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THE EFFECT OF VOID FRACTION CORRELATION AND HEAT FLUX ASSUMPTION ON REFRIGERANT CHARGE INVENTORY PREDICTIONS

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ABSTRACT

Ten void fraction correlations and four heat flux assumptions are evaluated for their effect on refrigerant charge inventory predictions. Comparisons between mass inventory predictions are made for condensers and evaporators over representative heat pump operating ranges of saturation temperature, mass quality, and mass flux. The choice of void fraction model is found to have a major effect on refrigerant inventory prediction. The maximum variation of predictions ranges from a factor of 10 for low-ambient, heating-mode evaporators to 4.2 for cooling-mode evaporators and 1.7 for high-ambient cooling-mode condensers assuming no subcooling. The correlations of Hughmark, Premoli, Tandon, and Baroczy are found to give the highest predictions and closest agreement to measured total system charge. The choice of heat flux assumption is shown to be insignificant for forced-flow evaporators and of secondary to possibly equal importance to choice of void fraction model for condensers. Implications for charge balancing, off-design and transient performance prediction, and unit reliability are discussed.

INTRODUCTION

Analytical prediction of the refrigerant charge inventory in a heat pump is a potentially valuable aid in system design that has received limited attention to date. Most heat pump systems are charge-sensitive in the sense that the off-design performance is determined, to some degree, by the amount of total charge in the unit. Finding the charge that gives good operation at both design and off-design conditions has been largely a trial-and-error experimental evaluation. The capability to model such off-design effects analytically opens the way to search for more optimal refrigerant charge/flow control balances as a part of the initial system configuration studies rather than after a given hardware design has been selected. This capability also provides a design tool for determining ways to minimize total system refrigerant charge. As noted by Bonne et al. [1980], reductions in total unit charge should improve system cycling performance. Improved recovery rates from defrost cycle reversal are also expected benefits, as well as improved compressor reliability from minimization of compressor slugging conditions [Moore 1978].

A major difficulty in charge inventory analysis is proper prediction of the refrigerant mass in the two-phase regions of the condenser and the evaporator. This is because of two basic uncertainties: the degree of vapor-to-liquid slip at each cross section in the two-phase region, and the variation of refrigerant quality with length through the two-phase region.

The simplest approach is to assume zero slip (a slip ratio of 1) and constant heat flux (quality varying linearly with length) in the two-phase region. Stoecker et al. [1981] followed this approach with the further simplification of ignoring the vapor contribution. Other modelers assuming zero slip include James and Marshall [1973], Daniels and Davies

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[1975], Dhar and Soedel [1979], Bonne et al. [1980; *], Farr,[†] and MacArthur [1984]. These investigators used a variety of constant and discretely variable heat flux approaches in their heat exchanger analyses. Other researchers [Otaki 1973; Otaki and Yoshii 1975; Rigot 1973; Ahrens 1983] have included slip effects but assumed constant heat flux. Domanski and Didion [1983] included slip effects and computed the two-phase region in a tube-by-tube manner, thus having discretely varying heat flux. Only Domanski and Otaki included comparisons of analytical charge inventory results to experiment. Domanski found significant differences in both condenser and total inventory comparisons, while Otaki's results were generally within $\pm 10\%$ of measured total charge for a variety of vapor compression machines.

The major emphasis of this paper is to survey and compare a wide variety of slip ratio models (more generally represented by void fraction correlations) and four different heat flux variation assumptions. Both void fraction correlations developed for annular flow and those proposed as reasonably accurate without regard to flow regime are included. A brief review of each correlation from the author's perspective is provided followed by comparisons of their predictions for cases of direct application to heat pump charge inventory analysis. This is done with regard to the effect on refrigerant mass predictions over the two-phase region of condensers and evaporators operating at representative conditions.

The basic equations for charge inventory calculation are given first for both single-phase and two-phase regions of a heat exchanger. The relationship of void fraction correlation and heat flux assumption to these equations is developed here. The various void fraction correlations to be considered are presented next along with sample predicted profiles vs. mass quality. The specifics of the four considered heat flux assumptions are then developed followed by total two-phase mass inventory predictions for the various correlations for a constant heat flux. The effect of variable heat flux on the predictions is examined and compared to the influence of void fraction correlation. The results are compared with available charge inventory measurements. Last, possible implications of the range of predictions and applications of the results are discussed with regard to system charge inventory control and off-design and transient performance prediction.

BASIC EQUATIONS

Single-Phase Refrigerant Sections

The single-phase refrigerant mass, m, contained in a length of tubing, L, of cross-sectional area A_c and total volume, V is given by

$$m = \int_0^L \rho \cdot dV = A_c \int_0^L \rho \cdot dl \quad , \tag{1}$$

where ρ is the local single-phase refrigerant density along the tube. Equation 1 can be rewritten as

$$m = A_c L \frac{\int_0^L \rho \cdot dl}{\int_0^L dl} = V \cdot \rho_{ave} \quad ,$$

where ρ_{ave} is a suitably averaged[‡] refrigerant density over the tube length. These equations would be used to calculate the refrigerant mass in the subcooled liquid or superheated vapor sections of a heat exchanger.

*Ulrich Bonne, Honeywell, Inc., Corporate Technology Center, Bloomington, MN, personal communication, 1982.

†Richard A. Farr, R&R Supply Company, Inc., Columbus, OH, personal communication, 1983.

‡The choice of averaging method used for nonisothermal single-phase regions is of secondary importance when compared to the uncertainties in the two-phase calculations.

Two-Phase Refrigerant Sections

The two-phase refrigerant mass contained in a length of tubing is obtained by summing the gas g and liquid f contributions occupying each cross-sectional area over the length of the region. These contributions are given separately by:

$$m_g = \int_0^L \rho_g \cdot dV_g = \rho_g \int_0^L A_g \cdot dl \ , \tag{2}$$

and

$$m_f = \int_0^L \rho_f \cdot dV_f = \rho_f \int_0^L A_f \cdot dl \ , \tag{3}$$

where

A_g = cross-sectional area occupied by vapor,
A_f = cross-sectional area occupied by liquid, and
A_c = $A_g + A_f$ at each cross section.

Relationship to Void Fraction

Introducing the void fraction $\alpha = A_g/A_c$, Equations 2 and 3 can be rewritten as:

$$m_g = \rho_g A_c \int_0^L \alpha \cdot dl \ , \tag{4}$$

and

$$m_f = \rho_f A_c \int_0^L (1 - \alpha) \cdot dl \ . \tag{5}$$

The total mass, m_t, in the two-phase section can be obtained from Equations 4 and 5 in terms of tube volume V as:

$$m_t = V \cdot \left[\rho_g \int_0^L \alpha \cdot dl + \rho_f \int_0^L (1 - \alpha) \cdot dl\right] / \int_0^L dl \ . \tag{6}$$

Density Weighting Factors

The void fraction α is generally represented as some function of refrigerant quality, x. Therefore, to evaluate m_t from Equation 6 for a given void fraction equation, i.e.,

$$\alpha = f_\alpha(x) \ , \tag{7}$$

the tube length variable, l, must be related to mass quality, x, in some manner. This relationship is obtained from an assumption regarding the heat flow variation, dQ, with differential length, dl, in the two-phase region, that is,

$$dQ = \dot{m}_r h_{fg} dx = f_Q(x) \cdot dl \ , \tag{8}$$

where

$\dot{m}_r$ = refrigerant mass flow rate,
h_{fg} = enthalpy of vaporization, and
$f_Q(x)$ = assumed heat flux equation (i.e., equation for local heat flow per differential length*).

*Because the heat flow per differential length is functionally equivalent to the heat flux dependence in a tube assuming no radial dependence, *heat flux* variation will be used herein instead of local heat flow per unit length.

In terms of the representations $f_\alpha(x)$ and $f_Q(x)$ given by Equations 7 and 8, Equation 6 can be rewritten as:

$$m_t = V\cdot\left[\rho_g W_g + \rho_f(1 - W_g)\right] , \tag{9a}$$

where

$$W_g = \frac{\int_{x_i}^{x_o} f_\alpha(x)/f_Q(x)\,dx}{\int_{x_i}^{x_o} 1/f_Q(x)\,dx} , \tag{9b}$$

and x_i and x_o are inlet and outlet refrigerant qualities. The normalized integral W_g is the refrigerant gas density weighting factor.* Thus the evaluation of the two-phase refrigerant mass is reduced to the problem of evaluating the integrals given in Equation 9b for selected void fraction correlations [$f_\alpha(x)$'s] and heat flux assumptions [$f_Q(x)$'s].

Since the refrigerant liquid term is the major contributor to the total mass in the heat exchanger, it is convenient for discussion purposes to rewrite Equations 9a and 9b in terms of the refrigerant liquid density weighting factor, W_f, i.e.,

$$m_t = V\left[\rho_f W_f + \rho_g(1 - W_f)\right] , \tag{10a}$$

where

$$W_f = \frac{\int_{x_i}^{x_o} f_{1-\alpha}(x)/f_Q(x)\,dx}{\int_{x_i}^{x_o} 1/f_Q(x)\,dx} , \tag{10b}$$

and $f_{1-\alpha}(x) = 1 - f_\alpha(x) =$ liquid fraction (or holdup) correlation.

Equation 9a or 10a thus gives the total refrigerant mass in the two-phase section of a heat exchanger.

VOID FRACTION REPRESENTATIONS

The void fraction is generally represented as a function of mass quality, x, and combinations of various types of property indexes (which remain constant for a given average evaporator or condenser saturation temperature). Only in a few cases have dependences on mass flow rate been correlated. Existing correlations reviewed by the author were classified into four categories:

— homogeneous,
— slip-ratio-correlated,
— X_{tt}-correlated, and
— mass-flux-dependent.

Homogeneous

The homogeneous model is the most simplified. The model considers the two phases as a homogeneous mixture, thereby traveling at the same velocity. In this model, the relationship between void fraction, α, and mass quality, x, is straightforwardly derived as:

$$\alpha = \frac{1}{1 + \left(\frac{1 - x}{x}\right) P.I._1} , \tag{11}$$

*W_g can alternatively be described as the heat-flux-averaged void fraction over a given mass quality range.

where the property index $P.I._1 = \rho_g/\rho_f$. This formulation has been used by James [1973], Daniels [1975], Dhar [1979], Stoecker [1981], Bonne [1980], and MacArthur [1984].

Slip-Ratio-Correlated

A slightly more involved approach is to assume that the liquid and vapor phases are separated into two streams that flow through the tubes with different velocities, u_g and u_f, the ratio of which is given by the slip ratio $S = u_g/u_f$. A modified form of Equation 11 has been used to include this effect, i.e.,

$$\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right)\frac{\rho_g}{\rho_f}\cdot S} = \frac{1}{1 + \left(\frac{1-x}{x}\right) P.I._1 \cdot S} , \tag{12}$$

where S is estimated differently by various investigators.

