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THE GRAVITY ASSISTED HEAT PIPE WITH APPLICATION TO CONCRETE SHELL STEAM CONDENSERS

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Abstract—The manner of heat transfer in conventional steam condensers has been altered so that the cooling water and steam are separated by a concrete wall. The gravity assisted heat pipes which pass through the concrete membrane are used in transferring the heat between steam and water.

The heat pipe is evacuated and charged with pentane with an amount equal to forty percent of the volume of the pipe. Analytical and experimental investigations are done on the slope and on the evaporator length ratio of the pipe for maximum steam condensation rate. These investigations are carried out for two temperatures of water: (1) at 20°C cold water, and (2) at 63°C hot water. The optimum inclination angle is found to be twenty degrees for both cases. The evaporator length is two-fifths of the total length of the pipe in the cold water case; however, in the hot water case, the ratio is scattered between one-fourth and two-fifths of the total length.

In terms of the optimum slope and the evaporator length ratio, experimental and analytical results are found to be in good agreement. The magnitude of axial heat fluxes however are higher in analytical estimations. This deviation is attributed to the assumption of a non-flooded evaporator in theoretical investigations.

NOMENCLATURE

$\hat{A}$	heat transfer area [m^2]
A_o	inside cross sectional area of the pipe [m^2]
c_p	constant pressure specific heat [kJ kg^{-1} °C^{-1}]
g	gravitational acceleration [m s^{-2}]
h	heat transfer coefficient [W m^{-2} °C^{-1}]
h_{fg}	latent heat of vaporization [kJ kg^{-1}]
k	thermal conductivity [W m^{-1} °C^{-1}]
L	length of the pipe [m]
Nu_{ci}	inside condensation Nusselt number ($2h_{ci}r_o/k_l$)
P_m	mean pressure inside the pipe [kpa]
q	axial heat flow through the pipe, equation (1) [W]
r_o	inside radius of the pipe [m]
r_h	hydraulic radius [m]
ΔT	temperature difference, $T_w - T_s$ [°C]
ΔT^*	temperature difference, $T_{st} - T_{wa}$ [°C]
U	overall heat transfer coefficient [W m^{-2} °C^{-1}]
v	volume [m^3]
β	inclination angle of the pipe
δ	pipe wall thickness [m]
λ	evaporator length ratio (L_e/L)
ϕ	vapor subtented half angle, see Fig. 2
μ	dynamic viscosity [kg m^{-1} s^{-1}]
ρ	mass density [kg m^{-3}]
Ω	axial rate of change of condensate flow [m^3 m^{-1} s^{-1}]

Subscripts

bi	inside boiling
c	condenser
ci	inside condensation
co	outside condensation
e	evaporator
fc	forced convection
l	liquid
s	liquid–vapor interface
st	steam
v	vapor
w	wall
wa	water

