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An Experimental Study on Convective Condensation of Steam in a Horizontal Tube at Low Pressure with Twisted Tape Inserts

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Abstract: An experimental study is performed to investigate the enhancement of convective condensation heat transfer coefficient of steam at low pressure in a horizontal tube with helical twisted tape inserts. The results indicate substantial enhancement in the condensation heat transfer coefficient or the tape inserts with in the range $2.5 < \frac{H}{D} < 10$. Dimensionless correlation for design purposes are provided to predict the condensation heat transfer coefficient in terms of Nu, Nusselt number as a function of π-groups Re_L, superficial liquid Reynolds, ϕ_L the friction multiplier of Lockhart and Martinelli, Pr Prandtl number of the condensate and, $\frac{H}{D}$ pitch to diameter ratio of the tape. The correlations could satisfy the data with an accuracy of ±4% validating the correctness of the criteria employed in the regression analysis.

1 Introduction

Efficient design of condenser is basically dependent on the thermal resistances at the outer and inner peripheries of the condenser tubes. Thus, the unending attempt of the investigators is to increase the magnitudes of the heat transfer coefficients both at inner and outer walls of the tube. Bergles [1] gave an exhaustive review indicating the available contemporary literature related to methods to enhance film heat transfer coefficients. Studies relating to enhancement of convective heat transfer are abundantly available in the literature [3-10] for a wide range of flow Reynolds of single-phase fluid flows with twisted tape inserts. Several correlations are available together with theoretical analyses. Limited number of studies can be traced in the literature with regard to condensation of vapors in a horizontal tube with twisted tapes (20,21). Luu [23] studied augmentation of in-tube condensation of R-113 with twisted tape insets. Royal and Bergles [20] presented data pertaining to augmentation of horizontal in-tube condensation with twisted tape inserts and internal finned tubes. The correlation proposed by them indicated marked deviation from the data. Ramakrishna et al [19] investigated convective condensation in a horizontal tube with tape inserts and their attempt to correlate the data did not prove to be very successful since the deviation of data is found to be more than ± 30% from the correlation.

Thus, the present experimental investigation supplements additional experimental data to the existing information in the literature. In addition, the present data are subjected to regression analysis to evolve a dimensionless correlation. The correlation predicts the results with a deviation less than ±4%. Further, the investigation is aimed at locating the

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correlation in the literature for $100 < Re_L < 1500$ which can be employed in design to predict the convective condensation heat transfer coefficient. Carey's analysis [12] indicates that under identical system parameters, the predictions from various correlations indicate marked deviation in magnitudes. Consequently, it might be necessary to reaffirm the correct and proximate value nearer to the given system conditions of the present study. According to the two-phase separated flow models of Lockhart-Martinelli [2] the regimes are designated as a combination of the superficial flow Reynolds of the vapor-liquid phases. The combinations are variedly defined as liquid-viscous, vapor- viscous [abbreviated with the suffix as vv], liquid viscous and vapor turbulent [vt] and the third one liquid turbulent and vapor turbulent [tt]. In this regard the data related to liquid-viscous and vapor turbulent [vt] are quite meager in literature. Thus, the present study is undertaken to provide information and answer to some of these uncertainties and ambiguities.

2. Review of correlations

Many investigated the process of condensation and some of the correlations often referred to in the literature are listed below.

Correlation of Akers [16]:

$$Nu=0.026\,Re_L^{1/3}\left[\frac{\rho_L}{\rho_v}\right]$$

where $Re_L = \dfrac{4m(1-x)}{\pi D \mu_L}$

Correlation of Ananiev [17]:

$$Nu = 0.021\,Re_L^{0.4}\left[\frac{\rho_L}{\rho_m}\right]$$

where $\dfrac{1}{\rho_m} = \dfrac{1-x}{\rho_L} + \dfrac{x}{\rho_v}$

Correlation of Kutateladze [13]:

$$Nu = 0.0387\,Re_L^{0.8}\,Pr_L^{0.4}\left[\frac{\rho_L}{\rho_v}\right]^{0.5}\left[\frac{\mu_v}{\mu_L}\right]^{0.1}$$

Correlation of Shah [18]:

$$Nu = 0.023\,Re_L^{0.8}\,Pr_L^{0.4}\left\{1+3.8\frac{1}{P^{*0.38}}\left[\frac{x}{1-x}\right]^{0.76}\right\}$$

where $P^* = P/P_{cr}$

Sarma et al [14]:

$$Nu = 0.023\,\phi_L\,Re_L^{0.8}\,Pr^{0.4}$$

These correlations are shown plotted in figure 1 and for the range of superficial Reynolds number: $100<Re_L<1200$ i.e.