Rigot and Ahrens/Thom. Rigot [1973] suggested using an average value of 2 for slip ratio for his intended application. Ahrens [1983] recommended use of the steam/water data of Thom [1964] suitably generalized by the property index $P.I._2$ given by:

$$P.I._2 = \left(\frac{\mu_f}{\mu_g}\right)^{0.2} \cdot \frac{\rho_g}{\rho_f} = \left(\frac{\mu_f}{\mu_g}\right)^{0.2} \cdot P.I._1 . \tag{13}$$

In the Ahrens/Thom method, the slip ratio, S, is, in effect, dependent on the refrigerant operating pressure only and is thus independent of quality. The Thom method was developed along the same lines as the more well known Martinelli-Nelson [1948] approach for steam/water systems under boiling conditions. The Thom method represents a more extensive set of void fraction data and as such should be the preferred choice. Slip ratio values generalized from the Thom method in terms of $P.I._2$ ~~method~~ are given in Table 1. In Table 2, corresponding values of various property ratios including $P.I._2$ are given for an appropriate range of R-22 saturation temperatures. From Tables 1 and 2, it can be seen that in the Thom method, the slip ratio is predicted to range from about 1.5 at high condensing temperatures to about 2.5 at low evaporating temperatures.

Zivi. Zivi [1964] developed a void fraction equation similar in form to Equation 12 where S is given by:

$$S = \left(P.I._1\right)^{-1/3} . \tag{14}$$

This relationship was developed for annular flow based on principles of minimum entropy production under conditions of zero wall friction and zero liquid entrainment (100% liquid entrainment gives a slip ratio of 1).

Smith. Smith [1969] developed a correlation based on equal velocity heads of a homogeneous mixture center and an annular liquid phase. He obtained an equation for slip ratio S, dependent on the density ratio $P.I._1$, mass quality, and entrainment ratio K given by:*

$$S = K + (1 - K)\left[\frac{1/P.I._1 + K\left(\frac{1-x}{x}\right)}{1 + K\left(\frac{1-x}{x}\right)}\right]^{1/2} , \tag{15}$$

and where $K = 0.4$ was found to correlate well with the three sets of experimental data considered.

In summary, the slip-ratio-correlated equations all use Equation 12 for void fraction with the vapor slip ratio S given alternatively by

$S = 2,$ Rigot [1973],

*Zivi also developed more involved slip equations dependent on entrainment level and mass quality but did not attempt a general correlation to existing data.

or

$$S = f(P.I._2) \text{ in Table 1,} \qquad \text{Ahrens/Thom [1983],}$$

or

$$S = (P.I._1)^{-1/3}, \qquad \text{Zivi [1964],}$$

or

$$S = f(P.I._1, x), \text{ Equation 15,} \qquad \text{Smith [1969].}$$

X_{tt}-Correlated

Another group of correlations avoids the use of a form of the homogeneous equation by employing the Lockhart-Martinelli (L-M) correlating parameter X_{tt} defined as:

$$X_{tt} = \left(\frac{1 - x}{x}\right)^{0.9} P.I._2^{0.5} \ . \tag{16}$$

Lockhart-Martinelli. The well-known early L-M pressure drop work [1949] presented void fraction data as a function of X_{tt} on two-phase/two-component adiabatic flows near atmospheric conditions. These data were approximated by equations developed by Wallis [1969] and refined by Domanski and Didion [1983] for $X_{tt} > 10$. The equations are:

$$\alpha = f(X_{tt})$$

$$= (1 + X_{tt}^{0.8})^{-0.378} \qquad \text{for } X_{tt} \leqslant 10 \ , \tag{17}$$

$$= 0.823 - 0.157 \ln X_{tt} \qquad \text{for } X_{tt} > 10 \ . \tag{18}$$

These equations were used by Domanski [1983] in a charge inventory model for heat pump simulation.

Baroczy. A second general X_{tt} correlation is that developed by Baroczy [1965]. He added a direct functional dependence on the property index $P.I._2$, i.e., $\alpha = f(X_{tt}, P.I._2)$, beyond that already included in the L-M correlating parameter, X_{tt}, as given by Equation 16. This correlation was developed over a wider range of conditions than the L-M representation and like L-M is easily applied to different fluids through the property index formulation. The tabular representation of the Baroczy method in terms of liquid fraction $(1 - \alpha)$ is given in Table 3.

Baroczy [1966] noted in his comparisons with experimental data [Staub and Zuber 1964] that there was also an apparent mass flux effect that was not accounted for in his correlation. The observed effect of increased mass flux was to increase void fraction values at any given quality. The remaining methods to be surveyed include a mass flux effect.

Mass-Flux-Dependent

The mass-flux-dependent methods include one physically based model for annular flow [Tandon et al. 1985] and two empirically based correlations [Hughmark 1962; Premoli 1971].

Tandon. The model developed by Tandon et al. [1985] is an improvement for annular flow over the Zivi [1964] method in that the effect of wall friction is included. The Tandon method predicts void fraction results close to those of Smith [1969] yet does include a small mass flux effect. The correlation is of the form

$$\alpha = f(\mathrm{Re}_L, X_{tt}) \ ,$$

where Re_L is the liquid Reynolds number. The full equations are given in Appendix A for reference.

Premoli. The empirical correlation developed by Premoli et al. [1971] is of interest because it was optimized to minimize liquid density prediction errors.* Since density differences directly relate to refrigerant mass predictions, the

*The intended purpose was for determining reactivity effects of two-phase coolant in nuclear reactors.

approach could be well suited for the intended application. The model is developed in terms of the modified homogeneous equation (Equation 12) where the slip ratio, S, is represented by

$$S = f(x, P.I._1, \mathrm{Re}_L, We) ,$$

where We is the Weber number (dependent on mass flux, diameter, and surface tension). The complete correlation is also given in Appendix A. The Premoli correlation was developed for a large variety of conditions for two-phase mixtures flowing upwardly in vertical adiabatic channels.

Hughmark. The empirical Hughmark [1962] correlation is a generalization of the work of Bankoff [1960], which assumed a bubble flow regime with a radial gradient of bubbles across the channel. Hughmark was concerned with prediction of liquid fraction (holdup), $(1 - \alpha)$, in the pipes of oil refineries.

Although developed for vertical upward flow with air-liquid systems near atmospheric pressure, the correlation was found by Hughmark to do equally well for horizontal flow, for much higher pressures, and for other flow regimes. This approach has been used by Otaki [1973] and Otaki and Yoshii [1975] for the prediction of the refrigerant charge in refrigerating, air-conditioning, and heat pump systems.

In the correlation, void fraction is given by a correction factor K_H to the homogeneous equation, i.e.,

$$\alpha = \frac{K_H}{1 + \left(\frac{1 - x}{x}\right) P.I._1} , \tag{19}$$

where $K_H = f(Z)$ given in Appendix A, and Z (also given in Appendix A) is dependent on a viscosity-averaged Reynolds number, the Froude number, and the liquid volume fraction, i.e.,

$$Z = f(x, P.I._1, G, D_i, \mu_f, \mu_g, \alpha)$$

where

G = mass flux,
D_i = tube inside diameter,
μ = kinematic viscosity.

Since Z contains a dependence on void fraction α through the averaged Reynolds number, Equation 19 must be iteratively evaluated to obtain the void fraction at each refrigerant quality.

Liquid Fraction (Holdup) Comparisons

Liquid fraction results are shown in Figures 1 and 2 as a function of refrigerant quality for a sampling of the discussed void fraction methods. The results were generated for refrigerant saturation temperatures representing a low-ambient heating-mode evaporator condition [5 F (−15°C)] and a high-ambient cooling-mode condenser condition [130 F (54.4°C)]. Liquid fraction is shown rather than void fraction, since the refrigerant mass contained in the two-phase section of a coil is overwhelmingly determined by the liquid distribution (due to the order-of-magnitude density difference).

From the comparisons, it is seen that the more involved models generally predict a larger liquid presence than the homogeneous baseline. The only exception is at qualities less than 0.2 for the high-temperature, condensing case where the Lockhart-Martinelli model predicts liquid fraction values lower than for the homogeneous case. This implies predicted slip ratios of less than one for the L-M method at low qualities.

Comparison of the general levels between Figures 1 and 2 shows that at higher saturation pressures the heat exchanger mass quality range is occupied by more liquid. This is because of an increase in the gas-to-liquid density ratio ($P.I._1$) which decreases void fraction at a given quality level [from Equations 11 and 12].

The curves shown in Figures 1 and 2 represent the $f_{1-\alpha}(x)$ correlations to be used in Equation 10b along with heat flux weighting factors, $f_Q(x)$'s, to obtain the two-phase mass inventory. Various choices of heat flux weighting factors to be applied to these curves are described next.

TWO-PHASE HEAT FLUX ASSUMPTIONS

Constant Heat Flux

By far the most common heat flux assumption used is that of a constant heat flux [Otaki 1973; Rigot 1973; Otaki and Yoshii 1975; Stoecker 1981; Ahrens 1983]. From Equation 7, this assumption gives

$$dQ = \dot{m}_r h_{fg} dx = f_Q(x) \cdot dl = \text{constant} \cdot dl ,$$

i.e., a linear relationship between quality and tube length. Since $f_Q(x) = \text{constant}$, each increment of refrigerant quality is weighted equally in averaging the liquid fraction, given by $f_{1-\alpha}(x)$, in Equation 10b.

Constant Tube Wall Temperature

A second possible assumption is that of constant tube wall temperature along the tube length. Here

$$dQ = [U_R(x) \Delta T_{ref-tube} \pi D_i] \cdot dl , \tag{20}$$

where $f_Q(x)$ can be given simply* by $U_R(x)$, the local refrigerant-side heat transfer coefficient. This approach was used by Hiller and Glicksman [1976] in obtaining average heat transfer coefficients for two-phase regions of condensers and evaporators.

Constant Average Air-to-Refrigerant ΔT

A third approach analyzed was to assume a constant average air-to-refrigerant ΔT along and across the tube length.† By analogy with Equation 20, this sets $f_Q(x)$ to $U_T(x)$, the local overall air-to-refrigerant heat transfer coefficient.

