JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER
Vol. 20, No. 3, July–September 2006

Review of Condensation Heat Transfer in Microgravity Environments

Yongping Chen*
Southeast University, Nanjing 210096, Jiangsu, People's Republic of China and Rensselaer Polytechnic Institute, Troy, New York 12180
C. B. Sobhan†
National Institute of Technology, Calicut, Kerala 673 601, India
and
G. P. Peterson‡
Rensselaer Polytechnic Institute, Troy, New York 12180

A review of recent investigations of condensation in microgravity environments is presented in order to better understand the influence of microgravity, shear stress, surface tension, and/or capillary and centrifugal forces on condensation phenomenon. Based on a thorough review of the available literature on condensation in the absence of gravitational body forces, the potential of using the vapor shear stress to remove the condensate in systems designed for spacecraft applications was assessed. Other removal methods, such as the use of wall suction are presented, discussed and evaluated. It is apparent from the literature that the gravitational body force in condensation processes could effectively be replaced by the centrifugal forces generated by a rotating system. The possibility of utilizing surface tension and/or capillary forces to enhance the condensation phenomenon for use in spacecraft applications is also evaluated and discussed.

Nomenclature

a	=	impinging density
c_p	=	specific heat at constant pressure
f_i	=	flow friction factor
g	=	gravity acceleration
h_{fg}	=	latent heat
h_x	=	heat transfer coefficient
l	=	plate length
$\dot{m}''_c$	=	condensate mass flux
Nu	=	Nusselt number
Pr	=	Prandtl number
p	=	pressure
q	=	local heat transfer rate
Re	=	Reynolds number
r	=	radius of curvature
U_∞	=	freestream velocity
w	=	mass flow rate
X_{tt}	=	Martinelli parameter
X_{vt}	=	Martinelli parameter
$\bar{\alpha}$	=	average heat transfer coefficient
ΔT	=	Temperature difference between saturated vapor and wall
δ^*	=	dimensionless film thickness
λ	=	thermal conductivity
μ	=	absolute viscosity
ρ	=	density
σ	=	surface tension
τ	=	shear stress
ω	=	angular velocity

Subscripts

G	=	gas
L, l	=	liquid
v	=	vapor

I. Introduction

CONVENTIONAL thermal control systems, consisting of mechanically pumped, single-phase fluid loops, require large amounts of power to overcome friction and meet the high heat-removal demands of advanced spacecraft thermal control systems. With the development of more advanced applications, these spacecraft thermal control systems must be able to transport and dissipate large quantities of heat over moderate to long distances with very small external power requirements, and require more flexible and effective thermal management systems than are currently available.[1] As a result of the combination of high heat flux capacity, low weight-to-power-dissipation ratio, and high degree of temperature uniformity, phase-change condensing radiators offer an effective alternative to the single-phase pumped fluid loops currently in use in most spacecraft thermal control applications. In these applications, condensation heat transfer could be used to transfer heat from the source to the radiators, where, through phase change, the desired heat transfer can be achieved and the waste heat ultimately dissipated from the radiator surface to space. Although the ultimate heat dissipation from the radiator is by thermal radiation, the use of phase change and in particular the condensation process that occurs in the condenser regions serves to spread heat uniformly over the radiator surface. Effective condensing radiators are capable of dissipating high heat fluxes with small temperature differences and have high rejection-to-weight ratios. The successful design of these systems requires a fundamental understanding of the parameters that govern the mechanisms of condensation in reduced-gravity environments. To assist in the development of this understanding, a thorough review of the current literature has been conducted.

Condensation under a normal ($1g$) gravitational field has been investigated extensively[2–5] and a comprehensive review of the

Received 13 July 2005; revision received 20 September 2005; accepted for publication 23 September 2005.

*Professor, Key Lab of Clean Coal Power Generation and Combustion Technology of Ministry of Education, Department of Power Engineering; ypchen@seu.edu.cn.

†Assistant Professor, Department of Mechanical Engineering, Center for Nanotechnology; csobhan@nitc.ac.in.

‡Professor, Department of Mechanical, Aerospace and Nuclear Engineering, Office of the Provost, Troy Building; peterson@rpi.edu. Fellow AIAA.

Table 1 Variation of heat transfer coefficient with heat flux[7] (see Fig. 1)

Curve	Orientation	Surface material	Surface finish	Promoter	Venting arrangement	Gas concentration
1	Vertical	Copper	Mirror-smooth	Stearic acid	Unknown	Unknown
2	Vertical	Copper	Mirror-smooth	Benzyl mercaptan and oleic acid mixed	Continuous bleed	Unknown
3	Vertical	Chromium plated on copper	Highly polished	Oleic acid	Unknown	Unknown
4a, 4b, 4c	Vertical	Copper	Mirror-smooth	Dioctadecyl disulfide	Blow past surface, various velocities	2 ppm
5	Vertical	Copper	Mirror-smooth	Dioctadecyl disulfide	Close by venting	Very small
6	Vertical	Copper	Mirror-smooth	Dioctadecyl disulfide	Blow past surface	Very small
7	Vertical tubes	Chromium plated on copper	Unknown	Oleic acid	Continuous bleed	Unknown
8	Vertical	Copper	Unknown	Oleic acid	Blow past surface	Unknown
9	Vertical tubes	Copper	Unknown	Benzyl mercaptan	Continuous bleed	Unknown
10, 10a	Vertical	Copper	Highly polished	Benzyl mercaptan	Blow past surface	Unknown
11	Vertical	Copper	Polished	Cupric oleate	Unknown	Unknown
12	Vertical	Copper	Unknown	Benzyl mercaptan	Blow past surface	Unknown

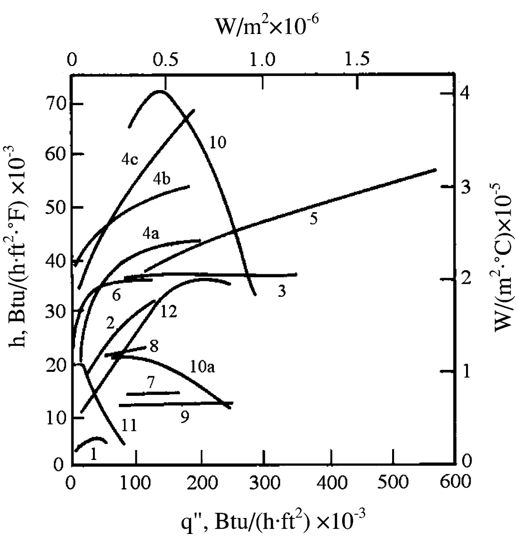


Fig. 1 Dropwise condensation heat transfer data[7] (see Table 1).

