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Numerical Modeling and Simulation of a Double Tube Heat Exchanger Adopting a Black Box Approach

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ABSTRACT
The double tube heat exchangers are commonly used in industry due to their simplicity in design and also their operation at high temperatures and pressures. As the inlet parameters like temperatures and mass flow rates change during operation, the outlet temperatures will also change. In the present paper, a simple approximate linear model has been proposed to predict the outlet temperatures of a double tube heat exchanger, considering it as a black box. The simulation of the heat exchanger has been carried out first using the commercial CFD software FLUENT. Next the linear model of the double tube heat exchanger based on lumped parameters has been developed using the basic governing equations, considering it as a black box. Results have been generated for outlet temperatures for different inlet temperatures and mass flow rates of the cold and hot fluids. The results obtained using the above two methods have then been discussed and compared with the numerical results available in the literature to justify the basis for the assumption of a linear approximation. Comparisons of the predicted results from the present model show a good agreement with the experimental results published in the literature. The assumptions of linear variation of outlet temperatures with the inlet temperature of one fluid (keeping other inlet parameters fixed) is very well justified and hence the model can be employed for the analysis of double tube heat exchangers.
Keywords – Black Box, Heat Exchanger, Linear Model, Numerical Modeling

NOMENCLATURE

T	Temperature of fluid (°C)
$\dot{m}$	Mass flow rate of fluid (kg/s)
C_p	Specific heat capacity of fluid (J/kg-°C)
A	Area (m^2)
U	Overall heat transfer coefficient (W/m^2-°C)
h	Convection coefficient (W/m^2-°C)
k	Thermal conductivity of tube material (W/m-°C)
d	Diameter (m)
L	Length (m)
Re	Reynolds number (-)
Pr	Prandtl number (-)
Nu	Nusselt number (-)
ρ	Density of fluid (kg/m^3)
μ	Dynamic viscosity of fluid (kg/m-s)
λ	Thermal conductivity of fluid (W/m-°C)
ε	Effectiveness of heat transfer (-)

Subscripts

1	Inlet, interior
2	Outlet, exterior
h	Hot fluid
c	Cold fluid
in	Internal tube
an	Annular space

I. Introduction

Heat exchanger is a device which controls the temperature of a system or a substance by adding or removing thermal energy. Although there are different types of heat exchangers with varying sizes [1, 2], they have a basic similarity. All of them use a thermally conducting element usually in the form of a plate or a tube to separate the two fluids, such that one can transfer thermal energy to the other even without being mixed. The double tube heat exchanger involves two concentric tubes, where generally, a hot fluid flows through the interior tube and a cold fluid flows through the annular space. This type of heat exchangers are widely used in food and oil refinery industries and are an important element of various types of installations like steam power labs, heating and air conditioning systems. The most widely used types of double tube heat exchangers are counter flow heat exchangers because of their high effectiveness. While there are many advantages of double tube heat exchangers like simple structures, operation in parallel and counter flow, operation at relatively low flow rates, low costs, etc., their disadvantages are related to low values of the overall heat transfer coefficients that leads to large heat transfer areas [3].

The mathematical modeling of heat exchangers has been treated extensively in literature.

Heat exchanger models to predict the outlet temperatures of the fluids [4] and a heat transfer coefficient calculation program based on the EES software for the double tube heat exchangers [5] have been developed. The solution of the mathematical models of the heat exchangers using numerical algorithms has been performed [6]. Simulators based on these models have been developed which allow the determination of the outlet temperatures of the two fluids between which the heat transfer is realized, by using data about heat exchanger geometry and the values of inlet temperatures and mass flow rates of the two fluids [7, 8].

In the present paper the authors proposed another approach for evaluating the outlet temperatures of the double tube heat exchangers in a simplified approximate linear form. In this work, the heat exchanger has been considered as a 'black box' to examine how the input variables affect the output.

1.1 Geometry and arrangement of the heat exchanger

Heat exchangers are typically classified according to flow arrangement and type of construction. The simplest heat exchanger is one for which the hot and cold fluids move in the same or opposite directions in a concentric tube (or double-tube) construction. The schematic diagrams of double tube heat exchanger have been shown in Fig. 1-a and Fig. 1-b for the counter flow and parallel flow arrangements of the two participating fluids respectively. In the parallel flow arrangement, the hot and cold fluids enter at the same end, flow in the same direction, and leave at the same end. In the counter-flow arrangement, the fluids enter at opposite ends, flow in opposite directions, and leave at opposite ends.

