PERGAMON

International Journal of Heat and Mass Transfer 42 (1999) 2943–2953

International Journal of HEAT and MASS TRANSFER

# Influence of sweeping on dropwise condensation with varying body force and surface subcooling

C. Yamali[a,*], H. Merte Jr.[b]

[a] *Middle East Technical University, Ankara, 06531, Turkey*
[b] *The University of Michigan, Ann Arbor, MI 48109, U.S.A.*

Received 20 March 1997; in final form 27 September 1998

**Abstract**

An experimental work is conducted to study the effect of body forces on dropwise condensation heat flux under controlled subcooling. For this purpose a centrifuge is designed. A gold plated condenser surface, which is placed horizontally on the centrifuge so that horizontal centrifugal force becomes parallel to the condenser surface, is used to obtain dropwise condensation. Experimental data is obtained at various acceleration levels while the coolant temperature is kept constant. To determine the noncondensable effect, data is collected with and without surface suction. Results of the experiments showed an increase in dropwise condensation heat transfer coefficient with an increase in body force. Some noncondensable effect is observed at low acceleration levels (1 and $5 \times g$). No definite effect of subcooling on heat transfer coefficient is observed. Variations in heat transfer coefficient with different subcoolings are believed to be due to the variation in nucleation site density. © 1999 Elsevier Science Ltd. All rights reserved.

**Nomenclature**

$a$ acceleration [m s$^{-2}$]
$g$ gravitational acceleration [m s$^{-2}$]
$q$ heat transfer rate [kW]
$r$ drop radius [m]
$B$ constant [m]
$K_1$ constant [W m$^{-2}$]
$K_2$ constant [m K]
$T$ temperature [K]
$\Delta T$ temperature difference [K].

*Greek symbols*
$\theta$ contact angle
$\rho$ density of the condensate [kg m$^{-3}$].

*Subscripts*
b base
d drop
sat saturation
t total

*Superscript*
$''$ flux.

## 1. Introduction

In this study, the effect of body forces on dropwise condensation heat flux under controlled subcooling is studied experimentally. Previously, some work was done to investigate the effect of varying body forces [1, 2], but no control on subcooling and noncondensables is introduced in these studies.

It is observed by Schmidt et al. [3] that heat transfer coefficient for the dropwise condensation of steam was an order of magnitude larger than the heat transfer coefficient obtained with filmwise condensation. A successful application of dropwise condensation requires a clear understanding of the physical events taking place during this phenomenon. To understand the dropwise condensation phenomenon, researchers followed two approaches; while some of them tried to explore the physical events making up dropwise condensation experimentally [4, 5], the others devised mathematical models [6, 7] to explain the phenomenon. It is one of the goals of this study, to put light on the effect of drop departure

* Corresponding author. Tel.: +90-312-210-2577; fax: +90-312-210-1266; e-mail: cemily@rorqual.cc.metu.edu.tr


PII: S0017–9310(98)00345–7

size on the dropwise condensation, heat transfer experimentally.

In dropwise condensation, many complicated events take place simultaneously. Drops, first, nucleate at locations called nucleation sites and grow by direct condensation and by coalescences taking place between them. When a drop grows to a size that can not be sustained by surface tension forces [8], it starts sliding on the surface leaving a bare surface behind it, on which new droplets form by nucleation.

Tanasawa et al. [9] used a wiper to remove all the drops on the condensing surface periodically. By using such a wiper, they could control the maximum drop size and also obtain a homogeneous state of condensation on the condensation surface. Measurements of their work showed that the heat transfer coefficient is proportional to $-0.22$ power of the maximum drop diameter.

Studies show that in dropwise condensation, condensation begins with the nucleation of the droplets at certain sites such as pits and grooves [10]. The majority of the heat transfer in dropwise condensation takes place through the droplets of very small sizes as these droplets grow as a result of condensation on their surface and the latent heat released is conducted through them, particularly through the regions close to the periphery of the droplets. Yamali and Merte [11] showed, by taking a droplet model with a uniform base temperature and uniform vapor temperature and circular heat flow lines in the vicinity of the periphery of the droplet and a variable interfacial heat transfer coefficient developed by Umur and Griffith [12], that it is possible to express the heat flux at the base of the droplet as

$$q''_{\mathrm{d}}(r_{\mathrm{d}}) = \frac{2K_1 B}{r_{\mathrm{b}}^2 T_{\mathrm{sat}}}[r_{\mathrm{b}}\Delta T_{\mathrm{t}} - K_2 \sin\theta \times \left[\left(1+\frac{B}{r_{\mathrm{b}}}\cos\theta\right)\ln\left(1+\frac{r_{\mathrm{b}}}{B}\right) - \cos\theta\right] \quad (1)$$

Analysis and measurements of Graham and Griffith [13] showed that at atmospheric pressure more than 90% of the heat is transferred through drops smaller than 100 $\mu$m indicating the need for an accurate relation for predicting the heat conduction through the droplets particularly at the small sizes.

## 2. Apparatus

A centrifuge is designed to vary the body forces during dropwise condensation. The condenser surface is placed horizontally on the centrifuge so that horizontal centrifugal force becomes parallel to the condenser surface. A circular, gold plated copper surface is used to produce dropwise condensation. Steam which is used for condensation is produced in stainless steel boilers and circulated through a rotating chamber to obtain solid body rotation. Deionized water is used in the boilers and steam pressure is kept above atmospheric to prevent air entering into the steam chamber.