Variable Air-to-Refrigerant ΔT (With Constant Entering Air Temperature)

The effectiveness/NTU equation [Kays and London 1964] for a heat exchanger with one fluid at constant temperature can be used to more accurately represent (under certain conditions) the air temperature change across a single tube or parallel bank of tubes at each given tube cross section. For a uniform entering air temperature $t_{air,in}$ along the tube length, the equation is:

$$dQ = [1 - e^{-\text{NTU}(x)}](t_r - t_{air,in}) C_{air} \cdot \frac{dl}{L} , \tag{21}$$

where

C_{air} = air capacity rate,
$\text{NTU}(x)$ = $U_T(x) \cdot A/C_{air}$, and
A = heat transfer surface area over which U_T and C_{air} apply.

Elimination of the constant terms gives

$$f_Q(x) = 1 - e^{-\text{NTU}(x)} .$$

Because of the variety of complex circuiting arrangements possible, it is difficult to generalize as to which of the above methods is more representative when applied to the two-phase region of a heat exchanger as a whole. For certain

*This is because the term $\Delta T_{\text{ref-tube}}/L$ is constant for constant tube wall and refrigerant temperatures and the constant terms cancel out in Equation 10b. Similarly, $f_Q(x)$ can be given by 1.0 rather than a specific constant value for the constant heat flux case.

†Since the refrigerant temperature is constant, this implies the use of a constant average air temperature for the analysis.

simplified heat exchanger circuiting assumptions [Hiller and Glicksman 1976], Equation 21 is the more exact treatment. However, for circuiting arrangements where the tubes are serpentine in the airflow direction, Equation 21 is only appropriate when applied one tube at a time with the proper average value of $t_{air,in}$ used for each tube. The approaches used by Daniels and Davies [1975] and Domanski and Didion [1983] approximate this treatment by dividing the two-phase region of the heat exchanger into sections or tube lengths and applying Equation 21 with an average NTU to each section.

In summary, the four heat flux assumptions to be considered result in heat flux weighting factors $f_Q(x)$ of:

1. $f_Q = 1$ for constant heat flux (Q = constant),
2. $f_Q = U_R(x)$ for a constant wall temperature,
3. $f_Q = U_T(x)$ for a constant air-to-refrigerant ΔT, or
4. $f_Q = 1 - e^{-\mathrm{NTU}(x)}$ for a variable air-to-refrigerant ΔT ($Q \propto [1 - e^{-\mathrm{NTU}(x)}]$) ,

for use in either Equation 9b or 10b to compute integrated gas or liquid density weighting factors, respectively.

The latter three heat flux assumptions are attempts to account, in varying degrees, for the expectation that proportionally more of the heat exchanger length (in the two-phase region) will be required for those sections that have the lower refrigerant-side heat transfer coefficients. The application of these four heat flux assmptions show the range of possible effects from a uniform weighting to a weighting inversely proportional to the local value of U_R. Good heat exchanger design practice is expected to keep the range of possible variation within these bounds and more toward the constant-heat-flux-assumption end of the spectrum.

MASS INVENTORY PREDICTIONS

Constant Heat Flux, Quality Range of 0 to 1

The results to be shown for mass inventory predictions are given in the general terms of average two-phase refrigerant density, ρ_{TP}, which is directly proportional to the total refrigerant mass in the two-phase section of the heat exchanger, i.e.,

$$\rho_{TP} = m_T/V \ , \tag{22}$$

where the constant of proportionality is $1/V$, the inverse of the heat exchanger volume in the two-phase section. From Equation 10a, ρ_{TP} is given by:

$$\rho_{TP} = \rho_f W_f + \rho_g(1 - W_f) \ , \tag{23}$$

where W_f is the liquid density weighting factor. Because of the constant proportionality, average R-22 density and refrigerant mass inventory are used synonymously in the following discussion.

Cumulative Values. In Figure 3, the cumulative amounts of liquid, vapor, and total mass are shown for a condenser condition for the two limiting* void fraction cases—homogeneous and Hughmark for a low G value. The homogeneous method predicts about 50% as much total mass as Hughmark with nearly 60% of that mass in the 0.0 to 0.30 mass quality section of the heat exchanger. In contrast, Hughmark has a much larger liquid mass contribution above 30% quality. Overall, liquid accounts for about 70% of the homogeneous total mass as compared to nearly 90% in the Hughmark case.

Mass-Flux-Independent Comparisons. In the next series of figures, the total R-22 inventory values predicted over the quality range of 0.0 to 1.0 for the ten void fraction methods are compared. This is done for a range of saturation temperatures and flow rates under a constant heat flux condition. The saturation temperature range chosen covers the region of most application to heat pumps.

*Limiting based on total mass predictions as shown later.

The first comparisons in Figure 4 include the seven mass-flux-*independent* methods. For reference, the curves representing pure R-22 vapor and liquid densities are included as lower and upper boundaries. The Baroczy and Zivi methods are the highest-predicting of the group (and are nearly identical) giving low-temperature evaporator densities more than 2.6 times larger than the homogeneous case. Condenser densities are predicted 40% to 50% larger than homogeneous. The Smith prediction gives slightly lower values than the first two but with the same approximate slope as all the methods except for Rigot and Lockhart-Martinelli. The Rigot method of a constant slip ratio of 2 matches the Thom method at 40 F (4.4°C)* but overpredicts compared to Thom at 130 F (54.4°C) where a slip ratio of only 1.44 is predicted.

The Lockhart-Martinelli method is seen to approach the homogeneous results at high condenser temperatures. This follows from the Lockhart-Martinelli lower and then higher predictions of liquid fraction values (as compared to the homogeneous), as shown earlier in Figure 2. The Thom method more closely represents the alternative Martinelli-Nelson [1948] approach, which contained interpolated curves more appropriate for such high-pressure applications than those of Lockhart-Martinelli.

Mass-Flux-Dependent Comparisons. The ranges of mass inventory predictions for the three mass-flux-*dependent* models are shown in Figure 5. The regions shown for each method are bounded by selected lower and upper limits of mass flux, G, for conceivable ranges of heat pump operation and coil circuiting. These values of G and corresponding inner diameter, D_i, values were set at:

$$\text{low } G = 2 \times 10^4 \text{ lbm/h}\cdot\text{ft}^2 \, (2.712 \times 10^1 \text{ kg/s}\cdot\text{m}^2)$$
$$D_i = 0.0458 \text{ ft } (14.0 \text{ mm})$$

$$\text{mid } G = 1 \times 10^5 \text{ lbm/h}\cdot\text{ft}^2 \, (1.356 \times 10^2 \text{ kg/s}\cdot\text{m}^2)$$
$$D_i = 0.0367 \text{ ft } (11.2 \text{ mm})$$

$$\text{high } G = 5 \times 10^5 \text{ lbm/h}\cdot\text{ft}^2 \, (6.781 \times 10^2 \text{ kg/s}\cdot\text{m}^2)$$
$$D_i = 0.258 \text{ ft } (7.86 \text{ mm})$$

The results for the Lockhart-Martinelli and homogeneous methods from Figure 4 are repeated as reference lines. Although not shown in Figure 5, the Zivi line from Figure 4 cuts diagonally across the Tandon region in Figure 5 from just below the lower left corner to just above the upper rightmost corner.

The Tandon and Premoli methods are seen to give similar predictions for evaporator conditions with the Premoli method consistently predicting higher values for condenser cases. The Hughmark method only agrees with the other two mass-flux-dependent correlations at conditions of low evaporator temperature and high mass flux; elsewhere, the Hughmark predictions are always higher—increasing in difference as the mass flux is lowered.

Summary Comparison of Methods. The various two-phase density predictions for the case of constant heat flux and a quality range of 0 to 1 are summarized in Figure 6. Here the Hughmark predictions are shown in more detail and form the upper boundary of the density predictions. The mid-G Hughmark line is the most representative of average heat pump flow rates and as such is the best line for comparison to the mass-flux-independent curves in general. However, heat pump operation does tend to approach the low G values at low evaporator temperatures, which is toward the direction of highest density prediction difference. Figure 6 once again emphasizes the disparate slope of the Lockhart-Martinelli line. It is also of interest to note that the Zivi and especially the Baroczy results from Figure 4 coincide closely with the Hughmark curve for high mass flux.

Constant Heat Flux, Evaporator Quality of 0.2 to 1.0

Summary Comparison of Methods. The curves shown in Figure 6 are repeated in Figure 7 with only the range of integration changed for the evaporator conditions. For heat pump operation, the evaporator inlet quality, rather than 0.0, is typically around 0.2 after isenthalpic expansion from condenser exit conditions. The effect of this adjustment is to further widen the range of predicted differences. The low-G Hughmark case is a factor of 10 above the homogeneous case at the lowest evaporator temperature,† a factor of 6 above the Thom method, and 2.7 times that of Lockhart-Martinelli, whereas before in Figure 6 the ratios were 5, 3, and 2, respectively.

*As can be checked using Tables 1 and 2.

† Where the low-G condition is most likely.

For typical cooling mode evaporator conditions of 45 F (7.2°C), the mass predictions of the more likely, mid-G Hughmark line are ratios of 4.2, 2.9, and 1.8 over the homogeneous, Thom, and Lockhart-Martinelli methods, respectively. For high-temperature condenser conditions, the range of predictive differences narrows considerably where the mid-G Hughmark ratios are 1.7, 1.4, and 1.6 over the homogeneous, Thom, and Lockhart-Martinelli methods, respectively.

In Figure 8, the remaining void fraction correlations of interest are shown for the evaporator quality range of 0.2 to 1.0. The Hughmark, Premoli, and Tandon methods are given for mid-G values for the most representative comparison to the Baroczy and Smith methods.

The wide range of predicted evaporator densities shown in Figures 7 and 8 results in a similar variation in the predicted proportions of mass to be found in the condenser relative to the evaporator. The ratios predicted by the various methods for low and moderate ambient heating conditions and a high-ambient cooling condition are given in Table 4. The condenser-to-evaporator mass ratios predicted range from a high of 7.46 to a low of 1.82 for the low-ambient heating case with the range reducing to 4.95 to 1.74 for the high-ambient cooling case. The three mass-flux-dependent methods along with the L-M method show the least change in ratio with change in operating conditions.

The curves in Figures 7 and 8 also show additional points worthy of mention. First, comparison of Figure 7 to Figure 4 shows the Zivi and the Lockhart-Martinelli predictions switch order for evaporator inlet qualities of 0.2. The Zivi method apparently obtains relatively more mass from the 0.0 to 0.2 quality range. The Lockhart-Martinelli and Thom (approximating Martinelli-Nelson) methods become noticeably more dissimilar in evaporator predictions than in Figure 4. The Zivi and Baroczy methods show a similar widening of difference for evaporator conditions (comparing Figures 7 and 8 to Figure 4).

