Exergy analysis of double tube heat exchanger for parallel flow arrangement.

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Abstract. This paper presents the analysis of exergy according to second law of thermodynamics for a double concentric tube heat exchanger experimentally. For this purpose to study the heat exchange process between two fluid mediums separated by a solid partition for parallel flow arrangement, an experimental setup was fabricated as per designed considered to measured data. In this considered double concentric tube heat exchanger, hot water and cold water flowing through inner and outer tube respectively and heat exchange takes place from hot fluid medium to cold fluid medium. Here the wall tube conductive resistance is assumed as negligible. The investigation was performed for mass flow rate of cold water ranging from 0.0326 kg/s to 0.572 kg/s and mass flow rate of hot water ranging from 0.016 kg/s to 0.479 kg/s. The effects of various parameters such as rate of entropy generation, number of entropy generation, effectiveness of heat exchanger, and losses of exergy are observed by exergy analysis. The governing equations have been employed to solve analytically for obtaining the energy and exergy analysis parameters. It is expected that this study might be helpful for further related work.

1. Introduction
The fundamentals of thermodynamics deal with of energy, transfer of energy, and entropy change, at equilibrium condition. First law of thermodynamics represents conservation of energy. It does not provide idea about losses and place of losses and it also does not give any direction for a particular process. The limitations of first law are overcome by the second law of thermodynamics which is qualitative approach and deals with energy degradation, entropy generation, and exergy losses during a process. Exergy is the maximum useful work under the given environmental condition and it is the tool which indicates how far the system departs from equilibrium state. Exergy is mainly affected by the state of the system and environment under consideration. For concentric tube in which one is stationary and other is rotating, the different flow regimes inside the annulus have been presented [1]. The flow regimes at high ranges of rotation of both the cylinders have been explained and also examine the transition between the different flow regimes [2]. The experiments for double pipe heat exchanger by varying width of twisted tapes has been carried out and observed that heat transfer is more compared to plain tube [3]. The steady state analysis of fluid flow and heat transfer through numerical calculations in a zero boil off cryogenic liquid hydrogen tank has investigated [4] and the transient analysis of fluid flow and heat transfer through numerical calculations in a zero boil off
cryogenic liquid hydrogen tank has investigated [5]. The entropy generation and losses of exergy investigated experimentally and theoretically for a micro fin concentric tube heat exchanger placed horizontally by central finite difference approach [6]. The rate of heat transfer, friction factor and loss of exergy is affected by different swirl generators having different number of circular holes and also affected by diameter of inner and outer tube of heat exchanger. They have found experimentally that the rate of heat transfer increases directly with number of circular holes and inversely with diameter [7]. The second law analysis of heat transfer and fluid flow mechanism for steady incompressible fluid inside the two parallel plates of circular cross section has been studied [8]. The performance of heat exchanger in the thermodynamic vent system (TVS), based on the ideal gas state equation and energy conservation equation studied for further improvement [9]. In case of turbulent flow in a smooth pipe subjected to constant heat flux, the generation of entropy and power for pumping the fluid has been investigated [10]. The exergy analysis based on second law for the air conditioning system has been studied theoretically and experimentally [11].

2. Experimental set up and procedures.
The schematic configuration of used experimental setup is shown in figure 1. It comprises of a test section, tank for cold water and hot water, temperature sensors, flow control valve, temperature display, pump, thermostatic water heater.

The glass wool as insulating material was provided on the outer side of tube of test section to prevent loss of heat to the surrounding. All the components of the test section and connections are designed in such a way that they can be easily assembled, disassembled and repaired. Table 1 indicates the details of test section.

| Table 1. Details of test section. |
|----------------------------------|
| Dimensions | Outer tube | Inner tube |
|------------|------------|------------|
| Outer diameter | 30 mm | 18 mm |
| Inner diameter | 26 mm | 15 mm |
| Wall thickness | 4 mm | 3 mm |
| Length in | 3.6 m | 3.6 m |
| Materials used | Stainless steel | Copper |
To measure flow rate of hot and cold water flow meters are separately used. The thermocouples were used to measure the water temperature at different section of test specimen such as at the inlet, outlet, and middle section. Experiments were performed by varying the temperature and mass flow rate of hot and cold water at inlet to the test specimen. The temperatures of hot and cold water at the inlet were varied by the help of electric heater equipped with temperature controller. The data were recorded after the system reached at the steady state.

3. Mathematical modelling.
Some principle governing equations based on first law analysis were used for calculating heat transfer rate, heat transfer coefficient.