An overall view of the experimental apparatus is shown in Fig. 1. Apparatus employed to achieve high gravitational accelerations parallel to the surface of dropwise condensation, has a centrifuge to rotate a test piece, a test piece assembly to produce dropwise condensation, a boiler with electric heaters to generate steam, a cooling system including a refrigeration unit and controls to maintain the coolant at constant temperature at the desired level, an auxiliary condenser to condense the excess vapor and to concentrate the noncondensable gases, a suction unit to remove the noncondensable gases from the vicinity of the test surface, a temperature measuring system, using thermocouples and an associated mercury slip ring assembly, as basic components. The suction unit to remove the noncondensable gases from the vicinity of the test surface is shown in Fig. 1.

A 38 inch long steel shaft, with a cylindrical chamber attached to its upper end and a rotating union at its lower end, constitutes the centrifuge of the experimental apparatus. The main shaft, not only rotates and supports the cylindrical chamber that carries the test piece in it, but also has the function of transferring the coolant and thermocouple signals by means of the coolant channels and thermocouples in it. The rotating chamber which has a 15.56 inch inside diameter has the function of carrying the test piece assembly and also acts as a channel to spread the water vapor radially (Fig. 2). The thermocouples coming from the main shaft are also conveyed to the test piece through the rotating chamber and are inserted into the test holes on the test piece.

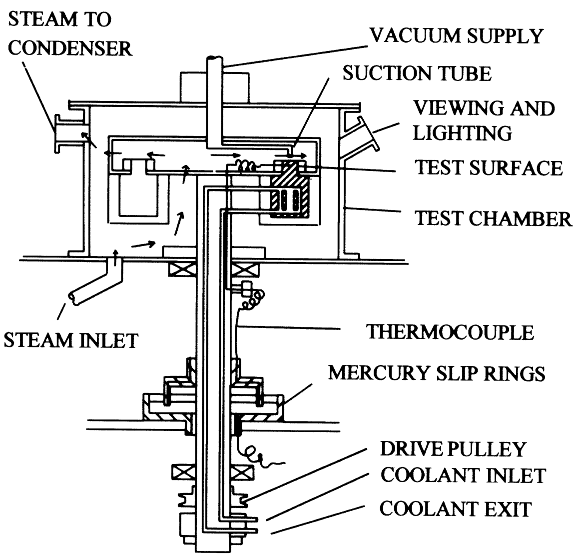


Fig. 1. Overall test assembly.

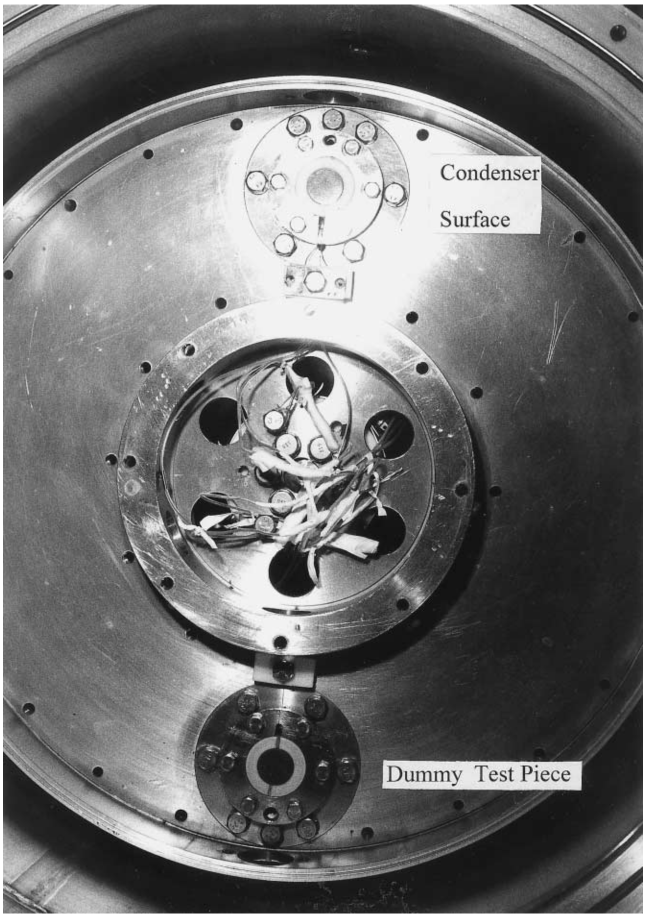


Fig. 2. Top view of the rotating chamber (mirror support plate is removed).

The rotating chamber has a small height to diameter ratio which cause the water vapor to spread axisymmetrically and because of this one-dimensional radial flow water vapor flows parallel to the surface of condensation on the test piece.

Slow radial velocity of the vapor over the surface of condensation helps to remove the noncondensable gases there, but as it will be shown in the results, this velocity is not sufficient to obtain a surface free from noncondensable effect, specially, at low centrifugal speeds.

