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An Evaluation of a Two-Evaporator Refrigerator-Freezer Using Nonazeotropic Refrigerant Mixtures

Final Report

Report Prepared by

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under
Subcontract 86X-47952C
for
OAK RIDGE NATIONAL LABORATORY
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the

U.S. DEPARTMENT OF ENERGY

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ABSTRACT

The temperature of a nonazeotropic mixture of refrigerants changes during evaporation and condensation at constant pressure. Refrigerant mixtures offer the potential of better matching the two temperature levels (approximately 38°F in the fresh food compartment and 5°F in the freezer) in a household refrigerator-freezer than a single refrigerant. This feature may reduce energy consumption.

The objective of this project was to investigate, through laboratory testing, the feasibility of using refrigerant mixtures to save energy. The unit selected for testing was the Amana ESTR-18D, two evaporator refrigerator-freezer developed jointly under U.S. Department of Energy sponsorship by Arthur D. Little, Inc., and Amana Refrigeration, Inc.

Tests using two mixture concentrations of R22 and R114 refrigerants demonstrated reliable refrigerator performance but no appreciable energy savings. An analytical model was developed to identify modifications to the refrigeration system in order to realize the benefits of mixed refrigerants. Based on the model, the expansion device was modified to reduce the system pressure ratio. Predicted benefits in evaporator heat exchanger effectiveness were observed, but were offset by the reduced heat capacity and density of the mixtures compared to the standard refrigerant. No energy savings were demonstrated with the ESTR-18D test unit.

EXECUTIVE SUMMARY

A. INTRODUCTION

The temperature of a nonazeotropic mixture of refrigerants (MR) changes during evaporation and condensation at constant pressure. As a result, MR offers the potential for operating the evaporators at different temperatures to better match the desired component temperatures. By improving the heat exchanger refrigerant-to-air temperature difference, MR may provide equivalent refrigeration capacity with lower energy consumption. An eighteen cubic foot refrigerator-freezer (Model ESTR-18D) developed by Arthur D. Little and Amana Refrigeration, Inc., under U.S. Department of Energy sponsorship through a subcontract with the Oak Ridge National Laboratory (ORNL), is different from conventional designs. It employs a two-evaporator refrigeration system, which may effectively utilize MR at two refrigeration temperatures.

The objective of this project was to investigate the energy savings of using MR in this refrigerator-freezer. Another purpose was to obtain practical experience using MR, including an evaluation of the effect of MR on heat exchanger and fan system design and identification of the controls necessary for effectively utilizing MR in refrigerator-freezers.

B. BASELINE LABORATORY TESTS

The purposes of the baseline tests were:

- to assure proper operation of the ESTR-18D, instrumentation, data acquisition system, and test facility under normal 90°F closed-door test conditions using the standard single refrigerant R12;
- to observe the performance of the ESTR-18D when two different mixtures of R22 and R114 (40%/60% and 60%/40% by weight) were used in place of R12, and the total charge was varied; and
- to identify changes in performance and energy efficiency due to the refrigerant mixtures solely, i.e., without hardware modification.

The results are summarized in Table 1 (page 13). The refrigerant mixtures were chosen based on discussions with duPont and other researchers which pointed to these as having the most similarity to R12 operationally and offering the most potential for efficiency improvement.

After significant effort was expended on initial experimental testing and modifications, the refrigerator and instrumentation operated predictably and repeatably. Several runs were necessary to determine the proper refrigerant charge for the mixtures, so that adequate condenser subcooling was achieved with no liquid refrigerant being returned to the compressor suction inlet. We observed the performance with R12 and R22/R114 and found that both 60/40 and 40/60% wt mixtures operated at evaporator and condenser pressures reasonably close to R12. We were unable to measure any experimentally-significant energy savings with mixtures using the unmodified refrigerator freezer.

C. ANALYTICAL MODEL AND DESIGN CHANGES

Because the laboratory tests (Table 1, page 13) showed no appreciable energy savings benefit from the MR mixtures in an unmodified refrigerator, it was decided to use a simple computer model to determine operational changes that could be made to enhance the performance.

A simple simulation based on the dual evaporator configuration was developed. Since the high side pressure was chosen as an independent variable, the effect of condenser design cannot be predicted. The solution schematic is shown in Figure 5 (page 23). A series of freezer-evaporator temperatures are selected until the heat flow to the refrigerant in the evaporators is equal to the heat flow to the air passing over the evaporators.

The analytic study focussed on the evaporator configuration since that is where the benefit of MR was expected. Changes to the evaporators were simulated. An increase in the air flow rate from 42 CFM to 100 CFM was found to reduce the single refrigerant (SR) evaporator effectiveness from .49 to .25 but increase the heat transport capability (effectiveness x heat flow capacity) by 21%. The MR heat exchanger also could benefit from increased air flow rate but to a lesser degree. Table 6 (page 27) shows a 2% improvement for MR compared to a 4% improvement for SR brought about by the increased airflow. The simulation, therefore, shows that increased air flow rate does not improve the relative performance of MR and SR.

In the case of the SR, the high-side pressure and the condensing temperatures are uniquely related by saturation conditions of temperature and pressure. For the MR, the pressure and temperature are composition dependent, so we exercised the model to attempt to use this feature to advantage. Table 7 shows the predicted benefit of reducing the high-side pressure by adjusting the throttling. At 140 psia, the MR system is predicted to yield a COP of 1.05 versus an SR system value (at normal high-side pressure of 160 psia) of 0.92, or a 14% improvement in steady state efficiency and a 10% reduction in energy consumption. Therefore, it was decided to evaluate experimentally whether the reduced head pressure (140 psia) could be achieved while maintaining the condenser heat rejection rate for the MR.

D. SYSTEM MODIFICATION TESTS

The analytical model predicted that equivalent refrigeration could be provided at decreased system pressure ratio, thus reducing compressor power requirements. Therefore, the length of capillary tube not soldered to the suction line was removed and replaced with a precision needle valve. The results of testing are shown in Table 7 (page 30). The range of system pressure ratio adjustment was limited by condenser performance. When the valve was opened to reduce the pressure ratio, the condenser was unable to subcool the increased refrigerant flow rate. This lead to the formation of vapor bubbles at the condenser outlet. Attempts to eliminate the problem by increasing refrigerant charge until liquid was observed at the compressor suction inlet did not significantly alleviate the problem. The range of pressure ratio adjustment was therefore limited, and we were unable to demonstrate the improved performance we had expected.

E. CONCLUSIONS

Simple substitution of two refrigerant mixtures for R12 in a two-evaporator refrigerator-freezer showed no measurable energy savings benefit.

At equal suction pressures, the R22/R114 MR showed a substantially lower refrigerant temperature than R12. This suggested a greater available refrigeration capacity per energy input. Enhanced heat exchanger effectiveness provided by the MR further appeared to favor the MR. However, during testing, the lower heat capacity of the MR was observed to offset these benefits resulting in about the same heat transfer.

For the condenser, the MR appears to have less desirable characteristics than the SR. In the tests conducted, the 40/60 mixture of R22/R114 has higher condensing pressures for lower condensing temperatures, which means that the compressor must work slightly harder with a MR to create the necessary heat rejection temperatures.

Providing forced air over the condenser would only worsen the prospects for MR by increasing the minimum heat capacity (and effectiveness) of the SR and not improving the MR heat exchanger performance.

Tests with the 40/60% wt of R22/R114 mixture showed that the evaporator heat exchanger performance closely correlated with NTU-effectiveness relations and revealed no significant difference in refrigerant heat transfer coefficients between R12 and the mixture.

In summary, expected benefits in evaporator heat exchanger effectiveness were observed, but were offset by reduced heat capacity and density effects, resulting in no apparent increase in refrigeration efficiency. Modifications to the expansion valve did not improve MR performance in the ESTR-18D demonstrator and no energy savings were predicted through evaporator heat exchanger modifications.

I. INTRODUCTION

A. BACKGROUND AND OBJECTIVE

The temperature of a nonazeotropic mixture of refrigerants (MR) changes during evaporation and condensation at constant pressure. As a result, MR is well matched for non-isothermal refrigeration. By reducing the heat exchanger refrigerant to air temperature difference, MR offers the potential to provide equivalent refrigeration capacity with lower energy consumption.

Residential refrigerator-freezers provide refrigeration at two temperature levels (approximately 38°F in the fresh food compartment and 5°F in the freezer). Most use a single refrigerant with one evaporator and distribute the air to each compartment to achieve the desired compartment temperature.

An eighteen cubic foot refrigerator-freezer (Model ESTR-18D) developed by Arthur D. Little and Amana Refrigeration, Inc., under U.S. Department of Energy sponsorship through a subcontract with ORNL [1,2], is different from conventional designs. It employs a two-evaporator refrigeration system, thus offering the possibility of effectively utilizing MR at two refrigeration temperatures.

The objective of this project was to investigate, through combined testing and analysis, the feasibility of using MR in this refrigerator-freezer to realize an energy savings over the same unit running with standard R12 refrigerant. Another purpose was to obtain practical experience using MR, including an evaluation of the effect of MR on heat exchanger and fan system design and identification of the controls necessary for effectively utilizing MR in refrigerator-freezers.

To meet the project objective, the following tasks were carried out:

Task 1

Develop analytical and experimentally-based preliminary estimates of refrigerator performance using MR.

Task 1a

Perform a literature review, interview current researchers and plan initial testing.

Task 1b

Develop an analytical model to simulate refrigerator-freezer performance using MR and validate it with the results of laboratory experiments.

Task 1c

Determine the effects of MR on the performance of the ESTR-18D two-evaporator refrigerator-freezer.

Task 2

Develop a methodology for optimizing MR performance.

Task 3

Develop recommendations for the MR system design.

Task 3a

Analyze and specify freezer and fresh-food evaporators and associated air-system designs.

Task 3b

Analyze and specify a condenser type, configuration and associated air system.

Task 3c

Analyze and specify an expansion device and liquid-to-suction heat exchanger.

B. LITERATURE SURVEY

The first task of the program involved a search and investigation of literature describing prior and current research in MR. The main purpose of this task was to assure that all publicly available work was identified to prevent duplication of effort. Prior to this project, a comprehensive literature search was conducted for ORNL by Stoecker and Launay [3,4]. Our search uncovered little new information. We were therefore assured that the information at hand reflected the state-of-the-art in the use of MR in refrigeration systems.

As part of this task, meetings and discussions were also held with Stoecker of the University of Illinois, Cooper of duPont, and Didion of the National Bureau of Standards to discuss the most attractive refrigerant mixtures and the most significant and recent research findings. These discussions was suggested that two mixtures offered potential for use in refrigerator-freezers:

- 40% (by weight) R22/60% R114, and
- 60% R22/40% R114.

We, therefore, decided to use these mixtures in the laboratory tests.

C. ANALYTICAL MODELS

Originally, it was planned that a publicly available computer model developed for ORNL by Stoecker would be used to assist in the design and analysis phases of this program. However, as a result of discussions with Stoecker, we concluded that the computer model would not be appropriate for ESTR-18D system performance predictions. This is because:

- liquid-to-refrigerant heat exchangers are modeled, not air-to-refrigerant;
- heat exchangers are modeled as pure counterflow with no alternate-coil-geometry capabilities; and
- no explicit modeling is done of compressor mass flow as a function of system conditions.

