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Heat Transfer and Fluid Flow in a Double Pipe Heat Exchanger, Part I: Experimental Investigation

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***Abstract* - Double pipe heat exchangers have assumed importance over the years owing to their simple construction, ease of maintenance and cleaning, and extensive use in applications involving sensible heating or cooling of process fluids. In this present study, a detailed comparison is made between the theoretical results and experimental data of the performance parameters of the heat exchanger. Experiments were performed for 60 ≤ Re ≤ 240 for two different inlet temperatures of the hot fluid, 50°C, and 70°C keeping the inlet temperature of the cold fluid constant at 31°C. Graphs were plotted between various performance parameters such as overall heat transfer coefficient, effectiveness, NTU, outlet temperatures of the hot and cold streams against mass flow rates of the fluid. Lastly, a comparison was done between the theoretical data and experimental results and they showed good accordance with a mean deviation of 10-12%.**
***Keywords*: Double pipe heat exchanger, heat transfer, effectiveness.**

NOMENCLATURE

m_h	mass flow rate of cold fluid	U	Overall Heat Transfer Coefficient [W/m^2-K]
m_c	mass flow rate of hot fluid	F	Correction Factor
C_{max}	Maximum heat capacity of fluid [W/K]	C_p	Specific Heat of Fluid [J-kg-K]
C_{min}	Minimum heat capacity of fluid [W/K]	Q	Heat Transfer Rate [W]
d_i	Inner Diameter of Inner Tube [mm]	h	Heat transfer coefficient [W/m^2-K]
d_o	Outer Diameter of Inner Tube [mm]	Superscripts and subscripts	
D_i	Inner Diameter of Outer Tube [mm]	c	Cold stream
D_o	Outer Diameter of Outer Tube [mm]	h	Hot stream
R_t	Total Thermal Resistance [K/W]	i	Inlet
U	Overall Heat Transfer Coefficient [W/m^2-K]	o	Outlet
T	Temperature [K]	b	Bulk
NTU	Number of Transfer Units	Greeks and symbols	
t	Thickness of tube [mm]	ε	Effectiveness
h	Heat transfer coefficient [W/m^2-K]	ρ	Density (kg/m^3)
R_t	Total Thermal Resistance [K/W]	μ	dynamic viscosity [kg/m s]

I. INTRODUCTION

Heat exchangers are the devices that provide the transfer of thermal energy between two or more fluids at different Temperatures. They find applications in power production, chemical and food industries, environmental engineering and waste heat recovery. Perhaps, the simplest of these heat exchangers is the double pipe heat exchanger. Advantages include ease of cleaning and maintenance and usage under severe fouling conditions. Fluids at high pressure can also be used. Limitations include difficulty in the cleaning of tubes due to fouling and a shell and tube heat exchanger of the same design is a better way for heat transfer. This present study will investigate the comparison of theoretical results with experimental data on a double pipe heat exchanger. Many researchers have investigated the thermo hydraulic performance of the double pipe heat exchanger which is presented in this section. Sivalakshmi *et al* [5]. analysed the effect of helical fins on the performance of a double pipe heat exchanger. The performance in terms of average heat transfer rate, heat transfer coefficient, and effectiveness of heat exchanger in a plain inner pipe is evaluated and compared with helical fins installed heat exchanger over the inner pipe. The average rate of heat transfer and effectiveness are increased to 38.46% and 35% respectively at a higher flow rate. Wang et al [6]. conducted experiments on coupled convection heat transfer of water in a double pipe heat exchanger at supercritical pressure. It was observed that the heat transfer coefficient of the outer pipe increases with the increase of water temperature at the inlet of the inner pipe when the mean bulk temperature is lower than the pseudo-critical temperature, due to the buoyancy effect decreases at the same time. A new heat transfer empirical correlation was proposed, which includes 94.01% of experimental data are within ±25% error bars. Ghani *et al* [7]. proposed the use of double pipe heat exchangers to meet the requirement of today's HVAC industry. Through experimental investigation, it was found that the double pipe system decreased the compressor work by 34-53% and the operating current by 16-30%. The COP of the system was increased by 52.5% while the double-pipe being sensitive to the evaporator working fluid flowrate. Salem *et al* [8]. experimentally examined the hydrothermal performance of a double pipe heat exchanger without helical tape insert. Compared with plain annulus-case, the results showed that using the HTI increases both annulus average Nu and f, with average increases of 69.4–164.4% and 48.6–113.1%. Correlations for Nu and f were proposed with HTI in the annuli as a function of the investigated parameters. Majidi *et al* [9]. experimentally studied the heat transfer of air in a double-pipe helical heat exchanger. To augment the heat transfer rate, a copper-wire fin is soldered on the outside area of the internal tube. The effect of different streams was found on the overall heat transfer coefficient and it was found out that the cold stream has a stronger influence on the overall heat transfer coefficient because the wire fin is installed on the annulus of the tube. Ali *et al* [10]. tried to enhance the heat transfer of a double pipe using rotating of variable eccentricity inner pipe. The results showed a significant enhancement by 223% in the heat transfer rate due to the eccentricity change up to 40 mm and at the inner pipe rotation of 500 rpm. However, the pressure penalty through the heat exchanger increase by 53% in the rotational speed of inner pipe up to 500 rpm and eccentricity of 40 mm and it is accepted penalties. *Yassin et al* [11]. discussed the thermal characteristics for the flow through the annulus with an inner finned pipe under stationary and rotating conditions. The results showed an increase of the heat transfer coefficient over six times for the height of 30 mm and rotational speeds of 400 rpm at Re = 60000 compared to the plain stationary pipe case. The rotational speed revealed no effect on the efficiency and effectiveness of the fin. Salem *et al* [12]. experimentally investigated the thermal performance of a double pipe heat exchanger with segmental perforated baffles. Experiments were performed for $1380 \leq Re \leq 5700$. It was found out that increasing SSPBs holes spacing and void ratios and inclination angle and decreasing single segmental perforated baffle's cut and pitch ratios increase the annulus average Nuas well as f. The thermal performance index (TPI) is calculated to compare the thermal performance of perforated baffled double pipe heat exchangers with un-baffled ones. Iqbal *et al* [13]. devised a process to optimize convective heat transfer in the double pipe using parabolic fins. A comparison was done with the same model without different configurations of fins such as trapezoidal and triangular. It was later concluded that no particular fin was best and it depended on the situation for which fin to choose. Syed *et al* [14]. studied friction factor and Nu against variations in diameters of inner and outer tubes.

