Enhancement Of Condensation Heat Transfer Coefficient Of Copper Tube In A Shell And Tube Condenser

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Abstract

The Condensation of steam on single horizontal copper tube in a shell and tube condenser has been investigated using experimental and theoretical methods in this study. The outer surface of the tube was modified by brazing it with a copper wire longitudinally and helically to enhance the heat transfer coefficient. The effect of varying the pitch of the helically brazed wire was studied with 25 mm and 35 mm pitch. Longitudinally Wire Brazed (LOWIB) and Helically Wire Brazed (HEWIB) Copper tubes with pitch 25 mm and 35 mm were found to increase the heat transfer coefficient by a factor of about 1.5, 2 and 1.2 respectively.

Key words: *Condensation, Enhancement of heat transfer coefficient, wire wrapped tube.*

"1. Introduction"

Achieving significant improvement in heat transfer coefficient by means of less complex, easily manufacture-able and economical solutions has always been a challenge for researchers in the field of condensation. Due to its wide variety of industrial applications such as automobiles, power plants, desalination plants and refrigerators, the research has reached greater levels to cater to the ever growing qualitative demands. The process of condensation of vapor to its liquid state is quite complex as it doesn't follow a simple scheme wherein a single mass of steam gradually converts into single mass of liquid without any interactions between the two phases [1]. This complex behavior further deteriorates the heat transfer process. Thus, continuous efforts are being made by many researchers to understand the variations in the condensation behavior in detail and reduce the resistances to the heat flow i.e. to improve the wall heat transfer coefficient of the condenser by making appropriate surface modifications according to the application.

Domingo [2] studied the condensation of refrigerant R-11 on several fluted, spiraled, roped and corrugated external surfaces and also in some cases internal surfaces of vertical tubes in comparison with smooth tube and found that flutes on external surface of tube increase the heat transfer coefficient by 5.5 times for a given heat flux and flutes on either sides of the tube render an additional 17% increase in heat transfer coefficient for a given overall temperature difference and water flow rate. Ravi Kumar et al. [3] investigated steam condensation over circular and spine integral fin tubes in comparison to plain tube and found that circular and spine integral fin tubes enhance heat transfer coefficient by 2.5 and 3.2 times respectively. Paisarn Naphon et al. [4] studied heat transfer and pressure drop characteristics by varying mass flow rates in horizontal double pipes with helical ribs of different height to diameter and pitch to diameter ratios and found that helical ribs significantly increase the heat transfer coefficient and also pressure drop. Thomas et al. [5] studied the effect of rectangular fins on external surface and detached promoter, twisted tape and fins inside a vertical aluminum tube on heat transfer and witnessed an increase in heat transfer coefficient by 2.5 to 4 times in case of fins on external surface and twisted tape inside the tube. Lemouedda et al. [6] numerically investigated the heat transfer performance of helical serrated finned tubes by twisting the outermost part of fin at different angles and by changing the number of fin segments per period and found that serrated finned tubes perform better than full fins. It was also found that serrated tubes with fins exteriors twisted by an angle in the range of 0 to 10° do not affect the heat transfer and increase in number of fin segments per period improves the heat transfer coefficient. Singh et al. [7] studied the heat transfer during condensation of steam over a vertical array of short horizontal integral fin tubes and developed a correlation between average heat transfer coefficient for n tubes and the first tube heat transfer coefficient. Belghazi et al. [8] investigated the local heat transfer coefficient of each row in a bundle of trapezoidal finned horizontal tubes during condensation of a pure fluid HFC 134a and several compositions of non-azeotropic binary mixture HFC 23 – HFC 134a and found that the heat transfer coefficient decreases significantly while using non-azeotropic mixture compared to pure fluid. It was also found that the while using non-azeotropic mixture HFC 23 – HFC 134a, the heat transfer coefficient increases in the first row due to the disturbance in diffusion layer by the condensate flowing from the upper rows. Takahiro Murase et al. [9] studied the condensation of steam, R113 and ethylene glycol on a horizontal wire-wrapped tube and reported enhancement ratios exceeding 3 for R113 and 2 for steam and ethylene glycol.

