Design and Construction of a Concentric Tube Heat Exchanger

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Abstract

The concentric tube heat exchanger was designed in order to study the process of heat transfer between two fluids through a solid partition. It was designed for a counter-flow arrangement and the logarithmic mean temperature difference (LMTD) method of analysis was adopted. Water was used as fluid for the experiment. The temperatures of the hot and cold water supplied to the equipment were 87^o and 27^{o}C, respectively and the outlet temperature of the water after the experiment was 73^{o}C for hot and 37^{o}C for cold water. The results of the experiment were tabulated and a graph of the mean temperatures was drawn. The heat exchanger was 73.4% efficient and has an overall coefficient of heat transfer of 711W/m^{2}K and 48^{o}C Log Mean Temperature Difference. The research takes into account different types of heat exchangers.

Keywords: *Counter-flow arrangement, logarithmic mean temperature difference, coefficient of heat transfer.*

Introduction

A heat exchanger is a device that transfers thermal energy from a high-temperature fluid to a low-temperature fluid with both fluids moving through the device. Examples in practice in which flowing fluids exchange heat are air intercoolers and pre-heaters, condensers and boilers in steam plant, condensers and evaporators in refrigerator units, and many other industrial processes in which a liquid or gas is required to be either cooled or heated (Eckert and Drakes 1974).

There are three main types of heat exchangers:

a. The Recuperative type in which the flowing fluids exchanging heat are on either side of a dividing wall.
b. The Regenerative type in which the hot and cold fluids pass alternately through a space containing a matrix of material that provides alternately a sink and a source for heat flow.
c. The Evaporative type in which a liquid is cooled evaporatively and continuously in the same space as the coolant.

This research paper is on recuperative type of heat exchanger, which can further be classified, based on the relative directions of the flow of the hot and cold fluids, into three types:

a. Parallel flow, when both the fluids move in parallel in the same direction.
b. Counter flow, when the fluids move in parallel but in opposite directions.
c. Cross flow, when the directions of flow are mutually perpendicular (Lienhard 2005).

The objectives of the research work are:

(i) to design and construct a concentric- tube heat exchanger in which two tubes are concentrically arranged and either of the fluids(hot or cold) flows through the tube and the other through the annulus.
(ii) to carry out test on the concentric- tube heat exchanger and obtain values which will be compared to theoretically determined ones.

Theory of Design and Analysis

Design Considerations

In designing heat exchangers, a number of factors that need to be considered are:

1. Resistance to heat transfer should be minimized
2. Contingencies should be anticipated via safety margins; for example, allowance for fouling during operation.
3. The equipment should be sturdy.
4. Cost and material requirements should be kept low.
5. Corrosion should be avoided.
6. Pumping cost should be kept low.
7. Space required should be kept low.
8. Required weight should be kept low.

Design involves trade-off among factors not related to heat transfer. Meeting the objective of minimized thermal resistance implies thin wall separating fluids. Thin walls may not be compatible with sturdiness. Auxiliary steps may have to be taken, for instance, the use of support plates for tubing, to realize sturdiness (Saunders 1981).

The Energy Balance

Since the flow in a tube is completely enclosed, an energy balance may be applied to determine how the mean temperature (T_m) x varies with position along the tube and how the total convective heat transfer Q_{conv} is related to the difference in temperatures at the tube inlet and outlet from Fig. 1.

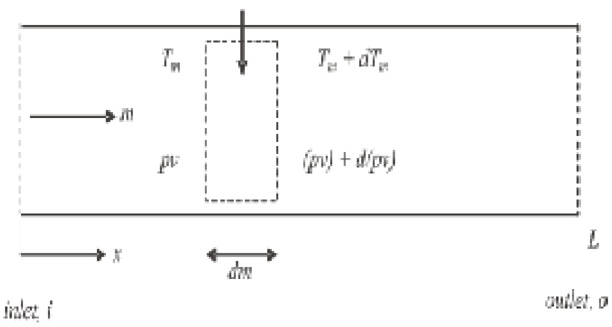

Fig. 1. Control volume for internal flow in the tube.

In Fig.1, fluid moves at a constant flow rate m, and convection heat transfer occurs at the inner surface. If the fluid kinetic and potential changes, as well as energy transfer by conduction in the axially direction are negligible. Therefore, if the fluid is doing no shaft work as it passes through tube, the only significant effects will be those associated with thermal energy change with flow work. Flow to move the fluid with pressure p and per unit mass of flow of fluid is given by the product of the fluid pressure p and specific volume $\acute{v}$ ($\acute{v} = 1/\rho$).

