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T. M. Doerr
Iowa State University

M. B. Pate
Iowa State University

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Thomas M. Doerr and Michael B. Pate, Iowa State University, Ames, Iowa

ABSTRACT

Heat transfer coefficient data was taken during in-tube evaporation of five differenct refrigerant mixtures. These measurements were performed in a 3/8" O.D. (9.7 mm) smooth copper tube at refrigerant temperatures of -2°C and 4°C. Four different mass fluxes ranging from 125 to 375 kg/s·m^2 were tested. On a mass basis, the five different mixtures tested were: R-125 (40%) / R-32 (60%), R-32 (30%) / R-125 (10%) / R-134a (60%), R-134a (90%) / R-32 (10%), R-125 (44%) / R-143a (52%) / R-134a (4%), and R-134a (75%) / R-32 (25%). The evaporative heat transfer coefficient for the mixtures were also compared to data for R-22.

The mixture R-32(60%) / R-125(40%) exhibited the highest heat transfer coefficients out of all the refrigerants tested at both 4°C and -2°C when compared on both an equal mass flux and an equal cooling capacity basis. The R-32/ R-125 mixture also had a lower pressure drop than R-22. The mixture R-134a(90%) / R-32(10%) had the lowest heat transfer coefficients at both 4°C and -2°C when compared on both an equal mass flux and an equal cooling capacity basis. In addition, this same mixture had the highest pressure drop. Except for the lowest performing refrigerant, all of the refrigerants performed either similar or better than R-22. An evaluation of temperature effects showed that the refrigerants had larger heat transfer coefficients at 4°C than at -2°C.

INTRODUCTION

Non-azeotropic refrigerant mixtures are being considered as potential replacements for R-22. In this study five different refrigerant mixtures were tested for evaporation heat transfer coefficients and compared against a baseline of R-22 data. These refrigerants were tested in an in-tube heat transfer test facility, which has been used in the past for evaporation and condensation tests of refrigerants R-12, R-113, R-22, and R-134a [Eckels and Pate, 1991, Doerr, et al., 1994]. Several problems are unique to testing non-azeotropic blends, such as a temperature glide which makes measurement of the inlet and outlet temperatures, where the quality can vary as much as 0 to 100%, more difficult than it does for a pure refrigerant.

Heat transfer coefficient data was taken during in-tube evaporation of refrigerant mixtures in a 3/8" O.D. (9.7 mm) smooth copper tube. The data were taken at temperatures of -2.0°C and 4.0°C. Five non-azeotropic refrigerant mixtures were tested and compared against baseline data for R-22. Four different mass fluxes ranging from 125 to 375 kg/s·m^2 were tested. On a mass basis, the mixtures tested were:

Mixture 1 R-125 (40%) / R-32 (60%)
Mixture 2 R-134a (90%) / R-32 (10%)
Mixture 3 R-134a (75%) / R-32 (25%)
Mixture 4 R-32 (30%) / R-125 (10%) / R-134a (60%)
Mixture 5 R-125 (44%) / R-143a (52%) / R-134a (4%)

TEST FACILITY

The test facility consists of four main parts: a refrigerant loop, a water loop, a data acquisition system, and a dual test section. For these tests only one side of the dual test section was used. The following sections provides detailed descriptions of the four main parts of the test facility. A schematic of the test rig is shown in Figure 1.

Test Section

The test section consists of a horizontal test tube and a surrounding annulus. The inner tube is a 3/8" O.D. (9.7 mm) copper tube which is 3.67 m long. The annulus which surrounds the tube is also 3.67 m long and is constructed of a copper tube with a 17.2 mm inside diameter. The test tube is centered in the annulus by a series of spacers. The spacers are constructed of three stainless steel rods spaced 120 degrees about the annulus. The spacers are held in place by a series of industrial PG Teflon glands.

The test section is instrumented with temperature and pressure sensors. The temperatures are measured with resistance type temperature probes, RTDs, which have been calibrated to an accuracy of ±0.05°C. The pressure is measured with a calibrated strain-gage type pressure transducer which is accurate to ±9 kPa. The pressure drop in the test section is measured with a strain-gage type differential pressure transducer accurate to ±0.2 kPa.

