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Discussion

Convective Condensation of Superheated Vapor[1]

A. F. Mills[2]. Webb (1998) has proposed a method for calculating the effect of vapor superheat on condensation heat transfer that is "theoretically based" and generally applicable. The purpose of this discussion is to show that the proposed method is an ad hoc model that is not based on theory, does not have correct limiting behavior, and is not generally applicable.

A theoretically sound method to account for vapor superheat was proposed more than 60 years ago by Ackermann (1937), and subsequently has been presented in textbooks such as those by Spalding (1963) and Mills (1995). This method is reviewed first as a basis for evaluating the Webb proposal. Convective heat transfer from the superheated vapor to the condensate surface is given by

$$q_{\text{conv}} = k \left.\frac{\partial T}{\partial y}\right|_s = h_c(T_b - T_s) \tag{1}$$

where T_s is the condensate surface temperature and T_b is the bulk or freestream vapor temperature. Equation (1) defines the heat transfer coefficient h_c. The Ackermann analysis is based on a stagnant film model (or equivalently, a Couette flow or Reynolds flux model) of the vapor flow, in lieu of solving exact conservation equations governing the vapor flow. For a pure vapor the result is

$$\frac{h_c}{h_c^0} = \frac{B_h}{\exp B_h - 1}\,;\; B_h = \frac{\dot{m}'' c_{pv}}{h_c^0} \tag{2}$$

where $\dot{m}''$ [kg/m^{2}s] is the mass transfer rate and is negative for condensation, and h_c^0 is the limiting value of h_c as $\dot{m}''$ goes to zero: h_c^0 is estimated from an appropriate impermeable surface convection correlation. B_h is usually termed a "blowing parameter." The Ackermann method is theoretically sound in the sense that, for the implied Couette flow model, the effect of mass transfer is mathematically correct. Of course, applications involve approximations related to true nature of the vapor flow, including the effects of the liquid film on the vapor flow.

The use of Eqs. (1) and (2) depends on the particular problem, and any further approximations that may be warranted. In particular, the two asymptotic limits of Eq. (2) are useful. First, for high condensation rates (strong suction limit), $B_h \to -\infty$, and

$$h_c = -\dot{m}'' c_{pv} \tag{3}$$

$$q_{\text{conv}} = -\dot{m}'' c_{pv}(T_b - T_s) \tag{4}$$

Secondly, for low condensation rates, $B_h \to 0$,

$$\frac{h_c}{h_c^0} = 1 - \frac{1}{2} B_h + O(B_h^2). \tag{5}$$

Notice that Webb's Eq. (13) results from the use of Eq. (4) in Nusselt's analysis for condensation on a vertical wall, as shown in Section 7.2.4 of Mills (1995). For condensation of refrigerants inside tubes at relatively low condensation rates, as considered in some detail by Webb, Eq. (5) applies and then Eq. (1) gives

$$q_{\text{conv}} = h_c^0(T_b - T_s) - \tfrac{1}{2}\dot{m}'' c_{pv}(T_b - T_s). \tag{6}$$

Turning now to Webb's proposed method, the essential element is his Eq. (7), which in the current notation, is

$$q_{\text{conv}} = h_c^0(T_b - T_s) - \dot{m}'' c_{pv}(T_b - T_s). \tag{7}$$

Comparing Eq. (7) to Eq. (6) we see that Eq. (7) is incorrect in the limit case of low condensation rates, overestimating the effect of mass transfer by 100 percent. Also, comparing to Eqs. (2) and (4) we see that Eq. (7) is not generally valid, and in the limit of high condensation rates overestimates q_{conv} by $h_c^0(T_b - T_s)$. It is important to recognize that Eq. (2) is based on a model that is a limit form of exact conservation equations, and it is well established in the literature that Eq. (2) is a good approximation to more exact solutions of the conservation equations for various types of forced flow. Since Webb's method is ad hoc and contradicts Eq. (2) it should be rejected as incorrect and unnecessary.

Other elements in Webb's development require comment.

1 Webb's Eqs. (1) and (2) are also ad hoc. They are good approximations for forced convection condensation in horizontal tubes, but are invalid for gravity dominated condensation in vertical tubes and on external surfaces.

2 Webb's Eq. (9) is not "identical" to the Colburn-Hougen model for condensation in the presence of a noncondensable gas. The Colburn-Hougen method accounts for sensible and latent heat gas phase resistances in parallel, implicitly in series with the liquid film thermal resistance, and is generally valid (see, for example, Example 10.9 of Mills (1995)). Webb's Eq. (9) cannot be given an equivalent circuit interpretation: It represents neither resistances in parallel or series. It is valid only if the thermal resistance of the condensate film is independent of superheat (which is generally not true) and is a consequence of the resulting linearity of the thermal problem.

3 Webb's relation $\dot{m}_v''/A = q_{\text{lat}}/i_{fg}$ is also generally invalid: It is untrue if the condensate film resistance depends on vapor superheat.

4 Webb's claim that the mass transfer correction would be significantly smaller for steam than for R-22 is also not generally valid. From his Eq. (9), or equivalently Eq. (2) of this discussion, the appropriate parameter scaling the effect of mass transfer is $B_h = \dot{m}'' c_{pv}/h_c^0$. Condensation of steam in practice

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usually occurs at low pressures for which h_c^0 is small, and with thin condensate films for which $\dot{m}''$ is large. The aforementioned Example 10.9 of Mills (1995) shows a value of $B_h = -6.77$ and $h_c/h_c^0 = 6.78$, which is not significantly lower than 1.4 as suggested by Webb. Of course, the effect of vapor superheat on the condensation rate remains small.