In most of the above cases, the heat transfer coefficient was improved by providing different types of fins or ribs on the external surface of the tube. The fins were either welded onto the tube surface or the tube was casted with integral fins. The methods requiring welding or casting of fins, ribs or machining grooves involve higher cost and manufacturing difficulty, moreover, such surface modifications end up increasing the weight and space requirements of the tube significantly. The aim of this work is to enhance the heat transfer coefficient of a copper tube by means of simple and economical surface modifications on the outside wall of the tube.

As shown in Table 1 most of the works were conducted employing tubes with diameter < 19mm, wire diameter < 3mm and pitch < 10mm. In this paper heat transfer characteristics of horizontal wire wrapped tubes with diameter > 19mm, wire diameter of 3mm and pitch > 10mm were studied.

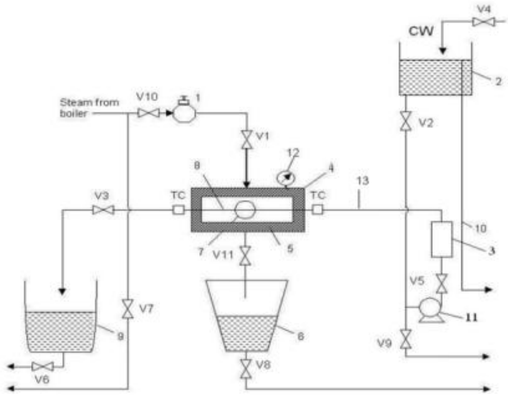
"Table 1. Comparison of earlier studies with present investigations"

Fluid	Reference	Outside diameter of the tube (mm)	Wire diameter (mm)	Pitch	Wire Material
Amonia	Rifert et al (1984)	-	1.5	8	-
R-11 and Ethanol	Fujii et al (1985)	18	0.3	2	-
Steam	Marto et al (1987)	19	1.6	4.62	-
Steam	Briggs et al (2003)	12.2	1	6	Steel
Steam	Present study	22	3	25	Copper

2. Experimental Setup

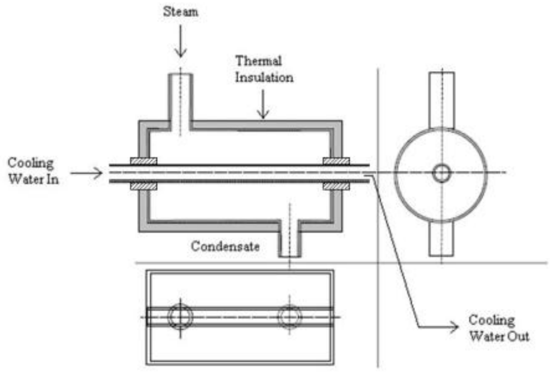
A detailed schematic diagram of the experimental set-up used in this study is shown in Fig. 1. Fig.

2 shows sectional view of the shell and tube condenser. The experimental set-up comprised of a shell and tube condenser (4) having 5 mm thickness, 325 mm shell diameter and 500 mm length. The shell is provided with a glass opening (7) to observe the condensation phenomenon.



(1) Pressure regulator, (2) Constant level cooling water tank, (3) Rotameter, (4) Shell and Tube condenser, (5) Thermal insulation, (6) Condensate vessel, (7) Viewing window, (8) Test section, (9) Hot water tank, (10) Overflow pipe, (11) Centrifugal Pump, (12) Pressure gauge, (13) Coolant flow tube TC – Thermocouple, CW – Cooling Water.