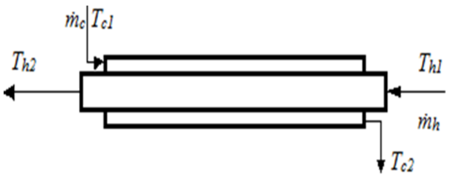

Figure 1-a: Flow arrangement of counter-flow double tube heat exchanger

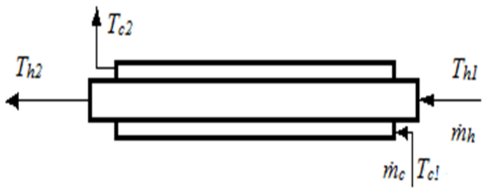

Figure 1-b: Flow arrangement of parallel flow double tube heat exchanger

The geometrical construction of the double tube heat exchanger is as shown in Fig. 2. The hot fluid is assumed to flow through the inner tube and the cold fluid is assumed to flow through the annular space.

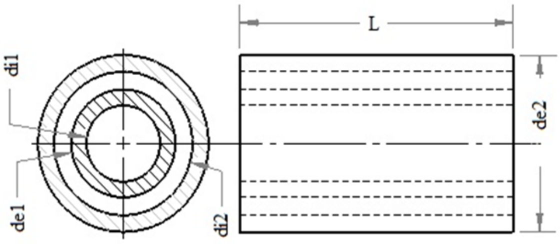

Figure 2: Geometrical dimensions of the double tube heat exchanger

II. Modeling And Simulation Using Ansys (Fluent)

The modeling and simulation has been carried out using a double tube counter-flow heat exchanger using water as the hot as well as the cold fluid. The geometrical dimensions and inlet parameters of the heat exchanger to be investigated are presented in Table 1 and Table 2 respectively. The tubes of the heat exchanger being examined are made of copper.

Table 1: Geometrical dimensions

Geometrical characteristic	Variable	Value
Interior diameter of the interior tube [m]	d_{i1}	0.027
Exterior diameter of the interior tube [m]	d_{e1}	0.030
Interior diameter of the exterior tube [m]	d_{i2}	0.040
Exterior diameter of the exterior tube [m]	d_{e2}	0.043
Length of the tube [m]	L	1

Table 2: Inlet parameters

No.	m_c [kg/s]	m_h [kg/s]	T_{c1} [ºC]	T_{h1} [ºC]
1	0.03	0.1	17	52
2	0.03	0.1	18	52
3	0.03	0.1	19	52
4	0.03	0.1	20	52
5	0.03	0.1	21	52
6	0.03	0.1	17	51
7	0.03	0.1	17	50
8	0.03	0.1	17	49
9	0.03	0.1	17	48
10	0.04	0.1	17	52
11	0.05	0.1	17	52
12	0.06	0.1	17	52
13	0.07	0.1	17	52

The heat exchanger has been modeled using Fluent in such a way that the interior of the heat exchanger was visible for the ease of defining the surfaces, symmetries, cell zone and boundary conditions. In modeling the heat exchanger, the outer surface was considered to be perfectly insulated. The model after meshing has been presented in Fig. 3.

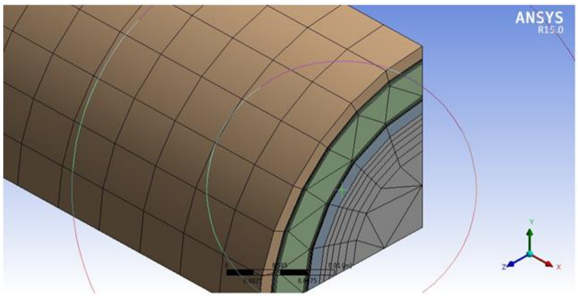

Figure 3: Meshed model

The temperature distributions for the third set of input as shown in Table 2 in the vicinity of the cold fluid entry side (and simultaneously hot fluid exit) and cold fluid exit side (and simultaneously hot fluid entry) have been presented in Fig. 4-a and Fig. 4-b respectively.

