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PRESSURE DROP DURING FORCED CONVECTION BOILING OF BINARY REFRIGERANT MIXTURES

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Abstract—An experimental investigation was carried out in an electrically heated horizontal tube to measure pressure drop for various flow rates and heat fluxes during forced convection boiling of pure refrigerant 12 and four compositions of refrigerants 13 and 12 mixtures. The Martinelli-Nelson correlation, using the properties of the flowing refrigerant mixture, could not predict satisfactorily the pressure drop data. Total, as well as frictional pressure drops were found to be function of concentration. Two separate models each for total, as well as frictional pressure drop were developed for predicting the corresponding pressure drop. In each case, the maximum per cent deviation between predicted and measured pressure drop was within ±30%.

INTRODUCTION

Mixtures of pure refrigerants when used in convectional vapour compression refrigeration systems result in non-isothermal boiling as well as condensation processes. This in turn reduces thermodynamic irreversibility involved in the phase change processes equipment. In addition, mixed refrigerants possess numerous other advantages over pure refrigerants, namely, improvement in miscibility of refrigerant and oil, better control and increase in refrigerating capacity and a wider cooling range Arora (1967), Arora (1970), Haselden (1952–53), Haselden & Klimek (1957–58), McHarness & Chapman (1962) and Tchaikoveski & Kuznetsov (1963). But inspite of these advantages, mixed refrigerants have not yet been put to commercial use. The major difficulty is that accurate design of heat transfer equipment is not possible because of the non-availability of reliable experimentally measured data for thermodynamic and transport properties of the binary refrigerant mixtures and also the design parameters, namely, the heat transfer and pressure drop characteristics.

Measured pressure drop data, during forced convection phase change in a horizontal duct, is available for commonly used pure refrigerants and many calculation models have been suggested for its determination. Altmann *et al.* (1960), Hatch & Jacobs (1962), Johnson & Chaddock (1964), Pierre (1964a, b). However, experimentally measured pressure drop data are scarce for binary mixtures of pure refrigerants. The present experimental investigation reports a correlation for pressure drop during forced convection boiling of mixtures of refrigerants 12 and 13.

EXPERIMENTAL APPARATUS AND DATA COLLECTION

A line diagram of the experimental set up is shown in figure 1. The test evaporator consisted of two identical horizontal stainless steel tubes of effective length 2.35 m each and having inside and outside diameters as 9.52 and 12.50 mm respectively. These tubes were connected by a smooth copper U-bend. The refrigerant flowed through these tubes in series. They were heated directly by passing a stabilized low voltage high intensity current through them.

The refrigerant circulation system employed a twin cylinder reciprocating compressor. An oil separator was installed in the discharge line to make circulating refrigerant effectively oil free. An evaporative type condenser was used to ensure complete condensation. A rotameter installed in the liquid line measured the flow rate of the circulating refrigerant mixture. A pre-heater was located in between the expansion valve and the test section inlet so as to give any desired quality of the working medium at the inlet to the test section. A back pressure regulating valve was used to maintain constant pressure at the test section outlet for various flow rates and heat fluxes. An afterheater ensured dry vapour reached the compressor inlet.

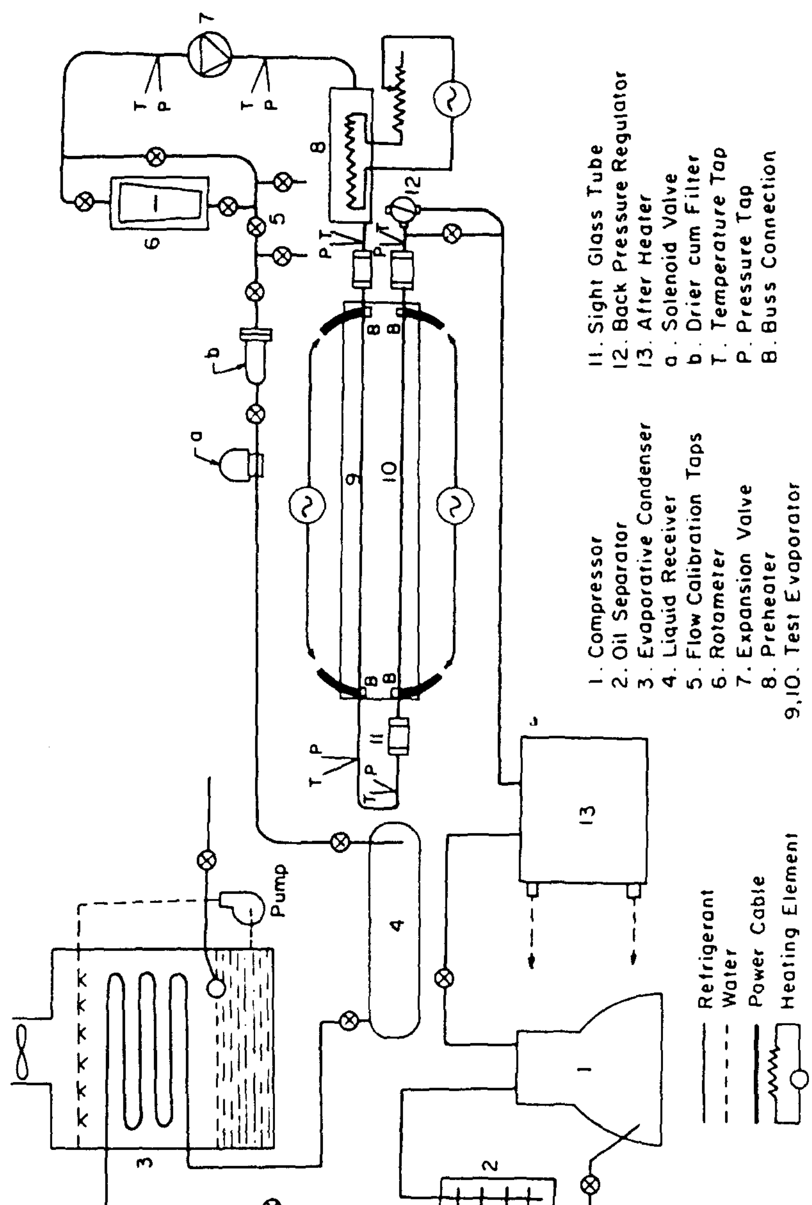

Figure 1. Line diagram of experimental apparatus.

Pressure tappings were provided at the inlet and the outlet of the two test sections and they were connected to a common distributor with four shut-off valves, one for each location of pressure measurement. The pressures were measured by a precalibrated pressure gauge.

After starting the refrigerating plant, a pre-calculated heat flux was applied to the test sections. Then adjustments were made to the expansion valve and the back pressure regulating valve till a steady state condition was achieved for a constant pressure at the test section outlet for any particular flow rate and the chosen heat flux.

Measurements of flow rates, power input to the test section and preheater, pressures and temperatures were recorded. Since flow boiling shows random fluctuations, three to four readings were taken for each run to get good representative mean steady state values. Although large fluctuations were observed in temperatures, these were insignificant for pressure measurements. In all 116 test runs were conducted using different combinations of the following variables—

Heat flux	:	5000–17,000 W/m^2
Flow rate	:	60–120 kg/hr (234–454 kg/m^2 s)
Concentration of R-13 in R-12	:	0–20% by mass at the interval of 5%
Temperature at test section outlet	:	-9 to $+5$°C

The measured pressure drops for all test runs, viz. at different mass velocity, heat flux, the inlet and the outlet vapour quality of each test section, are given in Table A1 of Appendix A. The data are listed separately for each composition of refrigerant charge. Slight variation in vapour quality is noticeable between the outlet of the first test section and inlet of the second tube. This occurs due to drop of pressure in the tube bend and not because of any heat addition.