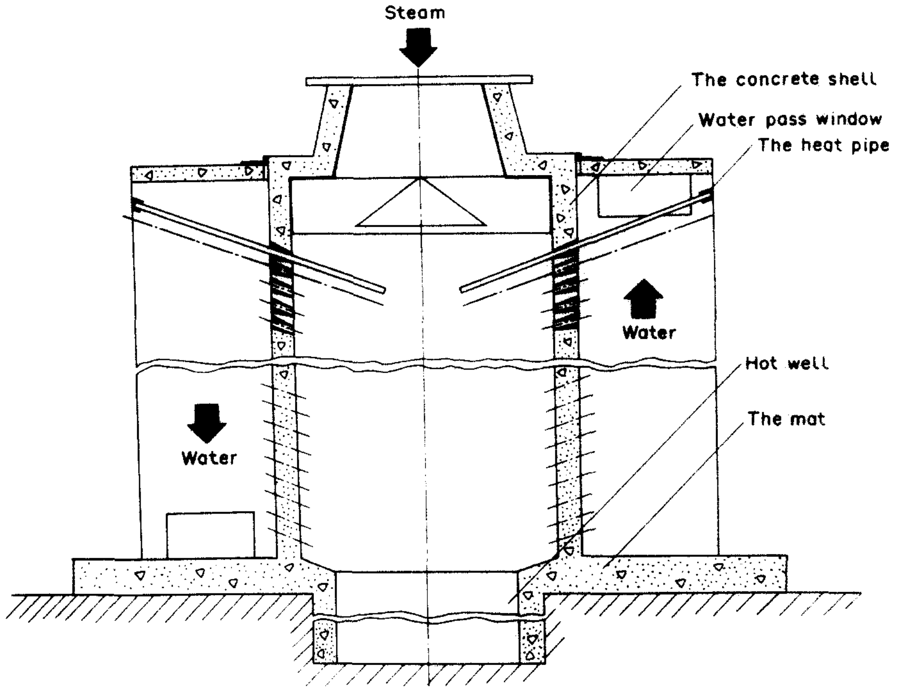

Fig. 1.

INTRODUCTION

In searching for simplicity in the design of conventional steam condensers, a solution to the thermal stresses resulting from the temperature differential between the tube bundle and the shell is a prime interest. It appears however than some of the drawbacks can be alleviated, and the use of expansion joints in the construction can be avoided by employing gravity assisted heat pipes as heat transferring elements. The essential components of such a modified design scheme are illustrated in Fig. 1. The cooling water flows from the top to the bottom around the circumference of a cylindrical shell. The steam condensed on the pipe surfaces is collected at the bottom of the condenser.

To render the contemplated configuration to experimentation and analysis, a first step is to study the heat transfer characteristics of a single heat pipe. The pipe working fluid is selected to be n-Pentane (C_5H_{12}). Within the considered temperature range, it can boil without requiring high vapor pressures. Furthermore, pentane has been tested and correlated extensively in pool and film boiling regimes [3]. In this preliminary design effort, the pipe is considered to be wickless. The prime purpose of the wick is to generate an adequate capillary pressure to transport the working fluid from condenser to evaporator region. In this study this has been accomplished by gravity effects. As shown in Fig. 2, in the evaporator region where film wise condensation of steam on the outer surface of the pipe takes place, pentane is boiled. Due to the density difference between liquid and

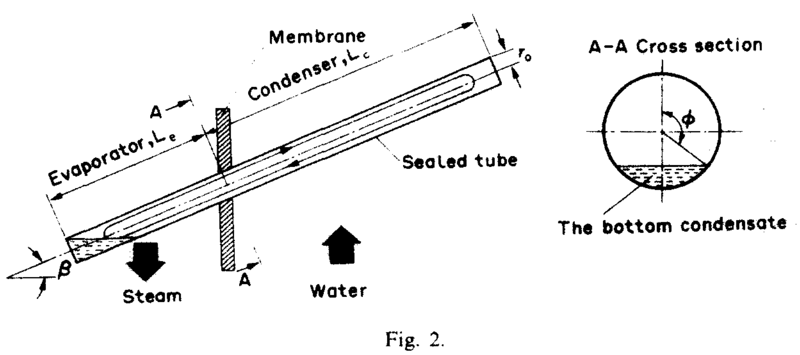

Fig. 2.

vapor, the vapor moves upward inside the inclined pipe. In the condenser region, the vapor is condensed along the periphery of the pipe and returned to the evaporator. Hence a closed loop of circulation of the working fluid in the pipe is established.

The objective of the present study is to investigate the effect of the pipe operating conditions; the inclination angle, β, the evaporator length ratio, L_e/L, and the temperature difference between warm and cold regions, ΔT^*, on the axial heat flux through the pipe. In both theoretical modelling and experiments, the pipe slope and the orientation are varied in a range which is compatible with steam condenser applications.

