

Optimized Design of Refrigerant Condensers

D. J. Kirshbaum and J. C. Chato

ACRC TR-100

June 1996

For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

*Prepared as part of ACRC Project 37
Effect of Geometric Variables and R-22 Alternatives
on Refrigerant-Side Evaporation and Condensation
J. C. Chato, Principal Investigator*

The Air Conditioning and Refrigeration Center was founded in 1988 with a grant from the estate of Richard W. Kritzer, the founder of Peerless of America Inc. A State of Illinois Technology Challenge Grant helped build the laboratory facilities. The ACRC receives continuing support from the Richard W. Kritzer Endowment and the National Science Foundation. The following organizations have also become sponsors of the Center.

Amana Refrigeration, Inc.
Brazeway, Inc.
Carrier Corporation
Caterpillar, Inc.
Dayton Thermal Products
Delphi Harrison Thermal Systems
Eaton Corporation
Electric Power Research Institute
Ford Motor Company
Frigidaire Company
General Electric Company
Lennox International, Inc.
Modine Manufacturing Co.
Peerless of America, Inc.
Redwood Microsystems, Inc.
U. S. Army CERL
U. S. Environmental Protection Agency
Whirlpool Corporation

For additional information:

*Air Conditioning & Refrigeration Center
Mechanical & Industrial Engineering Dept.
University of Illinois
1206 West Green Street
Urbana IL 61801*

217 333 3115

OPTIMIZED DESIGN OF REFRIGERANT CONDENSERS

Daniel J. Kirshbaum and John C. Chato

ABSTRACT

A Microsoft Excel spreadsheet program developed previously at ACRC to optimize condenser size with respect to surface area has been modified. A new method of modeling pipe bends has been developed incorporating a new pressure drop correlation. Estimation of total liquid and vapor masses has been added. The results using this new, more accurate method of modeling pipe bends proved to vary almost negligibly from those obtained from the previous, simpler method, confirming the usefulness of the older version of this program with the addition of the new pressure drop correlation and the estimation of the refrigerant inventory. The optimization method is also applicable to evaporator design with appropriate modifications.

INTRODUCTION

The use of a Microsoft Excel spreadsheet program to model refrigerant condensation in a heat exchanger was initially reported by Dobson et al. (1993), and followed by Chato et al. (1995). This program investigates the effect of tube diameter on the length required to completely condense a refrigerant, with fluid mass flow rate and the air side resistance held constant. The technique used to obtain a solution is to divide the heat exchanger into elements representing 5% quality decrements. The length for a given diameter is then calculated for each element using local parameters corresponding to the average quality in the element. Knowing the total length of tubing for a given diameter and for arbitrary inputs in a heat exchanger allows the surface area to be minimized by varying the tube diameter.

Unfortunately, the use of five percent quality changes may not accurately model all aspects of a heat exchanger. Generally, the element length corresponding to five percent quality decrements ranges between 100 and 500 cm. Because many calculations are dependent on average values from these relatively long sections, the accuracy of the final results using this technique may be questionable. Also, the pressure drop resulting from each pipe bend is difficult to incorporate into the calculations. Previous versions of the simulation program added a pressure drop at the end of each element to account for the effect of pipe bends. None of these approaches added a pressure drop at the correct location of the pipe bend. The addition of these pressure drops at their actual locations is the most realistic method of modeling the pipe bends.

The main foci of this study concern improvements to the previous techniques of modeling pipe bends and the addition of estimating the refrigerant inventory. A new, more accurate method for calculating pipe bend pressure drops is explored and compared to results using previous versions of the program. This comparison provides insight into the necessity for changing the method of modeling pipe bends. In addition, various other features have been added to the program. These include a new two-phase pressure drop correlation, and calculations of total liquid and vapor masses in the condenser.

THE SIMULATION PROGRAM

Due to the relative complexity of the simulation program, a discussion will be given to facilitate understanding of its main features. Assumptions made in the forming of the program include:

1. Inlet quality of 100%
2. Constant air side resistance per unit length
3. Uniform flow characteristics for each quality element

The previous version of the program does not use the type of refrigerant as an input. Therefore, separate programs exist for R-22 and R-134a, due to their different properties. Once the type of refrigerant is known, the program requires several operating conditions to be inputted by the user. These inputs, and the symbols used to denote them in the spreadsheet, are given below:

1. Refrigerant mass flow rate (\dot{m})
2. Inner tube diameter (D)
3. 180° return bend diameter (D_{rb})
4. Distance between return bends (z_{bend})
5. Inlet saturation temperature of refrigerant (T_{satin})
6. Air side resistance per unit length (R_{air})
7. Air temperature (T_{air})

The simulation program uses a main spreadsheet for inputs, outputs and simple calculations. This spreadsheet then calls Command Macros for more involved calculations. Each time any cell in the program is changed, the program runs completely through all the calculations to compute the total

length necessary to condense the fluid. In addition, the program returns several other useful outputs. All of the outputs and their symbols are shown below:

1. Two-phase length (L2ph)
2. Condenser surface area (SA)
3. Percentage of annular flow (%Ann)
4. Total pressure drop (dP)
5. Total temperature drop (dT)
6. Percentage of the temperature drop compared to the driving temperature difference (dT%)
7. Mean quality (xbar)
8. Mean refrigerant side resistance (R'ref)
9. Percentage of R'ref compared to total resistance per unit length (Rr/Rt)
10. Total mass of liquid refrigerant (mliq)
11. Total mass of vapor refrigerant (mvap)

An actual spreadsheet program's display of inputs and outputs is shown in Figure 1 for R-22.