Therefore, the initial scope of work was changed to include the development of a simple analytical model to predict the effects of refrigerant mixtures and component modifications on system efficiency. This model is described in Section III.

D. LABORATORY TESTS

Two sets of tests were conducted in this project. Baseline experiments using the single, standard refrigerant (R12) and MR tests using two different concentration mixtures of R22 and R114 were conducted to evaluate relative performance in an unmodified ESTR-18D (two evaporator refrigerator-freezer). Initially, normal cyclic tests were performed. However, the heat transfer agreement between the cooling of the air and the heating of the refrigerant, which should be identical, was poor. The problem was attributed to the thermal capacitance of the cabinet mass and the nonuniformity of the compartment temperatures caused by cycling.

Therefore, a calorimetric method of monitoring steady-state performance was developed using electric resistance heaters to maintain constant compartment temperatures (nominally 5°F in the freezer and 38°F in the fresh food compartment). The sum of the steady heater input and the calculated cabinet heat leak provides an accurate measure of the system heat load. Delivered refrigeration is determined by measuring the refrigerant mass flow and calculating the total refrigerant enthalpy difference between condenser outlet and compressor inlet. The thermodynamic properties for the mixtures were taken from duPont-supplied refrigerant property data. It was found that success in balancing the load and delivered refrigeration to acceptable accuracy (±5%) depends on extremely careful and accurate measurement of all parameters.

Tests using MR in the unmodified ESTR-18D were repeated using the calorimetric method. The results of this testing were analyzed and

used with the analytical output of the model to specify a second set of tests on a modified ESTR-18D refrigerator with an adjustable throttle valve (replacing the capillary tube) in an effort to demonstrate the savings predicted in the model by controlling overall system pressure ratio. A detailed discussion of the test results is contained in Sections II. and IV.

II. BASELINE LABORATORY TESTS

A. TEST FACILITY AND INSTRUMENTATION

The refrigerator test facility (see Figure 1) is composed of a temperature-controlled chamber which can be used to simulate a wide range of ambient conditions. In this program, all tests were run at 90°F, the standard condition for closed door testing accepted by the industry. The test facility has a data acquisition system based on a Digital Equipment Corporation PDP-11 computer with 52 channels of A/D and D/A capabilities. The data acquisition system can be tied to analog input sensors as well as digital and analog output accepting devices. The instrumentation used is listed in Table 1.

The ESTR-18D refrigerator (see Figure 2) is instrumented for power, refrigerant mass flow, pressure and temperature measurements. A schematic of instrumentation locations is presented in Figure 3.

B. PRELIMINARY TESTING

The purposes of the baseline tests were:

1) to assure proper operation of the ESTR-18D, instrumentation, data acquisition system, and test facility under normal 90°F closed-door test conditions using the standard single refrigerant R12;

2) to observe the performance of the ESTR-18D when two different mixtures of R22 and R114 (40%/60% and 60%/40% by weight) were used in place of R12, and the total charge was varied; and

3) to attempt to identify changes in performance and energy efficiency due to the refrigerant mixtures solely, i.e., without hardware modification.

The results are summarized in Table 2. The refrigerant mixtures of R22 and R114 at 40%/60% and 60%/40% by weight were chosen based on discussions with duPont which pointed to these as having the most similarity to R12 operationally and offering the most potential for efficiency improvement.

After expending significant effort on initial testing and modifications, the refrigerator and instrumentation operated predictably and repeatably. Several runs were necessary to determine the proper refrigerant charge for the mixtures, so that adequate condenser subcooling was achieved with no liquid refrigerant being returned to the compressor suction inlet. Charge was measured using a Robinair "Dial a Charge" charging cylinder. We observed the performance with R12 and R22/R114 and found that both 60/40 and 40/60% MR mixtures