"Figure 1. Schematic Diagram of Experimental Set-up"



"Figure 2. Sectional View of Shell and Tube Condenser"

Outer diameter of the coolant flow tube used was 22 mm with 1 mm wall thickness. A copper wire of diameter 3 mm was brazed on the external surface of the tube. In order to prevent any loss of heat to the surroundings, all the pipe lines are insulated with two layers of Asbestos rope. Then the complete test section is covered with Glass wool insulation. The system is experimented with steam and without cooling water supply to estimate the heat loss. In order to measure the test section tube wall temperature, three cromel alumel thermocouples of 36 gauge and nominal diameter 1mm were fixed on the tube wall at the top, side and bottom positions. Additionally two thermocouples (TC) were used to measure the cooling water inlet and outlet temperature and the steam pressure inside the test condenser was measured with the help of pressure gauge (12). The saturation temperature corresponding to the measured pressure gives an appropriate cross check over the vapour temperature measured using the thermocouples. The steam temperature was measured at two points, one above test section and other below the condenser. Before installation, the thermocouples were calibrated for an accuracy of 0.1°C. At the saturation pressure of steam, the temperature of both the thermocouples became equal when the air inside the test condenser was replaced by steam. It was reliably learnt that the steam inside the consider is free from non condensable gases at this point. The leak proof test has been conducted above atmospheric pressure before the data acquisition.

3. Experimental Procedure

Steam was generated using a 280 liters capacity cylindrical boiler at the rate of 70 kg per hour. The steam was routed into the condenser from the top and its flow rate was controlled with the help of control valve (V1). Steam pressure was reduced to the desired level of 1.01325bar by a regulator (1). The steam temperature at the inlet to the condenser was maintained at 100°C during all the conditions. The cooling water at atmospheric temperature was circulated inside the tube with help of a centrifugal pump (11) from a constant level over head tank (2). The flow rate of the cooling water was controlled with the help of a flow control valve (V2) and the flow rate was measured with the help of a Rotameter (3). During the test, the cooling water was passed through the tube at different predetermined flow rates ranging from 8 to 28 liters per minute (lpm) in steps of 4 lpm. Coolant inlet temperature, outlet temperature and outside tube wall temperature were recorded during each flow rate. Experiments were conducted with different coolant tube configurations as mentioned below and Table (2) shows the test section dimensions.

- Plain cobber tube with 20mm and 22mm inner and outer diameter respectively as shown in Fig. 3.
- LOWIB tube with copper wire brazed longitudinally on the plain tube as shown in Fig. 4.
- HEWIB tube with copper wire brazed helically with 35 mm pitch on the plain tube as shown in Fig. 5.
- HEWIB tube with copper wire brazed helically with 25 mm pitch on the plain tube as shown in Fig. 6. An experimental uncertainty analysis was carried out to compute the extent of uncertainty involved in the condensing side heat transfer

coefficient h_0 using the procedure detailed in reference [10]. The maximum uncertainty in the determination of heat transfer coefficient was found to be in the range of 8 to 12%.

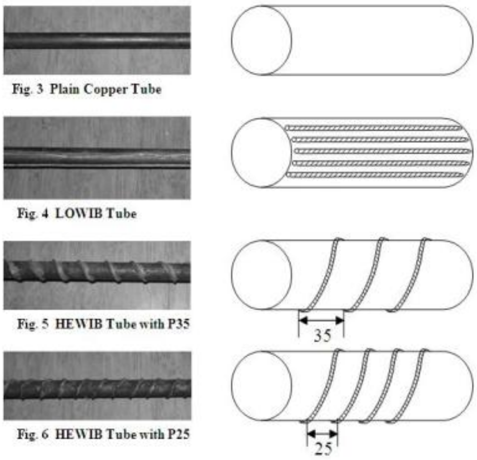


Fig. 3 Plain Copper Tube

Fig. 4 LOWIB Tube

Fig. 5 HEWIB Tube with P35

Fig. 6 HEWIB Tube with P25

"Table 2. Dimensions of the test section"

Parameters	Dimensions (mm)
Outer Diameter of the copper tube	22
Inner Diameter of the copper tube	20
Length of the tube	500
Thickness	1
Length of the shell	500
Dia of the shell	345
Thickness of the shell	5
Dia of copper wire	3
Pitches of helical rib	25, 35

4. Results and Discussion

Initially the experiments were conducted using Plain tube and the tube wall outside heat transfer coefficient (ho) was calculated using the experimentally measured coolant inlet temperature (Tci), outlet temperature (Tco) and Tube wall outside temperature (Two) using following Equations (1) and (2).

$$Q = m_c C_p (Tc_o - Tc_i) \qquad (1)$$

$$h_o = \frac{Q}{A_o (T_{sat} - T_{wo})} \qquad (2)$$

Further, the heat transfer coefficient was predicted by applying Nusselt's condensation theory using the relation (3).

$$h_o = 0.725 \left[\frac{k_f^3 g \rho_f (\rho_f - \rho_v) h_{fg}}{\mu_f D_o (T_{sat} - T_{wo})} \right]^{0.25} \qquad (3)$$

Subscript '*f*' represents condensate film i.e. water.