METHOD OF ANALYSIS

The physical model under consideration is a sealed cylindrical pipe, inclined from the horizontal position with an angle, β, and the evaporator and the condenser sections operating under the following assumptions:

1. The vapor and liquid phases are always assumed to be in thermodynamic equilibrium. Both phases are at the saturation temperature determined by the pipe pressure.
2. The bottom condensate layer inside the pipe is in laminar flow, i.e. $\mathrm{Re}_l < 3000$ where $\mathrm{Re}_l = 2\rho_l \Omega L_c/(\pi - \phi) r_o \mu_l$.
3. The effect of non-condensables (air) in the condensation phenomenon is neglected.
4. The transport properties of both phases are assumed to be constant and evaluated at the saturation temperature.
5. The evaporator and the condenser are surrounded by constant but different ambient temperatures, and are associated with uniform overall heat transfer coefficients between the liquid–vapor interface and the ambient.
6. The axial heat conduction through the pipe wall is neglected. Considering the above stated assumptions, the optimum evaporator length ratio, L_e/L, and the inclination angle, β, have to be determined. Accordingly, at a given temperature difference between warm and cold region, the axial heat flux through the pipe will attain a maximum.

With respect to the temperature difference between the two regions, the axial heat flow through the pipe can be expressed as,

$$q = (U\hat{A})\Delta T^*. \tag{1}$$

Since the energy transferred into the pipe in the evaporator section must be equal to the energy rejected out of the condenser section, the following relation holds:

$$\frac{1}{U\hat{A}} = \frac{1}{(U\hat{A})_e} + \frac{1}{(U\hat{A})_c}. \tag{2}$$

Where the overall conductance of both the evaporator and the condenser sections respectively are:

$$\frac{1}{(U\hat{A})_e} = \frac{1}{\lambda}\left[\frac{1}{\hat{A}_o h_{co}} + \frac{\delta}{\kappa \hat{A}_m} + \frac{1}{\hat{A}_i h_{bi}}\right] \tag{3}$$

$$\frac{1}{(U\hat{A})_c} = \frac{1}{1-\lambda}\left[\frac{1}{\hat{A}_o h_{fc}} + \frac{\delta}{\kappa \hat{A}_m} + \frac{1}{\hat{A}_i h_{ci}}\right] \tag{4}$$

Owing to the forced convective heat transfer on the outer surface of the condenser section and boiling in the pipe, the related heat transfer coefficients, h_{fc}, and h_{bi}, in equations (3) and (4) are not sensitive to inclination angle. However filmwise condensation which occurs on the outer and inner surfaces of the pipe depends on the inclination angle. Preliminary analysis shows that the heat transfer coefficient, h_{co}, due to condensation of steam on the outer surface of the evaporator is an order of magnitude greater than that of pentane condensing on the inner surface. Therefore film condensation inside the tube is the predominant factor in determining the slope for maximum heat flow rate.

Enclosure relations

The flow depth of the pentane condensate layer starts at zero at the top and reaches a maximum at the end of the condenser section. In determining the average condensation rate, the depth is assumed to remain constant and equal to the value calculated at the downstream half of the condenser section. Consider the dimensionless momentum equations of both vapor and liquid phases together with the energy-integral equation of the condensate. Chato [1] related the volumetric condensation rate with the flow characteristics of both phases. In a form applicable to the present problem, these relations are:

$$\begin{aligned}&0.5\left(\frac{L_c}{r_o}\right)\left(\frac{A_l}{r_0^2}\right)\sin\beta+0.000354\left(\frac{\rho_l}{\rho_v}\right)\left(\frac{\Omega^2L_c^2}{gr_o^5}\right)\left(\frac{L_c}{r_o}\right)\left(\frac{r_o^2}{A_v}\right)^2\sin\phi\\&-0.84\left(\frac{\Omega^2L_c^2}{gr_o^5}\right)\left(\frac{\mu_l}{\rho_l\Omega}\right)\left(\frac{r_o}{r_{hl}}\right)^2+0.000176\left(\frac{\rho_l}{\rho_v}\right)\left(\frac{\Omega^2L_c^2}{gr_o^5}\right)\left(\frac{A_l}{r_o^2}\right)\left(\frac{r_o^2}{A_v}\right)^2\left(\frac{L_c}{r_o}\right)\left(\frac{r_o}{r_{hv}}\right)\\&-0.25\left(\frac{\rho_l}{\rho_v}\right)\left(\frac{\Omega^2L_c^2}{qr_o^5}\right)\left(\frac{A_l}{r_o^2}\right)\left(\frac{r_o^2}{A_v}\right)-\left(\frac{r_o^2}{A_v}\right)\left(\frac{\Omega^2L_c^2}{gr_o^5}\right)=0\end{aligned}\tag{5}$$

