ABSTRACT

A systematic mathematical model and computational algorithm has been developed by using LMTD approach for the thermal analysis of tube-fin cross flow heat exchangers. The computer program has been developed by using C language. A detailed parametric study has been conducted for analyzing the influence of geometric and operating parameters on the thermal and hydraulic behavior of the exchanger. Effects of varying frontal air velocity, water temperature drop and air temperature rise on exchanger size, heat transfer rate and power requirements have been studied. Sensitivity analysis of inlet temperature of water and air on exchanger performance has been carried out. Design modification philosophy has also been discussed on the basis of given procedure for arriving at optimal core geometry and level of operating variables for achieving perfect design.

Key words: Heat Exchanger, Thermal, LMTD


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1. INTRODUCTION

The world today is striding towards the ways and means of conserving energy. In recent years it has become increasingly important to develop methods for the efficient transport of thermal energy from one location to another or from one medium to another, i.e. where the processes of heating and cooling occur, is known as heat exchanger. Therefore there is an urgent need for developing more and more efficient and reliable heat exchanger. A heat exchanger may be defined as a device built for efficient transfer of heat from one fluid to another; where either the two fluids are separated by a solid wall so that they never mix, or the two fluids come in direct contact with each other.

These are widely used in refrigerator, air conditioning, cooling tower, air cooler, space heating, electricity generation, chemical processing and transport application. One common example of exchanger is the radiator in a Car, in which a hot engine-cooling fluid, like water, transfer heat to air flowing through the radiator. When a heat exchanger is placed into a thermal transfer system, a temperature drop is required...
to transfer the heat. The size of this temperature drop can be decreased by utilizing a large heat exchanger, but this will increase the cost of the unit. Economic considerations are important in engineering design. Therefore in a complete engineering design of heat exchanger equipment, not only the thermal performance characteristics but also the pumping power requirement and the economics of total system are important. Thus economic, as well as such considerations as the availability and amount of energy and raw materials necessary to accomplish a given task should be considered.

2. HEAT EXCHANGER DESIGN CONSIDERATIONS

Design is an activity aimed at providing complete description of an engineering system, part of a system, or just of a single system component. These descriptions represent an unambiguous specification of the system/ component structure, size, and performance, as well as other characteristics important for subsequent manufacturing and utilization. The major design considerations are:

2.1. Process and Design Specifications

Process and design specifications consist of all necessary information needed to design and optimize a heat exchanger for a specific application. This includes:

- Heavy duty
- Composition and flow rate
- Inlet temperature and Pressure
- Maximum allowable Pressure
- Fouling characteristics

2.2. Thermal and Hydraulic Designs

Heat exchanger thermal / hydraulic design procedures involve exchanger rating (quantitative heat transfer and pressure drop evaluation) and / or exchanger sizing.

2.2.1. Sizing (or Design)

- Decision on construction type, flow arrangement, material.
- Determination of physical size to meet specified heat transfer rate and pressure drop within all specified constrains

2.2.2. Rating (or Performance Evaluation)

Determination of heat transfer and pressure drop performance of either an existing exchanger or an already sized exchanger. The rating problem is also sometimes referred to as the performance or simulation problem.

2.3. Mechanical Design

Mechanical design is essential for ensuring mechanical integrity of the exchanger under steady state, transient, startup, shutdown and part-load operating conditions during its design life.

A qualitative and quantitative analytical aspect of mechanical design includes:

- Selection of materials
- Compliance with national and international codes
- Method of bonding – brazing, welding, soldering, tension winding
- Thermal stress calculation to ensure desired life of heat exchanger for expected start up and shut down period and for part load operating condition
- Minimization and elimination of flow induced vibration
- Proper design of headers, tank, manifolds, nozzles, and inlet and outlet pipes to ensure uniform flow distribution through exchanger flow passages
• Maintenance requirements – cleaning, repairing, serviceability, and general inspections

