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THE SIMULATION AND PERFORMANCE OF A SYSTEM USING AN R-12/R-114 REFRIGERANT MIXTURE

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Prepared by
UNIVERSITY OF ILLINOIS AT URBANA-CHAMPAIGN
URBANA, ILLINOIS 61801

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ABSTRACT

THE SIMULATION AND PERFORMANCE OF A SYSTEM USING AN R-12/R-114 REFRIGERANT MIXTURE

The overriding objective of this combined analytical and experimental study is to determine energy conservation possibilities through the use of a refrigerant mixture in a refrigerator or heat pump. The system is one with two evaporators that are operated at different temperatures levels, and the basic approach was to maintain constant refrigeration loads and temperatures throughout a series of tests wherein the composition of the refrigeration mixture was altered.

The experimental measurements consisted of the determination of temperatures and pressures throughout the system, energy flow rates and compressor power, and boiling and condensing heat-transfer coefficients as the fraction of R-114 in the mixture varied from 0.0 to 0.6. In order to maintain the fixed refrigeration capacity in the two evaporators, the refrigerant flow rate was adjusted by means of a variable-speed compressor.

The refrigerant temperatures increased progressively through the evaporator and decreased through the condenser as is expected for a refrigerant mixture.

System simulation predicted an 8 percent saving in compressor power for an optimum 68 percent R-114 mixture (of R-12/R-114) in comparison to R-12 alone. The experimental results at this stage show about a 2 percent reduction in power in the range 10-15 percent R-114 in the mixture. The reason for the difference between the simulated and experimental results is

the reduction in heat-transfer coefficients in the condenser and evaporator. The drop in condensing coefficients was indeed confirmed by the measurements to be between 10 and 30 percent depending on the position in the condenser. The evaporator tubes in the test system are larger than optimal, resulting in low refrigerant velocities. The measured boiling coefficients did not decrease with the introduction of the refrigerant mixture, but the order of magnitudes of all boiling coefficients were lower than desired. The observations suggest that different optimum velocities that acknowledge the tradeoff between pressure drop and heat-transfer coefficients prevail for mixtures in contrast to single refrigerants.

CHAPTER 1

INTRODUCTION

1.1 Reason for using Refrigerant Mixtures

The interest in using nonazeotropic mixtures of refrigerants stems from the fact that constant pressure evaporation or condensation of a non-azeotropic mixture occurs with a change in temperatures. Azeotropic mixtures or single refrigerants would evaporate or condense at a constant temperature. When counterflow evaporators or condensers are used in a system with a non-azeotropic refrigerant mixture, the temperature profiles will be more parallel than they would be in a conventional system (Figure 1.1). Using parallel temperature profiles lowers the thermodynamic irreversibility of the system.

Reversible heat transfer occurs if the temperature difference between the two fluids is zero and the surface area approaches infinity. Large temperature differences result in large irreversibilities. Figure 1.1 shows a comparison between refrigerants when the required refrigeration duty is the same. The external fluid enters and exits at the same temperatures in both cases. But in the case of a pure refrigerant, there is a large difference between the inlet temperature of the external fluid and the exit temperature of the evaporating refrigerant. The refrigerant mixture (ideally chosen here to have a temperature profile parallel to the external fluid) has a constant difference in temperature between itself and the external fluid. Assuming constant area, heat transfer rate, and heat transfer coefficient, this constant temperature difference is given by:

$$\Delta T = T_A - T_{2'} = T_B - T_{1'}$$

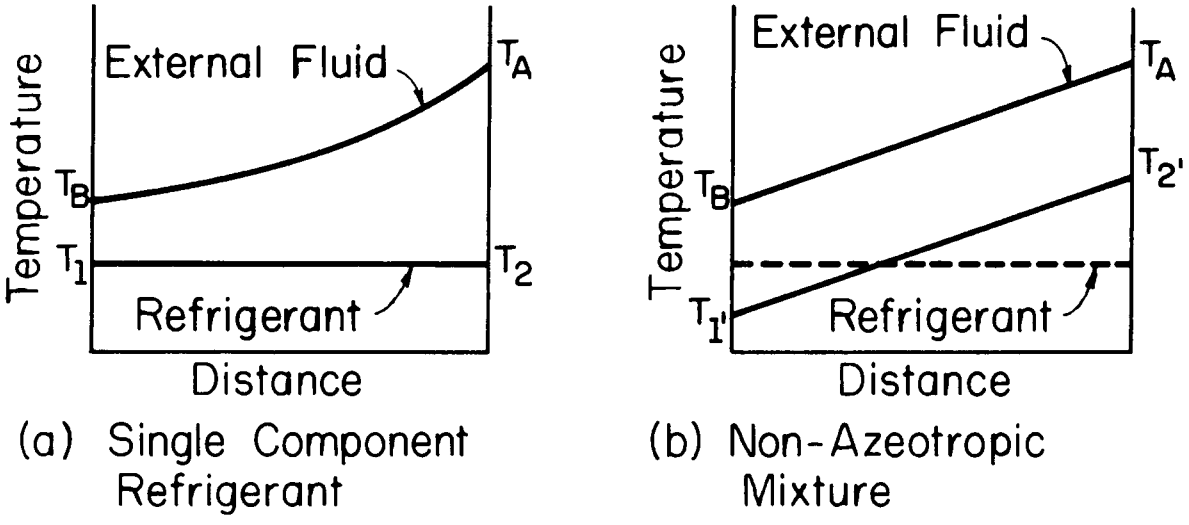


Figure 1.1 Evaporator Temperature Profiles in a Counterflow Heat Exchanger

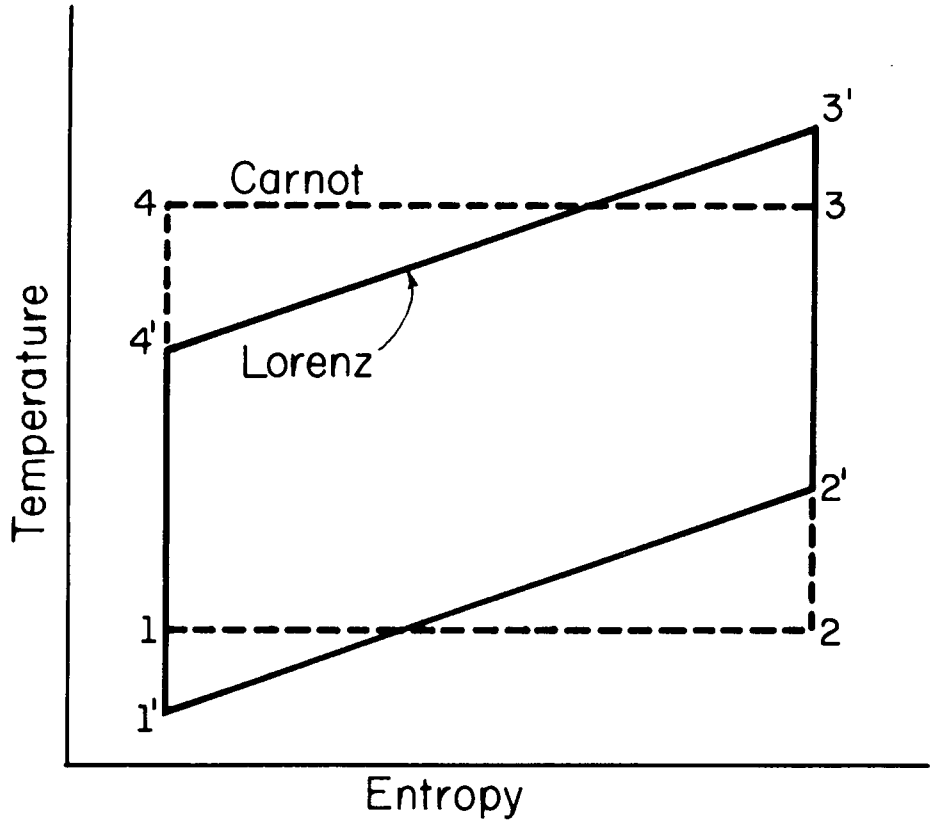


Figure 1.2 Carnot Cycle and Lorenz Cycle on a Temperature-Entropy Diagram

which equals

$$\frac{(T_A - T_2) - (T_B - T_1)}{\ln[(T_A - T_2)/(T_B - T_1)]}$$

in Fig. 1.1a.

In order for the refrigeration duties in the two cases to be identical, the following must be true:

$$U \times A \times \Delta T = U \times A \times \frac{(T_A - T_2) - (T_B - T_1)}{\ln[(T_A - T_2)/(T_B - T_1)]}$$

where U is the overall heat transfer coefficient and A is the heat transfer area.

If we look at both refrigeration cycles on a temperature-entropy diagram (assuming isentropic compression and expansion in both cases) and include the condensation processes, we can compare efficiencies by comparing areas (Figure 1.2). The cycle for the pure refrigerant is a Carnot cycle, while the cycle for the mixture is a Lorenz cycle [1]. A measure of refrigerating efficiency is the coefficient of performance (COP). It is defined as the refrigerating capacity divided by the power consumption. Although the area under the cycle diagram is not to scale, the coefficients of performance would be calculated as follows:

$$COP_{Carnot} = \frac{\text{Area under } 1 - 2}{\text{Area } 1 - 2 - 3 - 4} \quad COP_{Lorenz} = \frac{\text{Area under } 1' - 2'}{\text{Area } 1' - 2' - 3' - 4'} .$$

Therefore, the system employing a nonazeotropic mixture could theoretically have a higher COP. Unfortunately, the savings expected are lowered somewhat by the reduction in the heat transfer coefficients as the two refrigerants are mixed. This will be discussed in the next section.

1.2 Description of Mixture Properties

Thermodynamic properties of mixtures are much easier to define if the components mix ideally (i.e., obey Raoult's rule and Dalton's law of partial pressures). Fortunately, mixtures of refrigerants R-12 and R-114 can be described quite well using Raoult's rule. This is in part due to the fact that they are both non-polar [2]. If one of the refrigerants were polar, Raoult's rule would not predict properties very well. Equations used to calculate mixture properties for both the simulation program and for the data reduction program are based primarily on Raoult's rule and Dalton's law.

A table of temperature-pressure-concentration properties of the mixture R-12/R-114 is included as Appendix A; however, Figure 1.3 shows the temperature-concentration relationship at two pressure levels. This will give the reader a general idea of the relationship between temperature, pressure and concentration. As the pressure increases, the temperature difference between saturated liquid and saturated vapor (the bulge in the envelope) decreases. For the mixture R-12/R-114, the change in the maximum temperature difference (widest part of the envelope) is approximately 1°C per 500 kPa for the pressure range encountered during refrigeration processes (100 to 1000 kPa).

Heat transfer properties of the mixture R-12/R-114 have been measured by Arora [3], who found that the convective heat transfer coefficient of mixtures of the two refrigerants was lower than the concentration-weighted average of the heat transfer coefficient of each pure component. The exact physical mechanism responsible for this degradation of the mixture heat transfer coefficient is currently under investigation. This phenomenon should be addressed in any attempt to model refrigerant mixture performance. Some of Arora's results are presented in Figures 1.4 and 1.5.

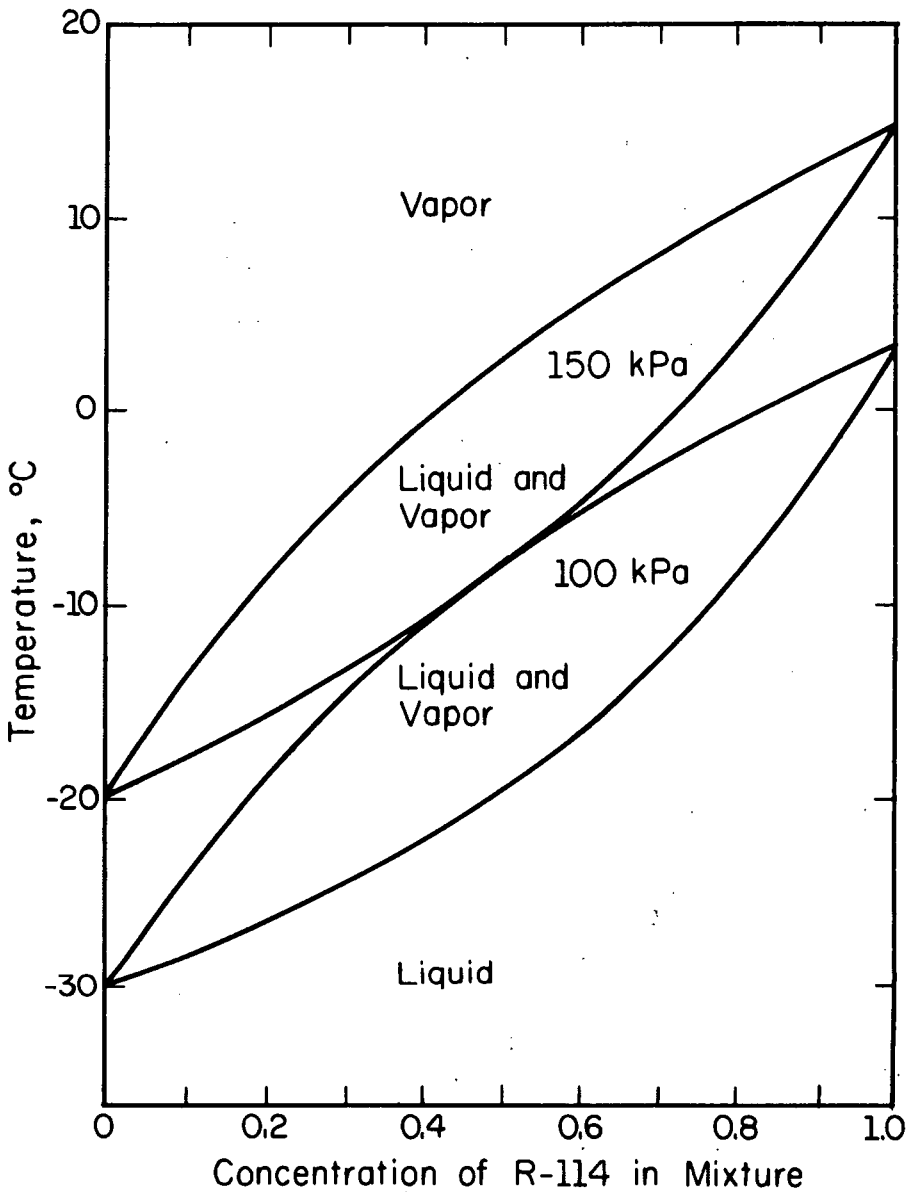


Figure 1.3 Temperature-Pressure-Concentration Diagram for R-114/R-12 Mixture

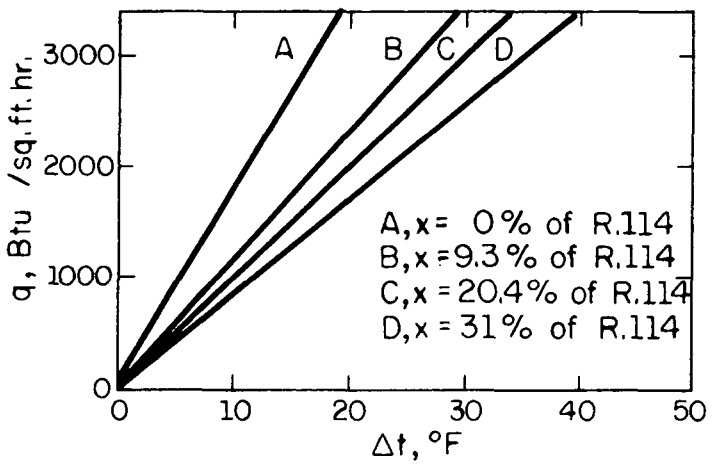


Figure 1.4 Heat Flux for Boiling as a Function of Temperature Difference [3]

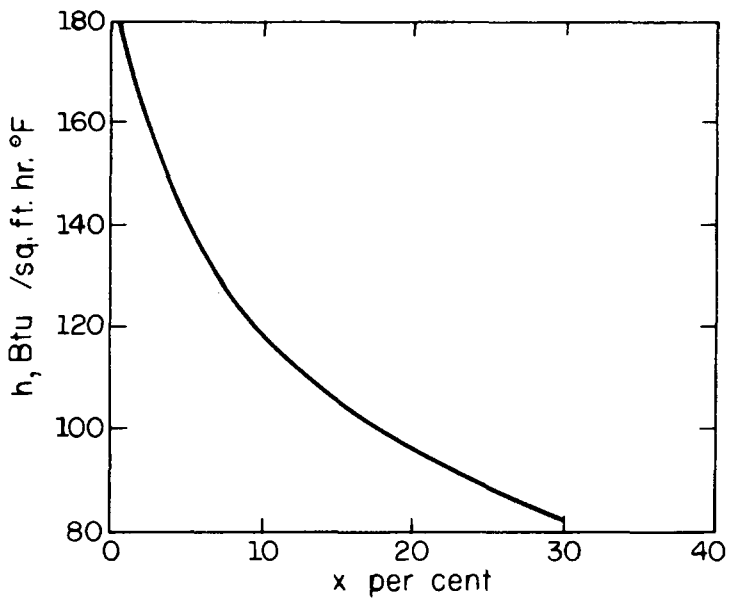


Figure 1.5 Boiling Heat Transfer Coefficient as a Function of Concentration of R-114 in the R-12/R-114 Mixture [3]

As a result, some of the savings expected because of the thermodynamic advantages of the mixture will be diminished by the degradation of heat transfer properties. Another penalty to the mixture is the increase in specific volume of the vapor on the suction side of the compressor when a low-density refrigerant is added to the original one. For a given refrigeration duty, the mixture of R-114 and R-12 will have to be at a lower evaporating pressure than would pure R-12, resulting in greater specific volume. As a result, either the compressor must be run at greater speed or a larger-displacement compressor must be used.

1.3 *Review of Work done by Philippe Launay*

The main focus of the work done by Philippe Launay [4] was development of a computer simulation model of a two-evaporator, two-condenser, two-heat exchanger, vapor compression refrigeration system using a mixture of refrigerants R-12 and R-114. His simulation model was incorporated into an optimization program which computed an optimum distribution of heat transfer areas among the heat exchanging components while keeping the sum of those areas constant. His results indicated that an optimized system using a mixture of R-12 and R-114 would require 9 percent less energy to run the compressor than an optimized system employing pure R-12. A test facility was constructed to demonstrate Launay's results, but no conclusive data had been collected by the time his thesis was completed.

1.4 Objectives of this Study

The major goals of this phase of the refrigerant mixture research were (1) to verify experimentally the possibility of energy conservation, and (2) to bring the simulation model into agreement with experimental data.

CHAPTER 2

TEST FACILITY

The results of the simulation study conducted by Philippe Launay [4] indicated that refrigerant mixtures could be used to save energy in a vapor-compression refrigeration system. An initial test facility had been constructed to demonstrate these results. Through the use of thermocouples, flow meters and a dynamometer, the system would be operated so as to duplicate refrigeration duty for various mixtures while recording the power necessary to perform that duty. However, Launay was unable to confirm the results of the simulation studies. The problems encountered were as follows:

(a) Excessive refrigerant pressure drops, especially in the evaporators.

(b) Inappropriate sizing of refrigerant and brine flow meters.

(c) Excessive refrigerant leaks past epoxy-sealed thermocouples.

(d) Inability to accurately control condenser water flow rates.

(e) No control over condenser water inlet temperatures.

(f) Insufficient refrigerant pressure taps

(g) No refrigerant oil separator after the compressor.

(h) Difficulty in obtaining accurate temperature measurements of the evaporator brine solutions.

For these reasons, the test facility was revised.

2.1 Description of Revised Test Facility

An overall diagram of the revised facility (Figure 2.1) shows that the original two-condenser, two-evaporator configuration was retained. However, some of the components were completely rebuilt. For clarity, components will be described as they appear in the refrigerant flow path.

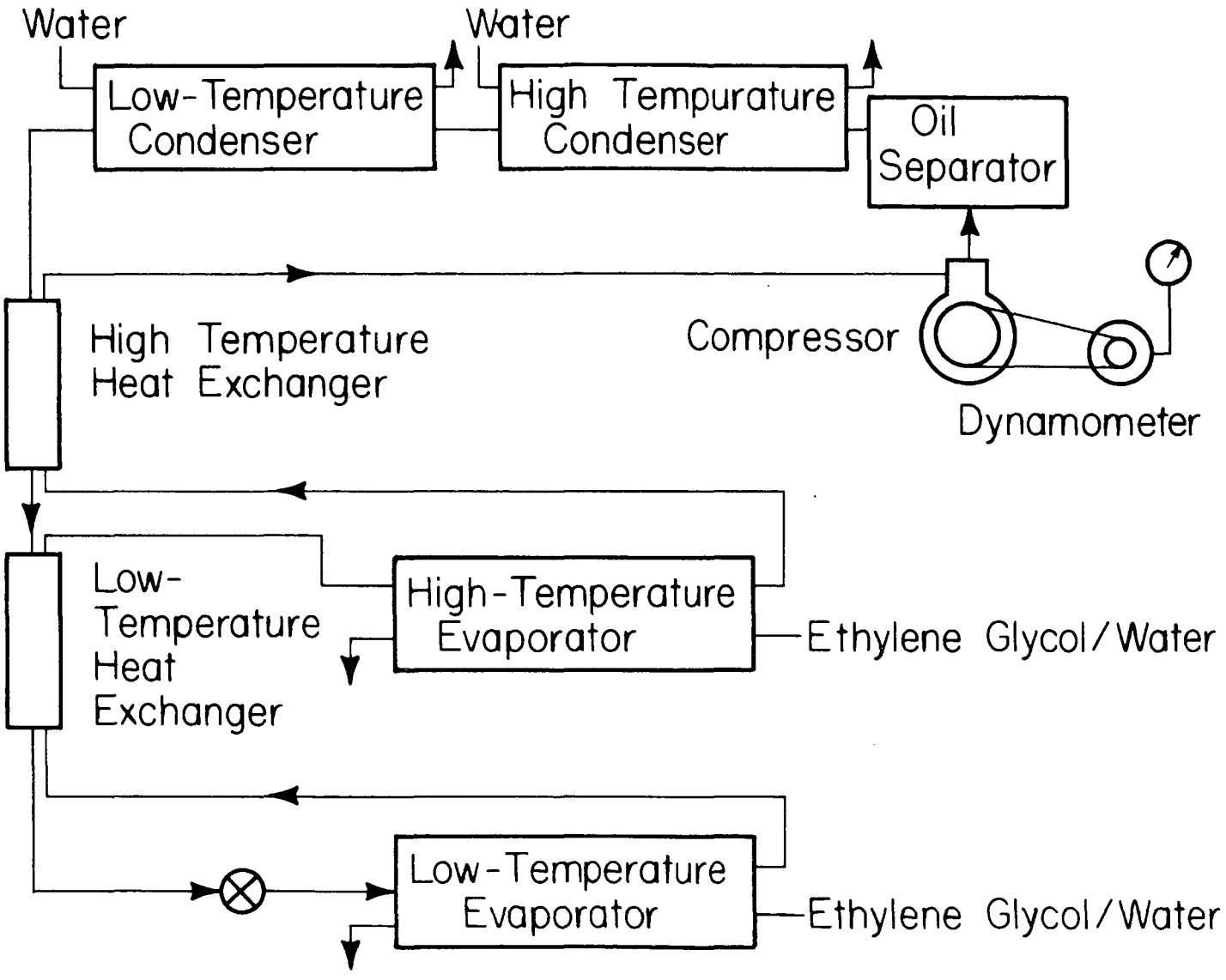


Figure 2.1 Diagram of Test Facility

2.2 Compressor and Oil Separator

The compressor is a Tecumseh Model CK, open type, with three cylinders (bore: 3.25 in. (82.6 mm), stroke: 3.0 in. (76.2 mm)), driven by three V-belts coming from a General Electric model 5T207 A6 NT48-34 variable-speed dc dynamometer. The dynamometer has a spring scale and a tachometer to allow for measurement of power input to the compressor. Vibration isolators are used in both the suction and discharge lines to prevent refrigerant leaks. A pressure relief valve is located immediately downstream from the vibration isolator in the discharge line. A Temprite Model 502 oil separator was installed downstream from the relief valve (Figure 2.2).

2.3 Condensers

The two condensers are identical in construction. Each is a tube-in-tube counterflow heat exchanger with water flowing in the annulus. The condensers are serpentine, each consisting of three passes (Figure 2.3). Water and refrigerant thermocouples are located at the end of each pass. A refrigerant thermocouple is located in the middle of each pass along with a thermocouple soldered to the outside surface of the refrigerant pipe. These last two thermocouples measure temperatures used in the calculation of heat transfer coefficients. Details are shown in Figure 2.4.