Figures 7 and 8 provide an overview of the predictive variation of the ten considered void fraction models for possible ranges of heat pump conditions with the assumption of *constant heat flux*. The relative effect of the second factor to be considered—*choice of heat flux assumption*—is shown next.

Heat Flux Weighting Factors

Assumptions. In Figures 9 and 10, local heat flux weighting factors are given for representative condenser and evaporator conditions. The curves were generated for the mid-G mass flux case and required use of mass-quality-dependent and mass-flux-dependent, refrigerant-side heat transfer coefficients. The local condensing coefficients were based on equations of Traviss et al. [1973]. The local evaporator coefficients were based on the work of Chaddock and Noerager [1966] for qualities up to 0.75 beyond which the heat transfer coefficient was assumed to fall off quadratically (due to dryout) to the pure vapor coefficient at a quality of 1 [Fischer and Rice 1983]. The dryout point of 0.75 was estimated from the work of Sthapak et al. [1976]. The air-side heat transfer coefficients and the relative refrigerant-to-air-side heat transfer areas, etc., were taken from the sample heat pump configuration given by Fischer and Rice [1983].

Condenser. The local condenser weighting factors shown in Figure 9 reflect the effects of an increasing condensing coefficient with increasing quality. This is because the weighting factors are the reciprocal of the U-dependent $f_Q(x)$ functions. Figure 9 shows that the constant wall temperature assumption ($f_Q = U_R$) results in the strongest weighting of the low mass quality region and the weakest in the high quality sections where the heat transfer coefficient is highest and quality changes most rapidly with length. The constant average and constant entering air temperature cases ($f_Q = U_T$ and $f_Q = 1 - e^{-\mathrm{NTU}}$, respectively) are progressively closer to the constant heat flux line since the constant air-side coefficients in U_T tend to moderate the effect of the local U_R variations.

Evaporator. For the evaporator in Figure 10, the curves are similar in shape to those for the condenser until the point of dryout ($x = 0.75$). At that point, the dropoff in heat transfer coefficient causes a switch from less than to greater than unity weighting as the lower pure vapor coefficients are approached.

Variable Heat Flux, Low Mass Flux

The local heat flux weighting factors were applied through Equation 10b to the void fraction distributions shown earlier in Figures 1 and 2. Comparative results of the integration for average density are shown in Figures 11 and 12 for the four heat flux assumptions for the homogeneous and Hughmark cases (which had the widest range of predictions for the constant heat flux case).

Evaporator. In Figure 11, both evaporator and condenser conditions are included for the case of low mass flux (as defined earlier). The range of evaporator quality used was from 0.2 to 1. The results indicate that neither void fraction method is significantly affected by the heat flux assumptions for the evaporator.

The different weightings throughout the evaporator quality range appear to approximately cancel each other. For the Hughmark method, the higher liquid fraction predictions in the mid-quality region were the most strongly lowered by the low weighting given by assumption 2. This made assumption 2 the lowest predicting of the four methods for the Hughmark evaporator case in contrast to the three other sets of comparisons where assumption 2 was the highest predicting. Note in all cases, however, that curves 3 and 4 move respectively closer to the constant heat flux case.

Condenser. For the condenser situation in Figure 11, a more significant effect is seen to result from the variable heat flux assumptions. The homogeneous results are increased up to 41% for curve 4 and up to 64% for curve 2. For the Hughmark method, increases of at most 19% to 31%, respectively, are seen.

Variable Heat Flux, Moderate to High Mass Flux

Condenser. In Figure 12, results for condenser conditions only are shown for moderate to high mass flux cases. (Evaporator variations were again insignificant.) Compared to the similar curves in Figure 11, curves 3 and 4 move downward closer to the constant heat flux curves. For mid-G conditions, curves 3 and 4 have moved almost equidistant between 1 and 2. At high G, curves 3 and 4 have moved almost all the way back to curve 1. Note that the homogeneous curves, 2, 3, and 4, are functions of mass flux through the dependence of the heat flux weighting factor on G. The shift of curves 3 and 4 toward curve 1 at higher G values reflects the decreasing dependence of the overall U values on the variable U_R value—instead depending more on the constant air-side U values.

In summary, the various heat flux assumptions considered have minimal effect on evaporator mass inventory predictions. For a condenser with low-G values, inventory values of at most 41% to 61% larger are predicted for the homogeneous method and at most 19% to 31% for the Hughmark method. For larger G values, the differences remain relatively constant for assumption 2 but reduce progressively to small values at high mass flux for assumptions 3 and 4.

DISCUSSION

From the preceding survey of possible methods, it is seen that the choice of void fraction model is of major significance in determining two-phase refrigerant charge inventory accurately. This is especially so in the evaporator. However, the choice of heat flux assumption appears to be insignificant for evaporators and of secondary to possibly equal importance to choice of void fraction model for condensers.

Comparison with Experiments

With regard to comparisons to measured charge in operating condensers and evaporators, only two sources of comparison were found.

Domanski and Didion [1983] compared total charge and condenser charge for one cooling mode condition with results calculated from their heat pump simulation model. Their model used the Lockhart-Martinelli correlation and a tube-by-tube heat exchanger model. As such, any inaccuracies due to heat flux approximations through the condenser should have been minimal. Their model underpredicted the total refrigerant charge by 26.5% and that determined to be in the condenser by 11%.* With subcooling accounting for probably 25% of the R-22 mass in the condenser, this would imply about a 15% underprediction for the mass in the two-phase region. From Figures 4 and 5, the methods of Thom and Tandon would predict values about 15% higher than the L-M method.

*Their procedure for determining the amount of mass in the condenser may have underestimated the amount of charge due to leakage past isolation valves.

Otaki [1973, 1975] made more extensive comparisons between model and experiment. He used the Hughmark method and a constant heat flux assumption and calculated the fraction of the heat exchanger length occupied by subcooled, two-phase, and superheated refrigerant. Comparisons of total charge only were made for one electric refrigerator, six window-type air conditioners (0.6 to 1.5 kW), six air-source heat pumps (3.7 to 11 kW), one water-source air conditioner (3.7 kW), one large bus cooler, and one trailer refrigeration unit. This gave a total of 17 units, some R-12 and most R-22, over 32 operating points with total mass inventory ranging from 0.22 to 22 lbm (0.10 to 10 kg). Agreement on total charge of ±10% was found, except for two points where the actual charge was underpredicted by about 15%. Some other comparisons were also made to specific heat exchanger tests, which suggested that the Hughmark method may tend to overpredict mass flux effects.

From the analysis presented here, it seems possible that such good agreement on total charge could have resulted in some part from canceling errors due to possible overprediction of condenser charge from the Hughmark method combined with underprediction of condenser charge from the constant heat flux assumption (and/or underprediction of the fraction of the condenser occupied by subcooled liquid). Similar results could possibly have been obtained with the Tandon or Premoli methods and a tube-by-tube model using heat flux assumption 3 or 4 for each tube. However, the results of Otaki certainly suggest that the higher predicting methods are likely to obtain better overall agreement. It is unproven whether this is due to better agreement in the two-phase region of the heat exchangers or because of heat exchanger inventory overprediction, which compensates for some consistent relative underprediction elsewhere in the vapor compression system.*

Recently, investigators have reported efforts to weigh the refrigerant charge in various individual components or sections of operating heat pumps under steady-state [Miller 1986] and dynamic conditions [Belth and Tree 1986]. These data, as available, could be used to further assess which of the methods presented here predict most accurately both absolute heat exchanger charge (compared to Figures 7 and 8) and relative condenser to evaporator charge (compared to Table 4).

Design Use of Inventory Results

Even though the specific preferred charge inventory method cannot be selected based on existing data, the indications are that the Hughmark, Premoli, Tandon, and Baroczy methods are leading candidates. Assuming perhaps the Premoli method as a reasonable average, Figure 8 and Table 4 could be used alone for charge inventory design considerations. Given assumptions on indoor and outdoor coil internal volumes and single-phase refrigerant fractions (especially the subcooled condenser fraction), the required charge for heating and cooling conditions can be estimated. Necessary means to reduce or eliminate charge inventory imbalance between heating and cooling (perhaps by adjusting indoor/outdoor coil internal volume) could be studied, as could the effectiveness of various means to reduce total system charge. Accumulators and receivers could be sized as a result of these considerations and those of conditions during defrost cycle operation [Otaki and Yoshii 1975].

Effect on Steady-State Off-Design Predictions

For steady-state modeling purposes, the charge inventory calculation can be used as a way to constrain system operating conditions to be consistent with the requirement of a single fixed refrigerant charge. Without such a requirement, there can be regions of operation where simplifying assumptions, such as a fixed degree of compressor inlet superheat, become inadequate. Proper off-design prediction of systems employing valves whose opening is dependent on the level of superheat (TXVs and PWMs) especially require use of some type of charge inventory analysis [Stoecker 1981].

Experience to date of the author† and others [notably Domanski and Didion (1983) and also Ahrens (1983)] suggests that absolute accuracy in charge prediction may be more important for dynamic heat pump models than for steady-state off-design predictions. For capillary tube systems with accumulators, close agreement in absolute charge prediction does not appear necessary for good off-design prediction [†; Domanski and Didion 1983], provided the predicted charge is

*Otaki [1973] did account for solubility of refrigerant in the compressor oil.

†Ongoing work on use of a charge inventory model with a steady-state heat pump design program [Fischer and Rice 1983] capable of handling capillary tubes, short-tube orifices, and TXV and PWM type expansion valves with or without accumulators.

calibrated to a known compressor inlet superheat condition at a cooling mode design point. More analysis is needed to see over what ranges and what flow control devices this calibration adjustment can be expected to suffice. Some considerations for evaluating off-design prediction accuracy with different charge inventory models include:

— the degree of effect of significantly different indoor and outdoor coil volumes on proportionality of charge distribution between heating and cooling modes, (i.e., are the off-design predictions as accurate when one coil has much more internal volume than the other);

— the degree of effect on valve openings in systems that control on superheat level, e.g., TXVs, TEVs, and PWMs;

— the degree of effect in systems that are capacity modulated over wide flow ranges;

— the possible use of an additional calibration adjustment to heating-mode design condition.

Obviously, the more confidence that can be shown in the absolute agreement of a given method, the less are the concerns for the generality and accuracy of off-design predictions. However, given the level of uncertainty shown by Figure 7, a weak dependence of off-design predictions on the accuracy in predicting the absolute amount of charge would certainly be desirable.

