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THE EFFECT OF IMPINGEMENT ON HEAT TRANSFER IN ROTATING CONDENSATION

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Abstract—A study has been made of the heat transfer for an impinging, condensing flow on a rotating disk. Results were obtained based on integral methods. Although the effect of impingement is to increase the heat transfer, the magnitude is insufficient to account for the larger experimental results reported in the literature.

NOMENCLATURE

a, impinging intensity;
B, dimensionless quantity, equation (28);
C, constant, equations (14) and (15);
C_1, flow constant, equation (19c);
c_p, specific heat;
E, dimensionless quantity, equation (14);
F, dimensionless velocity, equations (14) and (17);
G, dimensionless velocity, equations (14) and (16);
h, heat transfer coefficient, equation (30);
h_{dim}, dimensionless heat transfer coefficient, $h/k\,[\nu/(a^2+\omega^2)^{1/2}]^{1/2}[c_p\Delta T/Pr\,h_{fg}]^{1/4}$;
h_{fg}, heat of evaporation;
H, dimensionless velocity, $C_1/\sqrt{(\nu_V\lambda)}$;
k, thermal conductivity;
K, dimensionless quantity, equation (28);
P, dimensionless quantity, equation (28);
p, pressure;
Pr, Prandtl number;
q, heat flux;
Q, dimensionless quantity, equation (28);
r, radial coordinate;
T, temperature;
ΔT, temperature difference, $T_{\text{sat}} - T_w$;
V, velocity;
W, dimensionless quantity, equation (14);
z, coordinate normal to the disk surface.

Greek symbols

δ, condensate film thickness;
Δ, vapor layer thickness;
η, $= z - \delta$;
θ, tangential coordinate;
λ, $= [a^2 + \omega^2]^{1/2}$;
μ, viscosity;
ν, kinematic viscosity;
ρ, density;
τ, shear stress;
ω, angular velocity.

Subscripts

L, liquid layer;
sat, saturation value;
V, vapor layer;
w, surface of the disk.

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INTRODUCTION

FILM condensation on a rotating disk has been the subject of many investigations. In 1959, Sparrow and Gregg [1] made an analysis for film condensation on a rotating disk in an infinite medium of pure saturated vapor. An exact numerical solution of the Navier–Stokes and energy equations was obtained and results were given for the heat transfer, the condensate layer thickness and the temperature and velocity profiles in terms of the Prandtl number and the ratio $(c_p\Delta T)/h_{fg}$. Buoyancy viscous dissipation and the shear stress at the liquid–vapor interface were neglected.

In 1960, Nandapurkar and Beatty [2] performed an experiment with a rotating, horizontal, water-cooled disk which was used as the condensing surface for pure vapors of different substances. They measured the temperature at different locations on the surface and then calculated the heat transfer coefficient based on the measured temperature. Their results confirmed that the heat transfer was essentially constant over the surface. Their measured heat transfer coefficients were 25–30% less than those predicted theoretically. They attributed the discrepancy between the measured values and the predicted results to neglecting the effect of vapor drag on the condensate in the theoretical analysis.

Sparrow and Gregg [3] reconsidered the problem of laminar film condensation on a cooled rotating disk. They extended their previous work [1], where the assumption of zero shear at the interface had been involved, and now included interfacial shear. They solved the flow problems in both the liquid and vapor simultaneously. The analysis was carried out for condensation of pure, saturated vapor on an isothermal disk. The results showed that for Prandtl numbers greater than one, the vapor drag at the liquid–vapor interface did not have a significant effect on the heat transfer and Sparrow and Gregg [3] therefore concluded that the 30% deviation between theory and experiment must be laid to other causes.

