Optimal Sizing of Two-Phase Heat Exchangers

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ABSTRACT

Correlations for two-phase heat transfer and pressure drop developed by ACRC Project 01 are presented and discussed. In particular, these correlations are used to assess the impact of the tube diameter on the required length of heat exchanger to perform phase-change processes. The correlations are integrated into a spreadsheet for evaporation and condensation that predicts the required length of two-phase heat exchanger when supplied with the operating conditions. The spreadsheets and macro sheets which provide needed thermodynamic and transport properties are included with the report. The spreadsheets were used to study the impact of tube diameter on the required length of condenser and evaporator tube to transfer a fixed amount of heat as mass flow rate and air side resistance were held constant. The results revealed that over a wide range of diameters, the required length was relatively insensitive to the tube diameter. In this region, the required length increased slightly as the diameter was decreased. As the diameter became sufficiently small, pressure drop destroyed much of the driving temperature difference. At diameters below this, the required length of heat exchanger increased greatly with further decreases in diameter.

NOMENCLATURE

SYMBOLS

A_c	cross sectional area
c_p	specific heat at constant pressure
D	inner tube diameter
f	friction factor
F	evaporation heat transfer reduction parameter
Fr	Froude number
g	acceleration due to gravity
G	mass flux, $\frac{\dot{m}}{A_c}$
Ga	Galileo number, $\frac{gD^3\rho_l^2}{\mu_l^2}$
h	heat transfer coefficient
i_{lv}	enthalpy of vaporization
j_g^*	Wallis' dimensionless gas velocity
k	thermal conductivity
L	length
M	molecular weight
$\dot{m}$	mass flow rate
Nu	Nusselt number, $\frac{h\,D}{k_l}$
Pr	Prandtl number, $\frac{\mu\, c_p}{k}$
q	heat transfer rate
q"	heat flux
R'	thermal resistance per unit length
Re_l	superficial liquid Reynolds number, $\frac{G\,D\,(1-x)}{\mu_l}$
x	vapor quality
X_{tt}	Lockhart-Martinelli parameter, $\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}\left(\frac{1-x}{x}\right)^{0.9}$

NOMENCLATURE

GREEK SYMBOLS

α	void fraction
ΔP	pressure difference
ΔT	temperature difference
ϕ_l^2	two-phase friction multiplier
μ	dynamic viscosity
ρ	density

SUBSCRIPTS

c	cross section
cb	convective boiling
i	inlet
l	liquid
nb	nucleate boiling
o	outlet
v	vapor

INTRODUCTION

Refrigerant side evaporation and condensation have been major topics of research in the ACRC since its inception. The work to date has focused primarily on determining local heat transfer and pressure drop data and correlating that data based on the appropriate dimensionless groups. This work is important from both a fundamental and practical standpoint, since an understanding of the fundamental physics is necessary to make intelligent design recommendations. To equipment designers, though, the ultimate utility of this research is its application to design problems. A first effort towards this end is presented herein. The specific problem that is addressed is what tube diameter is optimal for a given mass flow rate, air side resistance, and driving temperature difference.

For a given mass flow rate, reducing the tube diameter has three primary effects on heat transfer: (1) increasing the heat transfer coefficient, (2) decreasing the heat transfer area per unit length, and (3) increasing the pressure drop. The increase in heat transfer coefficient clearly reduces the required length of heat exchanger, while the decrease in heat transfer area tends to counteract this effect. The increase in pressure drop also lengthens the heat exchanger since it destroys the temperature difference that drives the heat transfer. This mix of effects suggests, at least qualitatively, that an optimum tube diameter might exist for each set of operating conditions. This question is addressed quantitatively in this report.

First, the heat transfer and pressure drop correlations that have been developed as part of Project 01 are presented. These correlations are used to discuss the effect of various parameters on the heat transfer coefficient, primarily diameter, quality, and mass flow. These correlations were combined to develop a simple computer simulation that can be used to determine the required length of heat exchanger for a given set of operating conditions. This model was used to compute the required length of heat exchanger, holding mass flow rate and air side properties constant, as the diameter was varied.

