The Effect of Vapor Velocity on Condensation on a Vertical Surface

W. K. LEONARD and JOSEPH ESTRIN

Clarkson College of Technology, Potsdam, New York

Thin liquid films occur in many common types of equipment: wetted-wall towers, evaporators, and condensers. Frequently this thin film flow involves interfacial shear produced by a moving gas or vapor. A common example of this is condensation upon a cold solid surface from a moving vapor. In condensers high vapor velocities are often encountered especially in the entrance region where only a small portion of the total vapor entering is condensed. Much work yielding experimental film condensation heat transfer coefficients has been reported. However, the bulk of the data is for condensation with no shear on the condensate-vapor interface, and the scatter of the data is large. Generally, the experimental results fall above Nusselt's (*1*) classic prediction for steady laminar condensate flow. The limited data for condensation with interfacial shear also show wide scatter.

It has long been realized that high vapor velocities in the direction of gravity would increase the condensate film heat transfer coefficient for vertical condensers. Carpenter and Colburn (*2*) measured condensation coefficients ten times higher than those predicted by the no-shear theories. Several models exist to describe condensation from high velocity vapors. They include the Nusselt (*3*) theory for laminar condensate flow and the Rohsenow et al. (*4*) and Dukler (*5*) models for turbulent condensate flow. These models give distinctly different predictions of heat transfer coefficients for the Reynolds numbers and interfacial shear values commonly encountered. For example, at a Reynolds number (N_{ReL}) of 1,000 and no interfacial shear, Dukler's model predicts coefficients 100% higher than those of Nusselt and Rohsenow et al. Limited experimental data showing the effect of vapor velocity have been reported in the literature (*2, 6, 7*). Wide scatter of these data and variation in vapor velocity along the condensing length have not allowed verification of existing theories. The work described here (*8*) was undertaken to demonstrate condensation with and without significant vapor velocities, which hopefully would support an existing theory.

The discrepancy between the Dukler and Rohsenow et al. theories is chiefly due to the two different ways in which they describe the onset of turbulence in the condensate film. Rohsenow et al. assumed that turbulence began when a particular value of wall shear was exceeded. They used the universal velocity distribution to describe the turbulent flow. Dukler discarded the idea of a distinct transition from laminar to turbulent flow and considered a combined mechanism always to be present. He described the heat transfer in terms of the sum of components due to molecular and turbulent transport and used Deissler's and Von Karman's expressions for eddy viscosity.

In this work, average heat transfer coefficients were obtained experimentally with a condenser constructed specifically to investigate the effect of vapor velocity upon condensation. The equipment is shown schematically in Figure 1. Significant features of the equipment were the geometry of the condenser and the readily controlled vapor velocity; large cooling water flows ensured nearly constant wall temperatures. The control of the vapor velocity was accomplished by controlling the feed rate to the boiler and for some runs by using a gas blower to recirculate the vapor through the test section. Condensation occurred on the outside of a ⅝-in. O.D. stainless steel tube passing through the center of a 3-in. I.D. shell made of glass pipe. The 2½ ft. of stainless tube above the condensing section were insulated by a tightly fitting Teflon tube of ⅛-in. wall thickness. Below the 1-ft. long condensing section the stainless tube was surrounded by a ¾-in. I.D. pipe which served as a condensate collector. The condensate level was maintained at the top of this collector. The condenser geometry allowed observation of the entire condensing surface. This was important because the experimental work was carried out with steam which readily condenses by either a dropwise or filmwise mechanism.

Average condensation heat transfer coefficients were calculated from measurements of the condensate flow rate and condensing section wall temperature. The condensate flow rate was measured volumetrically and the wall temperature was measured by thermocouples embedded in it. Two of them were attached to the wall at locations 1 in. from the ends of the test section. The thermocouples were soft-soldered in grooves cut in the stainless tube. The soldered connections were ground and polished flush with the tube wall. Accuracy of the calculated heat transfer coefficients was limited by the wall temperature measurement. The maximum error in the measurement was estimated to be ± 4°F. Because of this uncertainty the error in the data obtained for low Reynolds numbers (less than 100) may be as large as 60%.