REFRIGERANT 22					
	INPUTS			OUTPUTS	
mdot	80	[lbm/hr]	D	4.5	[mm]
D	4.5	[mm]	m(liq)	0.0449	[kg]
Drb	76.2	[mm]	m(vap)	0.0080	[kg]
z Bend	500	[mm]	L2ph	9.612	[m]
Tsatin	43.33	[C]	S.A.	0.1359	[m^2]
R'air	0.042	[m-K/W]	dp	83.23	[kPa]
Tair	32.22	[C]	dT	2.12	[°C]
			dT %	19.08%	
			%Ann	100.0%	
			R'ref	0.014	[m-K/W]
			Rr/Rt	24.4%	
			xbar	0.45	
			dprb	18.4	
			Rr/Ra	0.322	

Figure 1 - Simulation program's display of inputs and outputs

CONDENSATION CALCULATIONS

Refrigerant Side Resistance

The refrigerant side resistance is given by the equation:

$$R'_{ref} = \frac{1}{\pi Nuk_1} \quad (1)$$

The Nusselt number is dependent on the type of flow regime present. To determine this flow characteristic, the program incorporates the Wallis dimensionless gas velocity [Breber et al., 1979]

$$j_g^* = \frac{Gx}{\sqrt{gD\rho_v(\rho_l - \rho_v)}} \quad (2)$$

It is assumed that annular flow exists for j_g^* values above 1.8, while wavy flow exists for j_g^* values below 1.8. The Nusselt number calculations corresponding to wavy and annular flow, respectively, are given below [Dobson et al., 1993]:

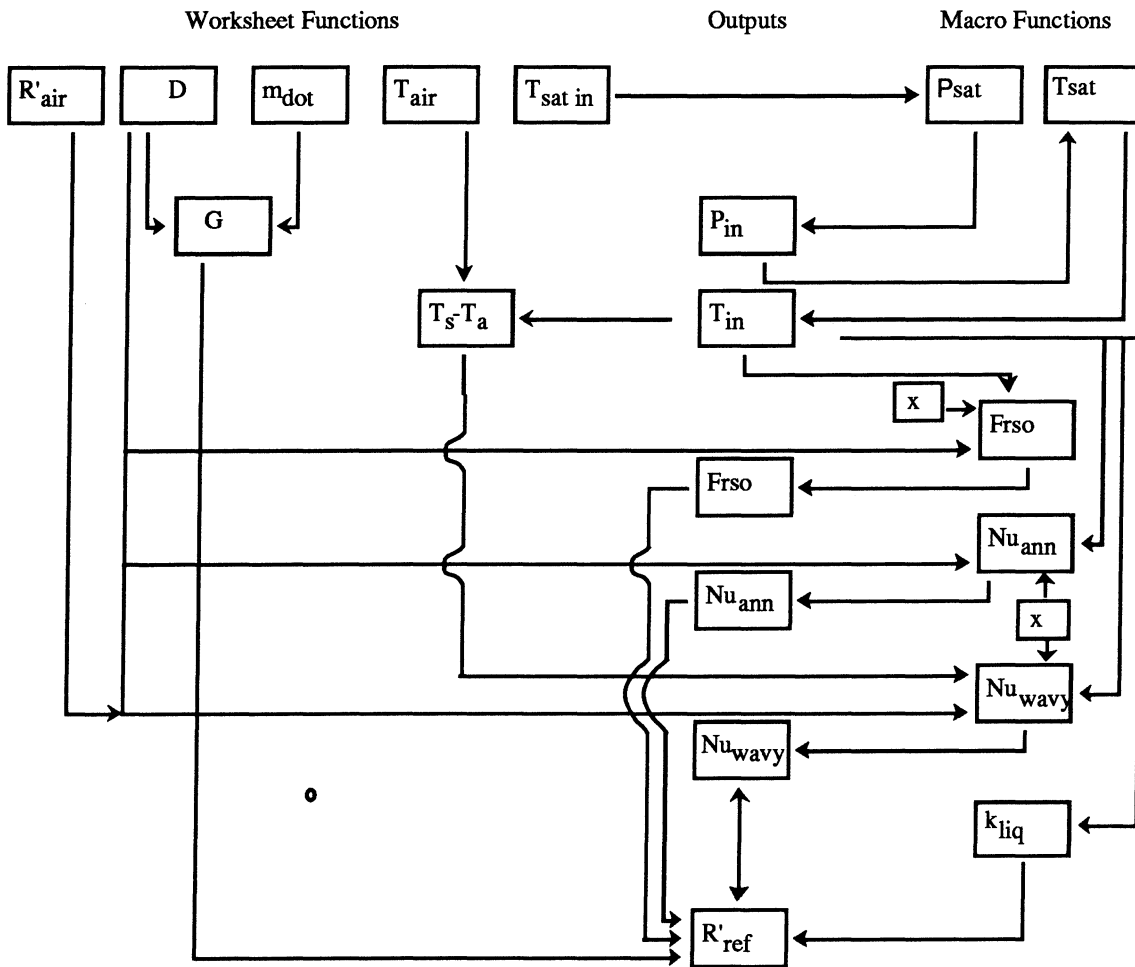
$$Nu_{wavy} = f(X_{tt}) * \left[\frac{g \rho_L (\rho_l - \rho_v) D^3 i_{lv}}{\mu \Delta T k_l} \right]^{0.25} \quad (3)$$

