

HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF EVAPORATING CARBON DIOXIDE IN MICROCHANNEL TUBES

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ABSTRACT

This paper provides evaporation heat transfer and pressure drop data for carbon dioxide (CO₂, R-744) in a flat, multiport extruded (MPE) aluminium tube. The tube had 25 circular ports/channels with 0.79 mm inner diameter. Data are presented for the following range of variables: Mass flux 200–600 kg/(m²s), evaporation temperature 0–20°C, and heat flux 5–20 kW/m². The overall heat transfer coefficient was measured for water-to-CO₂ heat transfer, and the CO₂-side heat transfer coefficient was arrived at by using a modified Wilson plot technique to calibrate an equation for the water-side heat transfer coefficient. The measured CO₂ evaporation heat transfer coefficients lay in the range between 5,000 and 15,000 W/(m²K). From a certain vapour fraction upwards, the heat transfer coefficient dropped significantly. This effect was most pronounced for high mass fluxes and evaporation temperatures. An accompanying paper (Pettersen et al., 2000a) presents the results from supercritical-pressure heat transfer and pressure-drop experiments

1. NOMENCLATURE

\dot{m}	Mass flux (kg/(m ² s))	α	Heat transfer coefficient (W/(m ² K))
p	Pressure (Pa)	Δ	Difference (–)
x	Vapour fraction (–)	σ	Deviation (–)

2. INTRODUCTION

Since the re-introduction of carbon dioxide (CO₂, R-744) as a working fluid (Lorentzen and Pettersen, 1992), it has been shown to have several favourable applications, for instance in heat pump water heaters (Nekså et al., 1998), laundry dryers (Schmidt et al., 1998), mobile air conditioners and heat pumps (Hafner et al., 1998), residential air conditioners (Pettersen et al., 1997), as well as in military environmental control units (Patil and Manzione, 1999).

The motivation of this study was to obtain heat transfer and pressure drop data for heat exchangers to be used for the transcritical CO₂ process, as well as to examine the applicability of common engineering correlations. This paper focuses on heat transfer and pressure drop during evaporation in a multiport extruded (MPE) aluminium tube. A cross-sectional sketch of the tested tube is shown in Figure 1. The dimensions are in millimetres. Heat exchangers built from MPE tubes are capable of withstanding high pressure while giving compact and thermally efficient design, and are therefore of great interest for CO₂ systems.

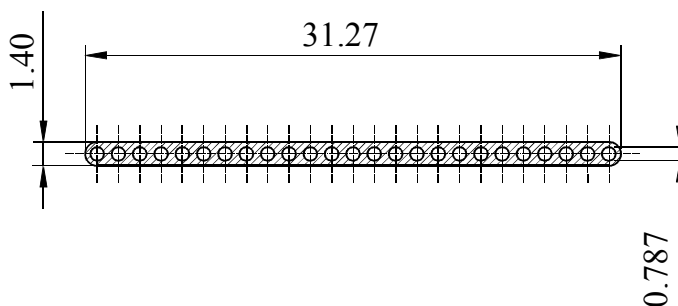


Figure 1: Cross-section of the multiport extruded (MPE) tube (Dimensions in mm)

As most two-phase engineering correlations for heat transfer and pressure drop were developed mainly for (H)CFCs, they cannot necessarily be expected to give accurate results for CO₂. The reasons for this are the differences in thermophysical and thermodynamic properties, such as thermal conductivity and specific heat capacity. In particular, CO₂ air conditioning and heat pump systems usually operate close to the critical point (31.1 °C, 73.8 bar). At saturation temperatures in the range between, say, 0 °C and 20 °C, the density ratio between liquid and vapour is significantly lower for CO₂ than for commonly used refrigerants. At 0 °C, the density ratio is 9.5 for CO₂ and 89 for R-134a. Furthermore, the surface tension of CO₂ in this temperature range is very low.