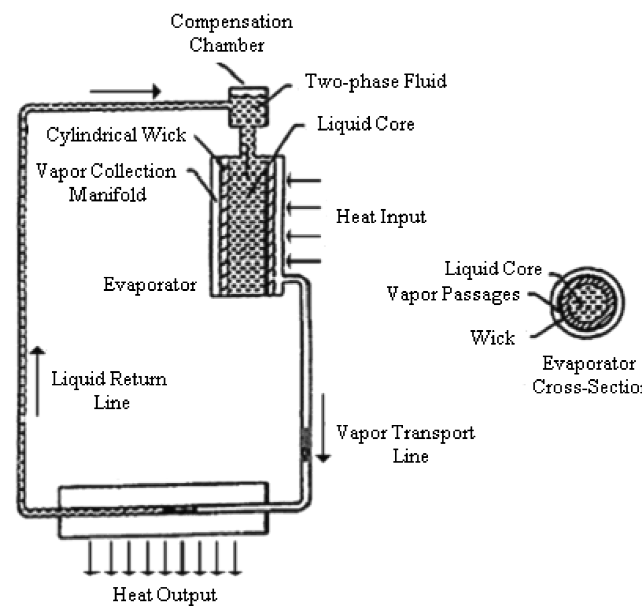


Fig. 2 Schematic of loop heat pipe.[11]

published literature of condensation heat transfer inside smooth and micro-finned tubes in normal gravitational fields has been presented by Garcia-Valladares.[6] The correlations presented were discussed and benchmarked with experimental data from a number of different investigations for several different fluids and flow conditions. The most accurate correlations, based on comparisons with experimental data for the different fluids and boundary conditions, were also identified and discussed.

Although it has been known since the 1930s that heat transfer coefficients for dropwise condensation are much higher than those for film condensation, until the 1960s, there were wide discrepancies among the results presented by various investigators. As shown in Fig. 1, the model comparison using either numerical sample calculation or graphical representation was presented to clearly show both the effect of the governing parameters and overall trends in prediction for dropwise condensation.[7] The conditions for these data are indicated in Table 1. All these curves refer to a pressure close to 1 atm. Rose[8] has reviewed the progress in dropwise condensation research from 1930 to the present and the theoretical aspects and measurements relating to heat transfer, with particular attention given to the effects of surface conditions. In this review, it was also noted that experimental measurements showed good consistency and provided better understanding of the mechanism and theory of dropwise condensation. Still, the practical problem of promoting stable dropwise condensation under industrial conditions remains unsolved despite considerable effort by numerous investigators.

Analytical and experimental investigations to determine the influence of the inclination on laminar condensation have been presented by Chato.[9] A momentum–energy integral method for condensate formation on the walls and condensate flow at the bottom were developed. It was noted that there should be an optimum slope of inclination, which produces maximum heat transfer for a tube operating under a set of prescribed conditions.

A review and comparative study of heat rejection systems for space platforms and space-based power systems including space-constructible heat-pipe radiators, variable-surface-area radiators, rotating solid radiators, moving belt radiators, rotating film radiators, liquid droplet radiators, Curie point radiators, and rotating bubble membrane radiators, has been presented by Chowdhury and Peterson.[10] The most technically feasible of these systems was found to be one that utilizes phase change and the associated large latent heat values obtained from evaporation and condensation of a fluid medium within a circulating loop, as in the loop heat pipe. Figure 2 illustrates a schematic diagram of a loop heat pipe.[11] To examine the performance of loop heat pipes, SINDA/FLUINT was developed by NASA.[12,13] This analyzer was specifically targeted toward modeling requirements of loop heat pipes and other capillary devices, and it is capable of simple steady-state sizing analysis as well as complex start-up transients including the effects of noncondensible gases. Developing such a circulating loop system for microgravity applications requires an in-depth understanding of condensation under microgravity and the resulting two-phase flow. In the absence of gravitational body force, that is, in space applications, the nature of the condensation process is quite different from that occurring on the ground. For example, in film condensation, the dominant heat flux is a function of the liquid film conductivity, so the thermal resistance of the liquid film is an important factor. It is very important to effectively drain the condensate liquid film from the heat transfer surface to prevent condenser flooding and stoppage of the vapor flow. The motion of the liquid film is driven by the gravitational body force on the ground; however, in the absence of acceleration-induced body forces, no similar driving forces exist in space environments. Utilization of the vapor flow shear stress, of suction through a porous wick surface, and of surface tension and the associated capillary driving force and application of centrifugal forces have all been presented and discussed in the literature as possible solutions to this problem, but only a few investigations have been reported on the behavior of in-tube condensation in the absence of a gravity force since the 1950s.

The effects of gravity, shear forces, and surface tension in internal flows with condensation have been investigated through direct computational simulations.[14] In this investigation, the two-dimensional governing equations for steady and unsteady laminar/laminar internal condensing flows were solved. The flow geometry, in normal or zero gravity, was chosen to be the inside of a rectangular channel with film condensation on one of the walls. Under normal gravity, film condensation was on the bottom wall of a tilted channel, where the tilt orientation was varied between vertical and horizontal. It was found to be important to know whether the exit conditions were constrained or unconstrained, because nearly incompressible vapor flow occurs only for exit conditions that are unconstrained. Compared to a vertical channel under normal gravity, shear-driven zero-gravity cases resulted in much larger pressure drops, much lower wave speeds, much larger noise-sensitive surface-controlled wave amplitudes, and much narrower flow regime boundaries within which the vapor flow could be considered incompressible.

The flow patterns during condensation of R12 inside a 0.5-in horizontal tube were visually and photographically studied by Soliman and Azer.[15] Nine different flow patterns were identified and the study concluded that the flow regime map originally developed for adiabatic two-phase flow was not adequate for nonadiabatic two-phase flow.

As stated above, the mechanism of condensation in the absence of body forces is much different from that experienced in the presence of a gravitational body force. In addition, because of the difficulties of conducting experiments in reduced gravity environments, only very limited information is available, which describes the behavior of the flow and the heat condensation characteristics in these types of conditions. In order to obtain better insight into the physics of condensation in microgravity, a review, with emphasis on the influences of microgravity, shear stress, surface tension, and centrifugal forces, is presented. The available literature on condensation in the absence of dominant body forces, such as gravity, is examined, discussed, and evaluated.

