Exergy modeling and simulation of a horizontal concentric tube heat exchanger

Ankit Upadhyay¹, Sankalp Gour²*, Ankul Kumar³, Deepak Kumar², Aseem Chandra Tiwari¹

¹Department of Mechanical Engineering, UIT-Rajiv Gandhi Proudyogiki Vishwavidyalaya Bhopal, Madhya Pradesh, India-462033
²Department of Mechanical Engineering, Maulana Azad National Institute of Technology Bhopal, Madhya Pradesh, India-462003
³E-mail: sankalpgour@gmail.com

Abstract. The present study deals with the exergy modeling and simulation of a horizontal concentric tube heat exchanger. In this context, a fundamental thermodynamics-based exergy model is developed here to predict the irreversibility and entropy generation number for various fluid flow arrangements of a concentric tube heat exchanger. To develop the same, a second law of thermodynamics-based approach is utilized. Additionally, a finite volume method-based simulation of the same heat exchanger is also performed using a computer-aided engineering software named ANSYS R15.0. At last, the analytical and simulation results are compared and validated with experimental results, and a good agreement is achieved.

1. Introduction

In general, the function of heat exchanger is the interaction of heat within the fluids due to change in temperature between them. Usually, they are based on the combination of conduction and convection without the interaction of external heat and work. The system can be taken as an isolated system, for example, Heat exchanger of concentric tube type, (i.e., neglecting the heat interaction and system work within the surrounding) [1]. However, heat transfer involves convection from hot water to an innermost layer of the tube wall, conduction within the tube wall (radially), and the tube wall’s outer shell to the cold fluid through convection. Generally, heat exchangers are employed in engineering processes like the food industry, chemical industry, air conditioning, and power generation [2]. However, due to low energy sources and higher energy costs, many efforts were being put into enhancing the efficiency of heat exchangers. The first step in this process was to measure the fundamentals thermodynamics parameters, which deal with energy, transfer, and conversion of energy and entropy change at equilibrium condition. At the same time, exergy analysis of heat exchangers also became very useful in developing a better understanding of thermal performance than energy analysis [3]. Exergy analysis prevails over the major limitation in analysis of energy that generally ignores the losses and direction for a particular process [4, 5]. This analysis typically includes a system analysis that uses the thermodynamics based second law approach, composed of the principals of conservation mass and energy, to design and analyze energy systems.

To mention some earlier works on the exergy analysis in case of heat exchangers, Moran and Scuibba [6] discussed the importance of exergy analysis to evaluate the true amount (magnitude) of waste and the loss of non-renewable energy resources. In line with that, Ahmetcan and Doganeryener [7] experiment on a double tube type of heat exchanger to
evaluate the significance of process variables on the Irreversibility or exergy losses. They observed that the rise in effectiveness might result in the exergy loss in both the paths at constant temperature ratios. Further, Adrian Bejan [8] suggested the use of maximum exergy for a hot stream is compatible with the integrative theoretical design of energy flow systems in the case of aircraft. Paisarn Naphon [9] performed a study by using thermodynamics based second law approach to evaluate the different heat transfer rates in case of horizontal concentric tube type heat exchanger and validated it with the experimental work. Colorado and Velázquez [10] also performed the exergy modeling on the vapor absorption refrigeration on cascade system and found that, on operating the system at lower irreversibility in the generator as well as an absorber, it may reach the maximum COP and energetic efficiency. Furthermore, Esfahani and Languri [11] performed the same exergy analysis by using graphene oxide nanofluids on a shell & tube type of heat exchanger and stated, On increasing hot fluid inlet temperature increases the exergy loss for turbulent flow & laminar flow regimes respectively.

In view of the simulation works, a CFD simulation is an efficient technique to design a heat exchanger system with all the modifications done in optimization. CFD possesses a fundamental methodology to resolve the complete system into small grids and proceeds with governing expressions on the discrete cells to obtain a numerical solution by validating the experimental work [12, 13]. In line with that, Correa et al. [14] suggested a multicell dynamic model that illustrates the transient behavior of the heat interaction in shell & tube type of heat exchangers. Further, Sunden [15] explained different feasible models that can be applied in the Computational fluid dynamics (CFD) analysis and performed a simulation on a plate type heatexchanger to show the prospect of CFD applications on various types of complications across heat exchangers. Huminic et al. [16] investigated that on using CuO and TiO₂ as nanoparticles with water as a base can increase the heat transfer in case of laminar flow region with increased water outlet temperature. At last, Congedo et al. [17] analyzed the configurations of three different geometry of ground heat exchangers with varying working conditions and learned that the velocity choice for the heat transfer fluid plays a vital role in the heat transfer performance for each arrangement.