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

$$Nu_{ann} = 0.023 Re_l^{0.80} Pr_l^{0.3} g(X_{tt}) \quad (4)$$

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

Logic statements are used to determine which Nusselt number to use in calculating R'_{ref} . Figure 2 indicates how the program travels from the main spreadsheet to Command Macros and ultimately calculates R'_{ref} .



x inputted automatically from program
 R'_{ref} , G, and T_{in} will be used to calculate L_{act}

Figure 2 - Flowchart describing calculation of R'_{ref}

This is the main point where the program must be modified for use in evaporator design, i.e. the correlations for evaporation must replace equations (3) and (4).

Pressure Drop

Previous versions of the program, described by Dobson et al. [1993] incorporated a pressure drop correlation developed by De Souza in 1993. However, the program now uses a new two-phase pressure drop developed by De Souza et al. [1995]. One important component of this correlation is the acceleration pressure drop:

$$\Delta P_{ac} = G^2 \left\langle \left| \frac{x_o^2}{\rho_v \alpha_o} + \frac{[1-x_o]^2}{\rho_l [1-\alpha_o]} \right| - \left| \frac{x_i^2}{\rho_v \alpha_i} + \frac{[1-x_i]^2}{\rho_l [1-\alpha_i]} \right| \right\rangle \quad (5)$$

where the Zivi [1964] void fraction calculation is used as:

$$\alpha = \frac{1}{1 + \left[1 - \frac{x}{X} \right] \left\langle \frac{\rho_v}{\rho_l} \right\rangle^{0.67}} \quad (6)$$

The pressure drop correlation also includes a frictional component:

$$\Delta P_f = \Delta P_{LO} \frac{1}{\Delta X} \int \phi_{LO}^2 dx \quad (7)$$

where the frictional pressure drop for the whole mixture flowing as a liquid is given by:

$$\Delta P_{LO} = \frac{2f_{LO}G^2L}{\rho_L d} \quad (8)$$

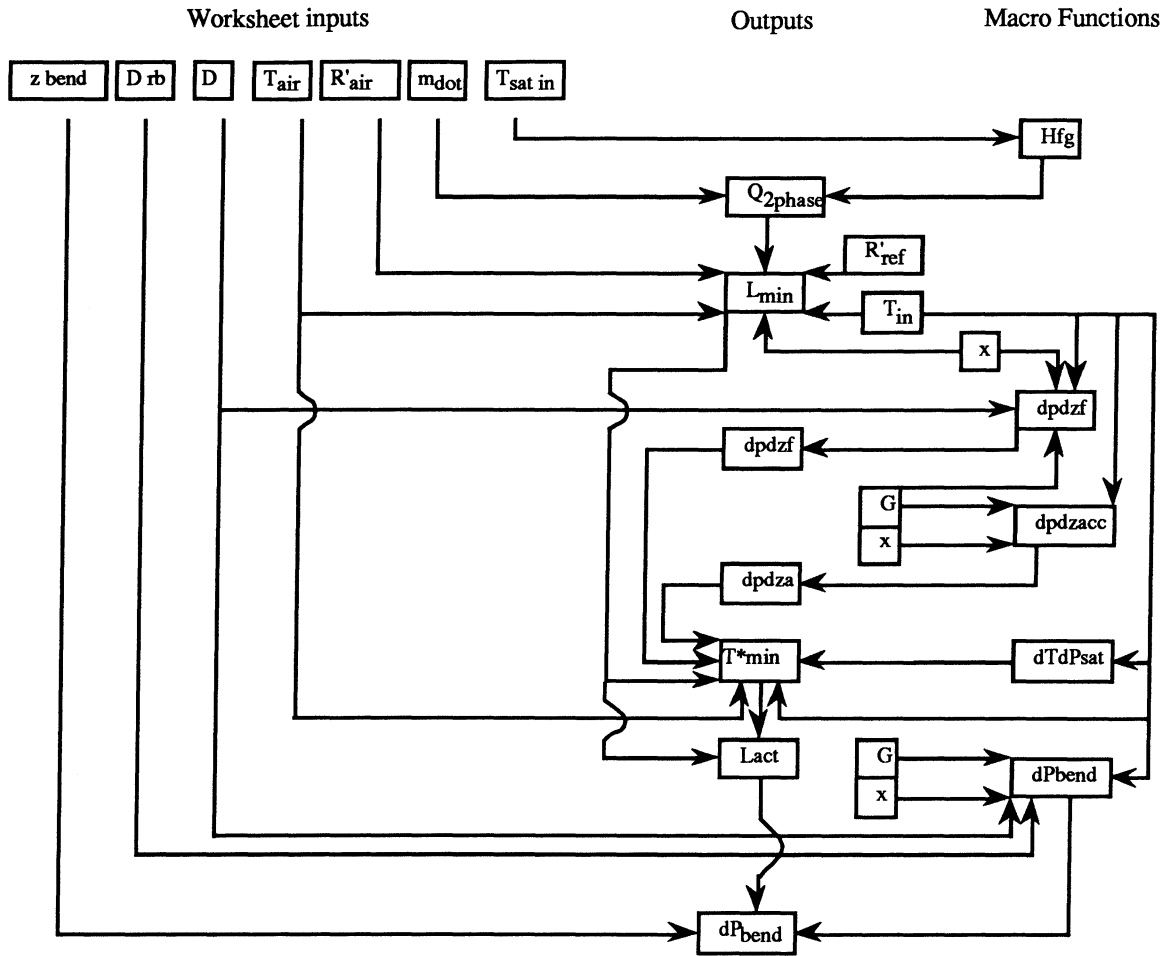
and the liquid Fanning friction coefficient is:

$$f_{LO} = \frac{0.079}{Re_{LO}^{0.25}} \quad (9)$$

The ratio of the two-phase pressure gradient to the friction pressure gradient if the total mixture flows as a liquid is given by:

$$\phi_{LO}^2 = 1 + (\Gamma^2 - 1)x^{1.75} \left(1 + 0.9524\Gamma X_u^{0.4126} \right) \quad (10)$$

To compute the total pressure drop per unit length for each section, the frictional and acceleration component are summed. The flowchart of Figure 3 shows the primary inputs and macro functions which help to calculate the pressure drop per unit length. This value is then multiplied by the length of each element and added to the pipe bend pressure drop to obtain the total pressure drop for each element. These values are summed over all elements to obtain the total pressure drop.



R'_{ref} , G , and T_{in} shown in previous flow chart
 x automatically inputted by program

Figure 3 - Flowchart describing calculation of pressure drop and element length

Condenser Length

Knowing R'_{ref} and the total pressure drop per unit length for each element, it is now possible to calculate the element length using the equations provided by Dobson [1994]:

$$L_{min} = \frac{\delta q (R'_{ref} + R'_{air})}{T_{ref} - T_{air}} \quad (11)$$

where $\delta q = mh_{lv} \Delta x$

$$T^* = \frac{\left(\frac{-\Delta P}{\Delta z} \right) \left(\frac{dT}{dP} \right)_{sat} L_{min}}{\Delta T_i} \quad (12)$$

$$L_{act} = \frac{\gamma/2 - \sqrt{(\gamma/2)^2 - \gamma T^* (1 + R^*)}}{T^*} \quad (13)$$

where $\gamma=2$ for linear temperature variations. The computation of the element length is also diagrammed in the flowchart of Figure 3. The element length is summed over all elements to determine the total condenser length.

Pipe Bend Pressure Drop

The pressure drop due to pipe bends is dependent on the length of each element, and is the last parameter of each element calculated by the program. The equation used for dP_{bend} depends on two components, friction and turning. The frictional component is given in equations (7) to (10), while the turning component was given by Christoffersen et al. [1993]:

$$\Delta P_t = \frac{G^2 v_{\text{ave}} \epsilon}{2}$$

$$\text{where } v_{\text{ave}} = (1-x)v_1 + xv_v \quad (14)$$

$$\text{and } \epsilon = \frac{C_3 X_{tt}^{C_4}}{2\delta \text{Re}_1}$$

The constants C_3 and C_4 are equal to 6.93×10^{-5} and -0.712 , respectively. The parameter d is defined as $\sqrt{D/D_{\text{RB}}}$.

Liquid and Vapor Masses

The total mass of liquid and vapor phases in the condenser is an output that has been added to the simulation program. This feature has been added by calculating the liquid and vapor masses for each quality element, then summing over all elements. Using equation (6) to compute the void fraction, the liquid and vapor masses for each element can then be found:

$$m_v = \alpha \rho_v \pi \frac{D^2}{4} \Delta L \quad (15a)$$

$$m_l = (1-\alpha) \rho_l \pi \frac{D^2}{4} \Delta L \quad (15b)$$

It is to be noted that equation (6) is only one of several equations proposed for the void fraction. However, at present there are not enough reliable experimental data available to decide which of these is the best. Consequently, we selected the one that gives reasonable values and is relatively simple.