PRESSURE DROP ESTIMATION

Total pressure drop in a duct Δp_{total} consists of frictional Δp_f, accelerational Δp_a and gravitational Δp_g components given by [1]

$$\Delta p_{\text{total}} = \Delta p_f + \Delta p_a + \Delta p_g \tag{1}$$

where Δp_{total} is the total two-phase pressure drop, Pa Δp_f is the frictional two-phase pressure drop, Pa Δp_a is the acceleration pressure drop, Pa and Δp_g is the gravitational pressure drop, Pa.

Equation [2] gives the total pressure drop in a horizontal pipe, since the pressure drop due to gravity effect is zero

$$\Delta p_{\text{total}} = \Delta p_f + \Delta p_a \tag{2}$$

The irreversible frictional pressure drop component, Δp_f, is generally the most important contribution to the total pressure drop. In a two-phase flow, it is usually estimated using a physical model incorporating the experimentally measured data. However, the acceleration pressure drop component Δp_a is generally small as compared to the frictional pressure drop component Δp_f and it is a reversible pressure drop. The Martinelli–Nelson correlation (1948) has been used widely to predict the pressure drop during convective boiling in a horizontal pipe. Wallis (1969) has used this development to derive [3].

$$\Delta p_{MN} = \frac{2 f_{f0} G^2 L}{D \rho_L}\left[\frac{1}{x}\int_0^x \phi_{f0}^2 \, dx\right] + \frac{G^2}{\rho_L}\left[\frac{x^2}{\alpha}\left(\frac{\rho_L}{\rho_G}\right) + \frac{(1-x)^2}{(1-\alpha)} - 1\right] \tag{3}$$

where Δp_{MN} is the Martinelli–Nelson total two-phase pressure drop, Pa, f_{f0} is the frictional coefficient considering total two-phase flow as liquid flow, G is the mass velocity, kg/m^{2}s, L is the length, m, D is the diameter, m, ρ_L is the density of liquid, kg/m^3, x is the vapour quality, ϕ_{f0}^2 is the two-phase frictional multiplier, α is the void fraction and ρ_G is the density of vapour, kg/m^3.

Equation [3] in general, requires a stepwise integration as it demands the local values of the two-phase frictional multiplier ϕ_{f0}^2 and the void fraction α. Steps employed to determine ϕ_{f0}^2 and α for convective phase change have been described in detail by Collier (1972) and Wallis (1969) and are as follows—

$$\left[\frac{1}{\phi_L^2}\right]^{1/n} + \left[\frac{1}{\phi_G^2}\right]^{1/n} = 1 \text{ with } n = 4 \text{ for turbulent flow}$$

where ϕ_L^2 is the two-phase frictional multiplier based on pressure gradient for liquid alone flow and ϕ_G^2 is the two-phase frictional multiplier based on pressure gradient for gas alone flow

$$X^2 = \frac{\phi_G^2}{\phi_L^2} \qquad [5]$$

where

$$X = \text{Lockhart–Martinelli parameter}$$

$$= \left(\frac{\rho_G}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_G}\right)^{0.1} \left(\frac{1-x}{x}\right)^{0.9}$$

μ_L is the viscosity of liquid, kg/ms and μ_G is the viscosity of vapour, kg/ms.

On simplifying, [4] and [5] give

$$\phi_L = \left(1 + \frac{1}{X^{0.5}}\right)^2. \qquad [6]$$

Frictional pressure drop, considering total flow as liquid, is given in [7]

$$\Delta p_{f,MN} = \left[\frac{2 f_{f0} G^2 L}{D \rho_L}\right] \phi_{f0}^2 \qquad [7]$$

where $\Delta p_{f,MN}$ is the Martinelli–Nelson frictional two-phase pressure drop, Pa.

The Blasius relation for smooth pipe turbulent flow is given in [8] and [9]. Further, equation [10] expresses the relation between ϕ_L given in [6] and ϕ_{f0} required in [3], Wallis (1969) and Collier (1972),

$$f_{f0} = 0.079\,(Re)^{-0.25} \qquad [8]$$

where

$$Re = \frac{DG}{\bar{\mu}_L} \qquad [9]$$

$$\phi_{f0}^2 = \phi_L^2 (1-x)^{1.75}. \qquad [10]$$

The Martinelli value of void fraction α at low pressures has been expressed by Wallis (1969) as,

$$\alpha = (1 + X^{0.8})^{-0.378}. \qquad [11]$$

The first term on the right hand side of equation [3] involves integration of ϕ_{fo}^2 with respect to vapour quality x. This has been done numerically by dividing each tube length into 55 subsections of 4 cm each. The values of x, X and the thermodynamic and transport properties were considered constant over this short length. However, it may be observed here that in case of binary refrigerant mixtures, the liquid and vapour compositions change continuously as the boiling proceeds non-isothermally. Therefore, the properties of the liquid and vapour phases change quite significantly along the test length.

Thermodynamic properties, namely, liquid enthalpy, vapour enthalpy and vapour density of the mixtures of refrigerants 13 and 12 have been determined using equations from Agarwal (1975). However, the thermodynamic and transport properties of pure refrigerants 13 and 12 have been taken from ASHRAE (1972) and ASHRAE (1973). Based on Kandlikar *et al.* (1975), transport properties of the refrigerant mixtures have been determined using [12]–[18].

$$\rho_{L,m} = \frac{\rho_{L,1}\rho_{L,2}}{W_{L,1}\rho_{L,1} + W_{L,2}\rho_{L,2}} \qquad [12]$$

where W_L is mass concentration in liquid phase, and subscripts m, 1 and 2 represent mixture and components 1 and 2 respectively.

$$C_{p_{L,m}} = W_{L,1}C_{p_{L,1}} + W_{L,2}C_{p_{L,2}} \qquad [13]$$

$$\mu_{L,m} = (\mu_{L,1})^{W_{L,1}}(\mu_{L,2})^{W_{L,2}} \qquad [14]$$

$$\mu_{G,m} = \frac{\mu_{G,1}}{1 + \left(\frac{W_{G,2}}{W_{G,1}}\right)\beta_{12}} + \frac{\mu_{G,2}}{1 + \left(\frac{W_{G,1}}{W_{G,2}}\right)\beta_{21}}. \qquad [15]$$

Here, the subscript G stands for gaseous phase. Further,

$$\beta_{12} = \frac{\left[1 + \left(\frac{\mu_{G,1}}{\mu_{G,2}}\right)^{0.5}\left(\frac{M_2}{M_1}\right)^{0.25}\right]^2}{2\sqrt{2}\left(1 + \frac{M_1}{M_2}\right)^{0.5}} \qquad [16]$$

$$\beta_{21} = \beta_{12}\left(\frac{\mu_{G,2}}{\mu_{G,1}}\right)\left(\frac{M_1}{M_2}\right) \qquad [17]$$

$$k_{L,m} = k_{L,1}W_{L,1} + k_{L,2}W_{L,2} + 0.72|k_{L,2} - k_{L,1}|W_{L,1}W_{L,2} \qquad [18]$$

where M is the molecular weight and c_p is the specific heat at constant pressure, J/kg K.