In 1968, Espig and Hoyle [4] studied the heat transfer from condensing steam on a cooled vertical flat rotating disk. Temperatures were measured at several locations on the surface and also in the interior of the disk. The heat flux was determined from these temperature measurements. A theoretical analysis was made by considering film condensation on the surface of an infinite rotating disk in a large medium of pure dry and saturated vapor. The film was assumed to be in axially symmetrical, steady, incompressible, laminar flow with no shear forces between the condensate film and the vapor. Integral methods were used to solve the momentum and energy equations in the liquid. Parabolic velocity distributions and a linear temperature profile were assumed. The theoretical results were in good agreement with those of Sparrow and Gregg [1]. The experimental results showed that the radial variation of the temperature was very small. The measured heat transfer coefficients of Espig and Hoyle [4] varied from 80–170% of the theoretical values. They proposed that the discrepancy between the predicted results for the heat transfer and the measured data was due to the presence of waves on the condensate film [4, 5].

In 1970, Butuzov, Pukhovoy and Rifert [6] constructed experimental equipment for the study of heat transfer attendant to sea water desalination in a centrifugal evaporator. The steam was supplied and condensed on the lower part of the disk. The vapor pressure, condensate temperature and disk wall temperatures at seven different locations were measured. The experimental results were presented in a subsequent paper in 1972 [7] and were stated to be in good agreement with theoretical results.

Interest in rotating evaporators has continued in connection with the use of wiper blades on the evaporation side of the disk. This results in higher performance of the evaporator and overall heat transfer coefficients as high as 8600 Btu/h ft^2 °F (48,800 W/m^2 K) have been reported by Tleimat [8]. Related measurements have also been carried out by Wang, Greif and Laird [9]. There is a clear, sustained interest in rotating flows with condensation. One effect that has not been investigated is the importance of an impinging flow on rotating condensation. Accordingly, this effect has been studied in the present work and results have been obtained based on integral methods.

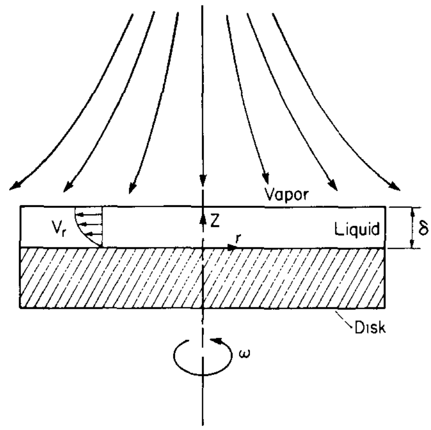

FIG. 1. Schematic of system with a liquid boundary layer.

ANALYSIS

The problem under consideration is that of an impinging steady, incompressible, laminar flow condensing on an isothermal disk rotating at constant angular speed about its symmetrical axis (cf. Fig. 1). The effects of gravity and viscous dissipation are neglected and the fluid is assumed to have constant properties.

Case I: zero shear stress at the liquid–vapor interface

The conservation equations of mass, momentum and energy are written in the thin liquid layer using the boundary layer approximations. These are given by

$$\frac{\partial V_r}{\partial r} + \frac{V_r}{r} + \frac{\partial V_z}{\partial z} = 0 \tag{1}$$

$$\rho\left(V_r \frac{\partial V_r}{\partial r} - \frac{V_\theta^2}{r} + V_z \frac{\partial V_z}{\partial z}\right) = -\frac{\partial p}{\partial r} + \mu \frac{\partial^2 V_r}{\partial z^2} \tag{2}$$

$$\rho\left(V_r \frac{\partial V_\theta}{\partial r} + \frac{V_r V_\theta}{r} + V_z \frac{\partial V_\theta}{\partial z}\right) = \mu \frac{\partial^2 V_\theta}{\partial z^2} \tag{3}$$

$$\rho c_p V_z \frac{\partial T}{\partial z} = k \frac{\partial^2 T}{\partial z^2}. \tag{4}$$

The no-slip boundary condition is assumed on the surface of the isothermal disk, $z = 0$. At the edge of the condensate layer, $z = \delta$, the shear stress is neglected and the saturation temperature is assumed. These boundary conditions are given by (cf. Sparrow and Gregg [1]).

$$\left.\begin{aligned} V_r &= 0 \\ V_\theta &= r\omega \\ V_z &= 0 \\ T &= T_w \end{aligned}\right\} z = 0 \qquad \left.\begin{aligned} \tau_{zr} &= 0 \\ \tau_{z\theta} &= 0 \\ T &= T_{sat} \end{aligned}\right\} z = \delta. \tag{5}$$

The remaining specification is that of the pressure gradient $\partial p/\partial r$. This is taken to be constant across the boundary layer and is determined by evaluating $\partial p/\partial r$

outside the condensate layer assuming an external vapor potential flow. The result is

$$-\frac{\partial p}{\partial r} = \rho a^2 r \quad (6)$$

where a is the impinging intensity [11, 12].