Refrigerant Loop

The refrigerant loop consists of an after-condenser, a positive displacement pump, an accumulator, and a boiler. The after-condenser is a co-axial heat exchanger which condenses and subcools the refrigerant leaving the test section. The water-glycol mixture for the heat exchanger is provided by a R-502 chilling unit. After being subcooled the refrigerant is circulated by a positive displacement pump. The pressure in the test section is controlled by the bladder accumulator. This accumulator also helps to dampen out pressure fluctuations that may occur in the system. The quality of the refrigerant entering the test section is set by a heater located directly upstream of the test section. The heater is a 12.7 mm O.D. by 2.63 m long stainless steel tube heated by direct current. The heater is electrically isolated from the rest of the system by a high pressure rubber hose. The refrigerant mass flow rate is measured by a coriolis type mass flow meter accurate to 0.15% of the mass flow rate plus 2.25e-05 kg/s.

Water Loop

The water loop consists of a centrifugal pump, an in-line electric heater, and a heat exchanger. The mass flow rate is controlled by a valve that restricts the flow of water. The temperature of the water entering the test section is controlled by an electric heater. The water mass flow rate is also measured by a coriolis type flow meter with an accuracy of 0.15% of the mass flow rate plus 2.25e-04 kg/s.

Data Acquisition

Data acquisition is done with a personal computer, a 40 channel scanner, and a multimeter. The controlling program on the personal computer is written in FORTRAN and controls the multimeter and the scanner via a IEEE-488 bus.

EXPERIMENTAL PROCEDURE

The test facility is allowed to come to steady state before the final data acquisition is done. This is achieved by setting the mass flow rates, the refrigerant quality, and the annulus water temperatures, with the latter controlling the outlet quality. The data acquisition system then scans for temperature, mass flow rate, and pressure fluctuations. When the fluctuations are minimal the final data acquisition program is run. Each of the channels is scanned a total of five times while the pressure is scanned 35 times because of pressure drop fluctuations. The inlet quality is maintained between 8 and 15% and the outlet quality is kept between 80 and 85%.

DATA ANALYSES

Raw data from the data acquisition system are analyzed for each run to determine the in-tube heat transfer coefficient and the quality. The main equations used in processing the raw data are based on energy balances. The energy transferred in the test section is computed from an energy balance on the water side.

$$Q_w = M_w \cdot Cp_w \cdot (T_{w,out} - T_{w,in}) \qquad (1)$$

The quality change in the test section is determined from the energy change of the water side.

$$\Delta X = Q_w / (M_r \cdot h_{fg}) \qquad (2)$$

The refrigerant-side heat transfer coefficient is determined from an overall heat transfer coefficient and the annulus-side heat transfer coefficient. The annulus-side heat transfer coefficient, h_o, was determined by using a modified Wilson plot technique over the range of flow rates and temperatures encountered during evaporation tests. The correlation for the annulus side heat transfer coefficient is a resulting from the Wilson plot technique is a Dittus-Boelter type equation. The overall heat transfer coefficient is determined from the energy balance on the test section.

$$U_o = Q_w / (A_o \cdot LMTD) \qquad (3)$$

The log mean temperature difference is determined from the inlet and outlet temperatures on the water and refrigerant sides.

$$LMTD = (\Delta T_1 - \Delta T_2) / \ln (\Delta T_1/\Delta T_2) \tag{4}$$

where

$$\Delta T_1 = T_{r,out} - T_{w,in} \tag{5}$$

$$\Delta T_2 = T_{r,in} - T_{w,out} \tag{6}$$

Assuming the thermal resistance of the copper tubing as negligible, the refrigerant-side heat transfer coefficient is then determined from

$$h_i = 1 / (1/U_o - 1/h_o) A_i/A_o \tag{7}$$

EXPERIMENTAL DATA

R-22 and five refrigerant mixtures were tested for heat transfer coefficients and pressure drops during evaporation at -2°C and 4°C. The mass fluxes tested ranged from 125 kg/s·m² to 375 kg/s·m².

Data Results

Heat Transfer Coefficients

The different refrigerants were compared on an equal cooling capacity basis as well as an equal mass flux basis to try to eliminate the differences due solely to the difference in their enthalpies of vaporization. Figures 2 and 3 shows heat transfer coefficients versus mass flux for all the refrigerants at both -2.0°C and 4.0°C. When the refrigerants are compared on a mass flux basis, mixture 1 performs the best having a heat transfer coefficient approximately 90% larger than R-22 for a mass flux of 375 kg/s·m² at 4°C and almost double the heat transfer coefficients of R-22 for a mass flux of 125 kg/s·m². Mixture 2 had the lowest performance of all the mixtures with a heat transfer coefficient almost 20% lower than R-22 at a mass flux of 375 kg/s·m² and a heat transfer coefficient almost 10% lower than R-22 for a mass flux of 125 kg/s·m².