Only a few studies of two-phase heat transfer for flow of CO₂ are openly available. Bredesen, et al. (1997) measured heat transfer and pressure drop during evaporation of CO₂ in an ID 7 mm round tube. They found, quite interestingly, that the heat transfer coefficient showed a decreasing trend for increasing vapour fraction at high evaporation temperatures. Rieberer (1998) measured heat transfer and pressure drop during evaporation of CO₂ in 10 mm inner diameter plain round tubes. Since the test apparatus was a laboratory heat pump, lubricant was present in the system. The heat transfer coefficients found were significantly lower than those of Bredesen et al. This was explained by the presence of lubricant, as well as the much higher heat fluxes in the tests by Rieberer.

3. EXPERIMENTAL RESULTS AND DISCUSSION

The test facilities are briefly outlined in an accompanying paper (Pettersen et al., 2000a). The test tube was heated by a water jacket, and a modified Wilson-plot technique was used to deduct the CO₂ heat transfer coefficient from the measured overall heat transfer coefficient. Details regarding the test rig, calibration procedure and data acquisition are available in an open technical report (Pettersen et al., 2000b).

3.1 Heat Transfer

Tests were carried out at evaporation temperatures of 0°C, 10°C, and 20°C, and at mass fluxes 200, 300, and 400 kg/(m²s), Figure 2. For all test runs in this series, the heat flux was 10 kW/m². The legend “t0/q10/m300” means that the data point was taken at an evaporation temperature of 0 °C, a heat flux of 10 kW/m², and a mass flux of 300 kg/(m²s). The horizontal lines are *not* error bars. Instead, they indicate the change in CO₂ vapour fraction through the test tube. The data point indicates the mean value with respect to the length of the test section.

From the first diagram (0°C evaporation temperature) one can see that for low vapour fractions, neither the mass flux nor the vapour fraction had any influence on the heat transfer coefficient. At all velocities, the measured heat transfer coefficient was about 9,000 W/(m²K). However, at high vapour fractions and a mass flux of 400 kg/(m²s), the heat transfer coefficient was significantly reduced.

A similar behaviour can be seen at 10°C (second diagram); at low vapour fractions the heat transfer coefficient was about 10,500 W/(m²K) and independent of the mass flux. In case of a mass flux of 200 kg/(m²s), this value was measured at all vapour fractions. But already at 300 kg/(m²s), a slight drop can be seen at high vapour fractions. At 20°C (third diagram) and low vapour fractions, a value of about 15,000 W/(m²K) was measured. The drop, however, occurred at lower vapour fractions. Indeed, at a vapour fraction of about 0.5, the measured heat transfer coefficient dropped from 15,000 W/(m²K) for a mass flux of 200 kg/(m²s) to 13,000 W/(m²K) for a mass flux of 400 kg/(m²s).

The effect of mass flux on heat transfer coefficient is further illustrated by the results shown in Figure 3. During these tests, the vapour fractions at the inlet and outlet of the test section were kept constant, while the mass flux was changed. This was realised by varying the heat flux. The heat fluxes required to keep the inlet and outlet vapour fractions constant were 10 to 26 kW/m² for the series with an inlet vapour fraction of $x_{in} = 0.3$ and an outlet vapour fraction of $x_{out} = 0.92$ (lower curve), and 14 to 42 kW/m² for the series with $x_{in} = 0$ and $x_{out} = 0.9$ (upper curve).

As can be seen, the influence of the mass flux was significant. In both tests, an increase from 200 to 300 kg/(m²s) resulted in a higher heat transfer coefficient. At higher velocities, a significant decrease was observed. The higher heat transfer coefficient in case of the lower inlet vapour fraction was thought to be due to the poor heat transfer at high vapour fraction (see Figure 2). An additional explanation might be the influence of heat flux (see below).

A possible explanation for the decreasing heat transfer coefficient with increasing mass flux is a dry-out phenomenon, in which the increased vapour velocity causes the liquid film at the wall to break down and become entrained as droplets in the gas core of the flow. VDI (1994) gives a calculation model for the so-called critical vapour fraction (x_{crit}) where dry-out occurs. Values calculated according to this model agreed fairly well to the ones in Figure 2, where the heat transfer coefficient can be seen to drop at a certain vapour fraction.

