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Collapse of vapor locks by condensation over moving subcooled liquid

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Abstract

Experiments were conducted for condensation taking place over thick layer of moving subcooled liquid. Measurements were done over a wide range of liquid subcool temperatures and Reynolds numbers. The measured values of condensation heat transfer coefficients were correlated with Reynolds number, Prandtl number and the degree of subcooling. The role of non-condensable gases in the liquid and condensation over surfaces other than the liquid surface was also determined. The condensation heat transfer coefficients obtained for the thick layers of moving liquid varied more strongly with variations of liquid temperature than those reported for film condensation. The effect of changes of flow velocity significantly influenced condensation at low values of subcool temperatures. Changes of subcool temperature were found to exert a greater influence on condensation than the changes in flow velocities. The reasons for the above influences are discussed.

Keywords: Vapor bubble; Condensation; Collapse time; Subcool temperature

1. Introduction

The condensation of vapor over a fairly thick layer of moving subcooled liquid has not been investigated unlike the condensation over walls and thin film of liquid. The problem of condensation over a thick layer of moving liquid is of relevance in different applications such as chilling and filling of complicated geometries involving feedlines and pumps. The persistence of vapor bubble in regions of adverse pressure gradients [1,2] could lead to two phase flow instabilities in the system. The collapse of vapor locks is also of relevance in nuclear reactor cooling where steam comes in contact with flowing subcooled water. The major parameters of interest that are likely to influence the condensation process leading to the collapse of vapor locks are liquid subcool temperature, flow velocity and stratification of temperature in the liquid. These aspects are investigated in the present study.

2. Previous work

There have been extensive studies dealing with heat and mass transfer in stratified two-phase flows. Most of these studies involved either gas absorption at the gas–liquid interface or condensation over a thin film, which is only slightly subcooled. A good review of the theoretical models proposed to account for these transfer rates is given by Herman [3]. Linechan et al. [4] made measurements of condensation over a subcooled thin film of liquid. Measurements of condensation on a turbulent liquid by Thomas [5] gave heat transfer coefficients of about 10 kW/m^2 K whereas the heat transfer coefficient of condensation of high speed steam jet reported by Young et al. [6] was of the order of 5×10^3 kW/m^2 K. The differences in heat transfer coefficients were attributed to the changes in turbulent intensity and the entrainment.

Dobson and Chato [7] conducted experiments in the different flow regimes corresponding to slug, plug and wavy flows for condensation of refrigerants in horizontal tubes. The focus was on variation of condensation heat transfer coefficients in the different flow regimes.

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Nomenclature

A	area (m^2)
C_p	specific heat capacity (J/kg K)
D	diameter of the test section (m)
d_i	diameter of the bubble (m)
h	convective heat transfer coefficient (W/m^2 K)
j	mass rate of condensation per unit area ($kg/m^2/s$)
Ja_l	Jakob's number
k	thermal conductivity of water (W/m K)
K	degree of subcooling
L	length of the test section (m)
$\dot{m}$	mass rate of condensate (kg/s)
M	molecular weight (kg/kg mol)
p	pressure (N/m^2)
Pr	Prandtl number
R	universal gas constant (8314 N m/kg mol K)
Re	Reynolds number
t	time for collapse of the bubble (s)
T	temperature (K)
u	average velocity of liquid in the test section (m/s)
v	vapor volume (m^3)
x	length parameter (m)

Greek symbols

α	thermal diffusivity of perspex wall (m^2/s)
Δ	thickness of perspex walls (m)
δ_T	thermal boundary layer thickness/thermal layer thickness (m)
ρ	density of vapor (kg/m^3)
λ	latent heat of vaporisation (J/kg)
ν	kinematic viscosity (m^2/s)

Subscripts

c	collapse
i	interface
l	liquid
s	surface of the test section
sat	saturated
v	vapor
w	water/wall surface

The heat transfer coefficient was seen to be nearly independent of mass flux.

Rosson and Myers [8] reported measurements of condensation for methanol and acetone in horizontal tubes. The condensation heat transfer coefficients were found to vary significantly over the circumference of the tube. The coefficient was independent of the amount of liquid in the tube in the upper circumference and correlated well when the effect of vapor shear was incorporated in the model.

Condensation measurements of Lim et al. [9] in horizontal co-current steam/water flows showed the heat transfer coefficients to vary from 1.3 to 20 kW/m^2 K depending on whether the liquid interface was smooth or wavy. Condensation heat transfer coefficients were observed to increase with increasing steam flow rates and water flow rates. The heat transfer coefficient was also observed to double when the vapor–liquid interface became turbulent.

Ichiro Tanasawa [10] reviewed the advances in condensation heat transfer. The importance of condensation over liquid surface and dropwise condensation were identified as the emerging new areas of research requiring further investigations. Chen et al. [11] conducted experiments to analyse vapor locking and heat transfer under transient and steady-state conditions in the cooling channels of a cryo-stable super-conducting magnet. No explicit measurements of the heat transfer coefficients were, however, made in his experiments.

Theoretical models have been proposed to determine the condensation rates and are reviewed by Collier [12]. Lin [13] and Sideman and Moalem-Maron [14] have brought out the complicated nature of the interface condensation for wavy and turbulent interfaces and for the drag induced convective heat transfer. Their studies point to significant increase in the condensation rates with increased convection in the liquid in the presence of higher shear stress. Gerner and Tien [15] modeled the effects of interfacial forces, subcooling of liquid, superheating of vapor, and the presence of non-condensable gas in the vapor. While subcooling of liquid was seen as a dominant mechanism enhancing condensation, the non-condensable gases in the vapor were shown to significantly reduce condensation.

A detailed analysis of condensation was provided by Lock [16]. Average heat transfer coefficients were presented for direct condensation on drops and jets. Empirical relations for predicting Nusselt number based on Prandtl number, Reynolds number and Jacob's number were obtained for forced laminar condensation in horizontal tubes. The analysis included the collapse of a spherical bubble surrounded by its own liquid. Analytical relationships were developed to predict the collapse time for bubbles with different levels of subcooling.