To relate the film thickness to known physical quantities, an overall energy balance is made over the disk extending from $r = 0$ to $r = r$ for a film thickness δ. The film thickness is taken to be a constant but unknown quantity. The result is given by [1]

$$2\pi \int_0^\delta h_{fg}\rho r V_r \, dz + 2\pi \int_0^\delta \rho c_p (T_{sat} - T) r V_r \, dz = k \left.\frac{\partial T}{\partial z}\right|_0 \pi r^2 \quad (7)$$

where the left-hand side is the heat transferred from the condensate to the surface. The first term on the left is the energy released as latent heat and the second term is the energy liberated by subcooling of the condensate. Conduction across the vapor–liquid interface has been neglected.

To obtain theoretical results for the heat transfer, integral forms of the conservation equations are used and these are given by

$$\nu \left.\frac{\partial V_r}{\partial z}\right|_\delta - \nu \left.\frac{\partial V_r}{\partial z}\right|_0 = -a^2 r\delta - \frac{1}{r}\int_0^\delta V_\theta^2 \, dz - \frac{V_{r,\delta}}{r}\frac{\partial}{\partial r}\left(r\int_0^\delta V_r \, dz\right) + \frac{1}{r}\frac{\partial}{\partial r}\left(r\int_0^\delta V_r^2 \, dz\right) \quad (8)$$

$$\nu \left.\frac{\partial V_\theta}{\partial z}\right|_\delta - \nu \left.\frac{\partial V_\theta}{\partial z}\right|_0 = \frac{\partial}{\partial r}\int_0^\delta V_r V_\theta \, dz + \frac{2}{r}\int_0^\delta V_r V_\theta \, dz - \frac{V_{\theta,\delta}}{r}\frac{\partial}{\partial r}\left(r\int_0^\delta V_r \, dz\right). \quad (9)$$

Equation (7) is the integral form of the energy equation.

To solve these equations, the following velocity and temperature distributions are assumed

$$V_r = V_{r,\delta}[2(z/\delta) - (z/\delta)^2] \quad (10a)$$

$$V_\theta = r\omega - (r\omega - V_{\theta,\delta})[2(z/\delta) - (z/\delta)^2] \quad (10b)$$

$$\frac{T - T_{sat}}{T_w - T_{sat}} = 1 - (z/\delta). \quad (10c).$$

Equations (10a), (10b) and (10c) approximate the physical profiles and satisfy the boundary conditions. Substituting these relations into the three integral equations, (7), (8) and (9) yields, respectively

$$\frac{30\nu}{\delta^2}\frac{V_{r,\delta}}{r} = 15a^2 + 3\omega^2 + 4\omega\frac{V_{\theta,\delta}}{r} + \frac{8V_{\theta,\delta}^2}{r^2} + \frac{2V_{r,\delta}^2}{r^2} - \frac{6V_{r,\delta}}{r}\left.\frac{\partial V_r}{\partial r}\right|_\delta \quad (11)$$

$$\frac{15\nu}{\delta^2}\left(\omega - \frac{V_{\theta,\delta}}{r}\right) = 3\omega\frac{V_{r,\delta}}{r} + \frac{3V_{r,\delta}V_{\theta,\delta}}{r^2} + \left(\omega - \frac{V_{\theta,\delta}}{r}\right)\left.\frac{\partial V_r}{\partial r}\right|_\delta + \frac{4V_{r,\delta}}{r}\left.\frac{\partial V_\theta}{\partial r}\right|_\delta \quad (12)$$

$$\frac{6c_p\Delta T}{h_{fg}} = \left[8 + 3\left(\frac{c_p\Delta T}{h_{fg}}\right)\right]\frac{Pr\delta^2}{\nu}\frac{V_{r,\delta}}{r}. \quad (13)$$