HEAT TRANSFER AND PRESSURE DROP CORRELATIONS

For steady-state, one-dimensional heat transfer with constant properties, heat transfer can be posed in terms of thermal resistances. If the thermal resistance of the high conductivity tube wall is neglected, we arrive at the following:

$$q = \frac{L * \Delta T}{R'_{AIR} + R'_{REF}} \tag{1}$$

where: $R'_{ref} = \frac{1}{\pi D h}$ = thermal resistance per unit length

For the sake of the analysis discussed here, we will assume that the air side resistance is known and focus on the effect of the refrigerant side heat transfer. Two rather simple conclusions can be drawn from Eq. (1): (1) we should attempt to keep the temperature difference as high as possible, which implies low pressure drop, and (2) we should focus on increasing the quantity h*D, not just h. The πDh term in the refrigerant side resistance is the heat transfer coefficient times the perimeter. This coupling of the heat transfer coefficient with the perimeter reminds us that decreasing the heat exchanger length by decreasing the tube diameter requires an increase in heat transfer coefficient to a power greater than 1. This additional diameter dependence can be eliminated by writing the refrigerant side resistance in terms of the Nusselt number, hD/k. The resulting expression is:

$$R'_{REF} = \frac{1}{\pi Nu k_L} \tag{2}$$

<u>Condensation Heat Transfer</u>

Condensation heat transfer can be broadly divided into two regimes: (1) gravity-driven, Nusselt-type condensation, and (2) forced convective condensation. At low flow rates, in the wavy and stratified flow regimes, Nusselt type condensation is expected to prevail. In this mode of condensation, heat transfer is accomplished primarily by conduction across the thin liquid film at the top of the tube. A recently developed correlation for wavy-flow condensation [Dobson et al., 1993] is:

$$Nu = f(X_{tt})^* \left[\frac{g\rho_L(\rho_L - \rho_V)D^3 i'_{lv}}{\mu_l \, \Delta T \, k_l}\right]^{0.25} \tag{3}$$

$$\text{where: } f(X_{TT}) = \frac{0.375}{X_{TT}^{0.23}}$$

Breaking this correlation down to study the effect of the individual parameters, we first notice that the Nusselt number is independent of the mass flow rate (or flux). This means that decreasing the diameter, thereby increasing the velocity, will have no positive effect on heat transfer as long as the flow regime remains wavy. In fact, the appearance of the $D^{0.75}$ term in the numerator shows that decreasing the diameter in the wavy flow regime will <u>increase</u> the thermal resistance to the 3/4 power. The other parametric effect of interest that is revealed by Eq. (3) is the appearance of the $\Delta T^{0.25}$ term in the denominator. Thus, increasing the driving temperature difference decreases the heat transfer coefficient in wavy-flow condensation. This is opposite to the effect of increasing ΔT in evaporation, where it serves to generate stronger nucleate boiling. The Nusselt number modestly increases with increasing quality due to an increasing void fraction. This results in more of the tube surface being covered by the thin film and less of it being occupied by the thick liquid pool at the bottom of the tube.

As the flow rate is increased, gravitational forces are overwhelmed by shear and inertial forces and forced convective condensation prevails. In this regime, the heat transfer is largely dominated by shear forces, so that increased velocities, mass fluxes, or mass flows will tend to increase the heat transfer coefficient. A recently presented correlation for annular flow, convective condensation is [Dobson et al., 1993]:

$$Nu = 0.023 Re_L^{0.80} Pr_L^{0.3} g(X_{TT}) \tag{4}$$

$$\text{where: } g(X_{TT}) = \frac{2.61}{X_{TT}^{0.805}}$$

This correlation consists of a liquid heat transfer coefficient, predicted by the Dittus-Boelter correlation for single-phase flow, times a two-phase multiplier, g, which is determined by the Lockhart-Martinelli parameter. The effect of both mass flow and diameter in this flow regime are determined by the superficial liquid Reynolds number, Re_L, which can be written in terms of mass flow rate as:

$$Re_L = \frac{4\dot{m}(1-x)}{\pi D \mu_L} \tag{5}$$

Thus, if we hold the mass flow rate constant, the Nusselt number is inversely proportional to $D^{0.8}$. This means that reducing the diameter at constant mass flow rate will decrease the thermal resistance, allowing a decreased tube length. This argument assumes that the temperature difference is held constant, which will be addressed shortly. The effect of quality is much more pronounced in the annular flow regime than in the wavy flow regime. This is because the heat transfer is driven primarily by interfacial shear stresses, which increase greatly as the velocity difference between the two phases increases.