In order to verify the reliability of the experimental apparatus, the tube wall outside temperature T_{wo} was determined using the Modified Wilson Plot Method [11][12]. The Sieder-Tate Equation (4) was used to determine tube inside wall heat transfer coefficient which is required to be used in Modified Wilson plot method(MWP). As the Sieder-Tate equation is meant to be used for longer tubes (L/Di ≥ 60), a correction factor recommended by Al-Arabi [13] determined using Equations (5) and (6) was applied to the Sieder-Tate equation for the shorter tube [3].

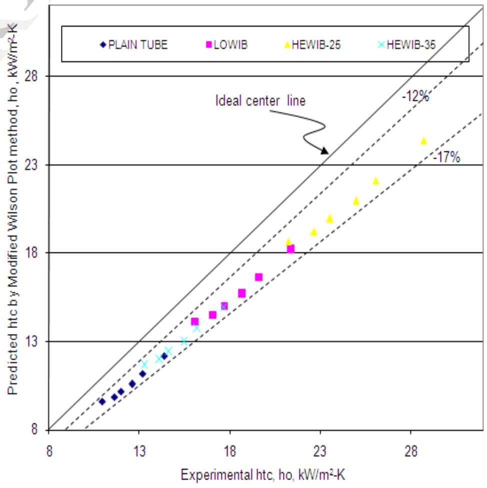
$$h_{i/longtube} = C_i \frac{k_i}{D_i} \mathrm{Re}_i^{0.8} \mathrm{Pr}_i^{0.333} \left[\frac{\mu_i}{\mu_{wi}}\right]^{0.14} \quad (4)$$

Subscript '*i*' represents internal fluid i.e. cooling water

$$SF = \mathrm{Pr}_i^{0.1667} \left[\frac{L}{D_i}\right]^{0.1} \left[0.68 + \frac{3000}{\mathrm{Re}^{0.81}}\right] \quad (5)$$

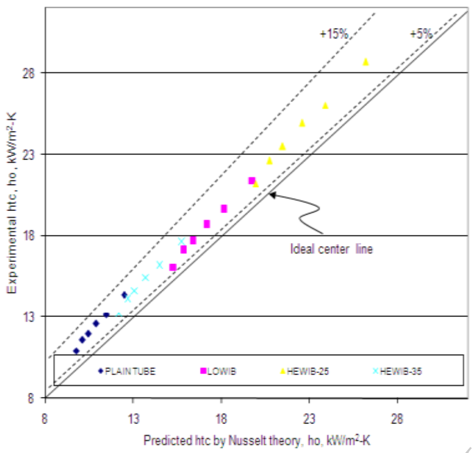
$$\text{For shorter tubes, } h_i = h_{i/longtube}\left[1 + SF\left[\frac{D_i}{L}\right]^{0.7}\right] \quad (6)$$

Fig. 7 shows the comparison between the experimental heat transfer coefficient and the one calculated using the tube outside wall temperature T_{wo} obtained using MWP method. It can be observed that the MWP method under-predicted the heat transfer coefficient by 12% to 17% which is in accordance with the other investigations in literature such as by Marto [14] and Ravi Kumar et al. [3].