It is not possible to apply the above results for quantitative prediction of condensation of vapor over a thick layer of moving liquid. A theoretical model for predicting the collapsing time of a vapor lock in cryogenic system was formulated by Nandi and Ramamurthi [1]. In the absence of experimental data on condensation over a moving layer of subcooled liquid, it is difficult to arrive at meaningful predictions for the time of collapse of a vapor column. Experiments are therefore conducted to generate data on condensation of vapor over subcooled moving liquid layer in a simple well-defined system. The system chosen is a vertically stratified pipe flow with water as the liquid.

3. Experimental set-up and instrumentation

A test rig was designed and constructed in which the time of collapse of a column of water vapor over a moving liquid column by condensation was measured. The set-up simulates condensing vapor bubble adhering to walls during flow through feedlines. A schematic diagram of the set-up is shown in Fig. 1. Water vapor of volumes varying between 4.5×10^{-6} and 12×10^{-6} m^3 is contained in a cylindrical test section. The walls of the test section are made of 5×10^{-3} m thick Perspex. The lower end of the test section is connected to a stainless steel shell adapter as shown in Fig. 1. It is provided with an inlet pipe of 8×10^{-3} m diameter through which warm water enters the Perspex test section. The water leaves the chamber, as shown through a pipe of the same diameter. Warm water is admitted at a steady rate from a tank kept elevated relative to the test section.

The tank is provided with a heater and stirrer, so that water at the desired temperature could be supplied to the test section. The tank is connected to the inlet of the test section by a flexible hose made of Teflon. The length of the hose is kept small in order to reduce the heat loss from the liquid. A flow control valve is provided in the line to vary the flow rate. The test section is also provided with a heater as shown in Fig. 1. The heater is connected to a regulated DC power supply. The positioning of this heater restricts the maximum volume of the vapor that could be generated in the apparatus.

The test section has a vent line fitted with a valve as shown. Chromel–alumel thermocouples are used for measuring the temperature of the warm water and vapor at different locations. The thermocouple leads are connected to a multi-channel, multi-purpose digital multimeter that has an accuracy of $\pm 0.2°C$. A thermocouple scanner card is used for multi-channel thermocouple measurements. An Infrared thermal-imaging camera is also used to map the temperature distribution on the surface of the test section. The IR radiation emitted from the surface is focussed on to a photovoltacic PbSn detector array and the resulting electrical signals are expressed as temperatures. The measurement accuracy of the system is $\pm 0.1°C$.

The experimental procedure comprises of initially opening the inlet valve and admitting water to fill the lines and the test section. The vent valve and the outlet valve are adjusted so that the entire test section is filled with water. The valves are thereafter closed. The vapor is generated by rapidly heating the water at the surface. The DC heater provided in the water column near the surface is switched on to generate steam of the required quantity and then switched off to avoid burn off of the heater coil. Simultaneously the inlet valve is opened to admit the warm water at a preset value of temperature and the time for the condensation of the vapor over the liquid surface is estimated using a stopwatch. The mass rate of condensation is estimated from the volume change of vapor per unit time. During the condensation, the pressure in the vapor column is taken to be constant since the liquid level in the tank is reasonably stationary. The walls of the test section are assumed to be adiabatic as Perspex has a very low thermal conductivity of the order of 0.5 W/m K.

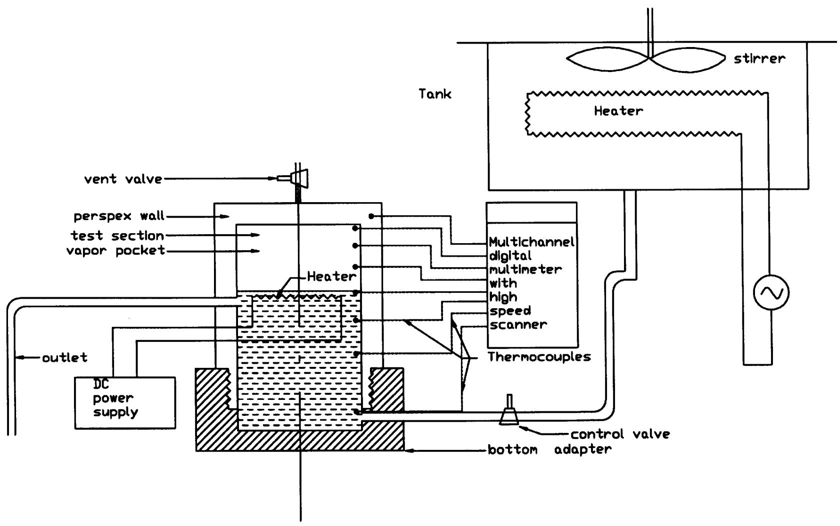

Fig. 1. Schematic of the experimental set-up.

A total of 59 experiments were performed, by varying the following parameters: (i) water flow rate (ii) vapor volume and (iii) degree of subcooling of the water. The water flow rate was varied over about an order of magnitude between 1.56×10^{-6} and 13.3×10^{-6} m^3/s. Initial vapor volumes of 4.91×10^{-6}, 9.81×10^{-6} and 12.27×10^{-6} m^3 were used. The effect of subcooling the water was determined by using water at temperatures between 30°C and 68°C in steps of 2°C approximately. All experiments were performed with the exit of the test channel open to atmosphere. The effect of superheat was very small because of the large values of the latent heat of vaporization and the steam temperature being maintained close to the saturation temperature of the liquid. Sparrow and Eckert [17] have also shown the effect of superheat on condensation to be small.

3.1. Uncertainties in measurement

The experimental set-up was simple requiring only basic measurements of time, volume and temperature. The time required for the collapse of vapor bubble was measured by a digital stopwatch, which had a least count of 0.01 s. The typical times for condensation of vapor was two orders of magnitude greater between 2 and 5 s. However, the response to the observation would be much slower than 0.01 s and repeated measurements showed it to be between 0.2 and 0.4 s. This translates to an uncertainty of 10–20% for the smaller values of condensation times. For large times of condensation, the accuracy would be significantly better (4–8%). Most of the experiments were conducted in regions of larger condensation times. Each experiment was repeated a number of times to ascertain the possibility of repeatable observation.