LIMITATIONS OF THE SIMULATION PROGRAM

Wavy Flow Correlation

Equation (3) shows that a DT term exists in the calculation of Nu_{wavy} . This term complicates the calculations significantly under wavy flow conditions because Nu_{wavy} and R'_{ref} have a mutual dependence, as shown in Figure 3. This means that when equation (2) indicates wavy flow, an iterative procedure must be used by the program to calculate R'_{ref} . Because iteration is a relatively time-consuming process, calculations for wavy flow take significantly more time than those for annular flow. In addition, Excel requires initial values in interdependent cells to begin iteration. If no number value exists in these cells, the program will not iterate successfully. Therefore, when the program fails to provide output values under wavy flow conditions, inputs corresponding to 100% annular flow should be initially used. Equation (3) shows that to produce annular flow, mass flux values should be increased. This can be easily accomplished by increasing the mass flow rate and/or decreasing tube diameter. The program will then generate results for annular or wavy flow situations. Table 1 provides input values that will yield completely annular flow for both R-22 and R-134a. These values also correspond to the high flow rates incurred in the condenser of a mobile air-conditioner.

Table 1: High flow rate input values producing completely annular flow

Input	Value
Refrigerant mass flow rate [lbm/hr]	80
Tube diameter [mm]	4.5
Distance between bends [mm]	500
Return bend diameter [mm]	76.2
Inlet saturation temperature of refrigerant [°C]	43.3
Air side resistance per unit length [W/m-K]	0.042
Air temperature [°C]	32.2

Pipe Bend Pressure Drop Calculation

The pressure drop resulting from pipe bends is difficult to incorporate into the solution methodology used in this program. Wattelet [1994] added the pressure drop of a full bend after each five percent element. This method is only accurate if each element corresponds to the distance between bends. However, the element length and distance between bends can be quite different, making this technique a relatively poor approximation. Dobson [1994] added a bend pressure drop after each element, but multiplied the full bend pressure drop by the ratio of ΔL to the distance between bends. This technique is a highly conservative approach, which calculates a slightly lower pressure than actually exists for each element, thereby underestimating the driving temperature difference. Although Dobson's approach was an improvement over earlier techniques, the return bend pressure drops are not placed in their actual location. Therefore, improvements can still be made in the modeling of pipe bends.

ANALYSIS

The method of calculating pipe bend pressure drops has been modified in an effort to more accurately model these effects. The pressure drop for a full pipe bend is now added to the total pressure drop at the end of the quality element immediately preceding the bend location. This provides a more accurate method of modeling pipe bends. Figure 4 shows physically why this is true. The pressure variation in an actual heat exchanger decreases in each straight section, and also undergoes drops in the pipe bends. For the method used by Dobson [1994] to calculate pipe bend pressure drop, the pressure variation along the tube is qualitatively shown in Figure 4(a), compared with that in an actual heat exchanger. This shows that the calculated pressure is generally lower than the actual pressure throughout the condenser, because scaled pressure drops are added after each quality decrement. This results in a smaller temperature difference between the fluid and the air over the length of the condenser. Equation (11) shows that a lower temperature difference results in a longer calculated length than actually needed. Figure 4(b) compares the new method of calculating pipe bend pressure drop to the situation in an actual heat exchanger. To locate the pipe bend more exactly, the quality change per element has been reduced to 2.5%. This method of modeling pipe bends is only effective if the element length is short enough (or the distance between bends long enough) to have several quality elements in-between each pipe bend. Otherwise it becomes very similar to earlier approaches.