The test piece (Fig. 3) is attached to the lower plate of the rotating chamber, with its axis of symmetry being parallel and 6 in away from the axis of rotation of the main shaft. The specially designed rotating chamber provides a solid body rotation for the vapor flowing radially in it, by eliminating the shear and turbulence effects of the surrounding vapor. Were the shear effects associated with rotation not eliminated by solid body rotation, the droplets would be spread over the condenser surface which would result in a reduction in the heat transfer coefficient, as occurs when dropwise condensation takes place on the outside of a rotating vertical shaft [14]. To expose the rotating chamber to vapor fully and also to provide an adequate vapor flow in it, in addition to providing safety at high rotational speeds, the rotating chamber is housed within a test chamber which consist of a cylindrical shell of inside diameter of 17 1/2 inch and a height of 13 1/2 inch. To reduce the heat losses through the walls of the test chamber, and consequently the heat load of the boiler, the surface of the test chamber is completely insulated. To prevent the diffusion of the air from outside into the test chamber, the main shaft inlet into the test chamber at the bottom and the suction tube inlet into the test chamber at the top is sealed by means of teflon and buna o-rings respectively. To further help to create a noncondensable-free environment in the test chamber, its pressure is always kept a couple of inches of water above the atmospheric pressure and the difference between the test chamber pressure and atmospheric pressure is continuously measured by a u-tube water

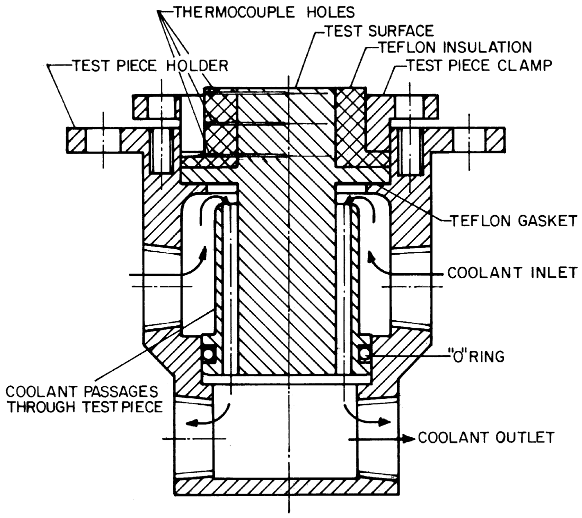


Fig. 3. Assembly of the test piece in the holder.

manometer. This differential pressure is used to calculate the saturation temperature of the vapor in the test chamber.

To keep the temperature of the coolant that enters into the test chamber as low as possible and also to keep the oscillations of the coolant and test piece temperatures at a minimum, the portion of the main shaft that remains in the test chamber and the coolant channels between the main shaft and the test piece are sealed by solid Teflon and Teflon tape. To prevent the leakage of condensed water between Teflon and the cold surfaces, hence the reduction in the insulation effect, solid Teflon and Teflon tape are held in place tightly by stainless steel clamps. Special attention is paid in insulating the cold surfaces such as test chamber surfaces and coolant channel surfaces, since on such areas condensation rate is very high which results in large amount of noncondensable gas accumulation which might effect the dropwise condensation measurements. Elimination of noncondensable gases accumulated at such cold spots is not possible by purging through the auxiliary boiler.

Water or Freon is used as a coolant to cool the test piece. When water is used for cooling, it enters the main shaft through the dual rotating union attached to the bottom end of the main shaft, coolant water flows through a coaxial tube at the center of the main shaft then it branches out to coolant channels to the test chamber housing. After completing the cooling process in the test piece, the coolant water returns to the main shaft through the coolant channels. After flowing downward annularly in the main shaft, coolant water exits at the dual rotating union. In the case of using freon as a coolant, the main shaft, together with the coolant channels in the test piece, act as the evaporator of the refrigerating unit employed.

## 3. Elimination of noncondensable gases

The accumulation of non-condensable gases on the condenser surface was reduced and minimized by means of three different elements; an auxiliary condenser, the centrifuging action of the system, and the use of an adjustable suction device near the condenser surface.

### *3.1. Auxiliary condenser*

An auxiliary condenser is connected to the test chamber shell and receives the vapor after passing through the test chamber. It consists of a stainless steel cylindrical vessel of 6 1/2 inch ID, containing stainless steel coiled tubing within which city water flows, producing condensation on the outside. The condensate returns to the boilers via a tube exiting from the bottom of the auxiliary condenser. Non condensable gases present in the test chamber steam, that accumulate on the surface of the coolant coils are carried out of the auxiliary condenser by continuous venting of the vapor which takes place from the top. The values of the heat transfer coefficients obtained with dropwise condensation indicate that most of the noncondensable gases present in the test chamber are eliminated by the extensive purging process which takes place before each test.

### *3.2. Centrifugal forces*

Continuous accumulation of non-condensable gases results in a concentration boundary layer on the condenser surface. The molecular weight of the non-condensables, basically air, are larger than that of the vapor, which results in a density variation in the direction perpendicular to the condenser surface, with a corresponding centrifuging action in the radial direction which is expected to reduce the non-condensable gases significantly.

Although the combined action of the auxiliary condenser and the centrifugal forces help to reduce the noncondensable on the condensing surface, these are not completely eliminated, as will be discussed with the presentation of the experimental results. In order to further remove the noncondensable gases, an interfacial suction system was designed and installed.

It is shown, both by analysis [15] and experiments [4] that the heat transfer coefficient is increased considerably if local suction takes place.