5 Webb's Fig. 1 shows a comparison of his model predictions with experimental data. There is no similar evaluation of the well-established Ackermann model. However, given the expected uncertainty in the experimental data due to random and bias error, and the scatter shown in Fig. 1, it is doubtful if any conclusion could be drawn from this comparison.

References

Ackermann, G., 1937, "Heat Transfer and Molecular Mass Transfer in the Same Field at High Temperatures and Large Partial Pressure Differences," *Forsch. Ing. Wes. VDI, Forschungesheft,* Vol. 8, p. 232.

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Author's Closure[1]

Prior to addressing Professor Mills comments it is helpful to review the chronology of the analysis methods referenced by Mills and in the present work. The 1934 Colburn and Haugen analysis was the first to address the effect of noncondensible gas on condensation. They proposed that the heat transfer rate is the sum of the latent (condensation) and sensible load (cooling of vapor to saturation temperature), which is the same as Eq. (2) in the present work. They worked an example for gas condensing on the outside of tubes. Their analysis did not include a correction for the effect of mass transfer on the sensible heat term. The 1937 analysis of Ackermann formulated a correction factor to account for the effect of mass transfer on the sensible heat term. Present day analysis (e.g., Collier and Thome (1994) and Hewitt et al. (1994)) of the noncondensible gas problem typically includes use of the "Ackermann correction factor" in the Colburn and Hougen equation.

We were not aware of Mills analysis of vapor velocity effect, which he references in his discussion. It appears that the Mills analysis is intended to treat the fundamentals of the problem by giving the differential equation that is to be solved, as opposed to giving a *solution* as is done in the present work. Mills analysis derives the Colburn and Haugen differential equation with the Ackermann correction factor.

There are two basic ways to solve convective heat transfer problems: one is to solve a differential equation (for a given flow geometry) by integrating the temperature profile across the flow field to obtain the temperature profile. Then, one calculates the heat flux by calculating $k(dT/dy)$ at the wall. This is the approach described by Mills, although he did not actually solve the differential equation. The other approach is to use heat transfer coefficients and thus calculate the heat flux using $q = h(T_{\text{sat}} - T_w)$, as is done in the present work. We have formulated a composite heat transfer coefficient using Eq. (2) and have given a generalized solution method (Eq. (9)) that is applicable for any flow geometry.

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Equation (2) properly states the underlying theory of the Colburn-Haugen equation as stated on page 1179 of the Colburn and Haugen paper, or the present Eq. (2). However, as noted above, Colburn and Haugen did not correct h_{fc} for mass transfer. We have included a correction factor for mass transfer as is included in Eq. (7).

It appears that the principal concern expressed by Professor Mills relates to our use of a different formulation to obtain the correction factor to the single-phase heat transfer coefficient to account for mass transfer. Our formulation could have been developed using the "Ackermann correction factor." However, we chose not to use the Ackermann formulation, which is based on the *assumption* of Couette (laminar between two parallel plates with the upper plate moving at a specified velocity). The present formulation is more general and is applicable to laminar or turbulent flow and is applicable to any geometry. The present formulation uses an energy balance to: (1) determine the mass flux to the interface, (2) write the bulk convection of sensible heat in terms of a heat transfer coefficient, and (3) define a composite heat transfer coefficient (h_{fc}^*) to include the effect of bulk convection to the single-phase term in Eq. (2). The result is Eq. (8). One may manipulate Eq. (8), which I will call Eq. (8a) here to yield

$$\frac{h_{fc}^*}{h_{fc}} = 1 + \frac{c_{pv}q_{\text{lat}}}{i_{fg}h_{fc}} . \qquad (8a)$$

Equation (8a) is equivalent in concept to the Akermann correction factor (Mills, Eq. (2)). Mills argues that Eq. (8a) does not have the same asymptotes as his Eq. (2). I agree that it is not asymptotic to his Eq. (2), because the detailed formulation leading to Eq. (8a) is different. There is no reason why it should have the asymptotes of his Eq. (2), because it is not based on the same assumptions or restrictions (e.g., the laminar flow model of Ackermann). Should the reader be interested in the Ackermann analysis, it is given in detail on p. 604 of Hewitt et al. (1994).

If one prefers, they could use the Akermann correction factor (Mills, Eq. (2)), rather than Eq. (8) to account mass transfer effects on the q_{sens} term in Eq. (2). Because the latent heat term dominates Eq. (2), it is possible that one would see little practical difference between the two formulations. However, I have not performed a sample calculation to evaluate the difference between the two formulations.

Mills item 3 criticism states that "the relation $(m_v/A) = q_{\text{lat}}/i_{fg}$ is generally invalid; it is untrue if the condensate film resistance depends on vapor superheat." We do not argue that the condensate film resistance should depend on superheat. Nothing in the present model says that.

Professor Mills is critical of the predictive ability of the Lee (1991) model, and he further implies that the Lee data are not very good. We feel that his criticism is unjustified. The predictive ability of the model is shown in Fig. 1 and is good. Professor Mills admits that "there is no similar evaluation of the well-established Akermann model."

In summary, we have presented a formulation based on use of heat transfer coefficients for condensation and single-phase heat transfer that is applicable to any flow geometry. If one prefers to use the Ackermann correction factor (rather than Eq. (8a) above), one may do so. We have shown that the predicted results are in good agreement with the convective condensation data of Lee.