2. Theoretical modeling
In this section, an analytical model based on horizontal concentric tube heat exchanger was developed to calculate the exergy loss and rate of entropy generation number. Further, To develop the same, a fundamental theory of heat transfer using thermodynamics based second law approach.

2.1 Problem definition
Consider a complete thermal unit composed of a test section having a concentric tube heat exchanger, refrigerant, cooling unit, heating unit, and data acquisition system shown in Figure 1. The water is considered a working fluid in the unit, and an R22 refrigerant is used to chilling the water. The piping and test connection system are connected for the easy removal or maintainace of parts at any time. The cooling unit includes reservoir tank , a voltage controlled electric heater, a stirrer , and a cooling coil immersed in the reservoir tank. For the working stage of the thermal unit, we may first adjust the temperature of water as required. Then, both the working fluids are pumped out of reservoir and allowed to flow through flow meter, testing unit and then collected back in the reservoir.

Next, we may control the water flow by regulating the valve and measuring the flow meters. Further, the temperature of water at the entry, exit, and mid section of heat exchanger as shown in Figure 1 and by using thermocouple probe we can estimate the inner tube wall temperature.
Figure 1. Schematic diagram of a thermal unit.

For a given working state of a thermal unit, the problem aims to derive an analytical expression of the exergy loss and entropy generation number following the thermodynamics based second law approach. The model is developed under the given assumptions

- Flow of hot fluid and cold fluid are assumed to be steady.
- The heat interaction between system and surrounding is assumed to be negligible.
- The convective heat transfer coefficients for annulus as well as tube sides are considered as constant.
- Thermal conductivity for construction material of the tube is assumed to be constant.

The primary objective of the defined problem is to verify the second law based developed model for concentric tube heat exchanger with simulation findings and existing experimental results.

2.2 Exergy model

For a given horizontal concentric tube type heat exchanger wherein heat interaction take place in both the fluids, the total rate of entropy generation rate $S_{\text{gen}}$ for the given system is defined as

$$S_{\text{gen}} = S_{\text{gen}}^h + S_{\text{gen}}^c,$$

where $S_{\text{gen}}^h$ and $S_{\text{gen}}^c$ denote the rate of entropy generation rates in hot and cold fluid, respectively.

From the theory of heat transfer [18], the above equation (1) may be rewritten as

$$S_{\text{gen}} = \dot{m}_h C_h \ln \left( \frac{\theta_{h,2}}{\theta_{h,1}} \right) + \dot{m}_c C_c \ln \left( \frac{\theta_{c,2}}{\theta_{c,1}} \right),$$

(2)
where \( \dot{m}_h \) represent the mass flow rate for hot-side and \( \dot{m}_c \) represent the mass flow rate for cold-side, \( C_h \) and \( C_c \) represent the hot and cold water specific heat capacities, and \( \theta_{h,2} \) and \( \theta_{h,1} \) are the outlet & inlet hot and cold side water temperatures, respectively.

### 2.2.1 Case-A \([\dot{m}_h C_h < \dot{m}_c C_c]\):

For a given minimum heat capacity rate in case of hot fluid \( C_{\text{min}} = \dot{m}_h C_h \) than the cold fluid \( C_{\text{max}} \), the effectiveness \( \epsilon \) for the heat exchanger is defined as

\[
\epsilon = \frac{\dot{m}_c C_c (\theta_{c,2} - \theta_{c,1})}{C_{\text{min}} (\theta_{h,1} - \theta_{c,1})} = \frac{(\theta_{h,1} - \theta_{h,2})}{(\theta_{h,1} - \theta_{c,1})}
\]  

(3)

On utilizing equation (3) in equation (2), we obtain

\[
\dot{S}_{\text{gen}} = C_{\text{min}} \ln \left[ 1 - \epsilon \left( 1 - \frac{1}{\theta_r} \right) \right] + C_{\text{max}} \ln \left[ 1 + \epsilon R (\theta_r - 1) \right]
\]  

(4)

where \( \theta_r = \frac{\theta_{h,1}}{\theta_{c,1}} \) and \( R = \frac{C_{\text{min}}}{C_{\text{max}}} \) denote the inlet temperature ratio and the heat capacity ratio, respectively. Following the Gouy-Stodola theorem [19, 20], an exergy loss or irreversibility \( I \) for a given working thermal system at a surrounding temperature \( T_0 \) is described by

\[
I = T_0 \dot{S}_{\text{gen}}.
\]  

(5)

Finally, For current case, the entropy generation number \( N_s \), can be written as

\[
N_s = \frac{\dot{S}_{\text{gen}}}{C_{\text{min}}} - \ln \left[ 1 - \epsilon \left( 1 - \frac{1}{\theta_r} \right) \right] + \frac{1}{R} \ln \left[ 1 + \epsilon R (\theta_r - 1) \right]
\]  