From vapor flow rate measurements interfacial shear was estimated. The experimental work of Bergelin et al. (*9*) showed that for the condensate flow rates and vapor velocities encountered here, the vapor phase pressure drop may be calculated by assuming no condensate to be pres-

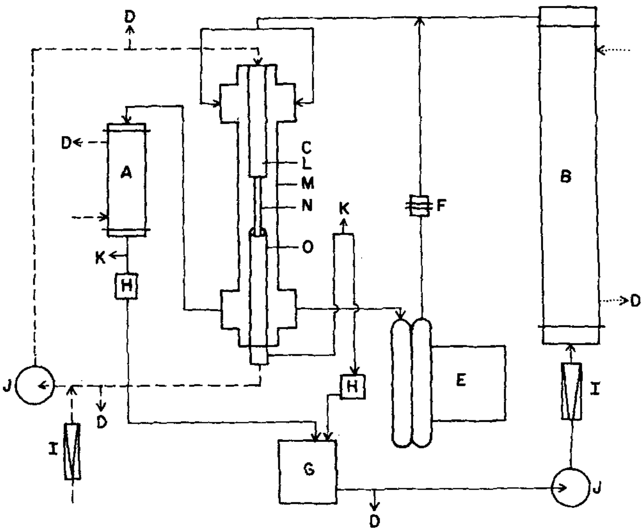

Fig. 1. Experimental apparatus. *A*, auxiliary condenser; *B*, boiler; *C*, test condenser; *D*, drain; *E*, blower; *F*, orifice meter; *G*, storage tank; *H*, volumetric measuring device; *I*, rotometer; *J*, pump; *K*, vent; *L*, Teflon insulation; *M*, glass pipe; *N*, condensing section, S.S. tubing; *O*, condensate collector; ——— vapor and condensate lines; ----- water lines; . . . . . steam lines.

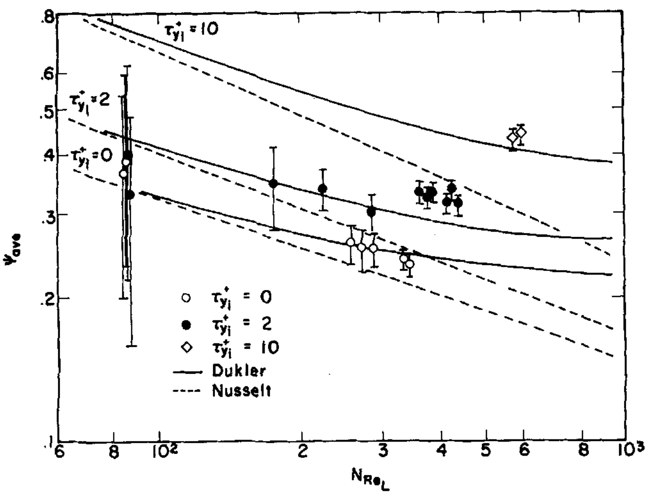

Fig. 2. Comparison of experimental data to theory N_{Pr} = 2.

ent. Therefore, a good estimate of the interfacial shear was obtained by assuming the interfacial shear to be equal to the calculated value of shear exerted on the wall with no condensate present. The shear on the inner wall of an annular duct was calculated by a method suggested by Knudsen and Katz *(10)*.

The maximum error in the vapor flow rate measurements was estimated as 15%. Much of this is attributable to fluctuations in the feed rate to the boiler. These fluctuations did not have a significant effect on the heat transfer coefficients. Their effect was hidden by the much larger uncertainties in the coefficients caused by the error in the wall temperature measurements. In all cases the momentum transferred across the interface by mass transfer was considered insignificant compared with the shear exerted by the motion of the vapor.

For the runs where the interfacial shear was estimated to be zero, calculations of condensate film surface velocity showed it to be of the same order of magnitude as the vapor velocity. Typical water surface and vapor velocities were 0.5 and 1.7 ft./sec., respectively. Because vapor velocities of 30 ft./sec. were needed to give dimensionless shear values of 2, the shear for a vapor velocity of 1.7 ft./sec. is certainly insignificant.

The experimental results are given in Figure 2. Because the condensate Prandtl numbers ranged from 1.8 to 2.6 a correction based on Dukler's theory was applied. It adjusted the data for comparison purposes to a Prandtl number of 2. With the exception of the results for condensate Reynolds numbers (N_{Re_L}) less than 100, the data obtained showed good reproducibility. To check if the equipment was operating properly, average heat transfer coefficients for the cooling water side of the test condenser were calculated. Since the cooling water flow rate was nearly the same for all experimental runs, a large deviation in the coolant side coefficient from the usual range [1,500 to 2,500 B.t.u./(hr.)(sq.ft.)(°F.)] was indicative of poor operation. Data for Reynolds numbers less than 600 and a dimensionless shear ($\tau^+_{y_i}$) of 10 were not obtained because dropwise condensation was encountered in this region.