The minimum and maximum increase in the product of friction factor and Reynolds number relative to the finless geometry is more than one time and more than 40 times respectively while the gain in the relative value of Nusselt number lies in the range 1–177. This provides evidence of more than four times enhancement in the heat transfer coefficient relative to that in the pressure loss as a result of extended fin surfaces. Zhang *et al* [15]. studied fluid flow characteristics for the shell side of a double-pipe heat exchanger with helical fins and pin fins. The results showed that the shell side only with helical fins at a large pitch, a pair of vortices near the upper and lower edge of the rectangular cross-section; the weakest secondary flow occurred at the center. By pin fins being installed, the three-dimensional velocity components in the helical channel were strongly changed. The extra two vortices emerged at the center of the cross-sections. Zhang *et al* [16]. studied the thermo hydraulic characteristics of turbulent flow in plain tube equipped with rotor-assembled strands of different geometries and leads. Experimental results revealed that rotor-assembled strands of different geometries employed improved heat transfer significantly with Nusselt number increased by 71.5–123.1% and friction factor increased by 37.4–74.8% compared with the plain tube. The best thermal performance factor was achieved when a helical blade rotor with ladders was manufactured with certain parameters (rotor diameter is 22 mm, rotor lead is 150 mm). Targui and Kahalerras [17]. analyzed the performance of a double pipe heat exchanger by use of porous baffles and pulsating flow.

The effects of the amplitude and frequency of pulsation, as well as the porous baffles permeability on the flow structure and the heat exchanger efficiency, are analyzed. The results reveal that the addition of an oscillating component to the mean flow affects the flow structure, and enhances the heat transfer in comparison to the steady non-pulsating flow. The highest heat exchanger performance is obtained when only the flow of the hot fluid is pulsating. Dizaji *et al* [18] experimentally studied heat transfer and pressure drop characteristics for new arrangements of corrugated tubes in a double pipe heat exchanger. Experiments were performed over $3500 \leq Re \leq 18{,}000$, based on the hydraulic diameter of the annular space between the two tubes. Findings indicated that the outer tube corrugations and arrangement type of corrugated tubes have a significant effect on thermal and frictional characteristics. Maximum effectiveness was obtained for heat exchanger made of the concave corrugated outer tube and convex corrugated inner tube. Corcoles *et al* [19]. numerically and experimentally studied the heat transfer process in a double pipe heat exchanger with inner corrugated tubes.

The highest corrugation height (H/D=0.05) and the lowest helical pitch (P/D=0.682) presented the highest pressure drops in both inner and annular tubes, being 4.15 and 1.27 times higher in the inner tube and the annulus side than in the smooth tube, respectively. The smallest helical pitch and an intermediate corrugation height (H/D=0.041) obtained the highest number of transfer units (NTU) value, which, under the experimental conditions of this work, resulted in an increase of 29% compared with the smooth tube.

II. GEOMETRY

TABLE 1 THE BELOW TABLE OUTLINES THE GEOMETRICAL DIMENSIONS OF THE DOUBLE PIPE HEAT EXCHANGER

Parameters	Magnitude (Unit)/ Description
Inner Diameter of Inner Tube	26 mm
Outer Diameter of Inner Tube	34 mm
Thickness of Inner Tube	4 mm
Inner Diameter of Outer Tube	68 mm
Outer Diameter of Outer Tube	76 mm
Thickness of Outer Tube	4 mm
Length of inner and outer tube	1200 mm
Material of Inner and Outer Tube	Galvanized Iron

Fig. 1 Experimental setup of the double pipe heat exchanger

III. MATERIALS AND METHODOLOGY

A. Theoretical Methodology

The goal of heat exchanger design is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing.