II. Theoretical Investigations

A. Vapor Shear Force

One of the earliest analyses of the governing equations for film condensation over a flat plate was presented by Cess,[16] with a simplifying assumption that neglected the convection and the inertial forces in the condensate liquid film. Subsequently, Koh[17] reported on a detailed analysis of laminar film condensation of a saturated vapor in forced flow over a flat plate, as shown in Fig. 3. The problem was formulated with boundary-layer simplifications. From the numerical solutions of the governing equations, it was found that the energy transfer by convection could not be neglected for high–Prandtl number liquids. The condensation rate, skin friction, and heat transfer were all presented as functions of $c_p \Delta T / Pr h_{fg}$ with $[(\rho\mu)_L/(\rho\mu)_v]^{1/2}$ and Pr as parameters.

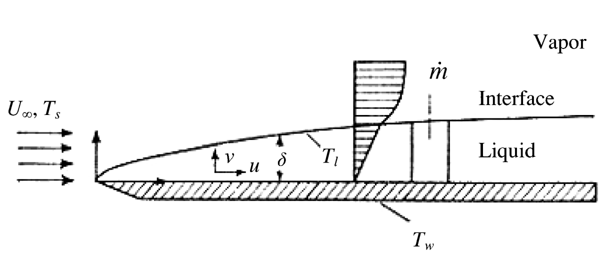


Fig. 3 Physical model for condensation on plate.[17]

It was observed, from the solution of this model and considering the convection heat transfer, that the dimensionless heat transfer, $Nu_x/\sqrt{(Re_x)}$ drops to a minimum value and then increases as $c_p \Delta T / Pr h_{fg}$ increases. Shekriladze and Gomelauri[18] also conducted research on condensing flow along a flat plate. Momentum transfer was shown to play a dominant role in the fluid dynamics of the process. A comparison between the numerical results and available experimental data was also presented and the following results were obtained:

1) For a constant wall temperature, the average heat transfer coefficient $\bar{\alpha}$ is given by

$$\bar{\alpha} = \sqrt{\frac{N}{N+1} \cdot \frac{\rho_L h_{fg} U_\infty \lambda}{\Delta T l}} \tag{1}$$

$$N = \frac{\lambda \Delta T}{h_{fg} \mu_L} \tag{2}$$

2) For a constant heat flux, the average heat transfer coefficient $\bar{\alpha}$ is given by

$$\bar{\alpha} = 1.41 \sqrt{\frac{\lambda^2 \rho_L U_\infty}{\mu_L l}} \tag{3}$$

Condensation in tube annuli is often used in the design of space applications where, in the absence of gravitational body force and buoyancy effects, the vapor condenses on the tube surface and the liquid attaches and creates a film on the tube inner wall due to the influence of surface tension. Under these conditions, a liquid film flows along the wall and a high-velocity vapor stream flows in the center of the tube. As the velocity increases, the liquid film is entrained and carried along and the interface between the vapor and the liquid forms a wavy surface, resulting in an increase in the pressure drop. A method for predicting the height of the wall layer and the interfacial drag in this annular flow in a circular tube was presented by Henstock and Hanratty.[19] For horizontal flows, the flow friction factor f_i based on the average shear stress τ_i around the circumference was represented by the relationship

$$f_i / f_s = 1 + 850F \tag{4}$$

where

$$f_s = 0.046 Re_G^{-0.20} \tag{5}$$

at low Re_{LF},

$$F = 0.0379 X_{vt}$$

at high Re_{LF},

$$F = 0.0379 X_{tt}$$

where

$$X_{vt} = 16.9 \left(\frac{\mu_L \rho_G W_L}{\mu_G \rho_L W_G} \right)^{0.5} Re_G^{-0.4}$$

$$X_{tt} = \left(\frac{\mu_L}{\mu_G} \right)^{0.1} \left(\frac{\rho_G}{\rho_L} \right)^{0.5} \left(\frac{W_L}{W_G} \right)^{0.9}$$

The possibility of using vapor shear stress to remove the condensate for space applications was assessed by Chow and Parish.[20] The film thickness was assumed to be small compared to the tube diameter and the inertial forces in the liquid were ignored. Examining the

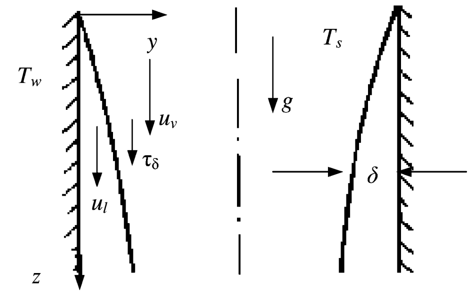


Fig. 4 Model of condensation on wall.[21]

momentum balance of a single fluid element inside the condensate film yields

$$\mu_L \frac{\partial^2 u_L}{\partial y^2} = \frac{\mathrm{d}p}{\mathrm{d}z} \tag{6}$$

and the shear stress at the interface can be expressed as

$$\tau_\delta = (c_f/2)_E \rho_v (u_v - u_{L\delta})^2 + \dot{m}_c''(u_v - u_{L\delta}) \tag{7}$$

For laminar vapor flow ($Re_v < 2.3 \times 10^3$), $(c_f/2)_E$ is given by

$$(c_f/2)_E = 8/Re_v \tag{8}$$

For turbulent vapor flow ($Re_v > 2.3 \times 10^3$), $(c_f/2)_E$ is given by

$$(c_f/2)_E = (c_f/2)(1 + 850F) \tag{9}$$

where

$$(c_f/2) = 0.023 Re_v^{-0.2}$$

$$F = \left[\left(1.414 Re_L^{0.5}\right)^{2.5} + \left(0.132 Re_L^{0.9}\right)^{2.5}\right]^{0.4} \mu_R \Big/ \left[(\rho_L/\rho_v)^{0.5} Re_v^{0.9}\right]$$

From a numerical solution of the governing equation, it was found that the vapor shear can introduce a liquid velocity that is about 10 to 20 times smaller than the vapor velocity. Assuming that in the absence of gravity, the motion of the condensate film is entirely due to vapor shear, Wang et al.[21] introduced a method for analyzing condensation heat transfer inside a tube in a microgravity environment. The schematic of the condensation flow model is shown in Fig. 4. Acceleration due to gravity was introduced into the momentum equation as a parameter varying in value from 0 to 9.8 m/s^2 in the form

$$\mu_L \frac{\partial^2 u_L}{\partial y^2} - \frac{\mathrm{d}p}{\mathrm{d}z} + \rho_L g = 0 \tag{10}$$

Condensate film thickness was found to increase and condensation heat transfer to decrease with a reduction in the gravitational body force, g. The model was benchmarked by experimentally measuring condensation heat transfer inside a vertical tube on the ground with normal acceleration due to gravity. Because no direct comparison was made with experimental data for microgravity, the accuracy of this method still needs to be ascertained.