The remaining obstacle to predicting condensing heat transfer coefficients is deciding appropriate criteria for predicting flow regime transitions. A simple method for predicting the transition from wavy or intermittent flow to annular flow is Wallis dimensionless gas velocity [Breber et al., 1979]

$$j_G^* = \frac{G\,x}{\sqrt{gD\rho_V(\rho_L - \rho_V)}} \tag{6}$$

Wallis dimensionless gas velocity represents the ratio of the vapor inertial forces to the gravitational forces on the liquid film. For low values of j_G^*, gravity forces are more significant and wavy flow results. For high values of j_G^*, the high vapor velocity overwhelms the gravity forces and annular flow results. Breber et al. [1979] stated that annular heat transfer correlations were applicable for values of $j_G^*>1.5$. Our own data indicate that a value of 1.8 is more appropriate. We recommend using Eq. (3) for $j_G^*<1.8$, and Eq. (4) for $j_G^*>1.8$.

Evaporation Heat Transfer

In general, flow boiling heat transfer takes place through two different mechanisms: (1) convective boiling at the liquid-vapor interface and (2) nucleate boiling at the tube wall. Depending on the mass flow rate, the heat flux, and the fluid, either or both of these heat transfer mechanisms may be significant.

The nucleate boiling contribution to heat transfer has been studied quite extensively in the literature. A correlation proposed by Cooper [1984] has been chosen because of its convenient form and accuracy. Cooper's nucleate boiling correlation can be written in the form:

$$Nu_{NB} = \frac{D}{k_L} * \frac{55q''^{0.67} P_R^{0.12}}{M(\log_{10}(P_R))^{0.55}} \tag{7}$$

For this correlation to work properly, the heat flux, q, must be in SI units of W/m². The main parameters affecting the nucleate boiling contribution are the heat flux, q, the reduced pressure, P_r, and the molecular weight, M. The nucleate boiling is increased by increasing q because this increases the wall superheat, which tends to activate more nucleation sites. Increasing the reduced pressure reduces the surface tension, which is a retarding force for bubble formation and growth. There is no effect of either mass flow or quality on the nucleate boiling contribution to heat transfer, while it is proportional to D. Thus, like the gravity driven condensation case, decreasing the diameter will <u>reduce</u> the nucleate boiling Nusselt number and thereby <u>increase</u> the thermal resistance.

The convective boiling correlation used in this study was recently proposed by Wattelet et al. [1993], and takes the form:

$$Nu_{CB} = 0.023 Re_L^{0.8} Pr_L^{0.4}\left[1+\frac{1.925}{X_{TT}^{0.83}}\right] * F \tag{8}$$

The convective correlation is equal to a liquid heat transfer coefficient, given again by the Dittus-Boelter correlation, times the bracketed term, which is a two-phase multiplier that is applicable for annular flow conditions, times the parameter F. The F parameter is given by:

$$\begin{aligned} &F = 1.0 \qquad \text{for } Fr_L >= 0.25 \\ &F = 1.32 Fr_L^{0.2} \text{ for } Fr_L < 0.25 \\ &\quad \text{where: } Fr_L = \frac{G^2}{\rho_L^2 g D} \end{aligned} \tag{9}$$

It can be seen from Eq. (8) that for high values of the Froude number, F=1.0 so that the bracketed function of the Martinelli parameter serves as the only two-phase multiplier. In the wavy flow regime, the parameter F is a reduction factor that accounts for the deviation from annular flow and the associated loss of turbulence.

The effect of reducing the diameter on the convective boiling contribution to the heat transfer must be addressed separately for the annular and wavy regimes. For the annular regime, decreasing D at constant mass flow increases the Nusselt number to the 0.8 power. This diameter dependence is identical to that for annular condensation, and the effect of quality is nearly identical. In the wavy flow regime, including the Froude number effect we find that the Nusselt number is proportional to $1/D^{1.0}$. Thus, decreasing the diameter results in increasing the Nusselt number for wavy, convective boiling to the 1.0 power. This is the strongest effect of diameter for any heat transfer mode.