EXPERIMENTAL PRESSURE DROP

Figure 2 shows the variation of the experimentally measured total pressure drop for refrigerant 12 plotted against those calculated using the method of Martinelli–Nelson. It is apparent from this graph that the Martinelli–Nelson correlation mostly overpredicts the pressure drop for refrigerant 12. A best fitting straight line on this graph has a slope of 1.185. Consequently the Martinelli–Nelson correlation did not predict satisfactorily the measured pressure drop data for pure R-12. Such observations also have been reported by several workers namely, Hatch & Jacob (1962), Johnson & Chaddock (1964), Pierre (1964a, b).

Further, figures 3–6 show a similar comparison for four mixtures of refrigerants 12 and 13. From the study of these graphs, it is apparent that the Martinelli–Nelson correlation underpredicts the two-phase pressure drop for mixtures. However, the deviation becomes larger as

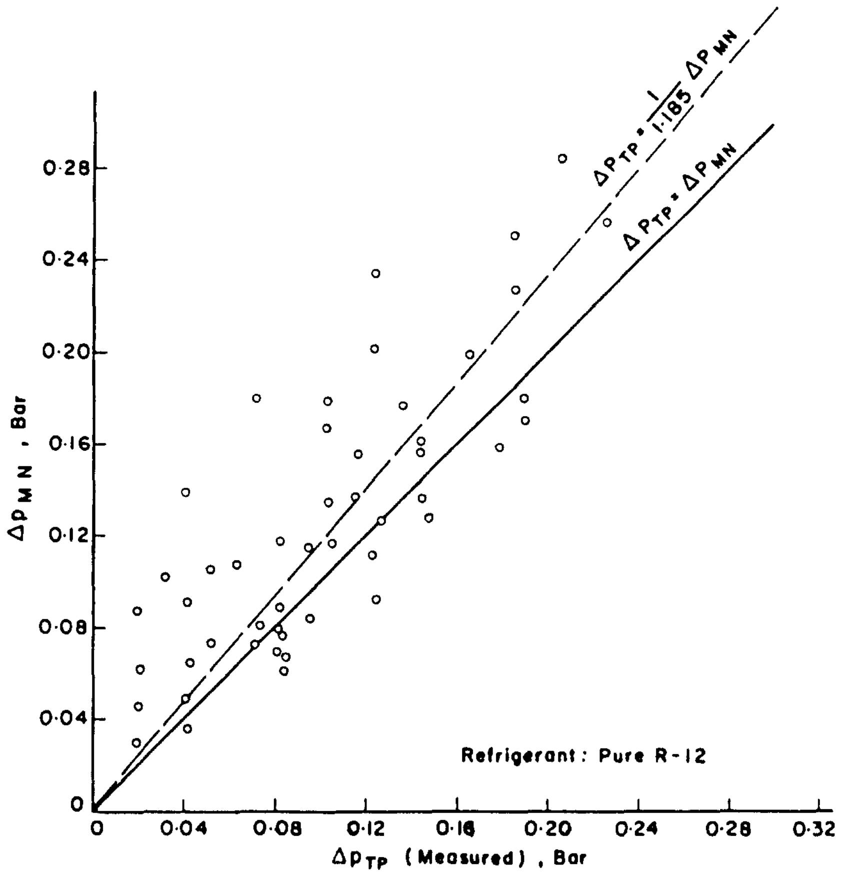

Figure 2. Comparison of Δp_{MN} vs Δp_{TP} (Pure R-12).

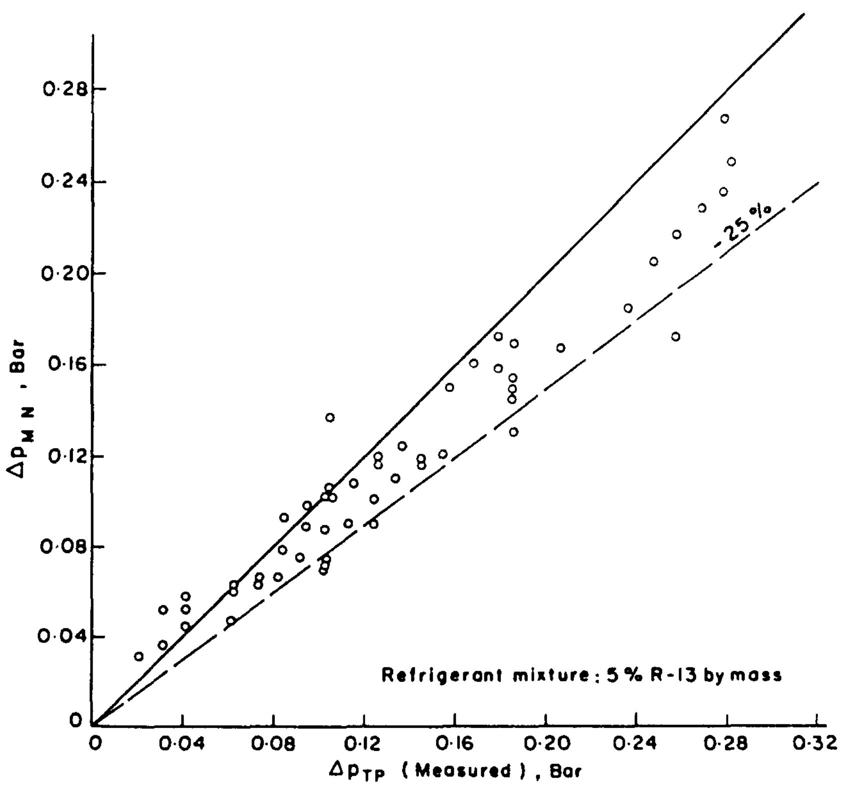

Figure 3. Comparison of Δp_{MN} vs Δp_{TP} (5% R-13, 95% R-12).

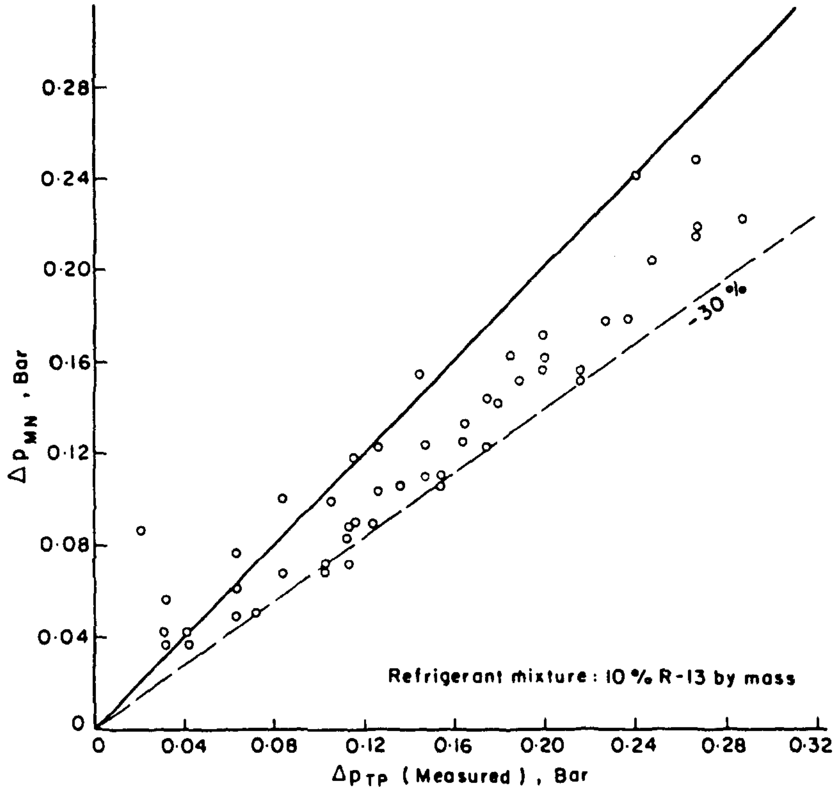

Figure 4. Comparison of Δp_{MN} vs Δp_{TP} (10% R-13, 90% R-12).