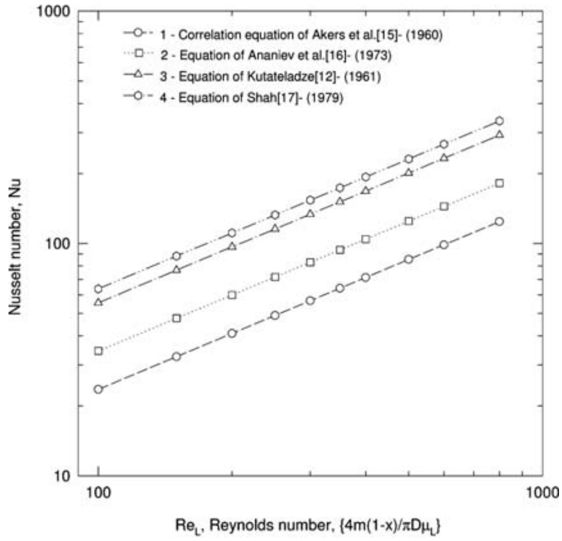

Fig1.Predicted values from various correlations

The magnitudes of Nusselt number calculated from these equations are found to vary substantially one from the other. To answer the question, which of these correlations is to be chosen as the one nearer to the present experimental study, the first phase of experimental study is performed in a tube without tube insertions. In addition, the first phase of study is aimed at establishing the reliability of functioning of the experimental set up for employing the same with the helical tape insertions in the tube during the second phase of investigation.

The detailed experimental procedure and description of the rig is given in [22]. However, a brief description at this location will be in order. The condenser tube serves as inner tube of an annular passage configuration with condensation occurring along the flow direction of the vapor in the tube. . Coolant water is admitted through the annular passage in counter direction to the flow of vapor. Temperature measurements are accomplished with the help of pre calibrated copper-constantan thermocouples welded at equidistant locations on the wall of the condenser tube with a provision to measure coolant temperatures at inlet and outlet locations. While partial condensation of steam occurs in the condenser tube at a given system pressure total condensation of the vapors is assured in the secondary condenser attached to the test section. This facilities the estimation of total flow rate of steam in the test section. Experiments are conducted such that the steam is under saturated conditions i.e. x = 1 at the inlet to the condenser. Such a physical situation of steam is ensured by throttling the steam in a specially designed nozzle from a high pressure of the package boiler to the condenser pressure. The temperature measurements at the inlet to the condenser tube is found to tally with the temperature corresponding to the system pressure noted with the help of a pressure gauge installed at the inlet. The exit dryness fraction at the outlet of the condenser tube is estimated from the balance

$$m h_{fg}[1 - x_0] = q_w \pi D L \qquad (1)$$

where m is the flow rate of steam and x_0 is the dryness fraction of steam at the outlet and h_{fg} the latent heat of condensation of steam at the system pressure. q_w is wall heat flux available at the periphery of the condenser tube.

The wall heat flux q_w is determined by computing the total heat taken away by the coolant circulating in the annular passage.

$$m_w\,[T_o - T_i] c_{pw} = q_w\, \pi\, D\, L \qquad (2)$$

Hence, the average overall heat transfer coefficient $h_{overall}$ is evaluated with the help of the equation.

$$h_{overall}\, \Delta T_m = q_w \qquad (3)$$

where $\Delta T_m = \dfrac{(T_s - T_i) - (T_s - T_o)}{\ln\left[\dfrac{T_s - T_i}{T_s - T_o}\right]}$

Similarly the heat transfer coefficient on the exterior wall of the condenser tube, h_o is by the expression

$$h_o \Delta T_1 = q_w$$

where $\Delta T_1 = \dfrac{(T_{w1} - T_i) - (T_{w2} - T_o)}{\ln\left[\dfrac{T_{w1} - T_i}{T_{w2} - T_o}\right]}$

Thus, h the average condensation heat transfer coefficient corresponding to the average dryness fraction of $x = (1 + x_0)/2$ is obtained from the relationship.

$$\frac{1}{h} = \left[\frac{1}{h_{overall}} - \frac{1}{h_o}\right]$$

The reliability and error analysis presented in [22] indicated that the maximum experimental error in the estimated heat transfer coefficient is not more than ±1.2%.