Effect on Transient Model Predictions

For dynamic modeling, the need for accurate prediction of the amount of refrigerant in each coil seems more inescapable. Here the off-cycle transient phenomena (and probably reverse-cycle, as well) are directly tied to the time required for refrigerant to flow into or out of the heat exchangers and adjoining accumulators and receivers, starting from the steady-state charge distribution [Murphy 1986]. Since there is this more direct interrelationship, it perhaps would be possible to use the comparison between predicted and measured off-cycle transient behavior as a means to determine the preferred void fraction model. These findings could be checked against absolute inventory data such as that of Belth and Tree [1986] and Miller [1986] for initial confirmation. This approach could avoid the need for involved component inventory measurements for each system against which inventory models were to be tested.

CONCLUSIONS

In summary, the following conclusions were reached.

1. The choice of two-phase void fraction model is of major significance in determining the absolute charge in condensers and evaporators, especially the latter. Figures 7 and 8 can be used as a guide to the inventory predictions of the various models. The fraction of the condenser occupied by subcooling and the relative size of the condenser and evaporator must also be considered in determining the effect on total charge prediction.

2. The choice of heat flux assumption is insignificant for forced convection evaporators and of secondary to possibly equal importance to choice of void fraction model for condensers. Ranges of possible effects are quantified in Figures 11 and 12.

3. Literature comparisons with experimental data suggest that the highest predicting void fraction models for condensers (where most charge is located), such as Hughmark, Premoli, Tandon, and Baroczy, will give closest agreement for total system charge. There are not sufficient data and independent comparisons at present to confidently recommend any one method. However, the Premoli method gives an approximate average of the above methods. The Hughmark method has been reported by one researcher to give good results.

4. For those currently using the homogeneous method, a switch at least to the Zivi method would be recommended on the basis of simplicity, since it has the same functional dependences and requires only a minor change to implement. Furthermore, for a constant heat flux assumption, the weighting factors W_g and W_f can be evaluated analytically [Wedekind 1977], thereby saving significantly in computational time.

5. Modeling experience to date suggests that off-design *steady-state* performance prediction may not be critically dependent on the absolute accuracy of charge predictions. The degree to which charge calibration to design conditions is sufficient for off-design prediction requires further study for the complete range of flow control types, relative heat exchanger sizes, and modulating conditions.

6. Accurate off-cycle and reverse-cycle *dynamic* predictions are expected to be most strongly dependent on accurate steady-state charge distribution predictions. Comparison of measured system off-cycle transient characteristics to predictions using the various inventory models is suggested as one approach to inventory method selection. This would be easier to perform for a variety of systems (and less system intrusive) than direct heat exchanger charge weighing for different systems.

It is hoped that this survey analysis will stimulate more open discussion and research on charge-inventory-related topics in the HVAC community. More understanding of this aspect of heat pump modeling is seen as a prerequisite to more extensive design consideration of flow control and charge inventory interactions, especially for modulating heat pumps now under development. It is believed that accurate transient modeling will require better knowledge of the charge inventory distribution than currently exists. A better understanding of control and refrigerant accumulator requirements during transient operation relates directly to improved equipment reliability through the avoidance or minimization of undesirable compressor operating conditions.

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APPENDIX A
REFERENCE EQUATIONS FOR THE MASS-FLUX-DEPENDENT VOID FRACTION CORRELATIONS

Tandon

The void fraction equations of Tandon et al. [1985] are given by:

$$\alpha = [1 - 1.928\,\mathrm{Re}_L{}^{-0.315}/F(X_{tt}) + 0.9293\,\mathrm{Re}_L{}^{-0.63}/F(X_{tt})^2] \;,\; \text{for } 50 < \mathrm{Re}_L < 1125 \;,$$

or

$$\alpha = [1 - 0.38\,\mathrm{Re}_L{}^{-0.088}/F(X_{tt}) + 0.0361\,\mathrm{Re}_L{}^{-0.176}/F(X_{tt})^2] \;,\; \text{for } \mathrm{Re}_L > 1125 \;,$$

where

$$F(X_{tt}) = 0.15(1/X_{tt} + 2.85/X_{tt}{}^{0.476}) \;,$$
$$\mathrm{Re}_L = \frac{GD_i(1-x)}{\mu_f} \;.$$

Premoli

The equations of Premoli et al. [1971] are given in terms of the slip ratio S to be applied to Equation 12.

$$S = 1 + F_1 \left(\frac{y}{1 + yF_2} - yF_2 \right)^{1/2} \;,$$

where

$$F_1 = 1.578\,\mathrm{Re}_L{}^{-0.19}(\rho_f/\rho_g)^{0.22} \;,$$
$$F_2 = 0.0273\,We_L \mathrm{Re}_L{}^{-0.51}(\rho_f/\rho_g)^{-0.08} \;,$$
$$y = \frac{\beta}{1 - \beta} \;,$$

and

Re_L = liquid Reynolds number, $\dfrac{GD_i}{\mu_f}$,

We_L = liquid Weber number, $\dfrac{G^2 D_i}{\sigma \rho_f g_c}$,

σ = surface tension,

g_c = gravitational constant,

β = volumetric quality, $\dfrac{1}{1 + \left(\dfrac{1-x}{x}\right)\left(\dfrac{\rho_g}{\rho_f}\right)}$

Hughmark

The equations for the Hughmark method involve use of another form of the homogeneous equation (Equation 11), i.e.,

$$\alpha = \frac{K_H}{1 + \left(\dfrac{1-x}{x}\right) P.I._1} = K_H \cdot \beta \;,$$

where K_H is the added dependence differentiating from the homogeneous equation where $K_H = 1$. The Hughmark flow parameter K_H is dependent on the correlating parameter Z, i.e., $K_H = f(Z)$, in the manner given by Table A.1. The parameter Z is in turn dependent on a viscosity-averaged, α-weighted Reynolds number, Re_α, the Froude number Fr, and the liquid volume fraction y_L, i.e.,

$$Z = \frac{\mathrm{Re}_\alpha^{1/6} Fr^{1/8}}{y_L^{1/4}} ,$$

where

$$\mathrm{Re}_\alpha = \frac{D_i G}{\mu_f + \alpha(\mu_g - \mu_f)} ,$$

$$Fr = \frac{V^2}{g D_i} = \frac{1}{g D_i}\left(\frac{Gx}{\beta \rho_g}\right)^2 ,$$

$$y_L = \frac{1}{1 + \left(\frac{x}{1 - x}\right)\frac{\rho_f}{\rho_g}} = 1 - \beta .$$

Combining terms, Z can be written as

$$Z = \left[\frac{D_i G}{\mu_f + \alpha(\mu_g - \mu_f)}\right]^{1/6} \left[\frac{1}{g D_i}\left(\frac{Gx}{\rho_g \beta(1 - \beta)}\right)^2\right]^{1/8} .$$

This equation for Z is used with

$$K_H = f(Z) \text{ , from Table A.1,}$$

and

$$\alpha = K_H \cdot \beta ,$$

to iteratively* evaluate α as $f(K_H, x, \beta, \mu_f, \mu_g, \rho_g, G, D_i)$.

For a given saturation condition the property dependence is eliminated and this reduces to $\alpha = f(x, G, D_i)$.

*Because of the need for iteration, the Hughmark method is the most difficult method to use of those considered here.

TABLE 1

Slip Ratios S for Property Index Values $P.I._2$
Generalized from Thom's Steam-Water Data [1964]

$P.I._2$	0.00116	0.0154	0.0375	0.0878	0.187	0.446	1.0
S	6.45	2.48	1.92	1.57	1.35	1.15	1.00

TABLE 2

Various Property Ratio Indices for a Representative Range
of R-22 Saturation Temperatures for Heat Pump Operation

	T_{sat}, F (°C)						
	−20 (−28.9)	0 (−17.8)	30 (−1.1)	45 (7.2)	90 (32.2)	110 (43.3)	130 (54.4)
$P.I._1$	0.00559	0.00869	0.0159	0.0210	0.0462	0.065	0.091
μ_l/μ_g	28.0	24.3	20.0	18.2	13.9	12.5	11.02
$P.I._2$	0.0109	0.0165	0.0289	0.0376	0.0782	0.107	0.146

TABLE 3

Generalized Liquid Fraction Correlation of Baroczy,
$f_{1-\alpha}(x) = f(X_{tt}, P.I._2)$

	X_{tt}										
$P.I._2$	0.01	0.04	0.1	0.2	0.5	1	3	5	10	30	100
	Liquid Fraction $(1-\alpha)$										
0.00002				0.0012	0.009	0.068	0.17	0.22	0.30	0.47	0.71
0.0001			0.0015	0.0054	0.030	0.104	0.23	0.29	0.38	0.57	0.79
0.0004		0.0022	0.0072	0.180	0.066	0.142	0.28	0.35	0.45	0.67	0.85
0.001	0.0018	0.0066	0.0170	0.0345	0.091	0.170	0.32	0.40	0.50	0.72	0.88
0.004	0.0043	0.0165	0.0370	0.0650	0.134	0.222	0.39	0.48	0.58	0.80	0.92
0.01	0.0050	0.0210	0.0475	0.0840	0.165	0.262	0.44	0.53	0.63	0.84	0.94
0.04	0.0056	0.0250	0.0590	0.1050	0.215	0.330	0.53	0.63	0.72	0.90	0.96
0.10	0.0058	0.0268	0.0640	0.1170	0.242	0.380	0.60	0.70	0.78	0.92	0.98
1.0	0.0060	0.0280	0.0720	0.1400	0.320	0.500	0.75	0.85	0.90	0.94	0.994

TABLE 4

Comparison of Predicted Condenser to Evaporator Two-Phase Inventory Ratios for Typical Heat Pump Conditions

Void Fraction Method	Evaporator/Condenser Temperatures 0/90 (F) −17.8/32.2 (°C)	30/110 (F) −1.1/43.3 (°C)	45/130 (F) 7.2/54.4 (°C)
	$\rho_{tp,cond} / \rho_{tp,evap}$ [a,b]		
Homogeneous	7.46	5.25	4.95
Thom	5.52	4.10	3.94
Zivi	4.29	3.31	3.13
Smith	3.60	2.92	2.86
Baroczy	3.39	2.74	2.53
Premoli (mid G)	2.58	2.38	2.28
Tandon (mid G)	2.53	2.25	2.25
Lockhart-Martinelli	2.49	2.21	2.21
Hughmark (mid G)	2.17	1.91	1.92
Hughmark (low G)	1.82	1.72	1.74

[a] Constant heat flux,
evaporator quality interval of 0.2 to 1.0,
condenser quality interval of 0.0 to 1.0.

[b] ρ_{tp} = average two-phase R-22 density.

