A Study of the Effect of Discrete Ring-Type Protrusions on the Filmwise Condensation of Steam

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Abstract :

Investigation has been done to study the effect of ring-type discrete protrusions having square and triangular geometry on the vapor-side heat transfer coefficient of steam during condensation process on a vertical tube. Specimens having different pitches of protrusions were studied. Visual study of the formation of condensate were done. In general Triangular rib geometry shows higher value of heat transfer coefficient compared to the square ribs. In both the cases optimum value of rib spacing is found to be around one inch for a rib height of approximately one-tenth of an inch.

Nomenclature :

C, K = Constants
P = Pitch, inches
P = Steam pressure, absolute, psia
P_o = Atmospheric pressure, psia
R = Total resistance to flow of heat, hr. ft².F/Btu
U = Average value of over-all heat transfer coefficient, Btu/hr. ft².F
h_k = Heat transfer coefficient of metal wall, Btu/hr. ft².F
U_o = Value of U from the intercept on the ordinate of Wilson's plot, Btu/hr. ft²;F
h_s = Heat transfer coefficient of metal scale, Btu/hr. ft². F
h_v = Heat transfer coefficient of the vapor film Btu/hr. ft². F
h_w = Heat transfer coefficient of water film, Btu/hr. ft².F
V = Velocity of water, ft/sec.
X = Distance along the test pipe, ft.

Introduction :

The phenomena of filmwise and dropwise condensation of steam and the associated heat and mass transfer processes have been studied by many investigators like Nusselt (1), Eckert (2) and others. Dropwise condensation has a very high heat transfer coefficient compared to film-wise condensation (of the order of 4 to 8 times). One of the methods to approach drop wise condensation is to introduce roughness in the path of the condensate film. This roughness hinders the formation of condensate film thereby increasing the heat transfer coefficient.

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Researchers Medwell (3), Friedman (4), Thomas (5), by using different types of roughness, have observed an increase in heat transfer coefficient and correlated it with the surface coverage. Iqbal (6) examined the effect of repeated rib roughness outside a condenser tube and observed pulsating made of flow and an increase in heat transfer coefficient.

Though plenty of work is done, little is, however, known about the growth and development of the film, the effect of roughness spacing and its geometry on the heat transfer coefficient, optimum size, shape and spacing of the roughness for a particular condenser.

The present work is an investigation of the phenomena of film growth and flow along the length of a vertical condenser tube having ring type roughness of (a) square and (b) triangular cross-section by visual observations and to study the effect of the rib spacing on the surface heat transfer coefficient and the heat transfer to the flowing water.

Mathematical Model :

The process of condensation is extremely complex to be defined mathematically. It involves heat, mass and momentum transfer and hence is defined mostly by empirical equations. The problem can be simplified by assuming that the condensate forms a film on the surface of the tube. Thus the total resistance to heat flow is due to (i) the condensate film (ii) the metal wall (iii) the scale on the metal wall and (iv) the water film. In equation form this can be written as :

$$R = \frac{1}{U} = \frac{1}{h_v} + \frac{1}{h_k} + \frac{1}{h_s} + \frac{1}{h_w} \qquad (1)$$

From equation (1), the value of h_v can be obtained provided we know the values of U, h_k, h_s, and h_w. Here U may be obtained from the heat transfer rate and LMTD while h_s may be taken as infinite for clean tubes and h_k may be known from the pipe metal characteristics. h_w may be obtained from established empirical equations.

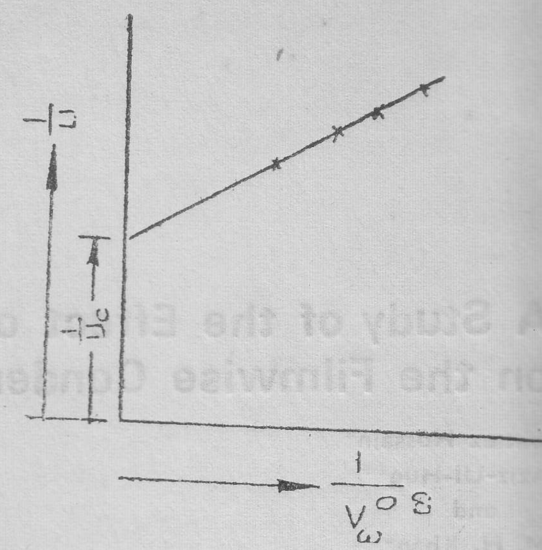


Fig. (i) Wilson's Plot.