$$\Omega=1.886\left[\frac{g(\rho_l-\rho_v)}{\mu_l}\right]^{1/4}\left[\frac{k_lr_o\Delta T}{\rho_lh_{fg}(1+0.68\epsilon)}\right]^{3/4}f(\phi,\beta)\tag{6}$$

where, ϵ, takes into account the subcooling effect on the condensate and is equal to $(c_{pl}\Delta T/h_{fg})$ $f(\phi,\beta)$ is a functional representation of the effect of condensate flow geometry and the pipe inclination angle on the condensation rate and defined as following:

$$f(\phi,\beta)=\left[\int_0^\phi\sin^{1/3}\phi\,\mathrm{d}\phi\right]^{3/4}(\cos\beta)^{1/4}.\tag{7}$$

The energy balance on the condensate layer gives the inner surface condensation heat transfer coefficient as

$$h_{ci}=\frac{\Omega\rho_lh_{fg}(1+0.68\epsilon)}{2\pi r_o\Delta T}.\tag{8}$$

The problem of estimating the maximum heat transfer coefficient is reduced to substituting into equation (8) the maximum volumetric condensation rates. For a given length ratio L_e/L, the maximum condensation rate occurs at a slope for which $f(\phi,\beta)$ is maximum. This in turn requires simultaneous solution of equations (5) and (6) for both condensation rate and the inclination angle at various vapor half angles of ϕ. The results for three different evaporator length ratios are shown in Fig. 3. The experimental observations reveal that when evaporator length is varied, the pipe pressure is increased or decreased depending on the tube movement. Increase of pressure causes the condensation rate to be increased. Due to the constant amount of working fluid in the pipe, the fluid circulation has to be increased. Since the motion is caused by gravity, the inclination angle must be increased. This behavior is in accord with the results of Fig. 3. The maximum values of Nusselt number, Nu_{ci}, slide to the right as the length ratio increases and occur at the inclination angles equal to 15, 18 and 21° for length ratios 0.166, 0.33 and 0.416 respectively.

In the evaporator section the essential parameters affecting the boiling heat transfer are two-fold: (i) The pressure effects, and (ii) The variation of wall superheat; $\Delta T=T_w-T_s$. Addoms [2] experimentally verified that at constant heat flux a decreasing required in the wall superheat for substantially higher pressures. Within the framework of this study, however, the pressure differentials are relatively small, i.e. in 3 bars range. The sole effect of pressure variation on boiling heat transfer can be neglected. The influence of wall superheat has been considered. The boiling of pentane on copper surfaces was experimentally studied by Berenson [3]. At various surface finish conditions, he graphically presented the heat flux variation with respect to wall superheat. Due to similar surface and fluid combination Berenson's results have been used in this study for estimating the boiling heat transfer coefficient at a particular pressure inside the pipe.