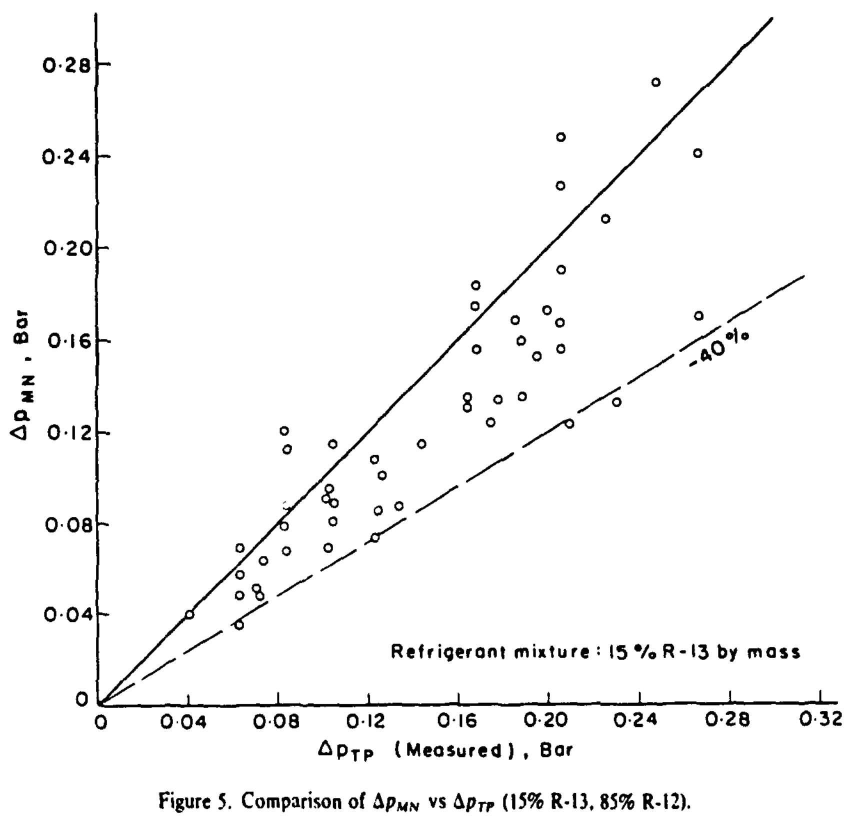

Figure 5. Comparison of Δp_{MN} vs Δp_{TP} (15% R-13, 85% R-12).

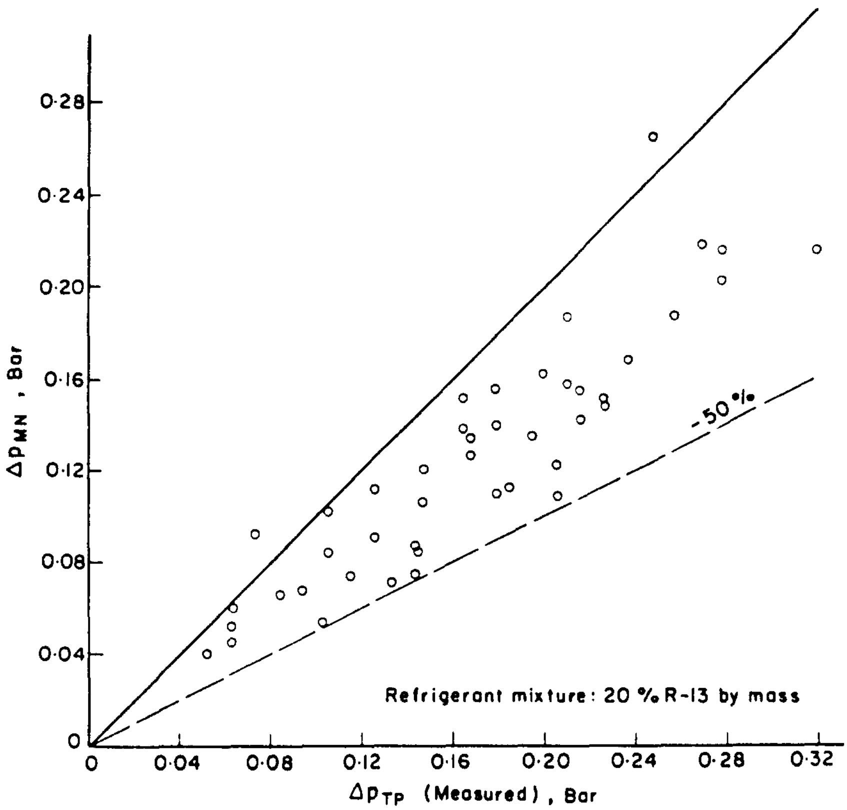

Figure 6. Comparison of Δp_{MN} vs Δp_{TP} (20% R-13, 80% R-12).

the concentration of refrigerant 13 increases, indicating also that the experimentally measured pressure drop is a function of mixture composition.

DEVELOPMENT OF CORRELATIONS

An attempt was made to develop a correlation for pressure drop during boiling of binary mixtures of pure refrigerants 13 and 12. It was found that the ratio of the measured pressure drop to that calculated by the Martinelli-Nelson correlation tended to be the same for a given composition. Consequently, it was concluded that a relation of the type given in [19] exists

$$\frac{\Delta p_{TP}}{\Delta p_{MN}} = f(C) \qquad [19]$$

where Δp_{TP} is the measured two-phase total pressure drop, Pa and C is the concentration of R-13 in the binary mixture.