To remove the noncondensable gases accumulating at the condensing surface a suction tube of 1/16 inch outer diameter is installed at the condenser surface with a differential screw system to adjust its distance from the condenser surface precisely. The tube is made as small as possible to reduce the disturbances on the condenser surface to a minimum. It is located above the condenser surface on an axis parallel to that of the test piece displaced 1/8 inch away from the center of rotation, since it is expected that the maximum concentration occurs at a point slightly away from the center of the test piece. A differential screw of 24 threads per inch on one end, and 32 threads per inch on the other end makes it possible to adjust the distance of the suction tube from the condenser surface to within 0.001 inch. An identical unit is installed symmetrical to the center of rotation to insure that the centrifuge is balanced. The suction tube is attached to a tube at the center of rotation which extends above the cover plate, sealed by means of a Teflon seal. The shaft is centered by a small bearing at the top, and a single flow rotating union is used to connect the rotating tube to a vacuum system. The mixture of vapor and noncondensable gases passes over two sets of coils cooled by ice water and a glass flask in ice before passing to a vacuum pump.

Water is used as a working fluid in this study. The fact that the ultimate goal of dropwise condensation research is to make it applicable to industry, especially to condensers in power plants where the condensation of water, in general, takes place is another reason to choose water as a working fluid in this work. To generate the vapor used in the experiments a stainless steel boiler with stainless sheathed electric heaters having a total capacity of 14 1/2 kW and with a total water capacity of about 30 gallons which provides vapor for a sufficiently long period of time to obtain experimental data for a set of values of coolant temperature without any interruption is used. The saturated vapor produced in the boiler enters the bottom of the test chamber through a 1.435 inch ID pipe which also serves to return the condensate from the test chamber to the boiler. Vapor enters the rotating chamber through holes at the bottom plate, then moves radially out, passing over the test surface horizontally, and leaves the rotating chamber through holes at the outer side. The vapor then enters the auxiliary condenser through a hole in the test chamber shell slightly above the rotating chamber. The distilled water used in the boilers is charged by means of a laboratory metering pump through ion exchangers to further purify it.

The refrigerating unit designed and constructed to control the temperature of the coolant has the options of (a) pumping water through the centrifuge whose temperature is regulated by either evaporating Freon-12 or hot or cold water in an auxiliary heat exchanger, or (b) evaporating the Freon-12 directly in the test piece so as to provide coolant temperatures down to $-20^{\circ}$C. The cooling system consist of compressor, evaporator, condenser, coolant storage tank and various piping and control valves with Freon-12 as the refrigerant. When Freon is used to cool the test piece directly by appropriate valving by the evaporator of the refrigeration system, Freon-12 bypasses the evaporator and goes directly to the centrifuge via the dual rotating union. After circulating through the test piece, which now acts as the evaporator, the Freon vapor returns to the compressors. To regulate the superheat of the Freon prior to entering the compressor, a flow control valve is provided between the condenser and the test piece, and to regulate the test piece temperature, an evaporator pressure regulator valve is connected between the test piece and the compressors. It was also observed that the temperature of the test piece oscillated due to two phase flow instabilities in the freon. These oscillations were eliminated by inserting two additional surge tanks in the flow path, one preceding and another following the rotating union. The test piece on which dropwise condensation takes place is machined from high purity copper and has essentially two components. The upper component has a cylindrical shape with a diameter of 3/4 inch, the end of which is coated with gold to a thickness of about $10^{-7}$ m to promote dropwise condensation. To obtain one-dimensional heat transfer in the axial direction so that the computation of the heat flux and extrapolation to the surface temperature can be made using the one-dimensional theory, the cylindrical upper component of the test piece is made long enough and is insulated by a Teflon piece which fits closely over it.

The cooling of the test piece takes place at the lower component which is inserted into a housing and is in contact with the coolant medium. To increase the heat transfer area radial holes are drilled parallel to the axis of the test piece, where coolant enters at the bottom and after flowing through these holes it surrounds the bottom component of the test piece and leaves the housing.

The following procedure is used to prepare the condensation surface. The surface is polished with fine emery paper, then with rouge covered cloth. To prevent rounding of the corners all polishing is done on a glass surface. The surface is then rubbed with a lens cleaning tissue soaked with a jewelers rouge and kerosene suspension. After polishing, the surface is cleaned by acetone and rubbed by dry cleaning tissue to remove any particles adhering to the surface. During the coating process a dark area is sometimes observed at the edges of the coated surface. It is thought that this dark area is caused by excessive heating of the sharp corners. In subsequent coating processes a circular collar was placed around the test piece surface to eliminate the sharp corners, and the dark ring on the surface was no longer observed.

## 4. Procedure

Experimental data is collected by varying centrifugal force while the coolant temperature is kept constant. At each rotational speed data is taken with and without surface suction to determine the noncondensable effect. Tests are done for two different suction heights in order to determine whether it has an effect on noncondensable present on the surface.

In all experiments conducted, special attention is given to maintain a vapor pressure at least 5 inch of water above atmospheric to insure that air does not enter the test chamber.

Due attention is given to make sure that the coolant system is completely leak tight within and around the rotating chamber. This becomes especially important when freon is employed as the cooling medium. A minute amount of oil in the freon, when this is used as the cooling medium, contaminates the steam. Freon, when it leaks, also interferes with the dropwise condensation, being a noncondensable gas. To make sure that the portion of the main shaft that remains in the test chamber, the coolant channels and the connections are leak-free, the test chamber is filled with distilled water completely and the coolant channels are pressurized with compressed air.