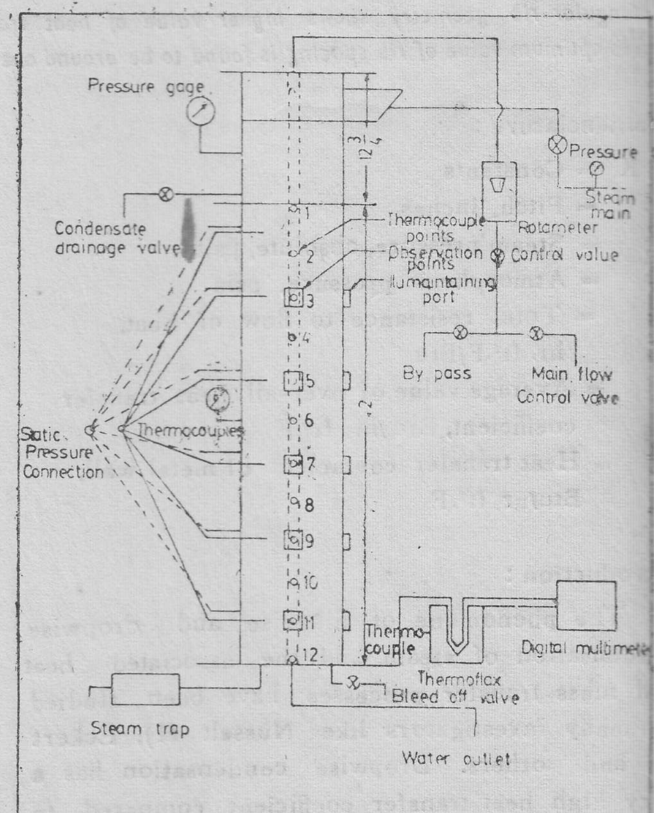


FIG (ii) SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET-UP

h_v can also be determined from Wilson's (7) plot as shown in figure (i), with the help of the following equation :

$$\frac{1}{U} = C + \frac{1}{KV_n} \qquad \text{(II)}$$

where $$C = \frac{1}{U_o} = \frac{1}{h_v} + \frac{1}{h_k} + \frac{1}{h_s}$$

All the calculations are based on equation (II) .

Experimental set-up and producer:

The experimental set-up, shown in figure (ii), consists of a pair of concentric tubes. Cold water flows vertically downwards through the inner tube having rings on the outside surface. Through the annular space a parallel flow of steam is maintained at a particular pressure and temperature. Rings having two types of geometry and different pitch were fabricated. Rib geometry and pitch are shown in figure (iii).

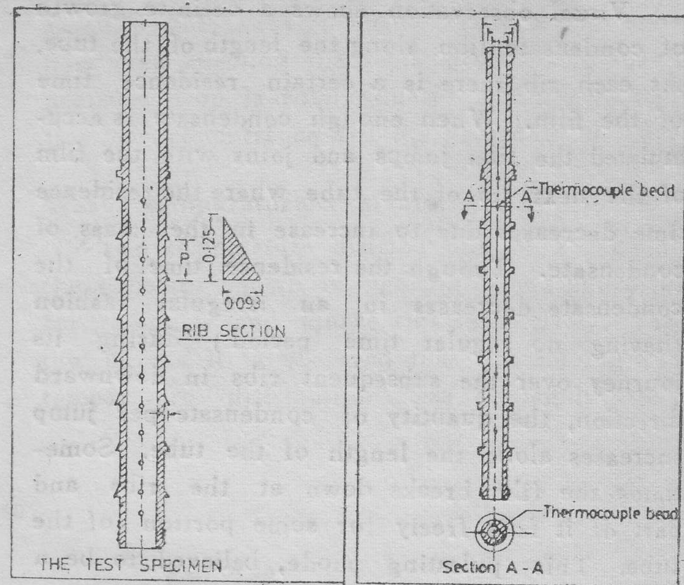
Pitch was varied from $\frac{1}{2}$ inch to 3 inch at an interval of $\frac{1}{2}$ inch.