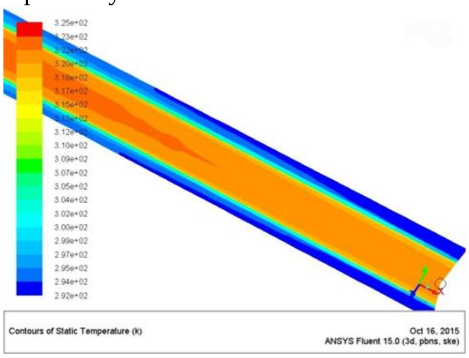

Figure 4-a: Contour of static temperature (cold fluid entry side)

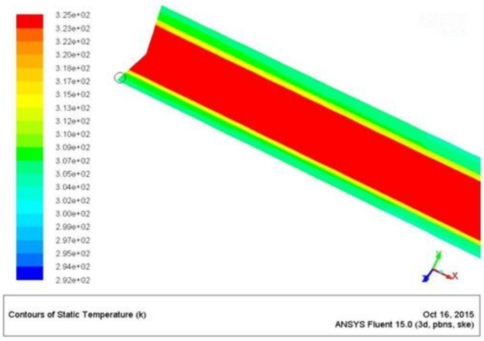

Figure 4-b: Contour of static temperature (hot fluid entry side)

The results of this simulation have been discussed in detail and also compared with the experimental results and that obtained using the proposed linear model in section 4.

III. The Mathematical Model

To mathematical model is based on certain assumptions. The flow is single phase and in steady state regime. The heat transfer to the surrounding environment is neglected, i.e., the outer surface of the heat exchanger is perfectly insulated. The heat exchanger is considered a system with lumped parameters. Considering the heat balance between the hot and the cold fluid and the heat transfer, the following equations can be written:

$$\dot{m}_h C_{ph} (T_{h1} - T_{h2}) = \dot{m}_c C_{pc} (T_{c2} - T_{c1}) \quad (1)$$

$$\dot{m}_h C_{ph} (T_{h1} - T_{h2}) = UAT_{lm} \quad (2)$$

The logarithmic mean temperature difference, T_{lm}, for counter-flow heat exchangers is given as:

$$T_{lm} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{ln\left(\frac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)} \quad (3)$$

and for parallel flow heat exchangers as:

$$T_{lm} = \frac{(T_{h1} - T_{c1}) - (T_{h2} - T_{c2})}{ln\left(\frac{T_{h1} - T_{c1}}{T_{h2} - T_{c2}}\right)} \quad (4)$$

The system of equations given by (1) and (2) represents a system of transcendental equations with two variables, having the form:

$$f_1(T_{h2}, T_{c2}) = \dot{m}_h C_{ph} (T_{h1} - T_{h2}) - \dot{m}_c C_{pc} (T_{c2} - T_{c1}) = 0 \quad (5)$$

$$f_2(T_{h2}, T_{c2}) = \dot{m}_h C_{ph} (T_{h1} - T_{h2}) - UAT_{lm} = 0 \quad (6)$$

The solution of the system of transcendental equations is rather complex. Therefore, in this paper an attempt has been made to simplify the set of equations, using certain assumptions to obtain the outlet temperatures of the hot and cold fluids as linear functions of the inlet temperatures.

3.1 Linear model using the black box approach

A black box model is a system which can be viewed in terms of its inputs and outputs (or transfer characteristics). The heat exchanger system as a black box model is shown in Fig. 5. The inputs are the inlet temperatures and mass flow rates of the hot and cold fluids and the outputs are the outlet temperatures of the two fluids. The black box approach of heat exchanger analysis has been attempted before [9]. It was specifically modeled for counter-flow heat exchangers having five constant coefficients. In the present work, an attempt has been made to put forward a general purpose linear relationship between the outlet temperatures, the inlet temperatures and heat capacity rate ratios using the black box model. Steady state, single phase flow, constant physical property values of the fluid and insulated outer surface conditions are assumed for the analysis. The logarithmic mean temperature difference has been approximated by arithmetic mean temperature difference (within about 1.4% error).

