Technical Notes

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Condensation Heat Transfer in a Microgravity Environment

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Nomenclature

c_p = specific heat
h_{fg} = latent heat of vaporization
Ja = Jakob number, $c_p(T_s - T_w)/h_{fg}$
k = thermal conductivity
$\dot{m}_c''$ = condensate mass flux
p = pressure
Pr_L = Prandtl number, $\mu_L c_p / k_L$
T = temperature
ΔT = $T_s - T_w$
u, v = velocity components along z, y
y = coordinate normal to wall
z = axial coordinate
δ = film thickness
ρ = density
τ = shear stress
μ = dynamic viscosity

Subscripts

L = liquid
s = saturation
v = vapor
w = wall
δ = at liquid/vapor interface

Introduction

CONDENSATION is the change of phase from vapor to liquid and is accompanied by heat transfer because the latent heat of condensation must be removed. In film condensation, the liquid condensate forms a continuous film that covers the condenser surface. The film usually constitutes the dominant resistance to heat transfer. If the condensate is not effectively removed, the condenser may flood, halting further condensation of the vapor. In an earth-based condenser design, the drainage of the condensate is typically provided by the gravitational force. In the absence of gravity, the condensate can be removed by suction, surface rotation, shear stress due to vapor flow, capillary pumping, or a mechanical wiper.

A review of pertinent literature revealed that data from two-phase heat transfer experiments with reduced gravity forces is very limited. A summary of related work is given in Ref. 1. The purpose of the present effort is to assess the potential of using vapor shear to remove the condensate for space applications. Effective space condensers should be capable of providing high heat fluxes with a small temperature difference. For example, for heat rejection in space, such a condenser will allow a maximum radiator temperature, resulting in lower weights for both the radiator and condenser.

Convective Condensation Inside a Tube

Consider a vapor at a temperature of T_s following inside a tube with a velocity of u_v. The tube wall is kept at a temperature of T_w, ($T_w < T_s$). At high mass flows or low liquid fractions, the flow regime is very likely to be annular. In the absence of gravity, the motion of the condensate film is entirely due to vapor shear and momentum transfer resulting from condensation. The motion of the condensate film is solved while ignoring the velocity distribution of the vapor. A friction coefficient based on the average vapor velocity is specified at the vapor/liquid interface. The approach is similar to the one used in Ref. 2.

Assuming the film thickness δ is small compared to the tube diameter d, and ignoring the inertia of the liquid, a momentum balance of a fluid element inside the condensate film is

$$\mu_L \frac{\partial^2 u_L}{\partial y^2} = \frac{\mathrm{d}p}{\mathrm{d}z} \tag{1}$$

The boundary conditions are $u_L = 0$ at $y = 0$ and $\alpha u_L / \alpha y = \tau_\delta / \mu_L$ at $y = \delta$.

The pressure will decrease with z due to friction at the wall. There is also a small pressure recovery due to the decreasing vapor velocity caused by condensation. However, the pressure changes are expected to be very small compared to the absolute pressure. Thus, the vapor temperature, vapor density, and vapor viscosity can be approximated to be constant.

The velocity u_L and the velocity at the interface $u_{L\delta}$ are $\tau_\delta y / \mu_L$ and $\tau_\delta \delta / \mu_L$, respectively. The liquid Reynolds number Re_L is given by

$$Re_L = \frac{\rho_L u_{L,\mathrm{av}} \delta}{\mu_L} = \frac{\rho_L \tau_\delta \delta^2}{2\mu_L^2} \tag{2}$$

where $u_{L,\mathrm{ave}}$ is the average velocity of the condensate.