A summary of results is listed in Table 1 where the ratio of the heat transfer coefficients for the mixture compared to R-22 for a mass flux of 300 kg/s·m² at temperatures of 4°C and -2°C is presented.

TABLE 1
Ratio of Heat Transfer Coefficients to R-22 at 300 kg/s·m²

	Mixture 1	Mixture 2	Mixture 3	Mixture 4	Mixture 5
-2°C	1.71	0.97	1.11	1.04	1.10
4°C	2.01	0.88	0.97	1.07	1.04

Table 2 shows the ratio of the heat transfer coefficients for the refrigerants at 4°C compared to -2°C for a mass flux of 300 kg/s m². All of the refrigerants tested had higher heat transfer coefficients at 4 C than at -2 C. For example, at a mass flux of 300 kg/s·m² the heat transfer coefficient of R-22 is approximately 12% larger at 4°C than at -2°C.

TABLE 2
Ratio of Heat Transfer Coefficients at 4°C Compared to -2°C at 300 kg/s·m²

R-22	Mixture 1	Mixture 2	Mixture 3	Mixture 4	Mixture 5
1.12	1.31	1.02	1.01	1.16	1.04

An approximate cooling capacity can be defined as the mass flow rate of the refrigerant multiplied by the refrigerants enthalpy of vaporization, which assumes a 100% quality change. Figures 4 and 5 show plots of refrigerant heat transfer coefficients versus cooling capacity. When the refrigerants are compared on an equal cooling capacity basis, then mixture 1, R-32(60%) / R-125(40%), performed the best, having a heat transfer coefficient almost 90% larger than R-22 for a cooling capacity of 35 kJ/s·m^2 at an evaporator temperature of 4°C. A comparison summary of the different mixtures are shown in Table 3 where the ratio of the heat transfer coefficients for the mixture compared to R-22 are listed for a cooling capacity of 45 kJ/s·m^2 at temperatures of -2°C and 4°C. As shown in Table 3, mixture 1 has the highest heat transfer coefficient when compared on an equal cooling capacity basis while mixture 2 has the lowest performance.

TABLE 3
Ratio of Heat Transfer Coefficients to R-22 at 45 kJ/s·m^2 Compared on an Equal Cooling Capacity Basis

	Mixture 1	Mixture 2	Mixture 3	Mixture 4	Mixture 5
-2°C	1.60	0.98	1.04	0.95	1.25
4°C	1.90	0.92	0.96	1.05	1.21

It should be noted that other sources of data have shown heat transfer performances of the R-32(60%) / R-125(40%) mixture somewhat lower than that reported in this paper. Specifically, Bivens, et al. (1993) reported that the heat transfer performance for R-32(60%) / R-125(40%) is 50% to 60% higher than R-22. The reasons for the differences from these two studies is still being investigated.

Pressure Drop

Pressure drop data were also taken for the refrigerant mixtures during evaporation, and the data are plotted in Figures 6 and 7. As can be seen from the graphs, the pressure drops for mixtures 2 and 3 are greater than R-22 while the pressure drops for mixtures 1 and 5 are less than R-22. These trends occur at both 4 and -2°C. The pressure drop for mixture 4 is the closest to R-22, with the mixture having a slightly larger pressure drop than R-22 at 4°C and about the same pressure drop as R-22 at -2°C. All of the refrigerants exhibited larger pressure drops at -2°C than at 4°C. This trend is probably due to the higher viscosity of the refrigerants at the lower temperature. The above comparisons are summarized in Table 4 where the ratio of the pressure drops for the mixtures compared to R-22 are listed for a mass flux of 300 kg/s·m^2 at a temperature of 4°C.

TABLE 4
Ratio of pressure drop versus R-22 at 4°C and 300 kg/s·m^2

	Mixture 1	Mixture 2	Mixture 3	Mixture 4	Mixture 5
4.0°C	0.88	1.29	1.22	1.10	0.66

Property Data

All of the thermodynamic and transport properties for the refrigerants used in this study were supplied by a refrigerant manufacturer [DuPont, 1993].