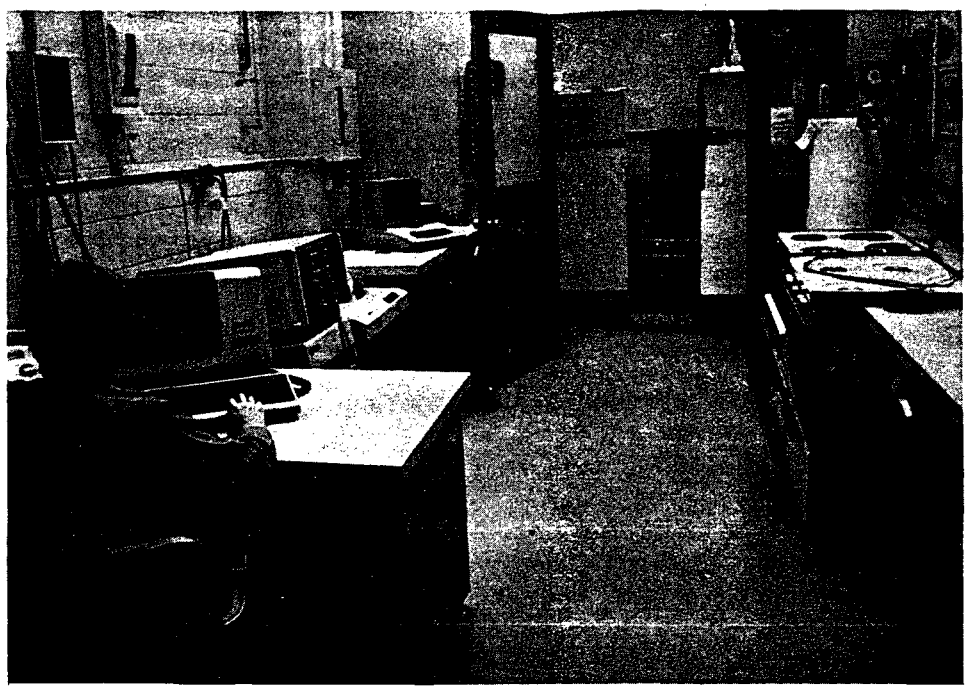

FIGURE 1 REFRIGERATOR LABORATORY

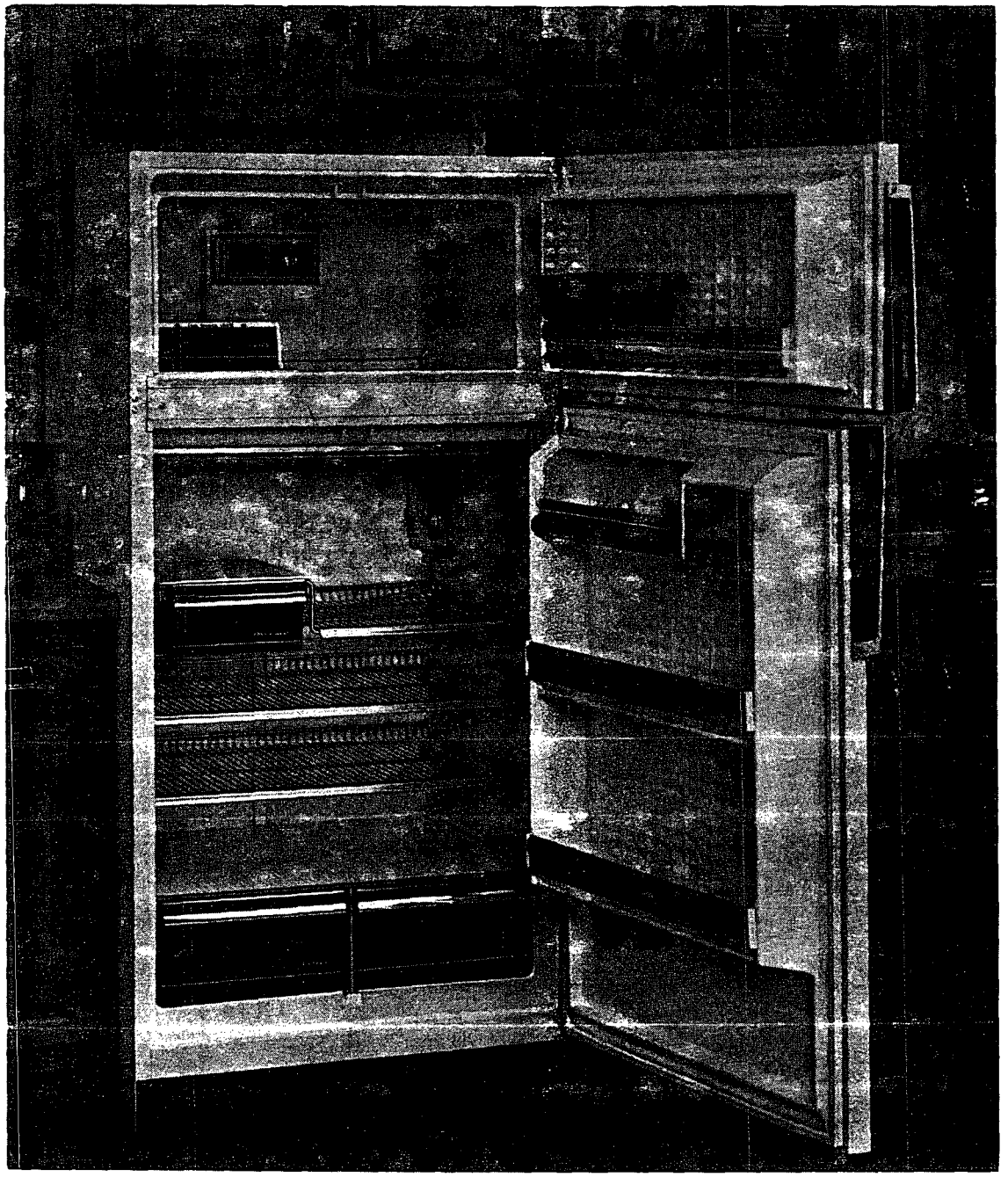

FIGURE 2 AMANA REFRIGERATOR-FREEZER, MODEL ESTR-18D

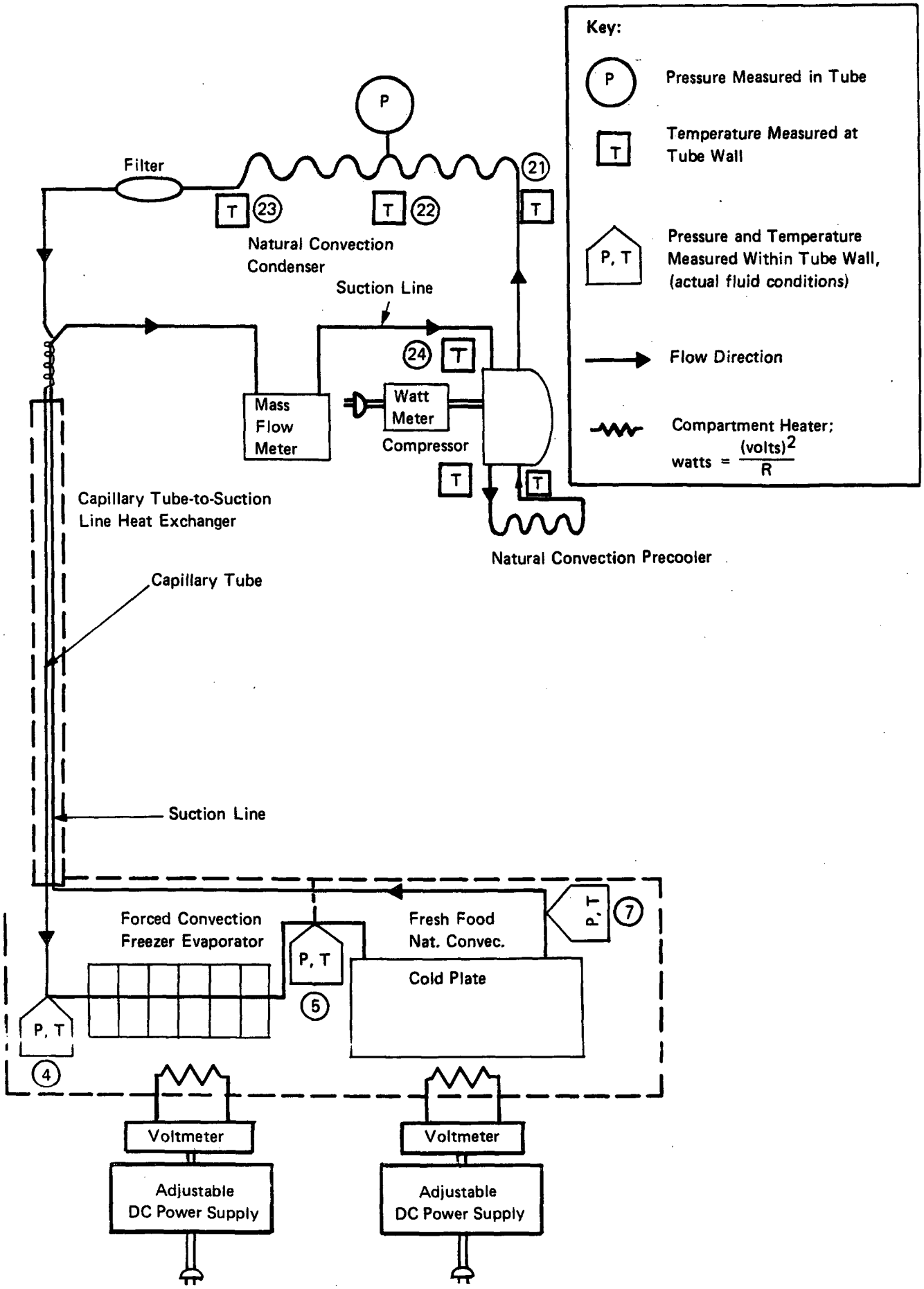

FIGURE 3 ESTR18-D REFRIGERATION AND INSTRUMENTATION DIAGRAM UNDER STEADY STATE (CALORIMETRIC) OPERATION

Table 1

INSTRUMENTATION

1. PRESSURE TRANSDUCERS

   - o High Pressure: Setra Systems Model 205-2, 0-500 psig (accuracy = .1170 F.S.)
   - o Low Pressure: Setra Systems Model 205-2, 0-100 psia (accuracy = .11% F.S.)

2. THERMISTORS

   - o High Temperature: Fenwal Model UUA32J3 (accuracy = ±.2°C, 0-70°C)
   - o Low Temperature: Fenwal Model UUD21J1 (accuracy = ±.2°C, -20-+50°C)

3. MASS FLOWMETER

   - o Matheson Model 603 Rotometer with stainless steel float (estimated accuracy of ±5%)

Table 2

CALORIMETRIC TEST RESULTS

Date	Refrigerant	Charge (Oz)	Delivered Refrigeration (Btu/hr)	Calculated Load (Btu/hr)	Compressor Power (Watts)	Mass Flow (Lb/hr)	REFRIGERANT PRESSURES Phi (psia)	Plow (psia)	AIR TEMPERATURES Ambient (°F)	FZ Bulk (°F)	FF Bulk (°F)	EER (Btu/w-hr)
1 2/16/83	R12	7½	468.9	478.1	155.8	8.25	159	17.8	90.4	5.6	37.1	3.01
1 2/19/83	R12	7½	470.6	481.1	156.8	8.25	159	17.8	91.3	5.4	35.5	3.00
1/13/84	R12	7½	449.8	478.7	154.0	7.9	159	16.6	90.8	3.5	38.7	2.92
1/19/84	40/60*	8½	515.7	478.5	153.9	8.75	172	16.0	90.6	3.9	38.3	3.35
2/15/84	R12	8½	490.5	503.2	159.0	8.5	154	18.7	90.6	3.2	39.7	3.08
2/17/84	40/60*	10½	468.2	473.1	150.7	7.9	161	17.3	89.7	3.4	37.7	3.11
2/27/84	40/60*	10½	450.8	479.7	147.7	7.7	160	16.9	90.8	4.0	38.0	3.05
3/5/84	40/60*	10½	477.9	483.8	152.8	8.1	158	17.5	89.6	2.9	37.0	3.13
3/12/84	60/40*	10½	635.9	601.3	189.8	11.1	172	24.1	90.9	3.1	36.5	3.35
3/14/84	60/40*	10½	609.5	553.7	186.4	10.6	168	22.9	91.6	4.5	36.5	3.27

*R22/R114 by wt

operated at evaporator and condenser pressures reasonably close to R12. We were unable to measure any experimentally significant energy savings with mixtures using the unmodified refrigerator freezer.

Table 3 shows the temperatures observed at various points in the system during the tests as well as the typical values we selected to represent the system performance in the analytical model. We observed a pressure drop in the freezer and fresh food evaporators of about 2 psi. This accounts for the difference between the saturation temperature and the measured temperature for the single refrigerant.

Table 3

SUMMARY RESULTS - TEMPERATURES

Date	Pressure Cond (psia)	Pressure Evap Out (psia)	Precooler In (°F)	Cond In (°F)	Cond Out (°F)	$\dot{m}$ (lbs/hr)	Evap In 4 (°F)	Mid 5 (°F)	Out 7 (°F)	Compressor Power (Watts)	Refrigeration Q_{FF} (BTUH)	Refrigeration Q_{FRZ} (BTUH)
Single Refrigerant (R12)												
12/16/83	159.0	17.8	196.1	145.7	107.0	8.2	- 4.9	- 4.1	7.1	156	178	300
12/19/83	159.0	17.8	196.1	147.0	106.0	8.2	- 4.9	- 3.9	1.5	156	183	297
1/13/83	159.0	16.6	192.5	147.0	107.6	7.9	- 7.6	- 6.7	11.0	154	169	310
2/15/83	153.8	18.6	184.9	133.0	103.7	8.5	- 9.0	- 7.6	10.4	159	133	365
Typical Values	159.0	18.0	195.0	146.0	106.0	8.3	- 7.0	- 6.0	+ 10.0	155	485	
								COP = .917				
Mixed Refrigerant (40/60)												
1/19/84	171.6	16.0	186.4	137.5	100.8	8.7	-15.4	- 1.5	1.8	154	169	309
2/17/84	160.7	17.3	181.1	124.0	98.7	7.9	-18.1	- 1.1	3.5	151	158	315
2/27/84	160.0	16.9	179.5	122.0	101.9	7.7	-17.7	- .2	4.4	148	160	320
3/5/84	158.4	17.5	175.5	122.0	98.3	8.1	-16.6	- 2.2	2.6	153	160	324
Typical Values	160.0	17.0	180.0	125.0	100.0	8.1	-16.5	- 2.0	+ 3.0	151	478	
								COP = .928 Based on Compressor Power				

III. ANALYTICAL MODEL AND POSSIBLE DESIGN CHANGES

The laboratory tests (Table 2) showed no appreciable energy savings benefit from the MR in an unmodified refrigerator. It was decided to use a simple computer model to identify operational changes that could be made to enhance the performance.

A. FUNDAMENTAL RELATIONS

Much work has been devoted to MR effects on energy consumption. While entropy (Second Law of Thermodynamics) analysis of the refrigeration cycle provide valuable insights into the cycle efficiency, it is First Law analysis which determines energy consumption. The First Law refrigeration efficiency is given by the coefficient of performance (COP):

$$\text{COP} = \frac{\text{Delivered Refrigeration in Btu/hr}}{\text{Total Electrical Energy Consumption in Btu/hr}} = \frac{Q_1}{W_1} \qquad [1]$$

Since a complete First Law model of the refrigeration system in this case would be extremely expensive to develop, it was decided to derive a simple analytical model of the key components. The model was written to reflect the expected effects of the MR in this project as outlined below. Refrigerant properties for SR and MR are critical to the analysis and were taken from the following sources:

SR Properties - taken from ASHRAE [5] "Thermodynamic Properties of Refrigerants," 1969.

MR Properties - taken from tabular data produced by duPont computer routine [6] which calculates properties based on the Redlich-Kwong-Soave equations of state (see Appendix B).

- <u>Compressor inlet pressure and density</u>.

The COP can be written very approximately as:

$$\text{COP} \propto \frac{\dot{m}\,(T_1 - T_4)}{\dot{m}\;\; P_1/\rho_1} \quad \begin{matrix}\text{See Equation 4, page 19} \\ \text{See Equation 5, page 19}\end{matrix}$$

At each evaporator inlet temperature T_4:

$$\text{COP}\ (T_4) \propto \frac{\rho_1}{P_1}$$

Table 4 shows a comparison of the pressure and density values of SR and MR at evaporator inlet temperatures experienced during the tests. The last two columns show the ratio of ρ_1/P_1 and indicates that there is little difference between MR and SR at each inlet temperature.

Table 4

COMPARISON OF MR AND SR PRESSURE AND DENSITY AT COMPRESSOR INLET

Inlet Evaporator Temperature °F	Suction Pressure (psia)		Density 90°F Inlet Temperature		ρ_1/P_1	
	R-12	MR (40/60)	R-12	MR	R-12	MR
-23	14.2	16.7	.292	.340	.0205	.0204
-18	15.9	19.3	.335	.40	.0210	.0208
-17	16.3	19.6	.350	.42	.0215	.0214
-10	19.1	22.5	.399	.47	.0210	.0210
- 8	20.0	24.0	.422	.52	.0211	.0214

o <u>Fluid enthalpy</u>.

MR have higher liquid-to-vapor change enthalpies than SR which can result in lower refrigerant mass flow rates and possibly reduced compressor power at equivalent refrigeration rates. The relations taken from the duPont mixture data used to characterize the refrigerants in the analysis were developed from curve fitting duPont refrigerant property data to produce the following lines which have an accuracy of about 2%:

$$\Delta h_r \text{ (R12) Btu/lb} = 74.32 - .119\ P_h \text{ (psia)} \qquad [2]$$

$$\Delta h_r \text{ (40/60 MR) Btu/lb} = 83.174 - .1489\ P_h \text{ (psia)} \qquad [3]$$

where P_h = condensing pressure in psia and the Δ enthalpies are based on the return gas to the compressor at 90°F.