3. Calibration of the Setup

Thus, several runs with different flow rates of steam have been conducted and the condensation heat transfer coefficients are evaluated for each run. The results are cast in dimensionless form and depicted in figure 2 between

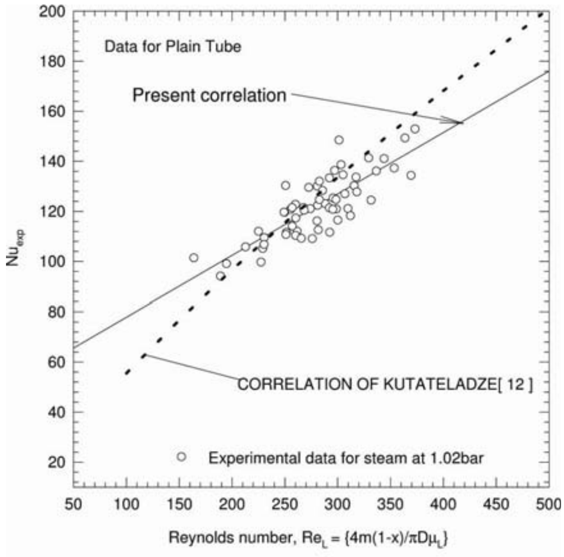

Fig. 2 Variation of Nusselt number with Reynolds number

$$\text{Nu vs } \mathrm{Re}_L = \frac{4m(1-x)}{\pi D \mu_L}.$$

It can be seen that the data fairly cluster with minimum deviation, on the same plot. The correlations of various investigations are also shown plotted. Evidently, the data fairly agree with the Russian correlation of Kutateladze [13]. Hence, for the working ranges and system conditions of the experimental setup, reliability in acquisition of data is ensured. Besides, the experimental data are subjected to regression analysis subject to the criteria as follows.

$$\mathrm{Nu} = F\,[\mathrm{Re}_L, \phi_L, \mathrm{Pr}_L]$$

where ϕ_L is the Lockhart-Martinelli friction multiplier. For liquid viscous and vapor- turbulent combination, ϕ_L is defined as

$$\phi_L = \left[1 + \frac{C}{\chi_{vt}} + \frac{1}{\chi_{vt}^2}\right]^{0.5} \qquad (5)$$

where $C = 12$.

$$\chi_{vt} = \left[\frac{f_l}{f_v}\right]^{1/2} \left[\frac{1-x}{x}\right] \left[\frac{\rho_v}{\rho_L}\right]^{0.5}$$

$$f_L = \frac{16}{\mathrm{Re}_L};\ \mathrm{Re}_L = \frac{4m(1-x)}{\pi D \mu_L}$$

$$f_v = \frac{0.046}{\mathrm{Re}_v^{0.2}};\ \mathrm{Re}_v = \frac{4mx}{\pi D \mu_v}$$

The resulting correlation of the present study is as follows

$$\mathrm{Nu} = 4.175\ \mathrm{Re}_L^{0.5}\ \mathrm{Pr}_L^{1/3}\ \phi_L^{0.1078} \qquad (6)$$

The correlation is shown in figure 3 with the experimental data scattering along the mean line with a deviation of ± 6%.