These equations are satisfied for velocities varying linearly with the radius. In terms of the following variables

$$W = c_p\Delta T/h_{fg}, \quad \lambda = \sqrt{(a^2 + \omega^2)}, \quad C = \delta\sqrt{(\lambda/\nu)} \quad (14)$$

$$F = V_{r,\delta}/r\lambda, \quad G = V_{\theta,\delta}/r\omega, \quad E = \frac{6W}{Pr(8 + 3W)}$$

the results are given by

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

$$G = \frac{15 - 4E}{15 + 6E} \quad (16)$$

$$F = \frac{E}{C^2}. \quad (17)$$

With these relations for C (or δ), G (or $V_{\theta,\delta}$) and F (or $V_{r,\delta}$), the results for the velocity and temperature distributions may be obtained from equations (10a), (10b) and (10c).

Case II: non-zero shear stress at the liquid–vapor interface

For this case it is assumed that a vapor boundary layer exists outside the condensate layer (Fig. 2). The interfacial shear stress at the liquid–vapor interface is no longer taken to be zero.

The conservation equations (1), (2), (3) and (4) are

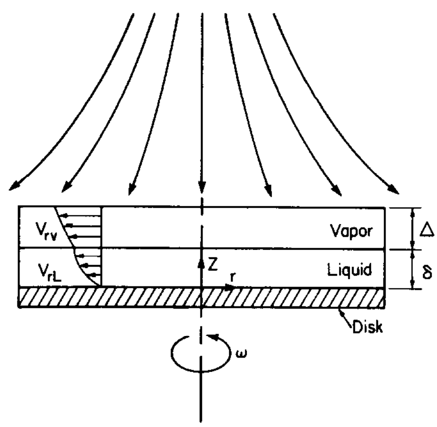

FIG. 2. Schematic of system with liquid and vapor boundary layers.

applied to the liquid, $0 < z < \delta$, and the vapor, $\delta < z < \delta + \Delta$, layers. The properties of the liquid and the vapor are constant. At the liquid–vapor interface, $z = \delta$, the boundary conditions are [3]

conservation of mass

$$\rho_L V_{zL,\delta} = \rho_V V_{zV,\delta}$$

equality of tangential velocity

$$V_{\theta L,\delta} = V_{\theta V,\delta} \equiv V_{\theta,\delta}$$

equality of radial velocity (18)

$$V_{rL,\delta} = V_{rV,\delta} \equiv V_{r,\delta}$$

continuity of tangential shear

$$\tau_{z\theta L,\delta} = \tau_{z\theta V,\delta} \equiv \tau_{z\theta,\delta}$$

continuity of radial shear

$$\tau_{zrL,\delta} = \tau_{zrV,\delta} \equiv \tau_{zr,\delta}$$

where the subscripts L and V denote liquid and vapor, respectively.

At the surface of the disk

$$V_r = V_z = 0, \quad V_\theta = r\omega \quad \text{at} \quad z = 0 \qquad (19a)$$

and at the edge of the vapor boundary layer

$$\tau_{zr} = \tau_{z\theta} = 0 \quad \text{at} \quad z = \delta + \Delta. \qquad (19b)$$

The liquid and vapor boundary layer thicknesses are assumed to be constant.

The temperature of the isothermal disk is T_w and the temperature at the liquid–vapor interface is T_{sat}. The vapor temperature is considered to be constant at its saturation value, T_{sat}.

The vapor outside the boundary layer, $z > \delta + \Delta$, is assumed to be in potential flow with

$$V_r = ar \quad \text{and} \quad V_z = -2az + C_1 \qquad (19c)$$

where C_1 is a flow constant which depends on the parameter $a/\sqrt{(a^2 + \omega^2)} = a/\lambda$ [11]. The radial pressure gradient, is assumed to be invariant in the direction normal to the disk and is given by $-\partial P/\partial r = \rho a^2 r$.