We will begin first, by discussing the basic principles of heat transfer for a heat exchanger. We may write the enthalpy balance on either fluid stream to give:

$$Q_c = m_c(h_{c2} - h_{c1})$$
$$Q_h = m_h(h_{h1} - h_{h2})$$

For constant specific heats with no change of phase, we may also write

$$Q_c = m_c c_{pc}(T_{c2} - T_{c1})$$

$$Q_h = m_h c_{ph}(T_{h1} - T_{h2})$$

There are two methods for the analysis of heat exchangers. One is known as the Logarithmic Mean Temperature Difference Method and the other is Effectiveness- Number of Transfer Units Method.

In the LMTD Method, all the temperatures such as cold fluid inlet, cold fluid outlet, hot fluid inlet, hot fluid outlet are known. With the help of these temperatures, total heat transferred between the two fluids is calculated.

In the second method, the inlet temperatures of the hot and cold fluid are known and the outlet temperatures of the hot and cold fluid are found out. After calculating the unknown temperatures, the total heat transferred is calculated based on enthalpy balance.

In our study we have taken the inlet temperatures of the hot fluid at two different points- 50 degrees and 70 degrees and then we have varied the mass flow rates of the hot fluid and cold fluid.

We have then compared the results of the experiment and theory.

B. Logarithmic Mean Temperature Difference Method (LMTD Method)

Logarithmic Mean Temperature Difference is defining as the temperature difference which, if constant, would give the same rate of heat transfer as occurs under variable conditions of temperature difference.

This method is used when both the inlet and outlet conditions are specified.

C. Assumptions

1. The overall heat transfer coefficient U is constant.
2. The flow conditions are steady.
3. The specific heats and mass flow rates of both fluids are constant.
4. There is no loss of heat to the surroundings, due to the heat exchanger being perfectly insulated
5. There is no change of phase either of the fluid during the heat transfer.
6. The changes in potential and kinetic energies are negligible.
7. Axial conduction along the tubes of the heat exchanger is negligible.

The logarithmic mean temperature difference is derived in all basic heat transfer texts. It may be written for a parallel flow or counter flow arrangement. The LMTD has the form:

Where Δ T1 and ΔT2 represent the temperature difference at each end of the heat exchanger, whether parallel or counter flow. The LMTD expression assumes that the overall heat transfer coefficient is constant along the entire flow length of the heat exchanger. If it is not, then an incremental analysis of the heat exchanger is required. The heat transfer rate for a Cross flow heat exchanger may be written as,

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$

$$Q = FUA\Delta T_{LMTD}$$

D. Correction Factor (F)

The expression $\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$, for LMTD is essentially Valid for single-pass heat exchangers. The analytical treatment of multiple pass shell and tube heat exchangers and cross flow heat exchangers is much more difficult than single-pass cases such cases may be analysed by, $Q = FUA\Delta T_{LMTD}$, where F is the correction factor.

Correction factor charts have been published in the form of graphs in various journals.

Temperature ratio, P is defined as the ratio of the rise in temperature of the cold fluid to the difference in the inlet temperatures of the two fluids. Thus, P is given by,

$$P = \frac{(T_{c2} - T_{c1})}{(T_{h1} - T_{c1})}$$

Where subscripts h and c denote hot and cold fluids respectively.

The temperature ratio P indicates cooling or heating effectiveness and it can vary from zero for a constant temperature of one of the fluids to unity for the case when the inlet temperature of the fluid equals the outlet temperature of the cold fluid.

Capacity ratio, R is defined as the ratio of the products the mass rate times the heat capacity of the fluids.

$$R = \frac{m_c c_{pc}}{m_h c_{ph}}$$

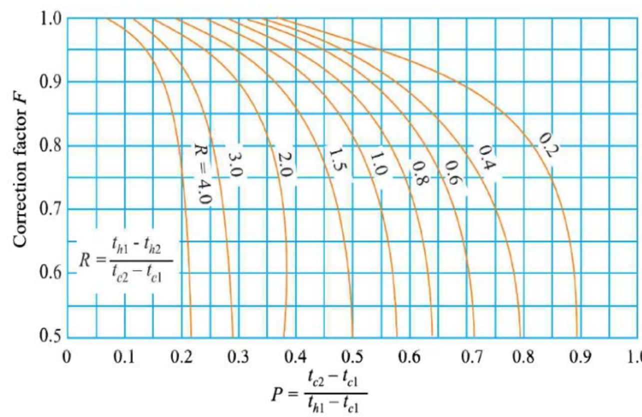


Fig. 2 Correction Factor Plot for Heat Exchanger with One Shell Pass and two, four, or any multiple of tube passes

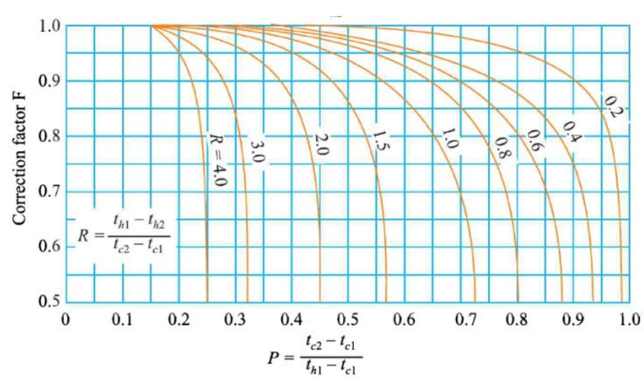


Fig. 3Correction Factor Plot for Heat Exchanger with Two Shell Passes and two, four, eight, or any multiple of tube passes