The temperature measurements were carried out using chromel–alumel thermocouple junctions connected to digital multimeter with high speed scanner. The measurement accuracy of the system was within ±0.2°C. These measurements were supplemented by mapping of temperatures using an IR thermal imager of ±0.1°C accuracy.

The vapor volume measurements were carried out by measuring the height of vapor column by an in-built scale having 0.001 m accuracy. This would lead to an uncertainty of 0.13% in the estimated vapor volumes and hence in the condensation rates.

The pressure of the condensing vapor column was assumed to be constant in the experiments. This assumption was reasonable considering that the water level in the storage tank was not affected by the small flow rates used in the experiments. The maximum variation of pressure in the test section during the collapse of bubble is less than 1.96 mbar, which corresponds to a deviation of 0.17%.

4. Results and discussions

4.1. Effect of non-condensables in vapor

Before interpreting the data, a study was carried out to estimate the effect of non-condensable gases and entrapped air on the condensation process. A set of initial experiments, conducted with progressively increasing temperature of water, revealed that the time for collapse of vapor volume is more for the initial experiments and decreases for the subsequent experiments. This is illustrated in Fig. 2. Here, the temperature of water is kept at 28°C in the first experiment and is progressively increased to 46°C, 58°C and 80°C in the subsequent experiments. The time taken for the vapor to condense is seen to increase as the temperature of the water is decreased which is contrary to expectation. The phenomenon is caused by the presence of non-condensable gases in the water. When water is heated to higher temperatures, some amount of non-condensable gases along with entrapped air is removed.

The results of experiments carried out with water initially heated to high temperatures and cooled thereafter are illustrated in Fig. 3. It is observed from the figure that the time of condensation is not only drastically reduced but the time for condensation also increases for the higher values of water temperature. All experiments in this study, for which the condensation rates are determined and used to interpret the heat transfer coefficients, were carried out after removing the non-condensable gases and entrapped air by preheating the water. Repeatable results for the condensation times were obtained in the experiments.

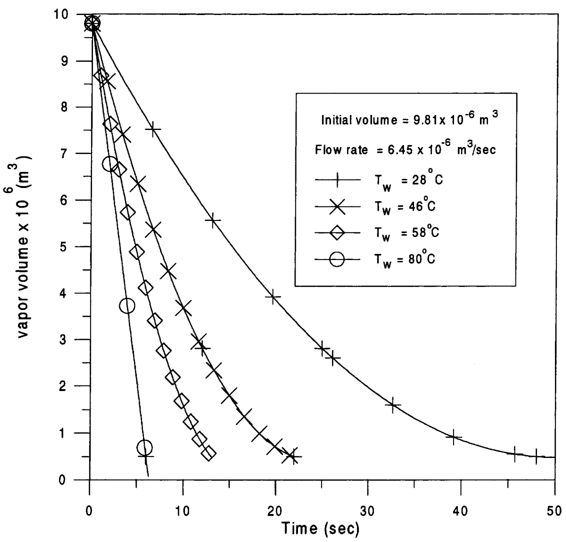

Fig. 2. Condensation in the presence of non-condensable gases in vapor.

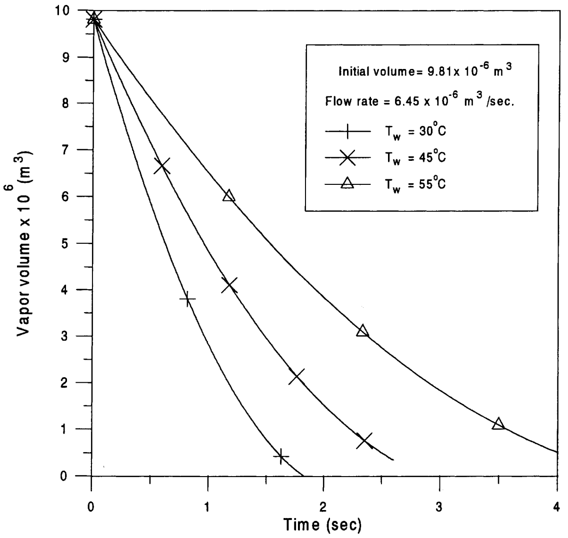

Fig. 3. Condensation in the absence of non-condensable gases in vapor.

4.2. Estimation of mass rate of condensation

The decrease in the volume of vapor is used to determine the mass of the condensate. The vapor density ρ is assumed to be constant during condensation. Since all the experiments are conducted at near-ambient pressures, the ideal gas equation $\rho = p/RT_{\text{sat}}$ is used to determine vapor density. The values of vapor density, estimated using steam table returns almost identical values. The mass rate of condensation $\dot{m}$ is calculated as:

$$\dot{m} = \rho v/t, \tag{1}$$

where t is the time for collapse of the vapor column of volume v.

4.3. Condensation over the walls of the test section

Experiments were carried out with varying proportions of wall surface area, A_s, for a constant area of water surface, A_w. Fig. 4 gives a plot of the mass rate of condensation, $\dot{m}$ as the wall surface area is increased. Different values of water temperature were used and the mean flow rate of water was kept at 6.45×10^{-6} m^3/s. It is seen that the mass condensed is remarkably constant over the entire range of wall surface area, A_s. The observation suggests that the condensation over the wall is small compared to the condensation over the surface of water. The major factor driving the condensation is the heat transfer from the interface to the liquid, which is about two orders of magnitude higher [9] than that from the vapor to the wall thereby leading to relatively insignificant wall condensation. The locally higher wall temperatures may also be responsible for the insignificant vapor condensation on the walls and this is discussed subsequently.