Although there have not been many theoretical investigations of condensation in a space or reduced-gravity environment, the available literature points to the conclusion that vapor shear can enable effective continuous condensation in a microgravity environment.

B. Wall Suction

A description of the condensation heat transfer process in a microgravity environment has been presented by Chow and Parish.[22] In this investigation, two mechanisms for condensate removal were analyzed by examining two quite different situations. The first of these involved film condensation on a flat porous plate with the condensate being removed by suction at the wall. The second involved the analytical prediction of the heat transfer coefficient for condensing annular flows with the condensate film driven by the vapor shear. It was concluded that both suction and vapor shear can effectively drain the condensate to ensure continuous operation of the condensers when they are operated in a microgravity environment.

Removal of the liquid film in a porous tube has also been considered in an investigation of vapor flowing on the outside of a horizontal tube.[23] Faghri, and Chow[24] analyzed the effects of suction on the condensation of a vapor flowing inside a tube in a microgravity environment. In this investigation, the properties of the fluid were assumed to be constant and the curvature of the tube was neglected. The boundary conditions were taken as uniform temperature and uniform suction at the wall. The local heat transfer coefficient h_x was obtained in terms of the Nusselt number through the expression

$$Nu = h_x D_h/\lambda = \left\{-e^{-\delta^*} + 1\right\}^{-1} Pr_L Re_w \tag{11}$$

where Re_w is the Reynolds number for suction at the wall.

Three values of the Reynolds number for suction at the wall, Re_w, of −0.79, −1.6, and −5.0 were used in the numerical solution. Figures 5 and 6 illustrate the variation of the normalized film thickness with respect to the normalized axial distance for Reynolds numbers of the vapor at the inlet, $Re_{v,e}$, of 5×10^3 and 50×10^3. From the results, it is apparent that the heat transfer was increased

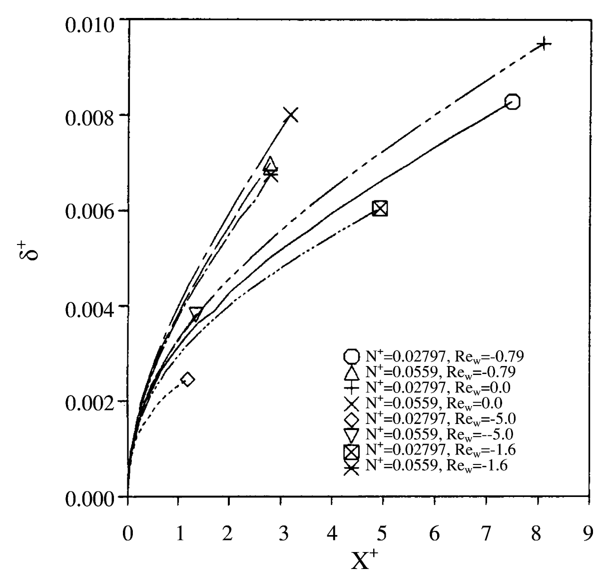


Fig. 5 The variation of δ^+ for $Re_{v,e} = 5 \times 10^3$ (Ref. 24).

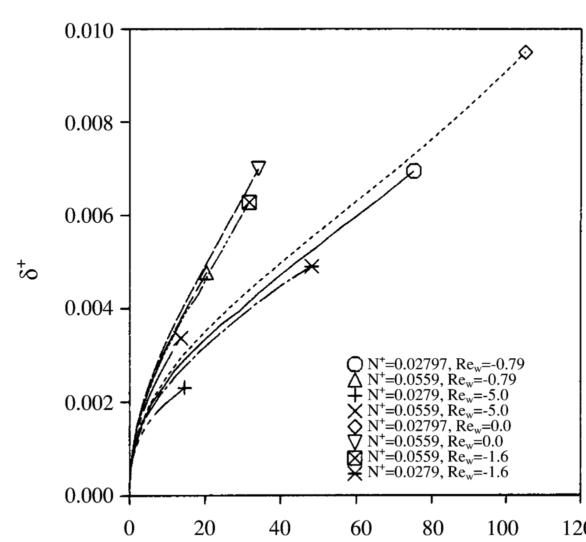


Fig. 6 The variation of δ^+ for $Re_{v,e} = 5 \times 10^4$ (Ref. 24).

by the use of suction and that the vapor condenses much faster in pipes with suction than without suction at the wall, for identical inlet vapor Reynolds numbers. This is clearly due to the reduction in the film thickness caused by the suction. The increases in heat transfer are smaller by comparison for small values of wall suction Reynolds numbers, but they become more significant when greater quantities of the condensate are extracted from the tube.

C. Centrifugal Force

It is well known that the gravitational force in a condensation process could be replaced by the artificial body forces caused by the centrifugal forces generated by a rotating system. According to Prenger and Sullivan[25] rotating film radiators were found to achieve a specific mass of 5.5 kg/kW or 3.5 kg/m^2, based on an analysis that considered the total emissivities.

Pioneering work on film condensation on a rotating disk situated in a large body of pure saturated vapor was presented by Sparrow and Gregg[26,27] and Sparrow and Hartnett.[28] The centrifugal field associated with the rotation of the plate was used to "sweep" the condensate outward along the disk surface. For cases where there were no gravitational body forces, two significant results were obtained:

1) The local heat transfer rate is proportional to the square root of the disk angular velocity ω:

$$q'' \propto \omega^{0.5} \tag{12}$$

2) The local condensation heat transfer coefficient is related to the temperature difference as expressed by

$$h(v/\omega)^{\frac{1}{2}}/\lambda = 0.904\left[(Pr/c_p\Delta T)^{\frac{1}{4}}\Big/h_{fg}\right] \tag{13}$$

The problem of laminar film condensation, including interfacial shear on a cooled rotating disk, was also considered by Sparrow and Gregg.[27] The analysis was carried out for condensation of a pure, saturated vapor on an isothermal disk. It was observed that for the case of $Pr > 1$, the vapor drag at the liquid–vapor interface can be neglected.

There are a few reported experimental investigations of condensation in the absence of gravitational body forces, the most significant of which was conducted by Butuzov and Rifert[29] in 1972. In this investigation, the heat transfer coefficients for steam condensing on a rotating surface were measured. The condensation surface was a copper disk, 0.3 m in diameter, rotating in the horizontal plane. With rotational speeds ranging from 10 to 224 rpm, the heat flux varied from 2×10^4 to 1.9×10^5 W/m^2. The heat transfer data were correlated by dimensionless expressions and a theoretical expression similar to the one developed by Sparrow and Gregg[26] was derived. The experimental data were compared with the theoretical analysis and demonstrated good agreement.