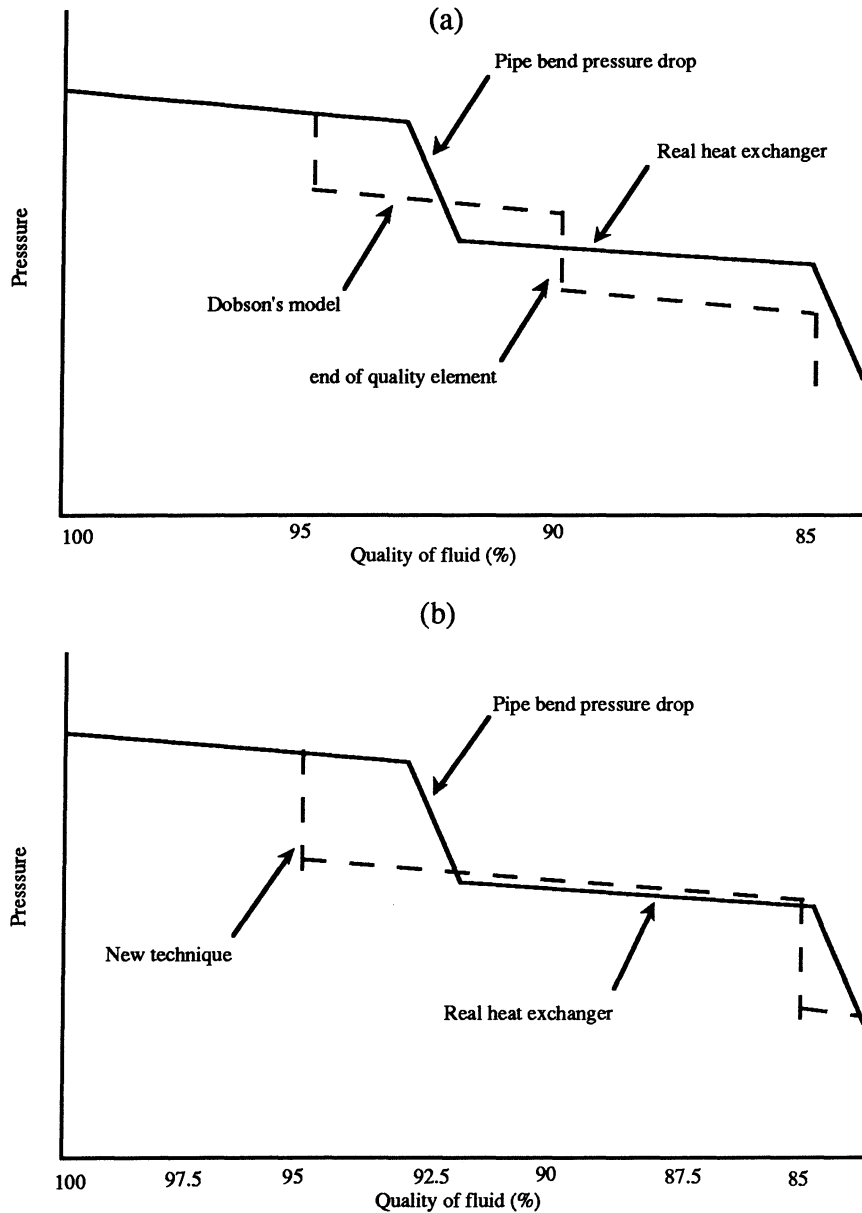


Figure 4 - Physical comparison of different approaches of modeling pipe bends

The analysis used in modifying the simulation program initially was focused on improving the method of modeling pipe bends. However, several other changes were made to the program used by Dobson [1994]. These changes include updating the program to incorporate a new two-phase pressure drop correlation developed by De Souza et al. [1995]. This correlation, shown in equations (5) to (10), is also applied to the pipe bend pressure drop.

In addition, a new feature has been added to the program. It is the calculation of total masses of liquid and vapor in the condenser, i.e. the charge inventory. Equations (15) are used for these calculations. As previously stated, the liquid and vapor masses in each section are calculated for each quality element, then summed over all elements and returned as outputs.

DISCUSSION

Although the new approach of modeling pipe bends provides more accurate results, several disadvantages exist with this technique. First, the time required to complete each run through the program increases considerably. Excel must calculate the length of each element before it adds in the pipe bend pressure drop. The modified program checks if a bend will be found in the next element, then adds in a pressure drop corresponding to a full bend at the outlet quality of the element. This approach makes iteration necessary, which dramatically increases computation time. In addition, reducing the quality decrement per element from 5 to 2.5% doubles the amount of calculations the program must perform. Both of these effects cause the calculation time to be extremely long for the modified program.

In spite of the excessively long calculation time, results using the modified program have been found. Figure 5 shows a comparison of optimization results using several different methods for modeling pipe bends. One technique shown which was not explicitly discussed uses Dobson's methods, but decreases the element length to 2.5%. This figure illustrates the negligible difference in the optimization results between old and new techniques. This theme is reinforced in Table 2, which compares the condenser lengths using three different approaches.

From the differences in the techniques of modeling pipe bends, one would expect the new method to calculate a smaller value for condenser surface area and length. However, it is apparent from Figure 5 that the values for R-22 are nearly identical to the previous methods. There are several explanations for this. First, the pipe bend pressure drop, for any given quality element, is between 10 to 25 percent of the frictional pressure drop. This means that the pipe bend pressure drop has a small effect compared to the frictional pressure drops. In addition, the new method of modeling pipe bends, although more accurate than previous methods, is limited by the length of each element. Although the quality decrement per element was decreased from 5 to 2.5%, this method still cannot insert pressure drops as close to the actual pipe bend location as desired. Therefore, both methods underestimate the driving temperature difference and calculate greater lengths and surface areas than in a real heat exchanger. It appears that the element length would have to be decreased even further to observe more obvious differences between old and new techniques.

Figure 6 and Table 3 respectively show the surface area and length comparisons for R-134a. The differences between the techniques of modeling pipe bends is very clear for this refrigerant. As expected, the condenser length and surface area is lower for the new method, indicating a less conservative approximation. The differences between results for R-22 and R-134a can be explained partially by the fact that the percentage of the total pressure drop per element caused by the pipe bends is significantly higher for R-134a. This makes the technique of modeling pipe bends more important in the condenser size. However, the optimization of surface area indicates the same optimum diameter for R-134a using all three approaches. For pure diameter optimization purposes, therefore, the Dobson method is just as effective as the new method.