CONCLUSION

The mixture R-32(60%) / R-125(40%) exhibited the highest heat transfer coefficients out of all the refrigerants tested at both 4°C and -2°C when compared on both an equal mass flux and an equal cooling capacity basis. The R-32/ R-125 mixture also had a lower pressure drop than R-22. The mixture R-134a(90%) / R-32(10%) had the lowest heat transfer coefficients at both 4°C and -2°C when compared on both an equal mass flux and an equal cooling capacity basis. In addition, this same mixture had the highest pressure drop. Except for the lowest performing refrigerant, all of the refrigerants performed either similar or better than R-22. Also of special note, all of the refrigerants had larger heat transfer coefficients at 4°C than at -2°C.

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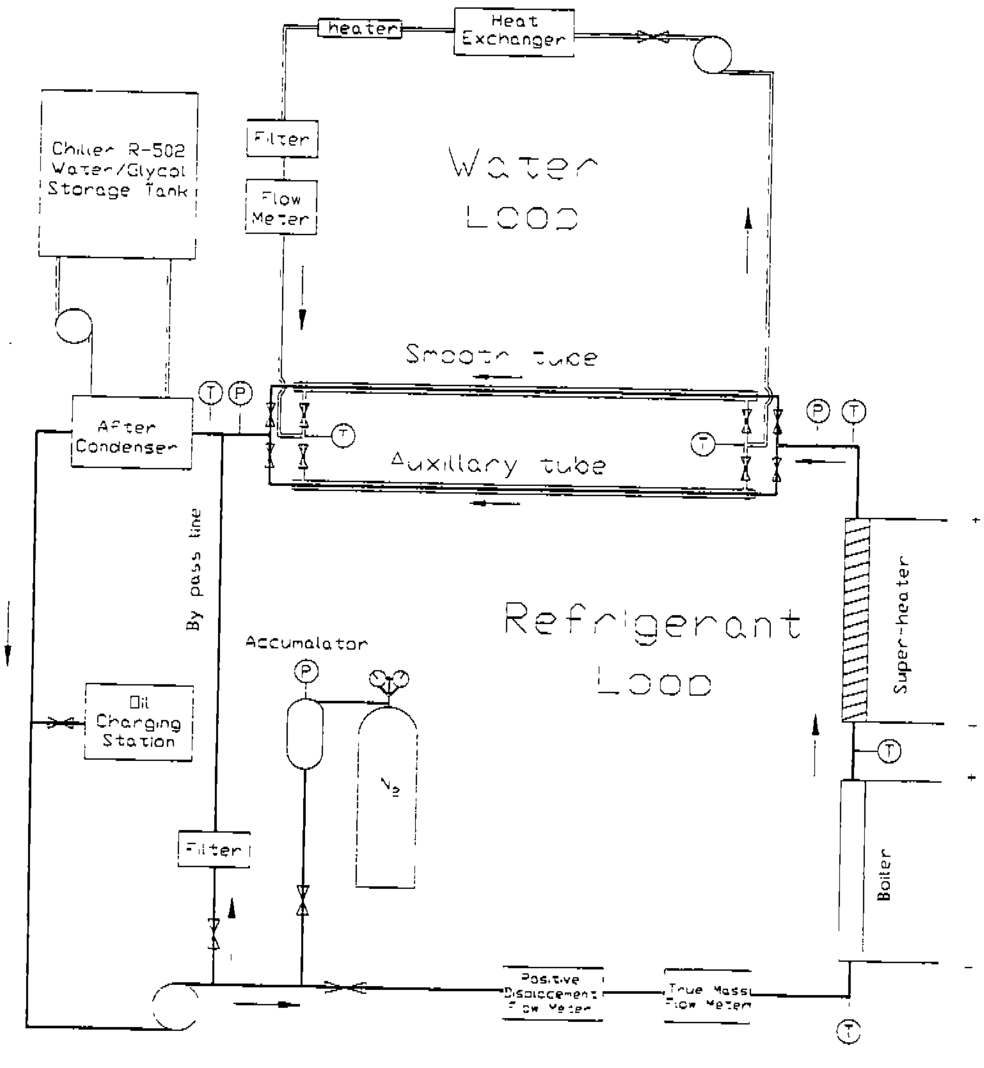

