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Performance and Analysis of Fin Heat Exchanger

G.Vadivel¹, G.Vijayakumar², M.Sivakumar³, K.Pasupathi⁴

U.G Student, Department of Mechanical Engineering, Muthayammal Engineering College, Rasipuram,
Tamil Nadu, India^{1,2,3}

Assistant Professor, Department of Mechanical Engineering, Muthayammal Engineering College,
Rasipuram, Tamil Nadu, India⁴

ABSTRACT: To develop an innovative fin tube heat exchanger. Copper fin is winded over the inner tube of the copper tube like a helical spring, fin tube has high heat transfer than the plain tube in tube heat exchanger. Another concept is artificial tabulation, in which the laminar flow is converted into turbulent flow. It can be implemented in thermal sectors. The inner pipe is a common helically coiled tube heat exchanger and the outer pipe is a helically coiled flow distributor which separates the colder water from the significantly hotter bulk fluid of the tank to decrease the entrainment effect induced by natural convection.

KEYWORDS: Heat transfer enhancement, Plate-Fin and tube heat exchanger,.

I.INTRODUCTION

Generally used to exchange heat between a gas and liquid, fin-tube heat exchangers are widely used in chemical processing plants, power plants, and home appliances, including small air conditioners and refrigerators [1]. While early fin-tube heat exchangers primarily adopted plain fins, extensive research on fin shape caused substantial progress in the development of high performance fins to produce compact heat exchangers. Current compact fin-tube heat exchangers adopt a slit or louver fin if the working fluid is not significantly corrosive, such as in the case of a fume gas.

Fin shapes for compact heat exchanger design are continuously studied to address related global environmental concerns and improve the efficiency of the energy system. One notable subject is the incorporation of vortex generators in fins. Early vortex generator research focused on the boundary layer control of air foils [2]. Vortex generators create longitudinal vortices in the main flow to re-energize the boundary layer that is developed adjacent to an air foil and, in turn, delay or prevent flow separation. The flow mixing caused by vortices enhances heat transfer, and thus researchers became interested in applying vortex generators to heat exchanger fin designs. The shape of existing vortex generators includes a rectangular wing, rectangular winglet, delta wing, and delta winglet. Recent research on the effects of vortex generators on the performance of fin-tube heat exchangers focused on delta winglet vortex generators (DWVG) rather than on other shapes.

Torii et al. [4] speculated that this configuration accelerates the flow between a tube and its vortex generators as well as delays flow separation; thereby reducing the wake region behind the tube. The effects of vortex generators on the performance of the heat transfer surface have been investigated in recent decades. Jacobi and Shah [5] reviewed previous works until the early 1990s. They explained physical phenomena and vortex characteristics associated with vortex generators on a fin as well as introduced experiments and analysis on the performance of heat exchangers with vortex generators. They categorized active and passive methods of vortex generation. Active methods generate vortices using external energy, such as electric or acoustic fields, mechanical device, or surface vibration. Passive methods generate vortices through structures and additional fluids. Regardless of method, vortex generators enhance heat transfer and simultaneous pressure drop due to form loss. The vortex generator has the interesting characteristic of

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having a small pressure loss compared with other interrupted fin designs. As such, most previous works focused on quantifying the improvement of heat transfer performance and additional pressure drop characteristics.

II. EXPERIMENTAL SETUP

FIN TUBE IN TUBE HEAT EXCHANGER

The heat exchanger was constructed from copper and aluminum pipe and 1 mm thickness of copper fin. The outer tube of the heat exchanger had an outer diameter of 38 mm and a wall thickness of 3 mm. The inner tube had an outer diameter of 19 mm with wall thickness of 1.5 mm. the cross sectional and lateral view of the exchanger. The pitch of the coil is 5 mm with the diameter is 1 mm. The total length of the exchanger is 1000 mm.



It consists of copper pipe inside the aluminum pipe with the annular space of 16 mm. both the ends of aluminum are sealed with thermo seal (m-seal), the aluminum pipe is bored with 5 holes. In which water proof digital thermometers are fitted in 3 holes to measure the inlet, outlet, flow temperatures.

III. EXPERIMENTAL PROCEDURE

Experiments were conducted under steady state conditions with hot water (aluminum pipe) and tap water (copper pipe) as the working fluids. The flow rate in the copper tube was varied over a range of 0.0125 – 0.0145 liter/sec for a constant flow rate.

Temperature data was recorded at every second by digital thermometer. Accuracy of the digital thermometer is $\pm 1^\circ\text{C}$. The copper tube flow Rate was kept constant at 0.0125 liter/sec and variation in the annulus flow rate was made. At the inlet of the copper tube of the heat exchanger the cooling water temperature was 26–28 °C, and it rises by 4–5 °C at the outlet of the outer tube.

During the experiments the ambient temperature was 27–28°C, therefore, there was not much heat loss from the outer wall of the heat exchanger.

The cold water from the tap flow into the copper tube. The hot water from the tank flow into the aluminum pipe in counter flow direction to the cold fluid. Here the heat exchange took place from the hot fluid into cold fluid. We have used 1 liter beaker and stop watch to find the mass flow rate of cold fluid. We know that the temperature between hot and cold fluids in the heat exchanger varies from point to point so, in order to find temperature, three thermometers are fitted in the aluminum pipe.