For the range of condensate Reynolds numbers and interfacial shears investigated, the Rohsenow et al. criterion for transition from laminar to turbulent flow is not exceeded, so their work gives the same results as Nusselt's. Figure 2 shows that within the limits of error the experimental results for zero interfacial shear are in good agreement with Dukler's theoretical predictions. Also, his theory predicts the increased heat transfer produced by cocurrent vapor and condensate flow with greater precision than Nusselt's theory.

For a dimensionless interfacial shear of 2 and for Reynolds numbers greater than 300, the results fall higher than Dukler's theoretical curve. This is also true for a dimensionless shear of 10. Many possibilities exist to explain condensation coefficients higher than predicted by Dukler's model. His model or the method of calculating the interfacial shear may be incorrect. The limited number and range of the data make it difficult to attribute the high coefficients to any specific cause. Probably a combination of effects is responsible. However, in general the agreement may be considered good.

Although the amount of experimental data is limited, it clearly shows Nusselt's condensation theory with or without the effects of interfacial shear to be inadequate in the Reynolds number range of 200 to 600. With the exception of the data below a Reynolds number of 200 (where the possible error is large), Dukler's theory predicts the experimental data much more closely than does Nusselt's theory.

NOTATION

Cp = specific heat at constant pressure, B.t.u./(lb.)(°F.)
g = gravitational acceleration, ft./sec.^{2}
g_c = gravitational constant, (32.17)(lb.)(ft.)/(sec.^{2})(lb.$_f$)
h_{ave} = average heat transfer coefficient, B.t.u./(sec.)(sq.ft.)(°F.)
k = thermal conductivity, (B.t.u.)(ft.)/(sec.)(sq.ft.)(°F.)
N_{Pr} = Prandtl number = $Cp\,\mu/k$
N_{Re_L} = Reynolds number of condensate at the bottom of the test section = $4\Gamma/\mu$

Greek Letters

ψ_{ave} = dimensionless average heat transfer coefficient = $h_{ave}\left(\frac{\mu^2}{\rho^2 g k^3}\right)^{1/3}$
$\tau^+_{y_i}$ = dimensionless shear stress = $\frac{g_c \tau_y}{g^{2/3}\mu^{2/3}\rho^{1/3}}$
Γ = condensate mass flow rate per unit perimeter, lb./(sec.)(ft.)
ρ = density, lb./cu.ft.
μ = viscosity, lb./(sec.)(ft.)
τ_y = shear stress, lb.$_f$/sq.ft.

Note: Physical properties refer to properties of the condensate film.

LITERATURE CITED

1. Nusselt as given in W. H. McAdams, "Heat Transmission," 3 ed., p. 329, McGraw-Hill, New York (1954).
2. Carpenter, F. G., and A. P. Colburn, "Proceedings of the General Discussion of Heat Transfer," p. 20, Inst. Mech. Engrs.-Am. Soc. Mech. Engrs., London (July, 1951).
3. Nusselt as given in C. C. Monrad and W. I. Badger, *Ind. Eng. Chem.*, **22**, 1103 (1930).
4. Rohsenow, W. M., J. H. Webber, and A. T. Ling, *Trans. Am. Soc. Mech. Engrs.*, **78**, 1637 (1956).
5. Dukler, A. E., *Chem. Eng. Progr.*, **55**, 62 (Oct., 1959).
6. Nicol, A. A., *Can. J. Chem. Eng.*, 278 (Dec., 1964).
7. Jakob, M., *Mech. Eng.*, **58**, 729 (1936).
8. Leonard, W. K., M.S. thesis, Clarkson Coll., Potsdam, N. Y. (1966).
9. Bergelin, O. P., P. K. Kegel, F. G. Carpenter, and Carl Gazley, Jr., "Heat Transfer and Fluid Mechanics Institute, Berkeley, Calif.," p. 19, Am. Soc. Mech. Engrs. (1949).
10. Knudson, J. G., and D. L. Katz, "Fluid Dynamics and Heat Transfer," McGraw-Hill, New York (1958).