The mass flow of the condensate increases with z due to condensation, and the rate of increase of Re_L is given by

$$\mu_L \frac{\mathrm{d}Re_L}{\mathrm{d}z} = \dot{m}_c'' \tag{3}$$

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The rate of condensation is determined by the heat transfer rate as the latent heat must be removed,

$$\frac{dRe_L}{dz} = \frac{k_L(T_s - T_w)}{\mu_L h_{fg} \delta} = \frac{Ja}{Pr_L \delta} \tag{4}$$

As the vapor condenses, the vapor mass flow rate decreases. Assuming the quality is unity at the inlet, then

$$\dot{m}_v + \dot{m}_L = \dot{m}_{vo} \tag{5}$$

where $\dot{m}_v$ and $\dot{m}_L$ are the vapor and liquid mass flow rates at any position z and $\dot{m}_{vo}$ is the vapor mass flow rate at the inlet. The vapor Reynolds number is

$$Re_v = \frac{\rho_v u_v d}{\mu_v} = Re_{vo}\left(1 - \frac{4Re_L \mu_R}{Re_{vo}}\right) \tag{6}$$

where Re_{vo} is the inlet vapor Reynolds number ($=\rho_v u_{vo} d/\mu_v$) and μ_R is the ratio of the liquid and vapor viscosities ($=\mu_L/\mu_v$). Since the vapor density and viscosity are assumed to be constant,

$$u_v = u_{vo} Re_v / Re_{vo} \tag{7}$$

As mentioned previously, the motion of the condensate is driven by the shear stress at the interface,

$$\tau_\delta = (c_f/2)_E \rho_v (u_v - u_{L\delta})^2 + \dot{m}_c''(u_v - u_{L\delta}) \tag{8}$$

where the first term in Eq. (8) is due to the shear stress between the vapor and the liquid and the second term is due to the faster-moving vapor condensing onto the slower-moving liquid.

For laminar vapor flow ($Re_v < 2{,}300$), $(c_f/2)_E$ is given by

$$(c_f/2)_E = 8/Re_v \tag{9}$$

For turbulent vapor flow ($Re_v > 2{,}300$), $(c_f/2)_E$ is given by[3]

$$(c_f/2)_E = (c_f/2)(1 + 850F) \tag{10}$$

where

$$(c_f/2) = 0.023\, Re_v^{-0.2}, \qquad F = \gamma \mu_R / (\rho_R^{0.5} Re_v^{0.9})$$

$$\rho_R = \rho_L/\rho_v, \qquad \gamma = [(1.414\, Re_L^{0.5})^{2.5} + (0.132\, Re_L^{0.9})^{2.5}]^{0.4}$$

Nondimensionalizing distances and velocities by the tube diameter d and the inlet vapor velocity u_{vo} respectively,

$$\bar{u}_v = u_v/u_{vo}, \quad \bar{u}_L = u_L/u_{vo}, \quad \bar{z} = z/d, \quad \bar{\delta} = \delta/d \tag{11}$$

and defining a nondimensional shear stress M, one can rewrite $u_{L\delta}$ and Eqs. (2), (4), (7), and (8) as

$$\bar{u}_{L\delta} = M\bar{\delta} \tag{12}$$

$$Re_L = \frac{M \rho_R Re_{vo} \bar{\delta}^2}{2\mu_R} \tag{13}$$

$$\frac{dRe_L}{d\bar{z}} = \frac{Ja}{Pr_L \bar{\delta}} \tag{14}$$

$$\bar{u}_v = Re_v / Re_{vo} \tag{15}$$

$$M = \tau_\delta d/(\mu_L u_{vo})$$

$$= \left(\frac{c_f}{2}\right)_E \frac{Re_{vo}}{\mu_R}(\bar{u}_v - \bar{u}_{L\delta})^2 + \frac{Ja}{Pr_L \bar{\delta}}(\bar{u}_v - \bar{u}_{L\delta}) \tag{16}$$

These equations can be solved together with Eqs. (6) and (9), or (10) to yield $\bar{\delta}$, Re_L, Re_v, $\bar{u}_v$, $\bar{u}_{L\delta}$, and M as functions of $\bar{z}$ with Re_{vo}, Ja/Pr_L, ρ_R, and μ_R as parameters, Results are presented only for the case of saturated steam condensing at one atmosphere ($T_s = 100°C$). The wall temperature is assumed to be at either 85 or 70°C. The liquid properties are evaluated at the average film temperature $(T_s + T_w)/2$. The values of the parameters are listed in Table 1.