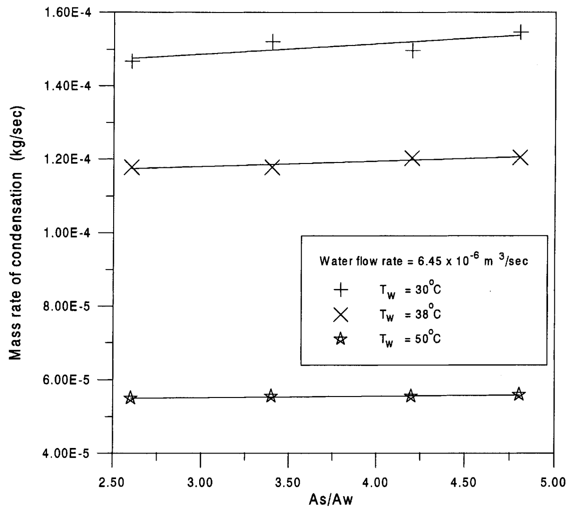

Fig. 4. Mass rate of condensation for varying wall surface areas.

In order to clarify the negligibly small condensation over the walls, experiments were done with detailed measurement of surface temperature over the walls of the test section. The thermal image of the surface of the perspex test section obtained before the commencement of vapor generation showed the entire surface to be at uniform temperature well below the operating temperature of water. The temperature distribution got modified after the vapor is formed in the test section by electrical heating of the water near the surface. This is illustrated in Fig. 5. Here the wall temperature is plotted as a function of the distance, (x) from the top of the test section, non-dimensionalized by the length (L) of the chamber. The wall temperatures of the upper region in contact with the vapor column increased to a value of about 63°C leaving the lower region containing the water column relatively uninfluenced at 44°C as shown in Fig. 5. Simple one-dimensional heat conduction analysis predict the inner wall temperature to be about 85°C which is very much higher than the liquid temperature thereby resulting in relatively insignificant wall condensation. Visual examination of the inner walls of the test section does not give any evidence of condensation taking place over it. It was also noted from the succeeding IR images that the outer wall temperature of the test section does not change during the course of condensation. This is due to the low thermal diffusivity of Perspex. The typical condensation times are in the range of 2–5 s whereas the characteristic wall heating times are $\sim \Delta^2/\alpha$ which is about a minute. Here Δ refers to wall thickness and α is the thermal diffusivity.

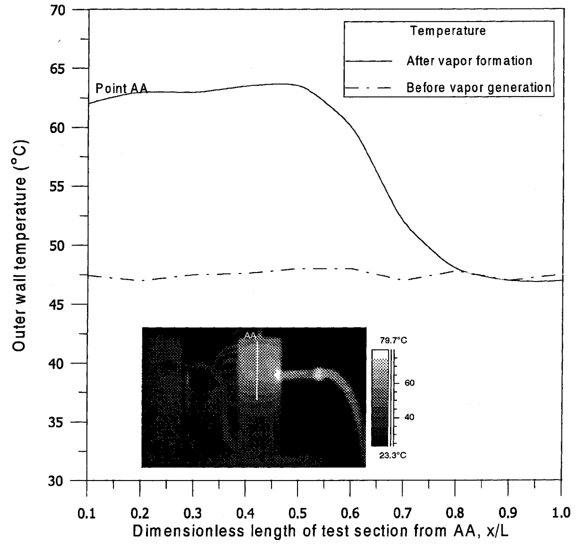

Fig. 5. Typical wall surface temperature distribution obtained by thermal mapping.

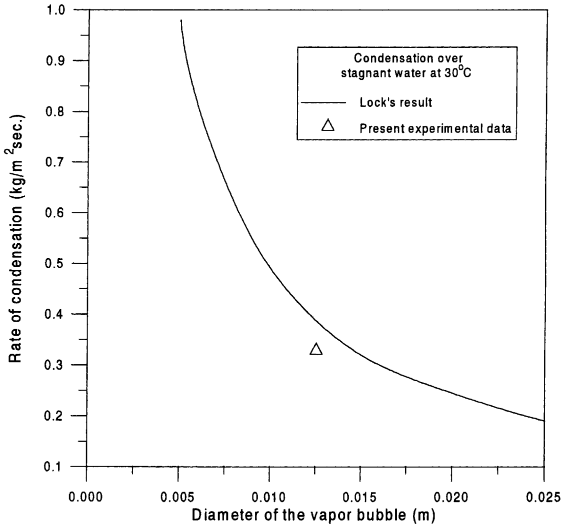

Fig. 6. Condensation rate for a stagnant water column using Lock's results and comparison.

With condensation predominantly taking place over the liquid surface in the present set of experiments, the mass flux of the condensate over the water surface during the collapse of the vapor column (hereafter termed as the rate of condensation) is written as:

$$j = \frac{\rho v}{A_w t}. \tag{2}$$

4.4. Comparison of the results for a stagnant water column

Lock [16] has modeled the collapse of a stagnant vapor bubble in a subcooled liquid by considering a symmetric thermal boundary layer to surround the collapsing bubble. He solved the energy and interface equations simultaneously resulting in the following:

$$\delta_T = \frac{d_i}{2}\left[\frac{\rho_v}{\rho_l Ja_l}\right] \tag{3}$$

and

$$t_c = \frac{d_i^2}{4k_l}\left[\frac{\rho_v}{\rho_l Ja_l}\right]^2. \tag{4}$$

Fig. 6 illustrates the condensation rate based on the above equations for bubbles of various diameters. It is noted that the condensation rate decreases exponentially with increase of bubble diameter. The diameter, d of the cylindrical liquid column in the present experiments is 0.025 m. Since condensation takes place predominantly over the surface of the liquid, the equivalent diameter of the spherical bubble (d_{eq}) corresponding to the surface area of the liquid is given by:

$$\pi d_{eq}^2 = \pi d^2/4. \tag{5}$$

This gives an equivalent bubble diameter of 0.0125 m. The measured rate of condensation for the stationary column of water at a temperature of 30°C in the present experiments is 0.345 kg/m^2/s. This is shown in Fig. 6. It is observed that the measured value of condensation is lower by about 25% compared to the prediction of Lock. Lock assumes an isothermal vapor volume in its own isothermal liquid which is not the case in the present experiments. The surface of water, being rapidly heated by the electrical heater, causes stratification in liquid temperature below the water surface.