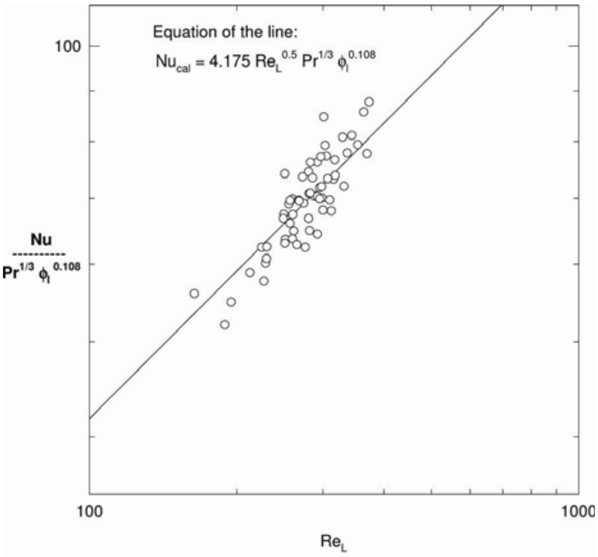

Fig. 3. Variation of predicted value of Nusselt with Reynolds number

4. Results with Tape Insertions

Three tapes with H/D = 2.5, 4, 10 are employed in the experimental study and the condensation heat transfer coefficients are obtained. The minimum tape configuration employed in the study is limited to H/D = 2.5 since tapes with less than this magnitude cannot be achieved with reasonable accuracy of uniform helical contours in the twist.

Results are respectively shown in figures 4, 5 and 6 for H/D = 2.5, 4 and 10. Solid lines representing the regression analysis are also shown plotted through the data of each plot and it can be seen that the data fairly cluster around these lines with minimum scatter. For assessing the relative enhancement in condensation heat transfer coefficients for varying magnitudes of H/D, the results are all comprehensively shown plotted in figure 7. It is evident that as H/D decreases, augmentation in condensation heat transfer coefficient can be achieved. All data are subjected to regression treating the independent term Nu as a function of the dependent system variables. Thus,

$$\mathrm{Nu} = F\,[\mathrm{Re}_L, H/D, \mathrm{Pr}, \phi_L] \tag{7}$$

$$\mathrm{Nu} = 7.175\ \mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3}\ \phi_L^{0.016}\,(1 + D/H)^{\,0.5681} \tag{8}$$

The resulting regression equation is shown plotted together with the data for all tapes. Evidently, the criteria under consideration yielded a successful correlation. To ascertain the accuracy, the equation is shown as generalized plot in figure 8 with experimental data. It can be seen that the data is correlated with an accuracy of ±4% by the equation. Besides another correlation as shown in figure 9 is evolved from the data to estimate the degree of augmentation or enhancement ratio.

$$\frac{\mathrm{Nu}_{H/D}}{\mathrm{Nu}_{\mathrm{Tube}}} = \lambda\ [\textit{Augmentation, Enhancement ratio}]$$

$$= 1 + 0.016\left[\mathrm{Re}_L\,\frac{D}{H}\right]^{0.6164} \tag{9}$$

For example, it can be seen from the equation that for Re_L = 800, H/D = 2.0, the enhancement ratio, λ = 1.642. Evidently, as H/D decreases the augmentation ratio, λ increases.

The relative merit of the present correlation is its ability to satisfy the data with a minimum deviation of ±4%.

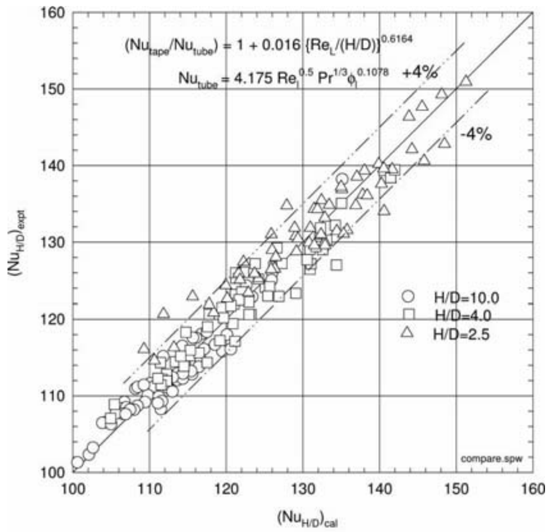

Fig. 9 Comparison between Calculated and Experimental Nusselt numbers for Twisted tape inserts

