($\rho$), isobaric specific heat ($C_p$) and thermal conductivity ($\kappa$) and maximum value of dynamic viscosity ($\mu$). The distribution of steam properties inside the inlet header is an important information for the designer.

Figure-8: Contour of steam properties at the middle plane inside the inlet header of the SH section of ‘type-3’ arrangement. (a) Density ($\rho$), (b) isobaric specific heat ($C_p$), (c) dynamic viscosity ($\mu$) and (d) thermal conductivity ($\kappa$).
3.4.2 Effect of Reynolds number

The effect of Reynolds number on flow mal-distribution has been assessed by varying the incoming mass flow rate. The ‘type-3’ arrangement is chosen for this study and three different values of Reynolds number are considered, they are 106x10⁵, 63x10⁵ and 147x10⁵. The %ENU predicted for the three values of Reynolds number are plotted in Fig. 9. As depicted in Fig. 9, there is no significant change in %ENU when the Reynolds number value changed from 63x10⁵ to 147x10⁵. Therefore, in the case of supercritical steam the Reynolds number does not influence the %ENU in header arrangements.

![Figure-9: Comparison of flow mal-distribution in header arrangement for three different values of the Reynolds number.](image)

The surface heat flux absorbed by the SH tubes for the three different values of Reynolds number is plotted in Fig. 10. As observed in the case of %ENU, here also the pattern of heat absorption by SH tubes does not change with Reynolds number. However, the absolute value of heat absorption by SH tubes changes with Reynolds number value.
High Reynolds number flow is characterized by a high mass flow rate of steam inside tubes. As discussed earlier, SH tubes with higher mass flow rates absorb more heat as heat absorption is a direct function of mass flow rate. Therefore, in Fig. 10 the flow with the highest value of Reynolds number i.e. $147 \times 10^{6}$ corresponds to the maximum heat absorption by SH tubes. In ‘type-3’ arrangement, since the SH tubes located at the central portion get the maximum value of mass flow (Fig. 5), the peak of surface heat flux in Fig. 10 is observed for centrally located SH tubes.

![Figure-10: Comparison of heat absorbed by the surface of the SH tubes for three different values of the Reynolds number.](image)

**4.0 CONCLUSIONS:**

In the present work, a real gas model is proposed which can be used to simulate the flow of supercritical steam in industrial boilers. The real gas model is a combination of the IAPWS relationships for the dynamic viscosity ($\mu$) and the thermal conductivity ($\kappa$), along with the ARKEOS real gas equation of state. The proposed real gas model can reproduce
the experimental data of Lemmon et al. [23]. Our model provides excellent agreement with
the experimental results when the steam parameters are above the critical point. In the
vicinity of the critical point (i.e. 220 bar pressure and 373 °C for water) as well, the results
are satisfactory in the allowance limit of industrial boilers. Nevertheless, the set of
equations incorporated into the model are generic in nature. Therefore, the model can be
used by any CFD solver, with minor syntax modifications.

Further, using the proposed real gas model, the flow of supercritical steam inside the
SH header arrangement of an industrial boiler is simulated. The effect of various inlet and
outlet arrangements on the flow mal-distribution of steam in SH tubes of the boiler is
assessed. The ‘type-3’ arrangement for the SH header is found to be the best arrangement
in terms of flow mal-distribution. There is a reduction of 19.7% in the maximum value
of %ENU and 17% in total pressure drop when the SH header arrangement changed from
‘type-1’ to ‘type-3’. The effect of Reynolds number on flow mal-distribution in the ‘type-3’
arrangement is observed to be negligible. The absolute value of heat absorption by SH
tubes increases with increase in Reynolds number value. The overall design effectiveness
of the boiler is assessed with the help of the developed real gas model.

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

<table>
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<th>Abbreviation</th>
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<tr>
<td>ARKEOS</td>
<td>Aungier-Redlich-Kwong equation of state</td>
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<td>BHEL</td>
<td>Bharat Heavy Electrical Limited</td>
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<tr>
<td>EOS</td>
<td>Equation of state</td>
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<td>ENU</td>
<td>Extent of Non-Uniformity</td>
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<td>H</td>
<td>Enthalpy</td>
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<td>IAPWS</td>
<td>The International Association for the Properties of Water and Steam</td>
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<tr>
<td>P</td>
<td>Pressure</td>
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<td>R</td>
<td>Universal gas constant</td>
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<td>RHVT</td>
<td>Ranque-Hilsch Vortex Tube</td>
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<td>Volume</td>
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<tr>
<td>Z</td>
<td>Compressibility factor</td>
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<td>c</td>
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<td>n</td>
<td>Critical point compressibility factor</td>
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<tr>
<td>k</td>
<td>Turbulent kinetic energy</td>
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<tr>
<td>ω</td>
<td>Acentric factor</td>
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<td>$P_R$</td>
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APPENDIX A.

Expression for enthalpy as suggested by Aungier [11]:

\[ H = H^0 + PV - RT + \frac{1}{b_0} \left( T \frac{\partial a(T)}{\partial T} - a(T) \right) \ln \left( \frac{V + b_0}{V} \right), \tag{A.1} \]

where \( H^0 (T) \) is the enthalpy function of a thermally perfect gas, which is given by

\[ H^0 (T) = \int_{T_0}^{T} C^0_P (T) dT, \tag{A.2} \]

\( C^0_P \) being the specific heat for a thermally perfect gas. The value of isobaric specific heat \( C_P \) is obtained by differentiating Eq. (5):

\[ C_P = \left( \frac{\partial H}{\partial T} \right)_P. \tag{A.3} \]

The expression for entropy as suggested by Aungier[11]:

\[ S = S^0 (T, P^0) + R \ln \left( \frac{V}{V_0} \frac{V - b_0 + c_0}{V} \right) + \frac{1}{b_0} \frac{\partial a(T)}{\partial T} \ln \left( \frac{V + b_0}{V} \right), \tag{A.4} \]

where \( S^0 (T, P^0) \) is the entropy function of a thermally perfect gas.

The speed of sound in the working fluid is given by [27]

\[ c^2 = \left( \frac{\partial P}{\partial \rho} \right)_S = - \left( \frac{C_P}{C_v} \right) \frac{V^2}{\left( \frac{\partial V}{\partial P} \right)_T}, \tag{A.5} \]
REFERENCES


[27] Ansys fluent theory and user’s guide (Release 15.0, 2013).