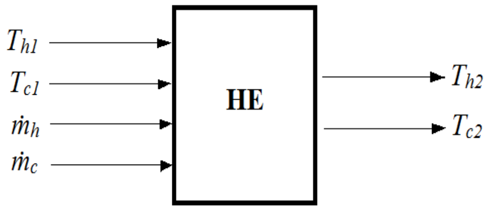

Figure 5: Inlet and outlet parameters of a double tube heat exchanger

The basic mathematical model remains the same, i.e., here the linear model is developed by simplifying the equations (1) and (2). The equation (1) remains the same and T_{lm} in equation (2) is replaced by the arithmetic mean temperature difference, T_{am}, which can be approximated for the analysis of counter-flow heat exchangers and is given as:

$$T_{am} = \frac{(T_{h1} + T_{h2}) - (T_{c1} + T_{c2})}{2} \quad (7)$$

Now, using equation (1), T_{h2} may be written in terms of T_{c2} as:

$$T_{h2} = -\left(\frac{\dot{m}_c C_{pc}}{\dot{m}_h C_{ph}}\right) T_{c2} + \left[\left(\frac{\dot{m}_c C_{pc}}{\dot{m}_h C_{ph}}\right) T_{c1} + T_{h1}\right] \quad (8)$$

Here it is assumed that the thermodynamic properties remain constant. Now, after some simplification, equation (2) may be written as:

$$\left(\frac{2\dot{m}_h C_{ph}}{UA}\right)(T_{h1} - T_{h2}) = 2T_{am} \quad (9)$$

The overall heat transfer coefficient is based on the cooler side and the heat transfer surface area, A, is given by the relation:

$$A = \Pi d_{e1} L \quad (10)$$

Here, two non-dimensional parameters, γ and C, are introduced for the heat capacity rate ratios:

$$\gamma = \frac{2\dot{m}_h C_{ph}}{UA} \quad (11)$$

$$C = \frac{\dot{m}_c C_{pc}}{\dot{m}_h C_{ph}} \quad (12)$$

Now, using equations (7) to (12), the cold fluid outlet temperature, T_{c2} has been obtained as:

$$T_{c2} = \frac{2T_{h1} + C(\gamma + 1)T_{c1}}{C(\gamma + 1) + 1} \quad (13)$$

Similarly, T_{h2} has been obtained using equations (8) and (13) and is given as:

$$T_{h2} = \frac{[1 + C(\gamma - 1)]T_{h1} + CT_{c1}}{C(\gamma + 1) + 1} \quad (14)$$

The effectiveness, ε, of the heat exchanger may be given as:

$$\varepsilon = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} \quad (15)$$

With the above said assumptions, the set of transcendental equations has been converted to a set of linear equations involving two unknowns. The overall heat transfer coefficient has been treated as a constant value for a particular set of data.

3.2 Estimation of overall heat transfer coefficient

The overall heat transfer coefficient, U, has a known expression and can be written as:

$$U = \frac{1}{\frac{1}{h_{in}}\left(\frac{d_{e1}}{d_{i1}}\right) + \left(\frac{d_{e1}}{2k}\right) ln \frac{d_{e1}}{d_{i1}} + \frac{1}{h_{an}}} \quad (16)$$

The calculation of the convection coefficients have been carried out by using the Reynolds, Prandtl and Nusselt similitude criteria relation. The Reynolds similitude criteria for the fluid flowing through the interior tube and the annular space are obtained from the following two equations respectively:

$$\mathrm{Re}_{in} = \frac{\rho_h v_h d_{i1}}{\mu_h} \quad (17)$$

$$\mathrm{Re}_{an} = \frac{\rho_c v_c d_{eh}}{\mu_c} \quad (18)$$

v_h and v_c are the velocities of the hot and cold fluids respectively, which are given as:

$$v_h = \frac{\dot{m}_h}{\rho_h A_{in}} \quad (19)$$

$$v_c = \frac{\dot{m}_c}{\rho_c A_{an}} \quad (20)$$

The cross sectional areas in the interior tube and the annular space are given as:

$$\left.\begin{aligned} A_{in} &= 0.25\Pi d_{i1}^2 \\ A_{an} &= 0.25\Pi(d_{i2}^2 - d_{e1}^2) \end{aligned}\right\} \quad (21)$$

The term d_{eh} in equation (18) is known as the equivalent hydraulic diameter and is given as:

$$d_{eh} = d_{i2} - d_{e1} \quad (22)$$

The Prandtl similitude criteria for the interior fluid and that flowing through the annular space are given by the following two equations respectively:

$$\mathrm{Pr}_{in} = \frac{C_{ph}\mu_h}{\lambda_h} \quad (23)$$

$$\mathrm{Pr}_{an} = \frac{C_{pc}\mu_c}{\lambda_c} \quad (24)$$

The Nusselt numbers in the circular section of the interior tube and the annular space has been calculated for the laminar, intermediate and turbulent flowing regime [10] and its expressions are given in equations (25) and (26) respectively.