- Heat exchanger effectiveness.

  The MR changes the effectiveness of the heat exchangers as a result of heat capacity and heat transfer coefficient changes. The literature suggests that the MR has a lower heat transfer coefficient than SR. Based on the evaporator test data we have obtained, it does not appear that the lower coefficient is significant enough to measure its effect on the evaporator refrigerant-to-air heat exchange.

Both evaporators are approximated by two-phase refrigerant flow. The heat exchanger metal temperature was found to be close to the evaporating refrigerant temperature in the freezer (T4 in Figure 4) and was observed to be close to the superheat vapor temperature (T4 + SH in Figure 4) in the fresh food evaporator because of the lower fin effectiveness of the fresh food evaporator. The radiative view factor of the fresh food evaporator is 1.0 since it is entirely enclosed by the fresh food compartment, hence, it also has an emissivity identical to 1.0. The freezer evaporator has no radiative contribution to the freezer compartment. The relationship between heat exchanger effectiveness and refrigeration capacity for the system shown in Figure 4 is given below.

$$Q = \underbrace{E1(T1-T4) \times \dot{m}Cp}_{\text{Freezer Hx}} + \underbrace{E2(T2-T4-SH) \times \dot{m}}_{\text{Fresh Food Hx}} + \underbrace{3.28 \times 1.71 \times 10^{-8} \times (T2^4 - (T4+SH)^4)}_{\text{Fresh Food Radiation}} \quad [4]$$

where:

Q	=	refrigeration capacity in Btu/hr (Q_1 = single refrigerant; Q_2 = mixed refrigerant)
E1, E2	=	heat exchanger effectiveness for low and high temperature evaporators, respectively
T1, T2	=	air temperatures (°F)
T4, T5, T7	=	refrigerant temperatures (°F)
$\dot{m}Cp$	=	air mass flow rate x heat capacity (Btu/hr °F) for SR ($\dot{m}1$); refrigerant mass flow rate x heat capacity (Btu/hr°F) for MR ($\dot{m}2$)
SH	=	degrees of superheat (°F) ($T7 - T7_{(sat)}$)
3.28	=	fresh food evaporator surface area

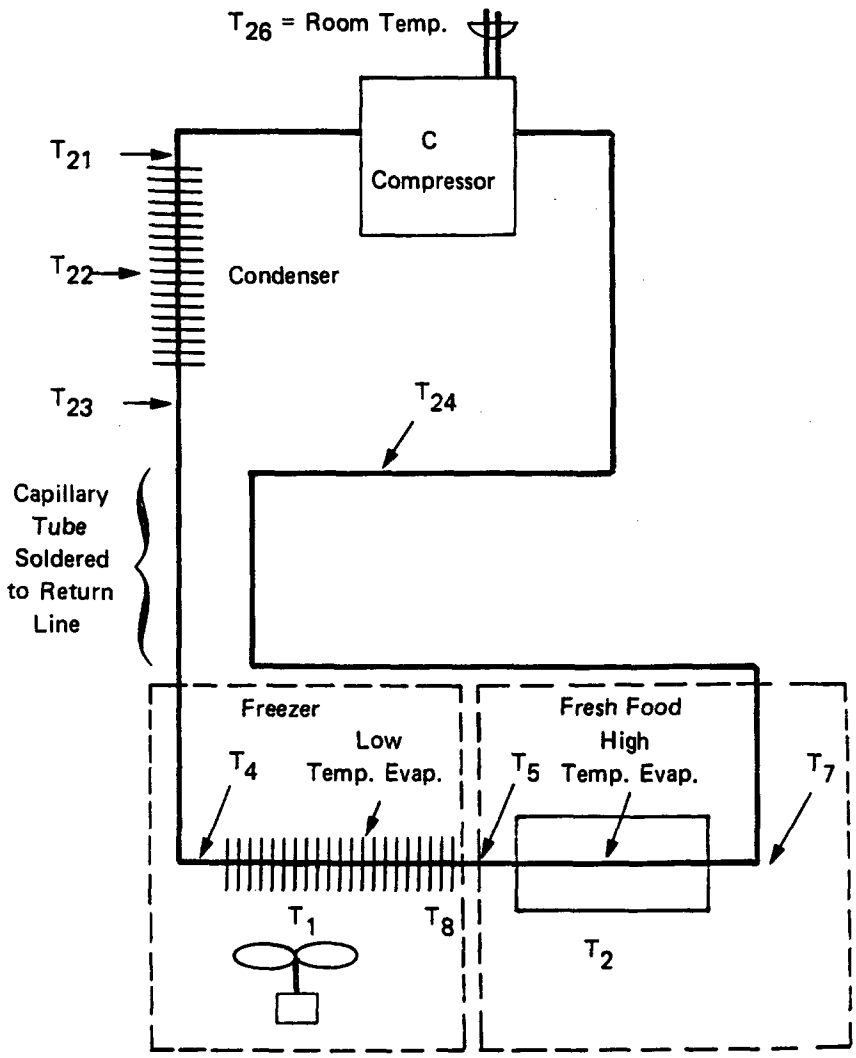

FIGURE 4 REFRIGERATION SYSTEM TEMPERATURES

The compressor energy consumption can be written as

$$W_1 = \dot{m}\ [N/(N-1)]\ P_1/\rho_1\ ((P_h/P_1)^{(N-1)/N}-1) \qquad [5]$$

where

$\dot{m}$ = refrigerant mass flow (lbm/hr)

N = polytropic coefficient of compressor = 1.03 for both SR and MR based on duPont property data and suction and discharge conditions shown in Table 2.

$P_1 P_h$ = low side and head pressure respectively (PSIA)

ρ_1 = suction gas density (lbs/cu ft)

The mass flow rate $\dot{m}$ is given by

$\dot{m}$ = ρ_1 x volumeric efficiency x displacement (ft^3/hr)

= ρ_1 x 0.8 x 28.5

$$= 22.82\ \rho_1 \qquad [6]$$

where the volumetric efficiency was derived from the compressor manufacturer's data [9] on R12 operation.

The proper solution of these plus detailed condenser and refrigerant inventory equations requires a computer model (more than 20 equations and variables), and is beyond the scope of this project. Therefore, a simple "cold end" computer model (without condenser or compressor equations) was developed using the above relations as well as the equations defining the enthalpy change of the refrigerant through the evaporators.

B. SIMULATION

A simple simulation based on the dual evaporator configuration and the equations presented in the previous section was developed. The high side pressure was chosen as an independent variable; therefore, the effect of condenser design cannot be predicted. The solution schematic is shown in Figure 5. A series of freezer evaporator temperatures are selected until the heat flow to the refrigerant in the evaporators is equal to the heat flow to the air passing over the evaporators.

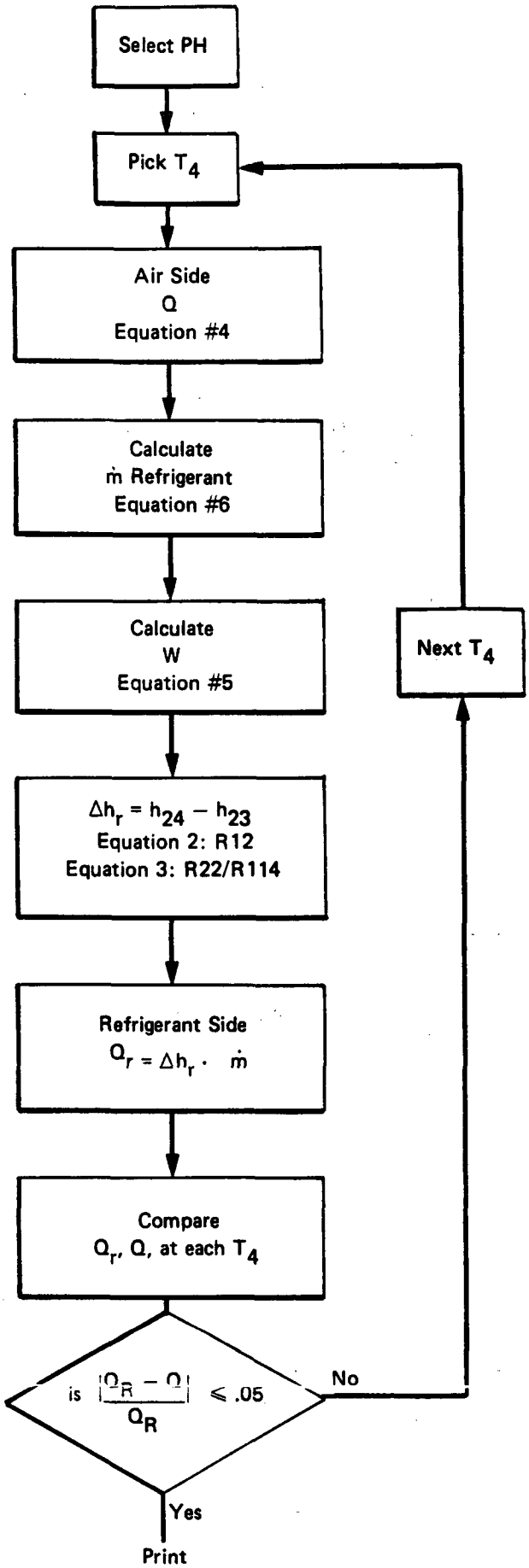

FIGURE 5 SIMPLE COMPUTER MODEL FLOWCHART

C. PARAMETRIC ANALYSIS

o Freezer Heat Exchanger Effectiveness

With SR, the effectiveness of the low temperature evaporator is governed by the air flow heat capacity and is constant:

$$1 - \exp(-35.3/52.1) = 0.492 \qquad [7]$$

where

35.3 = overall heat transfer coefficient for this freezer evaporator (Btu/hr°F) based on Amana test data (actual UA)

52.1 = freezer air flow heat capacity x freezer air flow (42.1 CFM) x heat capacity (Btu/hr°F) at 0°F

The experimental data for the single refrigerant, R12, agrees well with the expected performance (see Table 5) where the actual refrigeration is taken from the calorimeter tests and the experimental effectiveness is defined as the actual refrigeration divided by the maximum heat transfer possible (52.1 x (T8-T4)).

Table 5

EXPERIMENTAL EFFECTIVENESS WITH R12

R12 Test Date	Q_{Actual}	52.1(T8-T4)	Experimental Effectiveness (%)
12/16/83	300	615	48.8
12/19/83	297	599	49.6

The expected effectiveness for a 40/60 mixture (Test 1/19/84) is governed by the following relations.

$$\text{NTU} = \text{number of transfer units} = UA/(\dot{m}Cp)_{min} \qquad [8]$$

where

UA = 35.3 Btu/hr°F (ignoring differences between single and mixed refrigerant heat transfer coefficients; an assumption discussed later)

$(\dot{m}Cp)_{min}$ = calculated MR heat capacity @ 20 psia (2.22 Btu/°F- lbm) x refrigerant mass flow of (8.75 lbm/hr) = 19.42 Btu/°F- hr

The Cp is calculated by the relationship of $\Delta h_{MR}/\Delta T$.