The key to creating a successful flow boiling correlation is to determine the proper way to couple the convective and nucleate portions of the heat transfer into one correlation. Kutateladze [1961] was quite successful in using what is referred to as an asymptotic form, given by:

$$Nu_{TP} = [Nu_{CB}^{n} + Nu_{NB}^{n}]^{1/n} \tag{10}$$

The value of n used for this study was 2.5, as recommended by Wattelet [1993].

<u>Pressure Drop</u>

The two-phase pressure drop inside of a tube is composed of three components: (1) friction, (2) acceleration, and (3) gravity. For horizontal heat exchangers, the gravitational component of pressure drop is zero within each single tube, and small throughout the entire heat exchanger. Moreover, it is not dependent on diameter changes, the main item discussed herein. For this reason, it will be ignored.

The frictional pressure drop in a two-phase flow is a result of shear forces which are primarily generated at three places: (1) at the tube wall-liquid interface, (2) at the liquid-vapor interface, and (3) at the vapor-tube wall interface, if one exists. Correlation of the two-phase frictional pressure drop has been widely studied, with most researchers following the method set forth by Lockhart and Martinelli [1947]. Their basic hypothesis was that the two-phase pressure gradient was equal to that which would be experienced either by the liquid or vapor phase based on single-phase approaches, times a two-phase multiplier, ϕ^2. Focusing on the liquid phase, this takes the form:

$$\Delta P_{FRIC} = \Delta P_L \phi_L^2 \tag{11a}$$

$$\text{where: } \Delta P_L = \frac{2 f_L G^2 (1-x)^2 L}{\rho_L D} \tag{11b}$$

$$f_L = \frac{0.079}{Re_L^{0.25}} \tag{11c}$$

Furthermore, they hypothesized that this two-phase multiplier was a function of what is now called the Lockhart-Martinelli parameter, which is the ratio of the pressure drop expected for the liquid phase divided by that expected for the vapor phase. For the case of turbulent flow of both phases, and using the Blasius correlation for the friction factor, we arrive at the turbulent-turbulent Lockhart-Martinelli parameter:

$$X_{TT} = \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_V}\right)^{0.125} \left(\frac{1-x}{x}\right)^{0.875} \tag{12}$$

Other research since their pioneering work has found that the two-phase multiplier also varies systematically with mass flux. Recent work in the ACRC by de Souza et al. [1993] has developed an accurate correlation for the two-phase multiplier based on this approach. The correlation is:

$$\phi_L^2 = (1.376 + \frac{c_1}{X_{TT}^{c_2}}) \tag{13}$$

$$\begin{aligned}
&\text{For } 0 < Fr_L \leq 0.7 \\
&\qquad c_1 = 4.172 + 5.48 Fr_L - 1.564 Fr_L^2 \\
&\qquad c_2 = 1.773 - 0.169 Fr_L \\
&\text{For } Fr_L > 0.7 \\
&\qquad c_1 = 7.242 \\
&\qquad c_2 = 1.655
\end{aligned}$$

In the above correlation, the mass flux dependence is accounted for with the liquid Froude number, which has the effect of decreasing the two-phase multiplier due to loss of turbulence at low mass fluxes. The accuracy of the above correlation has been demonstrated by comparisons with data in several tubes with several refrigerants.

The acceleration pressure change, as it is most commonly discussed, results from acceleration or deceleration of the flow due to the phase-change process. For evaporation, the flow accelerates resulting in a pressure decrease. For condensation, the flow decelerates along the condensation path resulting in an increase in pressure. If we use appropriately averaged velocities for each phase, and a constant tube-diameter, we can obtain the acceleration pressure drop using a simplified, one-dimensional momentum equation. The resulting equation for a section with inlet properties denoted by the subscript i and outlet properties denoted by the subscript o is:

$$\Delta P_{ACC} = \frac{16\dot{m}^2}{\pi^2 D^4} \left\{ \left[\frac{x_o^2}{\rho_V \alpha_o} + \frac{(1-x_o)^2}{\rho_L (1-\alpha_o)} \right] - \left[\frac{x_i^2}{\rho_V \alpha_i} + \frac{(1-x_i)^2}{\rho_L (1-\alpha_i)} \right] \right\} \tag{14}$$

It is recommended that the void fraction, α, in Eq. (14) be computed using the correlation of Zivi [1964]:

$$\alpha = \frac{1}{1+\left(\frac{1-x}{x}\right)\left(\frac{\rho_V}{\rho_L}\right)^{0.67}} \tag{15}$$

We note from Eq. (14) that the acceleration pressure change is very sensitive to diameter, with a $1/D^4$ dependence. Thus, decreasing the diameter will have a dramatic effect on the acceleration pressure change.

