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MODELING OF TRANSIENT TEMPERATURE CHANGE IN FINNED TUBE HEAT EXCHANGER

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ABSTRACT

A transient one dimensional simulation of a finned tube heat exchanger was carried out to demonstrate the temperature variation and flow paths on the finned or tube sides. The finned tube heat exchanger model can also be used in cycle analysis where it is used in conjunction with other models. The objective of the simulation is to simulate the response of a finned tube heat exchanger to a specified temperature and pressure transient. Elements represent either the flow paths on the finned or tube sides were built up from an integrated network, or they model the heat transfer from the fluid on the tube side, through the tube wall and fins to the fluid on the finned side. Intersection between the finned side fluid path and the tube side fluid path is considered as a control volume. The thermal inertia of the solid tube wall and fin material and the fluid volume were taken into account in the modeling of transient simulations. It is shown that incorporating such simulation emphasizes that the tube wall temperature and the finned side air temperature stays constant for each tube pass through the heat exchanger.

KEYWORDS: Transient Temperature, Heat Exchanger, Finned Tube, Transient Model

INTRODUCTION

The finned tube exchanger model is a distributed model that is build up form an integrated network of one-dimensional elements. These elements represent either the flow paths on the finned or tube sides or they model the heat transfer from the fluid on the tube side, through the tube wall and fins to the fluid on the finned side. In figure 1, it is shown that each intersection between the finned side fluid path and the tube side fluid path is considered as a control volume and the typical element network of a single control volume is shown. The thermal inertia of the solid tube wall and fin material and the fluid volume are taken into account in the modeling of transient simulations.

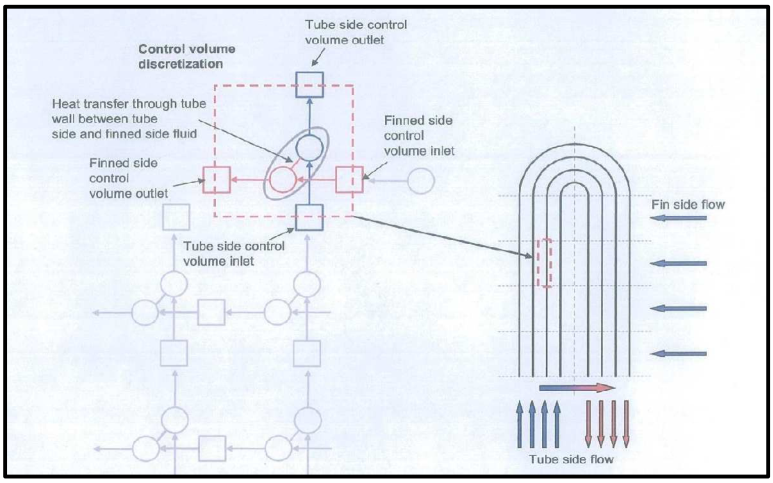

Figure 1: Finned Tube Heat Exchanger Model

The finned tube heat exchanger with two parallel circuits, ten rows and five passes are simulated. Both sides use air as working fluid. The inlet temperature and pressure of the air on the finned side is 150C^o and 350 KPa respectively. The inlet temperature and pressure of the air on the tube side is 30C^o and 350 KPa respectively. The finned side has a pressure drop of 1 KPa and the tube side has a pressure drop of 2.2 KPa. During the transient event the following events were specified:

- at 1 second the finned side inlet temperature is changed from 150 to 300 C^o .
- At 10 seconds the finned side inlet pressure is changed from 350 to 500 KPa.
- At 20 seconds the tube inlet pressure is changed from 350 to 500 KPa.
- At 25 seconds the finned side inlet pressure and temperature are changed from 500 KPa and 300C^o to 350KPa and 200 C^o respectively. At the same time the tube side inlet pressure and temperature are changed from 500KPa and 30 Co to 350KPa and 25C^o respectively.

Theory

In a heat exchanger, the two fluids namely: hot and cold, are separated by a metal wall. Under this condition the rate of heat transfer will depend on the overall resistance to heat transfer given by the equation:

$$\frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{x}{k A_{im}} + \frac{1}{h_o A_o} \tag{1}$$

Where

U_i = Overall heat transfer coefficient based on inner area [Kcal/hr m^2 C]

U_0 = Overall heat transfer coefficient based on outer area [Kcal/hr m^2 C]

Hi,h_0 = Inside and outside film heat transfer coefficients [Kcal/hr m^2 C]

A_i, A_0 = Inside and outside surface area [m2]

When viscous liquids are heated in a double pipe heat exchanger or any standard heat exchanger by condensing steam or hot fluid of low viscosity, the film heat transfer coefficient of the viscous liquid will be much smaller than that on the hot fluid side and will therefore, become controlling resistance for heat transfer. This condition is also present in case of air or gas heaters where the gas side film heat transfer coefficient will be very low (typically of the order of 0.01 to 0.0005 times) compared to that for the liquid or condensing vapor on the other side. Since, the heat transfer coefficient of viscous fluid or gas cannot be improved much, the only alternative is to increase the area available for heat transfer on that side so that its resistance to heat transfer can be reduced. To conserve space and to reduce the cost of equipment in these cases, certain type of heat exchange surfaces, called extended surfaces, have been developed in which outside area of tube is increased many fold by fins and other appendages.