The heat transfer in an impinging, condensing flow on a rotating disk was studied by Wang and Greif.[30] The problem considered impinging, steady, incompressible, laminar flow on an isothermal disk rotating at a constant angular velocity about its axis of symmetry. The effects of gravity and viscous dissipation were neglected and the fluid was assumed to have constant properties. The heat transfer results for the case without vapor shear were presented as

$$(h/\lambda)\left(v\Big/\sqrt{a^2+\omega^2}\right)^{\frac{1}{2}} = 1/C \tag{14}$$

$$C = \left(30E + 4E^2\Big/15\left(a\Big/\sqrt{a^2+\omega^2}\right)^2 + \left[1 - \left(a\Big/\sqrt{a^2+\omega^2}\right)^2\right]\right.$$
$$\left.\times\,(3+4[(15-4E)/(15+6E)]\{1+2[(15-4E)/(15+6E)]\})\right)^{\frac{1}{4}} \tag{15}$$

$$E = 6w/Pr(8+3w) \tag{16}$$

$$w = c_p\Delta T/h_{fg} \tag{17}$$

A study of the influence of the vapor shear was also conducted and concluded, as did Sparrow and Gregg,[27] that the vapor drag did not have a significant effect on the heat transfer in the case of condensation on a rotating horizontal flat disk.

Al-Baroudi and Klein[31] conducted an experimental investigation of the heat transfer characteristics of a rotating flat-plate radiator. Superheated steam was allowed to condense on the rotating flat surface, which was cooled to allow heat removal. Rotational speeds of 200, 300, and 400 rpm were used. The relationship between the overall heat transfer coefficient, the temperature difference between the working fluid and the cold environment, and the angular rotational speed of the plate were obtained. The empirical relationship developed is useful in selecting the optimal rotational speed for a rotating flat-plate radiator, given a desired heat rejection load.

III. Microfin Enhancement and Surface Tension

Surface tension plays a critical role and governs the size and shape of the interface in two-phase flows, and in reduced gravity environments it is the dominant parameter in determining the interfacial shape, and in so doing, strongly influences heat and mass transfer. In fact, the wall suction force described in the previous section is a manifestation of the surface tension. This effect becomes even more important in the absence of gravity, where buoyancy is strongly reduced and transport processes are determined by the properties at the interface alone.[32] In Georgia[33] it was recognized that surface tension on a curved surface could induce a pressure gradient many times larger than that induced by gravity. This discovery led to research on the possibility of utilizing the surface tension forces in thin condensate films to promote condensate removal in flow channels, thereby enhancing the condensation process. The role of surface tension in the condensate removal process has been studied by Masuda and Rose,[34] Marto,[35] and Yang and Chen.[36]

Due to the superior heat transfer performance, horizontal microfin tubes can be used effectively in the absence of gravity. Heat-transfer enhancement factors as high as 3.0 (relative to a smooth tube) have been reported as a result of this technique. This enhancement was considered to be the result of the combined effects of vapor shear and surface tension forces. Yang and Webb[37] proposed a semi-empirical model that considered the combined effects of vapor shear and surface tension forces for axially grooved microfin tubes. A stratified flow model for film condensation in helically grooved, horizontal microfin tubes was developed by Honda.[38] The height of the stratified condensate was estimated by extending the Taitel and Dukler[39] model for a smooth tube to a microfin tube. It was shown that the stratified flow model was applicable to a wide range of mass velocity and quality as long as the vapor to liquid density ratio was larger than 0.05.

A detailed analytical model was developed to investigate the condensation heat transfer phenomena occurring on microfinned surfaces with hydraulic diameters in the range of 5 to 10 μm, as shown in Fig. 7, by Li and Peterson.[40] The effects of vapor suction during the condensation process, the pressure difference between the inlet and the outlet, the capillary force induced by the curvature of the free liquid film, and the gravitational body force were all considered both in the groove and on the tip of the fin. When compared to a smooth flat surface of the same size, the condensation heat transfer coefficient of the microfinned surface was found to be significantly enhanced.

IV. Condensation in Microgrooves and Capillary Force

With the development of advanced machining methods, the use of microgrooves to enhance phase change and condensate flow has increased dramatically. The capillary force is the dominant factor in two-phase flow for flow in channels or tubes on the microscale, regardless of the magnitude of the gravitational body force. The capillary force developed at the meniscus in the microgroove corner along the tube can induce the working fluid to flow in the microgrooves, which is a manifestation of the surface tension effects. The geometric shape of a triangular microgroove is shown in Fig. 8. (see Ref. 41). The pressure difference at the liquid–vapor interface can be described by the Young–Laplace equation:

$$p_v - p_l \approx \sigma/r \tag{18}$$

Thus, the surface pressure of the liquid can be expressed as

$$F_{l,p} = -A_l d(p_v - \sigma/r) \tag{19}$$

Assuming that the vapor pressure and surface tension are constant, Eq. (19) can be simplified as

$$F_{l,p} = A_l \sigma d(1/r) \tag{20}$$

Micro heat pipes developed over the course of the past 15 yr are one of the devices that employ capillary flow resulting from small, sharp corner regions to induce flow. In these devices, the sharp corner regions serve as liquid arteries that induce flow along the length of the groves, and with localized heating can cause fluid to flow along the groove.

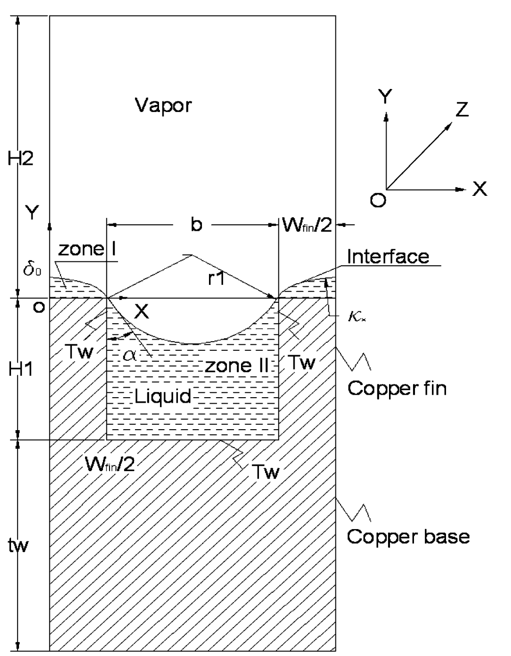


Fig. 7 Illustration of the liquid film profile in the cross section of the microchannel and coordinate system.[40]

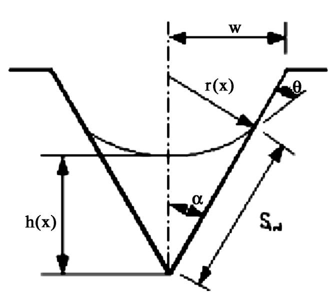


Fig. 8 The geometry shape of a triangular microgroove.[41]

The influence of the vapor flow on the capillary liquid flow in microgrooves was investigated by Ma et al.,[42] who found that as the vapor flowed over the condensate, the vapor flow directly affected the ability of the microgrooves to pump liquid toward or away from the condensing surface. This work also indicated that the interfacial forces, due to the interaction between the vapor and the liquid, must be considered in the analysis of the condensate flow, particularly in the case of counterflow such as in micro heat pipes, where the vapor flow can result in liquid being retained in the channels, thereby reducing the condensation heat transfer coefficient.