4.5. Effect of flow on condensation

A typical temperature profile, obtained along the depth of the liquid for the stagnant column of liquid and after the initiation of a flow at a rate of 6.45×10^{-6} m^3/s, is shown in Fig. 7. It is seen that the initial subcool water temperature of 30°C does not prevail in the near surface region of the liquid for the stagnant column of water. It is also seen that the temperature of the water layers reach values nearer to the initial water temperature immediately after the flow is commenced. It was not possible to measure the liquid temperature profile with time in the experiments. However, the time taken for the liquid temperature to reach values near to the initial subcool temperature was less than the collapse times of the vapor in the experiments that varied typically between one to five seconds. The stratification in the temperature of water, shown in Fig. 7, reduces the condensation rates.

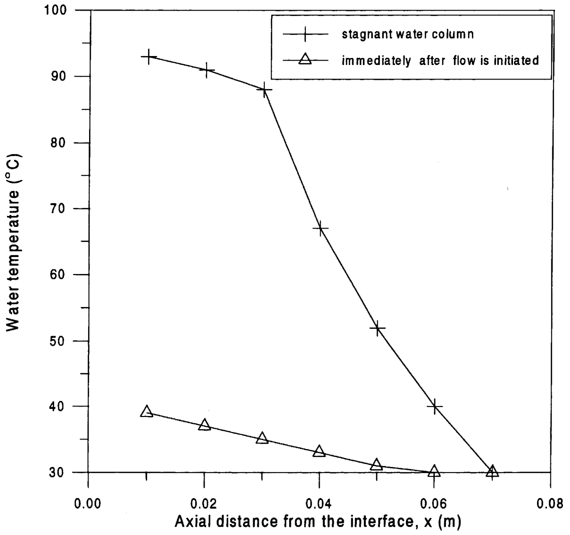

Fig. 7. Typical temperature distribution along stratified liquid layers.

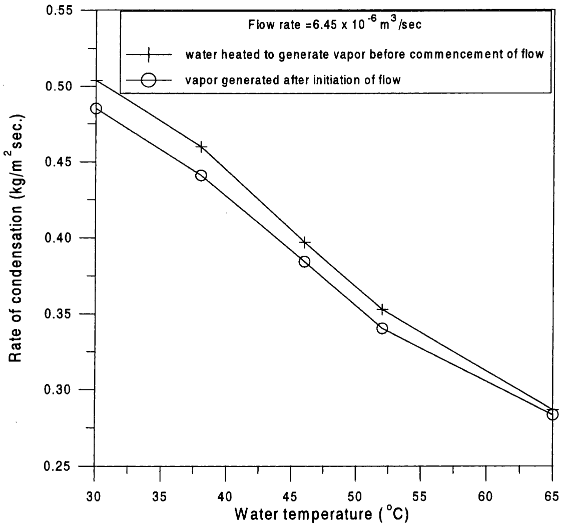

Fig. 9. Comparison of condensation rates for the two sets of experiments.

Experiments were done with two different levels of stratification obtained by heating the water near the surface through the electrical resistor either before or after the commencement of flow. Fig. 8 compares the temperature distribution in the water after initiation of flow for the two sets of experiments in which the vapor was generated before and after commencement of flow for a nominal flow rate of 6.45×10^{-6} m^3/s. It can be seen that the temperature profiles are slightly higher when water is heated before the commencement of flow.

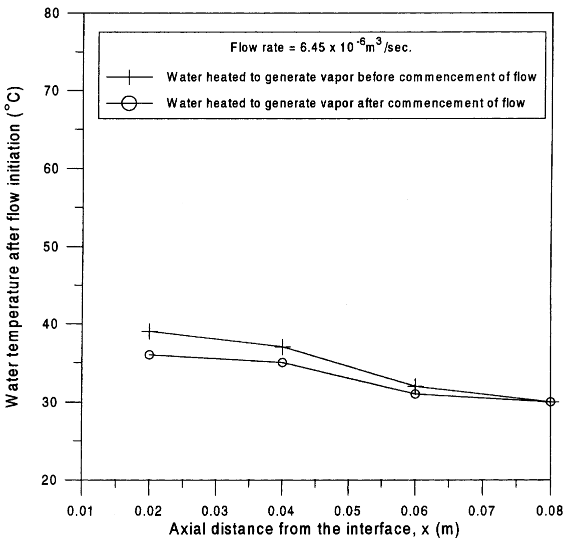

Fig. 8. Comparison of temperature profile of stratified layers for two sets of experiments.

Mixing within the liquid layers reduces the temperature when flow is initiated prior to the generation of vapor. The variation in condensation rates caused by the above difference in temperature distribution is shown in Fig. 9. The difference in the condensation rates in the two sets of experiments is seen to decrease as the water temperature increases, i.e., the time for condensation increases. The maximum difference is within 8%. The results suggest that though the present experiments may not be ideal to study condensation of vapor bubbles in stagnant liquids, good estimates of condensation over a thick layer of moving liquid are possible.

The effect of variation of liquid flow rate on condensation for the two stratification levels obtained with a water temperature of 30°C is illustrated in Fig. 10. A rapid increase in the rate of condensation is initially obtained with increasing liquid flow. The condensation rate depends on the ability of the liquid to transport thermal energy away from the interface into the liquid main stream, and this gets augmented as the flow rate of water increases. The deviations in rate of condensation between the two sets of experiments are negligible owing to reasons already discussed. In order to reduce effects of stratification, the vapor generated by resistance heating of the water at the surface was carried out after the initiation of flow.

4.6. *Influence of flow velocity and water temperature on condensation*

The dependence of water temperature on the rate of condensation is shown in Fig. 11 for flow rates of 6.45×10^{-6} and 10×10^{-6} m^3/s. It is observed that rate

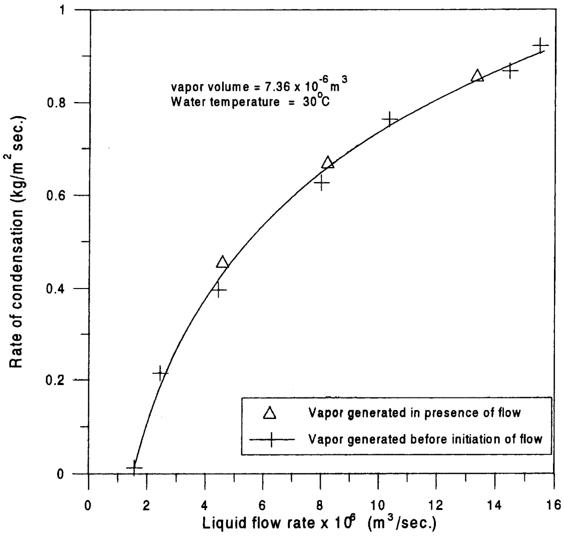

Fig. 10. Condensation rates as a function of liquid flow rate.