(6)

### 2.2.2 Case-B \([\dot{m}_c C_c < \dot{m}_h C_h]\):

For a given minimum heat capacity rate in case of cold fluid \( C_{\text{min}} = \dot{m}_c C_c \) than the hot fluid \( C_{\text{max}} \), the effectiveness \( \epsilon \) for the heat exchanger is defined as

\[
\epsilon = \frac{\dot{m}_h C_h (\theta_{h,2} - \theta_{c,2})}{C_{\text{min}} (\theta_{h,1} - \theta_{c,1})} = \frac{(\theta_{c,2} - \theta_{c,1})}{(\theta_{h,1} - \theta_{c,1})}
\]  

(7)

On utilizing equation (7) in equation (2), we get

\[
\dot{S}_{\text{gen}} = C_{\text{min}} \ln \left[ 1 + \epsilon (\theta_r - 1) \right] + C_{\text{max}} \ln \left[ 1 - \epsilon R \left( 1 - \frac{1}{\theta_r} \right) \right]
\]  

(8)

At last, the entropy generation number \( N_s \) for the current case may be stated as

\[
N_s = \frac{\dot{S}_{\text{gen}}}{C_{\text{min}}} = \ln \left[ 1 + \epsilon (\theta_r - 1) \right] + \frac{1}{R} \ln \left[ 1 - \epsilon R (1 - \frac{1}{\theta_r}) \right]
\]  

(9)
3. Simulation
In this section, a simulational study for a horizontal concentric tube type heat exchanger calculates the exergy loss as well as rate of entropy generation number. A finite volume method-based ANSYS R15.0 software is utilized to simulate the heat exchanger. This methods of simulation based upon computational fluid dynamics is an essential tool to make applications related to fluid flow and heat and mass transfer. The methods used in current work for the flow of working fluid, i.e., water inside a concentric tube heat exchanger with counterflow arrangement. The methods include certain additional assumptions to reduce the complications of calculations. These additional assumptions are

- Water is assumed to be incompressible as well as a Newtonian fluid.
- Steady-state condition is assumed.
- The effect of thermal radiation and gravity are being neglected.

3.1 Geometry of concentric tube heat exchanger
Two concentric parallel tubes are taken in which the hot fluid flows inside the inner tube, and the cold fluid flows into the annulus between the two tubes with the counter flow direction. The specifications of geometry and other details of the simulated heat exchanger are demonstrated in Table 1.

| Dimensions     | Outer tube | Inner tube |
|----------------|------------|------------|
| Tube material  | Copper     | Copper     |
| Outer diameter | 50 mm      | 22 mm      |
| Inner diameter | 44 mm      | 18 mm      |
| Wall thickness | 3 mm       | 2 mm       |
| Tube length    | 2400 mm    | 2400 mm    |

3.2 Simulation methods and boundary conditions
For simulation of the problem, the finite volume method is adopted. A pressure and velocity coupling formulation is utilized by a Semi-Implicit Method for Pressure-Linked Equations algorithm (i.e., SIMPLE algorithm) is utilized. An energy model is kept ON, and the Standard k-epsilon model is used with Standard wall functions. Water is considered as a working fluid for both the hot & cold fluid in material section. The material of the outer and inner tubes is copper. In spatial discretization, for momentum computation, we adopted a second-order upwind scheme, and for evaluation of dissipation rate as well as kinetic energy in turbulent regime, first-order upwind method is used. The Hybrid-Initialization solution method is applied before running calculations. The considered number of nodes and elements information is shown in Table 2. The entry and exit boundary constraints are applied as the mass flow at entry and pressure at exit for the inner tube and the annulus side. It can be seen in Figure 2 containing temperature variation contours at the outlet sections of both the tubes.

Various data like an inlet temperature of the hot fluid $\theta_{h,1} = 40 ^\circ C$ and an inlet temperature of the cold fluid $\theta_{c,1} = 20 ^\circ C$ are noted for a constant mass flow rates of cold $m_c = 0.03$ kg/s and hot $m_h = 0.0329$ kg/s fluids, respectively. Similar procedures are being adopted for all the parameters used in other conditions to get all the simulated plots of exergy loss and rate of entropy generation number.
Figure 2. Temperature contours at inner tube outlet (hot fluid) and outer tube outlet (cold fluid) with $\theta_{h,1} = 40 \, ^\circ\text{C}$, $\theta_{c,1} = 20 \, ^\circ\text{C}$, $m_h = 0.0329 \, \text{kg/s}$, and $m_c = 0.03 \, \text{kg/s}$.