Heat transfer effectiveness as a function of number of transfer units and capacity rate ratio; crossflow exchanger with one fluid mixed. [7]

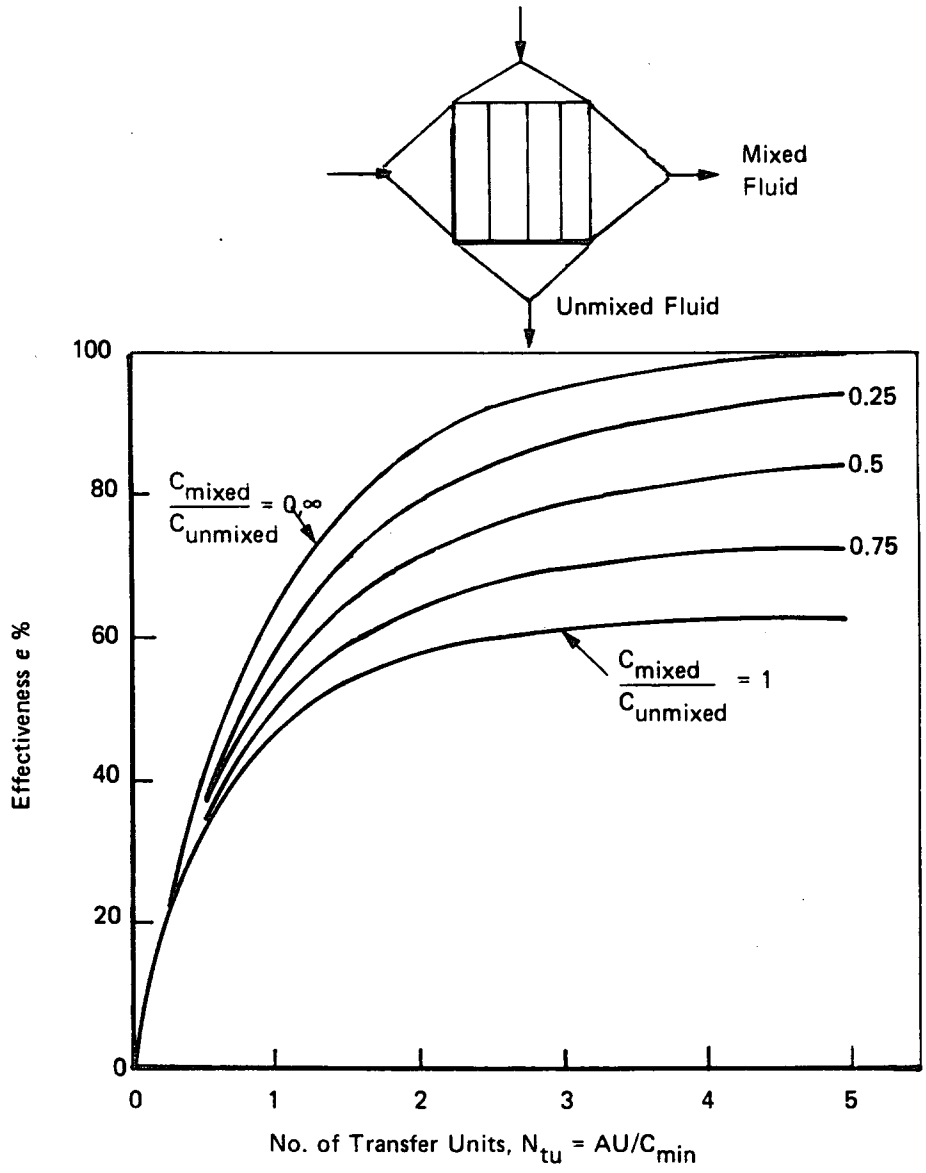

FIGURE 6 CROSSFLOW HEAT EXCHANGER

Therefore, NTU = 1.82

and

$$(\dot{m}Cp)_{MR}/(\dot{m}Cp)_{AIR} = 19.42/52.1 = .37 \qquad [9]$$

From Figure 6, the expected effectiveness is 75%.

The empirical effectiveness for the MR is given in Table 6 and is close to the predicted value.

Table 6

EXPERIMENTAL EFFECTIVENESS WITH MR

Test	Q_{Actual}	2.22 $\dot{m}$(T8-T4)	Experimental Effectiveness (%)
1/19	309	412	75.0

The agreement is sufficient to justify the use of the effectiveness relationship and to justify our previous assumption that the difference between MR and SR heat transfer coefficients is insignificant insofar as the measured performance is concerned. However, the results are limited in scope by the lack of modelling the condenser performance.

b. Fresh Food Heat Exchanger

The fresh food heat exchanger cannot be treated solely by an effectiveness-NTU relationship due to the large amount of radiant heat transfer to the surrounding warm walls. The effect of radiation must be included as shown in Equation 4. The effectiveness of the mixed refrigerant and single refrigerant are nearly the same for the fresh food evaporator since the air side of the heat exchanger limits the heat transfer.

$$\text{NTU (SR)} = \text{NTU (MR)} = UA/mCp)_{min} = \frac{0.83 \times 2 \times 3.28 \times (38-6)}{87.7} = 1.98 \qquad [10]$$

where mCp was estimated from laboratory tests in which the fresh food heat exchanger delivered 87.7 Btu/hr of cooling to the air (another 83.3 Btu/hr was radiation), which cooled to 6°F from 38°F. The overall heat transfer coefficient was estimated to be 0.83 Btu/hr-ft^2°F based on a measured face velocity of 1 ft/sec and a two-sided plate area of 3.28 ft^2.

$$\text{Effectiveness, SR } (C_{min}/C_{max} = 0) = 0.854 \qquad [11]$$

as taken from Figure 6.

Effectiveness, MR (C_{min}/C_{max} = 2.74/19.42 = .14) = 0.83 [12]

as taken from Figure 6.

In the SR situation, there was liquid refrigerant entering the fresh food evaporator with a transition to total superheat (gas) somewhere in the heat exchanger. If the entire fresh food heat exchanger had superheated refrigerant flow then NTU = 2.56; C_{min}/C_{max} = .42 and E(SR) = .78. The actual overall effectiveness is between .78 and .85; we choose to model the evaporator with SR as though it were entirely two-phase flow in order to conservatively estimate the benefit of MR.

With the evaporator effectiveness established, the computer model can be run to check it against the baseline data (Table 2). Table 7 shows the results of the simulation. The initial analysis is of the baseline SR Refrigerant and 40/60 (R22/R114) mixture. The predicted mass flow, refrigeration capacity and compressor power agrees well with the summary values of experimental data shown in Table 1, Section II.

A true cycle simulation must have the following features not contained in this simple model:

- A complete precooler, compressor case and condenser heat transfer model.
- Partitioning of heat exchangers into superheat and subcooling sections (except the low temperature evaporator).
- A fluid inventory model - mass conservation.

Inclusion of these features was beyond the scope of this activity. In the simple model developed, the condenser is decoupled from the system, and the high side pressure is selected as an independent variable. The analytic study focussed on the evaporator configuration since it is where the benefit of MR was expected.

Changes to the evaporators were simulated. An increase in the air flow rate from 42 CFM to 100 CFM would reduce the SR evaporator effectiveness from .49 to .25 but increase the heat transport capability (effectiveness x heat flow capacity) by 21%. The MR heat exchanger would also benefit from increased air flow rate. Table 7 shows a 2% improvement for MR and a 4% improvement for SR brought about by the increased airflow. The simulation shows that increased air flow rate does not improve the relative performance of SR and MR.

In the case of the SR, the high side pressure and the condensing temperatures are uniquely related by saturation conditions of temperature and pressure. For the mixed refrigerant, the pressure and

Table 7

SIMULATED PERFORMANCE

Variable	T_4 (°F)	Refrigerant Flow $\dot{m}$ (lb/hr)	COP	Q (Btu/hr)	Compressor Power (Watts)	kWh/Day*
Baseline						
MR E_1 = .83	-16	8.1	.93	478	150	1.92
SR E_1 = .49	- 7	8.3	.92	485	155	1.95
Increase Evaporator Air Flow from 42.1 CFM to 100 CFM, Phi = 160						
MR E_1 = .83	-15	8.24	.95	490	152	1.90
SR E_1 = .25	- 5.5	8.95	.93	490	152	1.88
Increase Evaporator Area 20%						
MR E_1 = .55	-15	8.24	.94	489	152	1.90
SR E_1 = .82	- 6.5	8.75	.96	486	150	1.90
MR P_h = 160	-15.5	8.14	.931	482	151	1.92
SR P_h = 160	- 7	8.65	.92	475	150	1.92
MR P_h = 150	-16	8.05	.98	486	145	1.82
SR P_h = 150	- 7.5	8.55	.99	485	144	1.84
MR P_h = 140	-17	7.87	1.05	492	138	1.73
SR P_h = 140	- 7.5	8.55	1.03	489	139	1.73

*Based on simplified relationship

$$\frac{kWh}{Day} = .024 \left(\frac{212.3}{Q\text{-Internal Load}}\right) (w+e) = \frac{5.0952}{Q-41} (W+15)$$

where

Q	=	refrigeration capacity in (Btu/hr)
Internal Load	=	41 = fan-related heat load (Btu/hr)
w	=	compressor electric power (watts)
e	=	parasitic fan and defrost timer (watts)
212.3	=	steady cabinet heat load (Btu/hr)

temperature are composition dependent, and we can attempt to use this feature to advantage. Table 7 shows the anticipated benefit of reducing the high side pressure by adjusting the throttling. At 140 psia, the MR system is predicted to yield a COP = 1.05 versus an SR system value (at 160 psia high side pressure) of 0.92 or a 14% improvement in steady state efficiency and a 10% reduction in energy consumption. Whether the reduced head pressure (140 psia) could be achieved while maintaining the condensing temperature and, therefore, the heat rejection rate in the MR situation had to be demonstrated in the laboratory as examined in the next section.

IV. SYSTEM MODIFICATION TESTS

Using the results of the analytical model in conjunction with the laboratory testing, we postulated that modification of the expansion device offered promise for energy savings when using MR. The model predicted that equivalent refrigeration could be provided at decreased system pressure ratio, thus reducing compressor power requirements. Therefore, the portion of capillary tube not soldered to the suction line was removed and replaced with a precision needle valve. The results of testing with this arrangement are shown in Table 8. Initially, runs were made with R12 to find the valve position equivalent to the original capillary tube. The needle valve subsequently was adjusted to vary system pressure ratio. The range of adjustment was limited by other constraints, namely condenser performance. When the valve was opened to reduce the pressure ratio, the condenser was unable to subcool the increased refrigerant flow rate. This lead to the formation of vapor bubbles at the condenser outlet. Attempts to eliminate the problem by increasing refrigerant charge until liquid was observed at the compressor suction inlet did not significantly alleviate the problem. The range of pressure ratio adjustment was therefore limited and did not appreciably affect performance at the higher refrigerant flow rates where we had expected improvement.