Proceeding in a manner similar to that previously discussed, integral forms of the conservation equations were used in the liquid and vapor boundary layers with the following velocity and temperature distributions

liquid region

$$V_{rL} = \left(\frac{\delta\tau_{zr,\delta}}{\mu_L} - V_{r,\delta}\right)\left(\frac{z}{\delta}\right)^2 - \left(\frac{\delta\tau_{zr,\delta}}{\mu_L} - 2V_{r,\delta}\right)\left(\frac{z}{\delta}\right)$$

$$V_{\theta L} - r\omega = \left[\frac{\delta\tau_{z\theta,\delta}}{\mu_L} + (r\omega - V_{\theta,\delta})\right]\left(\frac{z}{\delta}\right)^2 + \left[\frac{\delta\tau_{z\theta,\delta}}{\mu_L} + 2(r\omega - V_{\theta,\delta})\right]\left(\frac{z}{\delta}\right) \qquad (20)$$

$$\frac{T - T_{sat}}{T_w - T_{sat}} = 1 - \frac{z}{\delta}$$

vapor region

$$V_{rV} - V_{r,\delta} = \frac{\Delta\tau_{zr,\delta}}{\mu_V}\left(\frac{\eta}{\Delta}\right) - \frac{\Delta\tau_{zr,\delta}}{2\mu_V}\left(\frac{\eta}{\Delta}\right)^2 \qquad (21)$$

$$V_{\theta V} - V_{\theta,\delta} = \frac{\Delta\tau_{z\theta,\delta}}{\mu_V}\left(\frac{\eta}{\Delta}\right) - \frac{\Delta\tau_{z\theta,\delta}}{2\mu_V}\left(\frac{\eta}{\Delta}\right)^2$$

$$T = T_{sat} = \text{const.}$$

where $\eta = z - \delta$. Omitting the details, the problem reduces to the following six equations for the six unknown quantities, δ, Δ, $V_{r\delta}$, $V_{\theta s}$, $\tau_{z\theta,\delta}$ and $\tau_{zr,\delta}$

$$\frac{60}{C^2}(F - P) = 30N^2 + (Q^2 + 16G^2 + 8G - 3Q - 7GQ + 6)(1 - N^2) - 8F^2 + 11FP - 3P^2 \qquad (22)$$

$$\frac{30}{C^2}(Q - G + 1) - 2PQ - 7FQ - 3P + 8F - 2GP + 12FG \qquad (23)$$

$$F = \frac{2 + W}{8 + W}P + \frac{6W}{Pr(8 + 3W)C^2} \qquad (24)$$

$$30KPC + 2BC^2N^2 - (1 - N^2)(40^2B^3K^2 - 20GQB^2CK - 30)BC^2G^2 \times K^2P^2B^3 + 60KB^2CPF + 90BC^2F^2 + 5BC^2P^2K - 20K^2BC^2PI - 50KB^2CFP = 0 \qquad (25)$$

$$15KCQ + 8K^2B^3PQ + 20KB^2CFQ + 20KB^2CGP + 50BC^2FG - 5KC^3G(P - 4F) = 0 \qquad (26)$$

$$C^2K(P - 4F) + 6BCN - 3HC - 2B(BPK + 3CF) = 0 \qquad (27)$$

where

$$N = a/\lambda, \quad B = \Delta(\lambda/v)^{1/2}, \quad P = \delta\tau_{zr,\delta}/r\lambda\mu_L \qquad (28)$$

$$Q = \delta\tau_{z\theta,\delta}/r\omega\mu_L, \quad K = (\rho_L\mu_L/\rho_V\mu_V)^{1/2}.$$

The variable H is related to the potential flow outside the vapor boundary layer and values are tabulated by Hannah [11]. The six non-linear equations were solved numerically using subroutine ZSYSTM [10] on a CDC 6400 computer.