Figure 1 Schematic of In-Tube Test Facility

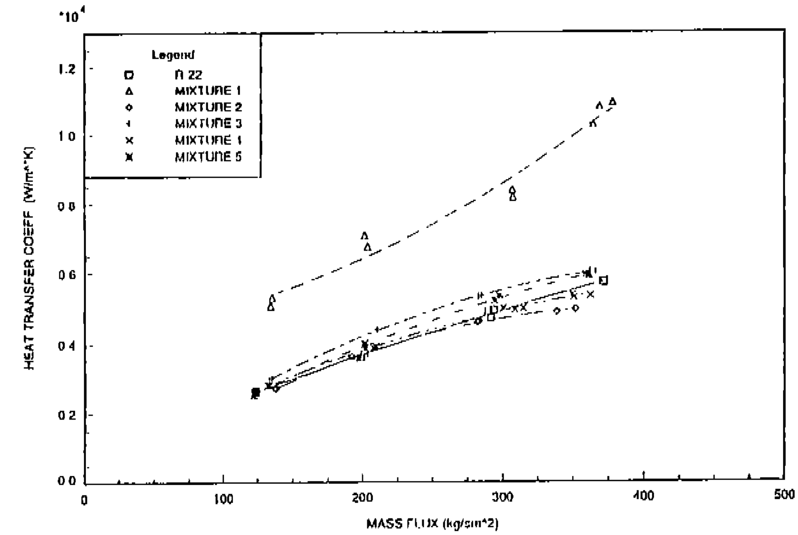

Figure 2 Heat Transfer Comparison at -2°C Based on Mass Flux

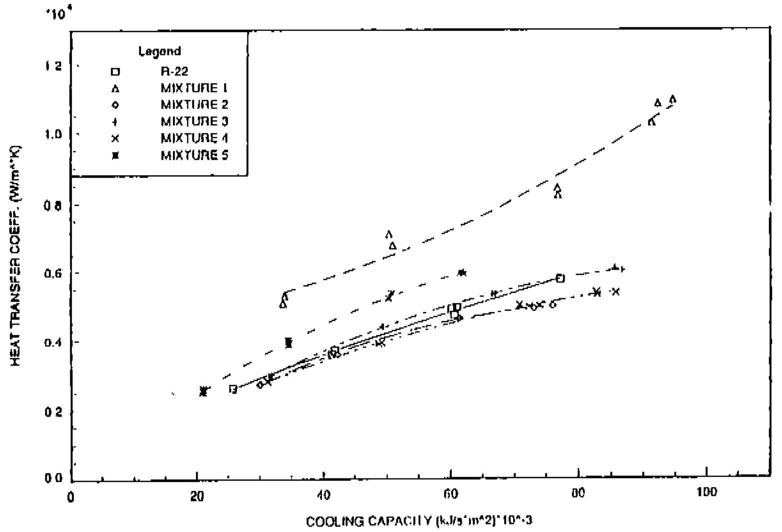

Figure 3 Heat Transfer Comparison at 4°C Based on Mass Flux

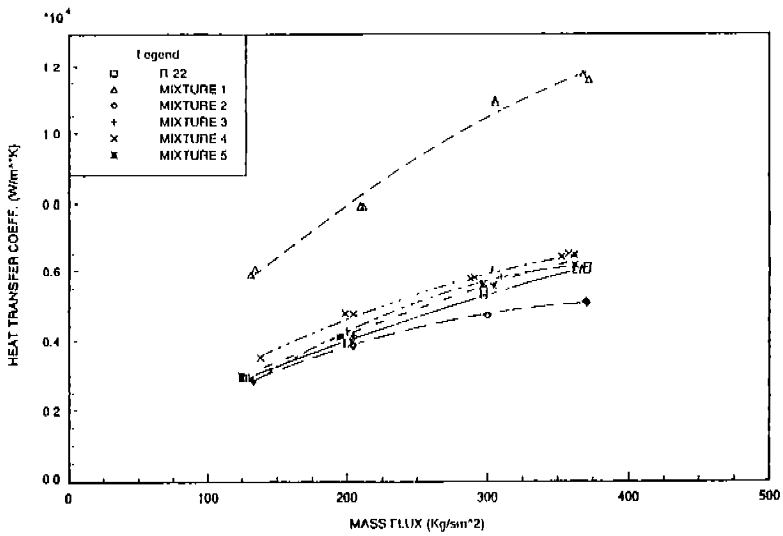