Table 8

CALORIMETRIC TEST RESULTS
Expansion Valve

Date	Refrigerant	Charge (Oz)	Delivered Refrigeration (Btu/hr)	Calculated Load (Btu/hr)	Compressor Power (Watts)	Mass Flow (Lbs/hr)	Phi (psia)	Plow (psia)	TEMPERATURE Ambient (°F)	FZ Bulk (°F)	FF Bulk (°F)	EER
4/9/84	R12	8.5 oz	486.2	494.8	156.7	8.5	148	18.2	89.6	2.6	37.7	3.10
4/10/84	R12	8.5 oz	459.4	494.8	153.1	7.8	153	17.7	91.2	5.2	41.7	3.00
4/11/84	R12	8.5 oz	510.8	526.8	162.0	9.0	153	19.4	92.0	4.5	36.9	3.15
4/13/84	R12	8.5 oz	483.6	500.9	156.6	8.5	151	18.3	91.2	3.5	39.0	3.09
4/17/84	R12	8.5 oz	438.5	451.0	146.7	7.4	152	16.6	90.2	4.1	44.0	2.99
4/20/84	40/60 wt % R22/R114	10.5 oz	463.9	457.7	150.3	7.7	165	17.4	91.5	4.5	36.7	3.09
5/15/84	"	"	475.2	473.4	148.3	7.8	156	17.1	90.2	2.8	38.3	3.20
5/15/84	"	"	528.7	494.5	164.8	9.9	158	20.0	89.3	2.4	38.2	3.21
5/17/84	"	"	368.9	372.2	131.9	6.0	159	14.5	88.6	2.9	38.8	2.80

V. CONCLUSIONS

Simple substitution of 40/60 wt% and 60/40 wt% R22/R114 refrigerant mixtures for R12 in an Amana ESTR-18D two evaporator refrigerator-freezer showed no measurable energy savings benefit.

At equal suction pressures, the R12/R114 MR show a substantially lower refrigerant temperature. This would suggest greater refrigeration capacity per energy input (suction pressure is the primary determinant for compressor energy consumption). Enhanced heat exchanger effectiveness over SR should further favor the MR. However, the lower heat capacity rate ($\dot{m}Cp$) of the MR was observed to offset these benefits resulting in about the same heat transfer as shown in Table 9.

Table 9

COMPARATIVE HEAT TRANSFER IN LOW TEMPERATURE EVAPORATOR

	T_{IN} (°F)	$\dot{m}Cp$(min) (Btu/hr°F)	E_1	Q(5° freezer)* (Btu/hr)
MR	-16	19.4	0.75	305
SR	- 7	52.1	0.49	306

For the condenser, the MR appears to have less desirable characteristics than the SR. In the tests summarized in Table 1, the 40/60 MR has higher condensing pressures for lower condensing temperatures which means that the compressor must work slightly harder with a MR to create the necessary heat rejection temperatures.

Providing forced air over the condenser would only worsen the prospects for MR by increasing the minimum heat capacity (and effectiveness) of the SR and not improving the MR heat exchanger performance (the minimum heat capacity for MR is the refrigerant not the air).

Tests with the 40/60 mixture of R22/R114 showed that the evaporator performance closely correlated with NTU-effectiveness relations and revealed no significant difference in refrigerant heat transfer coefficients between R12 and the mixture.

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*$Q = \dot{m}cp\ E_1(5-T_{IN})$

In summary, expected benefits in evaporator effectiveness were observed, but were offset by reduced heat capacity and density effects, resulting in no apparent increase in refrigeration efficiency. Modifications to the expansion valve did not improve MR performance in the ESTR-18D demonstrator and no savings were predicted through evaporator modifications.