3. DESIGN METHODOLOGY (LMTD METHOD)

• Input parameters include: Heat dissipation rate, Inlet temperature of fluids, Flow rates of fluids, Available space
• The temperatures of fluids in a heat exchanger are generally not constant but vary from point to point as heat follows from hotter to colder fluid. Even for a constant thermal resistance, the rate of heat flow will therefore vary along the path of the exchangers because its value depends on the temperature difference between the hot and the cold fluid in that section.
• Applying the steady flow energy equation, the intended heat duty $Q_i$ is given by
  \[ Q_i = M_w C_{pw} (T_{wi} - T_{wo}) = M_a C_{pa} (T_{ao} - T_{ai}) \]  
  (1)
• Total heat transfer rate occurring in the heat exchanger is the design value and is expressed as:
  \[ Q_d = U A \Delta T_m \]  
  (2)

4. DESIGN STEPS

4.1. Intended Heat Duty
Intended heat duty ($Q$) is known from input data.

4.2. Inlet and outlet Temperature for Water and Air
Inlet temperatures of both the fluids are selected based on the exchanger design and the environmental conditions. Air temperature rise and water temperature drop are assumed in accordance with standard norms. Initial specifications for temperature intervals for the two streams with due consideration to their inlet values then gives their outlet temperatures. Thus,
  \[ T_{wo} = T_{wi} - \Delta T_w \]  
  (3)
  \[ T_{ao} = T_a + \Delta T_a \]  
  (4)

4.3. Mean Temperature differential for Heat Transfer
Mean temperature difference $\Delta T_m$ is given by,
  \[ \Delta T_m = F_T (LMTD) \]  
  (5)
Where,
Correction factor, $F_T$ is given by,
  \[ F_T = 1 - \sum_{i=1}^{m} \sum_{k=1}^{n} a_{ik} (1 - Y_{lm})^k \sin(2i \arc\tan R) \]
  \[ Y_{lm} = (p - q) / \ln \{ (1 - q) / (1 - p) \} \]
  \[ p = (T_{wi} - T_{wo}) / (T_{wi} - T_{ai}) \]
  \[ q = (T_{ao} - T_{ai}) / (T_{wi} - T_{ai}) \]
  \[ R = (p / q) = (T_{wi} - T_{wo}) / (T_{ao} - T_{ai}) \]

4.4. Water and Air Flow Rates
Mass flow rate for water and air are determined from,
  \[ M_w = Q / (C_{pw} \cdot dT_w) \]  
  (6)
  \[ M_a = Q / (C_{pa} \cdot dT_a) \]  
  (7)

4.5. Air Velocity
The range of frontal air velocity should be known before designing a Heat exchanger. The frontal air velocity for automobile application is laid in the range of 5 to 10 m/s. since the objective is to minimize the
cross section and size of the heat exchanger, it is worth to begin with the higher frontal air velocity. This then determines frontal area of the heat exchanger core. Thus,

\[ A_{fr} = \frac{M_a}{(\rho_a \cdot V_{fr})} \]  

(8)

### 4.6. Core Cross Section Aspect Ratio
Relative dimensions of the heat exchanger core cross section in terms of height and width expressed as aspect ratio (H/W) depends on the space considerations where exchanger will be used and therefore may have values higher than, equal to or lower than unity depending upon the specific preference. Specifying \( F_{asp} \) makes,

\[ H = F_{asp} \cdot W \]

### 4.7. Core Dimensions
Frontal area of core is

\[ A_{fr} = H \cdot W \]

\[ W = \left( \frac{A_{fr}}{F_{asp}} \right)^{0.5} \]

### 4.8. Basic Surface Configuration
The surface configuration with given geometrical properties and known thermal and pressure drop characteristics intended to be used as exchanger core is selected at pressure this stage. The configuration designated as 9.1 – 0.737 – S by Kays and London.

### 4.9. Core Depth, Number of Tubes Rows and Total Number of Tubes
Core depth is prescribed at this stage. Then number of tube rows (\( N_r \)) that can be accommodated in the core is obtained by retaining integer part of the

\[ N_r = \frac{D}{P_t} \]

Similarly number of tubes per rows integer part of

\[ N_{tr} = \frac{W}{P_t} \]

Finally total number of tubes is obtained from

\[ N_t = N_r \cdot N_{tr} \]

### 4.10 Waterside Heat Transfer Coefficient
Waterside velocity through tubes comes from

\[ V_w = \frac{M_w}{(N_t \cdot A_w \cdot \rho_w)} \]