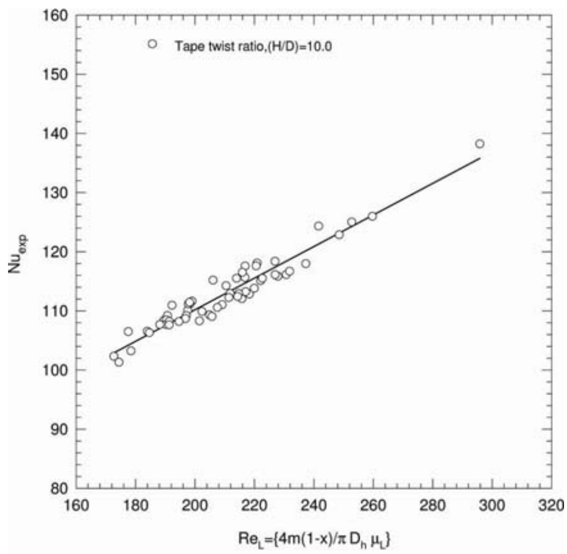

Fig.4. Variation of mean Nusselt number with Re_L for H/D=10

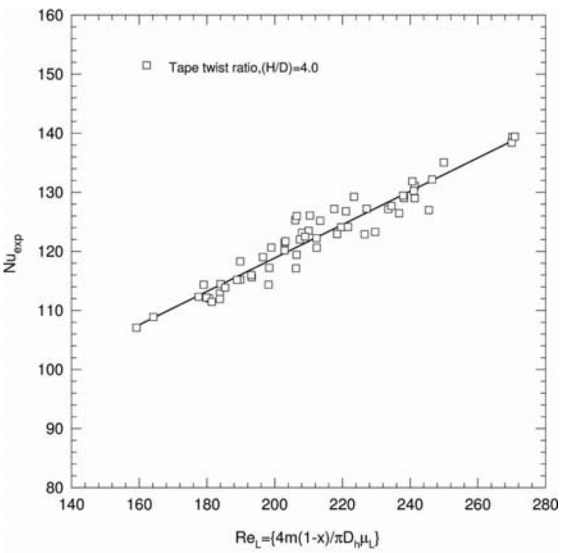

Fig.5 Variation of mean Nusselt number with Re_L for H/D=4.0

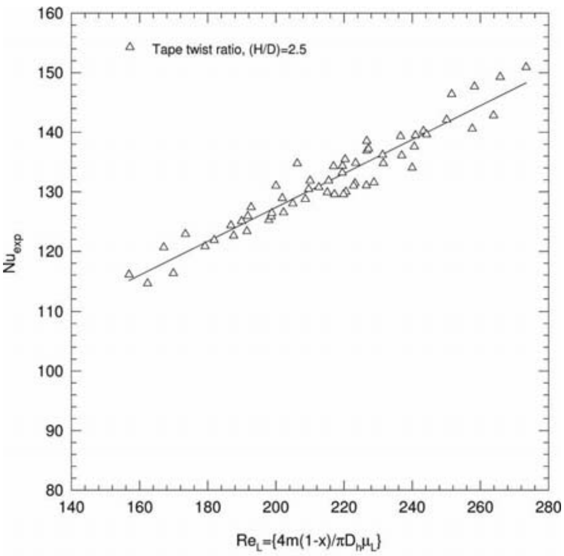

Fig.6 Variation of mean Nusselt number with Re_L for H/D=2.5

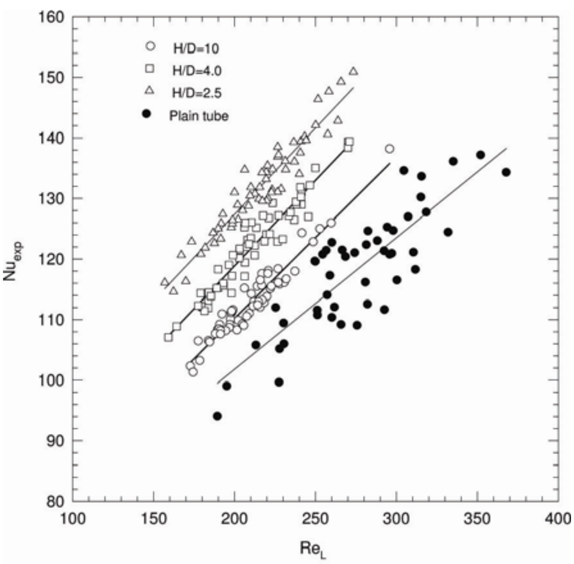

Fig 7. Variation of mean Nusselt number as a function of Re_L

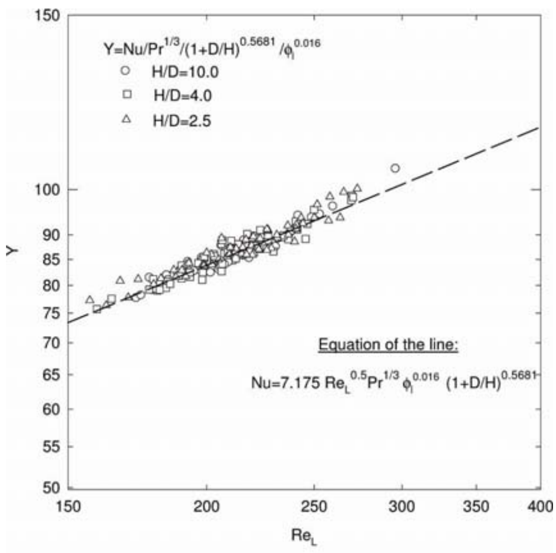

Fig.8 Generalized Plot: Y(as defined in plot) versus Re_L

5. Conclusions

The following conclusions can be made from the experimental study.

(i) For the following ranges of parameters: P = 1.05 – 2 ata, Re_L = 100 – 1000, Re_v = 9000 – 10^5, Pr_L = 2 – 3. Heat transfer coefficient can be predicted from equation (6). This equation gives more or less the same orders of the magnitudes as evaluated from the equation of Kutateladze [13].

(ii) The condensation heat transfer with tape inserts can be estimated from equation (8) and it is found advantageous to employ low values of H/D = 2.5 to achieve higher augmentation ratio.

(iii) The enhancement ratio λ for a given system conditions can be estimated from equation (9). Hence it is recommended that tapes with H/D = 2.5 can be employed to achieve better condensation heat transfer coefficient.

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Nomenclature:

C Constant in Lockhart-Martinelli equation

C_{pw} Specific heat of coolant at constant pressure, [J/Kg K]

D Diameter of the tube, [m]

f Friction coefficient

h_o Heat transfer coefficient on the coolant side [W/m K]