The test surface used in the experiments is coated with gold by sputtering technique thick enough ($2 \times 10^{-7}$ m) to obtain good dropwise condensation. Woodruff and Westwater [16] showed that a coating thickness of at least $10^{-7}$ m was necessary for adequate dropwise condensation. The diffusion of copper atoms into the gold coating reduces the hydrophobic character of the surface if the coating is not thick enough. It has also been demonstrated that a thin nickel barrier layer between the gold and the copper substrate reduces the diffusion of the copper atoms [17]. Such a coating procedure is used in initial trial experiments in the present work and found to improve the life of the gold coating, but was not used in subsequent tests for reasons of convenience.

To insure that dropwise condensation will indeed take place, the condensing test surface is exposed to steam at atmospheric pressure in a flask prior to assembly. Initially the surface is covered by a condensate film while with exposure over an extended period (about 3 h) a gradual transition from almost entirely filmwise condensation to dropwise condensation is observed.

Since the suction tube is attached to the cover of the test chamber, its position with respect to the test surface changes after each dismantling. A special procedure is followed to adjust the position of the suction tube without contacting and damaging the gold surface. A clean, dry cleaning tissue of known thickness is placed on the test surface and the suction tube is lowered until it touches the surface with a very slight pressure. The tube is then raised by a differential screw to a distance of 1.3 mm between the surface and the tip of the suction tube. The suction shaft is then centered to be concentric with the rotating shaft using an indicator.

## 5. Results

Results are presented as the variation of heat flux with surface subcooling at various acceleration levels and also as the variation of heat transfer coefficient with acceleration. Data taken with surface suction on and off at two different suction heights are presented on these graphs.

The variation of the heat flux with surface subcooling at various acceleration levels are seen in Figs 4–6 where the data appear to be clustered at various levels of surface subcoolings because of the different cooling systems used. The experimental data in these figures are represented by four different types of symbols. The circles and triangles represent the data when the surface suction is on whereas the rectangles and diamonds represent the data when the surface suction is off. Circles and rectangles are used for the situation where the surface suction is closer to the condensation surface (1.318 mm) and triangles and diamonds are used when the surface suction is farther from the surface (2.671 mm). Examination of Figs 4–6 show that heat flux data with surface suction falls approximately on a line passing through the origin over the heat flux range investigated in this study.

Figures 7–9 show the variation of the heat transfer coefficient versus centrifugal acceleration, each figure corresponding to data obtained with a particular range of heat flux as indicated. In these figures the coolant temperature is maintained approximately constant for each of the tests. However, since the heat transfer coefficient for dropwise condensation which changes with acceleration, is a small part of the total thermal resistance between the vapor and the coolant, the constant overall temperature difference implies an approximately constant heat flux.

Examination of Figs 7–9 shows that although the surface subcooling does not have an appreciable effect on the heat transfer coefficient, the measurements made with freon as the cooling medium and with a cooling medium at 0°C, give slightly higher heat transfer coefficients.

In this study, heat flux on the condensation surface is obtained by means of three chromel–constantan thermocouples located along the axis of the cylindrical part of the test piece. Two different methods are used to obtain the heat flux from the temperature measurements. In the first method the heat flux is computed by taking a linear temperature gradient with the readings of the two thermocouples closest to the condenser surface. In the second method heat flux is obtained from a least square curve fit of the measured three temperatures along the axis of the cylindrical part of the test piece. The condenser surface temperature is extrapolated from the computed temperature gradient. The purpose of computing the temperature gradient with two different methods is to see whether the third thermocouple farthest from the condenser surface is effected by two-dimensional effects in the lower part of the test piece. Comparison of the two methods showed that such two-dimensional effects are negligible. Each experimental run consists of the data obtained by starting at the lowest acceleration level ($1 \times g$) and gradually increasing it to the highest acceleration level ($100 \times g$). Each time the coolant tem-

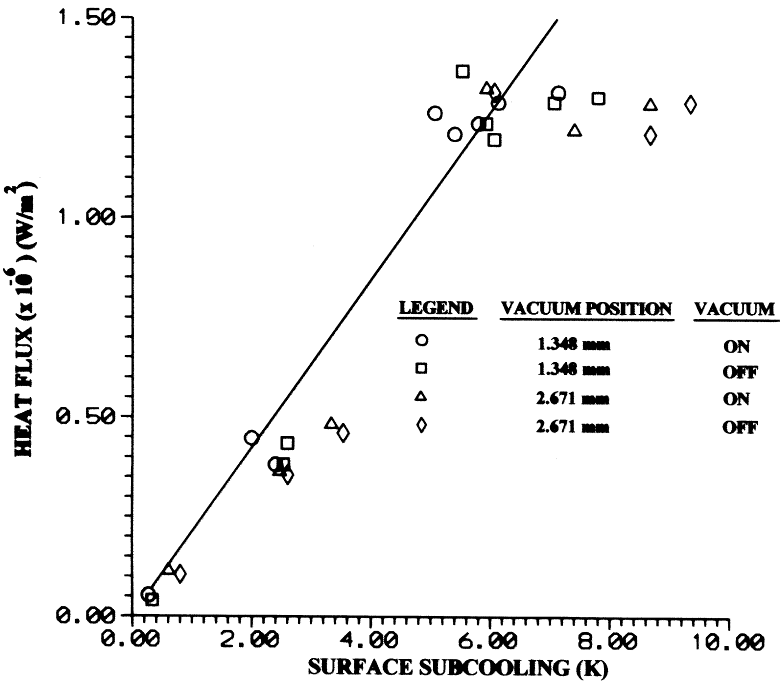


Fig. 4. Measurement of heat flux and surface subcooling ($a/g = 1$).