TABLE A.1

Hughmark Flow Parameter K as a Function of Z, $K_H = f(Z)$

Z	K_H
1.3	0.185
1.5	0.225
2.0	0.325
3.0	0.49
4.0	0.605
5.0	0.675
6.0	0.72
8.0	0.767
10	0.78
15	0.808
20	0.83
40	0.88
70	0.93
130	0.98

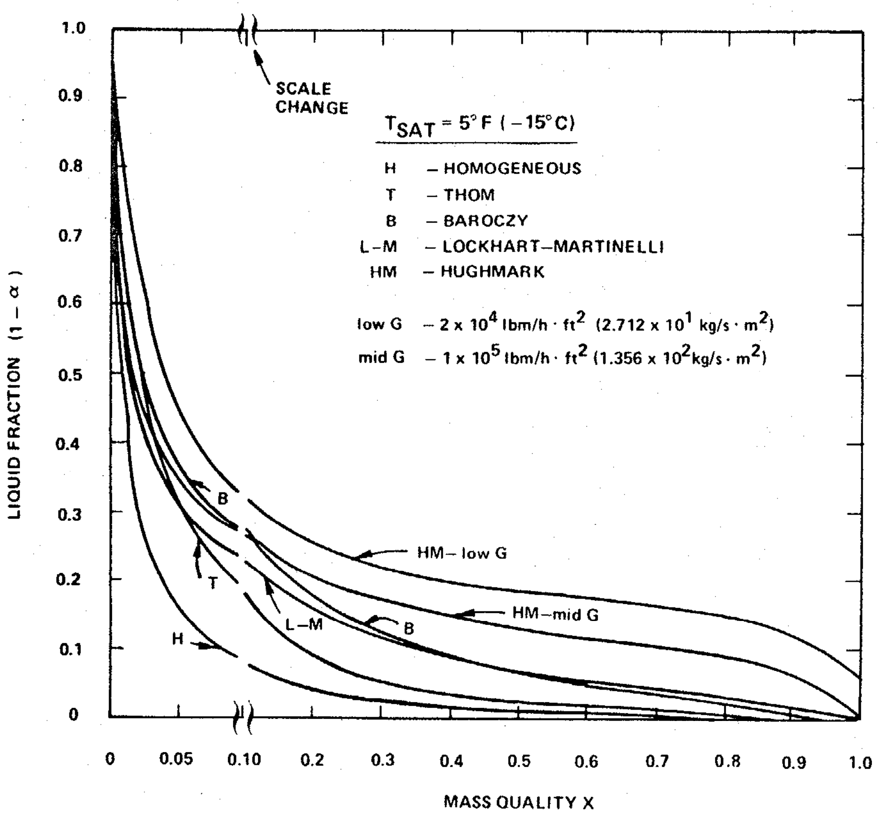

Figure 1. Comparison of local R-22 liquid fraction (holdup) predictions at a low ambient, heating-mode, evaporator condition

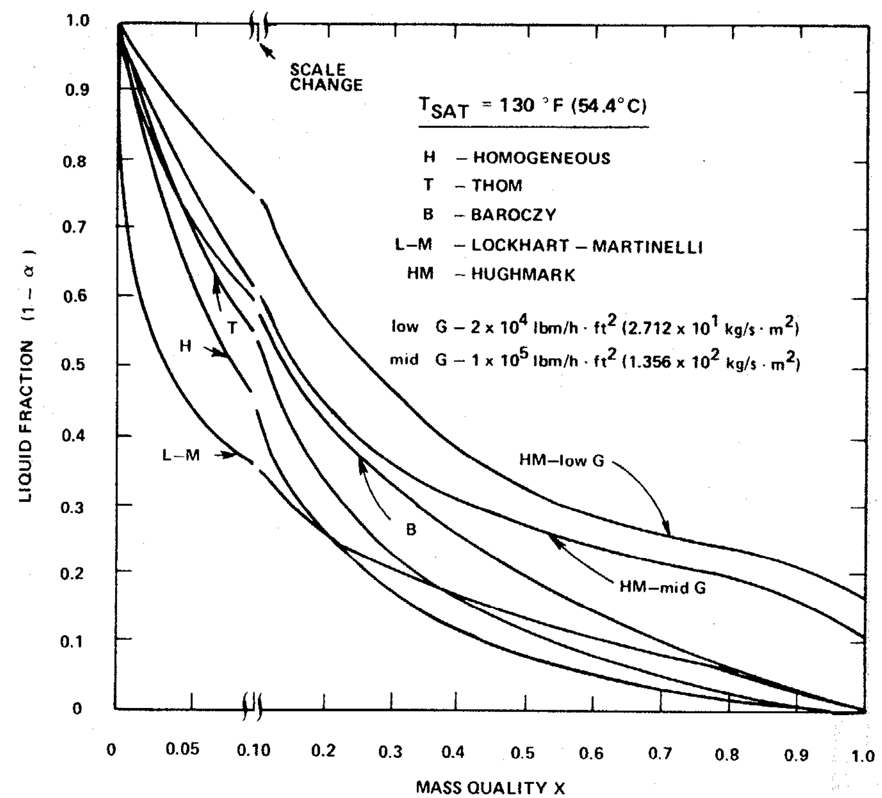

Figure 2. Comparison of local R-22 liquid fraction (holdup) predictions at a high ambient, cooling-mode, condenser condition

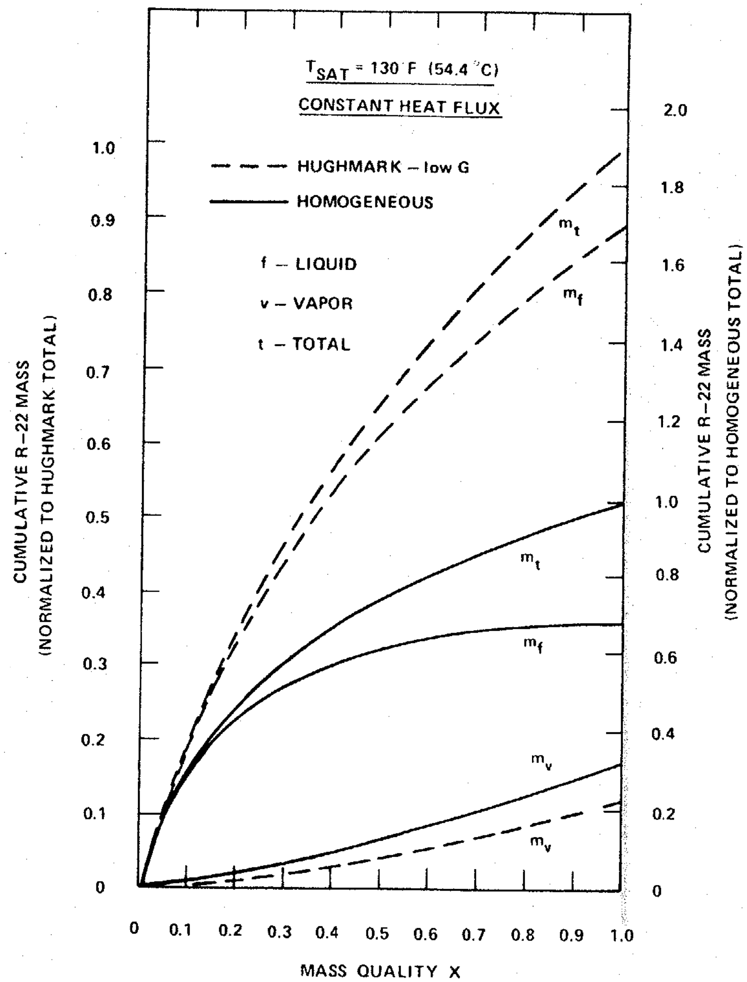

Figure 3. Comparison of cumulative R-22 mass distribution predictions for a high ambient, cooling-mode, condenser condition

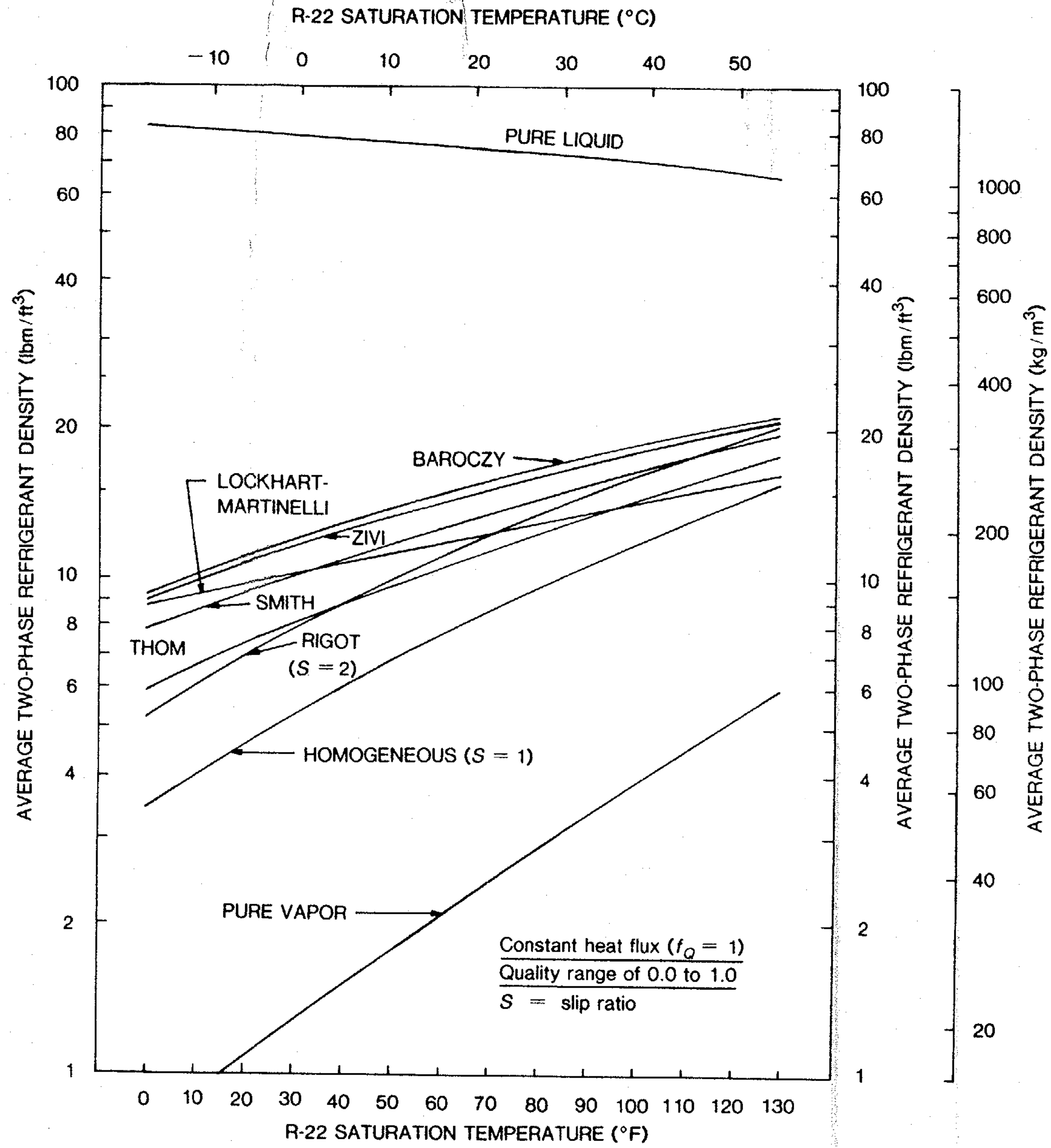

Figure 4. R-22 two-phase density predictions for mass-flux-independent void fraction models: f_Q = 1 and x = 0.0 to 1.0