RESULTS AND DISCUSSION

The most important results are the heat transfer to the disk which are calculated from

$$q = k\left(\frac{\partial T}{\partial z}\right)_{z=0} = \frac{k(T_{sat} - T_w)}{\delta}. \quad (29)$$

Introducing the heat transfer coefficient, h, defined by

$$h \equiv \frac{q}{T_{sat} - T_w} = \frac{k}{\delta} \quad (30)$$

and using equation (14) this becomes

$$\frac{h}{k}\left(\frac{\nu}{\sqrt{(a^2 + \omega^2)}}\right)^{1/2} = \frac{1}{C} \quad (31)$$

where C is defined in equation (15).

Results obtained from equation (31) are presented in Figs. 3, 4 and 5 for the dimensionless heat transfer coefficient for Prandtl numbers of 1, 10 and 100. It is noted that for the limiting case of no impingement the present results (which are the same as those of Espig and Hoyle [4]) are in good agreement with those of Sparrow and Gregg [1]. In detail, the results are virtually identical for $c_p\Delta T/h_{fg} = 0.001$ and differ for $c_p\Delta T/h_{fg} = 1$ by 1%, $Pr = 1$, by 5% at $Pr = 10$ and 6% at $Pr = 100$.

It is noted that the results have been based on the assumption of no vapor shear at the outer edge of the condensate layer; that is, they are based on the analysis designated as Case I. Results were also

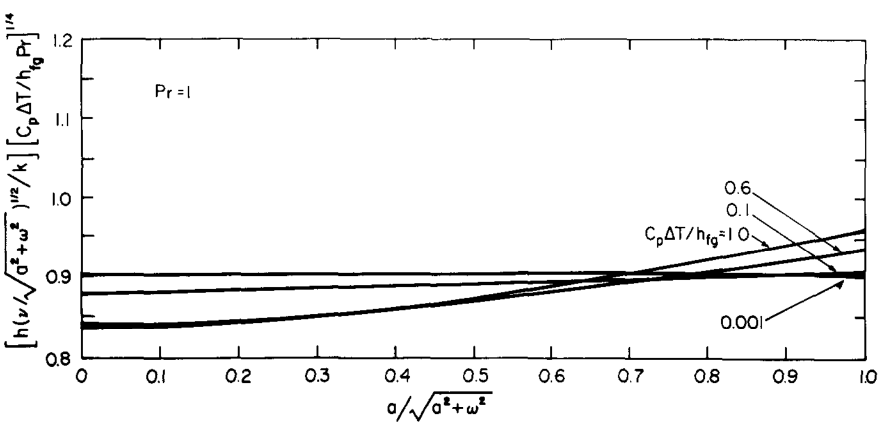

FIG. 3. Heat transfer results for $Pr = 1.0$.

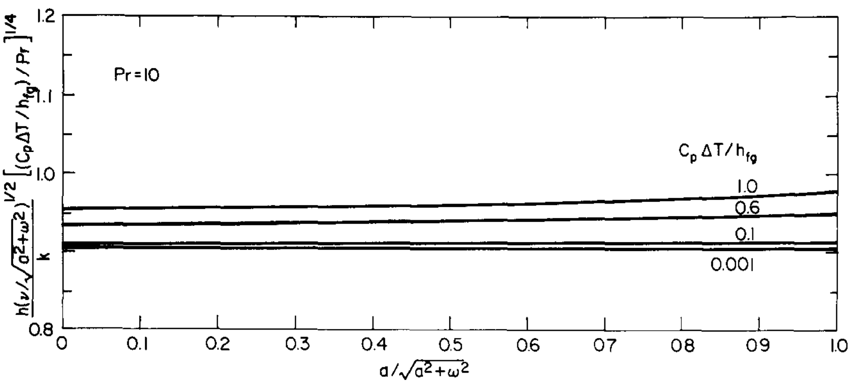

FIG. 4. Heat transfer results for $Pr = 10.0$.