"Figure 7. Comparison between the experimental heat transfer coefficient and the heat transfer coefficient predicted using Modified Wilson Plot (MWP) method"

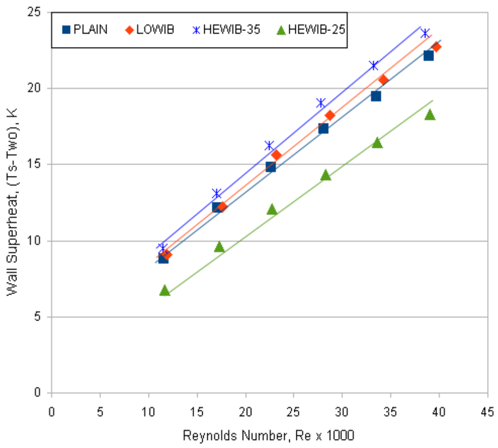
Fig. 8 shows the comparison between the experimental heat transfer coefficient and the one calculated using Nusselt's Condensation theory. It can be observed that the Nusselt's theory also under-predicted the heat transfer coefficient by 5% to 15% which is in accordance with the other investigations in literature such as by Mc Adams, W.H. [15] and Ravi Kumar et al. [3].



"Figure 8. Comparison between the experimental heat transfer coefficient and the heat transfer coefficient predicted using Nusselt's Condensation theory"

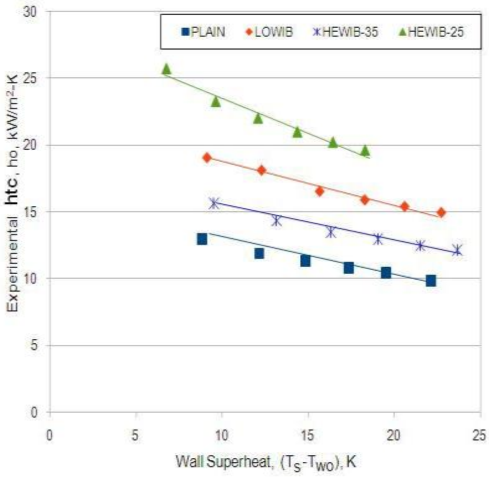
With the help of the above two observations shown in Fig. 7 and Fig. 8, it can be concluded that the experimental apparatus used to study the condensation phenomena is free from any irregularities as the results are in-line with the one observed by other investigators in literature and the deviations are within the acceptable limits.

The effect of change in cooling water flow rate on the wall superheat (T_s - T_{wo}) is shown in Fig. 9. The change in water flow rate or water flow velocity is represented by the dimensionless flow Reynolds number (Re). It can be observed that as the flow Reynolds number increases, the wall superheat also increases for a given vapor condensation temperature (for a given vapor pressure) which indicates a faster drop in wall temperature (Two) with increased water flow rate. Further, it can also be observed that the wall superheat is much lower for the case of HEWIB-25 compared to other tubes at all the water flow rates for a given vapor pressure indicating that the heat transfer performance of HEWIB-25 is better than other cases. The wall superheat with the case of HEWIB-25 was found to be 20 to 30 percent higher than that of Plain tube whereas LOWIB and HEWIB-35 showed very small change in wall superheat. The wall superheat (T_s-T_{wo}) versus Reynolds number (Re) lines were seen to be almost parallel to each other as shown in Fig. 9 indicating that the rate of change of wall superheat (Ts-Two) with Reynolds number (Re) is independent of the type of tube being used [3] for the given vapor pressure and within the range coolant velocities studied.



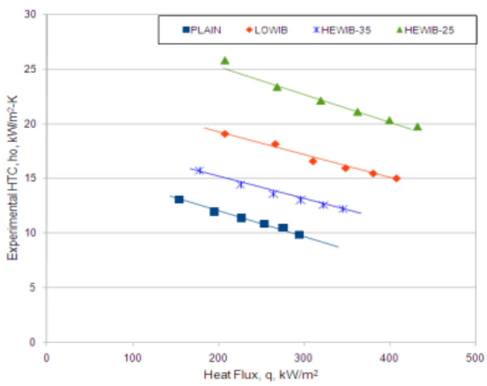
“Figure 9. Change in Temperature difference with Reynolds number”

Fig. 10 shows the comparison of change in heat transfer coefficient with wall superheat (T_s - T_{wo}) for Plain, LOWIB and HEWIB tubes. It can be observed that the heat transfer coefficient has improved due to the modifications on the external surface of the tube. The heat transfer coefficient with LOWIB, HEWIB-25 (Pitch = 25 mm) and HEWIB-35 (Pitch = 35 mm) is found to be 1.5, 2 and 1.2 times respectively higher than that of Plain tube. It can also be observed that with the increase in wall superheat which is due to the increase in coolant flow rate, the condensation heat transfer coefficient decreases. Such a decrease in heat transfer coefficient can be attributed to the increase in the resistance to the heat transfer. As the coolant flows faster, the condensation rate also increases which causes increase in condensate deposition on the tube leading to thickening of condensate film over the tube causing higher resistance to the heat transfer.