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IV. CALCULATION OF HEAT TRANSFER COEFFICIENTS

LOGARITHMIC MEAN TEMPERATURE DIFFERENCE (LMTD)

The overall heat transfer coefficient U was calculated from the temperature data and the flow rates using equation and is given below:

$$U = Q / A \Delta T_{LMTD}$$

$$Q = U A T_{LMTD}$$

Where, A is the surface area;
 Q is the heat transfer rate; and

ΔT_{LMTD} is the logarithmic mean temperature difference, based on the inlet temperature difference, ΔT_1 and the outlet temperature difference ΔT_2 using the following equation of White (1984):

$$\Delta T_{LMTD} = (\Delta T_2 - \Delta T_1) / \ln (\Delta T_2 / \Delta T_1)$$

Heat transfer coefficients for the outer pipe h_o , and for the inner Pipe side h_i , were calculated using traditional “Wilson plots” as described by Rose (2004). Wilson plots have been used in other heat exchanger studies (e.g., Briggs and Young, 1969; Shah, 1990).

Wilson plots allow the heat transfer coefficients to be calculated based on the overall temperature difference and the rate of heat transfer, without the requirement of wall temperatures.

This method was chosen to avoid the disturbance of flow patterns and heat transfer while attempting to measure wall temperatures.

Wilson plots are generated by calculating the overall heat transfer coefficients for a number of trials where one fluid flow is kept constant and the other is varied. In this work, the flow in the outer tube was kept constant and the flow in the inner tube was varied for the different flow rates mentioned above. The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients by the following equation (White, 1984)

$$1/U_o = A_o / A_i h_i + A_o \ln(d_o / d) / 2\pi k L + 1/h_o$$

Where,

d_o , inner is the inner diameter of the outer tube;

d is the diameter of the inner tube;

k is the thermal conductivity of the wall;

L is the length of the heat exchanger.

After calculating the overall heat transfer coefficients, the Only variables in that are unknown are the heat transfers Coefficients. By keeping the mass flow rate in the outer tube constant, it is then assumed that the outer heat transfer coefficient is constant. The inner heat transfer coefficient is assumed to behave in the following manner with the fluid velocity in the inner tube, V_i :

$$h_i = C V_i^n$$

It was placed into and the values for the constant C and the exponent n were determined through curve fitting. The inner and outer heat transfer coefficients could then be calculated. This procedure was repeated for each outer flow rate.

The mean annulus heat transfer coefficient is defined as

$$h_{LMTD} = Q / A_s \Delta T_{LMTD}$$

with A_s the wall surface area defined as

$$A_s = \pi d L$$

The average rate of heat transfer to the fluid, \dot{Q} , is given as

$$\dot{Q} = (\dot{Q}_i + \dot{Q}_o) / 2$$

The heat transfer rate to the annulus fluid (\dot{Q}_o) and inner-tube fluid (\dot{Q}_i) is calculated with

$$\dot{Q}_o = m_o C_p (T_{oo} - T_{oi}) \quad \dot{Q}_i = m_i C_p (T_{io} - T_{ii})$$

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Fluid properties were calculated from formulas proposed by Popiel and Wojtkowiak [10] at the average bulk temperature, approximated for the annulus as $T_b = (T_{oi} + T_{oo})/2$

The logarithmic mean temperature difference for the annulus side is defined as

$$\Delta T_{LMTD} = (T_{iw} - T_{oi}) - (T_{iw} - T_{oo}) / \ln[(T_{iw} - T_{oi}) / (T_{iw} - T_{oo})]$$

The dimensionless Nusselt number for the annulus is defined as

$$NuD_{h,LMTD} = hD_h / k$$

where ,

k is the thermal conductivity of the annulus fluid

D_h the hydraulic diameter,

D_h is defined as $D_h = D_o - D_i$

Fluids flowing in annuli: the equivalent diameter

When a fluid flows in a conduit having other than a circular cross section, such as an annulus, it is convenient to express heat-transfer coefficients and friction factors by the same types of equations and curves used for pipes and tubes.

To permit this type of representation for outer heat transfer it has been found advantageous to employ an equivalent diameter De . The equivalent diameter is four times the hydraulic radius, and the hydraulic radius is, in turn, the radius of a pipe equivalent to the annulus cross section.

The hydraulic radius is obtained as the ratio of the flow area to the wetted perimeter. For a fluid flowing in an outer tube and d the flow area is evidently

$$\pi/4(d_o^2 - d_i^2).$$

but the wetted perimeters for heat transfer and pressure the wetted perimeter is the outer circumference of the inner pipe with diameter $d_{i,outer}$, and for heat transfer in annuli

$$\begin{aligned} De &= 4rh = 4 \times \text{flow area} / \text{wetted perimeter} \\ &= 4\pi(d_{o,inner}^2 - d_{i,outer}^2) / 4\pi d_{i,outer} \\ &= (d_{o,inner}^2 - d_{i,outer}^2) / d_{i,outer} \end{aligned}$$

In pressure-drop calculations the friction not only results from the resistance of the outer pipe but is also affected by the outer surface of the inner pipe.

The total wetted perimeter is $\pi(d_{o,inner} + d_{i,outer})$ and for the pressure drop in annuli

$$\begin{aligned} D_e &= 4 \times \text{flow area} / \text{Frictional wetted perimeter} \\ &= 4\pi(d_{o,inner}^2 - d_{i,outer}^2) / (4\pi d_{i,outer}) \\ &= 4\pi(d_{o,inner} + d_{i,outer}) / (4\pi d_{i,outer}) \\ &= (d_{o,inner} + d_{i,outer}) / d_{i,outer} \end{aligned}$$

V. CONCLUSION

Using of Fin heat exchanger increased the heat transfer rate and overall heat transfer co-efficient.

Without using of Copper winding,

| | |
|------------------------------------|------------------------|
| Heat transfer rate | Q=1054.8 W |
| Overall heat transfer co-efficient | U = 931.2 W/m²k |

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With using of Copper winding,
Heat transfer rate

$Q = 1406.8 \text{ W}$
Overall heat transfer co-efficient $U = 1429.67 \text{ W/m}^2\text{k}$

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