The tubes are made of copper with dimensions given in Table 2.1. Silver solder was used to join the tubing. Pressure taps are located before and after each condenser. The piping for the cooling water is constructed to allow the water from the low-temperature condenser (LTC) to flow directly into the high-temperature condenser (HTC), thereby forming one continuous condenser. However, valves permit changing the water flow so that each

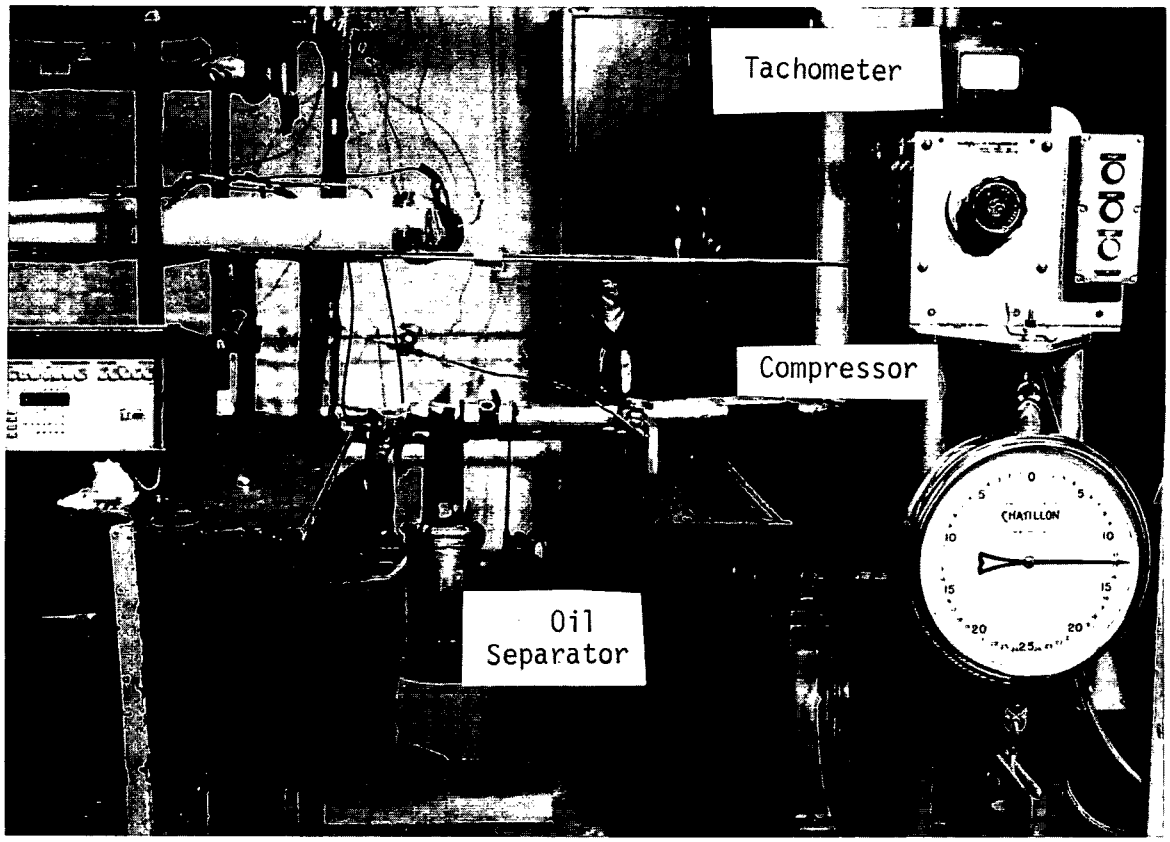


Figure 2.2 Photograph showing Compressor, Dynamometer and Oil Separator

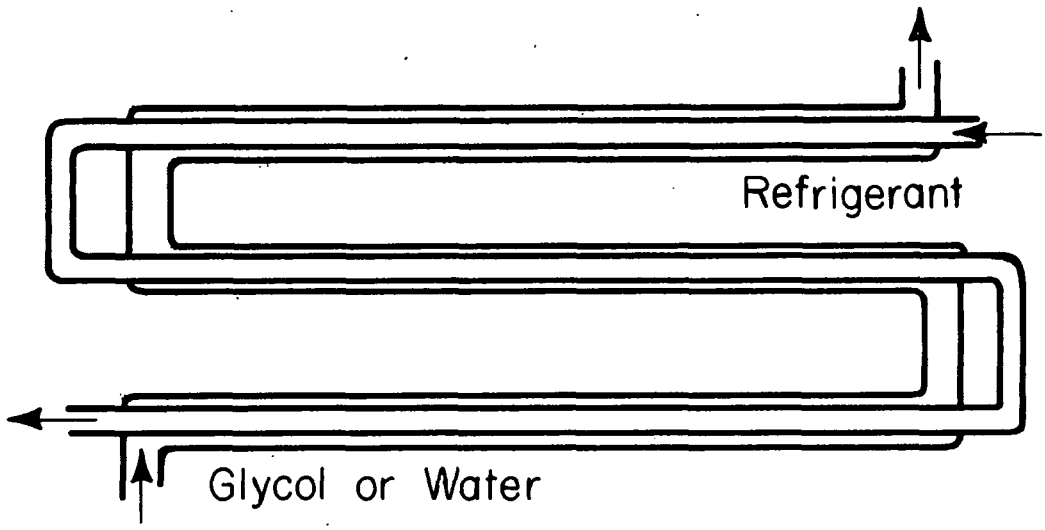


Figure 2.3 Condenser Configuration

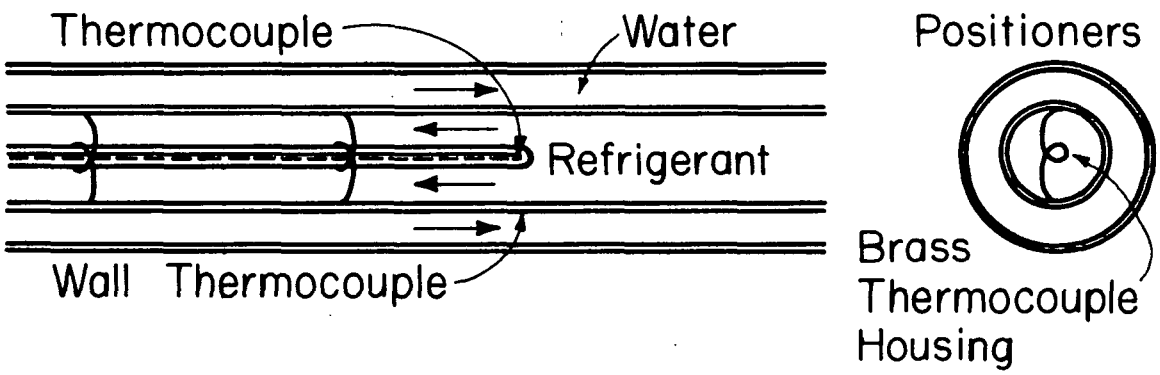


Figure 2.4 Thermocouple Positions at Center of Condenser Pass

condenser can be used independently (Figure 2.5). To provide control over the inlet water temperature, a separate heat exchanger uses warm discharge water to pre-heat inlet water. A bypass circuit is used to control the amount of warm water going to the heat exchanger, thereby providing temperature control. Brooks Model 1307 rotameters are used to measure the inlet water flow rate through each condenser. A refrigerant reservoir of approximately 250 cc with a sight glass at either end to permit verification of saturated liquid conditions is located after the LTC.

2.4 Refrigerant to Refrigerant Heat Exchangers.

The high-temperature heat exchanger (HX1) prevents saturated refrigerant from entering the compressor while slightly subcooling the liquid refrigerant. The low-temperature heat exchanger (HX2) subcools the liquid refrigerant even further, but a bypass circuit can be used to deactivate this component. Both heat exchangers are counterflow, tube-in-tube, with the liquid refrigerant flowing in the annulus. Dimensions of both are given in Table 2.1. Thermocouples are located before and after each heat exchanger.

2.5 Expansion Valve

Upstream of the expansion valve, the refrigerant passes through a filter-drier, then a rotameter which measures the liquid flow rate. A 3/8 in. (0.952 cm) Marsh needle valve (Model N-1313-FFC) is used as the expansion valve. Pressure taps on both sides of the valve are used to measure the pressure drop across the valve. The fitting used to charge the system is located downstream of the expansion valve.

Table 2.1 Specifications of condensers, evaporators, and refrigerant-to-refrigerant heat exchangers

Component	Number of passes	Length of refrigerant section of each pass, m	ID and OD of outer tube, mm	ID and OD of inner tube, mm	Refrigerant-side heat-transfer area m^2
Low temperature evaporator	3	1: 3.683 2: 3.683 3: 1.854	ID = 31.75 OD = 34.93	ID = 19.05 OD = 22.23	0.5518
High temperature evaporator	2	3.0734	ID = 31.75 OD = 34.93	ID = 19.05 OD = 22.23	0.3679
Low-temperature condenser	3	3.1242	ID = 25.40 OD = 28.58	ID = 12.70 OD = 15.88	0.3739
High-temperature condenser	3	3.1242	ID = 25.40 OD = 28.58	ID = 12.70 OD = 15.88	0.3739
Low-temperature heat exchanger	1	1.549	ID = 31.75 OD = 34.93	ID = 19.05 OD = 22.22	Inner: 0.0927 Outer: 0.1082
High-temperature heat exchanger	1	1.549	ID = 31.75 OD = 34.93	ID = 19.05 OD = 22.22	Inner: 0.0927 Outer: 0.1082

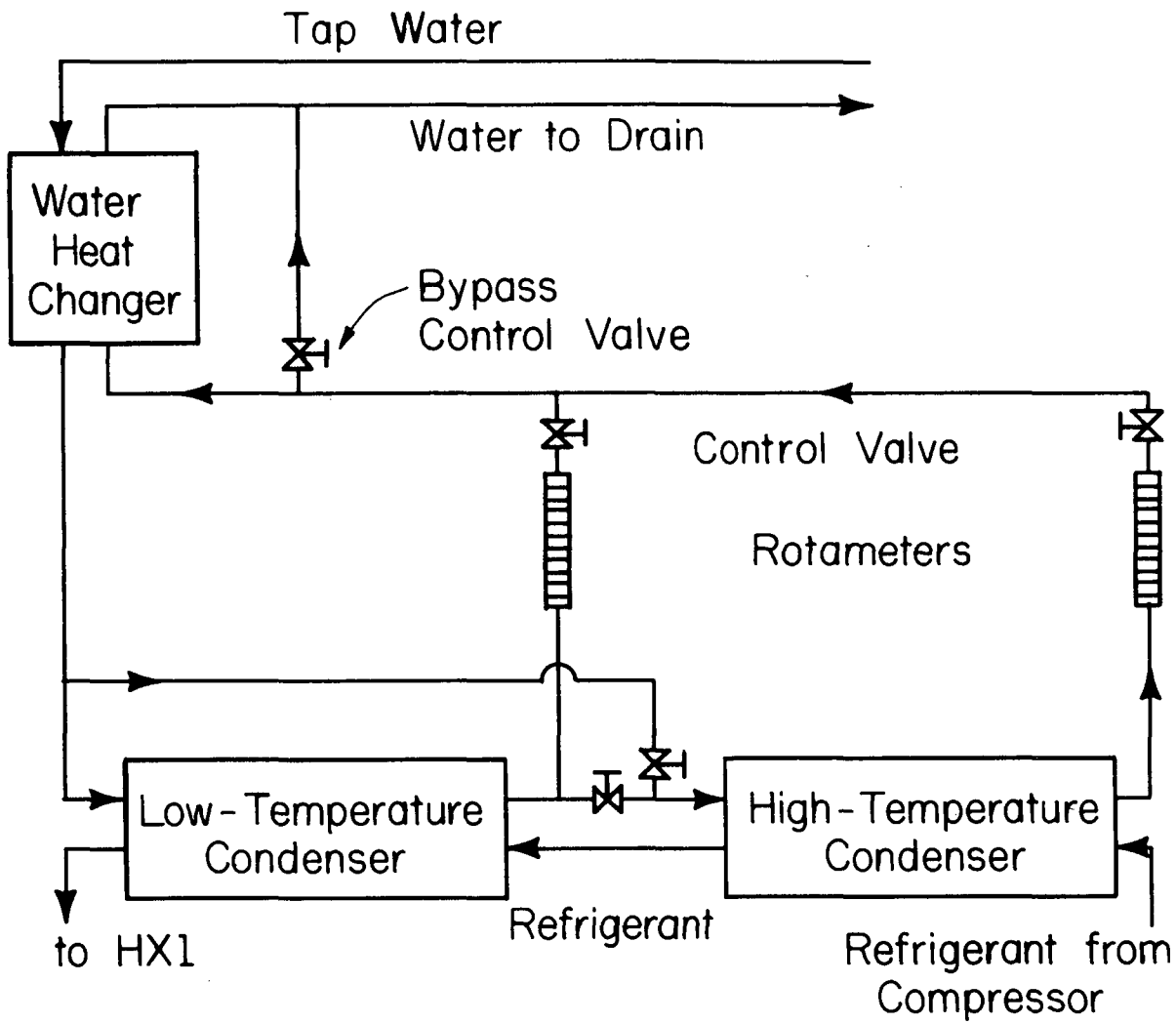


Figure 2.5 Condenser Water Control Arrangement

2.6 Low-Temperature Evaporator

The low-temperature evaporator (LTE) consists of two and one-half passes of a tube-in-tube heat exchanger. A mixture of ethylene glycol and water, referred to as glycol is the remainder of this report, flows in the annulus. Thermocouples are located at both ends and in the middle of each pass. To reduce the refrigerant pressure drop in the evaporator, the refrigerant thermocouple in the middle of each pass is positioned by a steel wire instead of the plastic positioners originally used (Figure 2.6). Dimensions are given in Table 2.1. The original 1/2 in. (1.27 cm) ID refrigerant piping has been replaced with 3/4 in. (1.905 cm) ID piping, which also helps to reduce the pressure drop. To improve the measurement of the glycol temperature, small diameter (3/8 in. (0.952 cm) OD) tubing was used to increase the glycol flow velocity before the thermocouple. This mixed the solution sufficiently to prevent the unsteady temperatures that had been recorded earlier. Insufficient mixing of the glycol had previously resulted in significant temperature stratification at the thermocouple location. The glycol is circulated by a 1/2 hp centrifugal pump. A variable transformer is used to control Chromalox heaters supplying the heat to the glycol solution. A rotameter measures the flow rate which is regulated by a valve.

2.7 High-Temperature Evaporator

The high-temperature evaporator (HTE) is constructed the same way as the low-temperature evaporator except that it only has two passes. The evaporator dimensions are listed in Table 2.1.

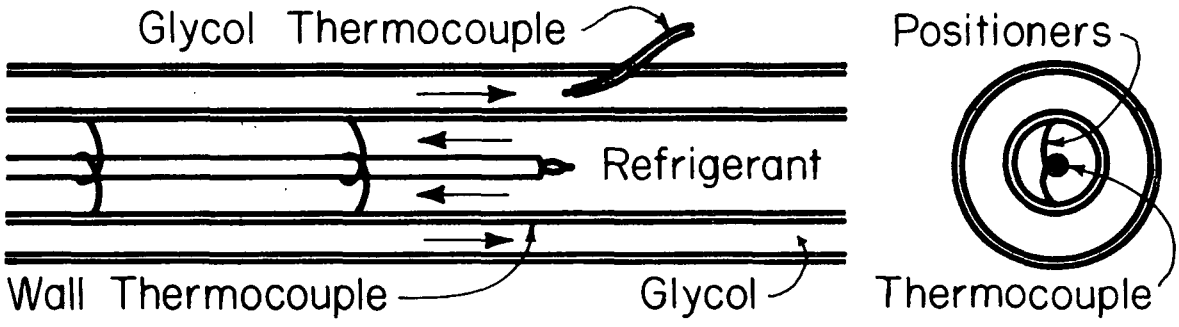


Figure 2.6 Thermocouple Locations at Center of each Evaporator Pass

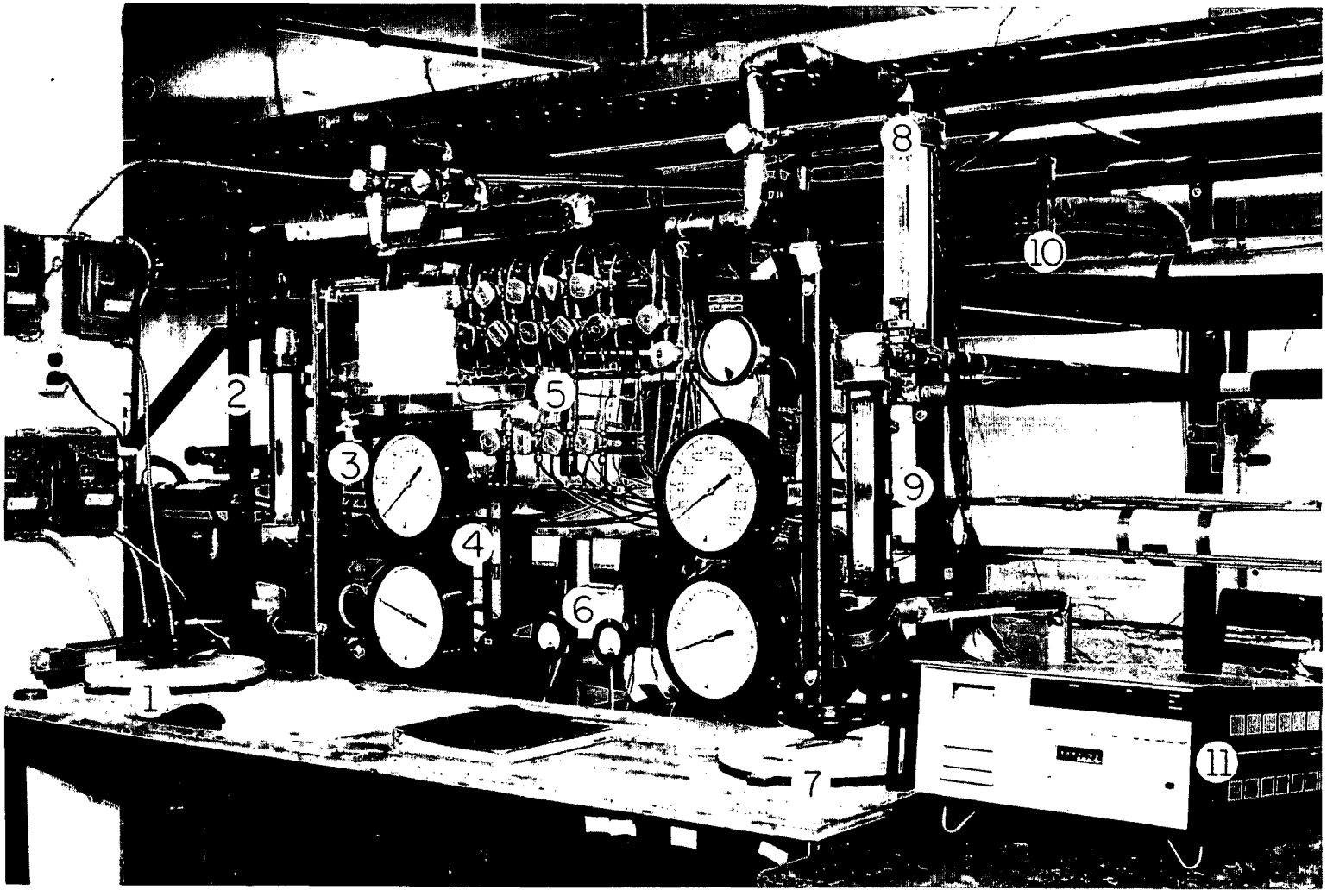
2.8 Insulation

All components and piping are covered with 0.5 in. (1.27 cm) foam rubber insulation and/or 1.0 in. (2.54 cm) fiberglass insulation. The condensers use two layers of foam rubber insulation. The evaporators and heat exchangers use a 1.0 in. (2.54 cm) layer of fiberglass insulation on top of the foam rubber. Some of the low-pressure refrigerant piping is also covered with 1.0 in. (2.54 cm) fiberglass. The two refrigerant-to-refrigerant heat exchangers are covered with 1.0 in. (2.54 cm) fiberglass.

2.9 Instrumentation

Most of the instruments and controls are located on the control panel (Figure 2.7). Figure 2.8 shows the location of pressure and temperature measurements in the system diagram. Flow rates are measured by rotameters. The refrigerant rotameter is a Fischer and Porter model 3555144 with an accuracy of 2 percent of full scale. The glycol rotameters are also Fischer and Porter, but they are slightly larger. The model number for both is 3555151, with an accuracy of 2 percent of full scale. Condenser water flow rates are measured by Brooks Model 1307 rotameters. The accuracy is also 2 percent of full scale. Repeatability is 0.5 percent of full scale for the Fischer and Porter rotameters, and 0.25 percent for the Brooks rotameters. Both the Fischer and Porter and the Brooks rotameters are calibrated in gallons of water per minute at 70°F. Manufacturers' recommended correction techniques are used to determine the flow rates of the liquids used in the test facility.

A Fluke Model 2200B Data Logger is used to measure the analog voltages from 60 thermocouples and to convert them to digital temperatures. The stated accuracy when using copper-constantan thermocouples is 0.5°C. Worst case resolution is 0.1°C. For reference, an ice-water bath in a Dewar flask is



1 LTE Heater Control
2 LTE Rotameter
3 Expansion Valve
4 Refrigerant Rotameter
5 Pressure Valves
6 Voltmeters and Ammeters
7 HTE Heater Control
8 HTC Rotameter
9 HTE Rotameter
10 Water Heater Exchanger
11 Fluke Datalogger

Figure 2.7 Control Panel

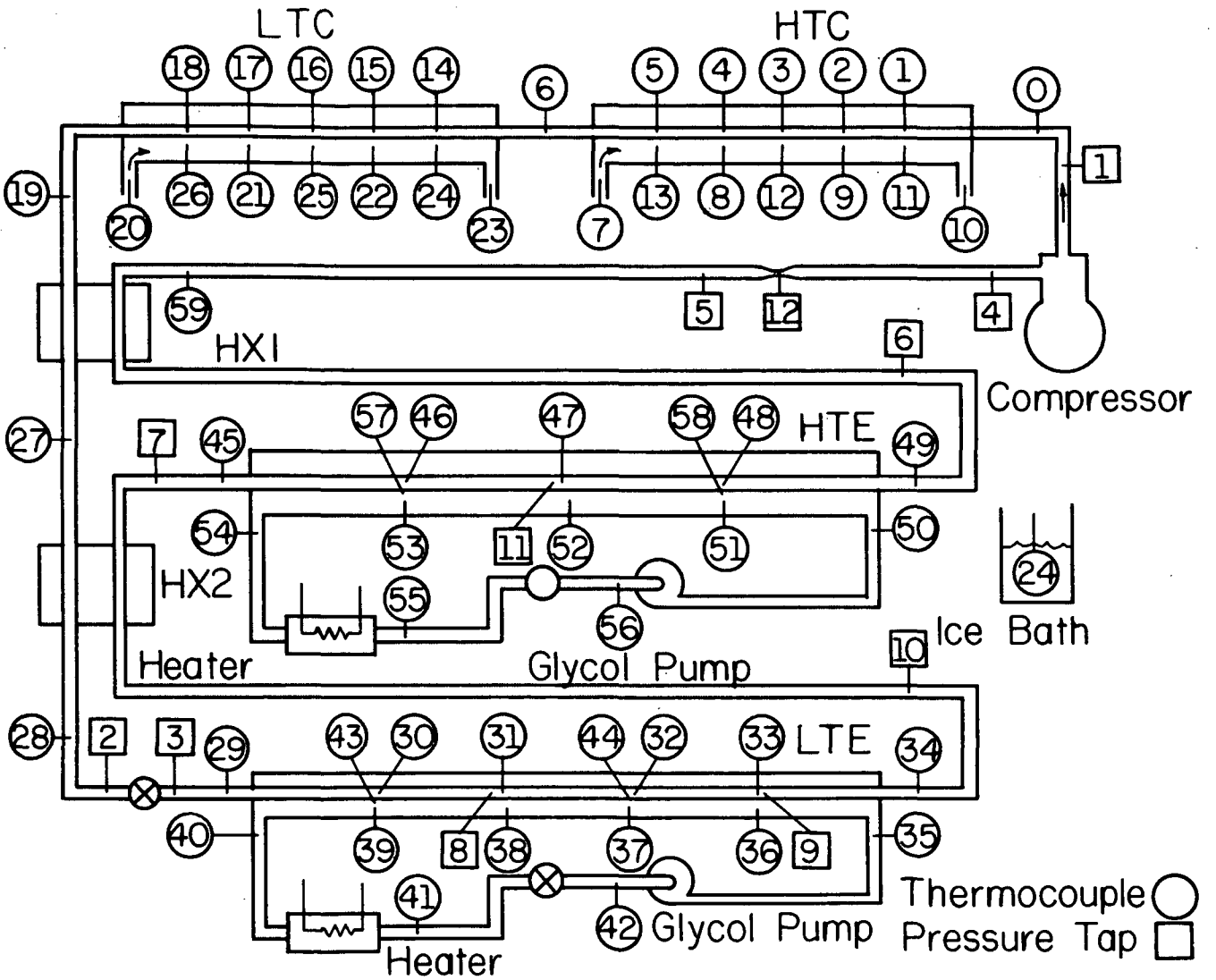


Figure 2.8 Test Temperature and Pressure Measurement

used as a 0.0°C reference. One of the thermocouples (No. 24) is kept in this reference bath as a check of the Fluke's accuracy. Most of the thermocouples are located in 1/8" (0.318 cm) OD brass tubes to prevent leaks and to steady the temperature readings. However, the thermocouples measuring the temperature of the refrigerant at the center of each evaporator pass are exposed to the refrigerant. They are made from 1/8 in. (0.318 cm) OD stainless steel sheathed copper-constantan thermocouple wire. The sheath is stripped off so that the thermocouple can be made. A magnesium oxide core prevents refrigerant leaks. The thermocouples thus constructed are about 7 feet (2.13 m) long. They were not replaced by the brass tube enclosed design because they did not leak and did not give unsteady readings.

The voltages and currents supplied to the glycol heaters are measured using Simpson voltmeters and ammeters. The voltage scale is 0 to 250 volts. The amperage scale is 0 to 10 amps.

Pressures are measured using two Weksler model AA18P gauges. The high pressures are measured with a 0 to 1000 kPa gauge. Low pressures are measured with a -100 to 200 kPa gauge. Scale divisions are 10 kPa on the high pressure gauge and 2 kPa on the low pressure gauge.

Compressor power consumption is measured by a General Electric model 5T207A6 NT48-34 dynamometer set (Figure 2.2). The tachometer has scale divisions of 100 rpm, while the spring scale has divisions of 0.1 lbs.

Alternative measurement of refrigerant flow rate was obtained by installing a Barco venturi in the suction line to the compressor. However, this proved to be a significant flow obstruction, especially during tests with 50 percent R-114. The high specific volume of the 50 percent mixture resulted in high velocities in the suction line where the venturi was installed. It was decided that this particular measurement had too much influence on system per-

formance, so it was removed. However, tests that were conducted with the venturi in place showed good agreement between flow rates measured by the venturi and those measured by the rotameter.

2.10 Data Reduction Methods

The raw test data is loaded into the University of Illinois central computer system by creating a data file at a computer terminal. The test data was then processed by a data reduction program within the computer. Equations obtained by fitting both first- and second-order least-squares curves to tabular and graphical data are used to calculate fundamental fluid properties (densities, specific heats, enthalpies, etc.). These values are then used to calculate flow rates and heat fluxes. The calculation of heat transfer coefficients is illustrated in Figure 2.9.

The work is calculated using the dynamometer load and rpm. When calculated values can be obtained by more than one method, a comparison is made between the two results. The results can then be viewed at a terminal or printed out at a remote line printer. Sample results are included in APPENDIX C.

2.11 Experimental Error

One obvious source of error is due to the assumption that no heat is lost to (or gained from) the environment. However, calculations based on actual test temperatures indicate that even in the most vulnerable component (low-temperature evaporator) the gain in heat from the ambient is only about 6 percent of the total heat transferred from the low temperature glycol to the refrigerant. The heat lost to ambient at the compressor does not introduce error in calculations because adiabatic compression is not assumed. Heat lost

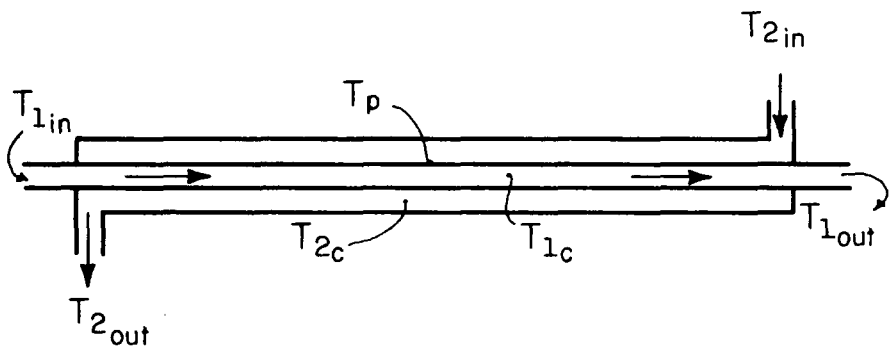


Figure 2.9 Temperatures used in Calculation of Heat Transfer Coefficients (h_c)

Let: Q = heat transfer rate (kW) for this section

q = Q/area on refrigerant side of pipe

LMTD = log mean temperature difference

$$= \frac{(T_{1_{in}} - T_{2_{out}}) - (T_{1_{out}} - T_{2_{in}})}{\ln[(T_{1_{in}} - T_{2_{out}})/(T_{1_{out}} - T_{2_{in}})]}$$

Then: $q_{local} = q \times (T_{1_c} - T_{2_c})/\text{LMTD}$

$h_c = q_{local}/(T_{1_c} - T_p)$

U = overall heat transfer coefficient = q/LMTD

Where T_{2_c} is not available, 1/2 $(T_{2_{in}} + T_{2_{out}})$ is used.

to ambient is roughly 3 percent of the heat transferred in the HTC and roughly 4 percent for the LTC. Heat loss is about 1 percent in HX1 and heat gain is negligible for HX2. The HTE gains about 2 percent.

A source of error is in measurement of the electric motor speed where the scale divisions are 100 rpm. Since the speed stays constant once steady state has been reached, a revolution counter that uses a pickup mounted directly on the motor pulley could provide very accurate measurements. The dynamometer load scale, however, has scale divisions of 0.1 lbs. which provides adequate accuracy for the torque component of the work measurement.

Error in pressure measurement is greatest for the high pressure readings where the scale divisions are 10 kPa. The low pressure gauge has scale divisions of 2 kPa, resulting in better accuracy.

The error in temperature measurement is broken into two categories. First, there is the error introduced due to variation in construction of the thermocouples themselves along with the error present in the Fluke Datalogger. This error is approximately 0.5°C. The second category is the error due to the positioning of the thermocouples. This error is most significant when glycol temperatures are measured. The glycol solution has a viscosity of about 15 centipoise in the LTE where the glycol velocity is only 0.125 m/s (0.37 fps). As a result, the mixing is very poor and the measurement of the glycol temperature depends on the position of the thermocouple in the flow stream. At the end of each evaporator pass, the glycol passes through a 6.35 mm (0.25 in) ID pipe section that mixes the stream before the temperature is measured, which eliminates the problem there. However, in the middle of each pass, the thermocouple is located in the annulus where the temperature could vary by 2 or 3°C (4 to 6 °F) depending on the exact position of the thermocouple in the flow stream. Fortunately, the most

important glycol temperatures (inlet and exit) are measured in well-mixed streams.