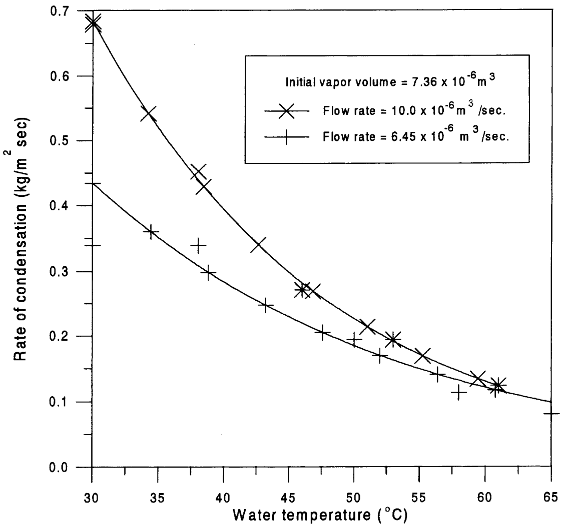

Fig. 11. Variation of rate of condensation at different liquid temperatures.

of condensation decreases with increase of the water temperature. The influence of velocity is more significant at lower temperature of water. For water temperature exceeding about 50°C, the two different flow velocities gave about the same rate of condensation. The effectiveness of flow velocity in increasing the condensation at the lower temperatures is to be anticipated since mixing with water at lower temperature leads to enhanced heat transfer from the gas–liquid interface to the liquid. The condensation heat transfer coefficients are obtained at different flow velocities and subcool temperatures following the earlier work of Collier [12] by defining the coefficient as:

$$h = j\frac{\lambda}{T_{\text{sat}} - T_l}. \tag{6}$$

The above equation is valid as the sensible heat carried away by the liquid surface is very small compared to the latent heat [12] and can be neglected. In the present set of experiments, vapor is not superheated and the vapor temperature corresponds to the saturation temperature. The condensation heat transfer coefficient so determined is plotted in Fig. 12. It is observed that the coefficient increases as water temperature increases. The observed increase with water temperature is in agreement with the findings of Collier [12] wherein average condensation coefficient over solid surface was also observed to increase as the temperature of the wall increases. Collier observed that the condensation heat transfer coefficient varies as $-1/4$th power of $T_{\text{sat}} - T_{\text{w}}$. The effect of flow velocity on the condensation heat transfer coefficient, shown in Fig. 12 is seen to be more significant at lower values of water temperature. The increased availability of subcool liquid near the interface which increases the ability of the liquid to transport heat energy from the surface is responsible for this increase. At elevated temperatures, the increased values of heat transfer coefficients are brought about by the enhanced water temperature rather than the increase in velocity.

Condensation of water vapor over a flat plate at 50°C at ambient pressure [12] yields a condensation coefficient of 3.7 kW/m^2 K. The present experiments on condensation of water vapor over a stagnant thick liquid layer at 50°C and ambient conditions resulted in a higher

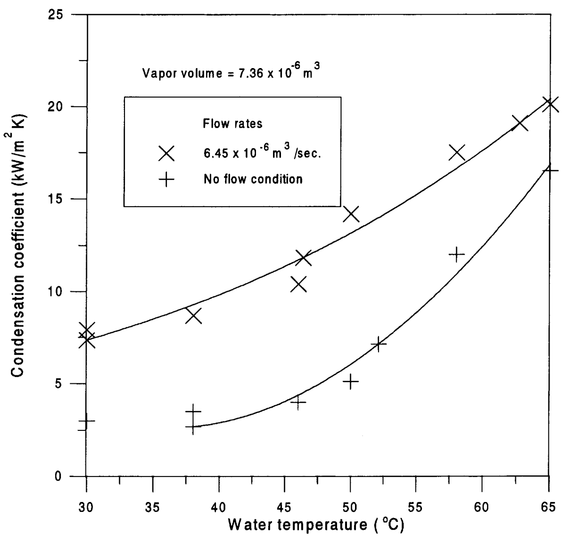

Fig. 12. Condensation coefficients as a function of water temperature for flow and no flow conditions.

value (5.1 kW/m^2 K) of condensation heat transfer coefficient as can be seen from Fig. 12. This is in agreement with the earlier assessment that condensation over the liquid surface is much higher than condensation over walls in view of the heat transfer from the interface to the liquid.

The influence of the temperature difference $T_{sat} - T$ (where T is either T_l or T_w as the case may be) on the condensation heat transfer coefficient is further illustrated in the log–log plot of the condensation heat transfer coefficient and $T_{sat} - T$ in Fig. 13. Here, the condensation heat transfer coefficient over a solid surface is compared with the results obtained from the present experiments for two liquid flow Reynolds numbers. The slope of the former is $-1/4$. However, the slope of the other two curves is higher and works out to be -0.35 and -0.37, respectively for flow Reynolds number values of 1328 and 970. A higher slope is indicative of the higher sensitivity of the liquid temperatures when condensation takes place over a moving liquid layer.

An effective interface temperature, T_i can also be calculated in terms of the interfacial mass flux j using the following expression derived by Collier [12] as:

$$j = \left(\frac{M}{2\pi R T_{sat}}\right)^{1/2} p \frac{(T_{sat} - T_i)}{2T_{sat}}. \tag{7}$$

Variations of interface temperature for different values of water temperature are shown in Fig. 14 for a flow rate of 6.45×10^{-6} m^3/s. The effective interface temperature increases with increase in water temperature and is to be anticipated. However, large variations in the water temperature result in only marginal changes in interface temperature.