E. NTU Method (Effectiveness- Number of Transfer Units Method)

Effectiveness- NTU Method: The effectiveness or number of transfer unit methods was developed to simplify several heat exchange design problems. Effectiveness is defined as the ratio of actual transfer rate to the maximum possible heat transfer rate if there were infinite surface area. Heat exchanger effectiveness depends on whether the hot fluid or cold fluid is minimum fluid. This method is employed when the inlet temperatures of the hot and cold fluid are known and the outlet conditions are not known.
The effectiveness or number of transfer unit methods was developed to simplify a number of heat exchange design problems. Effectiveness is defined as the ratio of actual transfer rate to the maximum possible heat transfer rate if there were infinite surface area. Heat exchanger effectiveness depends on whether the hot fluid or cold fluid is minimum fluid.

If the cold fluid is the minimum fluid, then the effectiveness is defined as

$$\varepsilon = \frac{C_h(T_{h1} - T_{h2})}{C_{min}(T_{h1} - T_{c1})}$$

And if the hot fluid is minimum then effectiveness is defined as

$$\varepsilon = \frac{C_c(T_{c2} - T_{c1})}{C_{min}(T_{h1} - T_{c1})}$$

Thus, we may now define the heat transfer rate as

$$Q = \varepsilon C_{min}(T_{h1} - T_{c1})$$

It is now possible to develop expressions that relate the heat exchanger effectiveness to another parameter referred to as the number of transfer units (NTU). The value of NTU is defined as:

$NTU = \frac{UA}{C_{min}}$For convenience the Effectiveness −NTU relationships are given for a simple double pipe heat exchanger for parallel flow and counter flow:

F. Parallel Flow

$$\varepsilon = \frac{1 - exp[-NTU(1 + C_r)]}{1 + C_r}$$

$$NTU = \frac{-ln[1 - \varepsilon(1 + C_r)]}{1 + C_r}$$

G. Counter Flow

$$\varepsilon = \frac{1 - exp[-NTU(1 - C_r)]}{1 + C_r exp[-NTU(1 - C_r)]}, when C_r < 1$$

$$\varepsilon = \frac{NTU}{1 + NTU}, when C_r = 1$$

H. Experimental Methodology

1. Start the flow on the hot side.
2. Start the flow through the annulus & the exchanger as a parallel flow unit.
3. Turn on the geyser.
4. Adjust the flow rate on the hot side between the rates of 1.5 - 4 lpm.
5. Adjust the flow rate on the cold side between ranges of 3 - 8 lpm.
6. Keeping the flow rate some wait till the steady-state conditions are reached.
7. Record the temperature on hot water & cold-water side & also the flow rates.
8. Repeat the experiment with a counter under identical flow conditions.

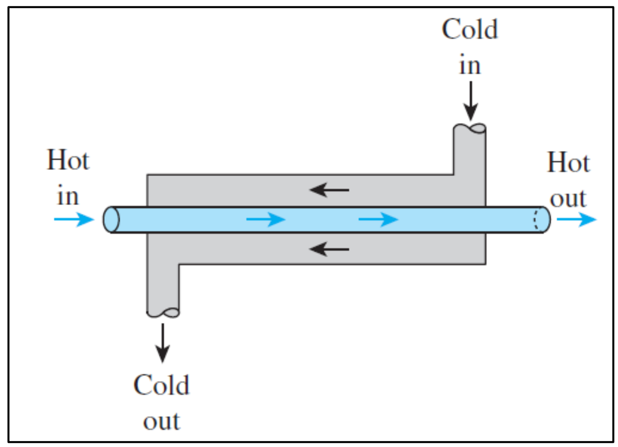


Fig. 4 Counter flow arrangement in a double pipe heat exchanger [4]

I. Experimental Matrix

TABLE IITHE FOLLOWING TABLE INDICATES THE EXPERIMENTAL MATRIX DEVELOPED FOR DIFFERENT INLET, FLOW, AND TEMPERATURE CONDITIONS

Q hot	(lpm)	0.5							
Q cold		0.5	1	1.5	2	0.5	1	1.5	2
Inlet Temperature	Thi	50				70			
	Tci	34	33	32	31	34	33	32	31

Q hot	(lpm)	1							
Q cold		0.5	1	1.5	2	0.5	1	1.5	2
Inlet Temperature	Thi	50				70			
	Tci	31	32	31	31	31	31	31	31

Q hot	(lpm)	1.5							
Q cold		0.5	1	1.5	2	0.5	1	1.5	2
Inlet Temperature	Thi	50				70			
	Tci	31	31	31	31	31	31	31	31

Q hot	(lpm)	2							
Q cold		0.5	1	1.5	2	0.5	1	1.5	2
Inlet Temperature	Thi	50				70			
	Tci	31	31	31	31	31	31	31	31

IV. RESULTS AND DISCUSSIONS

Temperatures of the hot and cold fluid are measured at the inlet and outlet of the double pipe heat exchanger under different inlet temperatures of the hot fluid such as 50 degrees and 70 degrees and the constant inlet temperature of the cold fluid being 31 degrees. Taking these temperature conditions, the flow rates of then the hot and cold fluid were varied and its effect on the performance parameters of the heat exchanger was observed.