Temperature rise of the flowing water along the tube was measured by 12 thermocouples and steam bulk temperature was measured by 6 thermocouples. Static pressure of the steam in the annular space was measured by 6 pressure tappings. Steam pressure was controlled by a control valve. Water flow rate was measured by weighing condensate directly.

Visual observations were made at five sections. At each of these sections, there were two windows placed at right angles to each other, one for illumination and the other for observation. A diagram of the section is shown in figure (iv).

Results and Discussions :

Results obtained from the experimental investigations are analysed and presented under the following heads :



FIG·(iii) SECTION OF SPECIMEN SHOWING RIB GEOMETRY & PITCH

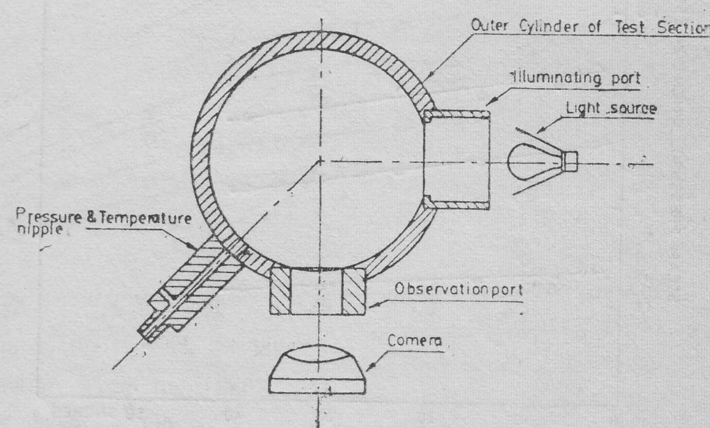


FIG (iv) Sectional view of Set-up showing illumination and Observation ports

Visual study of film for square and triangular ribs

Square rib :

Visual observation shows a definite growth of condensate film along the length of the tube. At each rib there is a certain residence time of the film. When enough condensate is accumulated the film jumps and joins with the film of the next rib of the tube where the residence time decreases due to increase in the mass of condensate. Though the residence time of the condensate decreases in an irregular fashion (having no regular time period) during its journey over the subsequent ribs in downward direction, the quantity of condensate per jump increases along the length of the tube. Sometimes the film breaks down at the ribs and part of it falls freely for some portion of the tube. This pulsating mode, believed to be a typical case for square ribs, is a major controlling factor in the heat transfer process.

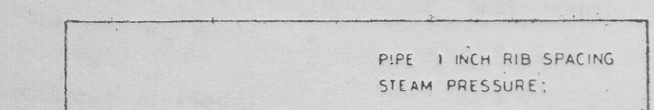


Fig. (v) Pressure drop Characteristic

Triangular rib :

The film had an increasing thickness along the tube upto the ribs. The quantity of the condensate flow increased along the length of the tube. The accumulated volume of the condensate at the ribs run down immediately. The residence time of the condensate on the ribs was almost negligible.

During its downward motion at the ribs a part of the condensate falls outside the tube surface directly into the bottom of the test section, while the remaining part joins the condensate flowing down. Thus breakage of the film at the ribs occurred, which is similar to that of the square ribs.