Applying conservation of energy to difference control volume to Fig. 1 and recalling the definition of the mean temperature, we have:

$$\mathrm{D}Q_{\text{conv}} + M(C_vT_m + pv) + [M(C_vT_m + pv)\ \mathrm{d}(pv)/\mathrm{d}x] = 0, \quad (1)$$

or

$$Q_{\text{conv}} = MC_p(T_{m,o} - T_{m,i}). \quad (2)$$

The rate of convective heat transfer to the fluid must equal the rate at which the fluid thermal energy increases plus the net rate at which work is done in moving the fluid through the control volume. A good approximation of incompressible fluid is:

$$\mathrm{D}Q_{\text{conv}} = MC_pT_m. \quad (3)$$

In this case, $C_p = C_v$, and v is very small, $\mathrm{d}(pv)$ is generally much less than $\mathrm{d}(C_vT_m)$. A special form of the equation above relates to conditions for the entire tube. In particular, integrating from the tube inlet, i, to the outlet, o, it follows that:

$$Q_{\text{conv}} = MC_p(T_{m,o} - T_{m,i}), \quad (4)$$

where Q_{conv} is the total tube heat transfer rate. It is a general expression that applies irrespective of the nature of the surface thermal or tube flow conditions.

The Overall Heat Transfer Coefficient

A heat exchanger typically involves two flowing fluids separated by a solid wall. Heat is first transferred from the hot fluid to the wall by convection through the wall by conduction and from the wall to the cold fluid again by convection. Any radiation effects are usually included in the convection heat transfer coefficients (Holman 2002).

For a double-pipe heat exchanger, $A_i = \Pi D_i L$ and $A_o = \Pi D_o L$, and the thermal resistance of the tube wall in this case is

$$R_{wall} = \ln(r_o/r_i)/2\Pi KL, \quad (5)$$

where K is the thermal conductivity of the inner pipe material and L is the length of the tube. Then the total thermal resistance becomes:

$$R = R_{total} = R_i + R_{wall} + R_o = [(1/h_i A_i) + \ln(r_o/r_i)/2\Pi KL + (1/h_o A_o)], \quad (6)$$

where A_i and A_o are the areas of inner and outer pipes respectively.

The rate of heat transfer between the two fluids is:

$$Q = \Delta T/R_{total} = UA\Delta T = U_i A_i \Delta T = U_o A_o \Delta T, \quad (7)$$

where U is the overall heat transfer coefficient (W/m^2 ^{o}C).

Equation (7) can be reduced to:

$$1/UA = 1/U_i A = 1/U_o A_o = R_{total} = 1/h_i A_i + R_{wall} + 1/h_o A_o. \quad (8)$$

If the thickness of the tube is small and the thermal conductivity of the tube, material is high, as is usually the case, the thermal resistance of the tube is negligible ($R_{wal} \approx 0$) and ($A_i \approx A_o \approx A$). Overall heat transfer coefficient becomes,

$$1/U \approx 1/h_i + 1/h_o, \quad (9)$$

where i and o represent inside and outside respectively (Yunus and Robert 2001).

Fouling Factor

The performance of heat exchanger usually deteriorate with time as a result of scaling or deposits from over the interior surface. Scaling or deposits on the inside and outside of the tubes is really a gradual build-up of layers of dirt due to impurities in the fluid, chemical reaction between the fluid and the metal, rust etc. The deposits can severely affect the overall heat transfer coefficient U. It is related to the overall heat transfer coefficient under clean conditions and under fouled conditions by the equation:

$$1/U_{foul} = R_f + 1/U_{clean}. \quad (10)$$

So, for fouled conditions

$$R_{total} = 1/h_i A_i + R_{fi}/A_i + \ln(r_o/r_i)/2\Pi KL + R_{fo}/A_o + 1/h_o, \quad (11)$$

where R_{fi} and R_{fo} are the fouling factors for both inside and outside of the tube respectively.

Analysis of Heat Exchangers

Logarithmic Mean Temperature Difference

The method used in the analysis of the heat exchanger in this research work is the Logarithmic Mean Temperature Difference (LMTD), and it is defined as that temperature difference which, if constant, would give the same rate of heat transfer as actually occurs under variable conditions of temperature difference.

In order to derive expression for LMTD, the following assumptions were made:

The overall heat transfer coefficient U is constant, the flow conditions are steady, the specific heats and mass flow rates of both fluids are constant, the is no loss of heat to the surroundings, there is no change phase either of the fluid during the heat transfer, the change in potential and kinetic energies are negligible, axial conduction along the tubes of the heat exchanger is negligible (Saunders 1981).