Mass velocity of water is given by

\[ G_w = V_w \cdot \rho_w \]

Waterside Reynolds number becomes

\[ \text{Re}_w = \frac{G_w \cdot d_w}{\mu_w} \]

And waterside Colburn Modulus is obtained from

\[ J_w = J_w (\text{Re}_w) \]

And waterside heat transfer coefficient then becomes

\[ h_w = J_w \cdot G_w \cdot C_{pw} \cdot P_{rw}^{-2/3} \]

### 4.11. Airside Heat Transfer Coefficient
Mass velocity of air is given by

\[ G_a = \frac{V_{fr} \cdot \rho_a}{\sigma} \]

Airside Reynolds Number is computed from

\[ \text{Re}_a = \frac{G_a \cdot d_a}{\mu_a} \]
And airside Colburn modulus is obtained from
\[ J_a = J_a (Re_a) \]

Then, Airside heat transfer coefficient is evaluated as
\[ h_a = J_a G_a C_{pa} P_{ra}^{-2/3} \]

### 4.12. Overall Heat Transfer Coefficient (U)
Waterside heat transfer area per unit of core volume \((\alpha_w)\) is obtained from
\[ \alpha_w = N_t P_t / (W.D) \]

Airside heat transfer area per unit of core volume \((\alpha_a)\) is known from surface configuration.

Then ration heat transfer area on waterside to that on airside \((F_{area})\), is given by,
\[ F_{area} = \alpha_w / \alpha_a \]

The overall heat transfer coefficient based on air side transfer area can now be computed from waterside and airside confidents, area ratio and airside and waterside fouling resistances. Tube wall thickness to heat transfer is comparatively very small and is therefore ignored. Thus,
\[ U = 1 / R_t \]

where,
\[ R_t = R_a + (R_w / F_{area}) + R_{fa} + (R_{fw} / F_{area}) \]

The component resistances are given by,
\[ R_a = 1 / (h_a, \eta_o) \quad \text{and} \quad R_w = 1 / h_w \]

and \(R_{fa}\) and \(R_{fw}\) are fouling resistances on airside and waterside respectively.

The overall fin effectiveness \(\eta_o\) is expressed as
\[ \eta_o = 1 - Y (1 - \eta) \]

where,
\[ \eta = \tanh (ml)/(ml) \]

With,
\[ m = 2h_a / k_f \delta_f \quad \text{And} \quad l = (P_t - t_w) \]

The value of \(Y\) comes from surface configuration. This completes the evaluation of \(U\).

### 4.13. Design Value of Heat Transfer Rate
The heat transfer rate obtained from selected geometry and operating variables is then given by,
\[ Q_d = U A \Delta T_m \]

where,
\[ A = \alpha_a V_c, \text{ with } V_c = A_{fr} D \]

### 4.14. Airside and Waterside Pressure Drop
Ignoring entrance, exit and flow acceleration losses, being comparatively small foe exchangers, airside core pressure drop is computed from,
\[ dP_a = (G_a 2f_a/2p_a) (A/A_c) \]

where,
\[ f_a = f_a (Re_a) \]

Waterside core pressure drop is evaluated from
\[ dP_w = f_w (L_w / d_w) (\rho_w V_w^2 / 2) \]
\[ f_w = f_w (Re_w) \]
4.15. Power Requirement
The total airside system pressure difference which includes the exchanger core, grills, etc., may be taken as double the core drop.

The air pumping power is obtained from,

\[ P_{fan} = \Delta P_{a, total} \left( \frac{M_a}{\rho_a} \right) / \eta_{fan} \]

where \( \Delta P_a \) is the total system pressure differential for the air circuit.

Similarly, the power required by the water pump is obtained from,

\[ P_{pump} = \Delta P_{w, total} \left( \frac{M_w}{\rho_w} \right) / \eta_{pump} \]

where \( \Delta P_{w, total} \) is the total system pressure differential for the water circuit.

5. DESIGN MODIFICATIONS PHILOSOPHY
The design rate of heat transfer \( Q_d \) is compared with intended heat duty \( Q_i \). If they match within the stipulated limit (say 100%), the design with considered geometric dimensions and the operating values is going to fulfill the desired objective and is therefore an appropriate choice for the task.