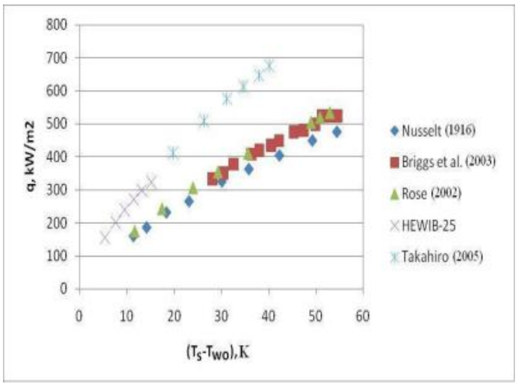
“Figure. 10 Change in Heat Transfer Coefficient with Temperature difference”

The change in condensation heat transfer coefficient with heat flux is shown in Fig. 11. It can be observed that heat transfer coefficient reduces with increase in heat flux which is due to increases in rate of condensation leading to formation of a thicker film around the tube augmenting the resistance to heat flow.



"Figure 11. Variation of Heat Transfer Coefficient with Heat Flux"

The values of experimental heat flux (q) obtained in the present investigation has been compared with other investigators in Fig. 12. It can be observed that the heat flux through the tube wall is better than plain tube by other investigators. It also found that the heat flux values observed in the present work are in good agreement with Takahiro Murase et al. [9].



"Figure 12. Comparison of experimental heat flux with those of other investigators and predicted by different models"

5. Conclusions

The surface modifications in the form of a thin copper wire brazed onto the external surface of the copper tube helped improve the wall heat transfer coefficient. The improvement in wall heat transfer coefficient using HEWIB-25 tube is found to be highest (2 times as compared to Plain tube) among the various modifications considered. The increase in heat transfer coefficient can be attributed to separation/ breakage of condensation film i.e. condensation phenomenon tending from film-wise condensation with Plain tube towards condensation with HEWIB-25 due to the surface intricacies/ discontinuities created by the thin helical braze of copper wire.

A simple surface modification of brazing a copper wire on the tube surface is found to be capable of yielding significant amount of improvement in heat transfer performance of the tube. The heat transfer coefficient is found to be increasing with decrease in helical pitch of the copper wire braze. This behavior observed in case of HEWIB-25 tube can be attributed the following two facts:

- The availability of higher heat transfer surface area enabling higher heat transfer rate.
- Increased surface intricacies/ discontinuities effecting onset of drop-wise condensation.

The rate of change of the Wall superheat (T_s-T_{wo}) with Reynolds number (Re) is found to be independent of the type of tube being used for the given vapor pressure and within the range coolant velocities studied. With the increase in coolant flow rate the increased condensate deposition rate leading to thickening of condensate film on the tube is found to deteriorate the heat transfer performance due to increase in resistance to the heat transfer.

6. Nomenclature

A_o - Tube outside surface area, m^2
CW Cooling Water
C_p - Specific Heat Capacity, kJ/kg-K
D_i - Inside diameter of the tube, m
D_o - Outside diameter of the tube, m
g - Acceleration due to gravity, m/s^2
htc - Heat Transfer Coefficient
HEWIB Helically Wire Brazed
h_{fg} - Latent heat of evaporation of water, kJ/kg
h_i - Tube inside heat transfer coefficient, kW/m^2-K
h_o - Tube outside heat transfer coefficient, kW/m^2-K
k - Thermal conductivity of condensate, kW/m-K
L - Tube length, m
LOWIB - Longitudinally Wire Brazed
MWP - Modified Wilson Plot
m_c - Cooling water flow rate, kg/s
Pr - Prandtl number
Q - Total heat transfer rate, kW
q - Heat Flux, kW/m^2
Re - Reynold's Number
SF - Correction factor
TC - Thermocouple
T_{ci} - Inlet Temperature of cooling water, K
T_{co} - Outlet Temperature of cooling water, K
T_{sat} - Vapor condensation temperature, K
T_{wo} - Tube wall outside temperature, K