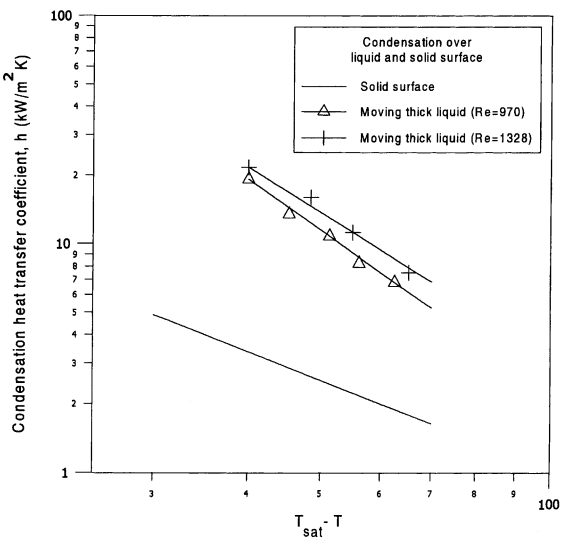

Fig. 13. Variation of condensation coefficients.

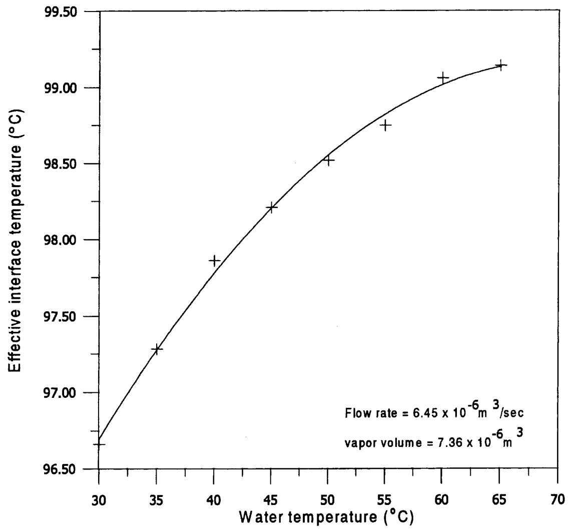

Fig. 14. Estimated effective interface temperature.

The energy balance at the interface may be written as:

$$\lambda \dot{m} = h(T_v - T_l) \approx k \frac{T_v - T_l}{\delta_T}, \tag{8}$$

where k is the thermal conductivity of water and δ_T is the equivalent thermal layer thickness.

Under steady-state conditions, the condensation is balanced by liquid being carried away in the film. Variation of condensation heat transfer with changes in water temperature is shown in Fig. 15. As reported by Lim et al. [9], the magnitude of the heat transfer is seen to decrease with increasing water temperature. An effective thermal layer thickness determined based on the above expression is plotted in Fig. 16 for different values of subcool liquid temperatures. It is noted that higher liquid temperatures invariably result in relatively larger thermal thickness and lower heat transfer. The availability of a potential sink (water at low temperature) in the immediate vicinity enhances the heat transfer.

The influence of liquid flow rate on the condensation coefficient is shown in Fig. 17 for two different values of water temperature. An increase in h is observed in both the cases as the flow rate of liquid is increased. The influence of velocity is seen to be larger at lower water temperatures. The gain achieved in condensation rates would therefore not be significant when higher flow velocities are employed with liquid temperatures nearer to the saturation temperatures.

The above results are plotted in terms of non-dimensional Nusselt numbers in Fig. 18 for different water temperatures at three different Reynolds numbers. The Reynolds number is defined with the diameter of the test

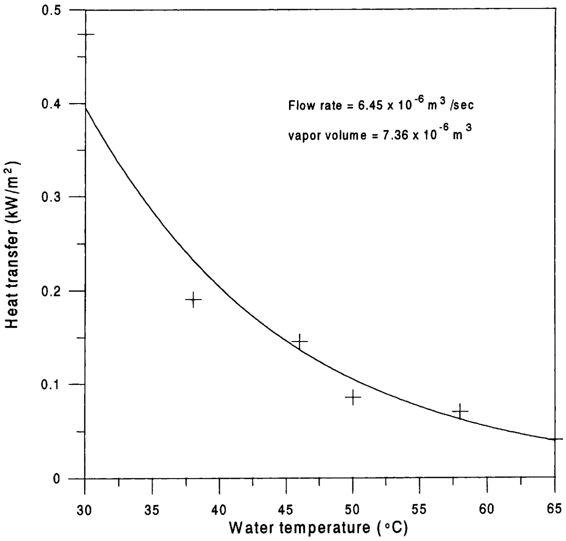

Fig. 15. A typical heat transfer profile with varying liquid temperatures.

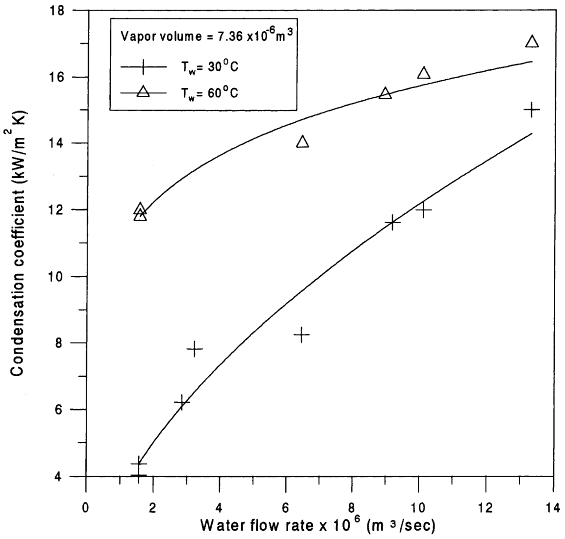

Fig. 17. Variation of condensation coefficients with changes in liquid flow rate.

section as the characteristic dimension ie., $Re = uD/\nu$. The choice of these dimensionless variables using constant coefficients and flow variables is justifiable as the experiments are conducted after establishing steady flow velocities. The vapor temperature, which equals the saturation temperature, would also not change during condensation. An average heat transfer coefficient is representative of the phenomenon during the collapse of the vapor bubble for the particular flow velocity and water temperature at which the experiment is done. Fig. 18 show the Nusselt number to be less sensitive to the changes of Reynolds number compared to the changes in water temperature.