DEVELOPMENT OF A HEAT EXCHANGER MODEL

The correlations presented above are accurate representations of the local behavior of the two-phase heat transfer and pressure drop. Simulating a real heat exchanger, however, requires integrating these quantities over the length as the quality is changed. To allow the correlations presented above to be integrated into a package that could provide realistic answers to design questions, we developed a Microsoft Excel spreadsheet that would accept inputs concerning the operating conditions and compute the length of the various regions of the heat exchanger as output. A spreadsheet was chosen for two main reasons: (1) it allowed iterative calculations to be implemented in a straightforward manner, and (2) it allowed access to a variety of interesting intermediate results without including many printing and formatting statements. To increase its range of usefulness, we developed macro sheets which contained fluid properties for R-134a and obtained all property information in the main spreadsheet by utilizing these add-in functions. This allows the spreadsheet to be utilized for any other refrigerant if a similar macro sheet is available for its fluid properties. We have developed such sheets for R-134a, R-12, R-22, and the 60%/40% azeotrope of R-32 and R-125. Instructions for obtaining and using the spreadsheet will be included in an upcoming technical memo.

The purpose of the spreadsheet is to compute the length of heat exchanger required to accomplish the phase-change process when the operating conditions are specified. The required inputs are:

1. Mass flow rate
2. Inner tube diameter
3. Air temperature
4. Air side thermal resistance per unit length
5. Inlet quality for the evaporator
6. Inlet pressure for condenser, outlet pressure for evaporator

Once the above inputs are made, the length of the heat exchanger can be computed. For the two phase region, Eq. (1) can be written in differential form as:

$$\Delta L = \frac{\delta q * (R'_{REF} + R'_{AIR})}{(T_{REF} - T_{AIR})} \qquad (16)$$

The two-phase region was broken up into quality increments of 5%, and the refrigerant side heat transfer coefficient, pressure drop, and heat transfer resistance were computed at the average quality for each of these elements. This fixed the resistance in the numerator of Eq. (16). The differential heat transfer, δq, was computed by Eq. (17):

$$\delta q = \dot{m}\ h_{LV}\ \Delta x \qquad (17)$$

The refrigerant temperature in the two-phase region was assumed to be equal to the equilibrium saturation temperature based on the pressure. Since the pressure drop for the element was dependent on the length, which was also unknown, iteration was required. This was accomplished with the iteration option in Excel. The result was that for each quality increment, the outlet pressure, outlet saturation temperature, and required length were computed. The refrigerant temperature in Eq. (16) was assumed to be equal to the arithmetic mean of the inlet and outlet values. Summing each of the differential lengths, we find the required length of heat exchanger to perform the phase-change process.

Condensers and evaporators have regions in which both single-phase and two-phase heat transfer occur. The single-phase regions will not be included in the present analysis. For evaporators, the superheat is a small part of the total heat transfer, so this section is expected to occupy relatively little of the evaporator length. For the condenser, on the other hand, the length of the superheated region may be appreciable. However, a substantial portion of the desuperheating section may have wall temperatures below the equilibrium saturation temperature. Both thermodynamic notions and experimental evidence indicate that condensation will occur in this type of situation, although the bulk enthalpy is above that of a saturated vapor. The heat transfer mechanism in this type of situation is very different from that of single-phase vapor, and will be studied in detail in the final stages of Project 01. Until this study is complete, we will offer no design recommendations for this region.

DISCUSSION OF ANALYSIS

To demonstrate the utility of the spreadsheet in addressing a design question, we set out to study the effect of changing the diameter on the required length of evaporator and condenser. Based on the correlations presented above, we suspect that an optimum diameter may exist since decreasing the diameter enhances several of the modes of heat transfer, primarily forced convection in the annular regime, while also dramatically increasing the pressure drop, which reduces the driving temperature difference. This type of situation often results in an optimum value at which the two competing effects are balanced properly.