In the circular section of the interior tube:

$$\left.\begin{aligned} &\mathrm{Re}_{in} < 2300 \text{ (laminar):} \\ &\qquad \mathrm{Nu}_{in} = 1.86[\mathrm{Re}_{in}\mathrm{Pr}_{in}(d_{i1}/L)]^{0.33} \\ &2300 \le \mathrm{Re}_{in} < 10^4 \text{ (intermediate):} \\ &\qquad \mathrm{Nu}_{in} = \\ &\qquad 0.023\,\mathrm{Re}_{in}^{0.8}\mathrm{Pr}_{in}^{0.4}[1\text{-}(6e5/\mathrm{Re}_{in}^{1.8})] \\ &\mathrm{Re}_{in} \ge 10^4 \text{ (turbulent):} \\ &\qquad \mathrm{Nu}_{in} = 0.023\,\mathrm{Re}_{in}^{0.8}\mathrm{Pr}_{in}^{0.4} \end{aligned}\right\} \quad (25)$$

In the annular space:

$$\left.\begin{array}{l} \mathrm{Re}_{an}<2300 \text{ (laminar):} \\ \quad \mathrm{Nu}_{an}=4.05\,\mathrm{Re}_{an}^{0.17}\mathrm{Pr}_{an}^{0.33} \\ 2300\leq\mathrm{Re}_{an}<10^4 \text{ (intermediate):} \\ \quad \mathrm{Nu}_{an}= \\ 1.86[\mathrm{Re}_{an}\mathrm{Pr}_{an}(d_{eh}/L)]^{0.33}[1-(6e5/\mathrm{Re}_{an}^{1.8})] \\ \mathrm{Re}_{an}\geq10^4 \text{ (turbulent):} \\ \quad \mathrm{Nu}_{an}=0.023\,\mathrm{Re}_{an}^{0.8}\mathrm{Pr}_{an}^{0.4} \end{array}\right\} \quad (26)$$

The convection coefficients h_i and h_e are determined respectively from the following equations:

$$h_{in}=\frac{\mathrm{Nu}_{in}\lambda_h}{d_{i1}} \quad (27)$$

$$h_{an}=\frac{\mathrm{Nu}_{an}\lambda_c}{d_{eh}} \quad (28)$$

Some representative values of the physical properties of some common fluids and thermal conductivities of some common tube materials have been incorporated into the simulation program in order to calculate the convection coefficients and hence the overall heat transfer coefficient using equation (16).

IV. Results And Discussion

The primary focus here was to justify and validate the assumption of a linear model by comparing the results with those obtained from ANSYS (Fluent) simulation and also the experimentally obtained data. The basis of the assumption of a linear model was that the Fluent simulation produced a similar linear plot (Fig. 6-9), by varying one inlet parameter (temperature of one fluid) and keeping other inlet parameters fixed. The simulations and linear model were primarily based on the counter-flow heat exchangers using water as the hot as well as the cold fluid. This is due to the fact that because of their high effectiveness, counter-flow heat exchangers are most widely used in practical applications. The results obtained from the existing simulator model [8, 9] (sim), Fluent simulation (ANS) and those obtained from the proposed linear model (sim1) were then compared to validate the proposed model. The double tube heat exchanger under consideration is made of copper, whose geometrical dimensions have been presented in Table 1. The inlet parameters have been presented in Table 2. The comparative results of the outlet temperatures of the heat exchanger are presented in Table 3.

Table 3: The comparison between sim, ANS and sim1 data

No.	T_{h2}[°C]			T_{c2}[°C]			ε		
	sim	ANS	sim1	sim	ANS	sim1	sim	ANS	sim1
1	49.06	48.50	48.22	26.78	31.00	29.61	0.279	0.400	0.360
2	49.14	48.60	48.26	27.54	31.60	30.46	0.281	0.400	0.366
3	49.21	48.70	48.30	28.29	32.20	31.32	0.282	0.400	0.373
4	49.28	48.80	48.35	29.04	32.80	32.17	0.283	0.400	0.380
5	49.36	48.90	48.39	29.79	33.40	33.03	0.284	0.400	0.388
6	48.16	47.60	47.30	26.47	30.60	29.32	0.279	0.400	0.362
7	47.25	46.70	46.39	26.16	30.20	29.03	0.278	0.400	0.364
8	46.34	45.80	45.48	25.85	29.80	28.74	0.277	0.400	0.367
9	45.43	44.90	44.57	25.55	29.40	28.45	0.276	0.400	0.369
10	48.61	48.50	48.02	25.47	29.25	26.95	0.242	0.350	0.284
11	48.25	48.50	47.88	24.49	27.50	25.24	0.214	0.300	0.235
12	47.95	46.75	47.78	23.74	26.63	24.04	0.193	0.275	0.201
13	47.71	46.75	47.69	23.13	25.75	23.15	0.175	0.250	0.176