Flow measurement of the glycol solutions and refrigerant by using rotameters has an error of 2 percent of the full scale according to the manufacturer, Fischer and Porter. Since our measurements were in the range of 40 to 50 percent, the error would be about 4 percent in measuring the flow rates. The condenser water flow rates were measured with Brooks rotameters which also have an accuracy of 2 percent of full scale. Since measurements were typically made at around 70 percent, the error in measurement would be about 3 percent.

The Simpson voltmeters have scale divisions of 5 volts, while the ammeters have scale divisions of 0.2 amps. Considering average measurements of 5 amps and 135 volts, the error in measurement would be about 2 percent each both current and voltage.

The masses of refrigerants charged into the system were measured using a Buffalo double-beam balance with scale divisions of 0.02 lbs. Since two measurements were required to charge the system to an average total charge of 6.35 kg (14 lbs.), the error would be about than 0.5 percent in measuring percent concentration.

The glycol solution concentration was measured using a Fischer Scientific precision hydrometer (Model 11556F) and precision thermometer (Model 15043B) with scale divisions of 0.001 specific gravity and 0.1°C, respectively. An equation derived by fitting a least-squares curve to graphical data is used to calculate concentration when specific gravity and temperature are known. The error in calculating percent concentration is approximately 1 percent.

CHAPTER 3

TEST PROCEDURE

The tests were conducted in the Mechanical Engineering Laboratory at the University of Illinois at Urbana-Champaign. The data recording sheet was constructed to assure that all relevent information would be recorded for each test. Tests typically required five hours to reach steady state at the target conditions. The steps in conducting a test were:

1. Fill out heading of data recording sheet.
2. Draw glycol sample from each of the evaporators.
3. Record specific gravity and temperature of each glycol solution using precision hydrometer and precision thermometer.
4. Return samples to system if concentrations are acceptable. Otherwise adjust the glycol concentration.
5. Connect Fluke 2200 Datalogger to system by attaching the six isothermal input cards.
6. Prepare ice-water reference bath in Dewar flask using thermocouple No. 24.
7. Turn on ac-dc converter and check zero reading on dynamometer scale.
8. Turn on condenser water valve and set to desired flowrate.
9. Turn on evaporator glycol pumps and set flow rates.
10. Start dynamometer and set speed to around 500 rpm.
11. Make sure expansion valve is open at least one-eighth turn.
12. With charging hose connecting container to system, weigh the R-114 container using the double-beam balance.
13. Set scale so that it will be level after desired amount of refrigerant has been charged into system.

14. Open charging valve and let refrigerant fill system until balance levels.
15. Repeat steps 10 through 12 for R-12 container.
16. Set refrigerant flow rate to desired value.
17. Set compressor speed to desired value.
18. Monitor glycol temperatures and turn on glycol heaters when temperatures start to approach the target temperatures.
19. Monitor condenser water inlet temperature and adjust heat exchanger bypass valve to maintain desired inlet water temperature (thermocouple No. 20).
20. Monitor temperature of refrigerant entering compressor to be sure there is at least 10°C superheat.
21. Once glycol flow rates and heater loads have been established, the compressor speed and expansion valve settings must be varied to achieve target temperature conditions.
22. When system gets very close to target conditions (condenser water inlet temperature and flow rate, glycol flow rates, glycol inlet and exit temperatures), the sight glasses at the low temperature condenser exit must be examined to determine if the refrigerant is saturated liquid there.
23. If vapor bubbles are present in both sight glasses, then the system charge must be increased until bubbles appear in the first sight glass only. The system was always started with a slight undercharge so that it would not be necessary to deal with the problem of venting-off a non-azeotropic mixture.

24. Once all target conditions have been met, the system should not be touched for at least 15 minutes to make sure that conditions are steady state.
25. When steady state target conditions have been achieved, the data recording sheet should be filled out completely and any comments should be noted in the space provided. A printout of all thermocouple readings by the datalogger should be pasted to the data recording sheet.
26. If results are also desired with HX2 bypassed, then the appropriate valves should be opened (or closed) and the variables (expansion valve and compressor speed) adjusted to meet target conditions.
27. Once steady-state conditions have been achieved with HX2 bypassed, the data should be recorded completely as described earlier.
28. To begin shutdown procedure, close the shut-off valves at the pressure gauges.
29. Close the expansion valve very gently (using very light finger pressure only) until the needle just touches the seat in the valve.
30. Shut off the dynamometer and the ac-dc converter turned off.
31. Glycol pumps and heaters should continue to run for at least 5 minutes to prevent freezing of glycol solution.
32. Turn off the condenser water.
33. If system had been run with HX2 bypassed, the valves which close off HX2 should be opened to prevent excessive pressure build-up when the refrigerant comes to room temperature.
34. Empty the ice water from the Dewar flask, and return the Fluke Datalogger to the locked cabinet.

35. Create a computer input file should be created with the results from the test so that the data can be reduced.

36. If the next test is to be run with a different refrigerant mixture, the system should be opened at the filling valve and the expansion valve should be opened slightly. Using the high pressure gauge, the pressure should be monitored until it drops to about 20 kPa gauge, at which time the evacuation pump should be connected to the filling valve. After overnight evacuation, the system is ready for another test.

CHAPTER 4

EXPERIMENTAL DATA

4.1 Test Conditions

The tests were run under identical refrigeration load conditions so that direct comparison could be made between them. The bottom line of figures on Table 4.1 gives the target settings of flow rates and temperatures.

The target values were based on results of preliminary tests. Initial tests indicated that compressor performance would limit the achievable conditions of the test. Increasing the motor speed past 2000 rpm, for example, did not increase the flow rate appreciably because of poor volumetric efficiency at higher speeds. Due to the change in specific volume of the suction vapor as the concentration of R-114 is increased, the compressor speed was increased by a factor of three when going from pure R-12 to a concentration of 70 percent R-114. By setting target conditions with the compressor motor running at about 600 rpm using pure R-12, it was possible to use concentrations as high as 70 percent R-114. A 500 rpm motor speed was chosen as a lower limit due to overheating of the motor at any slower speeds. An upper limit of 2000 rpm was chosen because compressor performance dropped off sharply at any greater speeds. Target flow rates and temperatures were then chosen so that refrigerant temperature profiles would be close to parallel with temperatures of the fluid being cooled. Flow rates of the glycol solutions were kept fairly high to keep the solutions well-mixed. The glycol concentration was kept as low as possible to reduce viscosity while providing a 10°C cushion against freezing.

Table 4.1 Test Control Parameters

Test No.	R-114 Percent	Percent Condenser Flow	Percent LTE Flow	Percent HTE Flow	T20 °C	T35 °C	T40 °C	T50 °C	T54 °C	LTE Percent Glycol	HTE Percent Glycol
48	10	70	42.6	47.7	34.3	-7.8	-13.6	-0.1	-6.5	41	31
49	20	69	41.8	48.0	34.3	-8.0	-13.7	-0.1	-6.7	42	33
50	40	70	41.9	47.9	34.2	-7.9	-13.8	-0.2	-6.6	42	31
51	30	70	41.9	48.2	34.4	-8.0	-13.7	-0.1	-6.7	40	32
52	50	70	42.0	47.9	34.3	-8.0	-13.6	-0.1	-6.6	42	32
53	0	70	41.9	47.9	34.4	-8.1	-13.6	-0.1	-6.8	42	32
54	0	71	41.7	48.0	34.2	-7.9	-13.8	-0.2	-6.7	40	32
55	50	70	42.0	48.0	34.3	-7.9	-13.7	-0.1	-6.6	40	32
56	0	69	42.2	48.0	34.4	-7.9	-13.7	-0.1	-6.7	41	31
63	70	70	42.0	48.0	34.0	-7.7	-13.4	-0.1	-6.4	37	28
64	70	70	42.0	47.5	33.9	-8.0	-13.7	0.0	-6.4	37	28
65	60	70	41.5	47.0	34.0	-8.1	-13.8	-0.2	-6.5	37	28
66	60	70	41.5	48.0	33.9	-8.0	-13.6	-0.4	-6.5	37	28
67*	60	71	41.0	48.0	34.2	-8.0	-13.7	-0.3	-6.5	40	43
68	40	72	41.7	47.8	34.2	-7.9	-13.7	-0.2	-6.7	39	38
69*	40	70	41.5	47.8	34.2	-7.9	-13.7	-0.1	-6.6	39	38
70	30	70	42.3	48.2	34.0	-7.9	-13.6	-0.4	-6.4	40	33
71*	30	70	43.1	48.0	33.9	-8.2	-13.5	-0.3	-6.4	40	33
72	20	70	41.9	47.3	34.1	-7.8	-13.5	-0.4	-6.3	39	31
73*	20	70	41.7	47.5	34.2	-8.1	-13.7	-0.4	-6.3	39	31
74	10	69	42.2	47.9	34.4	-8.0	-13.5	-0.5	-6.5	40	32
75*	10	70	42.0	48.0	34.3	-7.9	-13.7	-0.1	-6.1	40	32
76*	0	70	42.5	47.5	34.3	-8.0	-13.3	-0.3	-6.3	40	31
Targets		70	42.0	48.0	34.2	-8.0	-13.7	-0.1	-6.7	40	30

* HX2 Bypassed.
Temperature designation on Fig. 2.8.

4.2 Condenser Temperature Profiles

Figure 4.1 shows typical condenser temperature profiles for both pure R-12 and 50 percent concentration of R-114. The refrigerant temperature entering the condenser is higher when R-12 is used alone than for the 50-50 mixture. The reason is that R-114 inherently provides lower discharge temperatures from the compressor. With 100 percent R-12 the temperature of condensing refrigerant remains essentially constant as it passes through the two condensers until near the end where the temperature drops slightly due to subcooling. This subcooling occurred despite the attempt to assure that a few vapor bubbles were leaving the condenser. It is possible that there is a small amount of non-equilibrium between the liquid and vapor at this point.

The temperature profile for the 50 percent R-114 mixture showed the sharp drop in temperature during desuperheating initially upon entering the high-temperature condenser. Thereafter occurs the progressive drop in temperature as the binary mixture condenses. The drop in temperature during condensation is approximately 7°C.

The condenser cooling water enters the LTC at the point of refrigerant exit, so the flow of cooling water corresponds to a leftward direction of the temperature profile. While the entering water temperatures for the two refrigerant concentrations are the same and the leaving water temperatures are the same for both concentrations there is a spread within the temperatures. When operating with 100 percent R-12 the refrigerant temperature near the outlet is higher than with the mixture, so the rate of heat transfer at that position in the condenser is high. The rate of change of water temperature is thus greater with 100 percent R-12 near the refrigerant outlet. In the early stages of the HTE, however, the temperature difference between the mixture and water is highest, so the slope of the water temperature curve is steepest there for the 50 percent mixture.

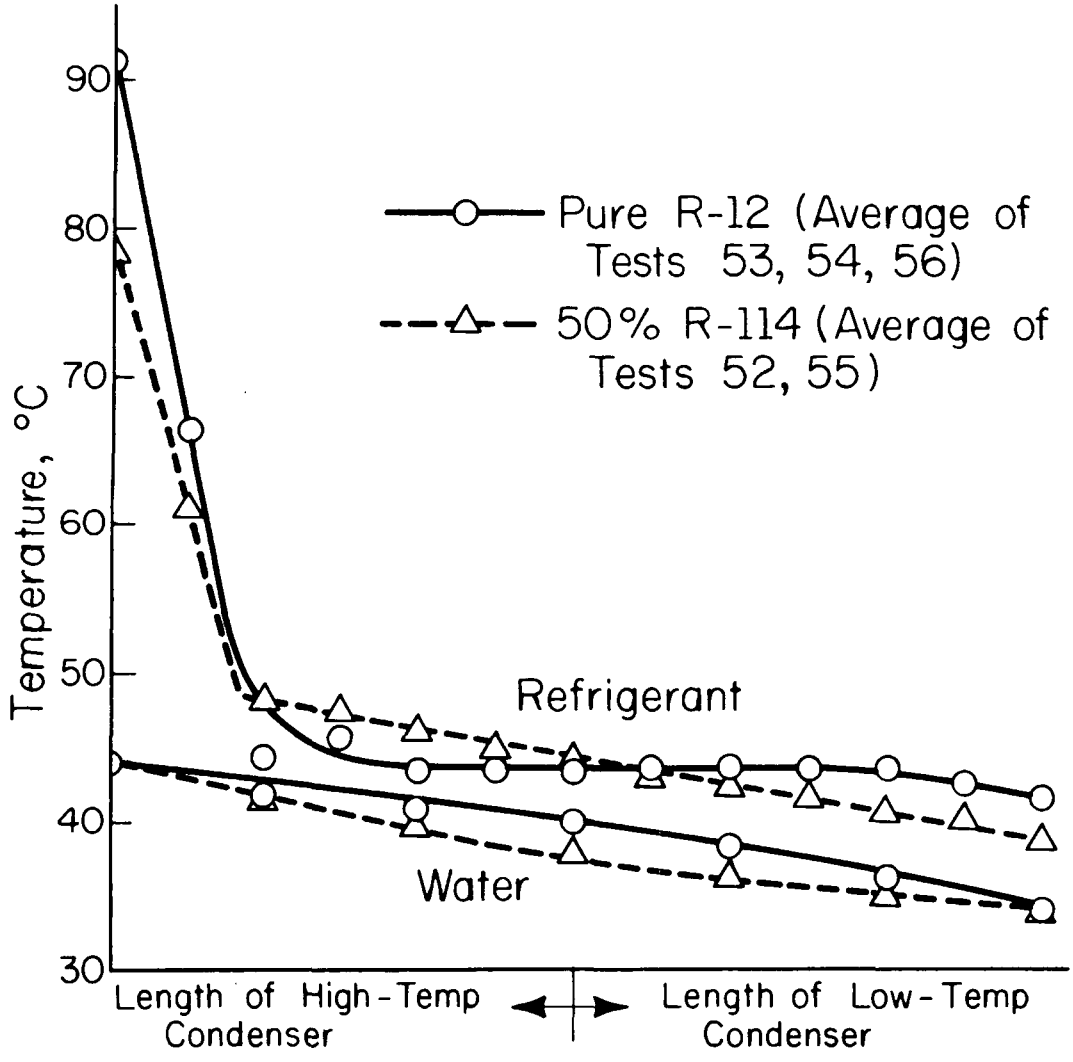


Figure 4.1 Condenser Temperature Profiles

4.3 Evaporator Temperature Profiles

Figure 4.2 shows temperature profiles of refrigerant and glycol for the evaporators for 0 and 50 percent R-114 concentrations. The temperature profile of the single R-12 refrigerant shows a constant-temperature boiling until the onset of superheating at the end of the HTE. The temperature of the 50-50 mixture, however, increases all the way through the evaporators and the HX2. The particular thermal loads imposed by the glycol were chosen such that all the HTE would be used for boiling when the R-114 concentration was 50 percent. The natural consequence of this choice was that in order to maintain the same thermal load with R-12 only, some of the HTE had to be devoted to superheating. Thus the mean temperature difference between glycol-water and refrigerant is approximately the same for both refrigerant concentrations. In fact, the same statement must be made for the LTE as well where the 50 percent mixtures enters at a lower but leaves at a higher temperature than the R-12 refrigerant.

4.4 Condensing Heat Transfer Coefficients

The measured condensing heat transfer coefficients were found to decrease as the R-114 concentration increased (Figure 4.3). The results indicate, however, that the heat transfer coefficients may start to improve again as the concentration approaches pure R-114. Compressor limitations prevented study of mixture concentrations greater than 70 percent. The change in the heat transfer coefficient is most dramatic in the range of 0 to 30 percent R-114. This is similar to results published by Arora [3]. This degradation of the condensing heat transfer coefficient may be the greatest penalty to mixtures of R-12 and R-114.

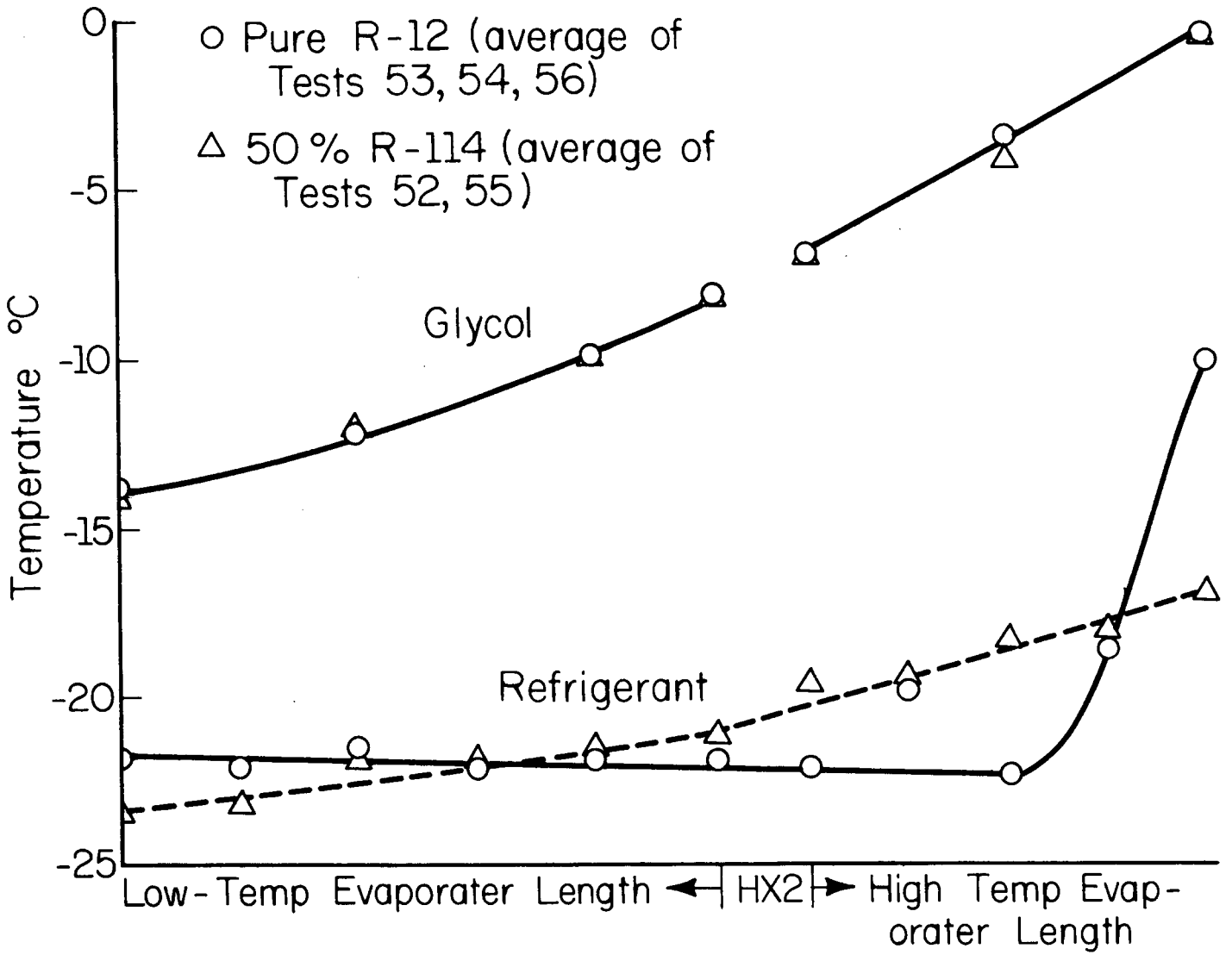


Figure 4.2 Evaporator Temperature Profiles

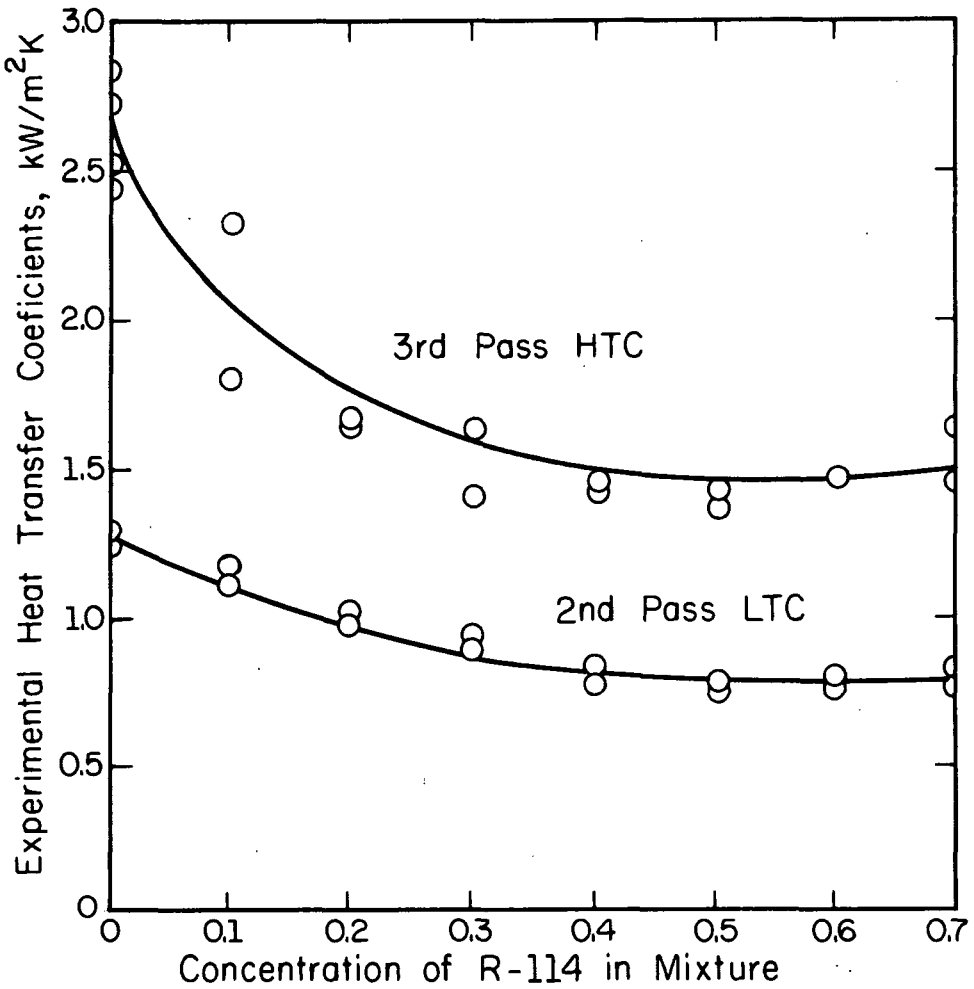


Figure 4.3 Condensing Heat Transfer Coefficients as Functions of Mixture Concentration for Constant Refrigerating Capacity

4.5 Boiling Heat Transfer Coefficients

Figure 4.4 shows some boiling heat-transfer coefficients measured in the two passes of both the LTE and HTE. In general, the coefficients increase as the refrigerant passes through the evaporator, with the highest coefficients being experienced in the second pass of the HTE. The measured points designated by triangles were made with the HX2 bypassed which resulted in refrigerant with a higher vapor fraction entering the first pass of the LTE in comparison to that if the heat exchanger were in operation. There was no definite reduction in the boiling heat-transfer coefficient indicated for the mixture in comparison to R-12 only. The coefficients are derived from measurements occuring with small temperature differences between the tube and refrigerant, so slight deviations in the temperatures can cause some scatter of the resultant coefficients.

4.6 Refrigerant Flow Rate

Figure 4.5 shows the refrigerant flow rate with the numbers alongside the points indicating the test number. There is a slight increase in the flow rate as the concentration moves from R-12 alone to the 50/50 mixture.

4.7 Compressor Power and Speed

One of the purposes of this phase of the refrigerant mixture study was to confirm the prediction from Launay's simulations that for a given refrigeration duty, the compressor power could be reduced by introducing R-114 into the R-12. Launay proposed that the minimum power requirement would occur with an R-114 concentration of 0.68. The reason for driving the compressor with a variable-speed dynamometer in the current tests was to be able to

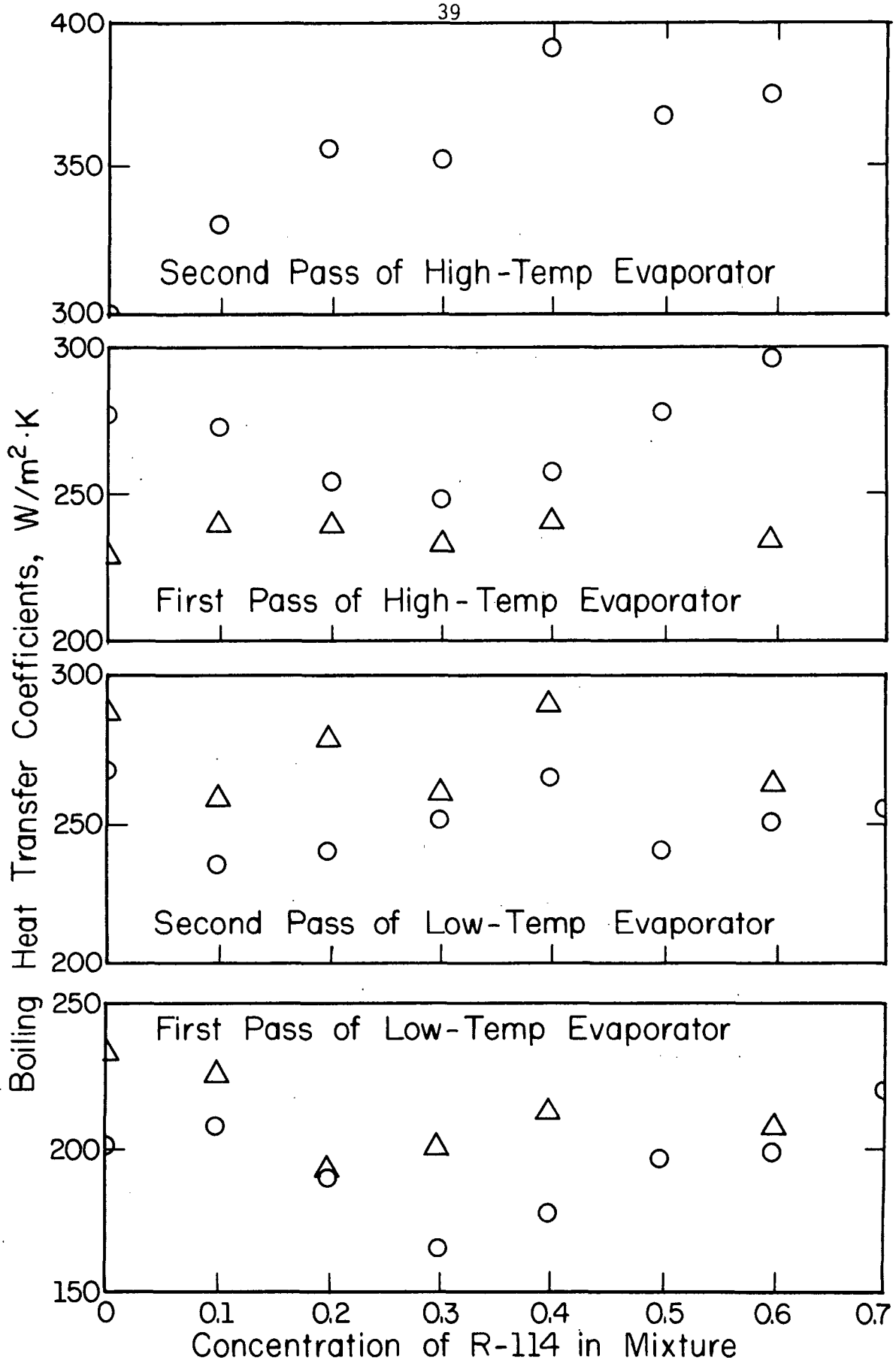


Figure 4.4 Boiling Heat Transfer Coefficients as Functions of Mixture Concentrations (Circles denote Low-Temperature Heat Exchanger in Operation; Triangles, Low-Temperature Heat Exchanger bypassed)

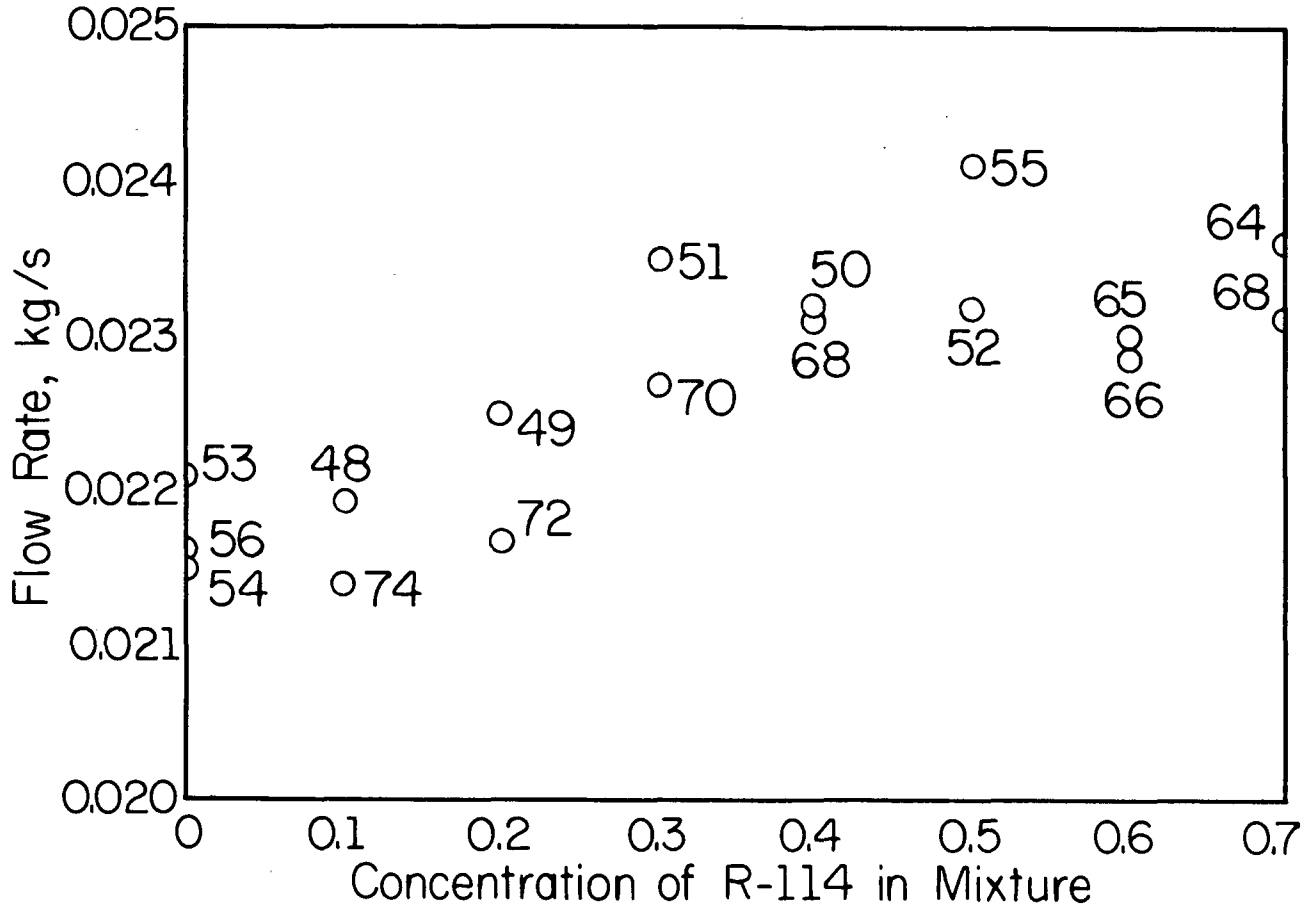


Figure 4.5 Refrigerant Flow Rate for Constant Refrigeration Capacity (Data Identified by Test Number)

adjust the compressor speed to compensate for the increased specific volumes as the concentration of R-114 increased, and to measure the power required by the compressor. It soon became clear that the power measured by the dynamometer increased as the concentration of R-114 increased, as Fig. 4.6 shows, rather than decreasing as expected. Figure 4.6 also shows the increase in motor speed required as the concentration of R-114 was increased. To explore whether the increase in power could be explained by a change in compressor efficiency, a measurement of compressor friction as a function of speed was conducted, as discussed in the next section.