A. Variation of performance parameters for different cold fluid flow rates

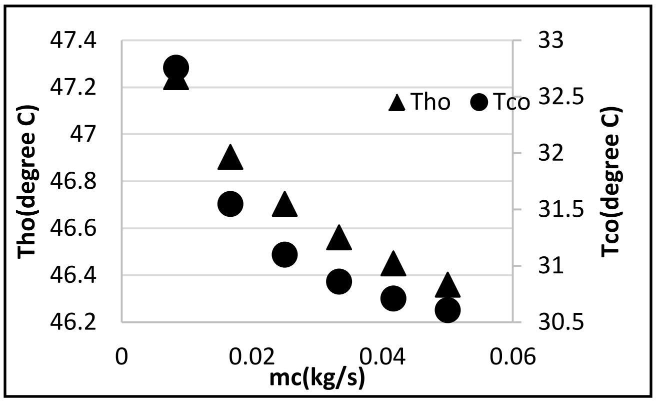


Fig. 5 Variation of outlet temperature of the hot and cold fluid with cold fluid

In the above figure, the graph is plotted between the outlet temperature of the cold fluid and the mass flow rate of the cold fluid and the outlet temperature of the hot fluid and the mass flow rate of the cold fluid. In the first case, we can observe that the outlet temperature of the cold fluid decreases as the mass flow rate of the cold fluid increases. This is due to the fact the heat lost by the hot fluid is equal to the heat gained by the cold fluid. So, if the specific heat of the cold fluid increases the value of ΔT should decrease to maintain the equilibrium and hence such nature of the graph. The same explanation can be observed for the outlet temperature of the hot fluid.

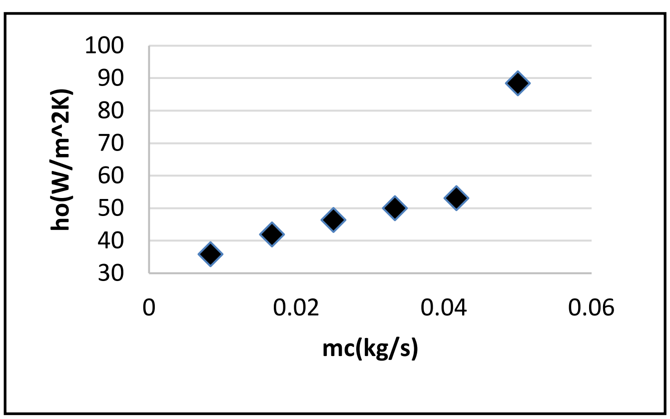


Fig. 6Variation of external heat transfer coefficient with cold fluid.

The above graph is plotted with the mass flow rate of the cold fluid on the x-axis and the external heat transfer coefficient on the y axis. According to the graph, the external heat transfer coefficient increases linearly with an increase in the mass flow rate of the cold fluid and then there is an exponential rise. We know that the external heat transfer coefficient depends on the Nusselt number and the Nusselt number is a function of Reynold's number and Prandtl's number. Reynolds number depends on the mass flow rate of the fluid and cold fluid flows in the external tubes of the shell and tube heat exchanger. So basically, we can say that the external heat transfer coefficient is a function of the mass flow rate of the fluid which flows in the inner tubes. As cold fluid flows in the external tubes of the heat exchanger, it is a linear function of that parameter. Increasing or decreasing the mass flow rate of the fluid will result in the corresponding increase or decrease in the external heat transfer coefficient. The abrupt change like the graph indicates the transition of the fluid flow from laminar to turbulent.

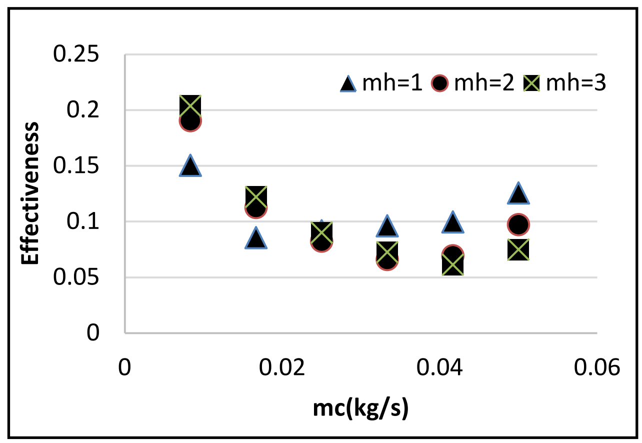


Fig. 7 Variation of the effectiveness of a heat exchanger with different cold fluid rates for different flow rates of hot fluid.