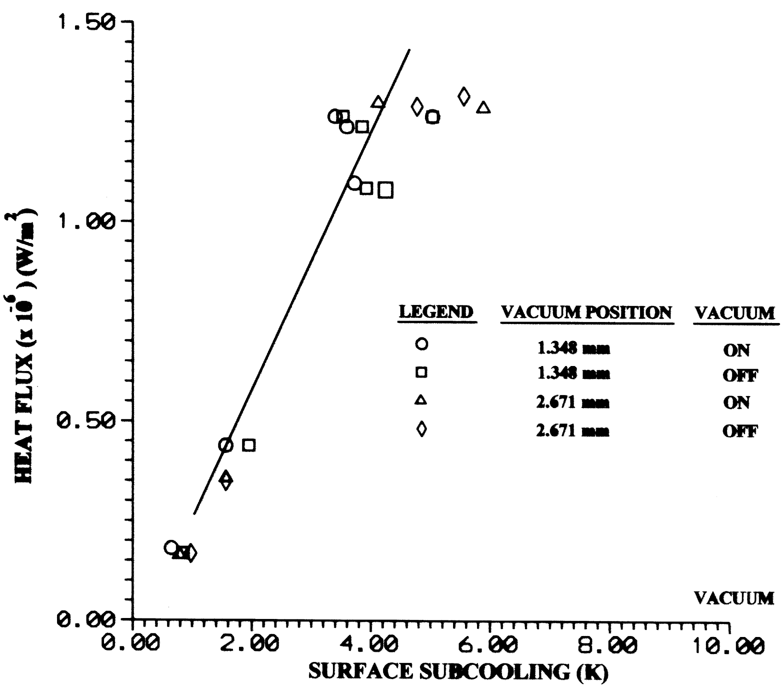


Fig. 5. Measurement of heat flux and surface subcooling ($a/g = 10$).

perature or the coolant system is changed, the condensation surface prepared and plated again. Consequently surface characteristics of the condensation surface in each run is expected to be different. This difference in surface characteristics and therefore in the number of the nucleation sites might be the major reason in

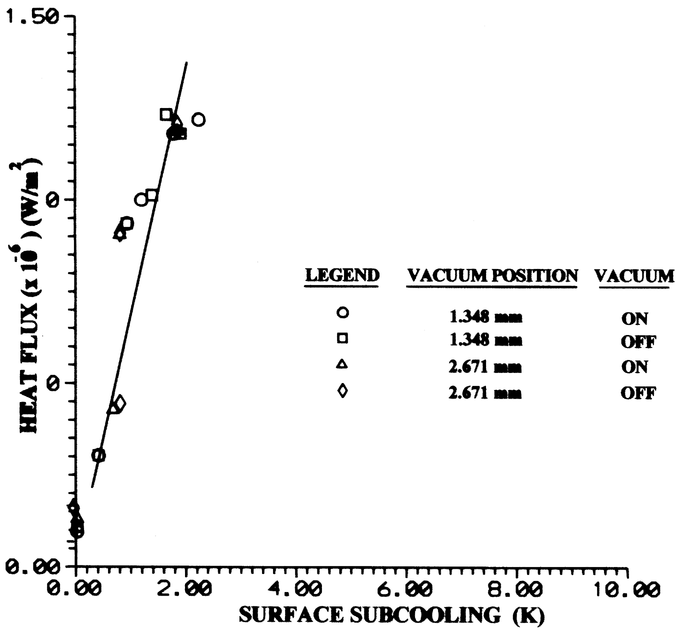


Fig. 6. Measurement of heat flux and surface subcooling ($a/g = 100$).

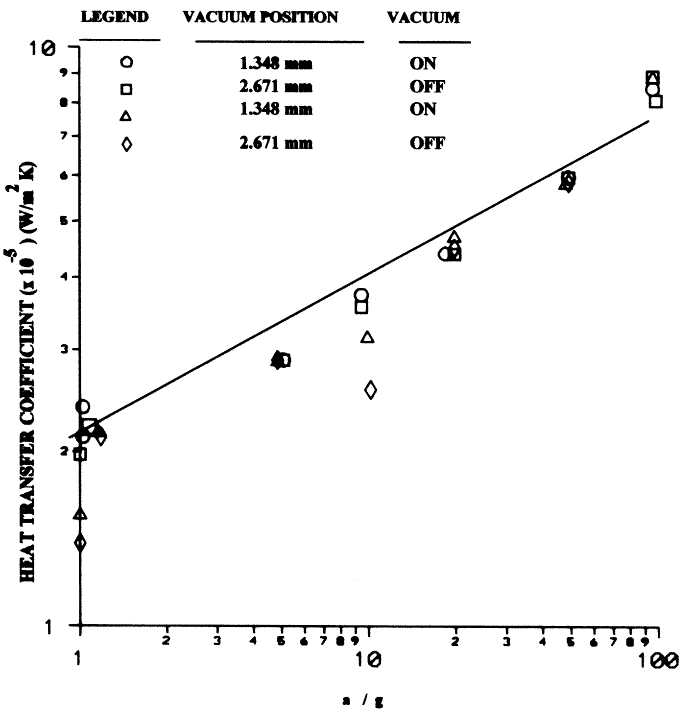


Fig. 7. Heat transfer coefficient versus acceleration for range of heat flux = $0.137 \times 10^7$–$0.94 \times 10^6$ W m$^{-2}$ (Freon cooling system).

the small variations in the experimental results at different experimental runs. In most cases repeat measurements were made at $a/g = 1$, following each measurements at higher acceleration and serve as an anchor point to check the reproducibility of the results.