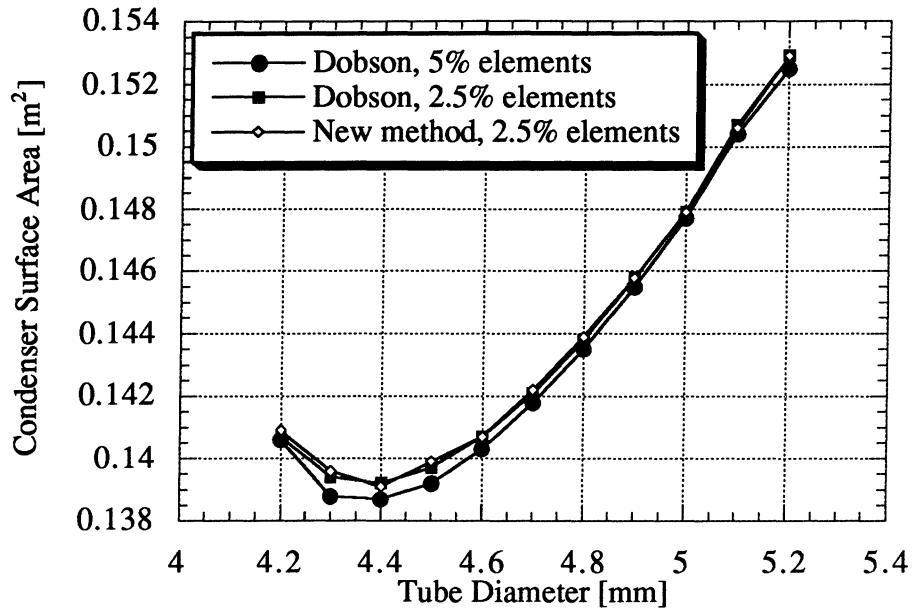


Figure 5 - Comparison of optimization results using different techniques of modeling pipe bends for R-22 with high flow rate inputs

Table 2 - Comparison of condenser lengths, in meters, using different techniques of modeling pipe bends for R-22 with high flow rate inputs

D [mm]	Dobson, 5%	Dobson, 2.5%	New method
4.2	10.66	10.66	10.68
4.3	10.27	10.32	10.33
4.4	10.03	10.07	10.06
4.5	9.85	9.88	9.92
4.6	9.71	9.74	9.74
4.7	9.60	9.62	9.63
4.8	9.52	9.54	9.54
4.9	9.45	9.47	9.47
5	9.40	9.42	9.42
5.1	9.39	9.41	9.40
5.2	9.34	9.36	9.36

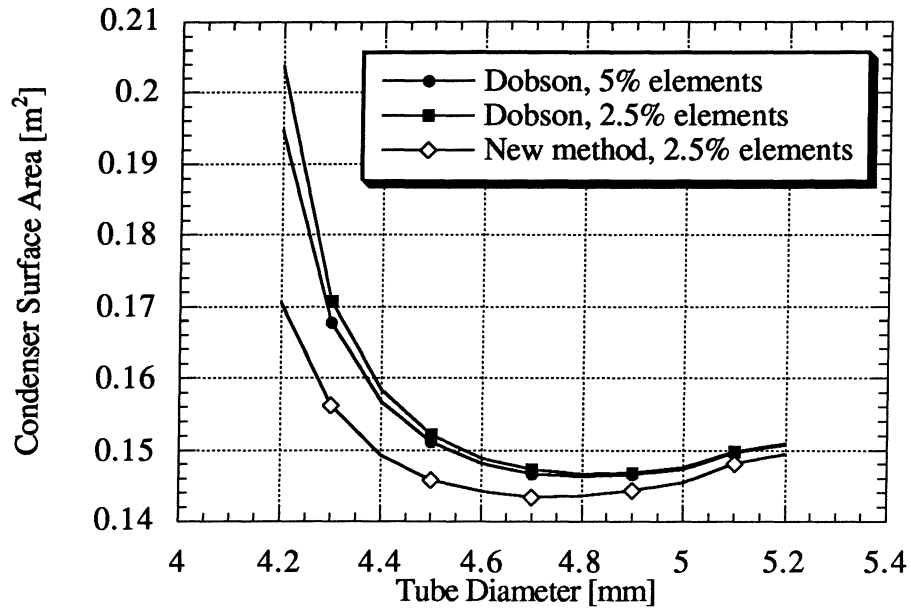


Figure 6 - Comparison of optimization results using different techniques of modeling pipe bends for R-134a with high flow rate inputs

Table 3 - Comparison of condenser lengths, in meters, using different techniques of modeling pipe bends for R-134a with high flow rate inputs

D [mm]	Dobson, 5%	Dobson, 2.5%	New method
4.2	14.76	15.44	12.93
4.3	12.42	12.64	11.56
4.4	11.34	11.46	10.79
4.5	10.69	10.76	10.32
4.6	10.25	10.30	9.98
4.7	9.93	9.97	9.71
4.8	9.70	9.73	9.52
4.9	9.52	9.54	9.37
5	9.38	9.39	9.26
5.1	9.34	9.35	9.25
5.2	9.23	9.24	9.15