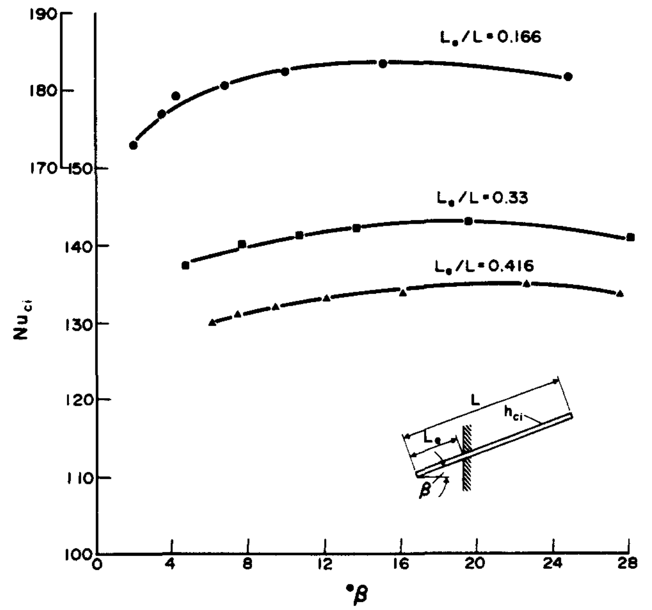

Fig. 3.

Using the results presented in Fig. 3 together with the analysis of Berenson, for a particular length ratio L_e/L, and inclination angle β, the overall conductance of the pipe can be determined by equation (2). The resulting axial heat flux distributions for inclination angles for 10 and 20° are presented in Fig. 4. The effect of inclination angle is not discernible for length ratios smaller than 0.3. Due to increase of L_e/L, decrease in condensation film coefficients (Fig. 3) is surpassed by the increase in boiling heat transfer coefficients. In Fig. 4, the heat flux approximately attains a maximum at a length ratio equal to 0.4.

THE EXPERIMENTS

The heat pipe

The pipe is made out of 1.27 cm inside diameter, 183 cm long, 2 mm thick copper tubing. The pipe was sealed at two ends by copper plugs. A spiral capillary 3 mm outside dia. copper tubing,

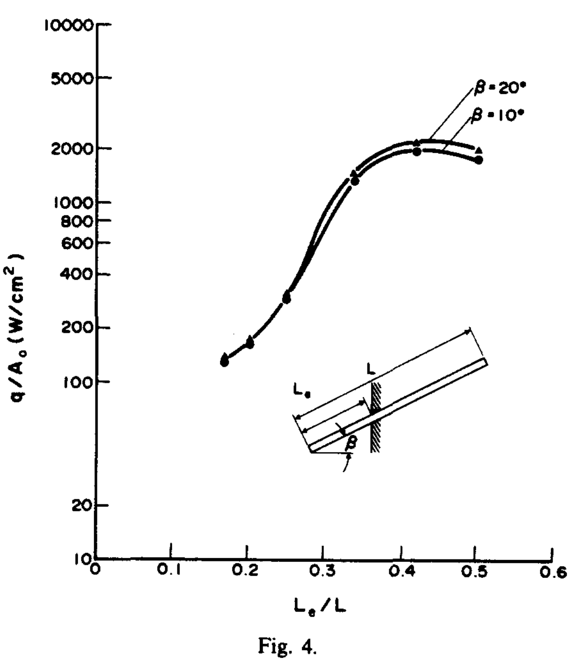

Fig. 4.