Two types of fins, are in common use: longitudinal fins and transverse fins. Longitudinal fins are used when the direction of flow of the fluid is parallel to the axis of tube and transverse fins are used when the direction of the flow of the fluid is across the tube. Spikes, pins, studs or spines are also used for either direction of flow.

The outside are of a finned tube consists of two parts: area of fins and the area of bare tube not covered by the bases of fins. A unit area of fin surface is not as efficient as a unit area of bare tube surface because of the added resistance to the heat flow by conduction through the fin at itsbase. The expression for fin efficiencies can be derived by solving the general differential equation of heat conduction with suitable boundary conditions. Generally three boundary conditions are used:

- Fin of infinite length so that there is no heat dissipation from its tip, or in other words
- Temperature at the tip of fin is same as that of the surrounding fluid.
- Insulated tip. This condition even though cannot be realized in practice, but considering that the tip area is negligible as compared to the total fin area, heat dissipated from tip can be neglected and hence, dt/dx is assumed to be zero at the tip.
- Finite heat dissipation from the tip. Even though the assumption of insulated tip is invalid, most of the fins are treated under this category, and longitudinal fin efficiency for this case is given by the expression.

$$\eta_{fin} = \frac{\tanh(mL)}{mL} \tag{2}$$

Where

$$m = \sqrt{\frac{hC}{kA}} \tag{3}$$

h = film heat transfer coefficient from the fin surface [Kcal/hrm^2.C]

C = circumference of the fin [m]

K = thermal conductivity of fin material [Kcal/hr m C]

A = cross-sectional area of fin [m^2]

`From the above equation, it can be seen that the fin efficiency is a function of mL, and as the value of mL increases, the fin efficiency decreases. A reasonable value of fin efficiency will be around 50 to 75% for which mL should have a value between 1 and 2. If the fin height L should be sufficient (of the order of 5 to 8 cm), then it can be seen that the value of h should be around 10 to 20 which can be given by air in natural convection. The value of film heat transfer coefficient for any other liquid in natural convection, or any gas in forced convection will be much higher than 20. Thus, the given set-up is used for heat transfer to air in natural convection region.

Calculations were based on:

- Circumference of fin (C):

  C=2(w+b) (m)

- Cross-sectional area of fin (A):

  A = wxb (m^2)

- Fin area available for heat transfer:

  A_F= CxLxN (m^2)

- Tube area available for heat transfer in finned tube heat exchanger:

  AB = (l l D – Nb) xw (m^2)

- Total area of finned tube heat exchanger:

  $A_t = A_F + A_B$ (m^2)

- Heat given out by steam through finned tube heat exchanger (Q1):

  Q_1= (m_1 x)xλ (Kcal / hr)

- Heat given out by steam through bare tube (Q2):

  Q_2= (m_2 x)xλ (Kcal / hr)

  Where λ = latent heat of vaporization of water at steam pressure (Kcal/kg)

- Film heat transfer coefficient from bare tube (h):

$$h = \frac{Q_2}{(Ax\,\Delta T)}$$

(kcal/hrm^{2}C^o)

A=11DL (m^2)

$\Delta T=(T_{steam}-T_{ambient})$ (C^o)

- Amount of heat actually dissipated by fin:

$$Q_{fin} = Q_1 - (A_B\,xhx\,\Delta T\,(kcal\ / hr)$$

- Amount of heat that can be dissipated by ideal fin:

$$Q_{ideal} = A_F\,xhx\,\Delta T\,(kcal\ / hr)$$

- Observed value of fin efficiency:

$$\eta_{observed} = \frac{Q_{fin}}{Q_{ideal}}$$

RESULTS AND DISCUSIONS

The results are compared to a distributed model in XNet. The comparison between the temperature results are shown in figure (2). It can be seen that at 1 second when the finned side temperature increases to 300C^o, both the finned side outlet and tube side outlet temperatures increases gradually. At 10 seconds when the finned side inlet pressure is increased, the outlet temperature of both the finned side and tube side increases almost as quick as the specified transient event. This is due to the sudden increase in mass flow on the finned side. At 20 seconds the tube side pressure is also increased and this causes the finned side outlet and tube side outlet temperature to decrease suddenly. This is due to the sudden increase of mass flow on the tube side that is able to remove more heat received from the finned side. At 25 seconds both sides experience a pressure and temperature transient.

The quick response of the outlet temperatures are again due to the sudden change in mass flows. Thereafter the gradual decrease of the outlet temperatures were changed the outlet temperature on both sides only changed gradually due to the thermal capacitance of the tube wall solid material. It can be seen that the thermal inertia of the solid material is much larger that the thermal inertia due to the fluid volume by comparing the transient temperature response of the outlet temperatures when a step input was specified at the inlet.