The above graph is plotted between effectiveness and mass flow rate of cold fluid for different hot fluid mass flow rates. Hot fluid rate is set at 1, 2, and 3 and respective effectiveness vs. mass flow rates of cold fluid curves are plotted for mc varying from 0.5 to 3 at a step of 0.5 kg/s. From the graph, it can be seen that with increasing mc the effectiveness first decreases then increases with a comparatively slower rate. This is because the effectiveness of a counter flow heat exchanger is a function of the negative exponential component of the negative of NTU. Thus, effectiveness first decreases till the minimum and increases after that. Refer 3.2.2. the point worth noting is that for mh=1 graph the minimum point appears earlier than mh=2 graph which in turn appears earlier than mh=3 graph. This is because of the values of Cr i.e. ratio of minimum specific heat to maximum specific heat (Cmin/Cmax). Now for mh=1, the value of Cr increases from 0.5 to 1 and then decreases slowly, similar is the case for mh=2 and mh=3. The value of Cr increases from 0.25 to 1 and then decreases for mh=2. Similarly, the value of Cr increases from 0.167 to 1 and then decreases. As effectiveness is inversely proportional to Cr, an increase in Cr will lead to decrease in the effectiveness and vice versa. Hence, we see a decrease till a particular point and further, the effectiveness keeps increasing gradually.

B. Variation of Performance Parameters with Respect to Different Hot Fluid Flow Rates

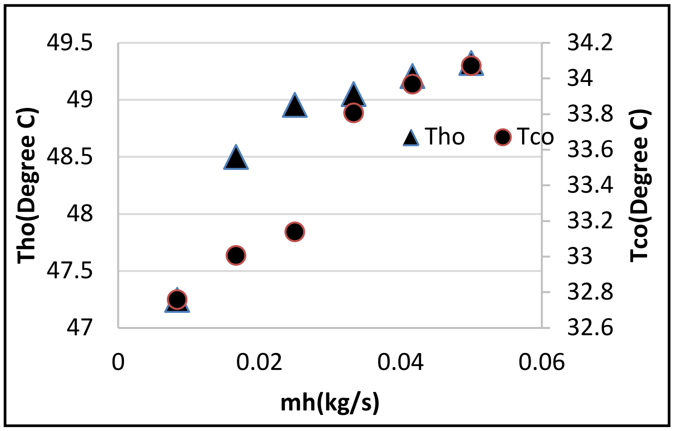


Fig. 8Variation of outlet temperature of the hot and cold fluid with hot fluid

The above graph illustrates the variation of the outlet temperature of the hot fluid to the increase in the mass flow rate of the hot fluid and also the variation of the outlet temperature of the cold fluid to the mass flow rate of the hot fluid. It is observed that as the mass flow rate of the hot fluid increases the outlet temperature of the hot fluid coefficient increases linearly with increase in the mass flow rate of the hot fluid and then there is a sudden rise. We know that the internal heat transfer coefficient depends on Nusselt number and Nusselt number is a function of Reynold's number and Prandtl's number. Reynolds number depends on the mass flow rate of the fluid and hot fluid flows in the internal tubes of the shell and tube heat exchanger. So basically, we can say that the internal heat transfer coefficient is a function of mass flow rate of the fluid which flows in the inner tubes.

increase. This is due to the fact that the heat lost by the hot fluid can be written as the product of specific heat of the hot fluid and the corresponding decrease in temperature. If the specific heat of the hot fluid increases the corresponding drop in temperature should be less to maintain proportionality. The outlet temperature of the cold fluid increases on increasing the mass flow rate of the hot fluid. This is because if there is a decrease in the temperature drop of the hot fluid there should be a corresponding increase in temperature rise across the cold fluid to maintain equilibrium.

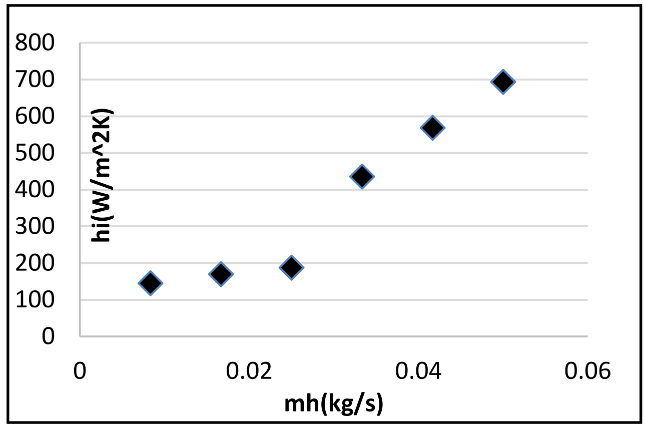


Fig. 9Variation of internal heat transfer coefficient with mass flow rate of hot fluid.

The above graph is plotted with mass flow rate of the hot fluid on the x axis and internal heat transfer coefficient on

They axis. According to the graph the internal heat transfer As hot fluid flows in the inner tubes of the heat exchanger, it is a function of that parameter. Increasing or decreasing the mass flow rate of the fluid will result in the corresponding increase or decrease in the internal heat transfer coefficient. The abrupt change in the nature of the graph indicates the transition of the fluid flow from laminar to turbulent.