Table 4: Statistical parameters of the double tube heat exchanger (linear model)

Statistical parameters	Output Variables			
	against ANS		against sim	
	T_{h2}	T_{c2}	T_{h2}	T_{c2}
Max. absolute deviation [°C]	1.03	2.6	0.97	3.24
Max. relative deviation [%]	2.20	10.09	1.97	10.87

Using Table 3, the statistical parameters associated to output variables of the proposed linear model have been presented in Table 4.

The adoption of a linear model for the double tube heat exchanger has been validated by comparing the variation of outlet temperatures with inlet temperature of one fluid (hot or cold), keeping other inlet parameters like mass flow rates and the inlet temperature of the other fluid constant, and hence plotting the results obtained from our relation with that from Fluent and sim data and comparing them (Fig. 6-9). As can be seen from these figures, the outlet temperatures obtained from the existing simulator model as well as those obtained from Fluent simulation too vary approximately linearly with the inlet temperatures of both the fluids, provided other inlet parameters are kept constant. The slight differences in the values of the outlet temperatures arise due to the assumptions incorporated in developing the linear model.

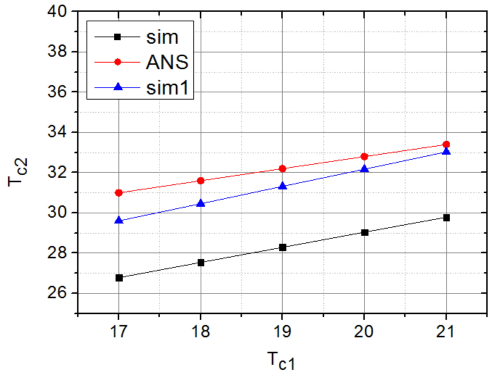

Figure 6: T_{c2} vs T_{c1} (other parameters constant)

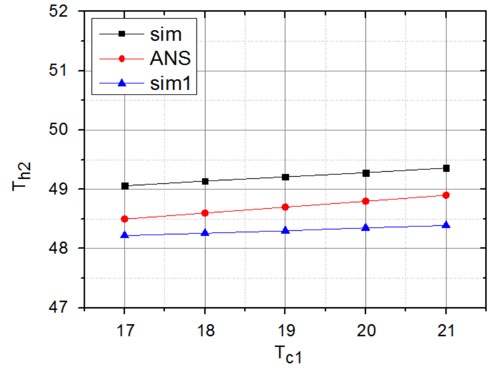

Figure 7: T_{h2} vs T_{c1} (other parameters constant)

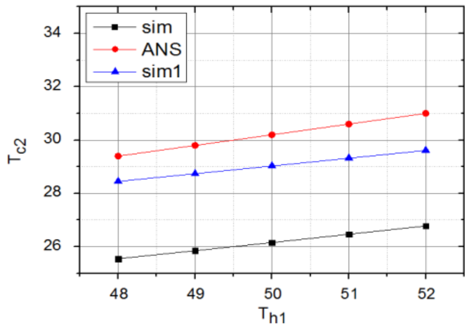

Figure 8: T_{c2} vs T_{h1} (other parameters constant)

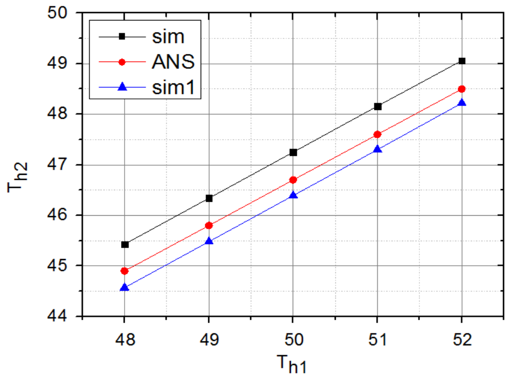

Figure 9: T_{h2} vs T_{h1} (other parameters constant)

The effectiveness of the heat exchanger as obtained from Fluent simulation and the proposed linear model have been presented in Table 3 and as seen from Fig. 10, the effectiveness values of the two models are linear and parallel to the line (y = x) (but not along the line due to certain errors as already discussed) which indicates that the proposed linear model is fairly accurate.