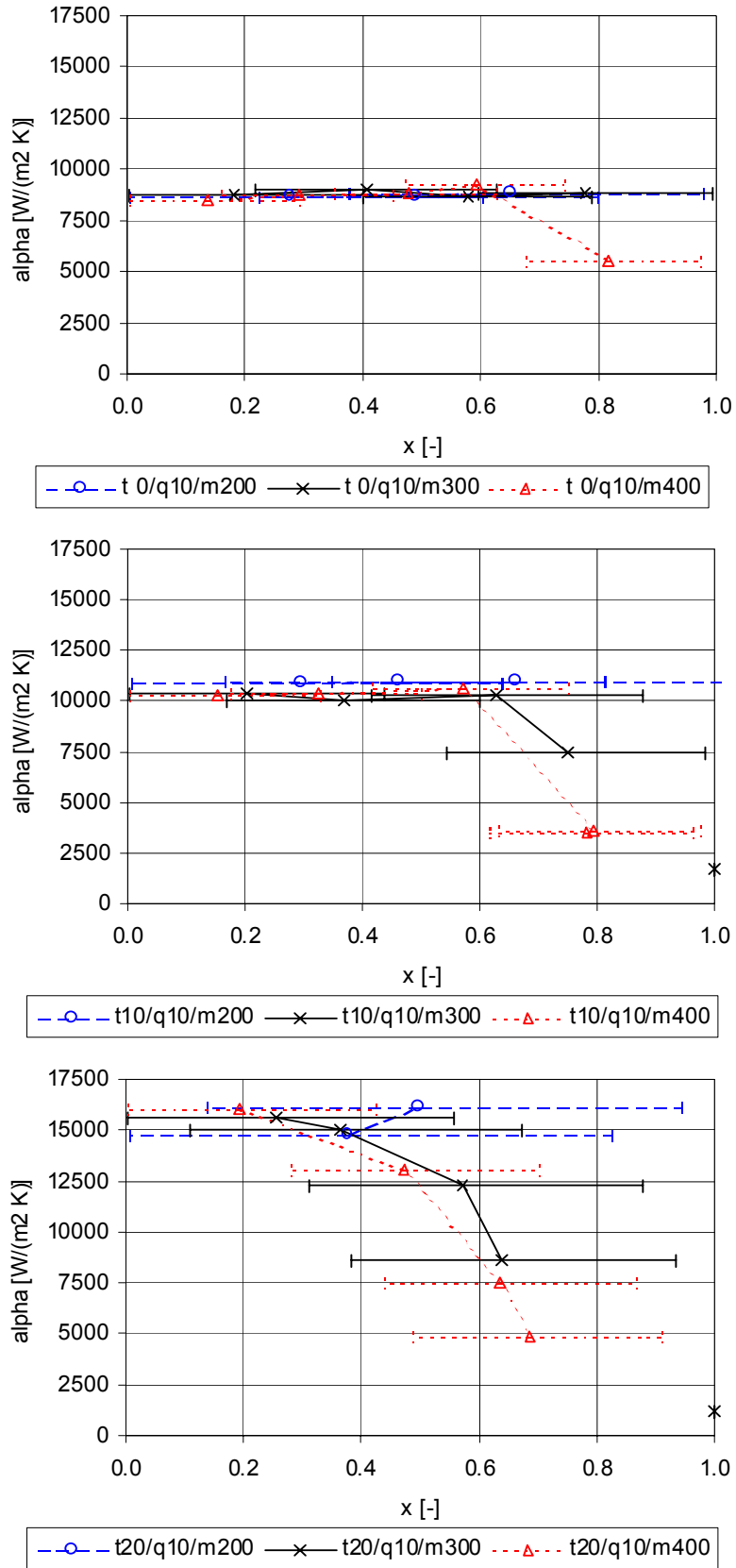


Figure 2: Influence of mass flux on the heat transfer coefficient at different evaporation temperatures