The film thickness and the vapor Reynolds number for $Re_{vo} = 5000$ are plotted as functions of $\bar{z}$ in Fig. 1. The vapor Reynolds number gives an indication of the amount of vapor that remains at a given $\bar{z}$. For $\Delta T = 15°C$, about 50% of the vapor remains at $\bar{z} = 8$, while it only takes $\bar{z} = 3$ to condense 50% of the vapor for $\Delta T = 30°C$. As shown in Fig. 1, the film thickness is indeed very small, with $\bar{\delta} < 0.01$.

Similar results for $Re_{vo} = 50{,}000$ are plotted in Fig. 2. However, it takes up to $\bar{z} = 50$ to condense 50% of the vapor with a ΔT of 15°C. The corresponding condensing length for a ΔT of 30°C is approximately 20.

For $Re_{vo} = 50{,}000$, the vapor velocity $\bar{u}_v$ and the liquid velocity at the interface $\bar{u}_{L\delta}$ are plotted in Fig. 3. The vapor shear can induce a liquid velocity that is about 10 to 20 times smaller than the vapor velocity.

Table 1 Parameters chosen for convective condensation

	$\Delta T = 15°C$	$\Delta T = 30°C$
μ_R	24.92	25.98
ρ_R	1614	1620
Ja	0.02797	0.0559
Pr_L	1.91	1.99

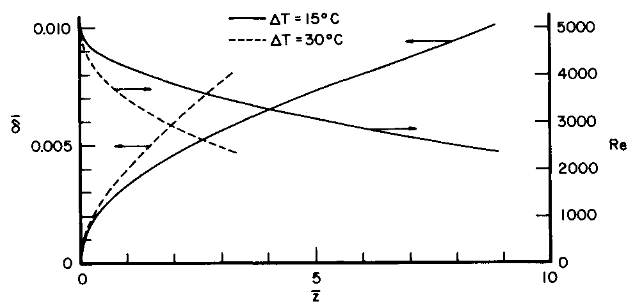

Fig. 1 Film thickness and vapor Reynolds number along the tube for Re_{vo} = 5000.

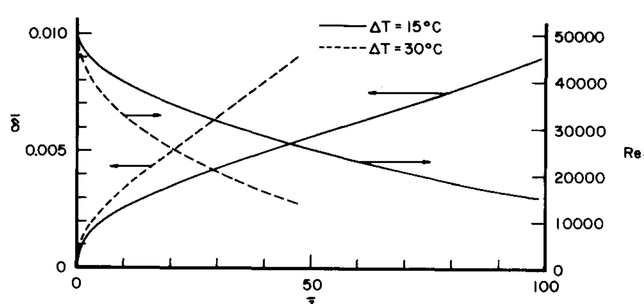

Fig. 2 Film thickness and vapor Reynolds number along the tube for Re_{vo} = 50,000.

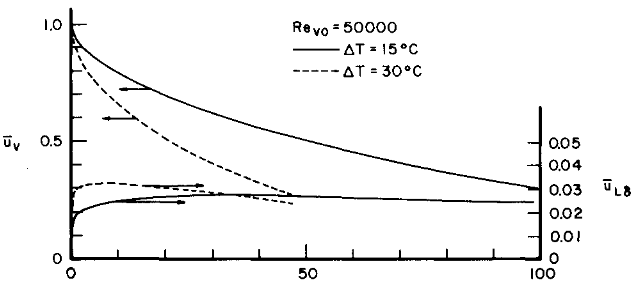

Fig. 3 Vapor velocity and liquid velocity at interface along the tube.

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Fluid Loss from a Puncture of a Space Radiator

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Introduction

FUTURE space missions will require the development of lightweight, large surface area radiators capable of rejecting large quantities of waste heat. A lightweight expandable bellow radiator has been suggested for this purpose.[1] Unlike radiators with rigid structures, this radiator is only exposed to the space environment when it is in operation. Even so, the susceptibility of the radiator to damage from micrometeoroids must be evaluated. These tiny particles travel through space at an average velocity of approximately 20 km/s. Since the radiator is lightweight and flexible, it is likely that punctures will occur. No previous research has been conducted to determine the mass loss of the radiator working fluid associated with the puncture of a bellow in a space environment. The present experiment was designed to simulate this situation so that the extent of potential damage and the effects on radiator performance could be evaluated.