3.1. The second law (Exergy) Analysis
The rate of entropy generation from the system can be written as

\[ S_{gen} = (mc_p)_h \ln \left( \frac{T_{ho}}{T_{hi}} \right) + (mc_p)_c \ln \left( \frac{T_{co}}{T_{ci}} \right) \]

(1)

3.1.1. Case (A) When \((mc_p)_h\) is less than \((mc_p)_c\) then effectiveness of heat exchanger may be as follows:

\[ \mathcal{E} = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}} \]

(2)

Putting the value of Eq. (2) in Eq. (1), we get the rate Entropy generation as,

\[ \dot{S}_{gen} = C_{\min} \ln \left[ 1 - \mathcal{E} \left( 1 - \frac{1}{T_R} \right) \right] + C_{\max} \ln \left[ 1 + \mathcal{E} \mathcal{C}_r \left( T_R - 1 \right) \right] \]

(3)

Number of Entropy generation, \(N_s\) can be given as,

\[ N_s = \frac{\dot{S}_{gen}}{C_{\max}} = \ln \left[ 1 - \mathcal{E} \left( 1 - \frac{1}{T_R} \right) \right] + \frac{1}{\mathcal{C}_r} \ln \left[ 1 + \mathcal{E} \mathcal{C}_r \left( T_R - 1 \right) \right] \]

(4)

3.1.2. Case (B) When \((mc_p)_c\) is less than \((mc_p)_h\), then effectiveness of heat exchanger may be as follows:

\[ \mathcal{E} = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}} \]

(5)

Putting the value of Eq. (5) into Eq. (1), and we get the rate Entropy generation as,

\[ \dot{S}_{gen} = C_{\min} \ln \left[ 1 + \mathcal{E} \left( T_R - 1 \right) \right] + C_{\max} \ln \left[ 1 - \mathcal{E} \mathcal{C}_r \left( 1 - \frac{1}{T_R} \right) \right] \]

(6)

Number Entropy generation, \(N_s\) can be given as

\[ N_s = \frac{\dot{S}_{gen}}{C_{\max}} = \ln \left[ 1 + \mathcal{E} \left( T_R - 1 \right) \right] + \frac{1}{\mathcal{C}_r} \ln \left[ 1 - \mathcal{E} \mathcal{C}_r \left( 1 - \frac{1}{T_R} \right) \right] \]

(7)

Now for both the cases the exergy losses may be calculated by using respective entropy generation rate in the following equation as

\[ I = T_p \dot{S}_{gen} \]

(8)
Here $T_o$ is the temperature of surrounding.

4. Result and discussion

In this study, number of experimental tests was carried out for heat exchangers with water as working fluid. The experiments were performed for parallel flow arrangement, for this purpose experimental readings have been taken. The unknown properties of the system have solved by using the equations given in the previous section (Eqn. 1 to 8). The coefficients of heat transfer for tube and shell side were calculated by using the correlations given by Gnielinski [12].

The experiments were carried out with two cases, in the first case keeping the hot fluid mass flow rate constant while varying the cold fluid mass flow rate. Where as in the second case keeping the cold fluid mass flow rate constant while varying the hot fluid mass flow rate. Table 2 indicates the ranges of operating parameters for both the cases.

| Parameters                     | Case-I         | Case-II        |
|--------------------------------|----------------|----------------|
| Inner tube water flow rate     | 0.032 kg/s     | 0.016-0.04 kg/s|
| Outer tube water flow rate     | 0.032-0.057 kg/s | 0.032 kg/s    |
| Inner tube inlet temperature   | 338-343 K      | 333-351 K      |
| Inner tube outlet temperature  | 335-330 K      | 324-342 K      |
| Outer tube inlet temperature   | 308 K          | 308 K          |
| Outer tube outlet temperature  | 317-320 K      | 313-321 K      |
| Heat transfer                  | 1.41 kW        | 1.3 Kw         |
| Entropy generation             | 0.0345 kW/°C   | 0.014 kW/°C    |
| Entropy generation number      | 0.25           | 0.10           |
| Exergy loss                    | 1.19 kW        | 0.5 kW         |

For the analysis of this study the different parameters have been calculated for each of the above cases and their graph depicted as follows.

Figure 2 Mean heat transfer rate variation with No of readings of both the cases
In the figure 2 which indicate the variations of the mean heat transfer rate with number of readings for case-I and case-II. As we know that, the rate of heat transfer varies with mass flow rate of hot water and temperature difference between the fluids flow. Thus for the case-II mean heat transfer rate is increases. Therefore the figure shows that the results are under acceptable value.

![Figure 3 Rate of generation Variation with No of readings of both the cases](image)

From the figure 3 that indicates the Entropy generation with No of Reading. It has been found that Entropy generation of case-I is higher to the Entropy generation of the case-II.

![Figure 4 Entropy generation number with No of readings of both the cases](image)

The figure 4 indicates the Number of entropy generation with No of Reading. Number of entropy generation may be evaluated by the use of either Eq. (4) or Eq. (7). It has been found that the growth of number of entropy generation for case-I is rapidly becomes higher than the Entropy generation number of the case-II as mass flow rate increases.

In the figure 5 which depicts the loss of exergy versus number of readings for both cases. As per obtained results the nature of the plot for the loss of exergy are similar to that of rate of entropy generation plot as shown in figure 3.
5. Conclusion
In the present investigation for Horizontal Double tube Heat exchanger the exergy analysis has been presented. The average increments in entropy generations, entropy generation numbers and Exergy losses are 0.02 kW/°C, 0.15, and 0.69 kW respectively in case-I as compared to case-II.

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