4.8 Compressor Friction

The mechanical frictional losses in a compressor vary with the rotative speed. In a reciprocating compressor, friction will increase with roughly the square of the speed. A test was conducted to determine the friction loss as a function of speed. The system was evacuated and the compressor was run with the suction valve closed at various speeds, recording the dynamometer load at each speed. The suction valve was closed so that no gas passed through the compressor. A second-order least squares curve was fit to the data resulting in the relationship in Figure 4.7. Friction power from Fig. 4.7 was deducted from the measured dynamometer power in Fig. 4.6 to eliminate the influence of the changes in compressor speed.

4.9 Refrigerant Pressure Drop

Pressure drop will both reduce the amount of temperature rise in the evaporation process and will increase the temperature drop in the condenser. The pressure drop data shown in Figure 4.8 indicates a general increase in pressure drop as the concentration of R-114 increases. The increase is

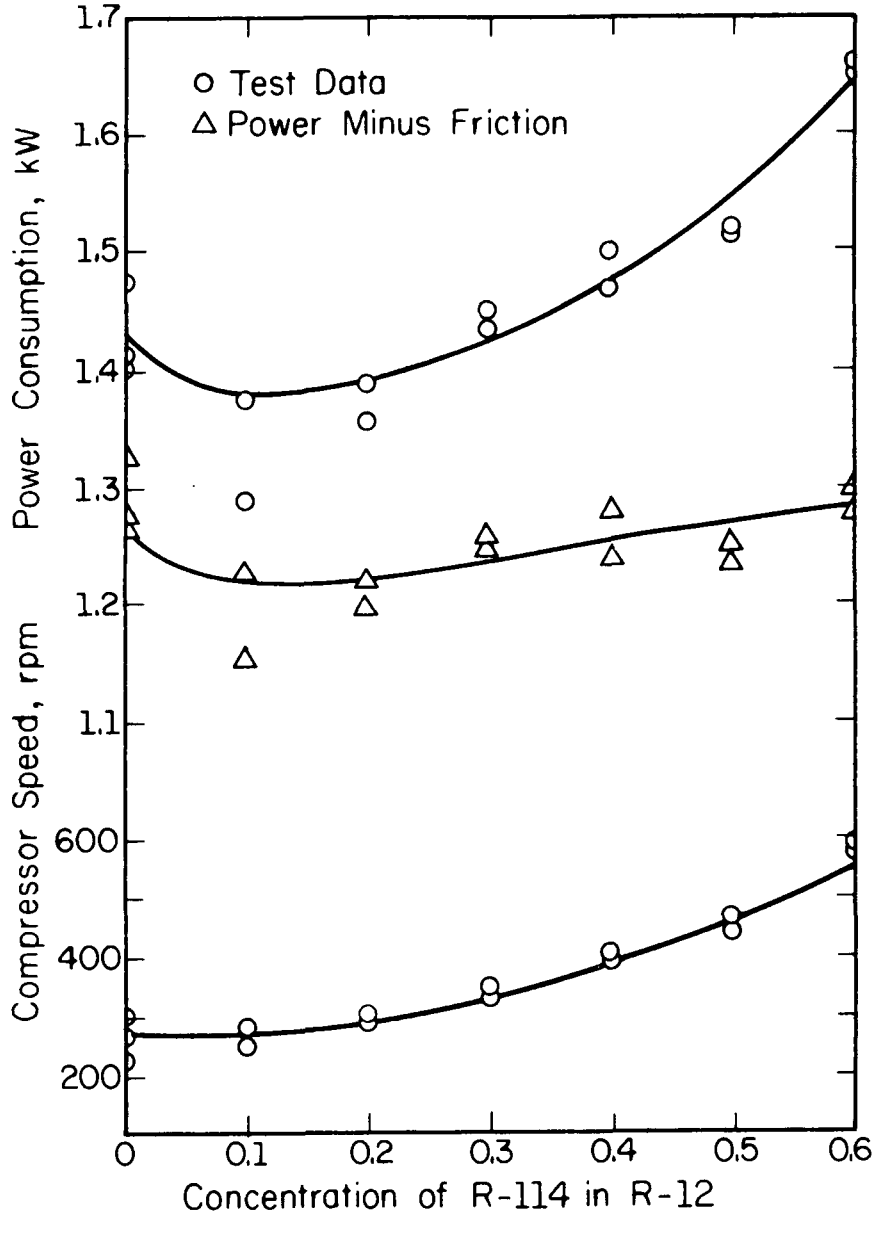


Figure 4.6 Compressor Power and Speed

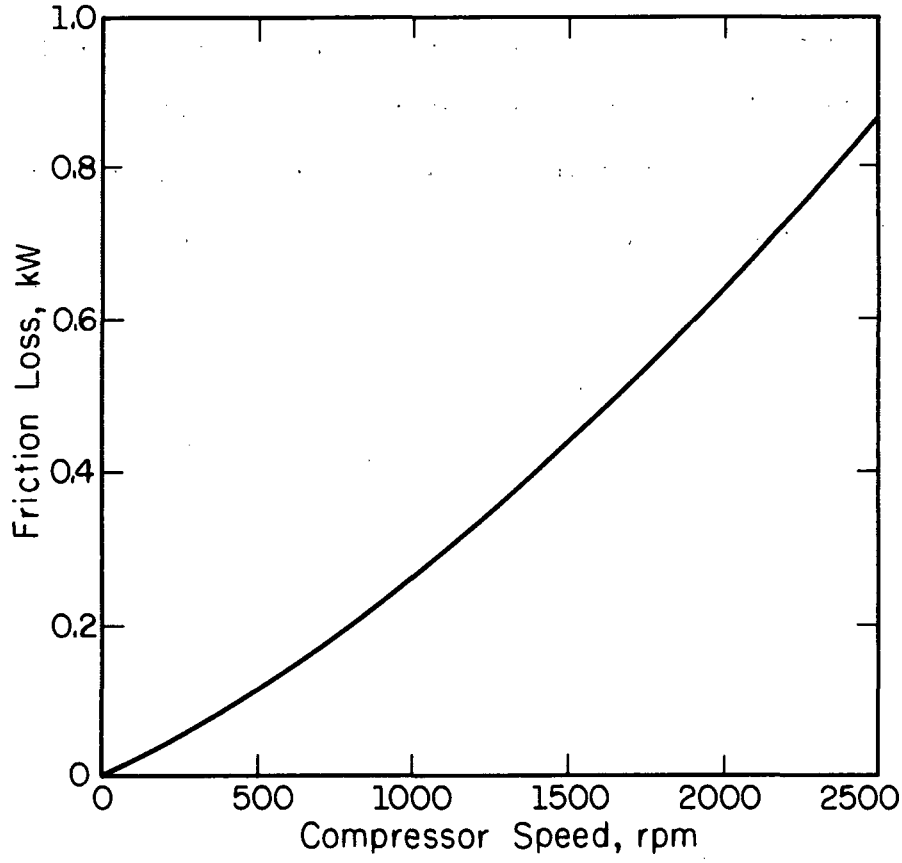


Figure 4.7 Compressor Friction Loss Based on Test Data

attributed to the higher velocities that prevail when the specific volume of vapor increases as the concentration of R-114 increases.

4.10 Refrigerant Pressures

The refrigerant pressures decreased with increasing concentration of R-114 as expected. Although the two pressure levels in Figure 4.9 are plotted using different scales, the ratio of discharge to suction pressures increases with increasing R-114 concentration.

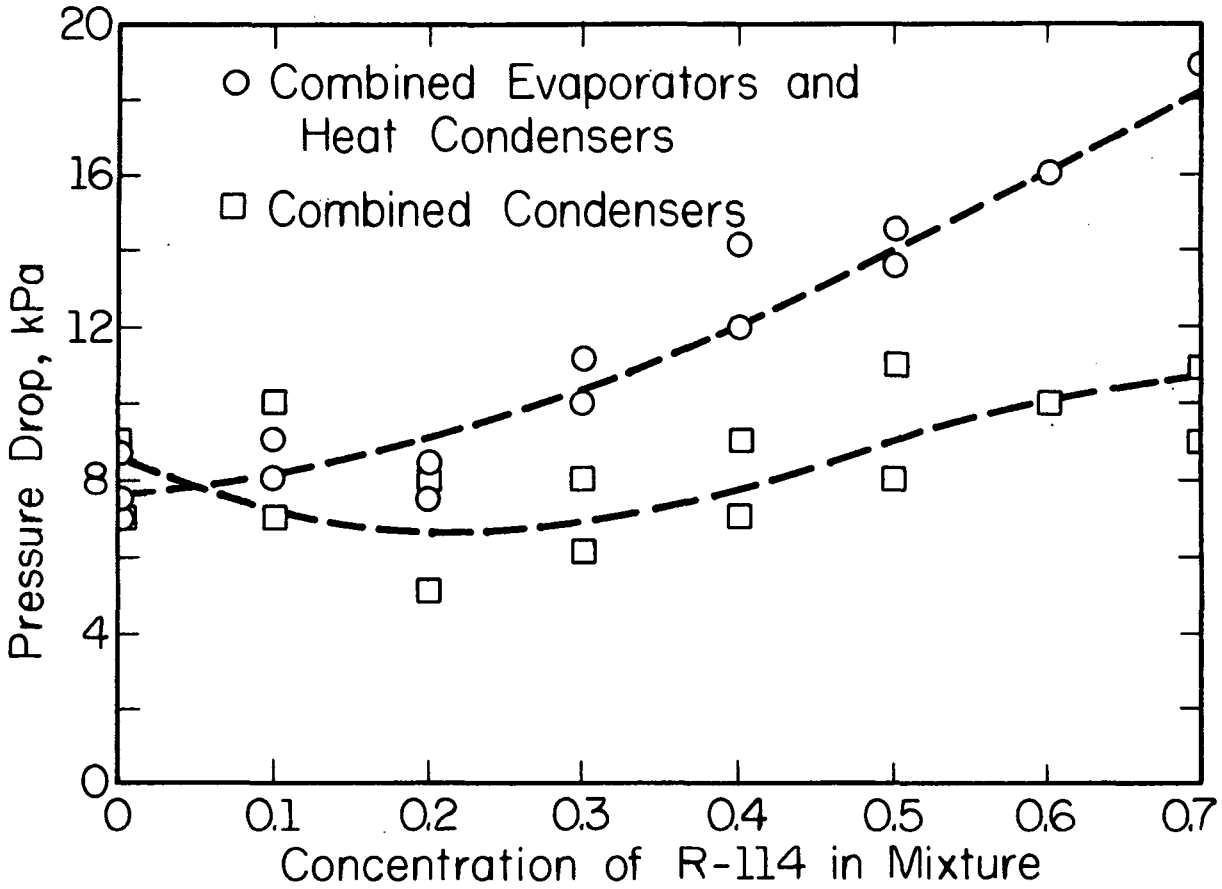


Figure 4.8 Refrigerant Pressure Drop

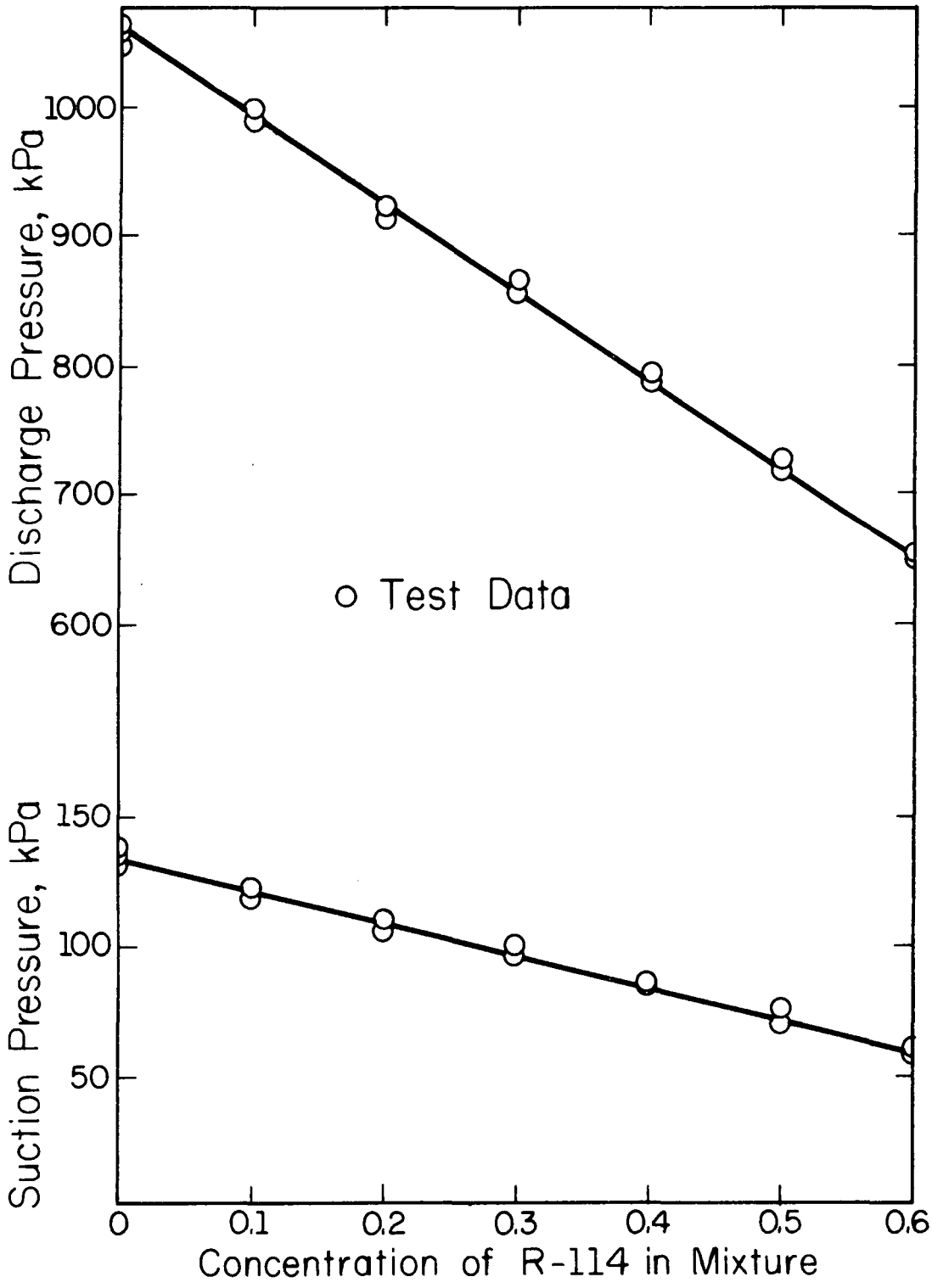


Figure 4.9 Compressor Suction and Discharge Pressures as Functions of Mixture Concentration for Constant Refrigerating Capacity

CHAPTER 5

COMPUTER SIMULATION

5.1. Simulation Technique

The test facility has been modeled mathematically using 45 equations with 45 variables. The equations are solved using the Newton-Raphson solution technique [7]. The 45 equations generally represent mass and energy balances as well as physical properties. However, some physical properties such as specific volumes are calculated in the computer program without including them as system variables. This was done to reduce computing costs without jeopordizing solution accuracy. The simulation program is based on the program written by Launay.

5.2. Revisions to Original Program.

The original program written by Launay had two purposes: first, to simulate a vapor-compression system employing nonazeotropic refrigerant mixtures; and second, to optimize the distribution of heat transfer surface areas in order to minimize compressor power consumption. It was an optimization program that used a simulation subprogram to solve the system equations for each iteration of the optimization. However, since it is not possible to readily redistribute surface areas among the components of the test facility, the optimization procedures are not a part of the revised simulation program. The major revisions to the simulation are:

1. Heat transfer coefficients are now based on actual test data and are functions of the mixture concentration.
2. Pressure drops (based on test results) have been included, also as functions of the concentration.

3. The point at which superheating begins has been moved into the HTE to correspond to test results.
4. The specific volume of the vapor entering the compressor is calculated so that the required compressor speed can be determined.
5. The power consumed due to mechanical friction in the compressor is included as a function of the compressor speed. The equation is a result of physical testing.
6. Volumetric and isentropic compression efficiencies are included as functions of the compressor speed.

5.3 Modeling of Actual Tests

During actual tests, the refrigeration duty, secondary fluid flow rates and secondary fluid inlet temperatures were fixed. This enabled direct comparison between tests. In addition, it was required during each test that superheat begin in the high-temperature evaporator. These fixed conditions are also a part of the simulation. They are read into the program as an array called DAT. These 17 fixed parameters are described as follows:

DAT(1) = CONC	Mass fraction of R-114 in R-12
DAT(2) = ALTE	Heat transfer area of LTE, m^2
DAT(3) = AHX1	Heat transfer area of HX1, m^2
DAT(4) = AHX2	Heat transfer area of HX2, m^2
DAT(5) = ALTC	Heat transfer area of LTC, m^2
DAT(6) = AHTC	Heat transfer area of HTC, m^2
DAT(7) = WLTE	Glycol flow rate in LTE, kg/s
DAT(8) = WHTE	Glycol flow rate in HTE, kg/s
DAT(9) = WLTC	Water flow rate in LTC, kg/s

DAT(10) = WHTC Water flow rate in HTC, kg/s

DAT(11) = QLTE Refrigerating capacity of LTE, kW

DAT(12) = QHTE Refrigerating capacity of HTE, kW

DAT(13) = TCLI Temperature of inlet water to LTC, °C

DAT(14) = TCHI Temperature of inlet water to HTC, °C

DAT(15) = TELI Temperature of inlet glycol to LTE, °C

DAT(16) = TEHI Temperature of inlet glycol to HTE, °C

DAT(17) = SPHX Superheat of vapor entering compressor, °C .

5.4. Simulation Variables

The 45 variables and their corresponding location in the system is shown in Figure 5.1. The variable descriptions are:

V(1) = W Refrigerant flow rate, kg/s

V(2) = PHI Compressor discharge pressure, kPa

V(3) = PLO Compressor suction pressure, kPa

V(4) = T1 Temperature at point 1, °C

V(5) = T2 Temperature at point 2, °C

V(6) = T2P Temperature at point 2P, °C

V(7) = T3 Temperature at point 3, °C

V(8) = T4 Temperature at point 4, °C

V(9) = T5 Temperature at point 5, °C

V(10) = T6 Temperature at point 6, °C

V(11) = T7 Temperature at point 7, °C

V(12) = T8 Temperature at point 8, °C

V(13) = T9 Temperature at point 9, °C

V(14) = T10 Temperature at point 10, °C

V(15) = T10P Temperature at point 10P, °C

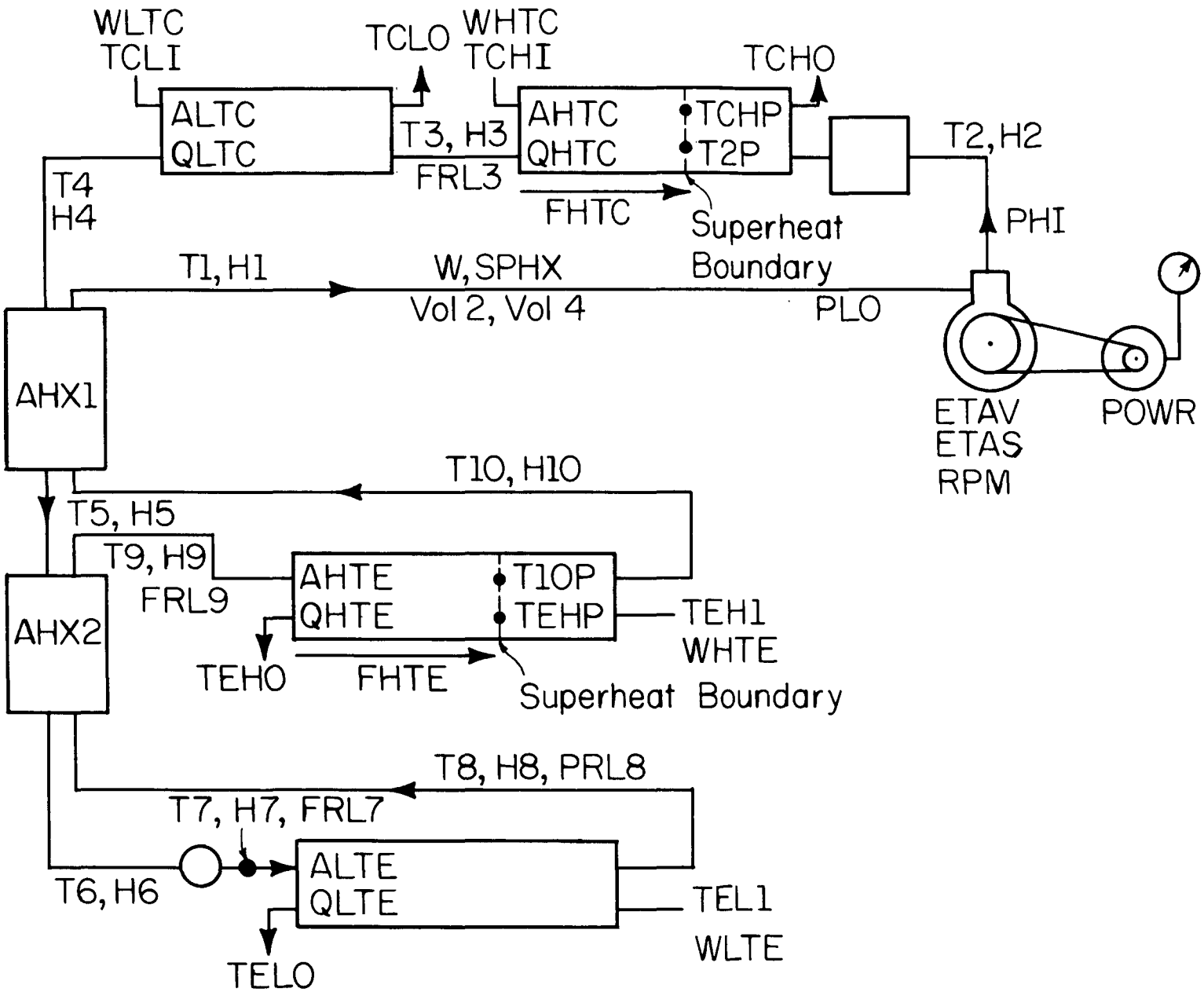


Figure 5.1 Simulation Variables

V(16) = H1	Enthalpy at point 1, kJ/kg
V(17) = H2	Enthalpy at point 2, kJ/kg
V(18) = H3	Enthalpy at point 3, kJ/kg
V(19) = H4	Enthalpy at point 4, kJ/kg
V(20) = H5	Enthalpy at point 5, kJ/kg
V(21) = H6	Enthalpy at point 6, kJ/kg
V(22) = H7	Enthalpy at point 7, kJ/kg
V(23) = H8	Enthalpy at point 8, kJ/kg
V(24) = H9	Enthalpy at point 9, kJ/kg
V(25) = H10	Enthalpy at point 10, kJ/kg
V(26) = FRL7	Fraction of liquid at point 7
V(27) = FRL8	Fraction of liquid at point 8
V(28) = FRL9	Fraction of liquid at point 9
V(29) = FRL3	Fraction of liquid at point 3
V(30) = FHTC	Fraction of area in HTC condensing
V(31) = FHTE	Fraction of area in HTE evaporating
V(32) = QLTC	Heat transfer rate in LTC, kW
V(33) = QHTC	Heat transfer rate in HTC, kW
V(34) = TCHP	Temperature of water opposite point 2P, °C
V(35) = AHTE	Heat transfer area of HTE, m^2
V(36) = TELO	Temperature of glycol exiting from LTE, °C
V(37) = TEHP	Temperature of glycol opposite point 10P, °C
V(38) = TEHO	Temperature of glycol exiting from HTE, °C
V(39) = TCLO	Temperature of water exiting from LTC, °C
V(40) = TCHO	Temperature of water exiting from HTC, °C
V(41) = POWR	Power required by compressor, kW
V(42) = RPM	Compressor speed, rpm

V(43) = ETAV Compressor volumetric efficiency

V(44) = VOL2 Partial volume of R-12 at point 1

V(45) = VOL4 Partial volume of R-114 at point 1

5.5 Simulation Equations

The Newton-Raphson technique requires that the equations be written with all terms on one side of the equal sign [7]. When each equation has been solved it will equal zero, and if not, then there will be some residual amount. The simulation subroutine PARDIF numerically calculates the partial derivatives of each residual with respect to each variable (45 x 45 = 2025 partial derivatives). The solution to the system of linear equations determines the magnitude of the correction of each of the variables. The complete listing of the 45 residual equations is included in the program listing in APPENDIX D, but a general description of the equations follows:

Equations 1 through 4 are refrigerant mass balances in the saturated regions.