Experimental Apparatus and Procedures

The experimental apparatus was designed to simulate puncture conditions for a flexible radiator. The test section consisted of a 3800-cc, double-walled, stainless-steel vessel with a membrane holder on the bottom. The test section was located in an evacuated bell jar. Either superheated steam (superheated steam case) or saturated steam with a condensate layer covering the membrane (condensate case) occupied the test section. The other side of the membrane was exposed to the near-vacuum environment of the bell jar. The temperature in the test section was controlled by circulating constant temperature water through the test section double wall. The membrane was punctured using a solenoid activated needle. The pressure and temperature inside the test section were monitored as a function of time after puncture. For a more detailed experimental description, see Ref. 2.

The mass loss rate for the condensate case is calculated by estimating the time it took for the entire condensate layer to leak out. Data required to compute the mass loss rates in this manner are given in Table 1, where T_{sat} is the saturation temperature, V_{add} is the volume of liquid added to the test section, and t is the estimated time it took to expel the entire layer of condensate. It is assumed that the temperature within the test section is uniform at saturation. The proper amount of water was added using a calibrated syringe. The high uncertainty in time is due to the difficulty in observing when all the condensate was expelled.

The mass loss rate for the superheated steam case is calculated by estimating the amount of mass in the test section just prior to puncture and at the end of the run. The difference is then divided by the time of the run. Since the volume of the test section is constant, the mass at each time can be calculated if the temperatures and pressures are known. Data collected for this case are given in Table 2. T_1 and P_1 are the temperature and pressure just prior to puncture. T_2 and P_2 are the temperature and pressure recorded t s after puncture. The time t for this case was determined by the pressure rise in the bell jar, which was read from the pressure thermocouple gage in the vacuum test machine. The time is measured using the strip chart recorder. As soon as the bell jar pressure rises above 1.0 mm Hg, the test is ended. If the test were run longer, the mass-loss rate would begin to decrease significantly due to the pressure drop in the test section.

Mass Loss Rate Determination

Condensate Case

The mass loss rate for the condensate case can be estimated from the experimental data given in Table 1. Since the volume of the test section is constant, an estimate of the mass of the condensate layer can be obtained by the following method. The volume of the test section is divided by the mass of the liquid added to obtain a total specific volume. This value, along with the specific volumes of the vapor and liquid corresponding to the recorded saturation temperature, is used to calculate the quality of the mixture and mass of the condensate layer. The mass loss rate $\dot{m}$ is obtained by simply dividing condensate mass by the experimentally observed time for the entire layer to be expelled. The resulting mass loss rates for this case are presented in Fig. 1.

The mass loss rate for the condensate case may also be predicted theoretically by using the orifice flow equation.[3] However, this prediction is not quite representative of the experiment due to the irregular puncture areas and irregular discharge observed experimentally. This prediction should only give an order of magnitude value. The mass loss rates calculated from the orifice velocity predictions and the measured puncture areas are approximately 2.5 to 3.0 times less than the experimentally determined values for all runs. The difference is attributed to the fact that the momentum of the liquid flow may have increased the hole area. The puncture areas discussed in the following section were measured after the experiment and with no liquid flowing through the holes.

Superheated Steam Case

To calculate the mass loss rate for the superheated steam case, the values for mass contained in the test section at the beginning and end of the run were computed using the ideal gas law. Superheated steam in the present pressure and temperature range behaves like an ideal gas, that is, the compressibility factor Z is approximately 1.0. The difference in these two values was divided by the time of the run to give the mass loss rate. The mass loss rates calculated in this manner from the data in Table 2 are plotted against the puncture areas. The slope is the average mass flux. This is shown in Fig. 2.

The mass flux for this case can be predicted theoretically using the choked flow equation.[4] The average experimental

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