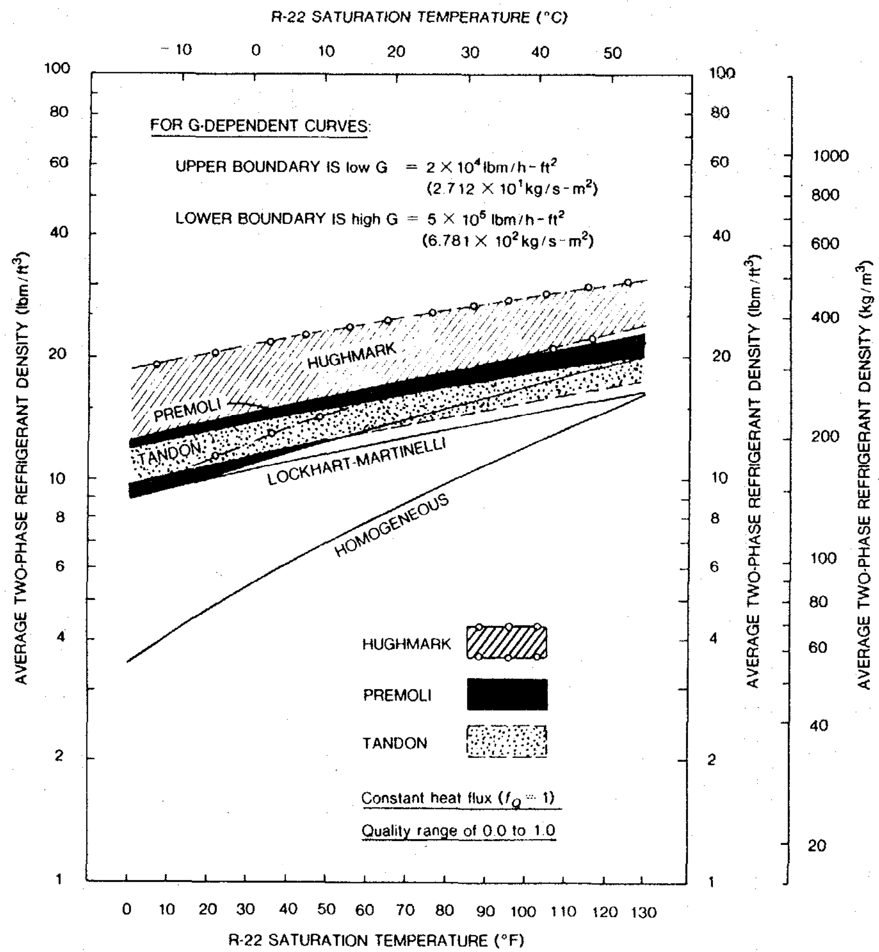

Figure 5. Range of R-22 two-phase density predictions for mass-flux-dependent void fraction models: $f_Q = 1$ and $x = 0.0$ to 1.0

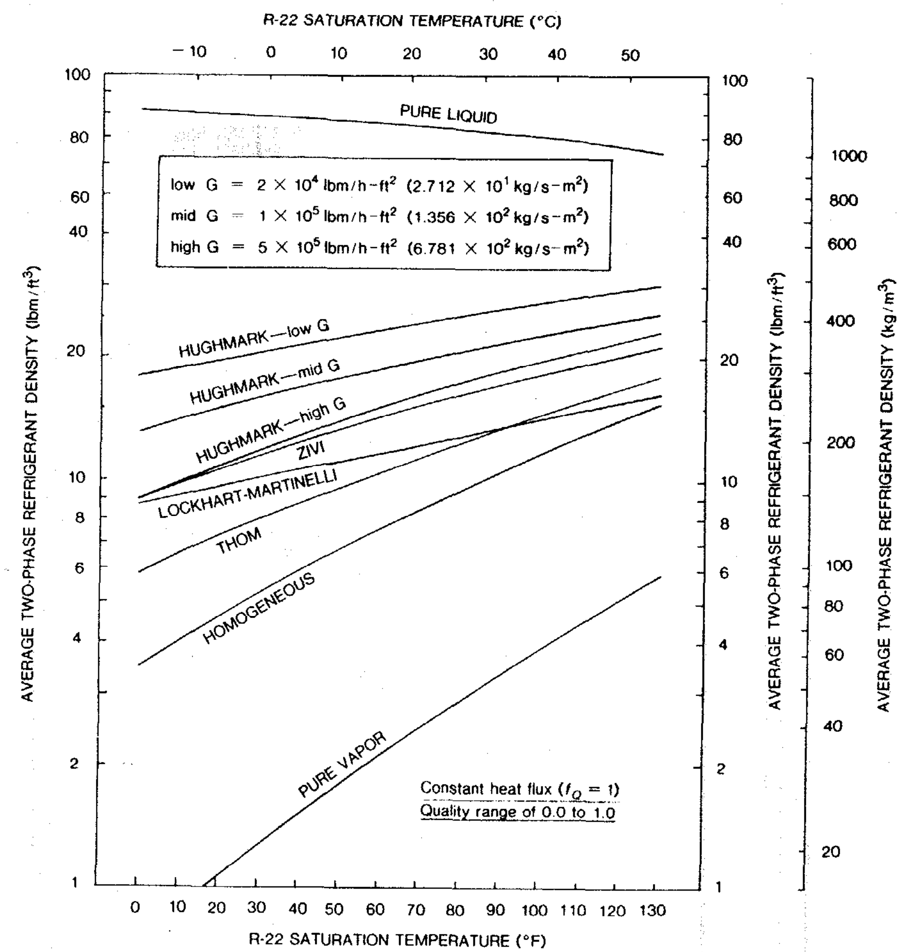

Figure 6. Summary of R-22 two-phase density predictions: $f_Q = 1$ and $x = 0.0$ to 1.0

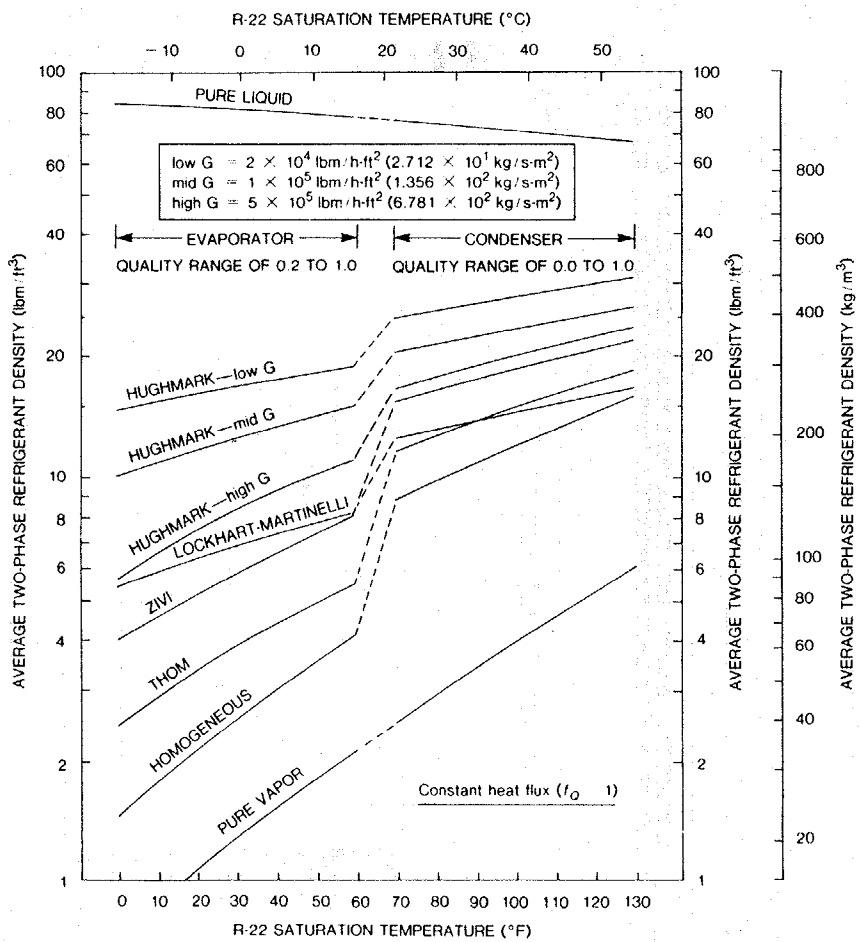

Figure 7. Summary of R-22 two-phase density predictions: f_Q = 1 and x = 0.2 to 1.0