The effect of surface suction on dropwise condensation

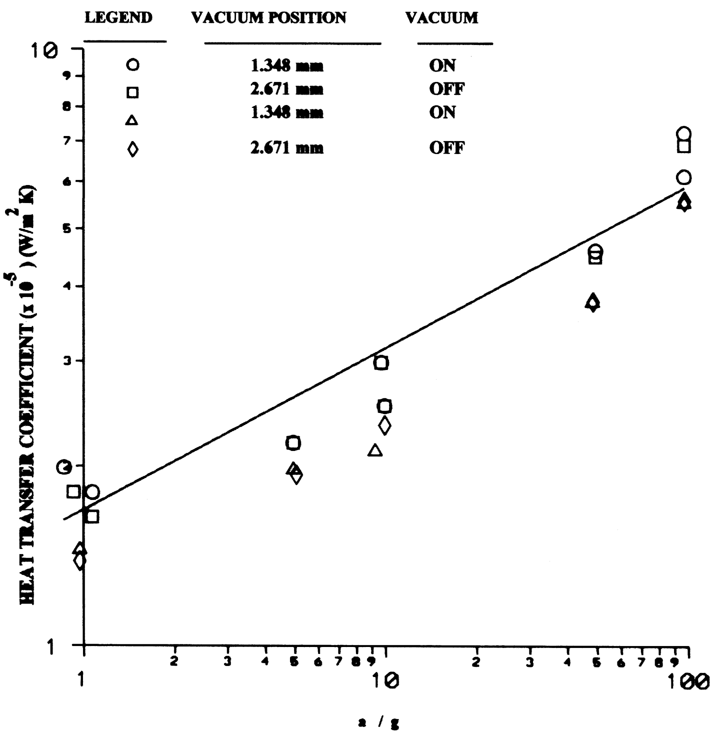


Fig. 8. Heat transfer coefficient versus acceleration for range of heat flux = 0.133 × $10^7$–0.118 × $10^7$ W m$^{-2}$ (coolant water at 25°C nominal temperature).

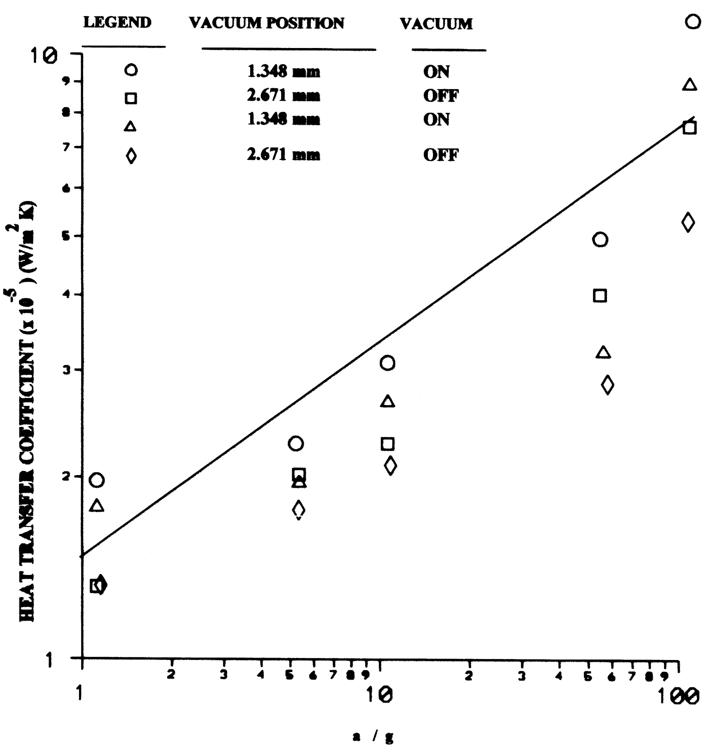


Fig. 9. Heat transfer coefficient versus acceleration for range of heat flux = 0.8 × $10^6$–0.45 × $10^5$ W m$^{-2}$ (coolant water at 95°C nominal temperature).

heat transfer is seen in Figs 4–9. It is clearly seen in Figs 4–9 that the difference between the heat transfer coefficients with the vacuum on and off decreases with increasing body force.

Although some exceptions exist, one notes that surface suction has a definite effect on the heat transfer coefficient at $a/g = 1$, a somewhat lesser effect at $a/g = 10$, and an insignificant effect at $a/g = 100$. These indicate that non-condensable gases present on the surface at $a/g = 1$ and 10 are removed by centrifugal forces at the higher acceleration levels.

In order to minimize the effect of the suction on the dropsize distribution the inside diameter of the suction tube was made quite small, the ratio of the suction tube area to the total area of the condensing surface is $7 \times 10^{-3}$. It is therefore expected that the effect of the suction on the dropsize distribution over the entire surface will be small. If the increase in the heat transfer coefficient with suction had been partly due to the disturbance of the droplets on the surface, this effect would have been observed at all times.