In this design, counter-flow LMTD was adopted (Fig. 2) because it is always greater than that for a parallel flow unit, hence counter-flow heat exchanger can transfer more heat than parallel-flow one; in other words a counter-flow heat exchanger needs a smaller heating surface for the same rate of heat transfer.

Let us consider an elementary area dA of the heat exchanger in Fig. 3. The rate of flow of heat through this elementary area is given by

$$dQ = U dA(t_h - t_c) = U dA \Delta t. \quad (12)$$

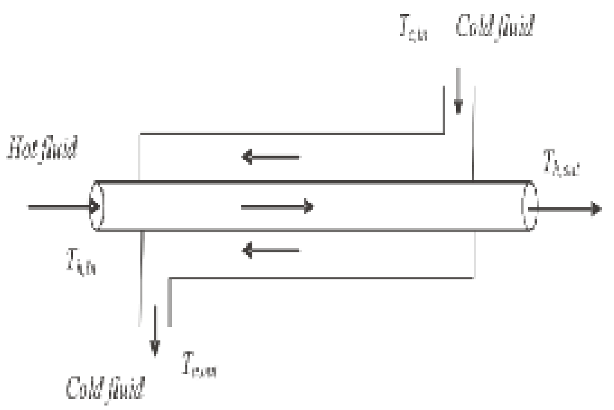

Fig. 2. Flow arrangement for a counter-flow.

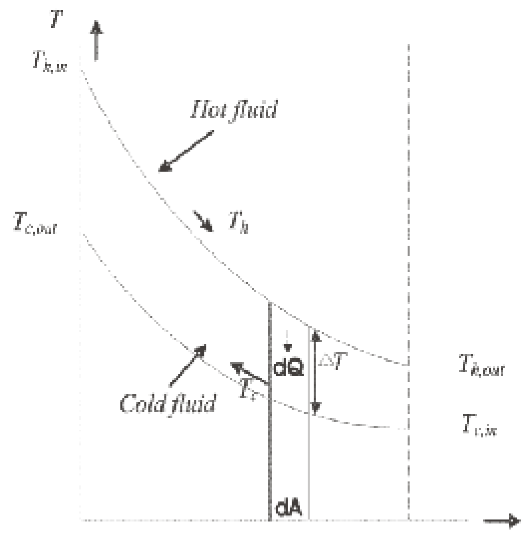

Fig 3. Temperature distribution for a counter-flow.

In this case also, due to heat transfer dQ through the area dA, the hot fluid is cooled down by dt_h whereas the cold fluid is heated by dt_c. The energy balance over a differential area dA may be written as:

$$dQ = -M_hC_{ph}dt_h = -M_cC_{pc}dt_c. \qquad (13)$$

In a counter-flow system, the temperatures of both the fluids decrease in the direction of heat exchanger length, hence the negative signs,

$$dt_h = -(dQ/M_hC_{ph}) \text{ and } dt_c = -(dQ/M_cC_{pc}). \qquad (14)$$

Let $M_hC_{ph} = C_h$ and $M_cC_{pc} = C_c$,

$$dt_h - dt_c = -dQ[(1/C_h) - (1/C_c)], \qquad (15)$$

or

$$d\theta = -dQ[(1/C_h) - (1/C_c)]. \qquad (16)$$

Inserting Eq. (12) into Eq. (16) above:

$$d\theta = -UdA(t_h - t_c)\,[(1/C_h) - (1/C_c)]$$
$$= -UdA\theta\,[(1/C_h) - (1/C_c)],$$

or

$$d\theta/\theta = -UdA\,[(1/C_h) - (1/C_c)]. \qquad (17)$$

Integrating Eq. (17) above from $A = 0$ to $A = A$:

$$\text{In}(\theta_2/\theta_1) = -UA\,[(1/C_h) - (1/C_c)]. \qquad (18)$$

Now, the total heat transfer rate between the two fluids is given by

$$Q = C_h(t_{h1} - t_{h2}) = C_c(t_{c1} - t_{c2}), \qquad (19)$$

or

$$1/C_h = (t_{h1} - t_{h2})/Q,$$

or

$$1/C_c = (t_{c1} - t_{c2})/Q.$$

Substituting the values of $1/C_h$ and $1/C_c$ into Eq. (17) above:

$$\text{In}(\theta_2/\theta_1) = -UA[(t_{h1} - t_{h2})/Q - (t_{c1} - t_{c2})/Q]$$
$$= -(UA/Q)\,[(t_{h1} - t_{h2}) - (t_{c1} - t_{c2})]$$
$$= -(UA/Q)\,(\theta_1 - \theta_2) = (UA/Q)\,(\theta_2 - \theta_1), \qquad (20)$$

or

$$Q = UA(\theta_2 - \theta_1)/\text{In}(\theta_2/\theta_1).$$

Since:

$$Q = UA\theta_m, \qquad (21)$$

then

$$UA\theta_m = UA(\theta_2 - \theta_1)/\text{In}(\theta_2/\theta_1).$$

Therefore:

$$\theta_m = (\theta_2 - \theta_1)/\text{In}(\theta_2/\theta_1) = (\theta_1 - \theta_2)/\text{In}(\theta_1/\theta_2), \qquad (22)$$

where: θ_m = Logarithmic Mean Temperature Difference (LMTD).

Heat Exchanger Effectiveness and Number of Transfer Units (NTU)

A heat exchanger can be designed by the logarithmic mean temperature difference (LMTD) when inlet and outlet conditions are specified. However, when the problem is to determine the inlet or exit temperatures for a particular heat exchanger, the analysis is performed more easily, by using a method based on effectiveness of the heat exchanger and number of transfer units (NTU).

The heat exchanger effectiveness (ξ) is defined as the ratio of actual heat transfer (Q) to the maximum possible heat transfer ($Q_{\max}$).

$$\xi = Q/ Q_{\max}. \tag{23}$$

If:

$$Q = M_h C_{ph}(t_{h1} - t_{h2}) = M_c C_{pc}(t_{c2} - t_{c1}), \tag{24}$$

and

$$Q_{\max} = C_h(t_{h1} - t_{c1}) \text{ or } Q_{\max} = C_c(t_{h1} - t_{c1}), \tag{25}$$

then:

$$\xi = [C_h(t_{h1} - t_{h2})/C_{\min}(t_{h1} - t_{c1})] = [C_c(t_{c2} - t_{c1})/C_{\min}(t_{h1} - t_{c1})]. \tag{26}$$

Once the effectiveness is known, the heat transfer rate can be very easily calculated by using the equation:

$$Q = \xi C_{\min}(t_{h1} - t_{c1}). \tag{27}$$

When using NTU method to determine the effectiveness of a heat exchanger (counter – flow), ξ is expressed as a function of three dimensionless parameters

$$\xi = \{1 - \exp[-\mathrm{NTU}(1 - R)]\}/\{1 - R\exp[-\mathrm{NTU}(1 - R)]\}, \tag{28}$$

where $R = C_{\min}/C_{\max}$, $\mathrm{NTU} = UA/C_{\min}$.

Experimental Procedure

Assembled Heat Exchanger is shown in appendix A. Hot water bucket (1) that contains the ring boiler having the power rating 2,790W was filled with cold water. Cold-water bucket (8) was also filled with water. Both pumps, each with power rating 0.5hp and flow rate of 35L/min that could pump water to a head of 35m at the right pressure, were then connected to the power source but not switched on.

The concentric tube heat exchanger (CHTX) consists of a copper pipe (6) of diameter 0.0127m that is enclosed by a bigger copper pipe (9) of diameter 0.0254m. The bigger pipe carries the cold water while the smaller copper pipe conveys the hot water in a counter-current flow. The cold water only flows through the annulus of the bigger pipe against the flow of the hot fluid. Initial temperature of the cold water was taken. When the power was turned on, some time was given for the water to boil. The temperature of the hot water was taken and recorded. Both gate valves were opened to permit the flow of water and both pumps were switched on to start pumping the water at the same time. The experiment was left for some time to allow proper heat transfer between the cold and hot water through the ½-inch copper pipe. The temperature of the hot water through the ½-inch pipe was taken at the outlet, the temperature of warm water through the 1-inch pipe was also taken, and both were recorded. The experiment was repeated for another four times to get an average value of temperature. At the end of the experiment, the switches were turned off and the valves were closed again.

Results

The results of the experiment are shown in Table 1.

Table 1. Results of the experiment.

S/No	$T_{h,i}$ (°C)	$T_{c,i}$ (°C)	$T_{h,o}$ (°C)	$T_{c,o}$ (°C)
1	87	27	72.5	38.0
2	87	27	73.0	37.0
3	87	27	73.5	37.0
4	87	27	73.0	36.5
5	87	27	73.0	36.5

Calculation of Mean Temperature

The mean temperatures of the outlet temperature of the hot water and that of the outlet temperature of the cold water are used for the determination of the *log* mean temperature difference:

Mean $\theta_{h,o} = (72.5+73.0+73.5+73.0+73.0)/5 = 73.0^{o}\mathrm{C}$,

Mean $\theta_{c,o} = (38.0+37.0+37.0+36.5+36.5)/5 = 37.0^{o}\mathrm{C}$.