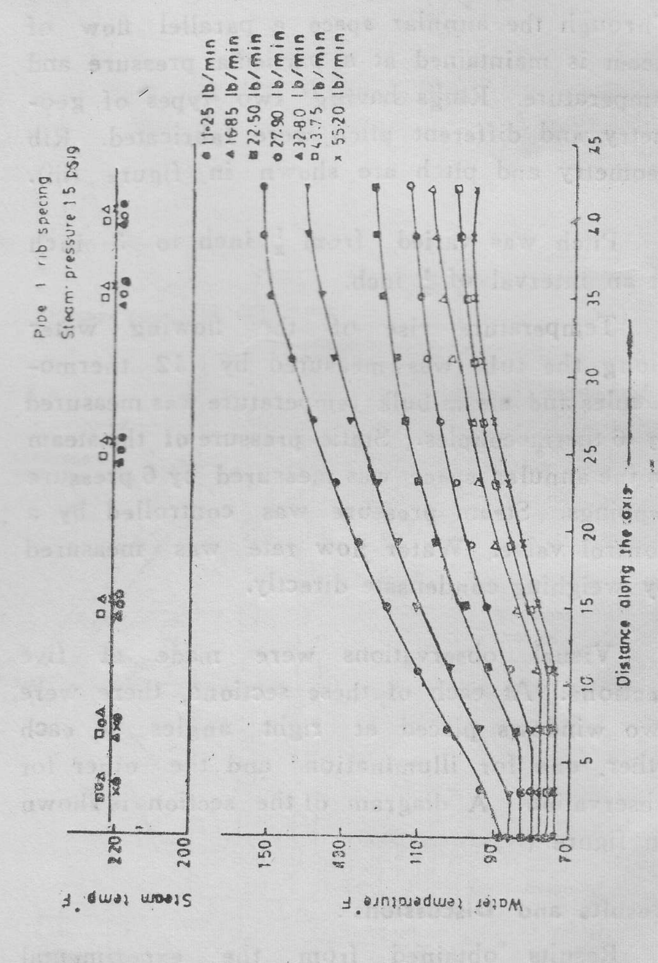


FIG (vi) TEMPERATURE DISTRIBUTION ALONG THE SPCEMEN TRIANGULAR RIB

At higher steam pressure the condensation rate increases, consequently the frequency of the film breakage at the ribs also increases.

Pressure drop characteristics on the steam side

Graphs plotted with $\frac{P}{P_0}$ and X as ordinate and abscissae respectively. (figure v), show the flow pattern of the steam with ribbedpipes at the three test pressures, The graphs show that the static pressure of steam decreased linearly along the flow direction. The condensate film thickness for the tubes and the steam boundary layer thickness on the inside surface of the outer tube were calculated and found to be very small compared to the width of the annulus and as such will not significantly reduce the steam flow area affecting the steam pressure. Thus the pressure drop is solely due to friction effect and this has concurrence with the nature of the curve.

Water temperature characteristics :

The temperature distributions of water along the tube are shown in figure (vi) and figure (vii) for triangular and square ribs respectively. From the curves it is observed that the cooling water temperature distribution has a definite shape.

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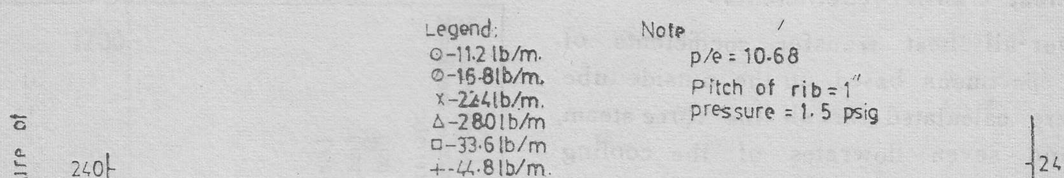

Fig(vii) Temperature distribution along the length of the specimen square rib

At lower flowrate of the coolant its temperature rise begins immediately after the entrance into the test section. After that it increases along a path of decreasing slope. But at higher flow-rates temperature of the cooling water remains constant upto a certain length of the specimen after the entrance and then it follows a path of exactly the same nature as those of the lower flow-rates. This nature of the curves is quite expected, because of the development of the thermal boundary layer upto a certain length of the tube and that larger length is required for the development of the thermal boundary layer at higher Reynolds numbers.