In reality, the effect of such a change would propagate throughout the system, thus requiring a system simulation to accurately predict the change in performance. For the purposes of this analysis, though, we wanted to isolate the effect to a single component. To accomplish this, yet still impose a penalty for pressure drop, we chose to fix the pressures at the inlet and outlet of the compressor. For the evaporator, then, pressure drop would require a higher inlet pressure and saturation temperature, which translates to a smaller temperature difference. For the condenser, pressure drop would lower the system pressure in the two-phase region and thereby reduce the temperature difference. The other significant assumptions in the analysis were:

1. The air-side resistance per unit length was constant as the tube diameter was changed.
2. Pressure drop in bends was not accounted for.

Two sets of conditions were simulated for both the condenser and evaporator: (1) a high flow rate case that would be applicable for mobile air-conditioners or unitary systems, and (2) a low flow rate case that would be applicable for refrigerators. The test conditions for the condenser and evaporator are listed in Tables 1 and 2, respectively. For each case, the inputs were made to the spreadsheet and the diameter was varied over a wide range. The required length of heat exchanger to complete the phase-change process was computed for each diameter. A lower limit on diameter was eventually reached at which the pressure drop became so great that the temperature difference was nearly eliminated.

Table 1 - Input Parameters for Condensation Simulations.

Quantity	High Flow Rate Case	Low Flow Rate Case
Mass Flow Rate	80 lb/hr	10 lb/hr
Inlet Saturation Temp.	110 °F	100 °F
Air Temperature	90 °F	75 °F
Air Side Resistance/Length	0.073 ft-R-hr/Btu	0.43 ft-R-hr/Btu

Table 2 - Input Parameters for Evaporation Simulations.

Quantity	High Flow Rate Case	Low Flow Rate Case
Mass Flow Rate	80 lb/hr	10 lb/hr
Inlet Quality	20%	20%
Outlet Saturation Temp.	41 °F	-4 °F
Air Temperature	73 °F	14 °F
Air Side Resistance/Length	0.073 ft-R-hr/Btu	0.43 ft-R-hr/Btu

The results of the condensation simulations are presented in Fig. 1 in the form of the required heat exchanger length for the phase-change process as a function of the inside tube diameter. The two sets of curves are for the high and low mass flux cases. The shape of both curves is quite similar. As the inside diameter is decreased, the required length of heat exchanger slowly decreases until eventually the curve takes a sharp upward turn. This can be explained physically as follows. In the part of the curve where the slope is the greatest, the flow regime is primarily annular and the pressure drop is very high. In fact, a substantial portion of the driving temperature difference (on the order of 20% to 80%, changing rapidly with D) is being destroyed due to pressure drop. Since pressure drop is very sensitive to diameter at constant mass flow rate, increasing the diameter even slightly preserves a substantial portion of the temperature difference and reduces the required length. As the slope of the curve decreases, pressure drop is becoming less important so that increasing the diameter has only a modest effect on the temperature difference. In this region, our first guess would be that the required length should <u>increase</u> as diameter increases, since the annular-flow refrigerant side resistance increases with increasing diameter. What happens to prevent this, though, is that the flow regime becomes wavy over a substantial portion of the heat exchanger. In the wavy flow regime, increasing the diameter <u>decreases</u> the thermal resistance, resulting in a slightly shorter heat exchanger. This same trend occurs at both flow rates, as exhibited by the similar shape of the curves. The low mass flow rate curve exhibits substantially less sensitivity to diameter because the refrigerant side resistance is much smaller when compared to the air side resistance.