This can be calculated using Eq. (22) above:

$$\theta_m = (\theta_1 - \theta_2)/\mathrm{In}(\theta_1/\theta_2),$$

$\Delta\theta_1 = \theta_{h,o} - \theta_{c,i} = 73 - 27 = 46^oC$,

$\Delta\theta_2 = \theta_{h,i} - \theta_{c,o} = 87 - 37 = 50^oC$.

Therefore:
θ_m = LMTD = (46 - 50)/In(46/50) = 47.97 $\approx 48^oC$.

Hence, LMTD = 48^oC = 321K.

Over Heat Transfer Coefficient for the Heat Exchanger

The overall heat transfer coefficient is given by:
$U_{ov} = (r_o/r_i)/h_i + (r_o/k)\ \mathrm{In}(r_o/r_i) + (1/h_o)$.
Data:
Length of the heat exchanger, L = 1.02m;
$h_i = 3106.67 W/m^2K$;
$h_o = 995.77\ W/m^2K$;
k = 386W/mK;
d_i = 0.0127m;
ri = 0.00635m;
thickness of the pipe, t = 0.0015m;
$d_o = d_i + 2t = 0.0127 + 2(0.0015) = 0.0157m$;
$r_o = 0.0157/2 = 0.00785m$.

Applying the above formula:
$1/U_{ov}$ = (0.00785/0.00635) (1/3106.67) + (0.00785/386) In (0.00785/0.00635) + (1/995.77),
and $U_{ov} = (1/0.00140648518) = 711 W/m^2K$.

Determination of the Efficiency of the Heat Exchanger

The relation determines the efficiency of the heat exchanger

$\eta = (Q_c/Q_h)$ x 100
= (24283.80/33078.40) x 100 = 73.4%.

Conclusion

This study as a whole offers an overview of an analytical method applicable to the design of concentric tube heat exchanger (counter-flow type) as shown schematically in the Appendix. Logarithmic mean temperature difference (LMTD) method was used in the design analysis. The overall heat coefficient and the efficiency were computed. Results obtained show that the heat exchanger was effective.

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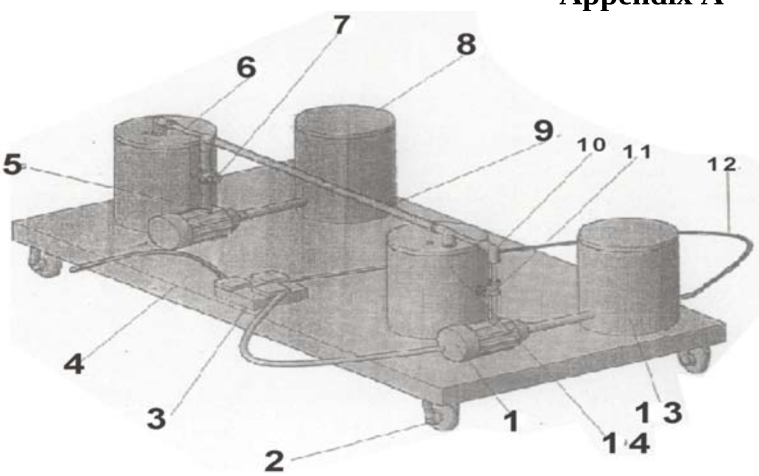
Appendix A



14	RESERVOIR	2	PLASTIC
13	HOT WATER BUCKET	1	PLASTIC
12	CABLE	3	PVC & COPPER
11	HALF INCH VALVE	2	COPPER
10	HALF INCH ELBOW	2	COPPER
9	HEAT EX CHAMBER	1	COPPER
8	COLD WATER BUCKET	1	PLASTIC
7	ONE INCH VALVE	1	COPPER
6	HALF INCH PIPE	1	COPPER
5	NIPPLE	2	GAVAL. STEEL
4	WOODEN FLOOR	1	WOOD .
3	PLUGS	3	CERAMICS
2	ROLLER	4	RUBBER & IRON
1	WATER PUMP	2	CAST
S/N	PART DESCRIPTION	QTY	MATERIAL

Fig. A1. Assembly Drawing of the Heat Exchanger.