Vapor-side heat transfer coefficients :

The over-all heat transfer coefficients of the various specimens based on the outside tube diameter were calculated for all the three steam pressures and seven flowrates of the cooling water by using the experimental data. Then the vapor side heat transfer coefficients were calculated for all the pipes by using Wilson's plot. The average values of the vapor-side heat transfer coefficient has been plotted against the pitch in figure (viii) and (ix), for triangular and square ribs respectively. From these plots it is noted that the rib spacing of one inch gives highest values of heat transfer coefficient than the other rib spacings at all test pressures. The increased heat transfer coefficient for the one inch pitch tube is because of the fact that the breakage of the condensate film is most efficient at this rib spacing for these particular geometries of the ribs. The same figure also shows that the higher heat transfer coefficient occurs at higher steam pressures. It is very much likely to obtain higher heat transfer coefficient at higher pressures since higher pressure saturated steam is always associated with higher temperature producing a steep gradient. The values of the vaporside heat transfer coefficient are also plotted against pressure in figure (x) and figure (xi). It appears from these figures that triangular rib shows higher heat transfer coefficient for all the ribbed specimens compared to the square ribs. This can be explained from the residence time of the condensate on the surface of the tubes. In the case of square ribs, since film cannot flow immediately, it will have a higher residence time and film growth will occur which will affect the heat transfer coefficient. In the case of triangular ribs the condensate film cannot grow due to the inclination of the rib surface, as a result high value of heat transfer coefficient will be obtained. This situation involving residence time and film growth was also confirmed by visual observations.

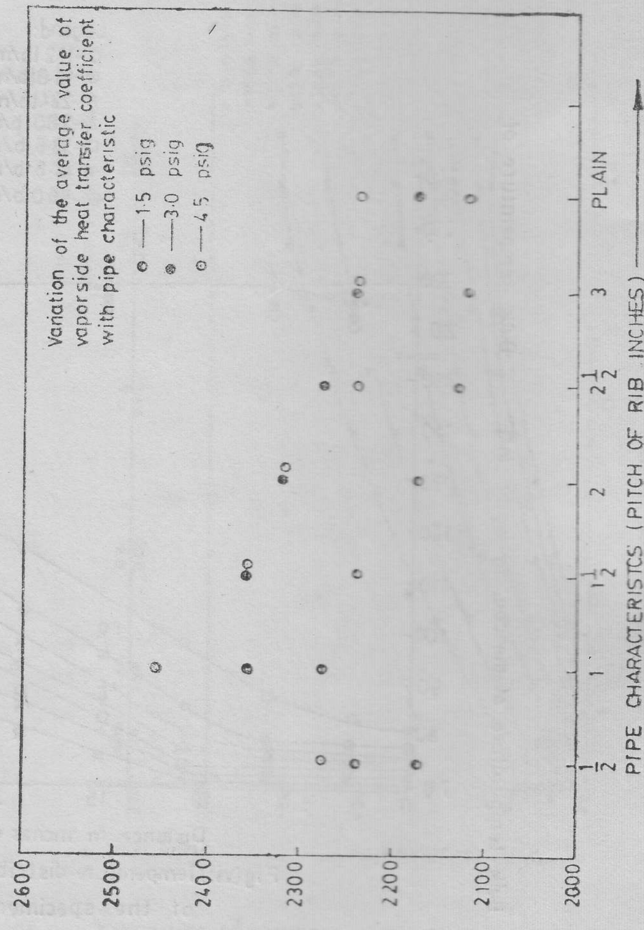

FIG.(viii) PLOT OF AVG. VAPOR SIDE HEAT TRANSFER COFFICIENT VS PITCH, TRANSFER RIB.