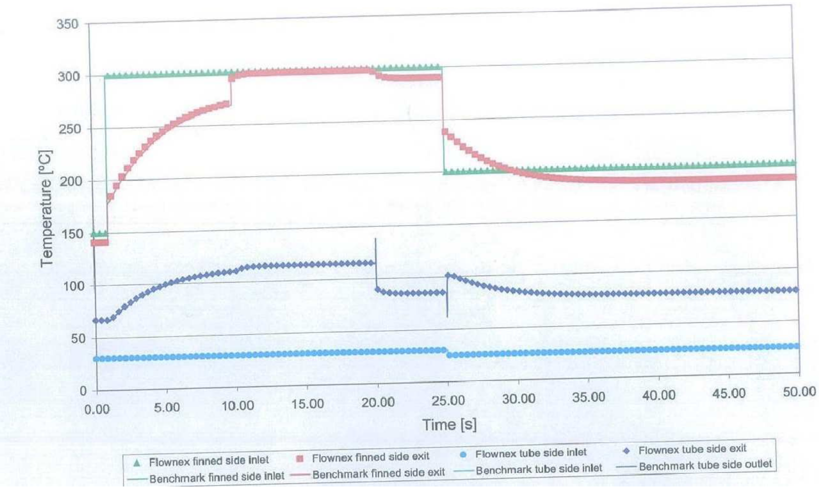

Figure 2: Finned and Tube Sides Temperatures for Transient Event

The pressure difference across the tube wall, at the bends at the end of each pass through the heat exchanger, during the transient event is shown in figure (3). The largest pressure difference across the tube wall is between 10 and 20 seconds and has a value of 15 kPa.

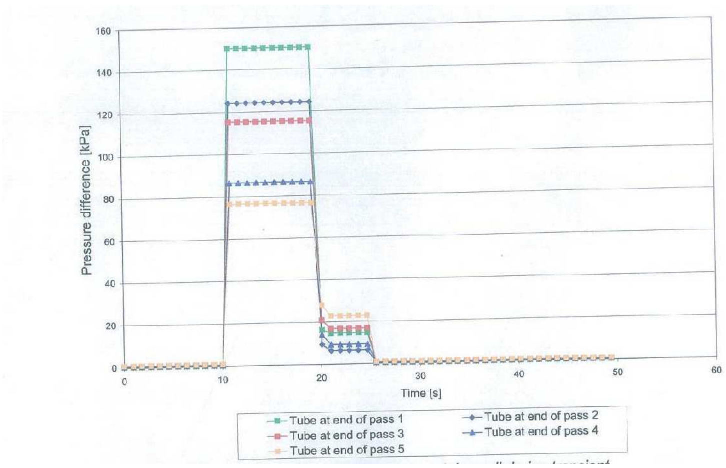

Figure 3: Pressure Difference across Tube Wall

The fluid temperature distribution on the fin and tube sides of the heat exchanger and the tube wall temperature for the last time step of the transient is shown in figure (4). It can be seen that the tube wall temperature and the finned side air temperature stays constant for each tube pass through the heat exchanger.

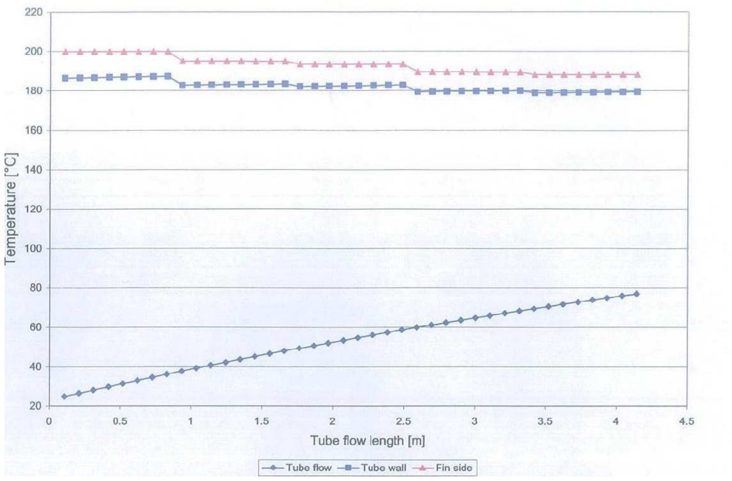

Figure 4: Temperature Distribution Along Tube Length

CONCLUSIONS

The thermal fluid transient simulation of a finned tube heat exchanger was discussed. The thermal inertia of the finned tube heat exchanger was visible in the temperature results. Depending on the design criteria, different variables can be monitored during the transient event, e.g. velocities, pressure difference between the fin and tube sides. From the temperature graph it can also be determined whether the heat exchanger would provide the design point outlet temperature for the specified transient even.

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