If the matching is not perfect, i.e. \( Q_d \) either exceeds or falls short of \( Q_i \) beyond the acceptable range, modification in core geometry and/or operating parameters values is in order and needs to be effected in an optimal fashion. The modification includes geometrical and operating parameters.

5.1. Geometrical Parameters
The first change to try towards achieving matched design may be make variation in the frontal area.

- If \( Q_d \) is higher than \( Q_i \), reducing core height comes as the first change. Varying height of the core while keeping its width at current value will minimize computational effort and should be attempted first. Doing so, will not change the total number of tubes mad waterside heat transfer coefficient will remain unaltered. Core size and heat transfer area are changed and thus new designed heat transfer rate is calculated and matched with intended duty, \( Q_d / Q_i \) falls within stipulated limit, design is perfect else modifying other geometrical parameters.

- Next geometrical variable worth for modification is the core width. During this exercise, the number of tubes per row will get altered but the number of rows will remain same. Waterside heat transfer coefficient and thus overall heat transfer coefficient altered. New designed heat transfer rate is calculated and matched with intend duty, if \( Q_d / Q_i \) falls within stipulated limit, design is perfect else modifying other geometrical parameters.

- Now modifying core depth, it will lead to change number of tubes, Waterside heat transfer coefficient and thus overall heat transfer coefficient.

- If geometric parameters variation does not yield the desired results, possible changes in operating variable levels may be attempted.

5.2. Operating Parameters

- Water temperature drop or outlet temperature of water may be varied, thus air temperature rise according to first law of thermodynamics (law of energy conservation) for moving towards matched design. Both water temperature drop and air temperature rise have a range whose value depends upon the exchanger rating, engine design and environmental conditions.

- Variation in frontal air velocity will also be one of the options for achieving perfect design, but it will change whole design of the exchanger.

6. RESULTS
A general computer program in “C” language programming has been developed for thermal design of cross flow heat exchanger. The results are displayed in the form of graphs when width and height of the core are considered equal for the sake of simplicity.

In the detailed case study the case of height not being equal to width is also covered.
The common conditions of all results which are shown in the graphs as follows:

\[ T_{ai} = 35 \, ^{0}\text{C}, \quad T_{wi} = 90 \, ^{0}\text{C} \text{ and } H = W \]

Impact of varying \( T_{wi} \) (85 \(^{0}\text{C}, 90 \, ^{0}\text{C} \text{ and } 95 \, ^{0}\text{C}) \) and \( T_{ai} \) (30 \(^{0}\text{C}, 35 \, ^{0}\text{C} \text{ and } 40 \, ^{0}\text{C}) \) is studied for the preferred set of \( Q, V_{fr}, dT_w \) and \( dT_a \) (30 kW, 10 m/s, 5 \(^{0}\text{C} \text{ and } 10 \, ^{0}\text{C})

**Figure 1** Variation of Core Width with Air Velocity

**Figure 2** Variation of Water Pressure drop with Air Velocity

**Figure 3** Variation of Fan Power with Air Velocity
Figure 4 Variation of Core width with Water Temperature drop

Figure 5 Variation of Water Pressure Drop with Water Temperature drop

Figure 1 & 2 are plotted when aspect ratio (F_{asp}) is one or H=W and inlet and outlet temperatures of both fluids are constant. In these figures different parameters are calculated with respect to different frontal velocity. In figure 1, core size is decreasing when air velocity increases because mass flow rate of air decreases as air velocity increase at all three heat duty.

In figure 2, water side pressure drop is increasing with air velocity, because velocity of water increases when number of tubes decreases due to reduction in W, although H decreases pressure drop slightly but effect of velocity raises pressure drop abruptly because pressure drop is directly proportional to square of velocity, thus small increase in air velocity, increases pressure drop drastically. In figure 3, fan power is plotted against air velocity, in this case all three curves are different because mass flow rates of air are different and fan power depends on flow rate of air. Figure 4 & 6 shows influence of water temperature drop on core size and pressure drop. Smaller dT_w (i.e. larger T_{wo}) makes water flow rate higher per tube casing h_w and U to increase, eventually core size decreases and dP_w increases. Due to reduction in core size P_f goes down for smaller dT_w.
REFERENCES


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