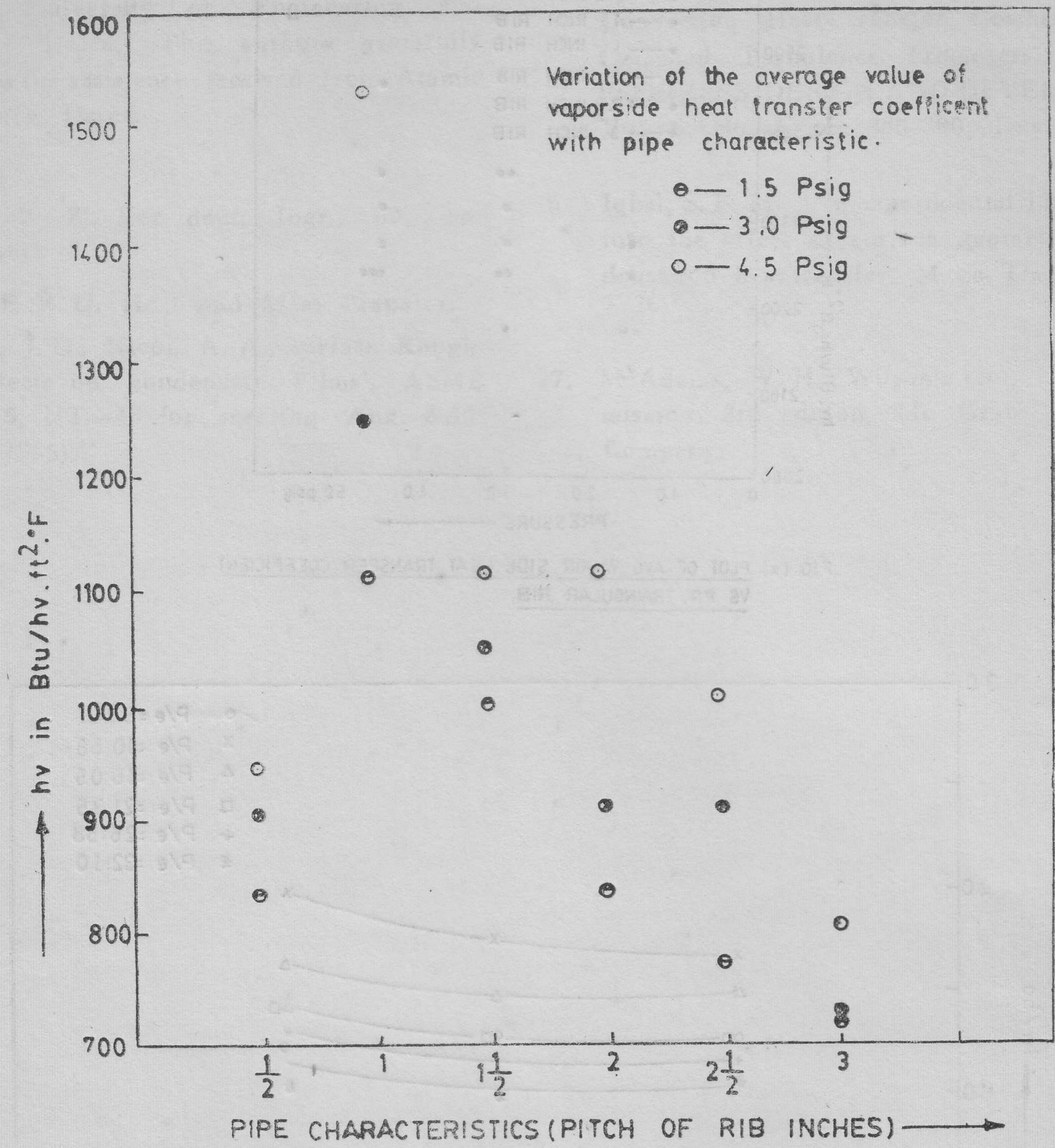

FIGURE (IX) PLOT OF AVG. VAPORSIDE HEAT TRANSFER COEFFICIENT VS PITCH, SQUARE RIB.