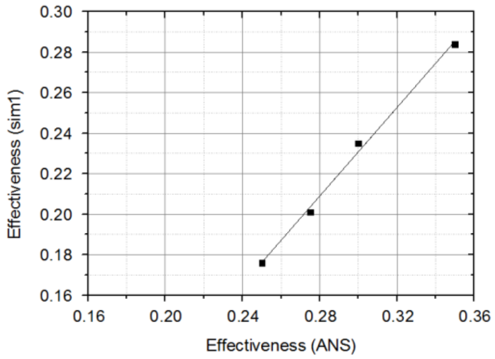

Figure 10: Comparison of Effectiveness

To further validate the results, the data from the linear model was compared with the experimental data [8] along with Fluent and the existing model data for a different heat exchanger whose geometrical dimensions and inlet parameters are provided in Table 5 and Table 6 respectively. Table 7 shows the comparative analysis which indicates that the results obtained for the linear model hold good and are somewhat better than the data predicted by the existing simulator model for this case.

Table 5: Geometrical dimensions

Geometrical characteristic	Variable	Value
Interior diameter of the interior tube [m]	d_{i1}	0.012
Exterior diameter of the interior tube [m]	d_{e1}	0.026
Interior diameter of the exterior tube [m]	d_{i2}	0.014
Exterior diameter of the exterior tube [m]	d_{e2}	0.028
Length of the tube [m]	L	0.935

Table 6: Inlet parameters

No.	m_c [kg/s]	m_h [kg/s]	T_{c1} [°C]	T_{h1} [°C]
1	0.0256	0.0528	11.8	55.3
2	0.0256	0.0583	11.8	55.3
3	0.0256	0.0597	11.7	55.3
4	0.0256	0.0639	11.7	55.3
5	0.0278	0.0528	11.6	55.3
6	0.0358	0.0611	11.5	55.3
7	0.0511	0.0639	11.0	55.3
8	0.0611	0.0667	10.9	55.3
9	0.0486	0.0681	10.7	55.3

Table 7: Comparison of sim, ANS and sim1 data against experimental results

No.	T_{h2}[ºC]				T_{c2}[ºC]			
	exp	ANS	sim1	sim	exp	ANS	sim1	sim
1	49.2	48.77	49.72	50.7	24.5	27.02	23.32	21.5
2	49.5	48.78	50.06	51.0	25.1	27.03	23.75	21.8
3	49.6	48.76	50.14	51.1	25.2	26.96	23.76	21.8
4	49.9	50.94	50.37	51.3	25.4	26.96	24.03	21.9
5	49.4	48.75	49.62	50.6	23.3	25.80	22.39	20.7
6	49.2	48.73	49.85	50.8	22.0	23.54	20.78	19.2
7	48.5	48.65	49.64	50.6	19.7	20.96	18.07	16.9
8	48.2	48.64	49.63	50.6	18.6	19.78	17.09	16.1
9	47.5	48.61	49.89	50.8	17.7	20.73	18.26	17.0

The slight differences compared with experimental results arise due to flow rate inconsistencies, heat losses in the experimental setup, small scale heat transfer apparatus and measurement inaccuracies. Also steady state flow is not perfectly realized in the laboratory setup.

V. Conclusion

In the present paper, a simpler approximate linear model was proposed for the determination of the outlet temperatures in a double tube heat exchanger. Water was used as heat transfer fluids. In the development of the linear model the heat exchanger has been treated as a 'black box'. The linear model has been validated by comparing the outlet temperatures of the two fluids, calculated with the proposed model, with the results from the existing simulator model, FLUENT simulation and finally with the experimentally obtained data for the same operating conditions. It is observed that the assumption of linear variation of outlet temperatures with the inlet temperature of one fluid (keeping other inlet parameters fixed) is very well justified. Even the slopes of the temperature plots are almost accurate. Taking into account the assumptions in the calculation of the output variables compared to experimentally determined values and those obtained from simulation models (ANS and sim), the deviation is seen to be small. Hence, taking all these factors into account, it may be considered that the proposed linear model can be employed for the analysis of double pipe heat exchanger.

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