A curve fit of the complete set of data obtained from the experiments by varying water flow rate, vapor volume and inlet water temperature in terms of Nusselt numbers, Reynolds numbers, Prandtl numbers and the degree of subcooling can be expressed in the form,

$$Nu = C Re^p Pr^q K^s, \tag{9}$$

where C, p, q, s are empirical constants. The degree of subcooling, K in the above expression is taken as:

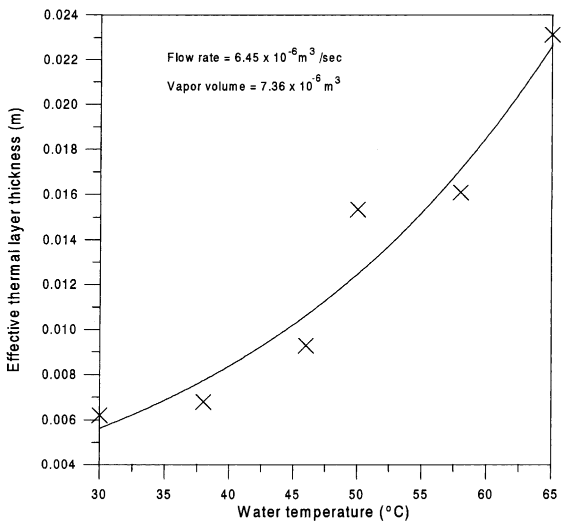

Fig. 16. Thermal layer thickness variation with different water temperatures.

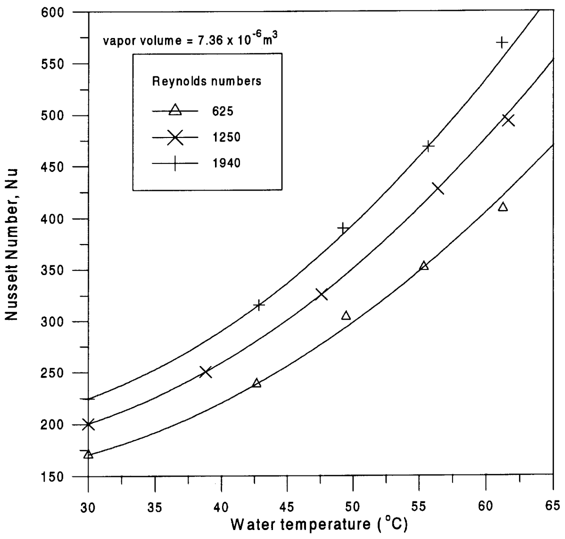

Fig. 18. Nusselt number variation with liquid subcool temperature.

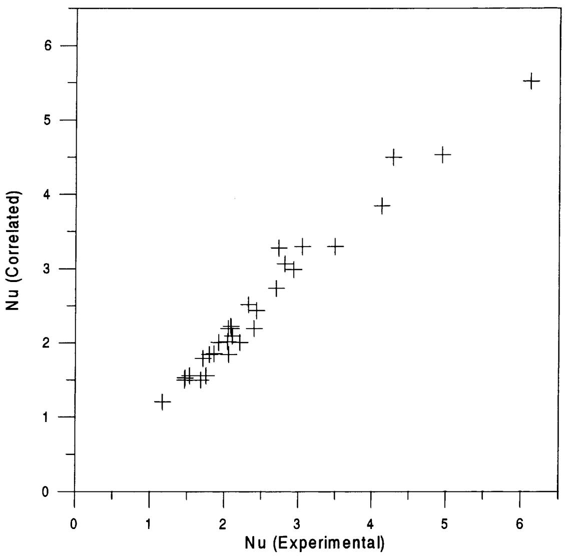

Fig. 19. Correlation of Nusselt number with Reynolds number, Prandtl number and degree of subcooling.

$$K = \frac{C_p(T_{sat} - T_l)}{\lambda}. \tag{10}$$

A least square scheme to locate the optimum values for C, p, q, and s, gives the following correlation:

$$Nu = 0.39Re^{0.641}Pr^{1.27}K^{-0.36}. \tag{11}$$

The correlation would be valid in the range:

$170 \leqslant Nu \leqslant 550$

$97 \leqslant Re \leqslant 2000$

and

$0.05 \leqslant K \leqslant 0.13.$

From the correlation it is seen that the heat transfer coefficient for condensation taking place over a moving liquid varies with the liquid temperature in proportion $h \sim (T_{sat} - T_l)^{-1/2.78}$. The dependence in the case of thin liquid films over solid surfaces [12] is much lower $(h \sim (T_{sat} - T_w)^{-1/4})$ as discussed earlier. Fig. 19 is a comparison of the above Nusselt number correlation with the experimental data. The least square fit has a standard error of 3.34% in the estimation of Nusselt number.

5. Conclusion

Condensation of water vapor over a moving thick layer of subcooled water in a stratified flow is measured and the heat transfer coefficients deduced. The effects of flow rate, temperature and presence of non-condensable gases on condensation rates are examined. Consistent results are shown to be obtained by removing the trapped gases in the liquid by preheating the water. The rate of condensation is shown to increase with decreasing temperatures of water.

For a stagnant column of water, the stratification in temperatures in the liquid has a significant influence on condensation rates. The stratification effects are less pronounced for moving column of liquid. The influence of changes in flow velocity of the liquid on condensation is not as strong as due to the changes in the temperature of the liquid. The increased availability of subcooled water in the vicinity of the interface promotes condensation. At elevated water temperatures, the heat transfer coefficient is less influenced by the flow velocity.

The values of condensation heat transfer coefficients obtained in the experiments are expressed in terms of Nusselt number and correlated with Reynolds number and degree of subcooling. The Nusselt number is shown to be more strongly dependent on the liquid temperatures for the case of thick layer of moving liquid than for film condensation. The work brings to focus the importance of subcool temperatures in promoting condensation of vapor locks in a flowing medium.

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