It may be noted that using [19] the ratio would be a constant for each composition. An attempt was made later to develop an expression for this ratio in terms of mixture composition. An examination of the measured pressure drop data in view of the relation proposed in [19], led to the following two simple expressions.

$$\frac{\Delta p_{TP}}{\Delta p_{MN}} = 0.87(1 + C)^{2.66} \qquad [20]$$

and

$$\frac{\Delta p_{TP}}{\Delta p_{MN}} = 0.89(1 - C)^{-2.12}. \qquad [21]$$

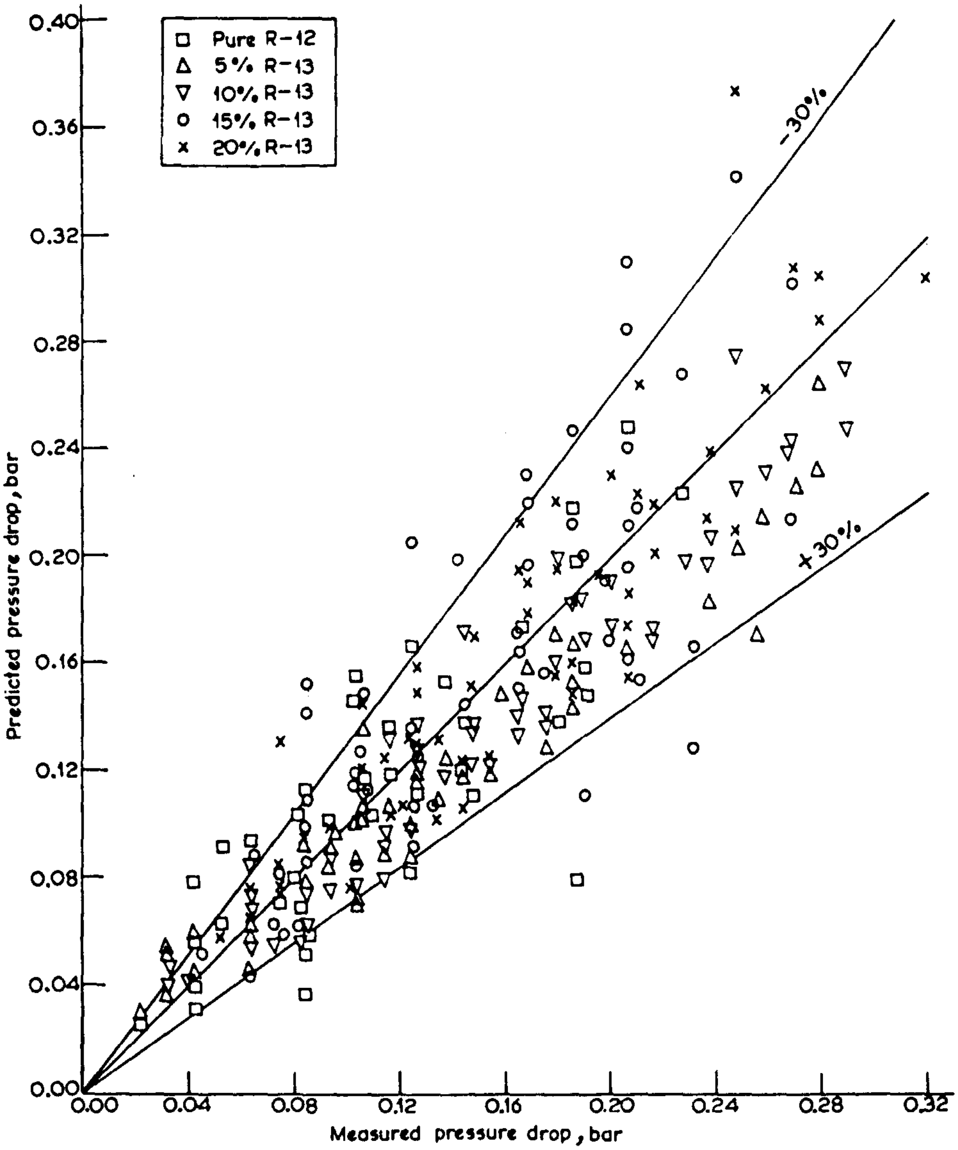

Figure 7. Comparison of predicted and measured pressure drops [20].

Figure 7 shows the variation of pressure drop predicted, by employing [20], versus experimentally measured pressure drop. From the study of this graph, it is apparent that almost the entire data have ±30% deviation. Similarly, the variation of pressure drop calculated with the use of [21] versus measured pressure drop is shown in figure 8. In this case too, the deviation is of the order of ±30%.

The percentage mean and standard deviations of total pressure drops calculated by using [20] and [21] from experimentally measured values were also calculated. Whereas the mean deviation was nearly zero for both these equations, the standard deviations were 23.1 and 23.2% respectively. This statistical analysis and figures 7 and 8 indicate that the two proposed correlations for total pressure drop are having the same degree of success. It may, however, be observed that the variables used, viz. $1 + C$ or $1 - C$ yield satisfactory correlation for pressure drop data with pure R-12 but the relation cannot possibly be extrapolated to pure R-13.

FRICTIONAL PRESSURE DROP CORRELATION

In the total pressure drop correlations described above the frictional as well as acceleration pressure drops calculated by the method of Martinelli–Nelson are multiplied by a corrective

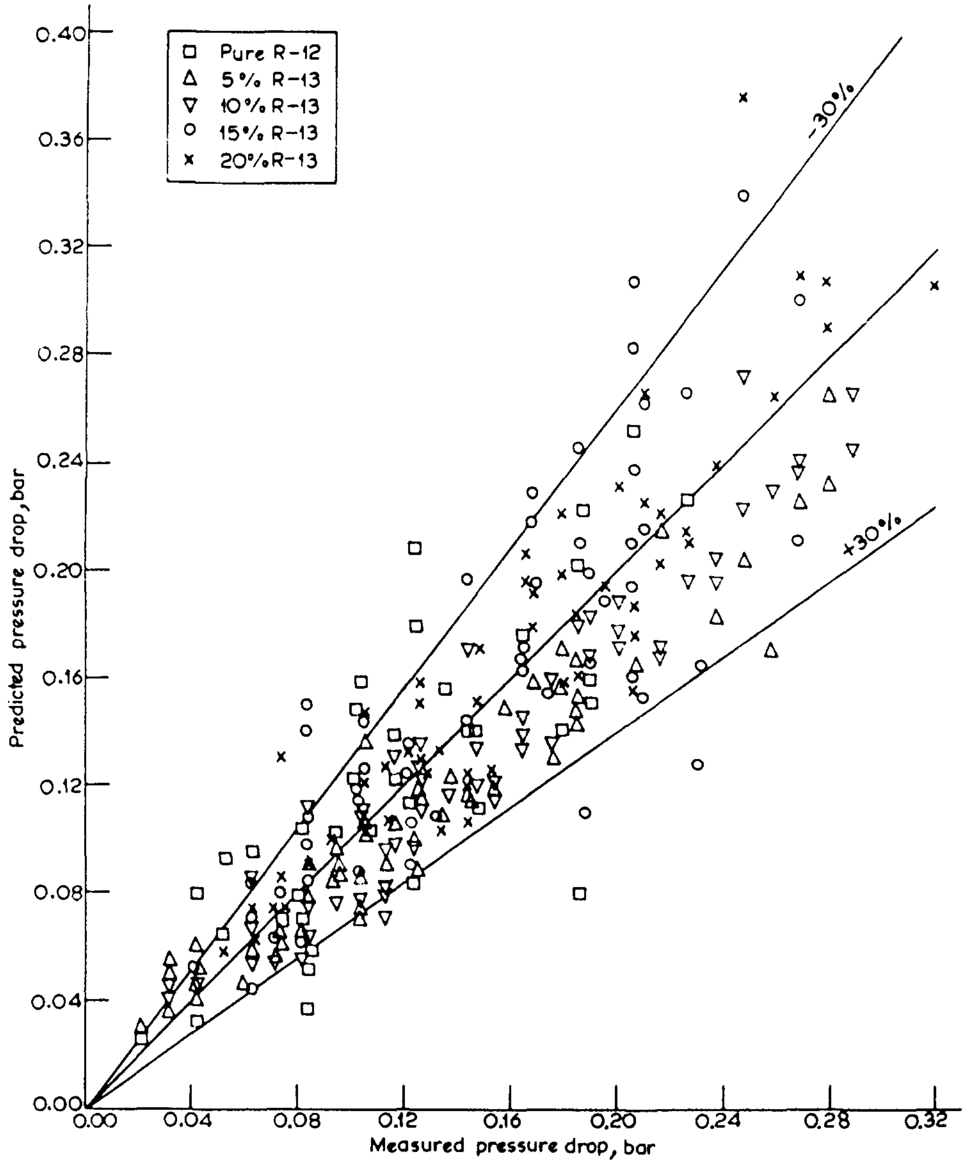

Figure 8. Comparison of predicted and measured pressure drops [21].