$h_{overall}$ Overall heat transfer coefficient [W/m K]

h Average condensation heat transfer coefficient [W/m K]
h_{fg} Latent heat of condensation [J/Kg]
k thermal conductivity of the condensate {W/m K]
L Length of the condenser tube
m Flow rate of steam [Kg/S]
m_w Flow rate of coolant [Kg/S]
P System Pressure [bar]
P_{cr} critical pressure of the condensate [bar]
q_w wall heat flux[W/m]
T_{w1} Temperature of the wall of the tube at inlet of the coolant
T_{w2} Temperature of the wall of the tube at the outlet of the coolant
T_i Temperature of the coolant at the inlet
T_o Temperature of the coolant at the exit of the test section
T_S Saturation temperature of the steam as system Pressure P
x_o Dryness fraction by weight at the exit of the condenser tube
x Average dryness fraction in the condenser tube i.e. $[1 + x_o]/2$
y Ordinate in figure 8

Dimensionless Parameters:
Nu Nusselt number, $[hD/k]$
Re_L Superficial Reynolds number of the condensate, $[4m(1-x)/\pi D\mu_L]$
Re_v Superficial Reynolds number of the wet steam, $[4mx/\pi D\mu\, v]$
Pr_L Prandtl number of the condensate
ϕ_L Lockhart -Martinelli friction multiplier
λ Augmentation- Enhancement ratio $[\mathrm{Nu}\ H/D/\mathrm{Nu}_{tube}]$
χ_{vt} Lockhart-Martinelli Physical property ratio parameter.

Greek letters:
μ Absolute viscosity [Kg/m S]
v Kinamatic viscosity [m^2/s]
ρ Density [Kg/m^3]

Subscripts:
L Condensate
v Vapor
i inlet
o outlet/Exit
w wall
H/D Value with tape insertion
$tube$ Value without tape insertion

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