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APPENDIX A

LISTING OF SIMPLE CYCLE MODEL

LISTING OF SIMPLE
CYCLE MODEL

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]LIST

5  REM  PROGRAM:MR3
10 T1 = 459 + 5:T2 = 459 + 37
20 TR = 90 + 459
30  REM  ASSUME FIXED CONDENSER P
     ERFORMANCE.
40  REM  CO=COMPRESSOR VOL.PARAME
     TER
50 CO = 22.82
60 SH = 5: REM  SR SUPERHEAT
70  REM  INPUT DATA*ON HEAT EXCHA
     NGERS
80 E1 = .49
90 M1 = .239 * .0863 * 42.1 * 60
100 E2 = .85
110 M2 = .239 * .076 * 2 * 60
120 E3 = .78
130 E4 = .83
140  REM  SELECT HIGH PRESSURE
150 PH = 160
160  REM  ENTHALPY DIFFERENCES
170 DH = 74.32 - .119 * PH
180 BH = 83.174 - .1489 * PH
190  REM  GAS CONSTANTS
200 R1 = .086
210 R2 = .086
220 A = .48421
230  FOR T4 = 441 TO 459 STEP 1
240  REM  INTERCHANGER HEAT LEAK
     QY
250 QY = 0
260  REM  SINGLE REFRIGERANT Q,P,
     M
270 Q1 = E1 * (T1 - T4) * M1 + E2
      * (T2 - T4 - SH) * M2 + 3.2
     8 * .171E - 8 * (T2 ^ 4 - (T
     4 + SH) ^ 4) + QY
280 P1 = 20.5 / 22.3 * (.51175 *
     T4 - 209)
290 P2 =  EXP (( - 5125.849 / T4)
      - 1.64059 *  LOG (T4) + 24.
     51755)
300 D1 = P1 / (R1 * 629)
310 M = D1 * CO
320 N = 1.1
330 W1 = A * M * (N / (N - 1)) *
     P1 / D1 * ((PH / P1) ^ ((N -
     1) / N) - 1)
340  REM  MIXED REFRIGERANT DENSI
     TY,M,Q
350 D2 = P2 / (629 * R2)
360 MR = D2 * CO
370 CR = MR * 2.5 / M1
380 NT = 32 / (MR * 2.5)
390 EX =  EXP ( - NT * (1 - CR))
400 T5 = T4 + E3 * (T1 - T4)
410 QX = 0
420 Q = E3 * (T1 - T4) * 2.5 * MR
      + E4 * M2 * (T2 - T5) + 3.2
     8 * .171E - 8 * (T2 ^ 4 - T5
      ^ 4) + QX
430 N = 1.25
440 W = A * 1 * (N / (N - 1)) * P
     2 / D2 * ((PH / P2) ^ ((N -
     1) / N) - 1)
450 W = .92 * W * MR
460  REM  FIX WATT EST. BY 8%
470  GOTO 500
480 D$ = ""; REM  D$=CONTROL D
490  PRINT D$;"PR#3"
500  PRINT T4 - 459;"F"
510  PRINT "E=";E1;"  ";E2;"  "E3
     ;"  ";E4
520  PRINT "AIR FLOW",M1;"  ";M2
530  PRINT "REFRIGERANT","SINGLE"
     ,"MIXED"
540  PRINT "INLET DENSITY",D1;"
     ";D2
550  PRINT "P(LO)",P1;"  ";P2
560  PRINT "MASS FLOW",M;"  ";MR
570  PRINT "COP",Q1 / W1;"  ";Q /
     W
580  PRINT "Q-HX",Q1;"  ";Q
590  PRINT "Q-REF",DH * M;"  ";BH
      * MR
600  PRINT "WATTS",W1 / 3.41;"  "
     ;W / 3.41
610  PRINT
620  IF J > 1 THEN  END
630  NEXT T4
```

APPENDIX B

MIXED REFRIGERANT THERMODYNAMIC PROPERTIES

**** NOTE **** THIS TABLE IS ARRANGED IN EVEN INCREMENTS OF PRESSURE AND GIVES BOTH THE BUBBLE POINT AND DEW POINT TEMPERATURES CORRESPONDING TO EACH PRESSURE. THE BUBBLE TEMPERATURE, VOLUME, DENSITY, ENTHALPY AND ENTROPY REFER TO THE BUBBLE POINT* FOR THE 40.00 WEIGHT PERCENT R22 MIXTURE, AND THE VAPOR TEMPERATURE, VOLUME, DENSITY, ENTHALPY AND ENTROPY REFER TO THE DEW POINT* FOR THE 40.00 WEIGHT PERCENT R22 MIXTURE.

**** THE VAPOR PROPERTIES ARE NOT FOR THE VAPOR IN EQUILIBRIUM WITH THE 40.00/60.00 WEIGHT PERCENT LIQUID.

SATURATED PROPERTIES OF FREON R22 / R114 — 40.00 WEIGHT PERCENT R22

PRESSURE PSIA	PSIG	TEMPERATURE (F) LIQUID	VAPOR	VOLUME (CU FT/LB) LIQUID	VAPOR	DENSITY (LB/CU FT) LIQUID	VAPOR	ENTHALPY (BTU/LB) LIQUID	LATENT	VAPOR	ENTROPY (BTU/LB-R) LIQUID	VAPOR
3.46	-8.23	-60.08	-24.1	.01023	5.7906	97.80	.1727	-4.70	83.67	78.97	-.0115	.1900
5.00	-5.79	-52.3	-16.4	.01030	4.7474	97.09	.2106	-2.90	82.93	80.03	-.0070	.1890
10.00	-4.70	-43.9	-8.0	.01038	3.8561	96.31	.2593	-.92	82.11	81.19	-.0022	.1881
12.00	-2.70	-36.7	-.9	.01046	3.2533	95.64	.3074	.77	81.40	82.17	.0018	.1874
14.00	-.70	-30.4	5.4	.01052	2.8173	95.04	.3549	2.25	80.78	83.04	.0053	.1866
16.00	1.30	-24.8	11.0	.01058	2.4869	94.50	.4021	3.59	80.22	83.81	.0084	.1864
18.00	3.30	-19.7	16.1	.01064	2.2275	94.01	.4489	4.80	79.71	84.50	.0111	.1860
20.00	5.30	-15.0	20.7	.01069	2.0183	93.55	.4955	5.91	79.23	85.14	.0136	.1857
22.00	7.30	-10.7	25.0	.01074	1.8457	93.12	.5418	6.95	78.78	85.73	.0159	.1854
24.00	9.30	-6.6	29.0	.01078	1.7010	92.72	.5879	7.91	78.36	86.27	.0181	.1852
26.00	11.30	-2.8	32.7	.01083	1.5777	92.34	.6338	8.82	77.96	86.78	.0201	.1850
28.00	13.30	.7	36.3	.01087	1.4714	91.98	.6796	9.68	77.58	87.26	.0221	.1848
30.00	15.30	4.1	39.6	.01091	1.3788	91.64	.7253	10.50	77.22	87.71	.0237	.1847
32.00	17.30	7.3	42.8	.01095	1.2974	91.31	.7708	11.27	76.87	88.14	.0253	.1845
34.00	19.30	10.4	45.8	.01099	1.2251	91.00	.8162	12.02	76.53	88.55	.0269	.1844
36.00	21.30	13.4	48.7	.01103	1.1606	90.69	.8616	12.73	76.20	88.93	.0284	.1842
38.00	23.30	16.2	51.4	.01106	1.1027	90.40	.9069	13.41	75.89	89.30	.0299	.1842
40.00	25.30	18.9	54.1	.01110	1.0503	90.12	.9521	14.07	75.58	89.66	.0312	.1841
42.00	27.30	21.5	56.6	.01113	1.0027	89.85	.9973	14.71	75.29	90.00	.0325	.1840
44.00	29.30	24.0	59.1	.01116	.9593	89.58	1.0424	15.33	74.99	90.32	.0338	.1840
46.00	31.30	26.4	61.4	.01120	.9195	89.32	1.0875	15.92	74.71	90.63	.0350	.1839
48.00	33.30	28.8	63.7	.01123	.8829	89.07	1.1326	16.50	74.43	90.93	.0362	.1839
50.00	35.30	31.1	65.9	.01126	.8491	88.82	1.1777	17.07	74.16	91.23	.0374	.1838
52.00	37.30	33.3	68.1	.01129	.8178	88.59	1.2227	17.61	73.89	91.51	.0385	.1837
54.00	39.30	35.4	70.2	.01132	.7888	88.35	1.2678	18.15	73.63	91.78	.0395	.1837
56.00	41.30	37.5	72.2	.01135	.7617	88.13	1.3129	18.68	73.38	92.04	.0406	.1836
58.00	43.30	39.5	74.2	.01138	.7365	87.91	1.3579	19.17	73.12	92.29	.0416	.1836
60.00	45.30	41.5	76.1	.01140	.7129	87.68	1.4029	19.67	72.88	92.54	.0426	.1835
62.00	47.30	43.5	78.0	.01143	.6906	87.47	1.4480	20.15	72.63	92.78	.0435	.1835
64.00	49.30	45.3	79.3	.01146	.6698	87.26	1.4930	20.62	72.39	93.02	.0444	.1835
66.00	51.30	47.2	81.6	.01149	.6501	87.05	1.5381	21.09	72.16	93.24	.0453	.1834
68.00	53.30	49.0	83.3	.01151	.6316	86.85	1.5832	21.54	71.92	93.47	.0462	.1834
70.00	55.30	50.8	85.0	.01154	.6141	86.65	1.6284	21.99	71.69	93.68	.0471	.1834
72.00	57.30	52.5	86.7	.01157	.5975	86.45	1.6736	22.43	71.46	93.89	.0479	.1834
74.00	59.30	54.2	88.3	.01159	.5818	86.26	1.7188	22.86	71.24	94.09	.0488	.1833

*****ATTENTION***** THIS TABLE IS ARRANGED IN EVEN INCREMENTS OF PRESSURE AND GIVES BOTH THE BUBBLE POINT AND DEW POINT TEMPERATURES CORRESPONDING TO EACH PRESSURE. THE *LIQUID* TEMPERATURE, VOLUME, DENSITY, ENTHALPY AND ENTROPY REFER TO THE *BUBBLE POINT* FOR THE 40.00 WEIGHT PERCENT R22 MIXTURE, AND THE *VAPOR* TEMPERATURE, VOLUME, DENSITY, ENTHALPY AND ENTROPY REFER TO THE *DEW POINT* FOR THE 40.00 WEIGHT PERCENT R22 MIXTURE.

**** THE VAPOR PROPERTIES ARE NOT FOR THE VAPOR IN EQUILIBRIUM WITH THE 40.00/ 60.00 WEIGHT PERCENT LIQUID.

SATURATED PROPERTIES OF FREON R22 / R114 40.00 WEIGHT PERCENT R22

PRESSURE PSIA	PSID	TEMPERATURE (F) LIQUID	VAPOR	VOLUME (CU FT/LB) LIQUID	VAPOR	DENSITY (LB/CU FT) LIQUID	VAPOR	ENTHALPY (BTU/LB) LIQUID	LATENT	VAPOR	ENTROPY (BTU/LB/(R)) LIQUID	VAPOR
76.00	61.30	55.9	89.9	.01162	.5669	86.07	1.7640	23.28	71.01	94.29	.0496	.1833
78.00	63.30	57.5	91.5	.01164	.5527	85.88	1.8093	23.69	70.79	94.49	.0504	.1833
80.00	65.30	59.1	93.0	.01167	.5392	85.70	1.8546	24.10	70.58	94.68	.0512	.1833
82.00	67.30	60.6	94.5	.01169	.5263	85.52	1.9000	24.50	70.36	94.86	.0519	.1832
84.00	69.30	62.2	96.0	.01172	.5140	85.34	1.9454	24.90	70.15	95.04	.0527	.1832
86.00	71.30	63.7	97.4	.01174	.5023	85.16	1.9908	25.29	69.93	95.22	.0534	.1832
88.00	73.30	65.2	98.9	.01177	.4911	84.99	2.0363	25.67	69.72	95.40	.0541	.1832
90.00	75.30	66.7	100.3	.01179	.4803	84.81	2.0819	26.05	69.51	95.56	.0548	.1832
92.00	77.30	68.1	101.6	.01181	.4700	84.64	2.1275	26.42	69.31	95.73	.0555	.1831
94.00	79.30	69.5	103.0	.01184	.4602	84.47	2.1732	26.79	69.10	95.89	.0562	.1831
96.00	81.30	70.9	104.3	.01186	.4507	84.31	2.2189	27.15	68.90	96.05	.0569	.1831
98.00	83.30	72.3	105.6	.01188	.4416	84.14	2.2647	27.51	68.70	96.21	.0576	.1831
100.00	85.30	73.6	106.9	.01191	.4328	83.98	2.3106	27.87	68.50	96.36	.0582	.1831
102.00	87.30	75.0	108.2	.01193	.4244	83.81	2.3565	28.21	68.30	96.51	.0589	.1831
104.00	89.30	76.3	109.4	.01195	.4162	83.65	2.4025	28.56	68.10	96.66	.0595	.1830
106.00	91.30	77.6	110.7	.01198	.4084	83.50	2.4485	28.90	67.90	96.80	.0601	.1830
108.00	93.30	78.9	111.9	.01200	.4009	83.34	2.4946	29.24	67.70	96.94	.0607	.1830
110.00	95.30	80.1	113.1	.01202	.3936	83.18	2.5408	29.57	67.51	97.08	.0614	.1830
112.00	97.30	81.4	114.3	.01204	.3865	83.03	2.5871	29.90	67.32	97.22	.0620	.1830
114.00	99.30	82.6	115.4	.01207	.3797	82.87	2.6334	30.23	67.12	97.35	.0625	.1830
116.00	101.30	83.8	116.6	.01209	.3732	82.72	2.6799	30.55	66.93	97.48	.0631	.1829
118.00	103.30	85.0	117.7	.01211	.3668	82.57	2.7264	30.87	66.74	97.61	.0637	.1829
120.00	105.30	86.2	118.8	.01213	.3606	82.42	2.7729	31.18	66.55	97.74	.0643	.1829
122.00	107.30	87.4	119.9	.01215	.3547	82.27	2.8196	31.50	66.36	97.86	.0648	.1829
124.00	109.30	88.5	121.0	.01218	.3489	82.13	2.8663	31.81	66.16	97.98	.0654	.1829
126.00	111.30	89.7	122.1	.01220	.3433	81.98	2.9132	32.11	65.99	98.10	.0659	.1829
128.00	113.30	90.8	123.2	.01222	.3378	81.84	2.9601	32.42	65.80	98.22	.0665	.1829
130.00	115.30	91.9	124.2	.01224	.3326	81.69	3.0071	32.72	65.62	98.34	.0670	.1828
132.00	117.30	93.0	125.3	.01226	.3274	81.55	3.0541	33.02	65.43	98.45	.0676	.1828
134.00	119.30	94.1	126.3	.01228	.3224	81.41	3.1013	33.31	65.25	98.56	.0681	.1828
136.00	121.30	95.2	127.3	.01231	.3176	81.27	3.1486	33.61	65.07	98.67	.0686	.1828