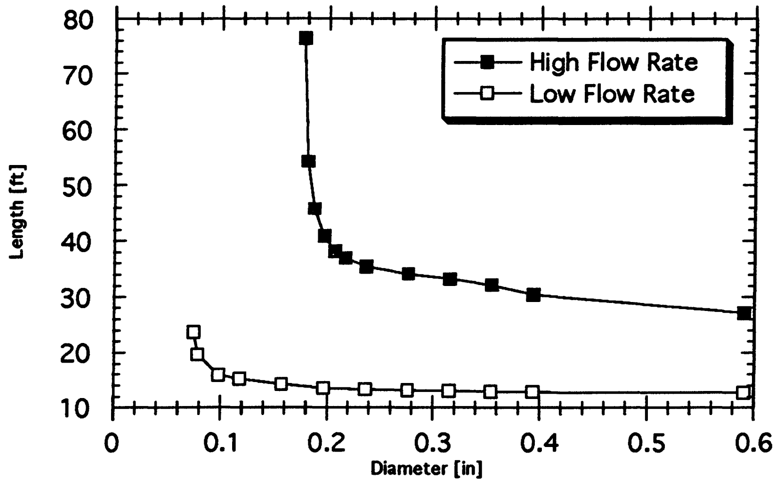

Figure 1 - Effect on inside tube diameter on required condensation length for a low and high mass flow rate case.

If we assumed that annular flow prevailed regardless of the flow rate and diameter, as just discussed, we would expect the required length to eventually increase as diameter was increased. Although this type of situation is impossible from a physical standpoint, it is quite important since it corresponds to the situation that one would predict when using supposedly generalized heat transfer correlations such as the one due to M.M. Shah [1979]. These correlations, although compared to a large amount of data in various flow regimes, reflect the physics of annular flow, forced convective condensation only. They do not account for the possibility of gravity-driven, Nusselt-type condensation. To study this situation, we ran the simulation for the high mass flow rate case using the annular flow heat transfer correlation, Eq. (4), no matter how large the diameter became. The results are compared with the correct method in Fig. 2. The results are as expected, with the required length beginning to increase at a diameter of approximately 0.23 inches. This shows that accounting for the change in flow regime has important effects on the results, since the use of annular flow correlations only falsely implies the existence of an optimum diameter and reduced design flexibility.

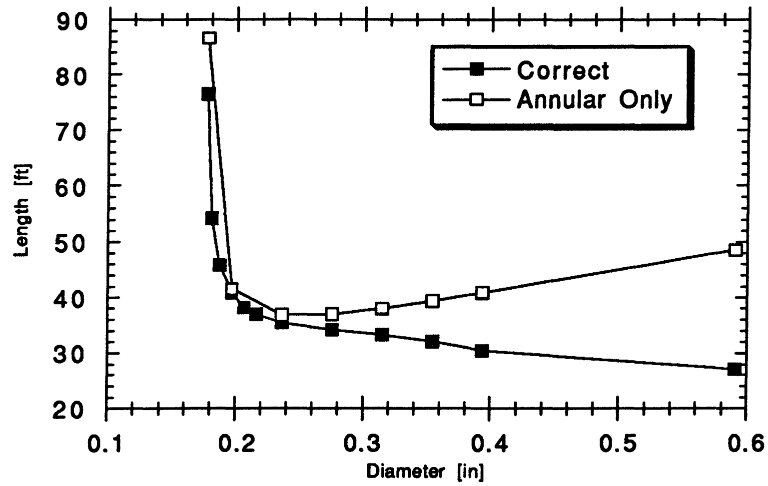

Figure 2 - Effect of inside diameter on condensation length when using annular flow correlations only as compared to accounting for the flow regime correctly.

Although the desuperheating region was not simulated, we can offer some qualitative insights into how it would affect the results. For single-phase heat transfer, the Nusselt number is proportional to $1/D^{0.8}$, identical to the case for annular flow. Thus, decreasing the diameter will always decrease the required length of the desuperheating region. When this effect is considered in conjunction with the two-phase region, the result will, at a minimum, decrease the negative slope of the L versus D curves in Fig. 2 and Fig. 3. Depending on the relative lengths of the desuperheating and two-phase regions, it is possible that the inclusion of the desuperheating region could even make the required length increase above some diameter. This effect will be most noticeable in the region where the length is relatively insensitive to diameter, and is unlikely to substantially change the diameter at which the length begins to increase dramatically. These same qualitative trends will hold for the subcooled liquid region, since the same type of heat transfer correlations are applicable.