Khrustalev and Faghri[43] developed a detailed mathematical model to describe the influence of the geometry on the capillary force in grooves. The effects of interfacial thermal resistance, disjoining pressure, and surface roughness for a given meniscus contact angle were investigated.

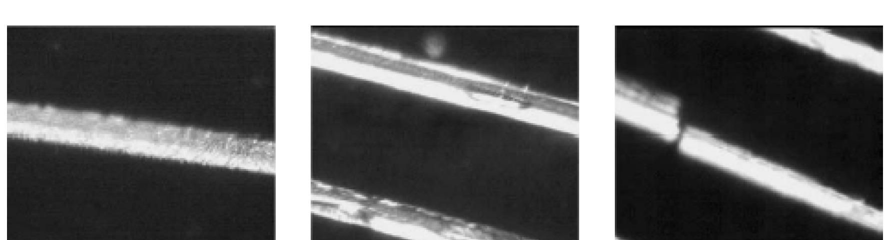
A detailed analysis of the capillary performance of triangular microgrooves was conducted by Sheu et al.[41] In this study, the curvature radius, cross-sectional area, and distribution of pressure and velocity of the working fluid in microgrooves were considered. The significance of the contact angle and hydraulic diameter in predicting the capillary performance of microgrooves was demonstrated by the proposed algebraic solutions.

Zhang et al.[44] presented a numerical investigation of the capillary blocking capability in forced convective condensation in horizontal minichannels. In this work, it was concluded that the film thickness and condensation length decreased as the hydraulic diameter or the distance between the parallel plates decreased. In addition, the condensation length was found to decrease significantly with a reduction in the total mass flow rate. The effect of various operational parameters on the condensation heat transfer coefficients was also investigated.

Techniques for determining the pumping power necessary for the removal of a predetermined quantity of heat using a two-phase flow system were presented and evaluated by Peterson.[45] Discussions of the flow regimes occurring in pumped two-phase fluids under microgravity were also presented. Six different concepts for achieving the required pumping power were presented and compared, based on various factors of performance.

Chen and Cheng[46] conducted visualization studies on dropwise condensation of steam in horizontal silicon microchannels. The microchannels had a trapezoidal cross section with a hydraulic diameter of 75 μm with saturated steam flowing through horizontal parallel microchannels, in which the walls were cooled by natural convection of air at room temperature. As shown in Fig. 9, stable droplet condensation was observed near the inlet of the microchannel. When the condensation process progressed along the microchannels, droplets accumulated on the wall and filled the microchannel. As the vapor core entrained and pushed the droplets, it developed into intermittent flow of vapor and condensate. It was predicted in this study that the droplet condensation heat flux of saturated steam at 225.5 kPa could reach values as high as 1,200 W/cm^2 at $\Delta T = 10°$C in a microchannel having a hydraulic diameter of 75 μm. This investigation also indicated that effective dropwise condensation can be obtained in microchannels in the absence of a body force.

One of the significant characteristics of condensation is the unsteady or inherent instabilities that occur, particularly when the length scale is small. Wu and Cheng[47] conducted simultaneous



a) Near the inlet **b) Near the midsection** **c) Near the outlet**

Fig. 9 Dropwise condensation in silicon microchannel.[46]

visualization and experimental measurements to investigate the condensation flow patterns for steam flowing through trapezoidal microchannels, having a hydraulic diameter of 82.8 μm. Different condensation flow patterns such as fully developed droplet flow, droplet/annular/injection/slug-bubbly flow, annular/injection/slug-bubbly flow, and fully slug-bubbly flow were observed in the microchannels. The wall temperature and the flow patterns were highly unstable and fluctuated significantly. The vapor injection flow and its induced condensation instabilities in microchannels were reported, and it was concluded the condensation instability must be considered for flow in microchannels having hydraulic diameters less than 100 μm at low mass fluxes.

Zhao and Liao[48] presented an analytical model for predicting film condensation of vapor flowing inside a mini triangular channel. In this investigation, three zones, thin liquid film flow on the sidewall, condensate flow in the corners, and vapor core flow in the center, all combined to compose the overall concurrent liquid–vapor two-phase flow field. The effects of interfacial shear stress and capillary force induced by the free liquid film curvature variation were studied. For the same inlet Reynolds number, the steam velocity was found to be higher in smaller channels, leading to a larger interfacial shear stress. The two-phase pressure drop increased with decreasing channel size and the heat transfer coefficients for triangular channels were always substantially higher than for round tubes having the same hydraulic diameter and the same inlet Reynolds number and inlet subcooling. The smaller the channel size, the higher the heat transfer coefficients in the entry region.

A multiple flow-regime model for pressure drop during the condensation of refrigerant R134a in horizontal round microchannels ranging in hydraulic diameter from 0.5 to 4.91 mm was presented by Garimella et al.[49] This model accurately predicted the condensation pressure drops in annular, disperse-wave, mist, discrete-wave, and intermittent-flow regimes.

Comparisons of the condensation in round minichannels ($D_h = 0.493$, 0.691, and 1.067 mm) and rectangular minichannels ($D_h = 0.494$, 0.658, and 0.972 mm) were experimentally presented by Shin and Kim.[50] It was reported that at lower mass fluxes, the heat transfer coefficients for the rectangular microchannels were higher than that obtained for round microchannels with similar hydraulic diameters. However, as the mass flux increased, the heat transfer coefficients of the round channels were larger than that of the rectangular channels and the local condensation Nusselt number of the round channels demonstrated a greater dependency on both the mass flux and the quality.