Figure 4 Heat Transfer Comparison at -2°C Based on Cooling Capacity

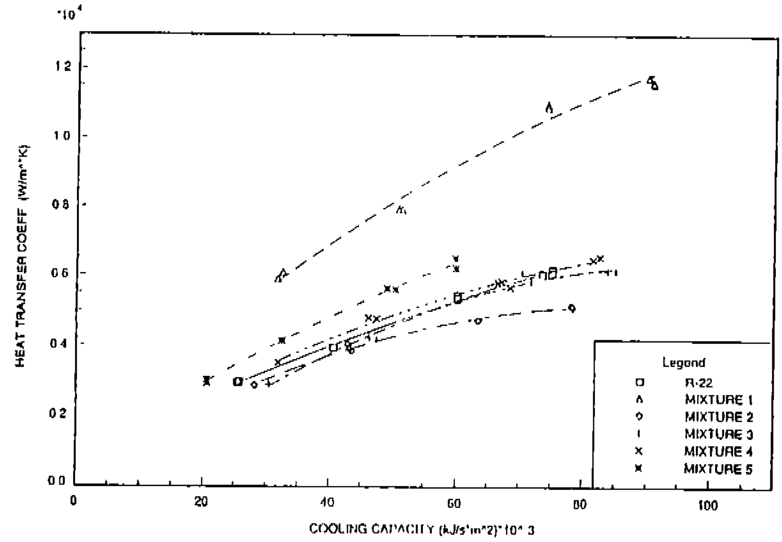

Figure 5 Heat Transfer Comparison at 4°C Based on Cooling Capacity

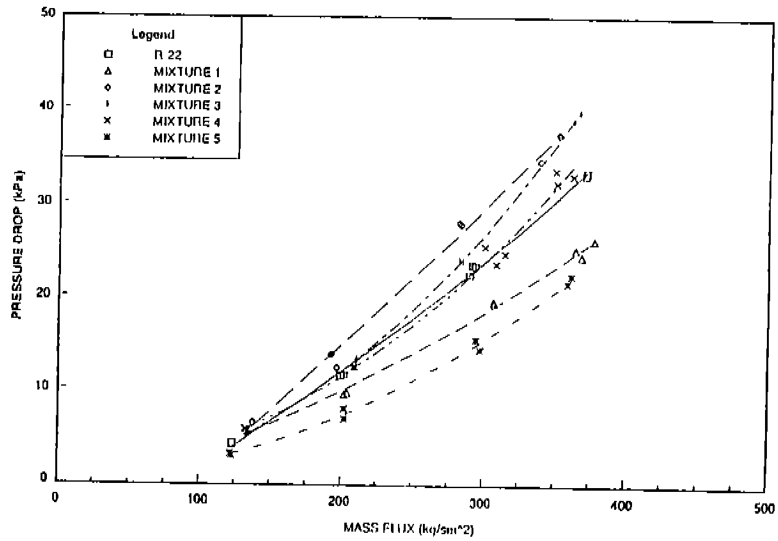

Figure 6 Pressure Drop Comparison at -2°C

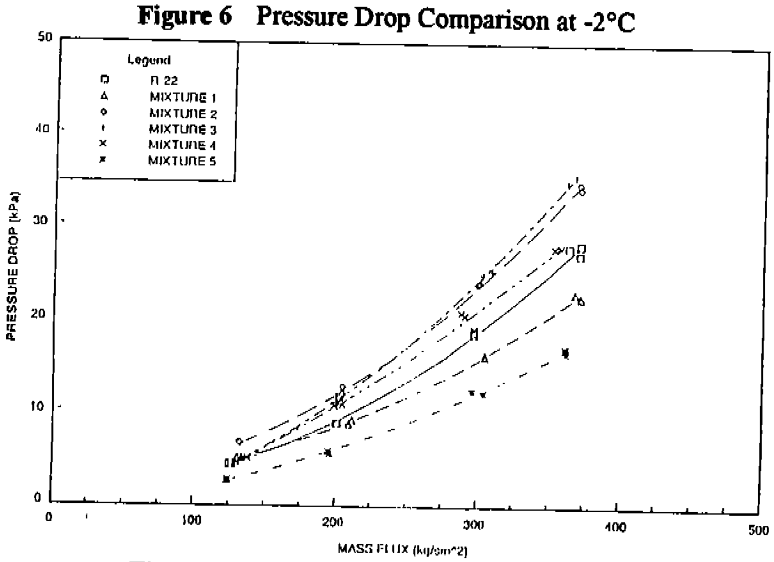

Figure 7 Pressure Drop Comparison at 4°C