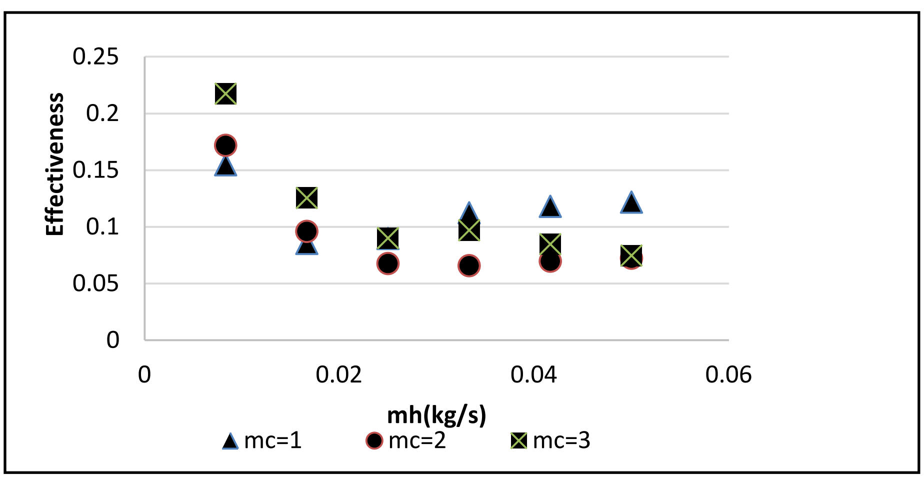


Fig. 10Variation of effectiveness of a heat exchanger with different hot fluid rates for different flow rates of cold fluid.

Above graph is plotted between effectiveness and mass flow rate of hot fluid for different hot fluid mass flow rates. Cold fluid rate is set at 1, 2 and 3 and respective effectiveness vs mass flow rates of hot fluid curves are plotted for mc varying from 0.5 to 3 at a step of 0.5 kg/s. From the graph it can be clearly seen that with increasing mh, the effectiveness first decreases then increases with a comparatively slower rate.

This is because the effectiveness for a counter flow heat exchanger is a function of negative exponential component of negative of NTU. Thus, effectiveness first decreases till the minimum and increases after that. The point worth noting is that for mc=1 graph the minimum point appears earlier than mc=2 graph which in turn appears earlier than mc=3 graph. This is because of the values of Cr i.e. ratio of minimum specific heat to maximum specific heat (Cmin/Cmax). Now for mc=1, value of Cr increases from 0.5 to 1 and then decreases slowly, similar is the case for mc=2 and mc=3.

The value of Cr increases from 0.25 to 1 and then decreases for mc=2. Similarly, the value of Cr increases from 0.167 to 1 and then decreases.

As effectiveness is inversely proportional to Cr, increase in Cr will lead to decreases in effectiveness and vice versa. Hence, we see decrease till a particular point and further the effectiveness keeps increasing at a slower rate.

C. Variation of performance parameters with respect to the number of transfer units

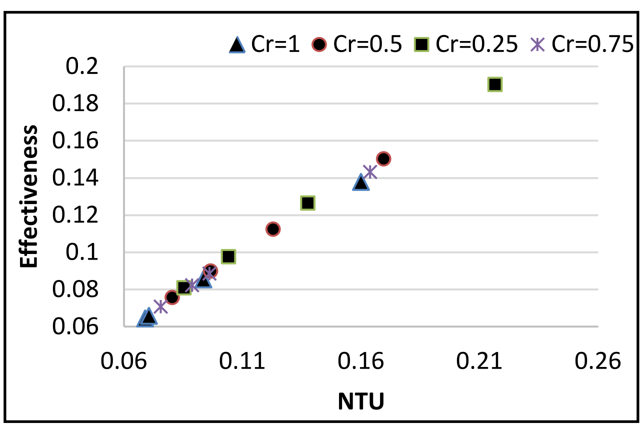


Fig. 11 Experimental Variation of effectiveness of a heat exchanger with NTU.

The above graph is plotted between effectiveness and NTU. The figure below that represents the standard graph between effectiveness and NTU for a double pipe heat exchanger. The curves are obtained by plotting effectiveness with an increase in NTU at different Cr i.e., ratio of minimum specific heat and maximum specific heat. From the figure it can be seen that effectiveness increases linearly with NTU till it reaches 0.5. After that it increases as an exponential function as effectiveness is a function of the negative exponent of the negative component of NTU.

After comparing the theoretical and experimental results the following graphs were plotted.