V. Conclusions

A review of the available literature pertaining to condensation in the absence of body forces similar to that experienced in microgravity environments is presented. The influence of the microgravity, shear stress, surface tension, and capillary and centrifugal forces on condensation is evaluated. The results of this review indicate that there is ample potential for the use of vapor shear to remove the condensate in space applications, as indicated in the literature. In addition, the removal of the liquid film with wall suction not only is possible, but presents a very viable solution for a wide variety of applications. Although the gravitational body force can be replaced by the centrifugal forces generated by a rotating system, the vapor shear at the vapor–liquid interface appears to have no significant influence on these rotating systems. As anticipated by numerous investigators, the surface tension and capillary forces play a significant role in two-phase systems for spacecraft thermal applications, and channel geometry can significantly affect the heat transfer and the resulting flow of condensation. These effects, however, are strongly size-dependent. For these situations, the heat transfer coefficients for triangular channels were always higher than for round tubes having the same hydraulic diameter, inlet Reynolds number, and inlet subcooling.

In actual applications, it is possible to utilize a combination of vapor shear, surface tension and/or capillary and centrifugal forces. For example, in a micro heat pipe, the fluid can be driven by capillary forces; the vapor shear stress will still play a significant role in the thin film flow. However, given the difficulty and expense of conducting extensive space-based experiments, experimental data are scarce, and the mechanisms of condensation in the absence of gravity are not adequately understood. Additional experimental and theoretical investigations are therefore necessary to better understand condensation in space applications and must be conducted before effective two-phase heat transport systems for thermal management of the microgravity environments can be validated.

Acknowledgments

The authors gratefully acknowledge the support provided by the NASA Glenn Research Center and a Key Project of the Chinese Ministry of Education No. 105082.

References

[1]Peterson, G. P., "Thermal Control Systems for Spacecraft Instrumentation," *Journal of Spacecraft and Rockets*, Vol. 24, No. 1, 1987, pp. 7–13.

[2]Nusselt, W., "Die Oberflachen Kondensation des Wasserdampfes," *Zeitsehrift Vereines Deutscher Ingeniuere*, Vol. 60, No. 2, 1916, pp. 541–569.

[3]Ghaderi, M., Salehi, M., and Saeedi, M. H., "Review of In-Tube Condensation Heat Transfer Correlations for Smooth and Enhanced Tubes," *Advances in Enhanced Heat/Mass and Energy Efficiency, Proceedings of the 1995 ASME Int'l. Mechanical Engineering Congress and Exposition*, HTD Vol. 320, San Francisco, 1995, pp. 83–94.

[4]Seban, R. A., and Hodgson, J. A., "Laminar Film Condensation in Tube with Upward Vapor Flow," *International Journal of Heat and Mass Transfer*, Vol. 25, No. 9, 1982, pp. 1291–1300.

[5]Kutateladze, S. S., *Fundamentals of Heat Transfer*, Academic Press, New York, 1963.

[6]Garcia-Valladares, O., "Review of In-Tube Condensation Heat Transfer Correlations for Smooth and Microfin Tubes," *Heat Transfer Engineering*, Vol. 24, No. 4, 2003, pp. 6–24.

[7]Rohsenow, W. M., Hartnett, J. P., and Ganic, E. N., *Handbook of Heat Transfer Fundamentals (II)*, 2nd ed., McGraw–Hill, New York, 1985, pp. 11–39.

[8]Rose, J. W., "Dropwise Condensation Theory and Experiment: A Review," *Proceedings of the I MECH E Part A, Journal of Power and Energy*, Vol. 216, No. 2, 2002, pp. 115–128.

[9]Chato, J. C., "Laminar Condensation inside Horizontal and Inclined Tubes," *ASHRAE Journal*, Vol. 4, No. 2, 1962, pp. 52–60.

[10]Chowdhury, A. H., and Peterson, G. P., "A Review of Advanced Radiation Technologies for Spacecraft Thermal Control," *Heat and Technology*, Vol. 11, Nos. 1–2, 1993, pp. 73–89.

[11]Parker, M. L., and Drolen, B. L., "Loop Heat Pipe for Spacecraft Thermal Control, Part 1: Vacuum Chamber Tests," *Journal of Thermophysics and Heat Transfer*, Vol. 18, No. 4, 2004, pp. 417–429.

[12]Cullimore, B., *SINDA/FLUINT, User Manual*, URL: http://www.crtech.com/docs/manuals/sf47main.pdf [cited 18 August 2005].

[13]Cullimore, B., and Bauman, J., "Steady-State and Transient Loop Heat Pipe Modeling," 30th International Conference on Environmental Systems, 2000-01-2316, Society of Automotive Engineers, Warrendale, PA, 2000.

[14]Liang, Q., Wang, X., and Narain, A., "Effects of Gravity, Shear and Surface Tension in Internal Condensing Flows: Results from Direct Computational Simulations," *Journal of Heat Transfer*, Vol. 126, No. 5, 2004, pp. 676–686.

[15]Soliman, H. M., and Azer, N. Z., "Flow Patterns During Condensation Inside a Horizontal Tube," *Journal*, Vol. 13, No. 1, 1971, pp. 57–58.

[16]Cess, R. D., "Laminar-Film Condensation on a Flat Plate in the Absence of a Body Force," *Zeitschrift für Angewandte Mathematik und Physik*, Vol. 11, No. 5, 1960, pp. 426–433.

[17]Koh, J. C. Y., "Film Condensation in a Forced-Convection Boundary-Layer Flow," *International Journal of Heat and Mass Transfer*, Vol. 5, No. 10, 1962, pp. 941–954.

[18]Shekriladze, I. G., and Gomelauri, V. I., "Theoretical Study of Laminar Film Comdemsation of Flowing Vapour," *International Journal of Heat and Mass Transfer*, Vol. 9, No. 6, 1966, pp. 581–591.

[19]Henstock, W. H., and Hanratty, T. J., "The Interfacial Drag and the Height of the Wall Layer in Annular Flows," *AIChE Journal*, Vol. 22, No. 6, 1976, pp. 990–1000.

[20]Chow, L. C., and Parish, R. C., "Condensation Heat Transfer in a Microgravity Environment," *Journal of Thermophysics and Heat Transfer*, Vol. 2, No. 1, 1988, pp. 82–84.

[21]Wang, W., Liu, Y., Zhang, L., and Zeng, X., "Study of Condensation Heat Transfer Inside a Tube in Microgravity," *Journal of Tsinghua University*, Vol. 37, No. 2, 1997, pp. 1–5.

[22]Chow, L. C., and Parish, R. C., "Condensation Heat Transfer in a Microgravity Environment," AIAA Paper 86-0068, Jan. 1986.

[23]Frankel, N. A., and Bankoff, S. G., "Laminar Film Condensation on a Porous Horizontal Tube with Uniform Suction Velocity," *Journal of Heat Transfer*, Vol. 87, No. 1, 1965, pp. 95–102.