The results of the evaporation simulations are presented in Fig. 3 which again show the required heat exchanger length for the phase-change process as a function of the inside tube diameter. Both high and low mass flux cases are shown. For decreasing inside tube diameter, the required length of heat exchanger slowly increases until eventually the curve takes a sharp upward turn. As inside tube diameter is decreased, both the heat transfer coefficient and pressure drop in the evaporator increase. For larger tube diameters, the pressure drop in the evaporator remains relatively low and does not destroy much of the temperature difference between the refrigerant stream and the air stream. Therefore, the length of the heat exchanger to transfer the given heat load remains relatively constant, increasing only marginally. When the inside tube diameter is further decreased to a critical value, the pressure drop rise eventually becomes large enough to destroy a substantial portion of the temperature difference between the refrigerant and air stream and the length of the evaporator necessary to transfer the required heat increases substantially.

The minimum inside diameter for the low flow rate case is smaller than for the high flow rate case, as can be seen in Fig. 3. The refrigerant-side resistance is much smaller compared to the air side resistance for the low flow rate case than the high flow rate case. In addition, the pressure drop in the evaporator is highly dependent on mass flow rate and is substantially reduced for a wavy flow pattern, which is the predominant flow pattern for the low flow rate case. Pressure drop is much higher for the annular flow pattern found in the high mass flow rate case. Both of these keep the temperature driving potential higher for the low flow rate case than the high flow rate case for the same diameter. This results in a lower inside tube diameter for the sharp upward turn of the curve in Fig. 3 for the low flow rate case.

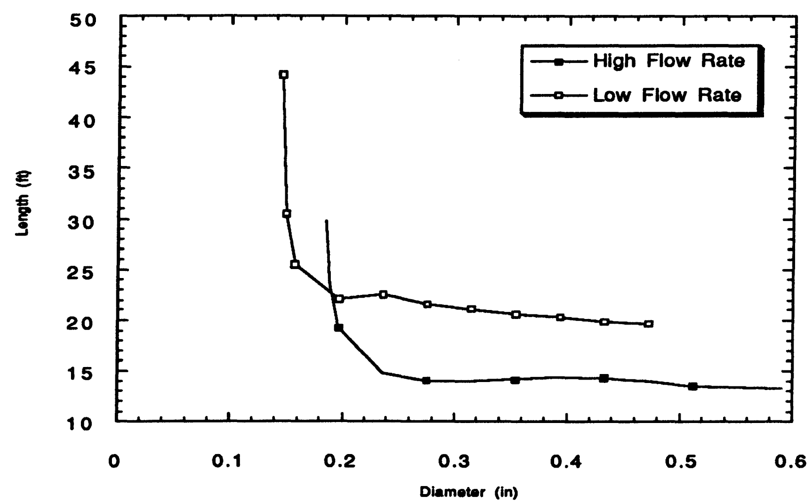

Figure 3 - Effect of inside tube diameter on required evaporation length for a low and high mass flow rate case.

An additional comment also needs to be made regarding the optimal values of inside tube diameter. For the high flow rate case found in stationary air conditioning evaporators, the optimal value for the evaporator inside tube diameter is about 0.3 inches. For the low flow rate case found in household refrigerator evaporators, the optimal value is about 0.23 inches. Both values appear to be close to what industry is driving towards. It should be noted that these values will increase slightly for the proper inclusion of bend pressure drop, which was not accounted for in this simulation.

SUMMARY AND CONCLUSIONS

Two-phase heat transfer and pressure drop correlations developed in the ACRC have been combined into a spreadsheet which allows simulation of the two-phase portion of evaporators and condensers. The spreadsheet allows simulation with four refrigerants: R-12, R-134a, R-22, and a 60%/40%, azeotropic mixture of R-32 and R-125. The spreadsheet was used to study the effect of altering the diameter of heat exchangers while keeping the air-side resistance and refrigerant mass flow rate constant. Simulations were performed for both a high and a low flow rate for condensers and evaporators. The results of all the simulations were similar. The required length of heat exchanger slowly decreased as the tube diameter was decreased over a wide range. As the diameter became sufficiently small, though, the required length of heat exchanger began to increase greatly as the diameter was decreased further. This occurred when a substantial portion of the driving temperature difference was being destroyed by pressure drop. This "critical diameter" was smaller for the lower flow rate cases than for the high flow rate cases because of decreased pressure drop. Utilizing supposedly generalized correlations in the wavy flow regime can lead to significant errors in the required length of heat exchanger.

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