Greek Symbols:
μ - Dynamic viscosity, kN-s/m2
μ_w - Dynamic viscosity at wall temperature, $kN\text{-}s/m^2$
ρ - Density, kg/m^3
ρ_v - Density of the steam, kg/m^3

7. References

[1] The Dynamics of condensation and vaporization, Thesis by Forrest Richard Gilmore, California institute of technology, Pasadena, California, 1951.
[2] Domingo, N., 1981, Condensation of R-11 on the outside of vertical enhanced tubes, , Engineering Technology Division, Oak Rindge

National laboratory, Fourth Miami International Conference on Alternative energy sources.

[3] Ravi Kumar, H.K. ,Verma, Bikas Mohanty, K.N. Agarwal., 1998, Augmentation of outside tube heat transfer coefficient during condensation of steam over horizontal copper tube. Int. Comm. Heat Mass Transfer, Vol.25, pp. 81-91.

[4] Paisarn Naphon, Manachai Nuchjapo, Jutarat Kurujareon., 2006, Tube side heat transfer coefficient and friction factor characteristics of horizontal tubes with helical rib. Energy Conversion & Management, Vol.47,pp. 3031-3044.

[5] David G. Thomas., Hayes P. H., 1970, Coefficient increased by combining rectangular fins on condensing side and twisted tape insert on flowing water side. Ind. Engg. Chem, Vol.62(2).

[6] Lemouedda, A., Schmid, A., Franz, E., Breuer, M., Delgado, A., 2011, Numerical investigations for the optimization of serrated finned-tube heat exchangers. Applied Thermal Engg.,vol. 31, pp.1393-1401.

[7] Singh, S.K., Ravi Kumar, Mohanty B., 2001, Heat transfer during condensation of steam over a vertical grid of horizontal integral fin copper tubes. Applied Thermal Engg.,vol. 21, pp.717-730.

[8] Belghazi, M. , Bontempsb, Marvilleta., 2003, Experimental study and modelling of heat transfer during condensation of pure fluid and binary mixture on a bundle of horizontal finned tubes. International Journal of Refrigeration, Vol.26,pp.214-223.

[9] Takahiro Murase., Adrian Briggs., Hua Sheng Wang., John W. Rose., 2005, Condensation on a Horizontal Wire-Wrapped Tube. Journal of Heat transfer, vol. 127/1207.

[10] Klines,S.J., McClintock. F.A., 1953, Describing uncertaninities in single sample experiments, Trans ASME, Mechnacial Engg., Vol.75,pp.3-8, (Referred from Holman,1984).

[11] Edwin H., Young and Dale E. Briggs., 1966, The Condensing of Low Pressure Steam on Vertical Rows of Horizontal Copper and Titanium Tubes, A.I.Chemical Engg., Journal., Vol.12, pp.31-36.

[12] Briggs, D. E., and E. H. Young. 1963, , Rept. No. 55,01592-149-T, Office Res. Admin., Univ. Michigan, Ann Arbor.

[13] Al-Arabi,M., 1982, Heat Transfer Engineering, Vol.3, Nos. 3-4, pp. 76-83.

[14] Marto, P.J., 1987, ASME Journal of Heat Transfer, Vol.110, pp. 1287-1305 .

[15] Mc Adams, W.H., 1954, Heat Transmission, McGraw Hill Book Company Inc. (Edited Book), 3rd ed., pp. 340.

[16] Vemuri, S. , Kim, K.J., 2006, An experimental and theoretical study on the concept of dropwise condensation. International journal of Heat and Mass transfer, vol. 49,pp.649-657.