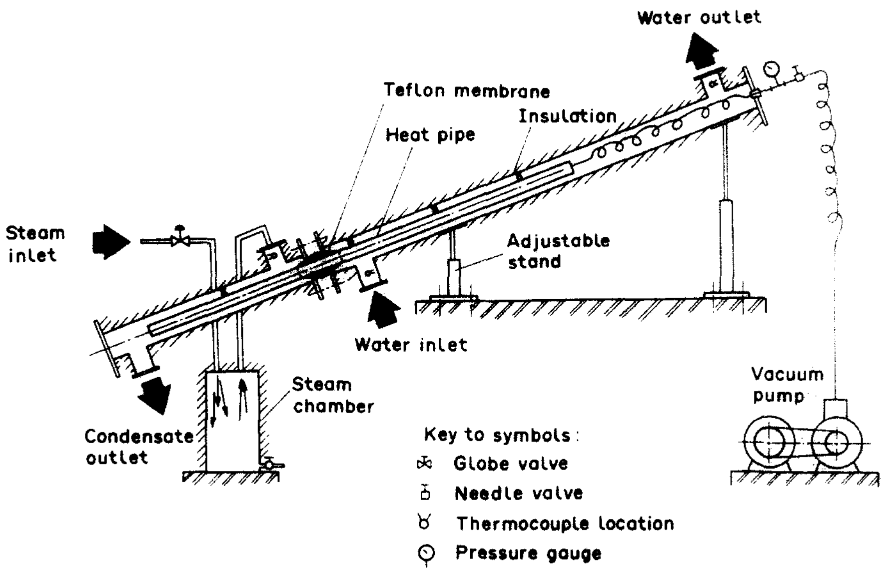

Fig. 5.

which is used for charging pentane, was silver soldered at one end of the pipe. Being spiral the capillary tubing is capable of extending or contracting when the evaporator length ratio of the pipe is altered. The other end of capillary tubing was connected to a Cenco type Megavac-R vacuum pump. On the line from the pipe end to the vacuum pump, type AR-96 compound vacuum pressure gauge and two miniature globe valves were placed. These valves in turn link the pipe either with a vacuum pump or with the pentane charging line.

Prior to experimental runs the pipe assembly was tested for any possible leaks. After an evacuation process of the pipe to 10.7 mm-Hg absolute pressure, for 72 h duration no pressure change was recorded and it was concluded that there was no leak.

The flow system

The basic flow apparatus as depicted in Fig. 5 consists of an outer shell which is 3.8 cm outside diameter, 275 cm long steel tubing and entirely surrounds the pipe. The shell was cut into two portions with one-third ratio. The 92 cm portion of the shell is the steam condensing region (the warm region), and the other 183 cm long section is the cooling water region (the cold region) of the apparatus. The length difference of 92 cm between the outer shell and the pipe provides evaporator length ratios varying from 0 to 0.50 in the experiments. A conical teflon membrane which has a tip to tip 3.8 cm diameter separates the two regions of the system. The separated regions of the outer shell are tightened to each other through flanges. These flanges are capable of sliding on the teflon surface. Consequently it is possible not only to press the teflon membrane circumferentially and hold the pipe at any desirable position, and also obtain a leak tight seal between the cold and the warm regions.

Dry saturated steam at a pressure of one bar was taken from central heating system of the Institute. To separate the water particles from steam, it first enters to a steam chamber, then flows into the system. The condensate at the outlet of the warm region is collected in a measuring burette. All parts of the system and the steam supply line including the steam chamber are insulated with 2.5 cm thick glass wool covered with an aluminum foil. A schematic of the experimental apparatus and instrumentation is presented in Fig. 5.

Instrumentation

Temperatures were measured at the inlet and outlet of water, at the entrance of steam, and at four points along the pipe surface. In order to avoid the contact of water or steam to the thermocouple beads by which the pipe wall temperatures are measured, 9 mm dia. cylindrical rubber corks are located to the clearance between the outer shell and the heat pipe. The thermocouple junctions are housed in stainless steel wells which pass through these corks and the

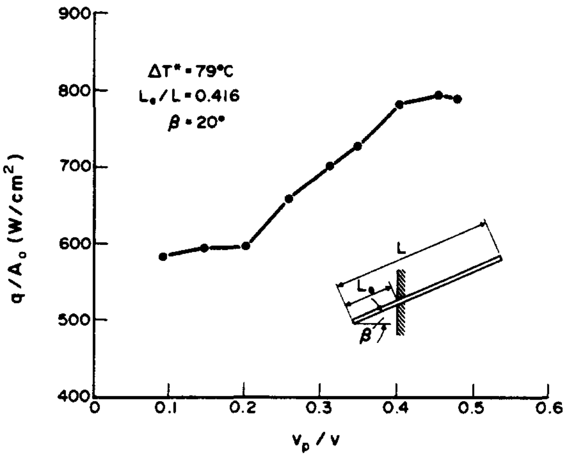

Fig. 6.

beads touch the wall. The thermocouple wells in turn are kept in their position by 3/8-inch fittings screwed on the outer shell. Temperature measurements are made using 30 Ga copper–constantan thermocouple wire. All wire is taken from a single spool to insure uniformity of the thermocouples.