term which is a function of mixture composition. Since the acceleration pressure drop component is the difference due to the change of momentum, the use of the correction term could not be justified in the case of the acceleration pressure drop. However, as the acceleration pressure drop component is usually quite small as compared to the frictional pressure drop, the effect of including the acceleration pressure drop term in the above development would be insignificant. It was, therefore, decided to study the variation of the actual frictional drop component for the binary mixtures. The experimental values of the frictional pressure drop were calculated by subtracting the calculated acceleration pressure drop component Δp_a from the measured total pressure drop for the given test conditions using [3]. It was then possible to develop the two correlations for Δp_f as given in equations [22] and [23].

$$\Delta p_f = 0.86\,\Delta p_{f,MN}(1 + C)^{2.86} \qquad [22]$$

and

$$\Delta p_f = 0.88\,\Delta p_{f,MN}(1 - C)^{-2.28}. \qquad [23]$$

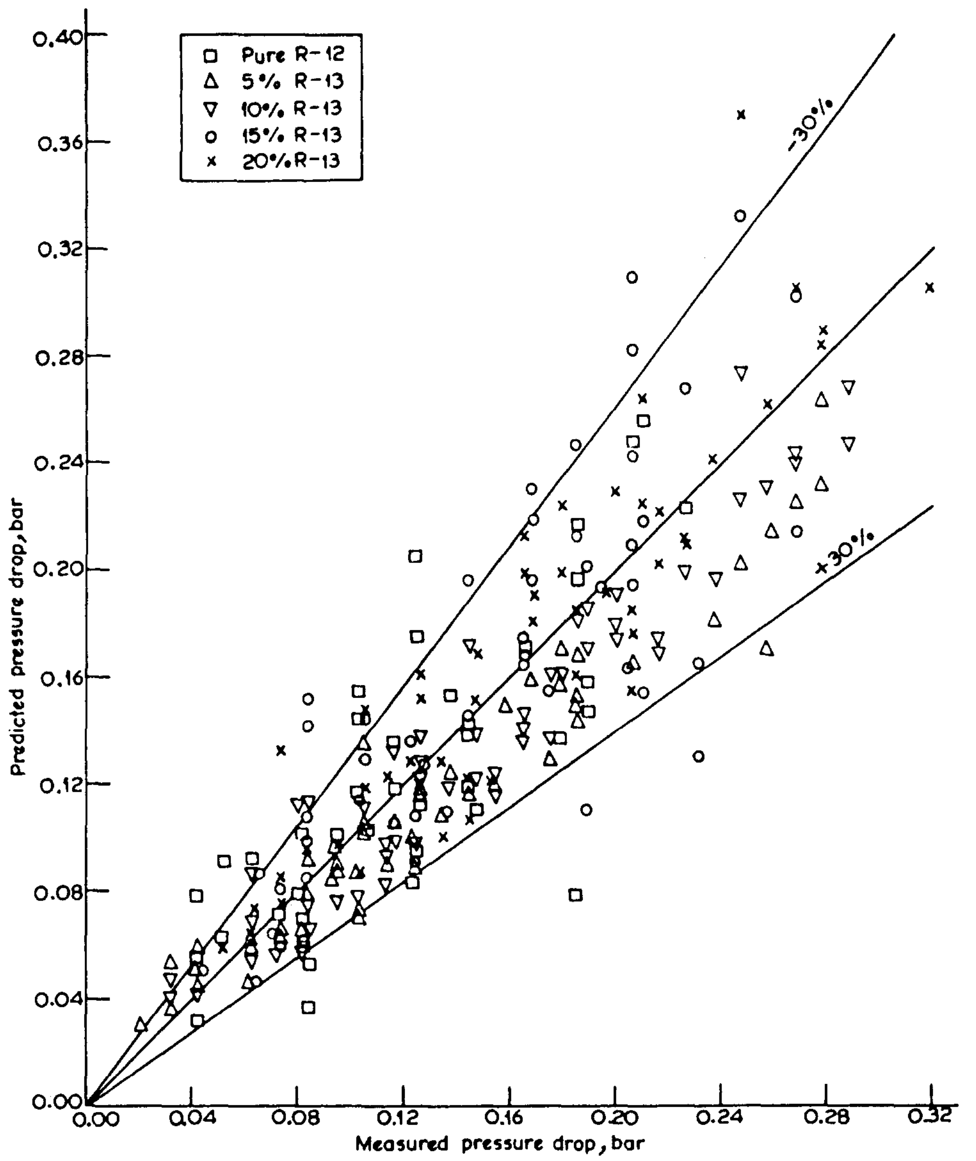

Figure 9. Comparison of predicted and measured pressure drops [22].

Comparing [22] and [23] with [7], it is observed that the two-phase friction multiplier, ϕ_{f0}^2, for the Martinelli–Nelson correlation has been modified for the pure refrigerant as well as binary mixtures. The total pressure drop can then be calculated by substituting the value of Δp_f from [22] and [23] into [3].

The variation of predicted pressure drop using equation [22] for Δp_f and the experimentally measured pressure drop is shown in figure 9. The dispersion of the entire data including that of refrigerant 12 is within $\pm 30\%$. Similarly figure 10 shows the variation of predicted pressure drop using [23] and measured pressure drop. Here in this case also, the percentage deviation for almost entire data is within $\pm 30\%$. The percentage mean deviation of the total pressure drops calculated on the basis of [22] and [23] with respect to those measured experimentally, was found to be nearly zero. The standard deviations were 23.1 and 23.2% respectively. The foregoing discussion indicates that these two correlations, developed on the basis of two-phase frictional pressure drop, also have the same degree of success as those based on total pressure drop.