Examination of the data points at $a/g = 1$ and $a/g = 10$ show that, in some cases, the difference between the data with and without suction is negligibly small. In general, changing the distance of the suction tube from the condenser surface produced only a small change in the dropwise condensation heat transfer coefficient. The spacing of 2.671 mm gives heat transfer coefficients slightly lower compared to that with a spacing of 1.348 mm. This shows that spacing of 2.671 mm was sufficient to adequately eliminate noncondensable gases. It is also noteworthy that the differences between the heat transfer coefficients with the two different spacing decreases as the body forces increases. This result also denotes that removal of non-condensables is helped by centrifugal forces.

A least square straight line fit to the data points for the closest position of the vacuum (circular symbols) are shown in each of Figs 4–9. In Figs 7–9 the slope of the least square straight line fit varies between 0.204 and 0.323, with an average value of 0.270. In a previous study on the effect of body forces on dropwise condensation [2] this slope was predicted to be 1/6 (0.167), based on another model. The difference between the slope obtained, experimentally in the present work, and that predicted in Ref. [2] is believed to be due to the omission of the effects of the sweeping process in the latter work. Since the increased sweeping frequency at higher accelerations causes an increase in the heat transfer coefficient, the omission of this phenomenon results in a lower slope in the plot of the heat transfer coefficient versus body force. Measurements of the effects of body forces on dropwise condensation have also been made by Tanasawa et al. [1]. A least square straight line fit to the data points of Ref. [2] and of Ref. [1] gives slopes of 0.223 and 0.243, respectively, on plots of heat transfer coefficient versus body force. This is to be compared with the experimental value of 0.27 obtained in the present work.

## 6. Conclusions

(1) On the basis of experimental measurements conducted, it was found that the heat transfer coefficient for dropwise condensation varied on the average as the 0.27 power of the body force.
(2) The effect of non-condensable gases might be significant at low acceleration levels.

## References

[1] I. Tanasawa, J. Ochiai, S. En-ya, Y. Utaka, Dropwise condensation—the effect of the critical size of drop detachment (Part I), English translation of the preprint submitted to the Japan Society of Mechanical Engineers, Tottori Conference, November 1973.
[2] S. Kodali, A study of dropwise and film condensation under variable gravity fields, Ph.D. thesis, The University of Michigan, Ann Arbor, MI, 1981.
[3] E. Schmidt, W. Schurig, W. Sellschop, Versuche uber die Kondensation von Wasserdampf in Film- und Tropfenform, Tech. Mech. Thermo-Dynam. 1 (1930) 53.
[4] E.J. Le Fevre, J.W. Rose, An experimental study of heat transfer by dropwise condensation. Int. J. Heat Mass Transfer 8 (1965) 1117–1133.
[5] J.L. McCormick, J.W. Westwater, Nucleation sites for dropwise condensation, Chemical Engineering Science 20 (1965) 1021–1031.
[6] E.J. Le Fevre, J.W. Rose, A theory of heat transfer by dropwise condensation, Proceedings of Third International Heat Transfer Conference vol. 2, 1966, pp. 362–375.
[7] L.R. Glicksman, A.W. Hunt, Numerical simulation of dropwise condensation, Int. J. Heat Mass Transfer 15 (1972) 2251–2262.
[8] H. Merte Jr., C. Yamali, Profile and departure size of condensation drops on vertical surfaces, Warme- und Stoffubertragung 17 (1983) 171–180.
[9] I. Tanasawa, J. Ochiai, Y. Funawatashi, Experimental study on dropwise condensation-effect of maximum drop size upon the heat transfer coefficient, Proceedings of Sixth International Heat Transfer Conference, vol. 1, 1968, pp. 74–81.
[10] J.L. McCormick, J.W. Westwater, Drop dynamics and heat transfer during dropwise condensation of water vapor on a horizontal surface, Chemical Engineering Progress Symposium Series 62 (1966) 120–134.
[11] C. Yamali, H. Merte Jr., Conduction heat transfer through condensation drops with variable interphase coefficients, ASME Winter Annual Meeting in New Orleans, LA, A.B.D., vol. 38, 1984, pp. 143–149.
[12] A. Umur, P. Griffith, Mechanism of dropwise condensation, Trans. ASME. J. Heat Transfer 87C (1965) 275–282.
[13] C. Graham, P. Griffith, Drop size distributions and heat transfer in dropwise condensation, Int. Heat Mass Transfer 16 (1973) 337–341.
[14] D.C.P. Birt, J.J. Brunt, J.T. Shelton, R.G.H. Watson, Methods of improving heat transfer from condensing steam

and their applications for condensers and evaporators, Trans. Instn. Chem. Engrs. 37 (1959) 289–296.

[15] A. Tamir, Y. Taitel, Improving condensation rate by interfacial suction and forced convection in the presence of noncondensable gases, Israel Journal of Technology 9 (1971) 69–81.

[16] D.W. Woodruff, J.W. Westwater, Steam condensation on electroplated gold: effect of plating thickness, Int. J. Heat Mass Transfer 22 (1979) 679–632.

[17] P. Waas, J. Straub, U. Grigull, The influence of thermal diffusivity of the condenser material on the heat transfer coefficient in dropwise condensation, Proceedings of Seventh International Heat Transfer Conference, Munich, Germany, vol. 5, CS5, 1982, 27–31.