An optical level with $\pm 1°$ resolution measures the inclination from horizontal position. Regulation of steam supply was accomplished by valve control to the steam chamber. Due to the clearance between the heat pipe and the shell inner surface, an annular flow develops in the water region. To satisfy the constant wall temperature requirement in this region, Reynolds numbers based on the hydraulic diameter of the annulus were in the range of 20,000. In experiments a maximum of ± 1°C temperature deviation was recorded along the pipe surface on the water side.

Experimental procedure

In the present study, the important parameters affecting the steam condensation rate are the inclination angle, β, the evaporator length, L_e, and the temperature difference between warm and cold regions, ΔT^*. The ranges over which these parameters were varied are as follows:

$$\beta \quad 10\text{–}20°$$
$$L_e \quad 30\text{–}90\ \text{cm}$$
$$\Delta T^* \quad 36\text{–}79°\text{C}.$$

In addition to the above related parameters, the optimum amount of working fluid for which the steam condensation rate assumes a maximum has to be determined.

The volume available for pentane to establish a circulatory flow and to transfer the heat is lessened by the presence of air. The heat pipe was evacuated to a pressure of 10 mm-Hg absolute. Then the desired amount of pentane was charged to the system through a pipette. In a first series of tests, the water inlet temperature was at 20°C. Subsequently it was raised to 63°C. This served the purpose of determining, at least qualitatively, the effect of temperature difference between warm and cold regions on heat pipe performance.

The experimental data comprised the orientation and the inclination angle of the pipe, the four wall temperatures, T_w, the inlet temperature of water, $T_{wa,i}$, the outlet temperature of water, $T_{wa,o}$, the inlet temperature of steam, $T_{st,i}$, the mean pressure inside the pipe, p_m, and the amount of steam condensate.

RESULTS

A preliminary round of heat transfer measurements was undertaken to estimate the optimum quantity of pentane to be charged. Thus, at specified geometrical conditions, a well established circulatory flow of the working fluid would be possible to exist in the pipe. The configuration chosen for these runs was with the pipe inclined 20° from horizontal position with 42% of the total length on the steam side. The charged amount was represented as a fraction of the total tube

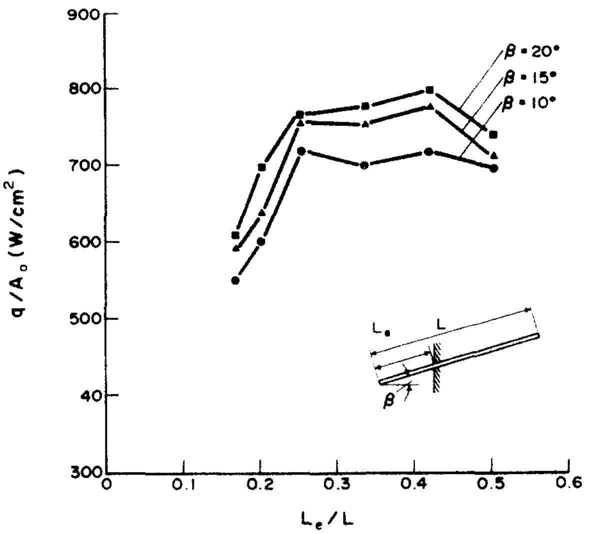

Fig. 7.