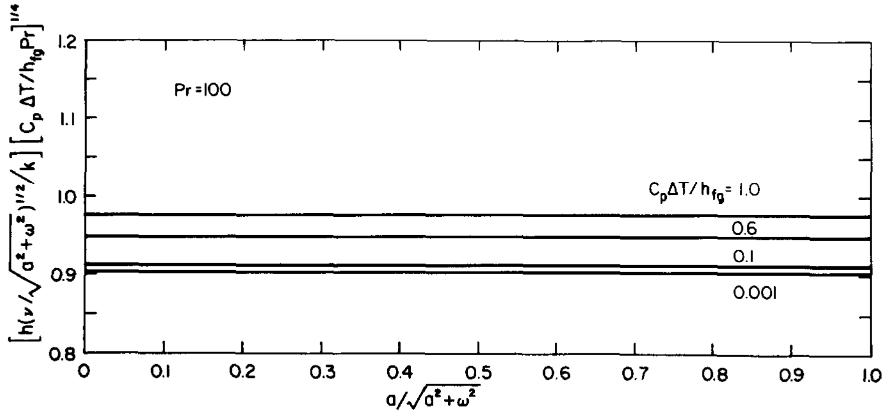

FIG. 5. Heat transfer results for $Pr = 100.0$.

Table 1. Heat transfer results

$\frac{\omega}{a}$	$\frac{a}{(a^2+\omega^2)^{1/2}}$	$\frac{c_p\Delta T}{h_{fg}}$	Pr	$\left(\frac{\rho_L\mu_L}{\rho_V\mu_V}\right)^{1/2}$	$\frac{h}{k}\left(\frac{\nu}{\sqrt{(a^2+\omega^2)}}\right)^{1/2}$ L & VBL	LBL	$\frac{h}{k}\left(\frac{\nu}{\sqrt{(a^2+\omega^2)}}\right)^{1/2}\left(\frac{c_p\Delta T}{Pr\,h_{fg}}\right)^{1/4}$ L & VBL	LBL
0	1	1.0	100	50	3.092	3.094	0.978	0.978
0	1	0.1	100	150	5.128	5.128	0.912	0.912
0	1	0.001	3	50	6.676	6.688	0.902	0.904
0	1	0.001	1	150	5.081	5.082	0.904	0.904
0.5	0.894	0.001	1	150	5.079	5.081	0.903	0.904
1	0.707	0.001	1	150	5.073	5.081	0.902	0.904
2	0.447	0.001	3	150	6.774	6.688	0.915	0.904
2	0.447	0.001	1	150	5.056	5.080	0.899	0.903

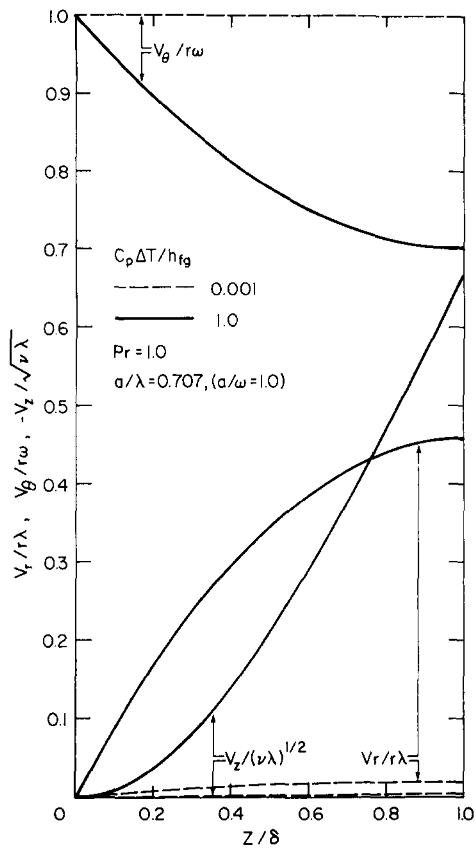

FIG. 6. Dimensionless velocity profiles.

obtained for non-zero shear stress at the liquid–vapor interface based on the analysis designated as Case II. These results are summarized in Table 1. For all the conditions studied, the inclusion of the vapor drag does not have a significant effect on the heat transfer. This is in accord with the conclusion of Sparrow and Gregg [3] for the case of no impinging flow.