Equations 5 through 12 calculate refrigerant enthalpies.

Equations 13 through 18 are heat balances in heat exchangers.

Equation 19 fixes the area of the high-temperature evaporator, which had been a variable in the Launay optimization.

Equation 20 is an energy balance across HX2.

Equations 21 through 23 are mole balances at the saturation points.

Equations 24 through 31 are heat balances.

Equations 32 and 34 calculate enthalpy of superheated vapor.

Equation 33 calculates compressor power consumption.

Equations 35 and 36 are heat balances for the portions of HTC and HTE

where superheated vapor is present.

Equations 37 and 38 are heat balances on the glycol in each evaporator.

Equations 39 defines the superheat of the vapor entering the compressor.

Equations 40 and 41 are heat balances on the water in each condenser.

Equation 42 calculates compressor speed.

Equation 43 calculates volumetric efficiency based on motor speed.

Equations 44 and 45 calculate partial specific volumes of the R-12 vapor and R-114 vapor entering the compressor.

APPENDIX E presents the results of the simulations of the performance for R-114 concentrations varying from 0 to 60 percent. Instead of using precisely 0 as the fraction of R-114 for 100 percent R-12, the value of 0.0001 was used. Zero concentration could not be used in the simulation because several equations designed with the expectations of mixtures would require division by zero.

CHAPTER 6

ANALYSIS OF RESULTS

6.1 Objectives of this Study

The major goals of this phase of the study were (1) to experimentally verify the possibility of energy conservation, and (2) to refine the simulation model to bring it into agreement with experimental measurements. The principal modifications of the simulation model were to incorporate actual heat-transfer coefficients, and to account for pressure drop in the condensers and evaporators.

The particular topics on which the following analysis concentrates are the refrigerant temperature profiles, heat transfer coefficients, refrigerant flow rate, compressor speed, and power.

6.2 Temperature Profiles

The comparative values of the temperature profiles from the simulation and from experiments are shown for the condensers in Figure 6.1 and the evaporators in Figure 6.2. Agreement of the two approaches is good for the condensers in Figure 6.1. For low concentrations of R-114, the simulations in the evaporators underestimate the refrigerant temperature in the region where superheating is expected to occur. The reason for this discrepancy is probably in the structure of the residual equations in the simulation which must be restudied.

By comparing the evaporating temperatures for the 50 percent mixture with those expected by looking at pressure-temperature-concentration information (APPENDIX A), it appears that the mixture entering the low-temperature evaporator is subcooled. When comparing data between tests with and without HX2 bypassed, the temperature of the refrigerant entering the LTE is higher

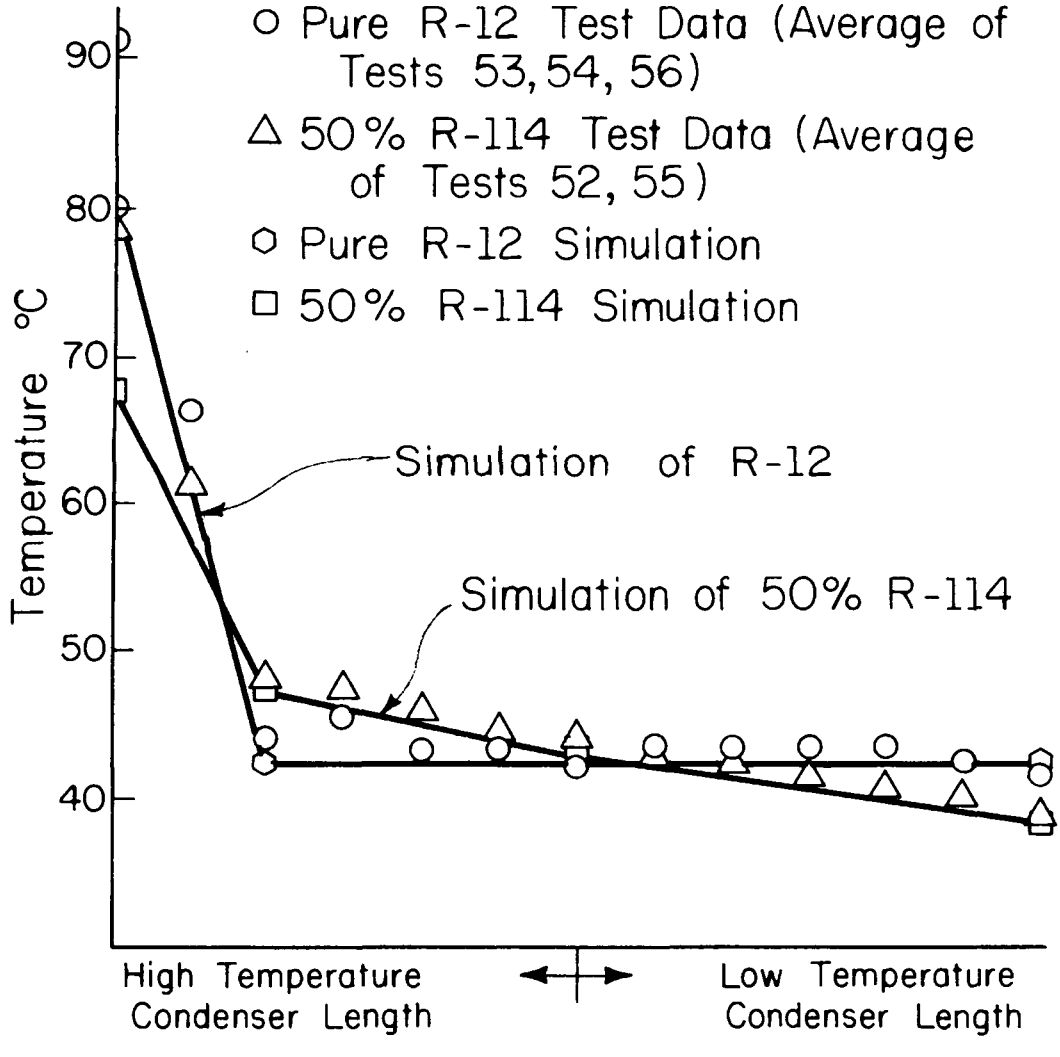


Figure 6.1 Comparison of Simulation and Actual Refrigerant Temperature Profiles in Condensers

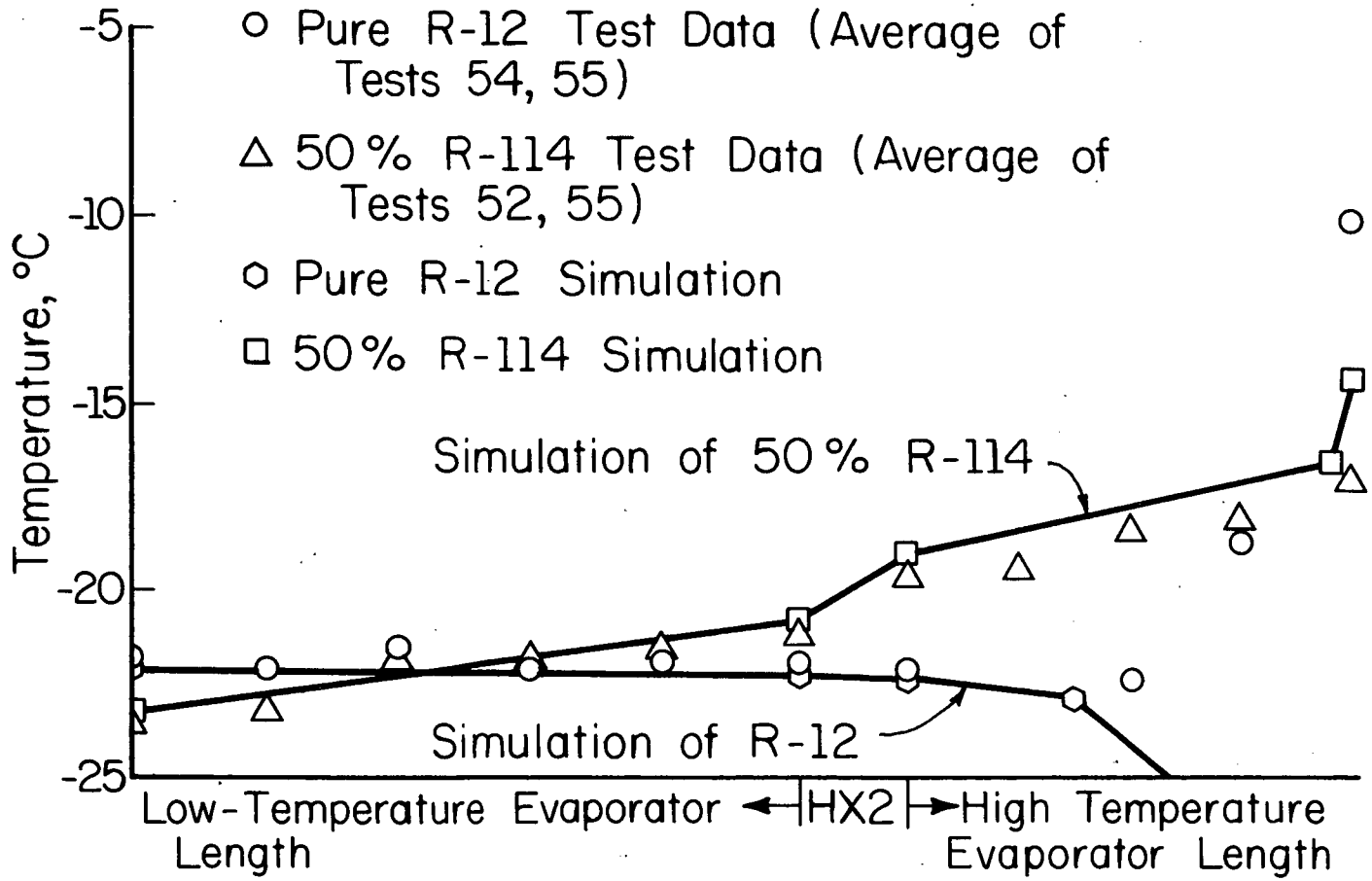


Figure 6.2 Comparison of Simulation and Actual Refrigerant Temperature Profiles in Evaporator

during the bypassed tests even though the evaporating pressures are the same or lower. This supports the contention that the refrigerant entering the LTE is slightly subcooled.

6.3 Heat Transfer Coefficients

The condensing heat transfer coefficients were in the range that was expected. However, the boiling heat transfer coefficients were quite a bit lower (Figure 4.4). This was in part due to the subcooling explained in Section 6.2, but is also a result of an effort to reduce the pressure drop in the evaporators. The piping cross-sectional area in the evaporators is 2.25 times greater than in the condensers. The resulting lower velocities help to keep pressure drop to a minimum, but penalize the heat transfer coefficients. The increase in the heat transfer coefficients of roughly 75 percent between the LTE and the HTE is in part due to the increasing flow velocity as some of the refrigerant vaporizes. But some of the increase is due to the greater heat flux in the high-temperature evaporator. There is a smaller area and a higher heat transfer rate in the HTE in comparison to the LTE.

The changing liquid and vapor concentrations during evaporation of the mixture should also influence the boiling heat transfer coefficients. This problem should be the focus of continuing research on the project. Improving the heat transfer characteristics of mixtures would surely improve the advantages nonazeotropic mixtures possess over single component refrigerants.

6.4 Refrigerant Flow Rate

The refrigerant mass flow rate can provide an indication of relative power requirement when the refrigerating capacity is the same. The test results in Figure 6.3 suggest that minimum power requirement would occur at the concentration of about 10 percent R-114.

The refrigerant mass flow rates calculated by the simulation show a similar relationship to those obtained by experiment (Figure 6.3). The simulation indicates a reduction in the flow rate of about 2.5 percent in going from pure R-12 to 20 percent R-114. Although these results tend to confirm the trend, the discrepancy between the test data and the simulation results is enough to prevent any strong conclusions. Employing more accurate refrigerant property equations [2] should yield more significant simulation flow rates.

6.5 Compressor Speed

The specific volume of the suction vapor increases as the concentration of R-114 increases. The compressor speed increases primarily because of this change in the specific volume (Figure 6.4). To compound the problem, the volumetric efficiency of the compressor decreases at higher compressor speeds. The increasing specific volume and subsequent decreasing volumetric efficiency account for the nonlinear increase in compressor speed as the concentration of R-114 increases. The increase in the refrigerant flow rate required to maintain a specified refrigeration load (Section 6.4) also contributes slightly to the increase in compressor speed.

The simulation results for compressor speed compare very well with the test data. This was expected because the correlations used in the simulation were based on the test results.

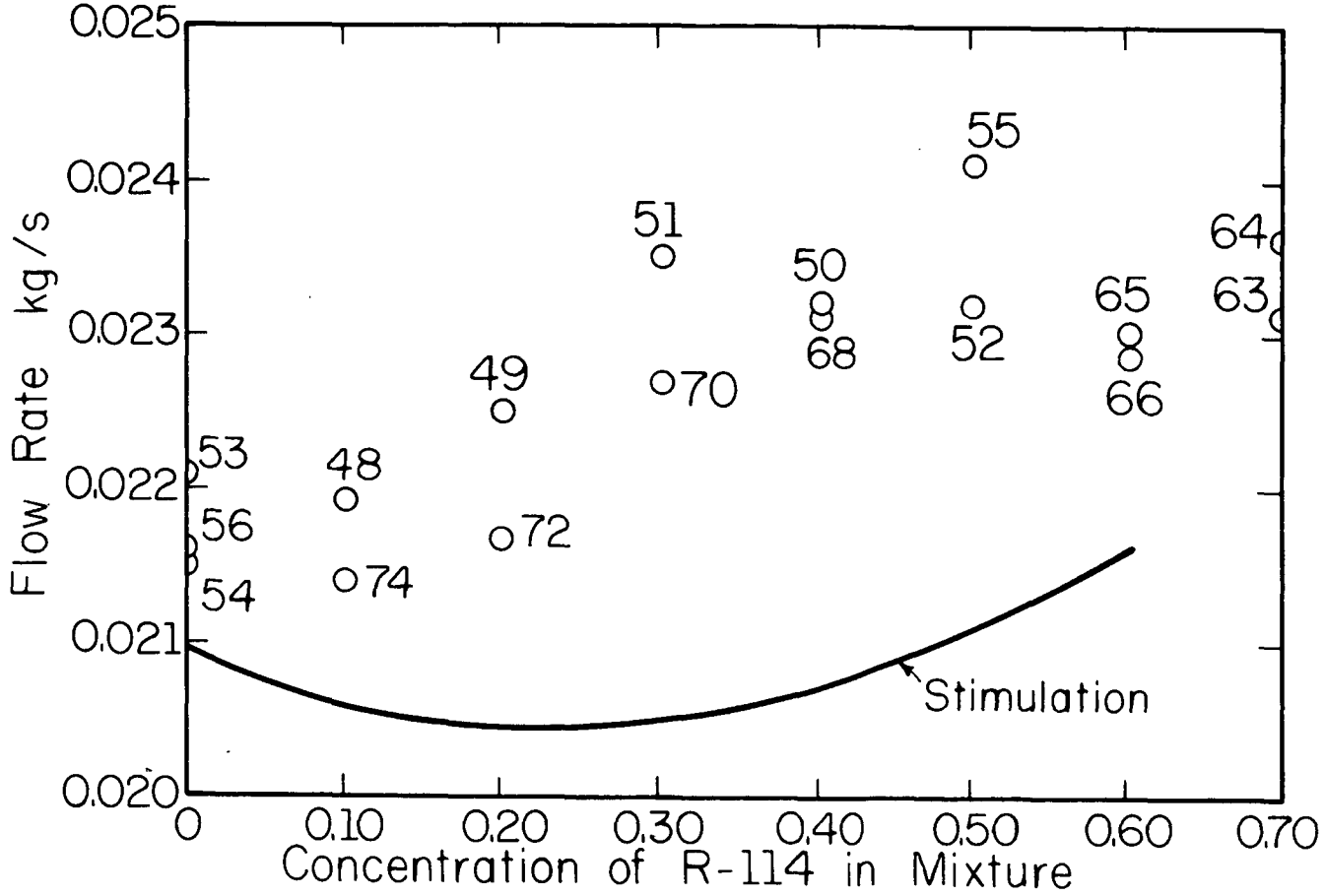


Figure 6.3 Refrigerant Flow Rate (Data Identified by Test Number)

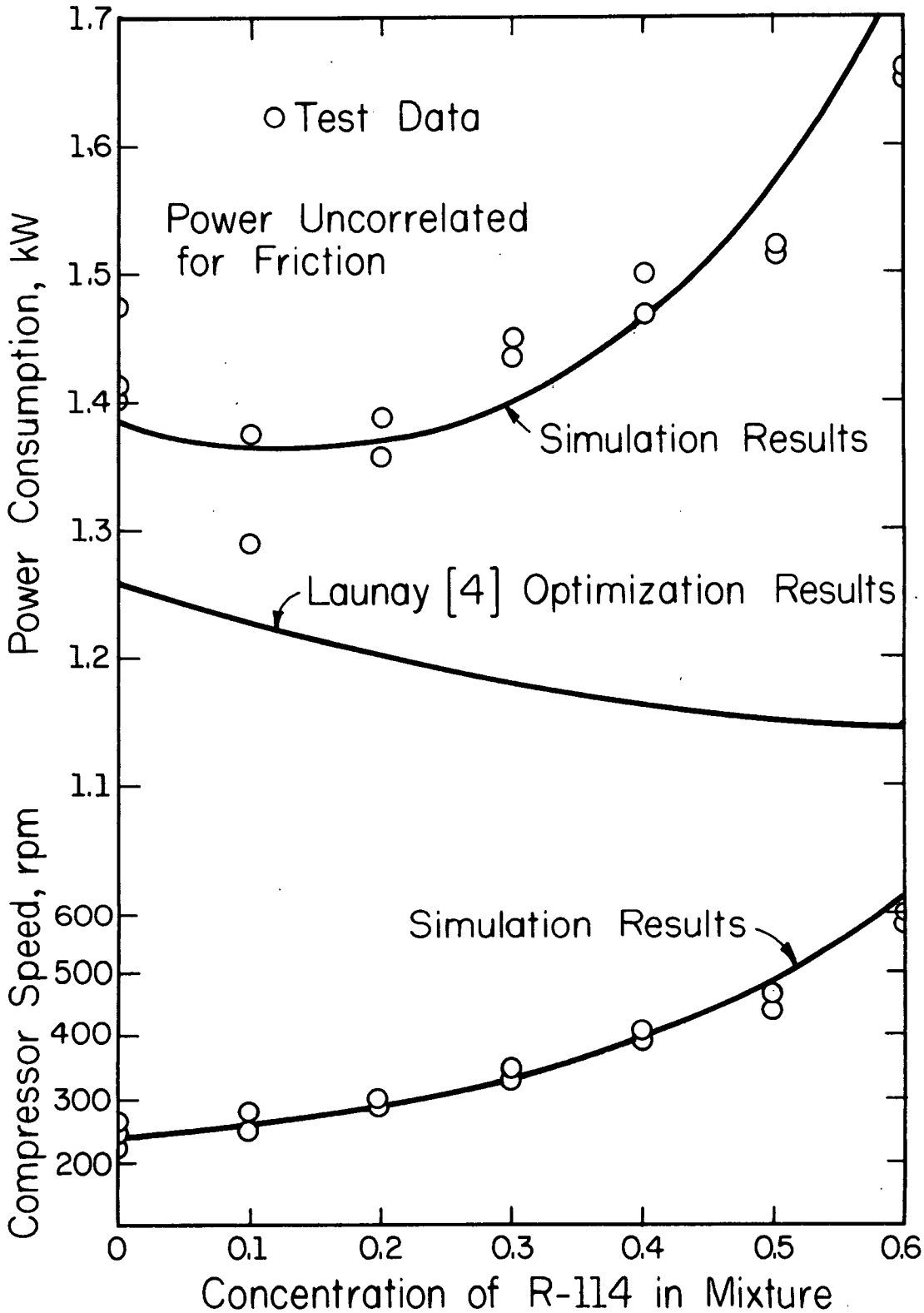


Figure 6.4 Compressor Power and Speed as Functions of Mixture Concentration for Constant Refrigerating Capacity

6.6 Compressor Power

The ultimate advantage that nonazeotropic mixtures must demonstrate is the reduction in the power requirement over single component refrigerants. The initial simulation and optimization study conducted by Launay indicated that power requirement for a given refrigeration duty would decrease with increasing concentration of R-114 to a minimum at about 68 percent R-114 (Figure 6.4). Energy savings would be about 8 percent over pure R-12. However, the test results did not agree. Instead, the optimum concentration was found to be around 15 percent R-114. The percent savings in power requirement was found to be about 2 percent (Figure 6.4).

The compressor was partly responsible for the power increase since the frictional losses increase with the square of the compressor speed. But there are ways to remove this influence from the results to see if it is the primary factor. The compressor power results can be plotted with the friction losses subtracted as was done in Figure 4.6. The power even with friction deducted still increases after reaching a minimum at around 10 percent R-114.

Another way to look at power consumption is to compute the power of isentropic compression. Enough properties are available to calculate the enthalpy and entropy of the suction vapor. The enthalpy change resulting from an isentropic increase in pressure from suction to discharge can also be calculated. Multiplying the enthalpy change by the mass rate of flow gives the power of isentropic compression. Measuring power consumption this way eliminates the influence of compressor behavior at different speeds. The results then become those that would be expected if an ideal compressor were used at each concentration of R-114. These isentropic results are presented in Figure 6.5. The power consumption is lower but still increases as the

concentration increases past an optimum of about 10 percent R-114. The increase in power consumption at greater concentrations is attributable to the lower heat transfer coefficients. Table 6.1 lists the values used to calculate the power of isentropic compression.

It is interesting to note the similarity between the Figures 4.6, 6.4, and 6.5. The plot of the compressor power minus the frictional losses (Figure 4.6) essentially shows what ideal compression would look like. Figure 6.5 shows the results assuming isentropic compression. The flow rate results in Figure 6.3 can give an indication of the expected relationship between power and concentration because the refrigerating capacity is the same in each test. In each figure there appears to be a minimum between 10 and 20 percent R-114, with a substantial power increase at higher concentrations.

6.7 Refrigerant Pressures

Figure 6.6 shows the comparison between pressures measured during tests and those resulting from the computer simulation. The discharge pressures did not match exactly but the relationship between pressure and concentration is the same.

The discrepency may be due to the lower accuracy of the high pressure gauge. The simulation recommendations in Chapter 7 may also help move the simulation results closer to the test data. The pressures decrease with increasing R-114 concentration so that the saturation temperatures remain at approximately the same level. It should also be mentioned that the pressure ratio (discharge/suction) increases with increasing concentration of R-114 in both the test results and the simulation results. This is a penalty that puts greater load on the compressor motor when higher concentrations of R-114 are used.

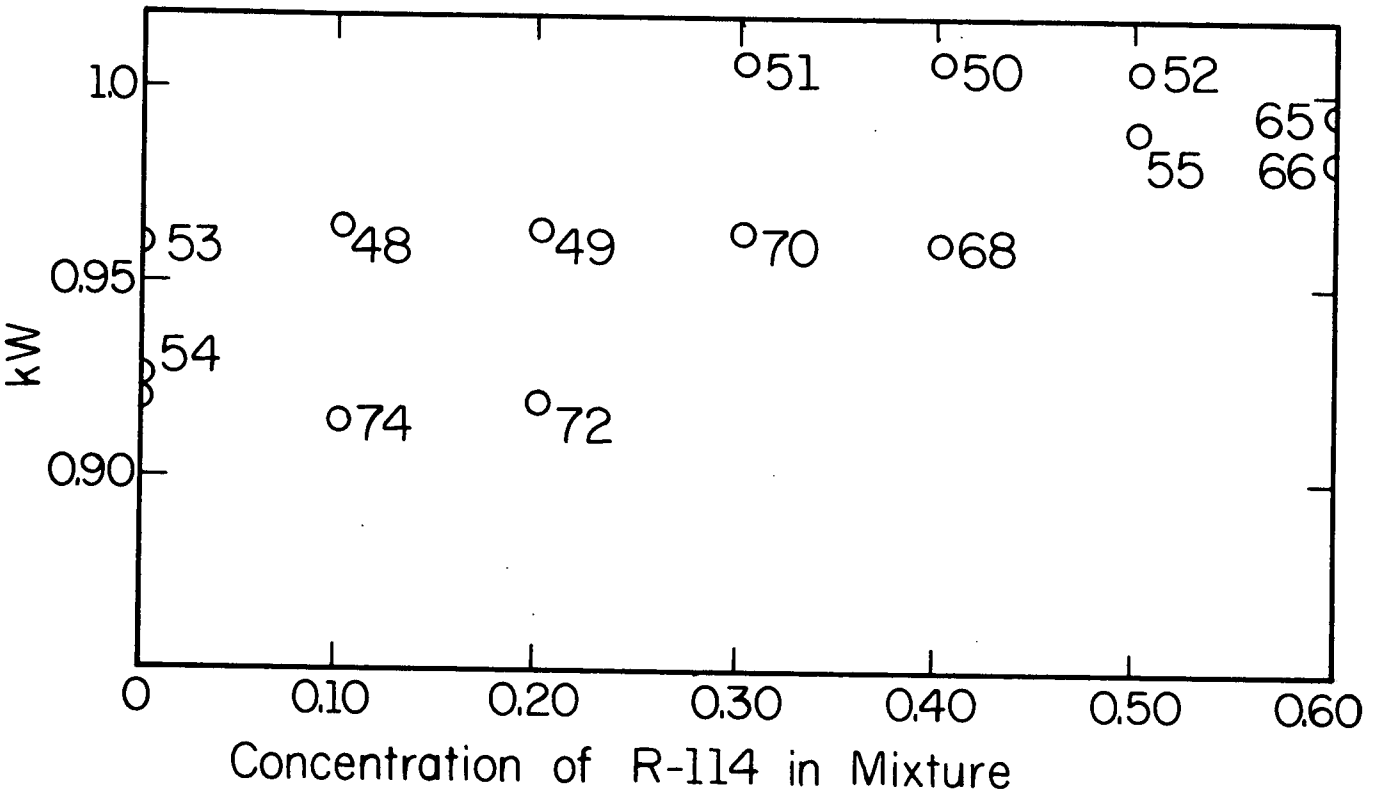


Figure 6.5 Power of Isentropic Compression (Data Identified by Test Number)

Table 6.1 Work of Isentropic Compression

Test Number	Percent R-114	Suction Pressure (kPa)	Suction Temperature °C	Discharge Pressure (kPa)	Isentropic Enthalpy Rise (kJ/kg)
48	10	116.5	11.5	989	44.04
49	20	108.9	9.2	922	42.78
50	40	81.4	8.1	786	43.46
51	30	95.0	6.6	857	42.86
52	50	69.4	5.9	720	43.29
53	0	132.0	11.4	1064	43.39
54	0	134.0	12.5	1058	43.08
55	50	76.4	3.0	727	41.02
56	0	137.0	11.4	1069	42.57
63	70	47.0	8.2	587	43.96
64	70	47.0	6.4	589	43.74
65	60	58.5	6.5	649	43.19
66	60	59.6	7.8	650	43.02
67*	60	59.4	5.2	729	44.93
68	40	85.2	1.6	791	41.57
69*	40	85.2	4.1	804	42.30
70	30	98.7	7.8	865	42.38
71*	30	97.2	0.5	877	41.84
72	20	110.6	9.9	914	42.34
73*	20	109.6	8.0	931	42.64
74	10	123.4	10.7	999	42.73
75*	10	123.9	9.5	1009	42.65
76*	0	131.2	10.0	1049	42.98

*HX2 Bypassed (not included in figures)

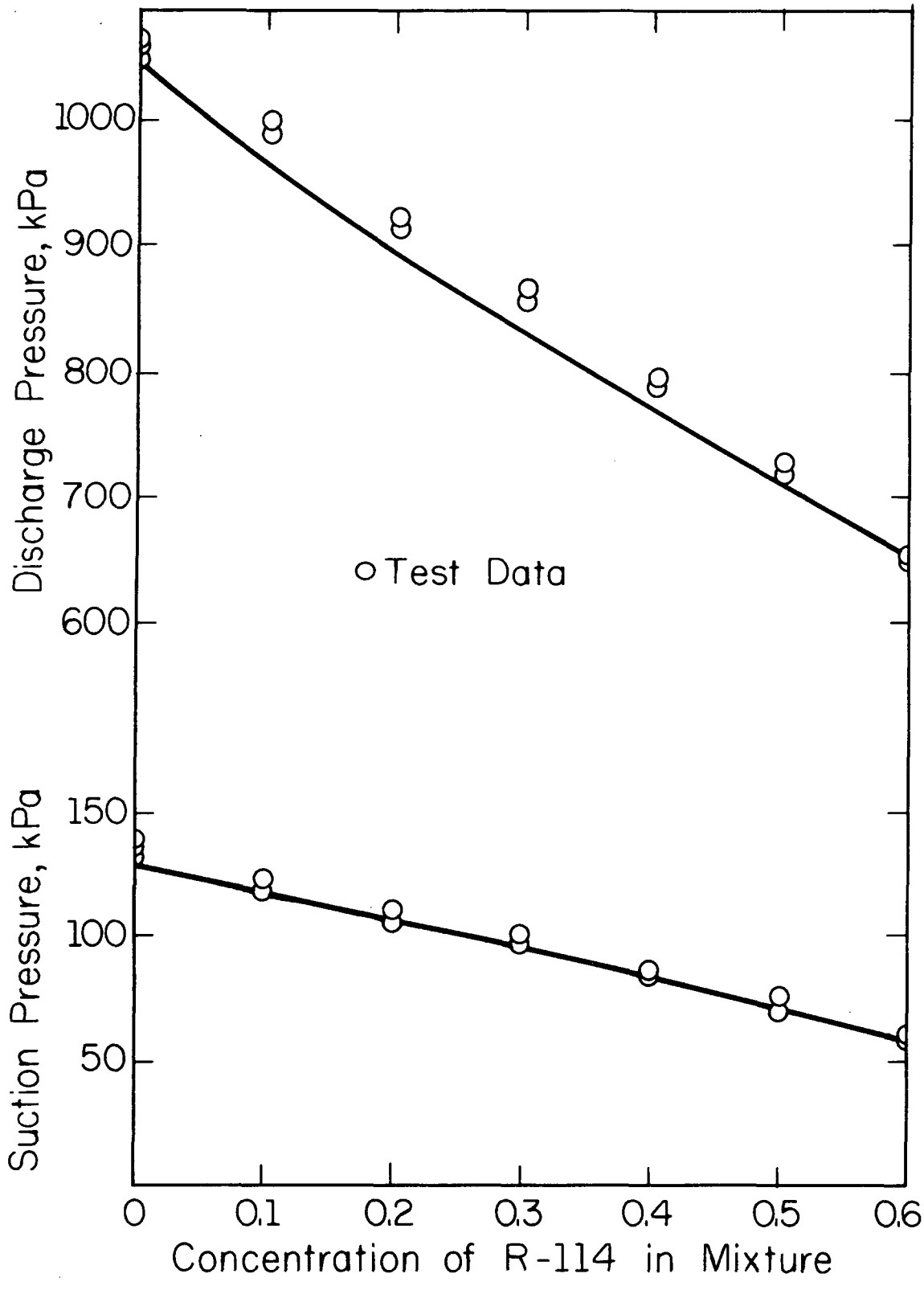


Figure 6.6 Compressor Suction and Discharge Pressures as Functions of Mixture Concentration for Constant Refrigerating Capacity

CHAPTER 7

SUMMARY AND RECOMMENDATIONS

7.1 Energy Conservation Possibility

At a concentration of about 15 percent R-114 there appears to be a slight reduction in power of about two percent. At higher concentrations, the low heat transfer coefficients, and high specific volume of the suction vapor combine to increase power requirement over pure R-12. If the heat transfer properties of the mixture can be improved, the savings will be more significant.