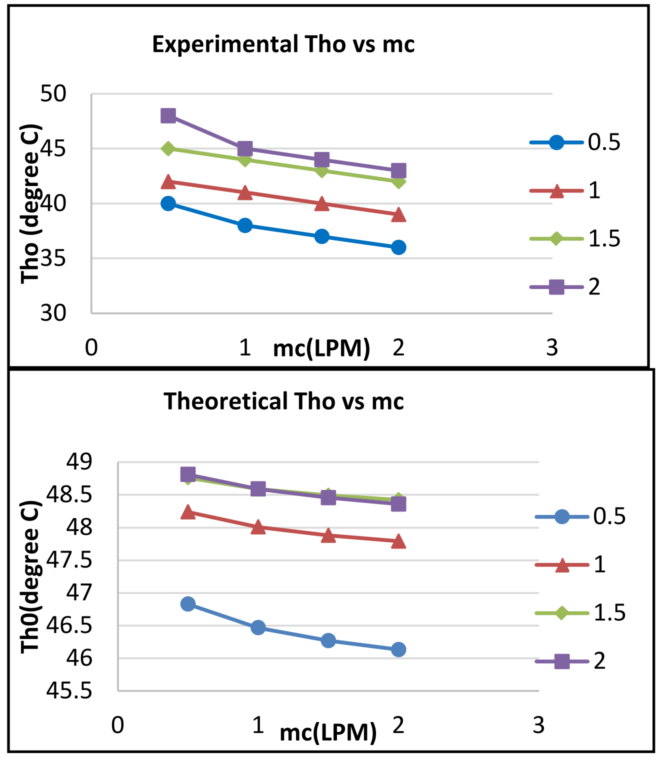
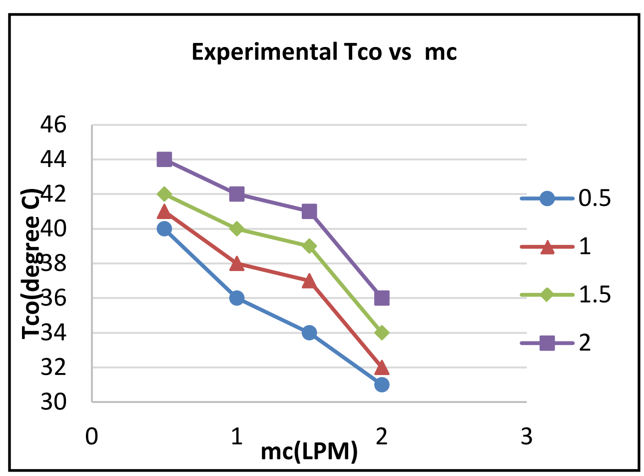


Fig. 12 Comparison of the variation of outlet temperature of hot fluid with mass flow rate of cold fluid for the experimental and theoretical data

The above comparison is done between experimental and theoretical temperatures of the hot outlet. It can be seen that temperature at hot outlet decreases with an increase in the mass flow rate of cold fluid. This is because the heat gained by the cold fluid should be equal to the heat lost by the hot fluid. So, if the specific heat of the fluid increases then the value of temperature difference should decrease to compensate for the change. The point worth noting is that the range of temperature at the hot outlet is minimum for minimum hot flow rate i.e., 0.5 and keeps on increasing till it reaches 0.5.



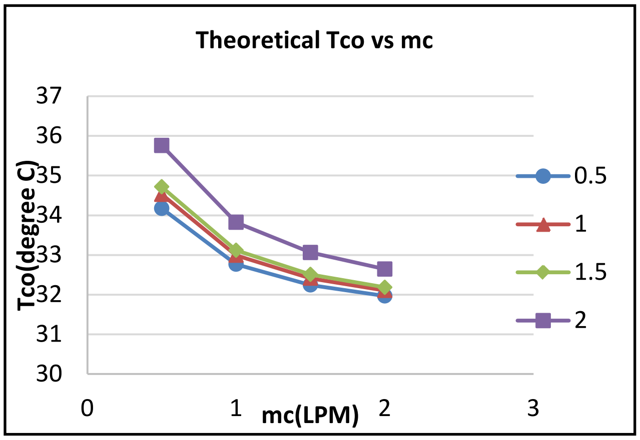


Fig.13 Comparison of the variation of outlet temperature of cold fluid with mass flow rate of cold fluid for the experimental and theoretical data

The above comparison is done between experimental and theoretical temperatures of the cold outlet. It can be clearly seen that temperature at cold outlet decreases with increase in mass flow rate of hot fluid. The point worth noting is that the range of temperature at hot outlet is minimum for minimum hot flow rate i.e., 0.5 and keeps on increasing till 2.

From the above comparisons, it can be concluded that temperatures at the outlet decrease hyperbolically with an increase in mass flow rate of cold fluid. This is because heat transfer is given by Q=mCp (ΔT) where ΔT=Tco-Tci if m= mass flow rate of cold fluid and Cp is the specific heat capacity of cold fluid (Cpc). Thus, the mass flow rate of the fluid is inversely proportional to ΔT which gives a hyperbolic nature.

V. CONCLUSION

The effect of hot fluid and cold fluid flow rate and hot fluid inlet temperature at 50°C and 70°C on the double pipe heat exchanger was studied using water as the working fluid at cold fluid inlet temperature of 31°C for the entire range of experimental conditions. The experimental results such as effectiveness, overall heat transfer coefficient, number of transfer units, and outlet temperature of the hot and cold fluid were compared with the theoretical predictions. A comparison of experimental results with the theoretical predictions shows that experimental results are higher by an amount of 10-12% for the entire range of the parametric space which is well within range. As the mass flow rate of the cold fluid increases, there is a drop in both the outlet temperature of the hot fluid as well as the cold fluid. As the mass flow rate of the hot fluid increases, there is a rise in both the outlet temperature of the hot fluid as well as the cold fluid. Increasing the mass flow rate of the hot fluid decreases the effectiveness first and then increases it at a comparatively slower rate. The overall heat transfer coefficient on the external side increases with increase in the mass flow rate of the cold fluid. The overall heat transfer coefficient on the internal side increases with an increase in mass flow rate of the hot fluid.

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