volume. In experiments the volumetric ratio of pentane, v_p/v, was varied from 0.09 to 0.45 in increments of 0.055. For each run the flow rate of water through the shell was constant, and the equilibrium conditions were assumed to exist when the energy released by steam condensation was equal to the energy absorbed by water. By measuring the steam condensation rate, the average heat flux through the pipe was determined. For each run the experiment was repeated seven-times and the results were averaged to a single point. The data so obtained are shown in Fig. 6. The addition of more pentane to the pipe resulted in an increase in the pipe pressure, p_m, and the axial heat flux. One explanation to this result is that a certain amount of pentane is always collected at the bottom end of the pipe, condensation and boiling processes occur through this source. The existence of this source increases the pressure in the pipe for better conditions. The increase in axial heat flux however is not indefinite. As shown in Fig. 6 for volumetric ratios in the vicinity of 0.4, the heat flux approaches a limiting value of 800 W cm^{-2}.

Pattern of heat flux distributions

A distinguishing feature of the present investigation is the experimental determination of the effect of inclination angle on axial heat flux distribution. Following the filling process of the pipe, at each length ratio the pipe inclination angle was varied from 10 to 20° with 5° increments. Throughout the experiments the temperature difference between warm and cold regions was constant and equal to 79°C. The results are exhibited in Fig. 7 in which at each inclination angle a monotonic increase of the heat flux over the low values range of length ratios can be recognized. For all inclination angles however the peak heat flux occurs at a length ratio equal to 0.416. This result is consistent with the theoretical findings presented in Fig. 4. A comparison of experimental results with the analysis shows that at low length ratios, i.e. $L_e/L < 0.3$, the heat flux variation in Fig. 4 is much steeper and a large deviation exists in peak values of the heat flux. These discrepancies mainly arise from assuming the pipe to be wetted but non-flooded in the analysis. In other words, the entire pentane content is assumed to circulate between the evaporator and the condenser section and the accumulation of a certain amount at the bottom is disregarded. Consequently in the analysis higher values of heat fluxes are obtained.

Figure 8 presents the effect of temperature difference between warm and cold region on the axial heat flux distribution. In these experiments the pipe material was aluminum with the same dimensions as copper tubing. Essentially the steam reacts with the aluminum resulting in the formation of a definite protective film on the surface [7]. Although the thermal conductivity of aluminum is less than of copper, the experimental results indicate that this does not appreciably effect the steam condensation rate. In these runs, the pipe inclination angle and the water flow rate through the shell being kept constant, only the temperature difference between the two regions was reduced to 36°C by increasing the water inlet temperature to 63°C. Then the steam condensation

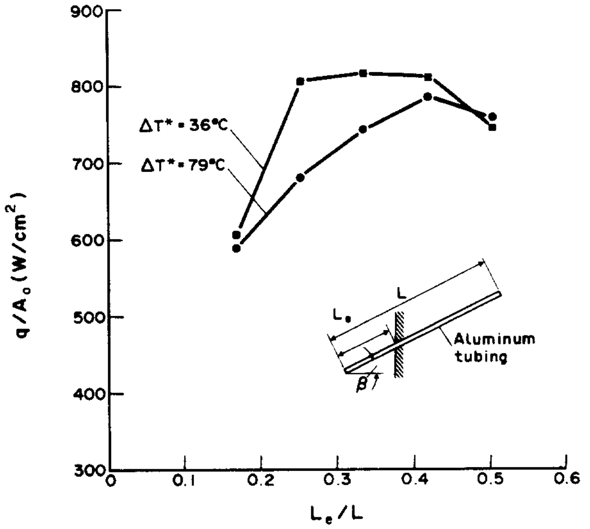

Fig. 8.

rate was measured. The purpose is, of course, to provide better operating conditions by increasing the pressure inside the pipe. As shown in Fig. 8, the heat flow rates at $\Delta T^* = 36°C$ are generally higher than the ones obtained by the temperature difference $\Delta T^* = 79°C$. In addition, at low temperature differences and at length ratios in the range of $0.25 < L_e/L < 0.42$, the axial heat flux is almost constant and independent from length ratio variations.

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