7.2 Future System Studies

More information on the heat transfer characteristics of mixtures is needed. The poor heat transfer coefficients appear to be a major drawback. As more data are collected on mixture heat transfer coefficients as a function of velocity, it will be possible to compute an optimum pipe diameter for the evaporating refrigerant. As the diameter is reduced the velocity increases which improves the heat transfer coefficients, however, if the velocity gets too high, the pressure drop becomes excessive. The present evaporator pipe diameter is probably larger than optimum, considering the low heat transfer coefficients and small pressure drop. The system also allows for the possibility of using different refrigerant pipe sizes in each of the evaporators. The optimum diameter for the LTE would probably be smaller than for the HTE.

Since the heat transfer coefficients vary with the heat flux, it would be helpful to conduct tests with different heat fluxes in both the condensers and evaporators. Changing the glycol temperature levels would accomplish this.

Since the refrigerant flow velocity has a significant effect, tests could be run at higher refrigerant flow rates.

The results obtained with the HX2 bypassed indicate that although some subcooling is desirable from a thermodynamic standpoint, the lower heat transfer coefficient reduction erases any thermodynamic gains. It would be best to conduct any future test series with the heat exchanger bypassed.

It would also be useful to conduct a series of tests from the orientation of a heat pump where the heat rejection rate is kept constant. There is reason to believe that nonazeotropic mixtures can save energy in heat pump applications as well as in refrigeration, especially if capacity modulation is employed.

7.3 Equipment Modifications

While we are not yet at a point of demonstrating substantial savings using mixtures of R-12 and R-114, other refrigerant mixtures may be superior. Pressure limitations of the test facility precluded testing of the mixtures R-22/R-114, for example, but these may be viable nonazeotropic refrigerant mixtures depending on their heat transfer properties. Should other mixtures show promise, the test facility can be modified to accommodate the higher working pressures. The high-pressure gauges must be replaced along with the refrigerant rotameter which has a pressure limit of 150 psi gauge (1034 kPa). At 40°C R-22 has a condensing pressure of over 1300 kPa gauge, which exceeds the limits of both the gauges and the rotameters. Once reliable heat transfer information is available, the simulation program can be modified to explore the use of these mixtures to see if they have potential for energy savings. Heat pump performance can also be investigated by simulation on the computer.

As mentioned earlier in the error analysis, the compressor motor speed must be monitored more accurately. Originally, an optimum concentration was expected at around 60 percent R-114. Since the compressor speed is quite high at higher concentrations of R-114, the power calculations are less sensitive to slight errors in the speed measurements. Now that the optimum appears to be at low concentrations of R-114 where the speed is slow, slight error in the speed measurement will have a greater effect on results.

A crucial measurement is the refrigerant flow rate, and the scatter of the measured values for what is intended to be an identical test is of the order of three to four percent. It may be necessary to install an additional flow measuring device. Also the actual temperatures of the chilled glycol solutions could never be precisely maintained from one test to another. It is appropriate to explore correcting the power for deviations in the actual refrigeration capacity.

7.4 Simulation Recommendations

The accuracy of the simulation is surely no better than the accuracy of the equations used to describe physical properties. The refrigerant equations published by Downing [5] should replace the equations currently employed. Properties which are not yet included as simulation variables should be so included. The refrigerant heat transfer coefficients which are expressed only as functions of concentration should be expressed as functions of flow velocity and heat flux, also. Pressure drops should be made a function of both concentration and flow velocity. However, the velocity term should dominate because the difference in viscosity between R-12 and R-114 is not as influential as the difference in specific volume at a given temperture. Volumetric efficiency would be better expressed as a function of three

variables: compressor speed, suction specific volume and the ratio of discharge to suction pressure. Test results have shown the degree of superheat on the suction side to vary from test to test. Therefore, it may be more appropriate to include superheat as a simulation variable rather than as a fixed parameter.

Since the simulated refrigerant temperature profiles in the evaporator were unrealistic at several of the concentrations, the simulation will have to be modified.

7.5 Conclusions Reached at this Stage of Refrigerant Mixture Study

Thermodynamic analysis shows the potential for conserving energy by substituting a nonazeotopic mixture of refrigerants for a single one. One of the goals of this current study is to experimentally verify that possibility. The test results indicate that a slight reduction in power occurs when 10 to 20 percent R-114 is added to R-12. With higher concentrations of R-114, the penalty due to reduced heat-transfer coefficients nullifies the potential energy saving.

This phase of the refrigerant mixture study brings us to greater realism in the struggle to achieve benefits in the use of refrigerant mixtures. It indicates that the task will not be easy and that the heat-transfer characteristics will have to be addressed along with continued system studies. The challenge is to maintain as nearly as possible the coefficients prevailing for the single working fluid. While there are some fundamental processes against which to work, there are also several approaches that now seem necessary. In the first place, a system study is required that attacks the tradeoff between high pressure drop and improved heat-transfer coefficients attributable to higher velocities in the tubes. That tradeoff is

one perpetually met by designers of heat-exchangers. There may be new considerations with mixtures that have not been a part of the single-refrigerant analysis. The second approach would be to explore techniques such as enhancing heat transfer (by artificial tube roughness) in order to determine whether the benefits are more pronounced with mixtures than with single refrigerants.

Energy conservation through the use of refrigerant mixtures still represents a prospect, although some of the difficulties in achieving the improved efficiency are now becoming apparent. Energy conservation is not the only reason that engineers are interested today in refrigerant mixtures. Two other potential benefits are (1) capacity leveling when air-source heat pumps operate at low ambient temperatues, and (2) tailoring the pressure-temperature characteristics for high-temperature heat pump [6]. In these applications as well as others, the cycle efficiencies will continue to be prominent. Continued research on the energy and heat-transfer characteristics is necessary to evaluate the most attractive applications for nonaqeotropic mixture of refrigerants.

REFERENCES

1. Lorenz, A., "The Utilization of Binary Refrigerant Mixtures in Compression Refrigeration Systems," Luft- und Kältetechnik, Vol. 9, No. 6, Dec. 1973, pp. 296-301.

2. Eiseman, Jr., B. J., "Multicomponent Refrigerants," Proc. of Meetings of Commissions 3 and 6, Int. Institute of Refrigeration, Prague, 1965.

3. Arora, C. P., "Boiling Heat Transfer data for Pure and Mixed Refrigerants," Proc. XII, Int. Congress of Refrigeration, Madrid, 1967.

4. Launay, P. F., "Improving the Efficiency of Refrigerators and Heat Pumps by using a Nonazeotropic Mixture of Refrigerants," Report No. ORNL/Sub 81/7762/1, Oak Ridge National Laboratory, October 1981.

5. Downing, R. C., "Refrigerant Equations," ASHRAE Transactions, Vol. 80, Part II, 1974.

6. Jakobs, R., and H. Kruse, "The Use of Non-Azeotropic Refrigerant Mixtures in Heat Pumps for Energy Saving," Proc. of Meeting of Commission B2, Int. Institute of Refrigeration, Delft, 1978.

7. Stoecker, W. F., Design of Thermal Systems, 2nd edition, McGraw-Hill, New York, 1980.

****** TABLE OF REFRIGERANT MIXTURES - R-12 AND R114 ******
****** CONDENSING AND BOILING TEMPERATURES ******
CALCULATED BY K.MATSUO SEP 10,1981
1ST LINE = CONDENSING TEMP. 2ND LINE = BOILING TEMP.

PRESS. (KPA)	WEIGHT FRACTION OF R114 0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
10.0	-74.22	-65.22	-60.04	-56.37	-53.58	-51.34	-49.52	-47.99	-46.71	-45.63	-44.75
	-74.22	-72.23	-70.07	-67.74	-65.21	-62.47	-59.45	-56.17	-52.63	-48.80	-44.75
20.0	-62.59	-53.66	-48.30	-44.45	-41.47	-39.09	-37.12	-35.49	-34.11	-32.95	-31.99
	-62.59	-60.43	-58.08	-55.57	-52.86	-49.93	-46.78	-43.37	-39.75	-35.94	-31.99
30.0	-55.14	-46.26	-40.78	-36.82	-33.74	-31.26	-29.22	-27.51	-26.07	-24.85	-23.84
	-55.14	-52.85	-50.41	-47.78	-44.95	-41.92	-38.67	-35.21	-31.56	-27.74	-23.84
40.0	-49.53	-40.67	-35.12	-31.07	-27.92	-25.37	-23.26	-21.51	-20.02	-18.77	-17.72
	-49.53	-47.15	-44.62	-41.90	-39.00	-35.89	-32.58	-29.07	-25.39	-21.58	-17.72
50.0	-44.97	-36.14	-30.53	-26.42	-23.20	-20.60	-18.44	-16.64	-15.12	-13.84	-12.76
	-44.97	-42.53	-39.92	-37.14	-34.17	-31.00	-27.64	-24.10	-20.40	-16.60	-12.76
60.0	-41.11	-32.29	-26.64	-22.47	-19.21	-16.56	-14.36	-12.53	-10.97	-9.67	-8.56
	-41.11	-38.61	-35.94	-33.10	-30.07	-26.85	-23.45	-19.88	-16.17	-12.38	-8.56
70.0	-37.74	-28.94	-23.24	-19.03	-15.72	-13.04	-10.81	-8.94	-7.36	-6.03	-4.91
	-37.74	-35.19	-32.47	-29.58	-26.50	-23.25	-19.81	-16.22	-12.50	-8.70	-4.91
80.0	-34.74	-25.95	-20.22	-15.97	-12.63	-9.91	-7.65	-5.75	-4.15	-2.80	-1.66
	-34.74	-32.14	-29.38	-26.44	-23.33	-20.04	-16.57	-12.96	-9.23	-5.43	-1.66
90.0	-32.03	-23.26	-17.50	-13.21	-9.83	-7.08	-4.80	-2.88	-1.25	.12	1.27
	-32.03	-29.39	-26.59	-23.61	-20.46	-17.14	-13.65	-10.02	-6.27	-2.49	1.27
100.0	-29.55	-20.79	-15.00	-10.69	-7.28	-4.50	-2.19	-.25	1.39	2.78	3.95
	-29.55	-26.88	-24.03	-21.03	-17.84	-14.49	-10.98	-7.33	-3.58	.20	3.95
125.0	-24.13	-15.40	-9.56	-5.17	-1.70	1.14	3.50	5.49	7.17	8.60	9.80
	-24.13	-21.38	-18.46	-15.37	-12.12	-8.71	-5.14	-1.46	2.30	6.08	9.79
150.0	-19.52	-10.81	-4.92	-.49	3.04	5.93	8.33	10.36	12.08	13.53	14.76
	-19.52	-16.70	-13.72	-10.56	-7.25	-3.79	-.19	3.52	7.29	11.06	14.76
175.0	-15.49	-6.80	-.87	3.61	7.18	10.11	12.55	14.62	16.36	17.84	19.09
	-15.49	-12.61	-9.57	-6.36	-3.00	.50	4.14	7.87	11.65	15.42	19.09
200.0	-11.89	-3.22	2.74	7.26	10.87	13.64	16.32	18.41	20.18	21.68	22.95
	-11.89	-8.96	-5.87	-2.62	.79	4.33	8.00	11.75	15.53	19.29	22.95
225.0	-8.64	.03	6.01	10.57	14.21	17.21	19.72	21.84	23.63	25.16	26.44
	-8.64	-5.65	-2.52	.78	4.22	7.80	11.49	15.25	19.04	22.80	26.44
250.0	-5.65	3.00	9.01	13.60	17.27	20.30	22.83	24.97	26.79	28.33	29.64
	-5.65	-2.63	.55	3.88	7.36	10.97	14.68	18.46	22.25	26.01	29.63
275.0	-2.89	5.74	11.77	16.39	20.10	23.15	25.71	27.87	29.71	31.27	32.58
	-2.89	.17	3.39	6.75	10.26	13.89	17.62	21.42	25.22	28.97	32.58
300.0	-.33	8.30	14.35	19.00	22.73	25.80	28.38	30.57	32.42	34.00	35.32
	-.33	2.77	6.03	9.42	12.96	16.62	20.36	24.17	27.97	31.72	35.32

PRESS. (KPA)	WEIGHT FRACTION OF R114 0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
325.0	2.08	10.70	16.76	21.43	25.19	28.29	30.89	33.09	34.96	36.55	37.89
	2.08	5.21	8.50	11.93	15.49	19.17	22.93	26.75	30.56	34.30	37.89
350.0	4.34	12.95	19.04	23.73	27.50	30.62	33.24	35.46	37.35	38.95	40.30
	4.34	7.51	10.83	14.28	17.87	21.57	25.35	29.17	32.98	36.72	40.30
375.0	6.49	15.08	21.19	25.89	29.69	32.83	35.47	37.71	39.61	41.22	42.59
	6.49	9.68	13.03	16.51	20.12	23.83	27.63	31.46	35.28	39.01	42.59
400.0	8.52	17.11	23.22	27.95	31.77	34.93	37.58	39.83	41.75	43.37	44.75
	8.52	11.75	15.12	18.62	22.25	25.99	29.80	33.63	37.45	41.18	44.75
425.0	10.46	19.04	25.17	29.91	33.75	36.93	39.60	41.86	43.79	45.42	46.81
	10.46	13.71	17.11	20.64	24.29	28.04	31.86	35.70	39.52	43.25	46.81
450.0	12.31	20.88	27.02	31.79	35.64	38.83	41.52	43.80	45.73	47.38	48.78
	12.31	15.59	19.01	22.56	26.23	30.00	33.83	37.68	41.50	45.22	48.78
475.0	14.09	22.65	28.80	33.58	37.45	40.66	43.36	45.65	47.60	49.26	50.66
	14.09	17.39	20.83	24.40	28.09	31.87	35.71	39.57	43.39	47.11	50.66
500.0	15.79	24.35	30.51	35.30	39.19	42.41	45.12	47.43	49.39	51.06	52.47
	15.79	19.12	22.58	26.17	29.87	33.67	37.52	41.39	45.21	48.93	52.47
525.0	17.43	25.98	32.15	36.96	40.86	44.10	46.82	49.14	51.11	52.78	54.20
	17.43	20.78	24.26	27.87	31.59	35.40	39.26	43.13	46.96	50.67	54.20
550.0	19.01	27.55	33.74	38.56	42.47	45.72	48.46	50.78	52.77	54.45	55.88
	19.01	22.38	25.88	29.51	33.25	37.07	40.94	44.81	48.64	52.35	55.88
575.0	20.54	29.07	35.26	40.10	44.02	47.29	50.04	52.37	54.37	56.06	57.49
	20.54	23.93	27.45	31.09	34.85	38.68	42.56	46.44	50.27	53.97	57.49
600.0	22.02	30.54	36.74	41.59	45.53	48.80	51.57	53.91	55.91	57.61	59.06
	22.02	25.42	28.96	32.63	36.39	40.24	44.12	48.01	51.84	55.54	59.06
625.0	23.45	31.97	38.18	43.03	46.98	50.27	53.04	55.40	57.41	59.12	60.57
	23.45	26.87	30.43	34.11	37.89	41.74	45.64	49.53	53.35	57.06	60.57
650.0	24.83	33.35	39.56	44.43	48.40	51.69	54.48	56.84	58.86	60.58	62.03
	24.83	28.28	31.85	35.55	39.34	43.20	47.11	51.00	54.83	58.53	62.03
675.0	26.18	34.69	40.91	45.79	49.77	53.08	55.87	58.24	60.27	62.00	63.46
	26.18	29.64	33.23	36.94	40.75	44.62	48.53	52.43	56.26	59.96	63.46
700.0	27.49	35.99	42.22	47.11	51.10	54.42	57.22	59.61	61.64	63.37	64.84
	27.49	30.97	34.58	38.30	42.12	46.00	49.92	53.82	57.65	61.35	64.84
725.0	28.77	37.26	43.50	48.40	52.40	55.73	58.54	60.93	62.97	64.71	66.18
	28.77	32.26	35.88	39.62	43.45	47.34	51.27	55.17	59.00	62.69	66.18
750.0	30.01	38.50	44.74	49.65	53.66	57.00	59.82	62.22	64.27	66.01	67.49
	30.01	33.52	37.16	40.91	44.75	48.65	52.58	56.49	60.32	64.01	67.49
775.0	31.22	39.71	45.95	50.87	54.89	58.24	61.07	63.48	65.53	67.28	68.77
	31.22	34.75	38.40	42.16	46.01	49.92	53.86	57.77	61.60	65.29	68.77
800.0	32.40	40.88	47.14	52.07	56.09	59.45	62.29	64.70	66.77	68.52	70.01
	32.40	35.94	39.61	43.38	47.24	51.17	55.11	59.02	62.85	66.54	70.01

APPENDIX B

Sample Data Recording Sheet

REFRIGERANT MIXTURE DATA SHEET

DATE 12-3-82, TIME 1:45

REFRIGERANT R-114, 20 % AND R-12, 80 %

RECORDER Fisher Test 72

GAUGE PRESSURES: (BAROMETRIC = 98.56 KPA)

1. COMPRESSOR DISCHARGE, 815 KPA
2. LEAVING HTC, 814 KPA
3. LEAVING LTC, 811 KPA
4. ENTERING EXPANSION VALVE, 810 KPA
5. LEAVING EXPANSION VALVE, 20.5 KPA
6. AFTER 1ST PASS LTE, 19.5 KPA
7. AFTER 2ND PASS LTE, 19.5 KPA
8. LEAVING LTE, 19.0 KPA
9. ENTERING HTE, 19.0 KPA
10. MIDWAY IN HTE, 17.5 KPA
11. LEAVING HTE, 14.5 KPA
12. ENTERING VENTURI, — KPA
13. VENTURI THROAT, — KPA
14. COMPRESSOR SUCTION, 13.0 KPA

FLOW RATES AT ROTAMETERS:

1. REFRIG. 44.5 % (100% = 0.67 GPM)
2. HTC WATER ~~47.3~~ 70 % (100% = 2.06 GPM)
3. LTC WATER 70 % (100% = 2.06 GPM)
4. LTE GLYCOL 41.9 %, GRAV: 1.049 @ 23.5 C (1.96 GPM)
5. HTE GLYCOL 47.3 %, GRAV: 1.038 @ 23.5 C (1.96 GPM)

LTE HEATER: VOLTS= 143, AMPS= 3.55

HTE HEATER: VOLTS= 135, AMPS= 6.65

DC MOTOR : SPEED= 550 +75 RPM, LOAD= 11.2 LB.

COMMENTS: Total charge 14.1 16.

59		9.9	C
58	-	4.9	C
57	-	6.5	C
56	-	2.5	C
55	-	2.5	C
54	-	4.3	C
53	-	5.3	C
52	-	5.5	C
51	-	1.8	C
50	-	9.4	C
49	-	11.4	C
48	-	16.8	C
47	-	16.8	C
46	-	18.3	C
45	-	20.5	C
44	-	14.2	C
43	-	15.0	C
42	-	10.3	C
41	-	10.6	C
40	-	13.5	C
39	-	11.5	C
38	-	11.8	C
37	-	9.2	C
36	-	9.6	C
35	-	7.9	C
34	-	21.3	C
33	-	21.7	C
32	-	22.0	C
31	-	21.7	C
30	-	22.5	C
29	-	22.5	C
28	-	1.0	C
27		29.1	C
26		35.6	C
25		37.1	C
24		0.1	C
23		38.9	C
22		37.2	C
21		35.6	C
20		34.1	C
19		41.0	C
18		39.7	C
17		41.7	C
16		42.1	C
15		42.6	C
14		42.9	C
13		40.9	C
12		42.0	C
11		43.0	C
10		43.4	C
9		41.5	C
8		40.0	C
7		38.4	C
6		43.3	C
5		43.9	C
4		44.4	C
3		45.6	C
2		41.6	C
1		62.5	C
0		81.9	C

13:42:09

APPENDIX C

Sample Results of a Data Reduction

```
 66.5   43.1   46.8   45.4   45.0   44.6   40.1   41.8   43.0   44.9
 44.2   42.9   42.1   44.1   43.9   43.5   43.3   42.6   41.4   34.4
 36.4   38.3   40.1    .1    38.3   36.2   29.9   -1.1  -22.2  -22.5
-21.7  -22.1  -21.7  -21.7   -7.7   -9.5   -8.8  -11.7  -11.3  -13.5
-10.6  -10.4  -15.0  -14.1  -21.1  -18.7  -20.8  -19.0   -9.9    0.0
 -1.1   -3.0   -2.8   -6.4   -2.2   -2.0   -6.4   -4.3   11.5   89.6
890.0  890.0  888.0  883.0   26.5   26.5   26.1   27.5   25.5   23.0
 20.5   -0.0    0.0   17.5   98.9
 45.0   70.0   70.0   42.6   47.7    1.1   26.9    1.0   26.9
142.5   -3.5  124.0    6.7
525.0   11.8
```

TEST# 48 DATE: 7- 7-82 TIME:14.25 MASS FRAC R114 IN R12: .100

TEMPERATURES T1 THRU T60, (CORRECTED IF NECESSARY)

```
 66.4   43.0   46.7   45.3   44.9   44.5   40.0   41.7   42.9   44.8
 44.1   42.8   42.0   44.0   43.8   43.4   43.2   42.5   41.3   34.3
 36.3   38.2   40.0    0.0   38.2   36.1   29.8   -1.2  -22.3  -22.6
-21.8  -22.2  -21.8  -21.8   -7.8   -9.6   -8.9  -11.8  -11.4  -13.6
-10.7  -10.5  -15.1  -14.2  -21.2  -18.8  -20.9  -19.1  -10.0    -.1
 -1.2   -3.1   -2.9   -6.5   -2.3   -2.1   -6.5   -4.4   11.4   89.5
```

ROTAMETER FLOWRATES (KG/S):
REFRIG: .0219 HTC: .0912 LTC: .0912 LTE: .0540 HTE: .0601

HEAT TRANSFER RATES (KW) FROM ROTAMETER FLOWRATES:
QHTC: 1.8292 QLTC: 2.1718 QLTE: 1.0389 QHTE: 1.4050
COOLING CAPACITY: 2.4440
HEAT PUMP CAPACITY: 4.0010
TOTAL HEAT BALANCE=(QR+WDYNO)/QC: .9005

REFRIGERANT HEAT TRANSFER COEFFICIENTS, (W/M**2*K):

```
HTC1:  804.   HTC2: 4239.   HTC3: 1800.   LTC2: 1103.
LTC3:  990.   LTE1:  207.   LTE2:  233.   HTE1:  285.
HTE2:  317.
```

REFRIGERANT-SIDE AREA U-VALUES (W/M**2*K):

```
HTC1:  795.   HTC2: 3758.   HTC3: 1289.   LTC1: 1094.
LTC2:  933.   LTC3:  880.   LTE1:  156.   LTE2:  162.
LTE3: -223.   HTE1:  250.   HTE2:  266.   HTC :  279.
LTC : 1027.   LTE :  169.   HTE :  314.
```

COMPRESSOR WORK FROM DYNO: 1.16KW FROM REF ENTHALPY: .92KW

HEAT REJ FROM REF IN HTC+LTC: 3.5565 % DEV FROM WATER-SIDE: -11.1

GLYCOL FLOWRATE FROM HEATER LOAD, AND % DEVIATION FROM ROTAMETER RATE:
LTE: .0519 KG/S, % DEV: -3.9 HTE: .0542 KG/S, % DEV: -9.8