Table 2. Nodes and elements specifications of concentric tube heat exchanger.

| Domains          | Nodes | Elements |
|------------------|-------|----------|
| Cold fluid       | 42688 | 31947    |
| Hot fluid        | 41064 | 33360    |
| Inner tube       | 31556 | 19180    |
| Outer tube       | 12768 | 6666     |
| All Domains      | 128076| 91153    |

4. Results and discussion

This section presents the analytical findings of the derived model and simulation results of ANSYS R15.0 software are analyzed and compared with the existing parallel experimental results [9] in the literature. In this context, Figures 3 and 4 illustrate the relationship between the rate of entropy generation and $m_h$ at different inlet conditions. On increasing $m_h$, we may observe the growth in the rate of entropy generation as expected and described via equation (4) or (8). The mass flow rate of hot water $m_h C_h$ is having higher heat capacity rate than that of cold water $m_c C_c$. Hence, the heat capacity in case of cold water is minimum, shown in equations (4) or (8).
Interestingly, a drop in this trend is observed at a higher value of the $\dot{m}_h$. Also, $\dot{S}_{gen}$ peaks constantly on the increase in the temperature at entry of hot water at steady temperature conditions for the cold water as well as the given $\dot{m}_h$. This observation shows that the $\dot{m}_h$ plays a fundamental role in the intensification of the rate of entropy generation. A similar observation may be noticed from figure 4. However, this grows closer for a lower value of the $\dot{m}_h$. One may also note that the theoretical and simulation findings agree well with the experimental data [9].
Figure 5 depicts the impact of hot water inlet temperature on the entropy generation number obtained from equations (6) or (9). Herein, increment in $N_s$ is observed on increasing $m_h$. However, the effect is more significant with higher inlet temperature for the same value of $m_h$. According to the anticipated and experimental results, the theoretical and simulation findings somewhat here overpredict the measured results.

The consequence of the $m_h$ on $N_s$ is presented in Figure 6. One may here observe that for $m_c = 0.03$ kg/s, with increment in $m_h$ we observe a continuous rise in the rate of entropy generation number. This phenomenon is interpreted by equations (6) or (9) that recommends, the rate of heat capacity for cold water is minimum. It means that with increment in $m_h$, the $N_s$ also rises. However in case of $m_c = 0.07$ kg/s, $m_h$ is lower than that of 0.07 kg/s. Hence, the $C_h$ is used in place of the minimum capacity rate. Therefore, one may identify a decreasing trend in the $N_s$ concerning the $m_h$. On the other hand, we recognize an increasing inclination of entropy generation number in regards to the $m_h$ beyond $m_h = 0.07$ kg/s. This behavior is obtained due to the change of the $C_{min}$ to $C_h$. Similarly, entropy generation number($N_s$) drops continuously at $m_c = 0.10$ kg/s with increase in $m_h$. Again, we can understand this phenomenon by help of equations (6) and (9), which show the minimum heat capacity rate is used as $C_h$. Hence, a decrease in $N_s$ is observed as increase in $m_c$.

From both of the figures, the theoretical and simulation findings now reasonably validate the experimental results [9].

**Figure 5:** Effect of hot water mass flow rate ($m_h$) on entropy generation number($N_s$) with varying inlet hot water temperatures.
Figures 7 and 8 depicts the dependence of the exergy loss on $m_h$ for different entrytemperature conditions for hot water with specific values of mass flow rate for the cold water, respectively. One may observe that the nature of the curves for the $I$ is identical to the $S_{gen}$, as displayed in Figures 3 and 4. Therefore, a similar description and interpretation may be followed for Figures 7 and 8 as well. Repeatedly, for both figures 7 and 8, the theoretical and simulation outcomes show a reasonable agreement with the experimental results [9].

**Figure 6.** Effect of hot water mass flow rate ($m_h$) on entropy generation number $N_s$ with varying cold water mass flow rates ($m_c$).

**Figure 7.** Effect of hot water mass flow rate ($m_h$) on exergy loss/irreversibility ($I$) with varying inlet hot water temperatures.
5. Concluding remarks

In the present work, an extensive study on the heat exchanger for horizontal concentric tube type has been performed following the thermodynamics based second law approach. The effect of exergy loss and entropy generation number is modeled and simulated for varying several parameters such as mass flow rates \( \dot{m}_h \) and inlet conditions for hot and cold water. The impact of these parameters upon the concentric tube heat exchanger design has been observed and analyzed. The theoretical and simulation findings of the current study are also successfully validated with the experimental results. Hence, the findings of the present work may help the researchers in designing the concentric tube type of other heat exchangers for various applications.

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