[24]Faghri, A., and Chow, L. C., "Forced Condensation in a Tube with Suction at the Wall for Microgravitational Applications," *International Symposium on Thermal Problems in Space-Based Systems*, HTD-Vol. 33, Association of Mechanical Engineers, New York, 1987, pp. 71–76.

[25]Prenger, F. C., and Sullivan, J. A., "Conceptual Designs for 100-Megawatt Space Radiations," *Symposium on Advanced Compact Reactors Systems*, National Academy of Sciences, Washington, D.C., Nov. 1982.

[26]Sparrow, E. M., and Gregg, J. L., "A Theory of Rotating Condensation," *Journal of Heat Transfer*, Vol. 81, No. 2, 1959, pp. 113–120.

[27]Sparrow, E. M., and Gregg, J. L., "The Effect of Vapor Drag on Rotating Condensation," *Journal of Heat Transfer*, Vol. 82, No. 1, 1960, pp. 71–72.

[28]Sparrow, E. M., and Hartnett, J. P., "Condensation on a Rotating Cone," *Journal of Heat Transfer*, Vol. 83, No. 1, 1951, pp. 101–107.

[29]Butuzov, A. I., and Rifert, V. F., "An Experimental Study of Heat Transfer during Condensation of Steam at a Rotating Disk," *Heat Transfer—Soviet Research*, Vol. 4, No. 6, 1972, pp. 150–153.

[30]Wang, C. S., and Greif, R., "The Effect of Impingement on Heat Transfer in Rotating Condensation," *International Journal of Heat and Mass Transfer*, Vol. 24, No. 7, 1981, pp. 1097–1104.

[31]AI-Baroudi, H. M., and Klein, A. C., "Experimental Simulations and Heat Transfer Parameter Measurements of Film Condensation on a Rotating Flat Plate," *Experimental Thermal and Fluid Science*, Vol. 10, No. 1, 1995, pp. 124–135.

[32]Straub, J., "Role of Surface Tension for Two-Phase Heat and Mass Transfer in the Absence of Gravity," *Experimental Thermal and Fluid Science*, Vol. 9, No. 3, 1994, pp. 253–273.

[33]Georgia, R., "Film Condensation on Finely Rippled Surfaces with Consideration of Surface Tension," *Zeitschrift für Angewandte Mathematik und Physik*, Vol. 5, No. 1, 1954, pp. 36–49.

[34]Masuda, H., and Rose, J. W., "Static Configuration of Liquid Films on Horizontal Tubes with Low Radial Fins: Implication for Condensation Heat Transfer," *Proceedings of the Royal Society of London Series A*, Vol. 410, No. 1838, 1987, pp. 125–139.

[35]Marto, P. J., "An Evaluation of Film Condensation on Horizontal Integral Fin Tubes," *Journal of Heat Transfer*, Vol. 110, No. 4(B), 1988, pp. 1287–1305.

[36]Yang, S. A., and Chen, C. K., "Role of Surface Tension and Ellipticity in Laminar Film Condensation on a Horizontal Elliptical Tube," *International Journal of Heat and Mass Transfer*, Vol. 30, No. 12, 1993, pp. 3135–3141.

[37]Yang, C. Y., and Webb, R. L., "A Predictive Model for Condensation in Small Hydraulic Diameter Tubes Having Axial Microfins," *Journal of Heat Transfer*, Vol. 119, No. 4, 1997, pp. 776–782.

[38]Honda, H., Wang, H., and Nozu, S., "A Theoretical Study of Film Condensation in Horizontal Microfin Tubes," *Journal of Heat Transfer*, Vol. 124, No. 1, 2002, pp. 94–101.

[39]Taitel, Y., and Dukler, A. E., "A Model for Predicting Flow Regime Transitions in Horizontal and Near Horizontal Gas-Liquid Flow," *AIChE Journal*, Vol. 22, No. 1, 1976, pp. 47–55.

[40]Li, C., and Peterson, G. P., "Capillary Condensation on Micro-Finned Surfaces," AIAA Paper 2004-162, Jan. 2004.

[41]Sheu, T. S., Kuo, J. S., Ding, P. P., and Chen, P. H., "Theoretical Analysis of Capillary Performance of Triangular Microgrooves," *Microsystem Technologies*, Vol. 10, No. 4, 2004, pp. 315–322.

[42]Ma, H. B., Peterson, G. P., and Lu, X. J., "The Influence of Vapor–Liquid Interactions on the Liquid Pressure Drop in Triangular Microgrooves," *International Journal of Heat Mass Transfer*, Vol. 37, No. 15, 1994, pp. 2211–2219.

[43]Khrustalev, D., and Faghri, A., "Heat Transfer During Evaporation on Capillary-Grooved Structures of Heat Pipes," *Journal of Heat Transfer*, Vol. 117, No. 3, 1995, pp. 740–747.

[44]Zhang, Y., Faghri, A., and Shafii, M. B., "Capillary Blocking in Forced Convective Condensation in Horizontal Miniature Channels," *Journal of Heat Transfer*, Vol. 123, No. 3, 2001, pp. 501–511.

[45]Peterson, G. P., "Thermal Control Systems for Spacecraft Instrumentation," *Journal of Spacecraft*, Vol. 24, No. 1, 1987, pp. 7–13.

[46]Chen, Y. P., and Cheng, P., "Condensation of Steam in Silicon Microchannels," *International Communications in Heat and Mass Transfer*, Vol. 32, No. 1–2, 2005, pp. 175–183.

[47]Wu, H. Y., and Cheng, P., "Condensation Flow Patterns in Silicon Microchannels," *International Journal of Heat and Mass Transfer*, Vol. 48, No. 11, 2005, pp. 2186–2197.

[48]Zhao, T. S., and Liao, Q., "Theoretical Analysis of Film Condensation Heat Transfer Inside Vertical Mini Triangular Channels," *International Journal of Heat and Mass Transfer*, Vol. 45, No. 13, 2002, pp. 2829–2842.

[49]Garimella, S., Agarwal, A. and Killion, J. D., "Condensation Pressure Drop in Circular Microchannels," *Heat Transfer Engineering*, Vol. 26, No. 3, 2005, pp. 28–35.

[50]Shin, J. S., and Kim, M. H., "An Experimental Study of Flow Condensation Heat Transfer Inside Circular and Rectangular Mini-Channels," *Proceedings of the Second International Conference on Microchannels and Minichannels (ICMM2004)*, Association of Mechanical Engineers, New York, 2004, pp. 633–640.