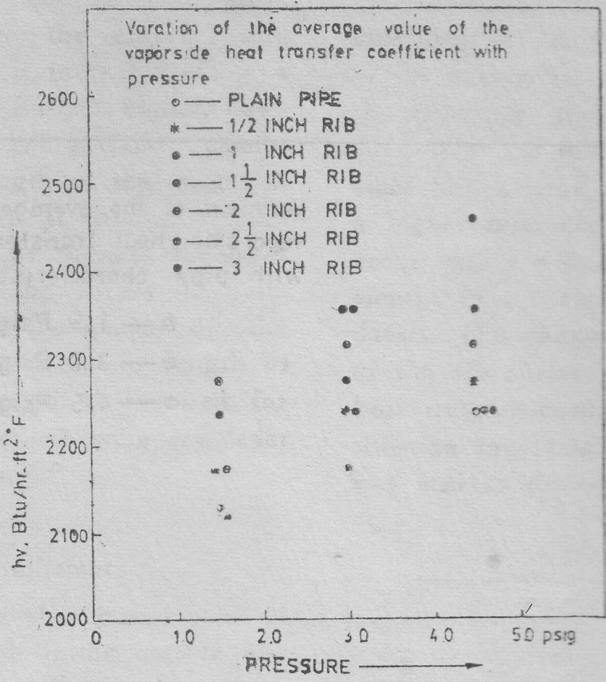

FIG (x) PLOT OF AVG. VAPOR SIDE HEAT TRANSFER COEFFICIENT VS. PR. TRAINGULAR RIB.

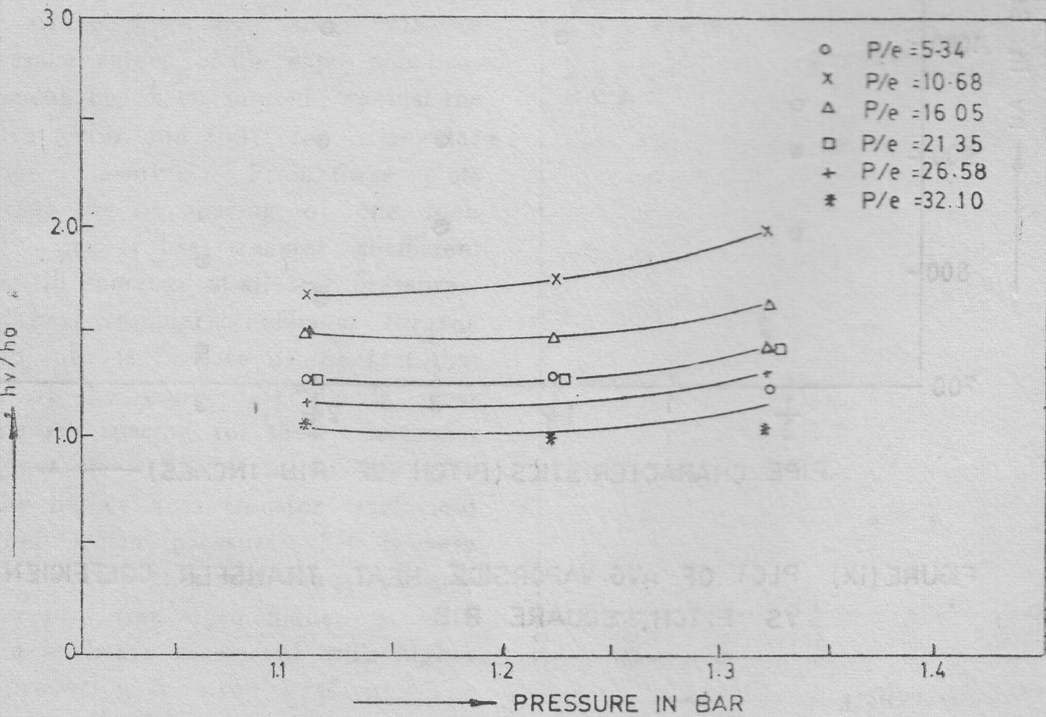

FIG (xi)—PLOT OF AVG. VAPOR SIDE HEAT TRANSFER COEFFICIENT VS. PRESSURE SQUARE RIB.

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