REF FLOWRATE FROM CAPACITY AND ENTHALPY, % DEV FROM ROTAMETER:
.0203 KG/S % DEV: -7.4

GLYCOL CONCENTRATION: LTE: 40.89 HTE: 31.15

PRINT (V9.8) REQUEST 82/07/09. 11.05.03. RJE*ME UN*3KURR38
FILE: OUTT /SEG

APPENDIX D

Simulation Program Listing

```
      PROGRAM SIMUL7(TAPE1,OUTPUT,TAPE6=OUTPUT)
C     ****************************************

C     LATEST REVISIONS MADE: 12-14-82
C
C     THIS PROGRAM SIMULATES THE PERFORMANCE OF A 2-CONDENSER,
C     2-EVPORATOR, 2-HEAT EXCHANGER VAPOR COMPRESSION REFRIGERATION
C     SYSTEM USING MIXTURES OF R-114 AND R-12.  IT IS BASED ON WORK
C     DONE BY PHILIPPE LAUNAY DURING COMPLETION OF HIS MASTERS THESIS
C     AT THE UNIVERSITY OF ILLINOIS IN 1980. THIS VERSION INCLUDES
C     VARIABLE HEAT TRANSFER COEFFICIENTS, VARIABLE PRESSURE DROP,
C     VARIABLE VOLUMETRIC EFFICIENCY AND VARIABLE ISENTROPIC EFFICIENCY.
C     THE FRICTION LOAD ON THE COMPRESSOR AS A FUNCTION OF SPEED IS ALSO
C     INCLUDED.  FOR FLOW DIAGRAM OF SIMULATED SYSTEM INCLUDING VARIABLE
C     DEFINITIONS, SEE "PERFORMANCE OF NON-AZEOTROPIC MIXTURES OF
C     DICHLORODIFLUOROMETHANE (R-12) AND DICHLOROTETRAFLUOROETHANE
C     (R-114) IN A VAPOR COMPRESSION REFRIGERATION SYSTEM"
C     C. I. MCCARTHY, MASTERS THESIS, UNIVERSITY OF ILLINOIS 1983
C
      DIMENSION V(45),DES(45),R(45),PD(45,45),VCORR(45),VD(45),RD(45),
     * DAT(17),DATNAME(17),DATUNIT(17),UNITV(45)

      INTEGER DAMPING,POPTION
      COMMON DAT


       NVAR=45
       TLRNCE=0.001
       ITMAX =15

C     THESE NEXT VARIABLES ARE REQUIRED BY SUBROUTINE SIMUL.
       DAMPING = 0
       POPTION=0
       ITER=0
C
CC    READING THE TRIAL VALUES STORED IN INPUT FILE (WHEN RUNNING
CC    ON CYBER "/LGO,INPUTFILENAME,OUTPUTFILEMANE")
C
```

```
      REWIND 1
      READ (1,10)
      READ (1,12) (DATNAME(I),DAT(I),DATUNIT(I),I=1,17)
      READ (1,10)
      READ (1,11) (DES(I),V(I),UNITV(I),I=1,NVAR)

C*****OPTIONAL PRINTOUT OF INPUT FILE INFORMATION
C     PRINT 19,(K,DATNAME(K),DAT(K),DATUNIT(K),K=1,17)
C     PRINT 17,(I,DES(I),V(I),UNITV(I),I=1,NVAR)

      CALL SIMUL(NVAR,TLRNCE,ITMAX,V,DES,R,PD,VCORR,VD,RD,DAMPING,
     *  POPTION,ITER)
      PRINT 20,ITER,(K,DATNAME(K),DAT(K),DATUNIT(K),K=1,17)
      PRINT 17,(I,DES(I),V(I),UNITV(I),I=1,NVAR)
C
      STOP
   10 FORMAT (/)
   11 FORMAT (9X,A4,3X,F21.5,3X,A5)
   12 FORMAT(20X,A4,3X,F8.4,3X,A5)
   17 FORMAT(45(1X,"V(",I2,") = ",A4," = ",F21.5,3X,A5,/),/)
   19 FORMAT(10(/),"1",4X,"VALUES FROM INPUT FILE",/,5X,27("*"),
     *  /,17(10X,"DAT(",I2,") = ",A4," = ",F8.4,3X,A5,/))
   20 FORMAT(10(/),10X,"AFTER",I3," ITERATIONS SOLUTION IS:",/
     * 10X,32("*")/17(10X,"DAT(",I2,") = ",A4," = ",F8.4,3X,A5/)/)
      END


      SUBROUTINE EQNS(NVAR,V,R)
C     ************************
C     THIS SUBROUTINE CONTAINS ALL OF THE EQUATIONS THAT DESCRIBE THE
C     SYSTEM.
```

```
      IMPLICIT REAL (A - Z)
      INTEGER NVAR,I
      DIMENSION  V(NVAR),R(NVAR),DAT(17),A2(5),A4(5),B2(5),B4(5),
     C           C2(5),C4(5)
      COMMON DAT

C  THESE ARE CONSTANTS USED IN THE EQUATIONS WHICH CALCULATE THE
C  SPECIFIC VOLUME OF THE TWO REFRIGERANTS IN THE MIXTURE (R12,R114).
C  SOURCE: R.C. DOWNING, ASHRAE TRANSACTIONS, VOL. 80, PART II, 1974.

      DATA A2/0.0,-3.40972713,0.0602394465,-5.4873701E-4,0.0/
      DATA A4/0.0,-2.3856704,0.034055687,-3.857481E-4,1.6017659E-6/
      DATA B2/0.0,1.59434848E-3,-1.87961843E-5,0.0,3.468834E-9/
      DATA B4/0.0,1.0801207E-3,5.333469E-6,0.0,6.2632341E-10/
      DATA C2/0.0,-56.7627671,1.31139908,0.0,-2.54390678E-5/
      DATA C4/0.0,-6.5643648,0.16366057,0.0,-1.0165314E-5/

C
CC    DEFINING USEFUL FUNCTIONS (SAT.PRESS.,ENTHALP.,ETC.......)
C     P. LAUNAY 1980.
C
      P12S(T) =EXP(14.861 - 2498.3/((T)+273.15))
      P114S(T)=EXP(15.407 - 2993.2/((T)+273.15))
      H12V(T) =351.48 + .42825*(T) - .00071*(T)**2 - .0000051*(T)**3
      H114V(T)=337.41 + .6234*(T) + .000086*(T)**2 - .00000385*(T)**3
      H12L(T) =200. + .9251*(T) + .00081*(T)**2 + .0000048*(T)**3
      H114L(T)=200. + .9545*(T) + .00116*(T)**2 + .000003*(T)**3
      MOLFRAC(X)=(X)/(1.4135 - 0.4135*(X))
      MASFRAC(X)=1.4135*X/(1.+0.4135*X)
      XCL(P,T)=(P-P12S(T)) / (P114S(T)-P12S(T))
      XCV(P,T,X)=P114S(T)*X/P
C
C-----WORK OF ISENTROPIC COMPRESSION FOR R12 & R114
C     P. LAUNAY 1980.
```

```
      C2N1(T,P)=(T + 273.15)*((.6933E-1 + .232E-4*T) +
     *           (-.2019E-4 + .757E-7*T)*P)
      C2N2(T,P)=(T + 273.15)*((.414E-2 - .148E-4*T) +
     *           (-.876E-5 + .231E-6*T)*P)
      WIR12(T,P1,P2)=(C2N1(T,P1) + C2N2(T,P1)*ALOG(P2/P1))*ALOG(P2/P1)
      C4N1(T,P)=(T + 273.15)*((.4949E-1 + .36E-5*T) +
     *           (-.3024E-4 + .4083E-6*T)*P)
      C4N2(T,P)=(T + 273.15)*((.75E-3 + .285E-4*T) +
     *           (.428E-5 + .365E-6*T)*P)
      WIR114(T,P1,P2)=(C4N1(T,P1) + C4N2(T,P1)*ALOG(P2/P1))*ALOG(P2/P1)
C
C-----ENTHALPY OF SUPERHEATED VAPOR OF R12 & R114
C     P. LAUNAY 1980.

      PP4(P)=(P-251.66)/244.77
      PP2(P)=(P-609.81)/554.19
      TP4(T)=(T-10.)/55.556
      TP2(T)=(T-10.)/50.
      AA4(P) =341.528 - 4.400*PP4(P) - .432*PP4(P)**2
      BB4(P) = 39.811 + 2.054*PP4(P) + .282*PP4(P)**2
      CC4(P) =  1.475 -  .535*PP4(P) + .012*PP4(P)**2
      HR114SH(P,T)=AA4(P) + BB4(P)*TP4(T) + CC4(P)*TP4(T)**2
      AA2(P) =351.740 - 12.954*PP2(P) - 2.367*PP2(P)**2
      BB2(P) = 35.957 +  8.406*PP2(P) + 2.484*PP2(P)**2
      CC2(P) =  -.619 -  2.379*PP2(P) -  .748*PP2(P)**2
      HR12SH(P,T)=AA2(P) + BB2(P)*TP2(T) + CC2(P)*TP2(T)**2



C
CC     REDEFINING V(I) TO MAKE EQUATIONS EASIER TO UNDERSTAND
C
```

```
W    =V(1)
PHI  =V(2)
PLO  =V(3)
T1   =V(4)
T2   =V(5)
T2P  =V(6)
T3   =V(7)
T4   =V(8)
T5   =V(9)
T6   =V(10)
T7   =V(11)
T8   =V(12)
T9   =V(13)
T10  =V(14)
T10P=V(15)
H1   =V(16)
H2   =V(17)
H3   =V(18)
H4   =V(19)
H5   =V(20)
H6   =V(21)
H7   =V(22)
H8   =V(23)
H9   =V(24)
H10  =V(25)
FRL7=V(26)
FRL8=V(27)
FRL9=V(28)
FRL3=V(29)
FHTC =V(30)
FHTE=V(31)
QLTC=V(32)
QHTC=V(33)
TCHP=V(34)
AHTE=V(35)
TELO=V(36)
```

```
      TEHP=V(37)
      TEHO=V(38)
      TCLO=V(39)
      TCHO=V(40)
      POWR=V(41)
C-----THESE VARIABLES WERE ADDED 11-18-82 BY CHRIS MCCARTTHY.
      RPM =V(42)
      ETAV=V(43)
      VOL2=V(44)
      VOL4=V(45)
C
CC    IMPOSED PARAMETERS
C
      CONC=DAT(1)
      ALTE=DAT(2)
      AHX1=DAT(3)
      AHX2=DAT(4)
      ALTC=DAT(5)
      AHTC=DAT(6)
      WGEL=DAT(7)
      WGEH=DAT(8)
      WLTC=DAT(9)
      WHTC=DAT(10)
      QLTE=DAT(11)
      QHTE=DAT(12)
      TCLI=DAT(13)
      TCHI=TCLO
      TELI=DAT(15)
      TEHI=DAT(16)
      SPHX=DAT(17)
C
C     INCLUSION OF PRESSURE DROPS BASED ON TEST DATA. (MCCARTHY 9-16-82)
C
```

```
      P2  = PHI -  1.0
      P3  = PHI -  2.0
      P4  = PHI -  5.0
      P7  = PLO + 7.9 + 14.1*CONC + 1.4*CONC**2
      P8  = PLO + 6.8 + 11.8*CONC + 1.4*CONC**2
      P9  = PLO + 6.3 + 11.6*CONC + 1.4*CONC**2
      P10 = PLO + 3.4 +  5.4*CONC + 0.3*CONC**2

C
CC    PRELIMINARY COMPUTATIONS
C
C-----MASS CONCENTRATION OF R114 IN LIQ. & VAP. AT 3,7,8,9
      XCL7=XCL(P7,T7)
      XCL8=XCL(P8,T8)
      XCL9=XCL(P9,T9)
      XCL3=XCL(P3,T3)
      XL7 =MASFRAC(XCL7)
      XL8 =MASFRAC(XCL8)
      XL9 =MASFRAC(XCL9)
      XL3 =MASFRAC(XCL3)
      XV7 =MASFRAC(XCV(P7,T7,XCL7))
      XV8 =MASFRAC(XCV(P8,T8,XCL8))
      XV9 =MASFRAC(XCV(P9,T9,XCL9))
      XV3 =MASFRAC(XCV(P3,T3,XCL3))
C
C-----MOLE CONCENTRATION OF R114 IN VAP. AT 10P,2P & LIQ. AT 4
      XCV10P=XCV(P10,T10P,XCL(P10,T10P))
      XCL4 =XCL(P4,T4)
      XCV2P=XCV(P2,T2P,XCL(P2,T2P))
C
C-----ENTHALPIES OF LIQ. & VAP. AT 3,7,8,9 (2-PHASE REGION)
      HL7 =XL7*H114L(T7) + (1.-XL7)*H12L(T7)
      HL8 =XL8*H114L(T8) + (1.-XL8)*H12L(T8)
      HL9 =XL9*H114L(T9) + (1.-XL9)*H12L(T9)
      HL3 =XL3*H114L(T3) + (1.-XL3)*H12L(T3)
      HV7 =XV7*H114V(T7) + (1.-XV7)*H12V(T7)
      HV8 =XV8*H114V(T8) + (1.-XV8)*H12V(T8)
      HV9 =XV9*H114V(T9) + (1.-XV9)*H12V(T9)
      HV3 =XV3*H114V(T3) + (1.-XV3)*H12V(T3)
```

```
C
C------PARTIAL PRESSURES OF R12 & R114 IN SUPERHEATED REGION
      PP4H    = MOLFRAC(CONC)*PHI
      PP2H    = PHI - PP4H
      PP4L    = MOLFRAC(CONC)*PLO
      PP2L    = PLO - PP4L
      PP4L10  = MOLFRAC(CONC)*P10
      PP2L10  = P10 - PP4L10

C
C     SPECIFIC VOLUME OF SUPERHEATED VAPOR.
C     SOURCE: R.C. DOWNING, ASHRAE TRANSACTIONS, #2313, VOL. 80,
C     PART II, 1974.
C     MIXTURE VOLUME APPROXIMATED BY AMAGAT'S MODEL FOR IDEAL GASSES

C     EQUATIONS ARE IN ENGLISH UNITS, SO SOME CONVERSION OF UNITS
C     TAKES PLACE.   C.I. MCCARTHY 11-17-82

      PLO2PSI = PLO / 6.894757
      PLO4PSI = PLO / 6.894757
      T1R2    = T1*1.8 + 32. + 459.7
      T1R4    = T1*1.8 + 32. + 459.69
      FACTOR  = 0.062427961
      TC2     = 693.3
      TC4     = 753.95
      K2      = -5.475
      K4      = -3.0
      SB2     = 0.0065093886
      SB4     = 0.005914907
      R2      = 0.088734
      R4      = 0.062780807

      CONST2  = 0.0
      CONST4  = 0.0
```

```
      DO 50 I=2,5
      CONST2  = CONST2 + (A2(I)+B2(I)*T1R2+C2(I)*EXP(K2*T1R2/TC2))/
     C                   (PLO2PSI*(VOL2-SB2)**(I-1))
      CONST4  = CONST4 + (A4(I)+B4(I)*T1R4+C4(I)*EXP(K4*T1R4/TC4))/
     C                   (PLO4PSI*(VOL4-SB4)**(I-1))
   50 CONTINUE
C     THE REST OF THE TERMS ARE IN EQUATIONS R(44) & R(45)


C-----COMPRESSOR CHARACTERISTICS  (MCCARTHY NOV 82)
C     ISENTROPIC EFFICIENCY (ETAS) IS NOT ANALYTICAL, BUT THIS TYPE OF
C     FUNCTION IS EXPECTED AND IT HELPS MATCH TEST DATA MORE ACCURATELY.
C     FRLOAD RELATION IS BASED ON ACTUAL DYNOMOMETER RESULTS.

      DISPL  = 1.3254E-3
      ETAS   = 0.72 - RPM/13000.
      SPEED  = 2.25*RPM
      FRLOAD = 2.07E-4 * SPEED + 5.63E-8 * SPEED**2

C-----WORK OF ISENTROPIC COMPRESSION  (P.LAUNAY)
      E12  = WIR12(T1,PP2L,PP2H)
      E114 = WIR114(T1,PP4L,PP4H)
C
C-----CP'S OF LIQUIDS KJ/KG K (LTE 40% GLYCOL, HTE 30% GLYCOL)
C     THESE GLYCOL EQUATIONS WERE OBTAINED BY FITTING A LEAST SQUARES
C     CURVE TO GRAPHICAL DATA.  (MCCARTHY NOV 82)
C
      TELAVG = (TELI+TELO)/2.
      TEHAVG = (TEHI+TEHO)/2.
      CPGL   = 3.397 + .0045*TELAVG - 2.03E-5*TELAVG**2
      CPGH   = 3.703 + .00234*TELAVG + 4.05E-5*TELAVG**2
      CPWA   = 4.19
C
C-----UA VALUES (KW/K) BASED ON ACTUAL TEST RESULTS.  (MCCARTHY NOV 82)
```

```
      ACO    = FHTC * AHTC
      ADSH   = (1.0 - FHTC) * AHTC
      UHTC   = 1.22 - 0.71*CONC
      ULTC   = 1.15 - 0.78*CONC
      UDSH   = 0.5
      ULTE   = 0.17
      UHTE   = 0.327 - .324*CONC + .40*CONC**2
      UACO   = UHTC * ACO
      UADSH  = UDSH * ADSH
      AEHTE  = FHTE * AHTE
      UAHTE  = UHTE * AEHTE
      ASH    = (1.0 - FHTE) * AHTE
      UHX1   = 0.09
      UHX2   = 0.20
      USH    = 0.09
      UAHX1  = UHX1 * AHX1
      UASH   = USH * ASH
      UALTE  = ULTE * ALTE
      UALTC  = ULTC * ALTC
      UAHX2  = UHX2 * AHX2
C
CC    COMPUTING RESIDUALS

C     THESE EQUATIONS HAVE BEEN REVISED BY C. MCCARTHY TO MOVE THE
C     SUPERHEAT POINT FROM INSIDE HX1 TO INSIDE THE HIGH TEMP EVAPORATOR
C     WHICH MODELS MORE CLOSELY THE PHYSICAL APPARATUS.
C     EQUATIONS R(42) THRU R(45) ARE A PART OF THE COMPRESSOR MODEL.

C
C------FRACTION OF LIQ. PHASE AT 3,7,8,9
      R(1) = -CONC + FRL7*XL7 + (1.-FRL7)*XV7
C
      R(2) = -CONC + FRL8*XL8 + (1.-FRL8)*XV8
C
      R(3) = -CONC + FRL9*XL9 + (1.-FRL9)*XV9
C
      R(4) = -CONC + FRL3*XL3 + (1.-FRL3)*XV3
```

```
C
C-----ENTHALPIES AT 3,7,8,9,10
      R(5) = -H7 + FRL7*HL7 + (1.-FRL7)*HV7
C
      R(6) = -H8 + FRL8*HL8 + (1.-FRL8)*HV8
C
      R(7) = -H9 + FRL9*HL9 + (1.-FRL9)*HV9
C
      R(8) = -H10 + CONC*HR114SH(PP4L10,T10) +
     *       (1.-CONC)*HR12SH(PP2L10,T10)
C
      R(9) = -H3 + FRL3*HL3 + (1.-FRL3)*HV3
C
C-----ENTHALPIES AT 4,5,6
      R(10)= -H5 + CONC*H114L(T5) + (1.-CONC)*H12L(T5)
C
      R(11)= -H6 + CONC*H114L(T6) + (1.-CONC)*H12L(T6)
C
      R(12)= -H4 + CONC*H114L(T4) + (1.-CONC)*H12L(T4)
C
C-----HEAT EXCHANGER EQUATIONS
      R(13)= -QLTC + UALTC*LMTD(T3,T4,TCLI,TCLO)
C
      R(14)= -QHTC*(TCHO-TCHP)/(TCHO-TCHI) +UADSH*LMTD(T2,T2P,TCHP,TCHO)
C
      R(15)= -QHTC*(TCHP-TCHI)/(TCHO-TCHI) + UACO*LMTD(T2P,T3,TCHI,TCHP)
C
      R(16)= -QLTE + UALTE*LMTD(TELI,TELO,T7,T8)
C
      R(17)= -QHTE + UAHTE*LMTD(TEHP,TEHO,T9,T10P)
     *      + UASH*LMTD(TEHI,TEHP,T10P,T10)
C
      R(18)= -W*(H1-H10) + UAHX1*LMTD(T4,T5,T10,T1)
C
      R(19)= -AHTE + 0.3679
C
      R(20)= -(T5-T9) +
     *      (T6-T8)*EXP(UAHX2*((T5-T9)-(T6-T8))/(W*(H9-H8)))
C
```

```
C-----SATURATION CONDITIONS
      R(21)= -MOLFRAC(CONC) + XCV10P
C
      R(22)= -MOLFRAC(CONC) + XCL4
C
      R(23)= -MOLFRAC(CONC) + XCV2P
C
C-----HEAT BALANCES
      R(24)= -QLTE + W*(H8-H7)
C
      R(25)= -H6 + H7
C
      R(26)= -QHTC + W*(H2-H3)
C
      R(27)= -QLTC + W*(H3-H4)
C
      R(28)= -QLTC - QHTC + (ETAS*POWR - FRLOAD) + QLTE + QHTE
C
      R(29)= -(H1-H10) + (H4-H5)
C
      R(30)= -QHTE + W*(H10-H9)
C
      R(31)= -(H5-H6) + (H9-H8)
C
C-----MISCELLANEOUS
      R(32)= -H2 + CONC*HR114SH(PP4H,T2) + (1.-CONC)*HR12SH(PP2H,T2)
C
      R(33)= -POWR + W*(CONC*E114 + (1.-CONC)*E12)/ETAS + FRLOAD
C
      R(34)= -H1 + CONC*HR114SH(PP4L,T1) + (1.-CONC)*HR12SH(PP2L,T1)
C
      R(35)= -W*(H10-H9)*(TEHP-TEHO)/(TEHI-TEHO) +
     *       W*(CONC*H114V(T10P) + (1.-CONC)*H12V(T10P) - H9)
      R(36)= -QHTC*(TCHO-TCHP)/(TCHO-TCHI) +
     *       W*(H2-CONC*H114V(T2P) - (1.-CONC)*H12V(T2P))
      R(37)= -QLTE + WGEL*CPGL*(TELI-TELO)
```

```
C
      R(38)= -QHTE + WGEH*CPGH*(TEHI-TEHO)
C
      R(39)= -SPHX + (T1 - T10F)
C
      R(40)= -QLTC + WLTC*CPWA*(TCLO-TCLI)
C
      R(41)= -QHTC + WHTC*CPWA*(TCHO-TCHI)
C
      R(42)= -RPM + (MOLFRAC(CONC)*VOL4+(1.-MOLFRAC(CONC))*VOL2)
     *       * FACTOR * W * 60.0 / (DISPL*ETAV)
C
C     THE VOLUMETRIC EFFICIENCY IS BASED ON TEST DATA.  (MCCARTHY NOV 82)
C
      R(43)= -ETAV + 0.72 - 3.5E-4*SPEED + 1.1E-7*SPEED**2
C
      R(44)= -VOL2 + CONST2 + SB2 + R2*T1R2/PLO2PSI
C
      R(45)= -VOL4 + CONST4 + SB4 + R4*T1R4/PLO4PSI
C
      RETURN
      END


      SUBROUTINE SIMUL(NVAR, TLRNCE, ITMAX, V, DES, R, PD, VCORR,VD,RD,
     *  DAMPING,POPTION,ITER)
C
C    THIS IS THE NEWTON-RAPHSON SUBROUTINE WHICH IS COMBINED WITH
C       THE PARDIF AND GAUSSY SUBROUTINES
C
C     USER MUST PROVIDE THE MAIN PROGRAM AND THE EQUATION SUBROUTINE
C
C     GLOSSARY OF TERMS USED
C
```

```
C     ITER      = NUMBER OF ITERATIONS
C     ITMAX     = MAXIMUM NUMBER OF ITERATIONS TO BE PERMITTED
C     NVAR      = NUMBER OF UNKNOWNS = NUMBER OF EQUATIONS
C     PD(I,J)   = PARTIAL DERIVATIVE OF FUNCTION I WITH RESP TO VARIABLE J
C     R( )      = RESIDUAL OF EQUATION
C     TLRNCE    = MAXIMUM FRACTION OF VALUE OF VARIABLES PERMITTED BEFORE
C                 ITERATION COMPLETE.  THUS TLRNCE = 0.01 REQUIRES CHANGE OF
C                 ALL VARIABLES TO BE LESS THAN 1 PERCENT FOR CONVERGENCE
C     V( )      = VALUE OF THE VARIABLE
C     VCORR( )  = CORRECTION IN THE VARIABLE DURING THIS ITERATION
C     DES( )    = DESIGNATION OF VARIABLE IN A4 FIELDS
C     DAMPING     IS THE FACTOR BY WHICH ITERATIONS ARE DAMPED IN STATEMENT
C                 NUMBER 43  (SET DAMPING=0 IF NO DAMPING IS DESIRED)
C     POPTION     IS A NUMBER REPRESENTING THE FOLLOWING PRINTING OPTIONS
C                 CORRESPONDING TO THE VALUE OF POPTION:
C             0 = NO PRINTING DESIRED FROM THIS SUBROUTINE
C             1 = PRINT LAST ITERATION ONLY
C             2 = PRINT ALL ITERATIONS
C             3 = PRINT PARTIAL DERIVATIVES ONLY
C             4 = PRINT LAST ITERATION PLUS PARTIALS
C             5 = PRINT ALL ITERATIONS PLUS PARTIALS
C

      DIMENSION VD(NVAR), RD(NVAR)
      DIMENSION V(NVAR),R(NVAR),PD(NVAR,NVAR),VCORR(NVAR), DES(NVAR)
      INTEGER DAMPING,POPTION

C
C   WRITING OUT THE INPUT DATA
C
      IF (POPTION.EQ.0) GO TO 24
      WRITE(6,20) NVAR
   20 FORMAT(1H1,///," NUMBER OF VARIABLES = ", I4)
      WRITE(6,21) TLRNCE
   21 FORMAT(" MAXIMUM FRACTION CHANGE FOR CONVERGENCE =", F10.4, //)
      WRITE(6,22)
   22 FORMAT("0VARIABLE NUMBER AND ITS TRIAL VALUE")
      WRITE (6,23)  (J, DES(J), V(J), J = 1, NVAR)
   23 FORMAT (" V(", I2, ") = ", A4, " = ", F15.5)
C
```

```
C    INITIALIZING THE ITERATION COUNTER
   24 ITER = 1
C    CALLING SUBROUTINES TO CALCULATE VALUES OF RESIDUALS, PARTIAL
C         DERIVATIVES AND CHANGES IN VALUES OF VARIABLES
C
   30 CALL EQNS(NVAR, V, R)
      IF (POPTION.NE.2.AND.POPTION.NE.5) GO TO 25
      WRITE (6,33)
   33 FORMAT("0EQUATION NUMBER       RESIDUAL")
      WRITE (6,35)  (I , R(I),  I = 1,NVAR)
   35 FORMAT (I10, F20.5)
   25 CALL PARDIF(NVAR, V, R, PD, VD, RD)
C
C     PRINTING OUT NON-ZERO VALUES OF PARTIAL DERIVATIVES
C
      IF (POPTION.LT.3) GO TO 55
      DO 54  I = 1, NVAR
      DO 53  J = 1, NVAR
      Z = ABS(PD(I,J))
      IF(Z - 0.00000001) 53, 53, 51
   51 WRITE(6,52) I, J, PD(I,J)
   52 FORMAT(" PD(",I2,",",I2,") =", F20.10)
   53 CONTINUE
   54 CONTINUE
   55 CALL GAUSSY(PD, R, VCORR, NVAR)
C     CORRECTING THE VALUES OF THE VARIABLES
      DO 44  L = 1,NVAR
      IF (DAMPING.EQ.0) GO TO 44
   43 VCORR(L) = VCORR(L)/(1.0+FLOAT(DAMPING-1)/(FLOAT(ITER)**2))
   44 V(L) = V(L) - VCORR(L)
C
C    WRITING OUT RESULTS OF THIS ITERATION
C
```

```
      IF (POPTION.NE.2.AND.POPTION.NE.5) GO TO 37
      WRITE (6,32) ITER
   32 FORMAT('ORESULTS AFTER', I4, '  ITERATIONS')
      WRITE(6,34)
   34 FORMAT('0',4X,'VARIABLE',16X,'VALUE',9X,'CHANGE FROM PREVIOUS')
      WRITE (6,36)  (I, DES(I), V(I), VCORR(I), I = 1,NVAR)
   36 FORMAT (' V(', I2, ') = ', A4,' = ', 2F21.5)
C
C   TERMINATING IF MAXIMUM NUMBER OF ITERATIONS REACHED OR OTHERWISE
C         INCREMENTING THE ITERATION COUNTER
   37 IF(ITER - ITMAX) 38, 99, 99
   38 ITER = ITER + 1
C
C   CHECK TO SEE IF CHANGE OF VARIABLE IS LESS THAN SPECIFIED TOLERANCE
C
      K = 1
   40 VAL = ABS(VCORR(K)) - ABS(TLRNCE*V(K))
      IF(VAL) 41, 30, 30
   41 IF(K - NVAR) 42, 99, 99
   42 K = K + 1
      GO TO 40
   99 IF (POPTION.NE.1.AND.POPTION.NE.4) RETURN
      WRITE (6,32) ITER
      WRITE (6,34)
      WRITE (6,36)  (I,DES(I),V(I),VCORR(I),I=1,NVAR)
      RETURN
      END


      SUBROUTINE PARDIF (NVAR, V, R, PD, VD, RD)
      DIMENSION VD(NVAR), RD(NVAR)
      DIMENSION V(NVAR), R(NVAR), PD(NVAR,NVAR)
C
C   GLOSSARY FOR SUBROUTINE PARDIF
C     DV = FRACTION OF VARIABLE CHANGE USED IN TAKING PARTIAL
C     VD =  V + DELTA V  =  V + V*DV
C     RD = R EVALUATED AT VD
C     THE PARTIAL DERIVATIVE IS  (RD - R)/(V*DV)
C
```

```
      DV = 0.001
C     SETTING ALL VD = V
      DO 550  K = 1,NVAR
  550 VD(K) = V(K)
      DO 560  J = 1, NVAR
C     ADDING DELTA TO VD(J)
C    CIRCUMVENT CASE OF V(J) = 0
      IF(ABS(V(J)) - 10.**(-30)) 551, 551, 552
  551 VD(J) = V(J) + 0.001
      GO TO 553
  552 VD(J) =(1. + DV)*V(J)
  553 CALL EQNS(NVAR, VD, RD)
      DO 558 I = 1, NVAR
      IF(ABS(V(J)) - 10.**(-30)) 555, 555, 556
  555 PD(I,J) = (RD(I) - R(I))/0.001
      GO TO 558
  556 PD(I,J) = (RD(I) - R(I))/(V(J)*DV)
  558 CONTINUE
C     RETURNING VD(J) TO V(J) VALUE
      VD(J) = V(J)
  560 CONTINUE
      RETURN
      END


      SUBROUTINE GAUSSY (A, B, X, N)
C
C     SOLUTION OF SIMULTANEOUS EQUATIONS BY GAUSS ELIMINATION
C
      DIMENSION A(N,N), X(N), B(N)
C     BEGINNING OF ELIMINATION PROCESS
      DO 28  K = 1,N
C     MOVING LARGEST COEFFICIENT INTO DIAGONAL POSITION
      AMAX = 0.
      DO 4  I = K,N
      IF(ABS(A(I,K)) - ABS(AMAX)) 4, 4, 2
```

```
    2 AMAX = A(I,K)
      IMAX = I
    4 CONTINUE
C     TESTING FOR INDEPENDENCE OF EQUATIONS
      IF(ABS(AMAX) - 0.1E-15) 10, 10, 14
   10 WRITE (6,12)
   12 FORMAT ('0 EQUATIONS ARE NOT INDEPENDENT')
      RETURN
C      EXCHANGING ROW IMAX AND ROW K
   14 BTEMP = B(K)
      B(K) = B(IMAX)
      B(IMAX) = BTEMP
      DO 18  J = K,N
      ATEMP = A(K,J)
      A(K,J) = A(IMAX, J)
   18 A(IMAX,J) = ATEMP
C     SUBTRACTING A(I,K)/A(K,K) TIMES TERM IN FIRST EQ FROM OTHERS
      KPLUS = K + 1
      IF(K - N) 22, 28, 28
   22 DO 24  I = KPLUS,N
      B(I) = B(I) - B(K)*A(I,K)/A(K,K)
      ACON = A(I,K)
      DO 24  J = K,N
   24 A(I,J) = A(I,J) - A(K,J)*ACON/A(K,K)
   28 CONTINUE
C      BACK SUBSTITUTION
      L = N
   32 SUM = 0.0
      IF(L - N) 34, 38, 38
   34 LPLUS = L + 1
      DO 36  J = LPLUS, N
   36 SUM = SUM + A(L,J)*X(J)
   38 CONTINUE
      X(L) = (B(L) - SUM)/A(L,L)
      IF(L - 1) 42, 42, 40
   40 L = L - 1
```

```
      GO TO 32
   42 RETURN
      END

C     THIS FUNCTION SAFELY CALCULATES LOG MEAN TEMPERATURE DIFFERENCE
C     MCCARTHY NOV 82.

      REAL FUNCTION LMTD(T1I,T1O,T2I,T2O)
      DT1  = T1I - T2O
      DT2  = T1O - T2I
      IF(ABS(DT1-DT2).LE.0.1) GO TO 2
      IF((DT1/DT2).LE.0.0) GO TO 2
      LMTD = (DT1-DT2)/ALOG(DT1/DT2)
      RETURN
    2 LMTD = 0.5*(DT1+DT2)
      RETURN
      END
```