CONCLUSIONS AND RECOMMENDATIONS

Two-phase heat transfer and pressure drop correlations developed at the ACRC have allowed a spreadsheet program to simulate the two-phase portion of condensers. The current program allows simulation with R-22 and R-134a, and is used to investigate the effects of altering the tube diameter while keeping the air side resistance and refrigerant mass flow rate constant. An effort has been made to facilitate the use of this program, through explanation of its solution methodology. In addition, modifications have been made to update the program, add more features, and improve the modeling of pipe bends. The changes made to the technique of modeling pipe bends, however, did not have a major effect on the tube diameter optimization results. For R-22 the results using old and new methods are nearly identical. For R-134a, however, the results are less conservative for the new approach, yet both methods yield the same optimum diameter. Therefore the method used by Dobson (1994) to model pipe bends will be recommended as the most effective method for optimizing tube diameter. The updated version of this program provides the total mass of both liquid and vapor refrigerant phases and a new pressure drop correlation. Additional features which will be addressed in the future include allowing variable air side resistance along the length of the tube, and including correlations for single phase flow of superheated vapor. Adapting this version of the simulation program for use with evaporators is also recommended for the future.

NOMENCLATURE

SYMBOLS

A_C	cross sectional area
c_p	specific heat at constant pressure
D	inner tube diameter
f	friction factor
Fr	Froude number
g	acceleration due to gravity
G	mass flux, $\frac{\dot{m}}{A_C}$
h	heat transfer coefficient
i_{LV}	enthalpy of vaporization
J_g^*	Wallis' dimensionless gas velocity
k	thermal conductivity
L	length
m	mass
\dot{m}	fluid mass flow rate
Nu	Nusselt number, $\frac{hD}{k_l}$
Pr	Prandtl number, $\frac{\mu c_p}{k}$
q	heat transfer rate
q''	heat flux
R'	thermal resistance per unit length

R^*	dimensionless resistance factor, $\frac{R'_{ref}}{R'_a}$
Re_{LO}	Liquid Reynolds number, $\frac{Gd}{\mu_1}$
Re_1	superficial liquid Reynolds number, $\frac{GD(1-x)}{\mu_1}$
T^*	dimensionless temperature drop
x	vapor quality
X_{tt}	Lockhart-Martinelli parameter, $\left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \left(\frac{1-x}{x}\right)^{0.9}$

GREEK SYMBOLS

α	void fraction
δ	pressure drop parameter
ϵ	dimensionless resistance factor
ΔP	pressure difference
ΔT	temperature difference
γ	nonlinear temperature drop factor
Γ	physical property index, $\left(\frac{\rho_l}{\rho_v}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.125}$
ϕ_{LO}^2	two-phase friction multiplier
μ	dynamic viscosity
ρ	density

SUBSCRIPTS

ac	acceleration
ann	annular flow conditions
f	friction
I	inlet
l	liquid
LO	liquid only
o	outlet
ref	refrigerant
RB	return bend
tot	total
v	vapor
wavy	wavy flow conditions

REFERENCES

- Breber, G., J.W. Palen, and J. Taborek, 1979, "Prediction of Horizontal Tubeside Condensation of Pure Components Using Flow Regime Criteria", *Condensation Heat Transfer*, ASME, New York, pp. 1-8.
- Chato, J.C. et al., 1995, "Prediction of Refrigerant-Side Heat Transfer and Pressure Drop for the Design of Evaporators and Condensers," *Proc. 46th International Appliance Technical Conference*, University of Illinois at Urbana-Champaign
- Christoffersen, B.C., J.C. Chato, and J.P. Wattlelet, 1993, "Evaporation of R-22 and its Alternatives in Horizontal Smooth and Micro-Fin Tubes", *ACRC Technical Report. 47*, University of Illinois at Urbana-Champaign.
- Dobson, M.K., J.P. Wattlelet, and J.C. Chato, 1993, "Optimal Sizing of Two-Phase Heat Exchangers", *ACRC Technical Report 42.*, University of Illinois at Urbana-Champaign.
- Dobson, M.K., et al., 1994, "Heat Transfer and Flow Regimes During Condensation in Horizontal Tubes", *ACRC Technical Report. 57*, University of Illinois at Urbana-Champaign..
- Souza, A.M., and Pimenta, M.M., 1995, "Prediction of Pressure Drop During Horizontal Two-Phase Flow of Pure and Mixed Refrigerants", *ASME Symposium on Cavitation and Multiphase Flow*, 210, pp. 161-171.
- Wattlelet, J.P., et al., 1994, "Heat Transfer Flow Regimes of Refrigerants in a Horizontal-Tube Evaporator", *ACRC Technical Report. 55*, University of Illinois at Urbana-Champaign.
- Zivi, S.M. 1964. "Estimation of Steady-State Steam Void-Fraction by Means of the Principle of Minimum Entropy Production", *Journal of Heat Transfer*, 86, pp. 247-252.