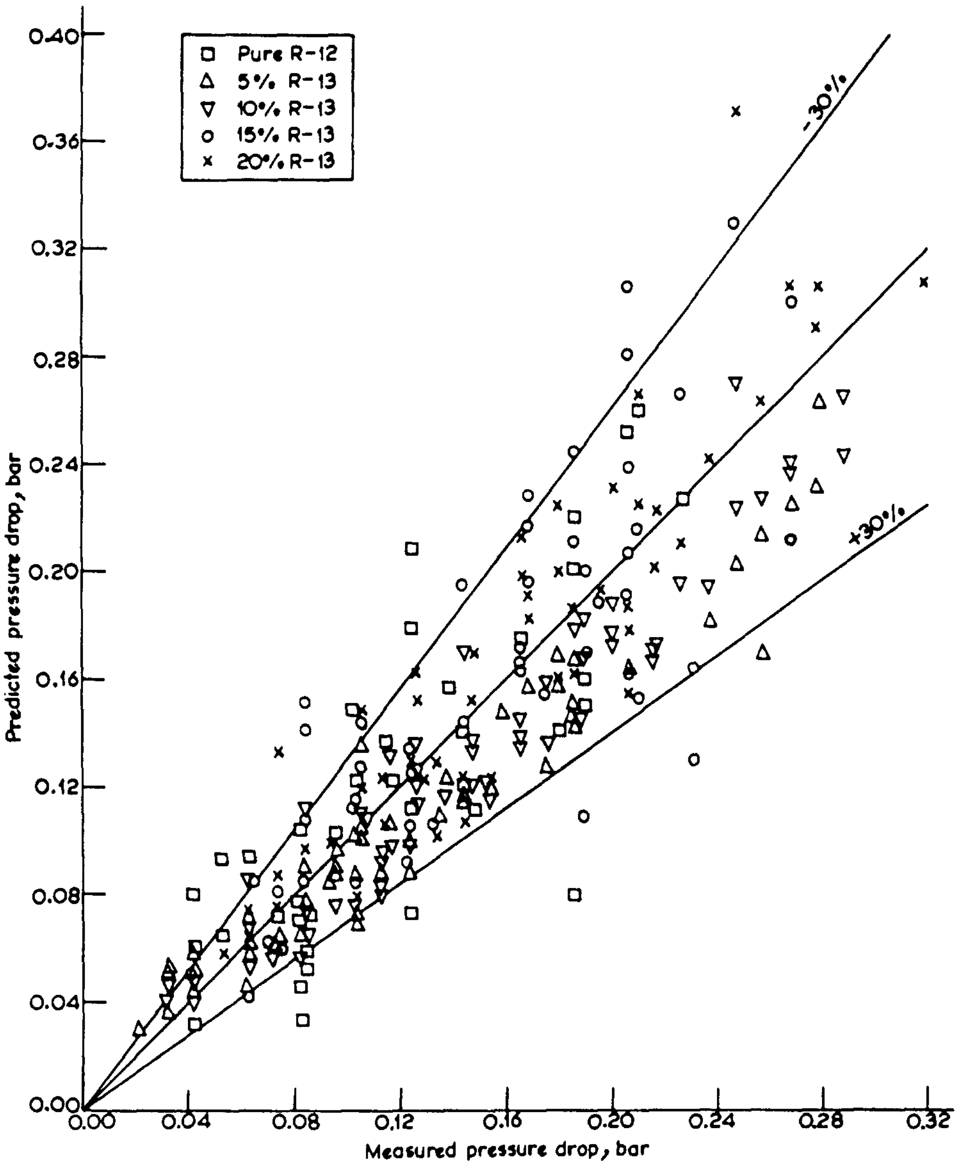

Figure. 10. Comparison of predicted and measured pressure drops [22].

CONCLUSIONS

From the study of predicted and experimentally measured pressure drop data, the following conclusions are drawn.

(1) The Martinelli–Nelson correlation mostly overpredicts the pressure drop data for pure R-12 and tends to underpredict the same for refrigerant mixtures of R-13 and R-12, the deviation increasing with increased fraction of R-13. Hence, the correlation does not predict satisfactorily the measured pressure drop data. Further, it was found that the pressure drop was a function of mixture composition also.

(2) Two correlations for total pressure drop as given below were found to produce an agreement with experimental measurements to within ±30%

$$\frac{\Delta p_{TP}}{\Delta p_{MN}} = 0.87(1 + C)^{2.66}$$

$$\frac{\Delta p_{TP}}{\Delta p_{MN}} = 0.89(1 - C)^{-2.12}.$$

(3) Two correlations have been developed by considering a modifying factor for binary refrigerant mixtures for the two-phase frictional pressure drop multiplier. The resulting correlations are given below—

$$\Delta p_f = 0.86\,\Delta p_{f,MN}(1+C)^{2.86}$$

$$\Delta p_f = 0.88\,\Delta p_{f,MN}(1-C)^{-2.28}.$$

The two correlations for frictional pressure drop have agreed with the pressure drop measurements to within $\pm 30\%$.

(4) All the four proposed correlations are nearly equally successful for the experimental pressure drop data of pure R-12 as well as that of the binary mixtures.

(5) The drawback of these correlations is that they do not allow extrapolation of the pressure drops to high concentrations of R-13, say above about 25%.

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APPENDIX

Table A1. Pressure drop data

Mass Velocity kg/m²s	Heat Flux W/m²	Tube 1 Inlet Vapour Quality	Tube 1 Outlet Vapour Quality	Tube 1 Pressure Drop Bar	Tube 2 Inlet Vapour Quality	Tube 2 Outlet Vapour Quality	Tube 2 Pressure Drop Bar
1	2	3	4	5	6	7	8
PURE R-12							
234	5000	0.12	0.26	0.0210	0.27	0.40	0.0412
234	5000	0.39	0.52	0.0841	0.54	0.67	0.0823
234	9000	0.12	0.35	0.0420	0.37	0.61	0.0823
234	9000	0.37	0.61	0.0420	0.63	0.87	0.1853
234	13000	0.12	0.47	0.0210	0.50	0.85	0.0823
234	17000	0.09	0.51	0.0841	0.51	0.92	0.1235
338	5000	0.16	0.26	0.0210	0.27	0.36	0.0206
338	5000	0.30	0.40	0.0420	0.41	0.50	0.0823
338	5000	0.38	0.48	0.0630	0.49	0.59	0.1029
338	9000	0.15	0.31	0.0841	0.33	0.49	0.1235
338	9000	0.38	0.56	0.1051	0.57	0.74	0.1441
338	13000	0.13	0.37	0.0526	0.39	0.64	0.1441
338	13000	0.39	0.65	0.1261	0.66	0.91	0.0720
338	17000	0.12	0.44	0.0736	0.46	0.77	0.1441
454	5000	0.17	0.24	0.0315	0.25	0.32	0.0412
454	5000	0.26	0.34	0.1156	0.35	0.42	0.1029
454	5000	0.32	0.40	0.1156	0.41	0.48	0.1647
454	9000	0.15	0.28	0.0526	0.29	0.41	0.1029
454	9000	0.30	0.44	0.1787	0.45	0.58	0.1853
454	9000	0.37	0.50	0.1366	0.52	0.65	0.1853
454	13000	0.14	0.33	0.0946	0.35	0.53	0.1235
454	13000	0.30	0.49	0.1892	0.51	0.69	0.2264
454	17000	0.14	0.38	0.1472	0.40	0.63	0.1235
454	17000	0.29	0.54	0.1892	0.56	0.80	0.2059
95% R-12, 5% R-13 MIXTURE							
234	5000	0.17	0.28	0.021	0.29	0.40	0.0618
235	5000	0.37	0.49	0.0315	0.50	0.61	0.1029
235	9000	0.17	0.37	0.0315	0.39	0.59	0.0823
234	9000	0.41	0.62	0.0736	0.64	0.85	0.1235
235	13000	0.18	0.49	0.042	0.52	0.84	0.1029
341	5000	0.16	0.23	0.0420	0.24	0.31	0.1029
339	5000	0.33	0.40	0.0946	0.41	0.48	0.1338
339	5000	0.37	0.45	0.1051	0.46	0.54	0.1544
339	9000	0.18	0.32	0.0631	0.33	0.47	0.1029
339	9000	0.38	0.52	0.1156	0.54	0.68	0.1853
338	13000	0.18	0.39	0.0315	0.41	0.63	0.1750
339	13000	0.39	0.61	0.1261	0.63	0.85	0.1853
340	17000	0.18	0.46	0.0841	0.48	0.76	0.1853
449	5000	0.19	0.24	0.0841	0.25	0.30	0.1441
454	5000	0.30	0.36	0.1366	0.38	0.44	0.2059
454	5000	0.32	0.38	0.1051	0.39	0.45	0.2573
447	9000	0.18	0.29	0.0946	0.30	0.41	0.1853
454	9000	0.33	0.44	0.1577	0.45	0.56	0.2470
451	9000*	0.32	0.50	0.1685	0.52	0.64	0.2676
454	13000	0.18	0.34	0.1051	0.36	0.52	0.2367
451	13000	0.31	0.49	0.1787	0.51	0.69	0.2779
451	17000	0.20	0.41	0.1261	0.43	0.63	0.2573
458	17000	0.34	0.55	0.1787	0.57	0.78	0.2779
90% R-12, 10% R-13 MIXTURE							
229	5000	0.23	0.33	0.0420	0.34	0.45	0.0720
230	5000	0.45	0.56	0.0841	0.57	0.68	0.1132
229	9000	0.23	0.44	0.0420	0.45	0.66	0.1029
229	9000	0.48	0.69	0.0841	0.70	0.91	0.1235
230	13000	0.23	0.54	0.0631	0.57	0.88	0.1132
330	5000	0.22	0.29	0.0631	0.30	0.37	0.1132
323	5000	0.31	0.40	0.0210	0.41	0.50	0.1544
327	5000	0.42	0.50	0.1051	0.51	0.59	0.1750
320	9000	0.24	0.38	0.0841	0.39	0.53	0.1544
324	9000	0.45	0.60	0.1472	0.61	0.75	0.1750
321	13000	0.23	0.46	0.0631	0.48	0.70	0.1647
323	13000	0.45	0.67	0.1156	0.69	0.91	0.1853
325	17000	0.24	0.52	0.1156	0.55	0.83	0.1441