For thin films, that is, small values of $c_p\Delta T/h_{fg}$, the dimensionless heat transfer coefficient is a constant equal to 0.904. Sparrow and Gregg [1] have pointed out that this result comes directly from omitting the energy convection and inertia terms in the conservation equations. Deviations from this constant value represent the effects of convection and inertia, the former causing an increase in the heat transfer and the latter tending to decrease it. For large values of the Prandtl number (Pr = 10, Fig. 4 and Pr = 100, Fig. 5) the convection effects become more important and h_{dim} is greater or equal to the limiting value of 0.904. Note that there is only a slight variation of h_{dim} with respect to variations in the impinging parameter $a/(a^2+\omega^2)^{1/2}$. For the smaller values of the Prandtl number the inertia effects become more significant. Thus, for Pr = 1 (Fig. 3) the value of h_{dim} for slow impinging flows is less than the limiting value of 0.904; the effect being more pronounced for the thicker films, i.e. $c_p\Delta T/h_{fg} > 0.10$. For fast impinging flows the effect of convection becomes more important causing an increase in the heat transfer coefficient which exceeds the 0.904 value for no impingement.

The velocity distributions across the condensate film are shown in Fig. 6. Across the thick liquid film the radial component of the velocity increases significantly while the tangential component decreases strongly. Also, for this case the axial velocity component carries a substantial mass to the disk. For the thin film case there are only slight changes in all the velocities across the film and the axial velocity now carries a much smaller amount of mass toward the disk surface. For completeness, it is noted that linear temperature profiles were assumed.

CONCLUSIONS

A study has been made of the heat transfer for an impinging, condensing flow on a rotating disk. Results based on integral methods show that the effect of impingement is to increase the heat transfer. However, over the range corresponding to the experimental studies of Espig and Hoyle [4], the present analysis, which includes the effect of impingement, cannot account for the larger experimental results for the heat transfer. Specifically, Espig and Hoyle [4, p. 437] report an increase of the heat transfer up to 170% which is much greater than the increase calculated in this study. It is also shown that the effect of vapor drag does not have a significant effect on the analytical prediction of the heat transfer. For completeness, it is noted that Espig and Hoyle [4] proposed that wave motion caused the larger experimental results for the heat transfer. It would appear that the effect of waves should be considered in future analyses.

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EFFET DE L'IMPACT SUR LE TRANSFERT THERMIQUE DANS LA CONDENSATION AVEC GIRATION

Résumé—On étudie le transfert thermique pour un écoulement incident qui se condense sur un disque tournant. Des résultats sont obtenus à partir de méthodes intégrales. Bien que l'effet de l'impact soit d'accroître le transfert de chaleur, il est insuffisant pour rendre compte des résultats expérimentaux rapportés dans la bibliographie.

DER EINFLUSS DER ANSTRÖMUNG AUF DEN WÄRMEÜBERGANG BEI KONDENSATION AN ROTIERENDEN SCHEIBEN

Zusammenfassung—Es wurde eine Studie zum Wärmeübergang einer auf eine rotierende Scheibe auftreffenden kondensierenden Strömung durchgeführt. Die Ergebnisse wurden durch integrale Methoden erhalten. Obwohl der Einfluß der Anströmung den Wärmeübergang verbessern soll, ist die Größenordnung nicht ausreichend, um die in der Literatur erwähnten höheren Werte zu erklären.

ВЛИЯНИЕ СОУДАРЕНИЯ ПОТОКА С ПРЕГРАДОЙ НА ТЕПЛОПЕРЕНОС ПРИ КОНДЕНСАЦИИ В ПРОЦЕССЕ ВРАЩЕНИЯ ТЕЛ

Аннотация — Проведено исследование теплопереноса импактной струи жидкости, конденсирующейся на вращающемся диске. Результаты получены с помощью интегральных методов. Несмотря на то, что соударение приводит к интенсификации теплообмена, с его помощью невозможно объяснить высокие экспериментальные значения, приводимые в литературе.