138.00	123.30	96.3	128.3	.01233	.3129	81.13	3.1959	33.90	64.88	98.78	.0691	.1828
140.00	125.30	97.3	129.3	.01235	.3083	80.99	3.2434	34.18	64.70	98.89	.0696	.1828
142.00	127.30	98.4	130.3	.01237	.3039	80.85	3.2909	34.47	64.52	98.99	.0701	.1828
144.00	129.30	99.4	131.2	.01239	.2995	80.71	3.3386	34.75	64.34	99.09	.0706	.1827

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**** THE VAPOR PROPERTIES ARE NOT FOR THE VAPOR IN EQUILIBRIUM WITH THE 40.00/ 60.00 WEIGHT PERCENT LIQUID.

SATURATED PROPERTIES OF FREON R22 / R114 40.00 WEIGHT PERCENT R22

PRESSURE PSIA	PRESSURE PSIG	TEMPERATURE (F) LIQUID	TEMPERATURE (F) VAPOR	VOLUME (CU FT/LB) LIQUID	VOLUME (CU FT/LB) VAPOR	DENSITY (LB/CU FT) LIQUID	DENSITY (LB/CU FT) VAPOR	ENTHALPY (BTU/LB) LIQUID	ENTHALPY (BTU/LB) LATENT	ENTHALPY (BTU/LB) VAPOR	ENTROPY (BTU/LB/RA) LIQUID	ENTROPY (BTU/LB/RA) VAPOR
146.00	131.30	100.4	132.2	.01241	.2953	80.57	3.3863	35.04	64.16	99.19	.0711	.1827
145.00	133.30	101.5	133.1	.01243	.2912	80.44	3.4341	35.32	63.98	99.29	.0716	.1827
150.00	135.30	102.5	134.1	.01245	.2872	80.30	3.4821	35.59	63.80	99.39	.0721	.1827
155.00	140.30	105.0	136.4	.01251	.2776	79.97	3.6025	36.28	63.35	99.63	.0733	.1827
160.00	145.30	107.4	138.6	.01256	.2686	79.64	3.7234	36.95	62.91	99.86	.0745	.1826
165.00	150.30	109.7	140.8	.01261	.2601	79.31	3.8450	37.61	62.47	100.09	.0756	.1826
170.00	155.30	112.1	143.0	.01266	.2521	78.98	3.9673	38.26	62.03	100.29	.0767	.1825
175.00	160.30	114.3	145.1	.01271	.2445	78.66	4.0903	38.90	61.59	100.49	.0778	.1825
180.00	165.30	116.6	147.1	.01276	.2373	78.35	4.2140	39.53	61.16	100.69	.0789	.1824
185.00	170.30	118.7	149.1	.01282	.2305	78.03	4.3385	40.15	60.73	100.88	.0799	.1824
190.00	175.30	120.9	151.1	.01287	.2240	77.72	4.4637	40.76	60.30	101.06	.0810	.1823
195.00	180.30	123.0	153.0	.01292	.2179	77.41	4.5897	41.36	59.87	101.23	.0820	.1823
200.00	185.30	125.0	154.9	.01297	.2120	77.10	4.7165	41.96	59.44	101.40	.0830	.1822
205.00	190.30	127.1	156.7	.01302	.2064	76.79	4.8442	42.55	59.01	101.56	.0839	.1822
210.00	195.30	129.1	158.5	.01307	.2011	76.49	4.9729	43.13	58.58	101.71	.0849	.1821
215.00	200.30	131.0	160.3	.01313	.1960	76.19	5.1018	43.70	58.16	101.87	.0859	.1821
220.00	205.30	132.9	162.1	.01318	.1911	75.88	5.2323	44.27	57.73	102.01	.0868	.1820
225.00	210.30	134.8	163.8	.01323	.1864	75.58	5.3634	44.84	57.31	102.14	.0877	.1820
230.00	215.30	136.7	165.5	.01328	.1820	75.28	5.4955	45.39	56.88	102.28	.0886	.1819
235.00	220.30	138.6	167.1	.01334	.1777	74.98	5.6286	45.94	56.46	102.40	.0895	.1818
240.00	225.30	140.4	168.7	.01339	.1735	74.69	5.7626	46.49	56.03	102.52	.0904	.1818
245.00	230.30	142.2	170.3	.01344	.1696	74.39	5.8976	47.03	55.61	102.64	.0913	.1817
250.00	235.30	143.9	171.9	.01350	.1657	74.09	6.0337	47.57	55.18	102.75	.0922	.1816
255.00	240.30	145.7	173.5	.01355	.1621	73.79	6.1707	48.10	54.76	102.86	.0930	.1816
260.00	245.30	147.4	175.0	.01361	.1585	73.50	6.3089	48.63	54.33	102.96	.0939	.1815
265.00	250.30	149.1	176.5	.01366	.1551	73.20	6.4482	49.15	53.91	103.06	.0947	.1814
270.00	255.30	150.7	178.0	.01372	.1518	72.90	6.5886	49.67	53.48	103.15	.0955	.1813
275.00	260.30	152.4	179.4	.01377	.1486	72.60	6.7301	50.19	53.05	103.24	.0964	.1813
280.00	265.30	154.0	180.8	.01383	.1455	72.30	6.8728	50.70	52.62	103.32	.0972	.1812
285.00	270.30	155.6	182.3	.01389	.1425	72.00	7.0168	51.21	52.19	103.40	.0980	.1811
290.00	275.30	157.2	183.7	.01395	.1396	71.70	7.1619	51.71	51.76	103.47	.0988	.1810
295.00	280.30	158.8	185.0	.01401	.1368	71.40	7.3084	52.22	51.33	103.55	.0996	.1809
300.00	285.30	160.3	186.4	.01407	.1341	71.09	7.4561	52.72	50.90	103.61	.1003	.1808
305.00	290.30	161.9	187.7	.01413	.1315	70.79	7.6052	53.21	50.46	103.67	.1011	.1807
310.00	295.30	163.4	189.0	.01419	.1289	70.48	7.7556	53.71	50.02	103.73	.1019	.1806

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**** THE VAPOR PROPERTIES ARE NOT FOR THE VAPOR IN EQUILIBRIUM WITH THE 40.00/ 60.00 WEIGHT PERCENT LIQUID.

SATURATED PROPERTIES OF FREON R22 / R114 40.00 WEIGHT PERCENT R22

PRESSURE PSIA	PRESSURE PSIG	TEMPERATURE (F) LIQUID	TEMPERATURE (F) VAPOR	VOLUME (CU FT/LB) LIQUID	VOLUME (CU FT/LB) VAPOR	DENSITY (LB/CU FT) LIQUID	DENSITY (LB/CU FT) VAPOR	ENTHALPY (BTU/LB) LIQUID	ENTHALPY (BTU/LB) LATENT	ENTHALPY (BTU/LB) VAPOR	ENTROPY (BTU/(LB)(R)) LIQUID	ENTROPY (BTU/(LB)(R)) VAPOR
315.00	300.30	164.9	190.3	.01425	.1265	70.17	7.9075	54.20	49.59	103.79	.1027	.1905
320.00	305.30	166.4	191.6	.01431	.1241	69.86	8.0607	54.69	49.15	103.84	.1034	.1904
325.00	310.30	167.9	192.9	.01438	.1217	69.54	8.2155	55.18	48.71	103.88	.1042	.1903
330.00	315.30	169.3	194.1	.01445	.1194	69.22	8.3718	55.66	48.26	103.92	.1049	.1902
335.00	320.30	170.8	195.4	.01451	.1172	68.90	8.5296	56.15	47.82	103.96	.1057	.1901
340.00	325.30	172.2	196.6	.01458	.1151	68.58	8.6891	56.63	47.37	104.00	.1064	.1900
345.00	330.30	173.6	197.8	.01465	.1130	68.25	8.8502	57.11	46.92	104.03	.1071	.1799
350.00	335.30	175.0	199.0	.01472	.1110	67.91	9.0129	57.59	46.47	104.05	.1079	.1798
360.00	345.30	177.8	201.3	.01487	.1070	67.23	9.3444	58.54	45.55	104.09	.1093	.1795
370.00	355.30	180.5	203.6	.01503	.1033	66.52	9.6826	59.49	44.63	104.12	.1107	.1793
380.00	365.30	183.2	205.8	.01520	.0997	65.78	10.0297	60.44	43.70	104.13	.1122	.1790
390.00	375.30	185.8	208.0	.01538	.0963	65.01	10.3831	61.38	42.75	104.13	.1136	.1787
400.00	385.30	188.4	210.1	.01558	.0931	64.18	10.7464	62.32	41.78	104.10	.1150	.1784

FREON R22 / R114 SUPERHEATED VAPOR 40.00 WEIGHT PERCENT R22

	18.00 PSIA (16.07 F)			20.00 PSIA (20.71 F)			22.00 PSIA (25.00 F)			24.00 PSIA (28.99 F)			
	V	H	S	V	H	S	V	H	S	V	H	S	
	2.2275	84.50	.1860	2.0183	85.14	.1857	1.8457	85.73	.1854	1.7010	86.27	.1852	
20.	2.2478	85.11	.1873	.0000	.00	.0000	.0000	.00	.0000	.0000	.00	.0000	20.
30.	2.2992	86.68	.1905	2.0616	86.60	.1887	1.8671	86.51	.1870	1.7050	86.43	.1855	30.
40.	2.3506	88.26	.1937	2.1081	88.18	.1919	1.9096	88.10	.1902	1.7442	88.02	.1887	40.
50.	2.4017	89.85	.1968	2.1544	89.77	.1950	1.9520	89.70	.1934	1.7833	89.62	.1919	50.
60.	2.4528	91.46	.2000	2.2006	91.39	.1982	1.9943	91.31	.1965	1.8223	91.24	.1950	60.
70.	2.5037	93.09	.2031	2.2467	93.02	.2013	2.0364	92.94	.1996	1.8612	92.87	.1981	70.
80.	2.5545	94.73	.2061	2.2927	94.66	.2043	2.0784	94.59	.2027	1.8999	94.52	.2012	80.
90.	2.6052	96.39	.2092	2.3385	96.32	.2074	2.1204	96.25	.2058	1.9385	96.19	.2043	90.
100.	2.6558	98.06	.2122	2.3843	97.99	.2104	2.1622	97.93	.2088	1.9770	97.86	.2073	100.
110.	2.7063	99.75	.2152	2.4300	99.68	.2134	2.2039	99.62	.2118	2.0155	99.56	.2103	110.
120.	2.7567	101.45	.2181	2.4756	101.39	.2164	2.2455	101.33	.2148	2.0538	101.26	.2133	120.
130.	2.8071	103.16	.2211	2.5211	103.11	.2193	2.2871	103.05	.2177	2.0921	102.99	.2162	130.
140.	2.8574	104.90	.2240	2.5665	104.84	.2222	2.3286	104.78	.2206	2.1303	104.72	.2191	140.
150.	2.9076	106.64	.2269	2.6119	106.59	.2251	2.3700	106.53	.2235	2.1684	106.47	.2220	150.
160.	2.9577	108.40	.2297	2.6572	108.35	.2280	2.4113	108.29	.2264	2.2064	108.24	.2249	160.
170.	3.0078	110.17	.2326	2.7025	110.12	.2308	2.4526	110.07	.2292	2.2444	110.01	.2278	170.
180.	3.0578	111.96	.2354	2.7476	111.91	.2336	2.4938	111.86	.2320	2.2823	111.80	.2306	180.
190.	3.1078	113.76	.2382	2.7928	113.71	.2364	2.5350	113.66	.2348	2.3202	113.61	.2334	190.
200.	3.1577	115.57	.2410	2.8378	115.52	.2392	2.5761	115.47	.2376	2.3580	115.42	.2361	200.
210.	3.2076	117.40	.2437	2.8828	117.35	.2419	2.6171	117.30	.2404	2.3957	117.25	.2389	210.
220.	3.2574	119.24	.2464	2.9278	119.19	.2447	2.6582	119.14	.2431	2.4334	119.10	.2416	220.
230.	3.3072	121.09	.2491	2.9727	121.04	.2474	2.6991	121.00	.2458	2.4711	120.95	.2443	230.
240.	3.3569	122.95	.2518	3.0176	122.91	.2501	2.7400	122.86	.2485	2.5087	122.82	.2470	240.
250.	3.4066	124.83	.2545	3.0625	124.78	.2527	2.7809	124.74	.2511	2.5463	124.69	.2497	250.
260.	3.4562	126.71	.2571	3.1073	126.67	.2554	2.8218	126.63	.2538	2.5838	126.58	.2523	260.
270.	3.5059	128.61	.2597	3.1520	128.57	.2580	2.8626	128.53	.2564	2.6213	128.48	.2550	270.
280.	3.5554	130.52	.2623	3.1968	130.48	.2606	2.9033	130.44	.2590	2.6588	130.40	.2576	280.
290.	3.6050	132.44	.2649	3.2415	132.40	.2632	2.9441	132.36	.2616	2.6962	132.32	.2601	290.
300.	3.6545	134.37	.2675	3.2861	134.33	.2657	2.9848	134.29	.2641	2.7336	134.25	.2627	300.
310.	3.7040	136.31	.2700	3.3308	136.27	.2683	3.0254	136.23	.2667	2.7710	136.20	.2652	310.
320.	3.7534	138.26	.2725	3.3754	138.22	.2708	3.0661	138.19	.2692	2.8083	138.15	.2678	320.
330.	3.8029	140.22	.2750	3.4200	140.19	.2733	3.1067	140.15	.2717	2.8456	140.11	.2703	330.
340.	3.8523	142.19	.2775	3.4645	142.16	.2758	3.1473	142.12	.2742	2.8829	142.09	.2728	340.
350.	3.9016	144.18	.2800	3.5090	144.14	.2782	3.1878	144.11	.2767	2.9202	144.07	.2752	350.
360.	3.9510	146.17	.2824	3.5536	146.13	.2807	3.2284	146.10	.2791	2.9574	146.06	.2777	360.
370.	4.0003	148.16	.2848	3.5980	148.13	.2831	3.2689	148.10	.2815	2.9946	148.06	.2801	370.
380.	4.0496	150.17	.2872	3.6425	150.14	.2855	3.3094	150.11	.2839	3.0318	150.07	.2825	380.
390.	4.0989	152.19	.2896	3.6869	152.16	.2879	3.3498	152.12	.2863	3.0689	152.09	.2849	390.
400.	4.1482	154.21	.2920	3.7313	154.18	.2903	3.3903	154.15	.2887	3.1061	154.12	.2873	400.