* Heat flux in tube 1 = 17000 W/m²

Table A1. (Contd)

Mass Velocity kg/m²S	Heat Flux W/m²	Tube 1			Tube 2		
		Inlet Vapour Quality	Outlet Vapour Quality	Pressure Drop Bar	Inlet Vapour Quality	Outlet Vapour Quality	Pressure Drop Bar
1	2	3	4	5	6	7	8
90% R-12, 10% R-13 MIXTURE							
435	5000	0.24	0.29	0.0841	0.29	0.34	0.1647
433	5000	0.31	0.37	0.1472	0.38	0.44	0.2161
434	5000	0.38	0.44	0.1787	0.45	0.51	0.2264
430	9000	0.24	0.34	0.1261	0.35	0.46	0.2161
428	9000*	0.34	0.50	0.1892	0.51	0.63	0.2470
418	9000*	0.38	0.57	0.1997	0.59	0.73	0.2676
416	13000	0.22	0.39	0.1366	0.41	0.58	0.2367
413	13000	0.41	0.59	0.1997	0.60	0.78	0.2882
443	17000	0.22	0.42	0.1261	0.44	0.65	0.2676
416	17000	0.41	0.64	0.1997	0.66	0.88	0.2470
* Heat flux in tube 1 = 17000 W/m²							
85% R-12, 15% R-13 MIXTURE							
231	5000	0.22	0.32	0.0631	0.33	0.43	0.0720
231	5000	0.42	0.53	0.0631	0.54	0.66	0.1235
231	9000	0.22	0.42	0.0420	0.44	0.64	0.1029
231	9000	0.46	0.67	0.0841	0.69	0.90	0.1029
231	13000	0.23	0.53	0.0631	0.56	0.86	0.1338
331	5000	0.23	0.30	0.0736	0.31	0.38	0.1235
336	5000	0.34	0.42	0.1051	0.43	0.52	0.1441
336	5000	0.41	0.48	0.1261	0.49	0.56	0.1750
336	9000	0.22	0.35	0.0631	0.36	0.50	0.1235
335	9000	0.44	0.58	0.1051	0.60	0.75	0.1956
335	13000	0.22	0.43	0.0841	0.45	0.66	0.1647
332	13000	0.41	0.64	0.0841	0.66	0.89	0.2059
333	17000	0.23	0.50	0.0841	0.53	0.80	0.2059
451	5000	0.21	0.27	0.2312	0.27	0.33	0.1647
439	5000	0.32	0.39	0.1892	0.40	0.46	0.1853
444	5000	0.38	0.45	0.1682	0.46	0.52	0.2059
455	9000	0.23	0.34	0.0841	0.35	0.46	0.2676
456	9000*	0.36	0.46	0.1892	0.47	0.58	0.2264
457	9000*	0.33	0.53	0.1682	0.55	0.67	0.2676
454	13000	0.28	0.38	0.2102	0.39	0.54	0.1853
454	13000	0.37	0.53	0.2102	0.54	0.70	0.2059
457	17000	0.23	0.42	0.2713	0.44	0.63	0.2059
456	17000	0.35	0.57	0.1682	0.59	0.80	0.2470
* Heat flux in tube 1 = 17000 W/m²							
80% R-12, 20% R-13 MIXTURE							
223	5000	0.29	0.39	0.0526	0.40	0.50	0.1029
224	5000	0.50	0.61	0.0736	0.62	0.73	0.1441
224	9000	0.28	0.49	0.0631	0.50	0.71	0.1338
224	9000	0.53	0.74	0.0946	0.76	0.97	0.1441
225	13000	0.28	0.59	0.0631	0.62	0.93	0.1544
309	5000	0.28	0.36	0.0841	0.36	0.44	0.1441
306	5000	0.44	0.52	0.0736	0.53	0.61	0.1853
307	5000	0.50	0.59	0.1051	0.60	0.68	0.2059
310	9000	0.29	0.43	0.1156	0.45	0.59	0.2059
309	9000	0.52	0.67	0.1787	0.69	0.84	0.2161
321	13000	0.28	0.50	0.1051	0.51	0.73	0.1956
307	13000	0.53	0.76	0.1472	0.77	0.99	0.1647
314	17000	0.29	0.58	0.1261	0.60	0.89	0.2264
404	5000	0.29	0.34	0.1261	0.35	0.40	0.1647
394	5000	0.39	0.46	0.1682	0.46	0.53	0.2161
392	5000	0.45	0.52	0.1787	0.52	0.59	0.2367
388	9000	0.30	0.41	0.1472	0.42	0.53	0.2264
415	9000*	0.46	0.56	0.1787	0.57	0.68	0.2779
409	9000*	0.41	0.61	0.1997	0.63	0.75	0.3191
408	13000	0.28	0.45	0.1472	0.47	0.64	0.2573
395	13000	0.47	0.64	0.2102	0.66	0.84	0.2779
422	17000	0.29	0.49	0.1682	0.51	0.72	0.2676
436	17000	0.43	0.64	0.2102	0.66	0.87	0.2470

* Heat flux in tube 1 = 17000 W/m²