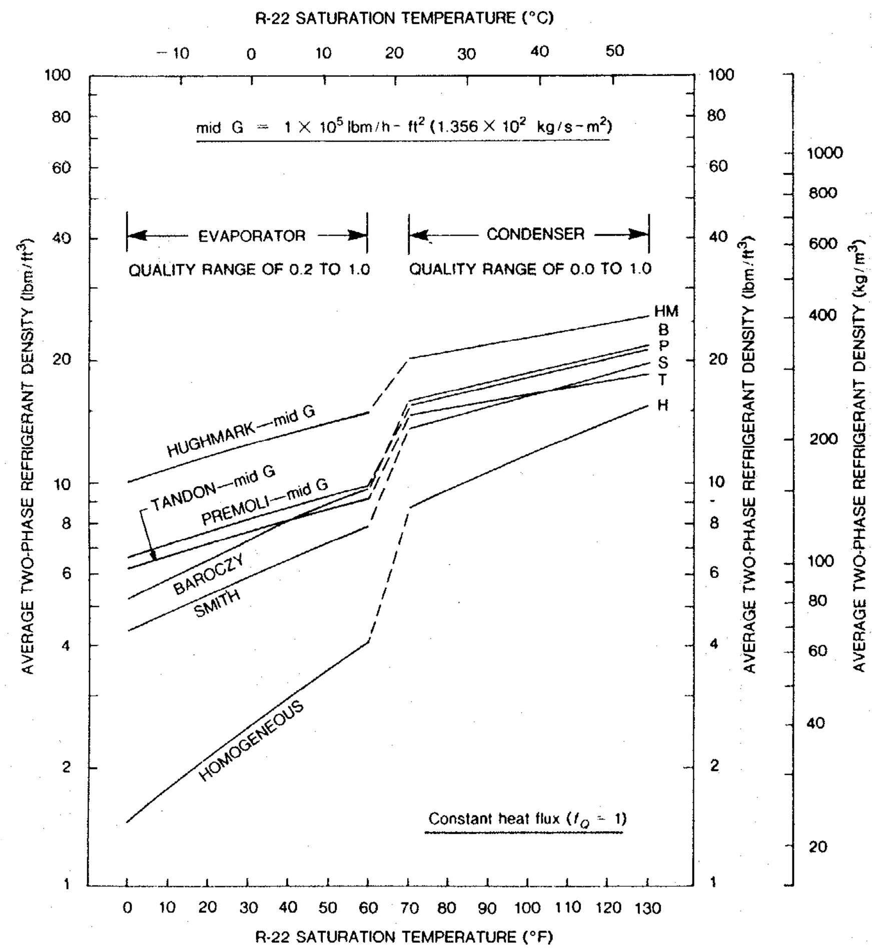

Figure 8. Supplemental comparisons of R-22 two-phase density predictions: f_Q = 1 and x = 0.2 to 1.0

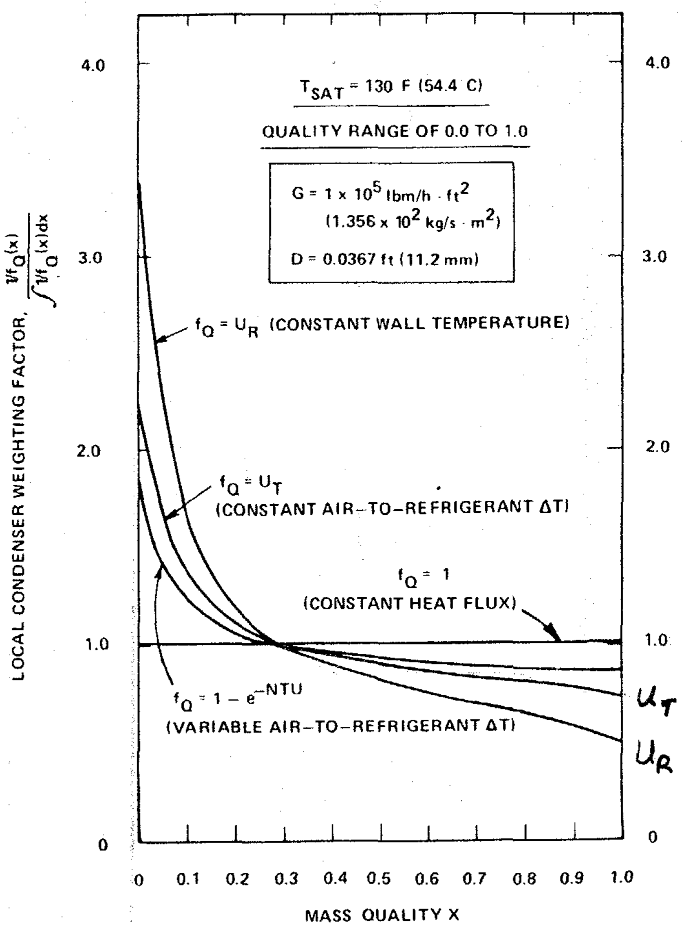

Figure 9. *Local condenser void-fraction weighting factors for four heat flux assumptions*

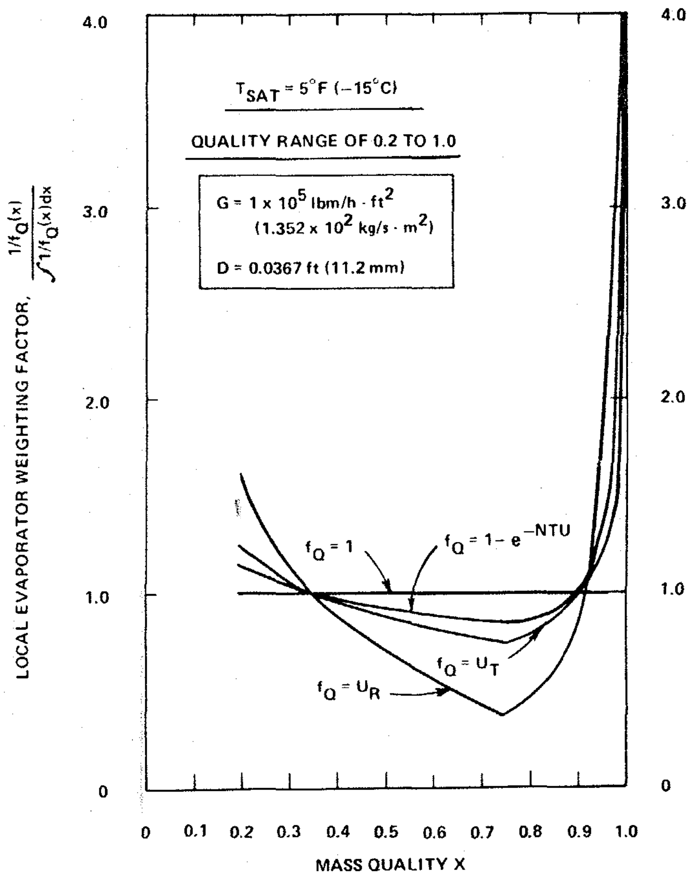

Figure 10. *Local evaporator void-fraction weighting factors for four heat flux assumptions*

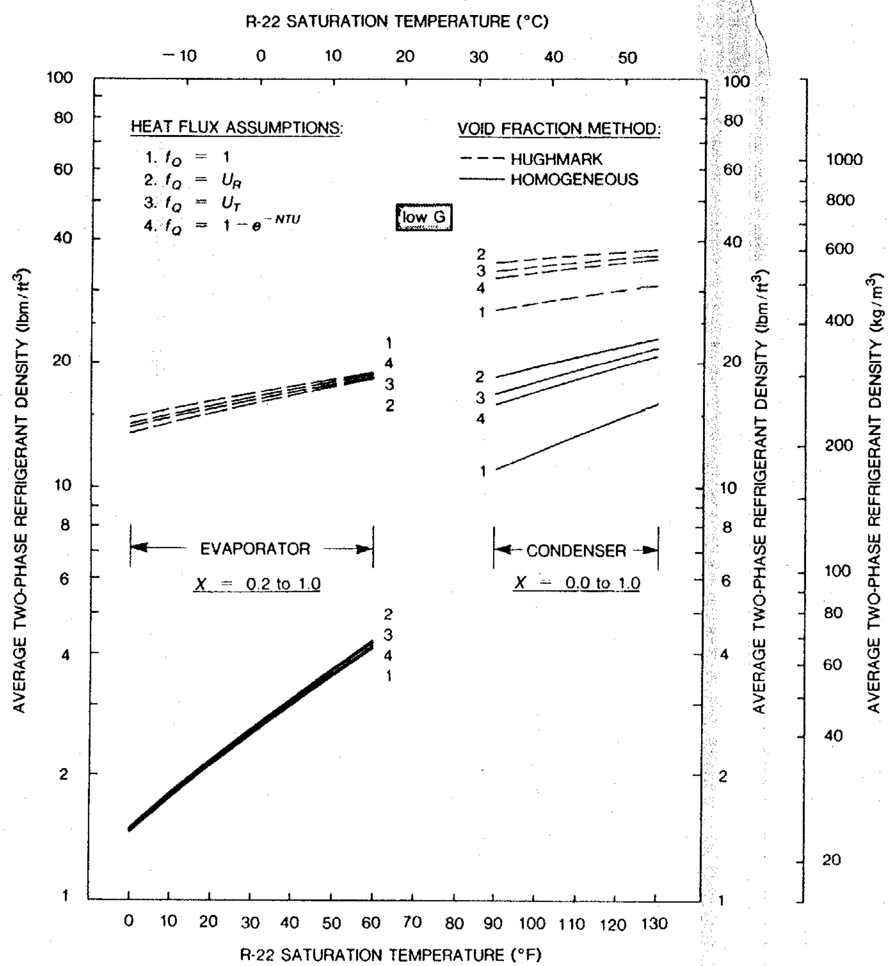

Figure 11. *Effect of heat flux assumption on R-22 two-phase density predictions: low G values, x = 0.2 to 1.0 for evaporator and 0.0 to 1.0 for condenser*

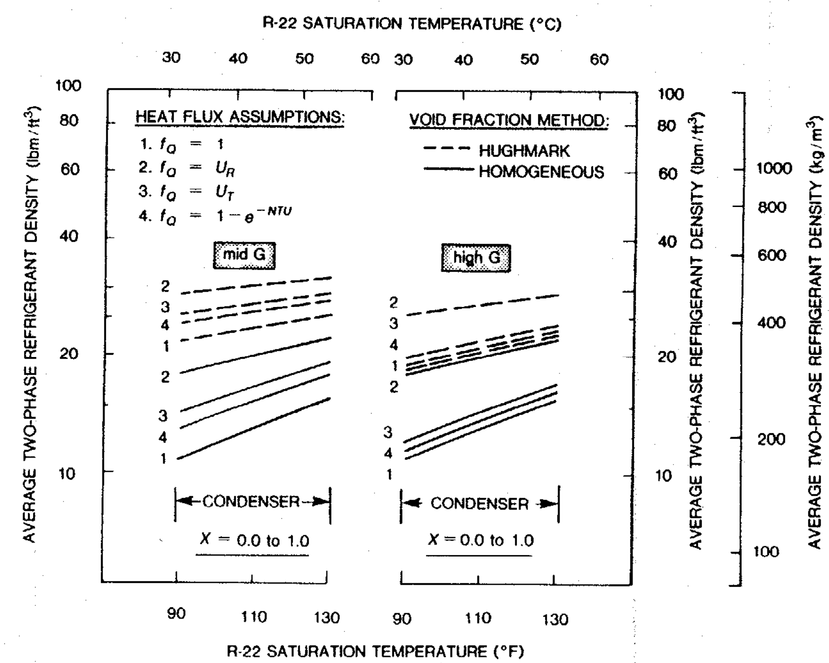

Figure 12. *Effect of heat flux assumption on R-22 two-phase density predictions: mid to high G values, condenser only*

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