APPENDIX E

Simulation Results for R-114 Concentrations from 0 to 0.6

AFTER 5 ITERATIONS SOLUTION IS:
****

DAT(1)	= CONC	=	.0001	
DAT(2)	= ALTE	=	.5513	M**2
DAT(3)	= AHX1	=	.0927	M**2
DAT(4)	= AHX2	=	.0927	M**2
DAT(5)	= ALTC	=	.3739	M**2
DAT(6)	= AHTC	=	.3739	M**2
DAT(7)	= WGEL	=	.0530	KG/S
DAT(8)	= WGEH	=	.0600	KG/S
DAT(9)	= WLTC	=	.0910	KG/S
DAT(10)	= WHTC	=	.0910	KG/S
DAT(11)	= OLTE	=	1.0300	KW
DAT(12)	= OHTE	=	1.3800	KW
DAT(13)	= TCLI	=	34.3000	DEG C
DAT(14)	= TCHI	=	34.3000	DEG C
DAT(15)	= TELI	=	-8.0000	DEG C
DAT(16)	= TEHI	=	-.1000	DEG C
DAT(17)	= SPHX	=	24.0000	DEG C

V(1)	= W	=	.02098	KG/S
V(2)	= PHI	=	1039.32535	KPA
V(3)	= PLO	=	127.91531	KPA
V(4)	= T1	=	1.10119	DEG C
V(5)	= T2	=	79.84129	DEG C
V(6)	= T2P	=	42.47086	DEG C
V(7)	= T3	=	42.43105	DEG C
V(8)	= T4	=	42.31383	DEG C
V(9)	= T5	=	23.41809	DEG C
V(10)	= T6	=	-4.38627	DEG C
V(11)	= T7	=	-22.05538	DEG C
V(12)	= T8	=	-22.25999	DEG C
V(13)	= T9	=	-22.35322	DEG C
V(14)	= T10	=	-30.44587	DEG C
V(15)	= T10P	=	-22.89881	DEG C
V(16)	= H1	=	355.80808	KJ/KG
V(17)	= H2	=	396.00751	KJ/KG
V(18)	= H3	=	340.40131	KJ/KG
V(19)	= H4	=	240.95862	KJ/KG
V(20)	= H5	=	222.17001	KJ/KG
V(21)	= H6	=	195.95743	KJ/KG
V(22)	= H7	=	195.95743	KJ/KG
V(23)	= H8	=	245.04247	KJ/KG
V(24)	= H9	=	271.25505	KJ/KG
V(25)	= H10	=	337.01947	KJ/KG
V(26)	= FRL7	=	.90100	
V(27)	= FRL8	=	.59673	
V(28)	= FRL9	=	.43445	
V(29)	= FRL3	=	.21734	
V(30)	= FHTC	=	.69760	
V(31)	= FHTE	=	.34371	
V(32)	= OLTC	=	2.08670	KW
V(33)	= OHTC	=	1.16684	KW
V(34)	= TCHP	=	41.29142	DEG C
V(35)	= AHTE	=	.36790	M**2
V(36)	= TELC	=	-13.80896	DEG C
V(37)	= TEHP	=	.31232	DEG C
V(38)	= TEHO	=	-6.34610	DEG C
V(39)	= TCLO	=	39.77275	DEG C
V(40)	= TCHO	=	42.83299	DEG C
V(41)	= POWR	=	1.38518	KW
V(42)	= RPM	=	240.02372	RPM
V(43)	= ETAV	=	.56306	
V(44)	= VOL2	=	2.27903	CF/LB
V(45)	= VOL4	=	1.58298	CF/LB

AFTER 5 ITERATIONS SOLUTION IS:
****

DAT(1)	= CONC	=	.1000	
DAT(2)	= ALTE	=	.5518	M**2
DAT(3)	= AHX1	=	.0927	M**2
DAT(4)	= AHX2	=	.0927	M**2
DAT(5)	= ALTC	=	.3739	M**2
DAT(6)	= AHTC	=	.3739	M**2
DAT(7)	= WGEL	=	.0530	KG/S
DAT(8)	= WGEH	=	.0600	KG/S
DAT(9)	= WLTC	=	.0910	KG/S
DAT(10)	= WHTC	=	.0910	KG/S
DAT(11)	= OLTE	=	1.0300	KW
DAT(12)	= OHTE	=	1.3800	KW
DAT(13)	= TCLI	=	34.3000	DEG C
DAT(14)	= TCHI	=	34.3000	DEG C
DAT(15)	= TELI	=	-8.0000	DEG C
DAT(16)	= TEHI	=	-.1000	DEG C
DAT(17)	= SPHX	=	24.0000	DEG C

V(1)	= W	=	.02059	KG/S
V(2)	= PHI	=	947.60405	KPA
V(3)	= PLO	=	117.34771	KPA
V(4)	= T1	=	4.22644	DEG C
V(5)	= T2	=	77.38789	DEG C
V(6)	= T2P	=	43.60270	DEG C
V(7)	= T3	=	42.05132	DEG C
V(8)	= T4	=	40.53678	DEG C
V(9)	= T5	=	23.62445	DEG C
V(10)	= T6	=	-4.28976	DEG C
V(11)	= T7	=	-22.25457	DEG C
V(12)	= T8	=	-21.97708	DEG C
V(13)	= T9	=	-21.62593	DEG C
V(14)	= T10	=	-23.76438	DEG C
V(15)	= T10P	=	-19.77356	DEG C
V(16)	= H1	=	356.37182	KJ/KG
V(17)	= H2	=	395.89013	KJ/KG
V(18)	= H3	=	326.18267	KJ/KG
V(19)	= H4	=	239.31603	KJ/KG
V(20)	= H5	=	222.45696	KJ/KG
V(21)	= H6	=	196.03412	KJ/KG
V(22)	= H7	=	196.03412	KJ/KG
V(23)	= H8	=	246.06211	KJ/KG
V(24)	= H9	=	272.48495	KJ/KG
V(25)	= H10	=	339.51275	KJ/KG
V(26)	= FRL7	=	.89888	
V(27)	= FRL8	=	.58969	
V(28)	= FRL9	=	.42653	
V(29)	= FRL3	=	.32688	
V(30)	= FHTC	=	.74406	
V(31)	= FHTE	=	.55225	
V(32)	= OLTC	=	1.78845	KW
V(33)	= OHTC	=	1.43517	KW
V(34)	= TCHP	=	41.24777	DEG C
V(35)	= AHTE	=	.36790	M**2
V(36)	= TELO	=	-13.80896	DEG C
V(37)	= TEHP	=	.03962	DEG C
V(38)	= TEHO	=	-6.34610	DEG C
V(39)	= TCLO	=	38.99053	DEG C
V(40)	= TCHO	=	42.75451	DEG C
V(41)	= POWR	=	1.36243	KW
V(42)	= RPM	=	259.60889	RPM
V(43)	= ETAV	=	.55309	
V(44)	= VOL2	=	2.52369	CF/LB
V(45)	= VOL4	=	1.75662	CF/LB

```
AFTER  5 ITERATIONS SOLUTION IS:
********************************
DAT( 1)  = CONC  =      .2000
DAT( 2)  = ALTE  =      .5518    M**2
DAT( 3)  = AHX1  =      .0927    M**2
DAT( 4)  = AHX2  =      .0927    M**2
DAT( 5)  = ALTC  =      .3739    M**2
DAT( 6)  = AHTC  =      .3739    M**2
DAT( 7)  = WGEL  =      .0530    KG/S
DAT( 8)  = WGEH  =      .0600    KG/S
DAT( 9)  = WLTC  =      .0910    KG/S
DAT(10)  = WHTC  =      .0910    KG/S
DAT(11)  = OLTE  =     1.0300    KW
DAT(12)  = OHTE  =     1.3800    KW
DAT(13)  = TCLI  =    34.3000    DEG C
DAT(14)  = TCHI  =    34.3000    DEG C
DAT(15)  = TELI  =    -8.0000    DEG C
DAT(16)  = TEHI  =    -.1000     DEG C
DAT(17)  = SPHX  =    24.0000    DEG C

V( 1)  = W     =        .02045    KG/S
V( 2)  = PHI   =     872.98916    KPA
V( 3)  = PLC   =     106.13258    KPA
V( 4)  = T1    =       6.28544    DEG C
V( 5)  = T2    =      75.13415    DEG C
V( 6)  = T2P   =      44.81681    DEG C
V( 7)  = T3    =      41.82175    DEG C
V( 8)  = T4    =      39.42032    DEG C
V( 9)  = T5    =      23.97212    DEG C
V(10)  = T6    =      -3.89289    DEG C
V(11)  = T7    =     -22.49366    DEG C
V(12)  = T8    =     -21.64820    DEG C
V(13)  = T9    =     -20.82068    DEG C
V(14)  = T10   =     -19.06969    DEG C
V(15)  = T10P  =     -17.71456    DEG C
V(16)  = H1    =     356.21523    KJ/KG
V(17)  = H2    =     395.16539    KJ/KG
V(18)  = H3    =     314.89313    KJ/KG
V(19)  = H4    =     238.33900    KJ/KG
V(20)  = H5    =     222.88443    KJ/KG
V(21)  = H6    =     196.38887    KJ/KG
V(22)  = H7    =     196.38887    KJ/KG
V(23)  = H8    =     246.76750    KJ/KG
V(24)  = H9    =     273.26307    KJ/KG
V(25)  = H10   =     340.76066    KJ/KG
V(26)  = FRL7  =        .89475
V(27)  = FRL8  =        .58434
V(28)  = FRL9  =        .42096
V(29)  = FRL3  =        .41433
V(30)  = FHTC  =        .77319
V(31)  = FHTE  =        .75282
V(32)  = OLTC  =       1.56516    KW
V(33)  = OHTC  =       1.64118    KW
V(34)  = TCHP  =      41.25605    DEG C
V(35)  = AHTE  =        .36790    M**2
V(36)  = TELO  =     -13.80896    DEG C
V(37)  = TEHP  =      -.14795     DEG C
V(38)  = TEHC  =      -6.34610    DEG C
V(39)  = TCLO  =      38.40491    DEG C
V(40)  = TCHO  =      42.70920    DEG C
V(41)  = POWR  =       1.36799    KW
V(42)  = RPM   =     288.71561    RPM
V(43)  = ETAV  =        .53906
V(44)  = VOL2  =       2.82191    CF/LB
V(45)  = VOL4  =       1.96802    CF/LB
```

AFTER 5 ITERATIONS SOLUTION IS:
****

DAT(1)	=	CONC =	.3000	
DAT(2)	=	ALTE =	.5518	M**2
DAT(3)	=	AHX1 =	.0927	M**2
DAT(4)	=	AHX2 =	.0927	M**2
DAT(5)	=	ALTC =	.3739	M**2
DAT(6)	=	AHTC =	.3739	M**2
DAT(7)	=	WGEL =	.0530	KG/S
DAT(8)	=	WGEH =	.0600	KG/S
DAT(9)	=	WLTC =	.0910	KG/S
DAT(10)	=	WHTC =	.0910	KG/S
DAT(11)	=	OLTE =	1.0300	KW
DAT(12)	=	OHTE =	1.3800	KW
DAT(13)	=	TCLI =	34.3000	DEG C
DAT(14)	=	TCHI =	34.3000	DEG C
DAT(15)	=	TELI =	-8.0000	DEG C
DAT(16)	=	TEHI =	-.1000	DEG C
DAT(17)	=	SPHX =	24.0000	DEG C

V(1)	=	W =	.02048	KG/S
V(2)	=	PHI =	807.72411	KPA
V(3)	=	PLO =	94.33161	KPA
V(4)	=	T1 =	7.48707	DEG C
V(5)	=	T2 =	72.90672	DEG C
V(6)	=	T2P =	45.93973	DEG C
V(7)	=	T3 =	41.81572	DEG C
V(8)	=	T4 =	38.75144	DEG C
V(9)	=	T5 =	24.31538	DEG C
V(10)	=	T6 =	-3.33733	DEG C
V(11)	=	T7 =	-22.76004	DEG C
V(12)	=	T8 =	-21.29479	DEG C
V(13)	=	T9 =	-20.02612	DEG C
V(14)	=	T10 =	-16.01792	DEG C
V(15)	=	T10P =	-16.51293	DEG C
V(16)	=	H1 =	355.46150	KJ/KG
V(17)	=	H2 =	393.88454	KJ/KG
V(18)	=	H3 =	306.75365	KJ/KG
V(19)	=	H4 =	237.81268	KJ/KG
V(20)	=	H5 =	223.31085	KJ/KG
V(21)	=	H6 =	196.89323	KJ/KG
V(22)	=	H7 =	196.89323	KJ/KG
V(23)	=	H8 =	247.17468	KJ/KG
V(24)	=	H9 =	273.59230	KJ/KG
V(25)	=	H10 =	340.95967	KJ/KG
V(26)	=	FRL7 =	.88942	
V(27)	=	FRL8 =	.58042	
V(28)	=	FRL9 =	.41733	
V(29)	=	FRL3 =	.47827	
V(30)	=	FHTC =	.79355	
V(31)	=	FHTE =	.91434	
V(32)	=	OLTC =	1.41223	KW
V(33)	=	OHTC =	1.78485	KW
V(34)	=	TCHP =	41.30339	DEG C
V(35)	=	AHTE =	.36790	M**2
V(36)	=	TELO =	-13.80896	DEG C
V(37)	=	TEHP =	-.27118	DEG C
V(38)	=	TEHO =	-6.34610	DEG C
V(39)	=	TCLO =	38.00383	DEG C
V(40)	=	TCHO =	42.68491	DEG C
V(41)	=	POWR =	1.40023	KW
V(42)	=	RPM =	331.04726	RPM
V(43)	=	ETAV =	.52033	
V(44)	=	VOL2 =	3.19977	CF/LB
V(45)	=	VOL4 =	2.23568	CF/LB

AFTER 5 ITERATIONS SOLUTION IS:
****

DAT(1)	= CONC	=	.4000	
DAT(2)	= ALTE	=	.5518	M**2
DAT(3)	= AHX1	=	.0927	M**2
DAT(4)	= AHX2	=	.0927	M**2
DAT(5)	= ALTC	=	.3739	M**2
DAT(6)	= AHTC	=	.3739	M**2
DAT(7)	= WGEL	=	.0530	KG/S
DAT(8)	= WGEH	=	.0600	KG/S
DAT(9)	= WLTC	=	.0910	KG/S
DAT(10)	= WHTC	=	.0910	KG/S
DAT(11)	= OLTE	=	1.0300	KW
DAT(12)	= OHTE	=	1.3800	KW
DAT(13)	= TCLI	=	34.3000	DEG C
DAT(14)	= TCHI	=	34.3000	DEG C
DAT(15)	= TELI	=	-8.0000	DEG C
DAT(16)	= TEHI	=	-.1000	DEG C
DAT(17)	= SPHX	=	24.0000	DEG C

V(1)	= W	=	.02068	KG/S
V(2)	= PHI	=	747.73109	KPA
V(3)	= PLO	=	82.05986	KPA
V(4)	= T1	=	7.91884	DEG C
V(5)	= T2	=	70.51554	DEG C
V(6)	= T2P	=	46.88345	DEG C
V(7)	= T3	=	42.06497	DEG C
V(8)	= T4	=	38.43121	DEG C
V(9)	= T5	=	24.60086	DEG C
V(10)	= T6	=	-2.70563	DEG C
V(11)	= T7	=	-23.02813	DEG C
V(12)	= T8	=	-20.95216	DEG C
V(13)	= T9	=	-19.35357	DEG C
V(14)	= T10	=	-14.43288	DEG C
V(15)	= T10P	=	-16.08116	DEG C
V(16)	= H1	=	354.16037	KJ/KG
V(17)	= H2	=	392.00423	KJ/KG
V(18)	= H3	=	301.49995	KJ/KG
V(19)	= H4	=	237.63936	KJ/KG
V(20)	= H5	=	223.68324	KJ/KG
V(21)	= H6	=	197.47208	KJ/KG
V(22)	= H7	=	197.47208	KJ/KG
V(23)	= H8	=	247.27151	KJ/KG
V(24)	= H9	=	273.48268	KJ/KG
V(25)	= H10	=	340.20426	KJ/KG
V(26)	= FRL7	=	.88344	
V(27)	= FRL8	=	.57775	
V(28)	= FRL9	=	.41515	
V(29)	= FRL3	=	.52098	
V(30)	= FHTC	=	.80925	
V(31)	= FHTE	=	1.00212	
V(32)	= OLTC	=	1.32083	KW
V(33)	= OHTC	=	1.87190	KW
V(34)	= TCHP	=	41.38519	DEG C
V(35)	= AHTE	=	.36790	M**2
V(36)	= TELC	=	-13.80896	DEG C
V(37)	= TEHP	=	-.34119	DEG C
V(38)	= TEHO	=	-6.34610	DEG C
V(39)	= TCLO	=	37.76410	DEG C
V(40)	= TCHO	=	42.67348	DEG C
V(41)	= POWR	=	1.46437	KW
V(42)	= RPM	=	393.36971	RPM
V(43)	= FTAV	=	.49639	
V(44)	= VOL2	=	3.69606	CF/LB
V(45)	= VOL4	=	2.58705	CF/LB

AFTER 4 ITERATIONS SOLUTION IS:
****

DAT(1)	=	CONC =	.5000	
DAT(2)	=	ALTE =	.5518	M**2
DAT(3)	=	AHX1 =	.0927	M**2
DAT(4)	=	AHX2 =	.0927	M**2
DAT(5)	=	ALTC =	.3739	M**2
DAT(6)	=	AHTC =	.3739	M**2
DAT(7)	=	WGEL =	.0530	KG/S
DAT(8)	=	WGEH =	.0600	KG/S
DAT(9)	=	WLTC =	.0910	KG/S
DAT(10)	=	WHTC =	.0910	KG/S
DAT(11)	=	OLTE =	1.0300	KW
DAT(12)	=	OHTE =	1.3800	KW
DAT(13)	=	TCLI =	34.3000	DEG C
DAT(14)	=	TCHI =	34.3000	DEG C
DAT(15)	=	TELI =	-8.0000	DEG C
DAT(16)	=	TEHI =	-.1000	DEG C
DAT(17)	=	SPHX =	24.0000	DEG C

V(1)	=	W =	.02104	KG/S
V(2)	=	PHI =	690.91124	KPA
V(3)	=	PLO =	69.49880	KPA
V(4)	=	T1 =	7.60525	DEG C
V(5)	=	T2 =	67.70846	DEG C
V(6)	=	T2P =	47.60430	DEG C
V(7)	=	T3 =	42.59200	DEG C
V(8)	=	T4 =	38.43766	DEG C
V(9)	=	T5 =	24.83574	DEG C
V(10)	=	T6 =	-2.04426	DEG C
V(11)	=	T7 =	-23.25324	DEG C
V(12)	=	T8 =	-20.67404	DEG C
V(13)	=	T9 =	-18.91923	DEG C
V(14)	=	T10 =	-14.23486	DEG C
V(15)	=	T10P =	-16.39475	DEG C
V(16)	=	H1 =	352.32448	KJ/KG
V(17)	=	H2 =	389.39668	KJ/KG
V(18)	=	H3 =	298.82806	KJ/KG
V(19)	=	H4 =	237.80049	KJ/KG
V(20)	=	H5 =	224.00793	KJ/KG
V(21)	=	H6 =	198.08288	KJ/KG
V(22)	=	H7 =	198.08288	KJ/KG
V(23)	=	H8 =	247.02882	KJ/KG
V(24)	=	H9 =	272.95387	KJ/KG
V(25)	=	H10 =	338.53192	KJ/KG
V(26)	=	FRL7 =	.87725	
V(27)	=	FRL8 =	.57620	
V(28)	=	FRL9 =	.41384	
V(29)	=	FRL3 =	.54494	
V(30)	=	FHTC =	.82287	
V(31)	=	FHTE =	.99153	
V(32)	=	OLTC =	1.28424	KW
V(33)	=	OHTC =	1.90589	KW
V(34)	=	TCHP =	41.50161	DEG C
V(35)	=	AHTE =	.36790	M**2
V(36)	=	TELO =	-13.80896	DEG C
V(37)	=	TEHP =	-.36401	DEG C
V(38)	=	TEHO =	-6.34610	DEG C
V(39)	=	TCLO =	37.66815	DEG C
V(40)	=	TCHO =	42.66669	DEG C
V(41)	=	POWR =	1.57314	KW
V(42)	=	RPM =	486.04739	RPM
V(43)	=	ETAV =	.46879	
V(44)	=	VOL2 =	4.37286	CF/LB
V(45)	=	VOL4 =	3.06605	CF/LB

AFTER 5 ITERATIONS SOLUTION IS:
****

DAT(1)	= CONC	=	.6000	
DAT(2)	= ALTE	=	.5518	M**2
DAT(3)	= AHX1	=	.0927	M**2
DAT(4)	= AHX2	=	.0927	M**2
DAT(5)	= ALTC	=	.3739	M**2
DAT(6)	= AHTC	=	.3739	M**2
DAT(7)	= WCEL	=	.0530	KG/S
DAT(8)	= WGEH	=	.0600	KG/S
DAT(9)	= WLTC	=	.0910	KG/S
DAT(10)	= WHTC	=	.0910	KG/S
DAT(11)	= OLTE	=	1.0300	KW
DAT(12)	= OHTE	=	1.3800	KW
DAT(13)	= TCLI	=	34.3000	DEG C
DAT(14)	= TCHI	=	34.3000	DEG C
DAT(15)	= TELI	=	-8.0000	DEG C
DAT(16)	= TEHI	=	-.1000	DEG C
DAT(17)	= SPHX	=	24.0000	DEG C

V(1)	= W	=	.02160	KG/S
V(2)	= PHI	=	636.46763	KPA
V(3)	= PLO	=	56.87837	KPA
V(4)	= T1	=	6.52703	DEG C
V(5)	= T2	=	64.20992	DEG C
V(6)	= T2P	=	48.09509	DEG C
V(7)	= T3	=	43.41148	DEG C
V(8)	= T4	=	38.82224	DEG C
V(9)	= T5	=	25.08894	DEG C
V(10)	= T6	=	-1.35784	DEG C
V(11)	= T7	=	-23.36880	DEG C
V(12)	= T8	=	-20.53452	DEG C
V(13)	= T9	=	-18.82980	DEG C
V(14)	= T10	=	-15.41498	DEG C
V(15)	= T10P	=	-17.47297	DEG C
V(16)	= H1	=	349.94085	KJ/KG
V(17)	= H2	=	385.89293	KJ/KG
V(18)	= H3	=	298.46783	KJ/KG
V(19)	= H4	=	238.35425	KJ/KG
V(20)	= H5	=	224.35314	KJ/KG
V(21)	= H6	=	198.72179	KJ/KG
V(22)	= H7	=	198.72179	KJ/KG
V(23)	= H8	=	246.41233	KJ/KG
V(24)	= H9	=	272.04367	KJ/KG
V(25)	= H10	=	335.93974	KJ/KG
V(26)	= FRL7	=	.87110	
V(27)	= FRL8	=	.57560	
V(28)	= FRL9	=	.41272	
V(29)	= FRL3	=	.55223	
V(30)	= FHTC	=	.83701	
V(31)	= FHTE	=	.88061	
V(32)	= OLTC	=	1.29831	KW
V(33)	= OHTC	=	1.88817	KW
V(34)	= TCHP	=	41.65744	DEG C
V(35)	= AHTE	=	.36790	M**2
V(36)	= TELO	=	-13.80896	DEG C
V(37)	= TEHP	=	-.34225	DEG C
V(38)	= TEHO	=	-6.34610	DEG C
V(39)	= TCLO	=	37.70504	DEG C
V(40)	= TCHO	=	42.65710	DEG C
V(41)	= POWR	=	1.74560	KW
V(42)	= RPM	=	618.71425	RPM
V(43)	= ETAV	=	.44594	
V(44)	= VOL2	=	5.33858	CF/LB
V(45)	= VOL4	=	3.74940	CF/LB