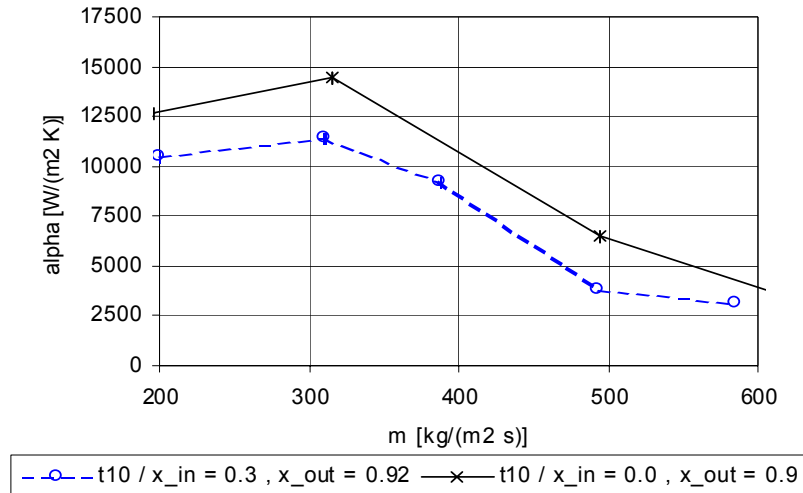


Figure 3: Heat transfer coefficient as a function of mass flux. Constant vapour fraction at inlet and outlet.

To investigate the influence of heat flux, heat transfer data were taken for heat fluxes ranging from 5 to 15 kW/m². The mass flux was kept constant at 300 kg/(m²s), while saturation temperature was varied. The experimental results are shown in Figure 4.

At 0°C saturation temperature (first diagram), the heat transfer coefficient increased slightly with heat flux. At 10°C (second diagram), the enhancement due to a higher heat flux was larger. At high vapour fractions, a direct comparison was not possible because of the different range of vapour fraction of each experiment depending on the heat flux. At 20°C, the change in vapour fraction was large in case of a heat flux of 15 kW/m². However, the lower heat transfer coefficients at high vapour fractions may be a confirmation that due to the higher heat flux, dry-out occurs at a lower vapour fraction. This would mean that the enhancement due to an increased heat flux does not compensate for the drop due to dry-out occurring at a lower vapour fraction.

As already mentioned, evaporation temperature influences the heat transfer characteristics of CO₂. Figure 5 compares measured heat transfer coefficients at constant heat flux and mass flux, and varying saturation temperature. As may be observed, the heat transfer coefficient was significantly higher for the highest saturation temperature at low vapour fractions. While the heat transfer coefficient was about 9,000 W/(m²K) at 0°C, it was higher than 15,000 W/(m²K) at 20°C.

Six common evaporation heat transfer correlations were chosen for a comparison with the experimental data. The average and mean deviations (Pettersen et al., 2000a) are listed in Table 1. As may be observed, the discrepancy between the calculated and measured heat transfer coefficients was significant. This was particularly so at high vapour fractions, where all models overpredicted the measured values. The bad fit might be caused by correlations that were not developed for CO₂, but also the small diameter of the MPE tube may be outside the range of the models. Rieberer (1998) found that the VDI (1994) correlation fitted quite well to the 7 mm tube data published by Bredesen, et al. (1997), for mass fluxes between 200 and 400 kg/(m²s). For the present tube, however, the VDI correlation gave completely wrong results, mainly due to the small diameter of 0.79 mm.

3.2 Pressure drop

The measured pressure drops were compared to values calculated using the following correlations: i) The Friedel (1979) correlation using the Premoli approach for the void fraction, as recommended by Thome (1997); ii) The Fuchs-Neraas correlation (Fuchs (1975), Neraas (1993)); iii) The VDI (1994) correlation; and finally iv) using a single-phase model, i.e., calculating the mean density for the two-phase flow assuming homogeneous flow and then calculating the frictional pressure drop from the Colebrook curve-fit to the Moody diagram. Thome's recommendation gave the best results. The mean deviation was nevertheless 22 %. The other correlations gave mean

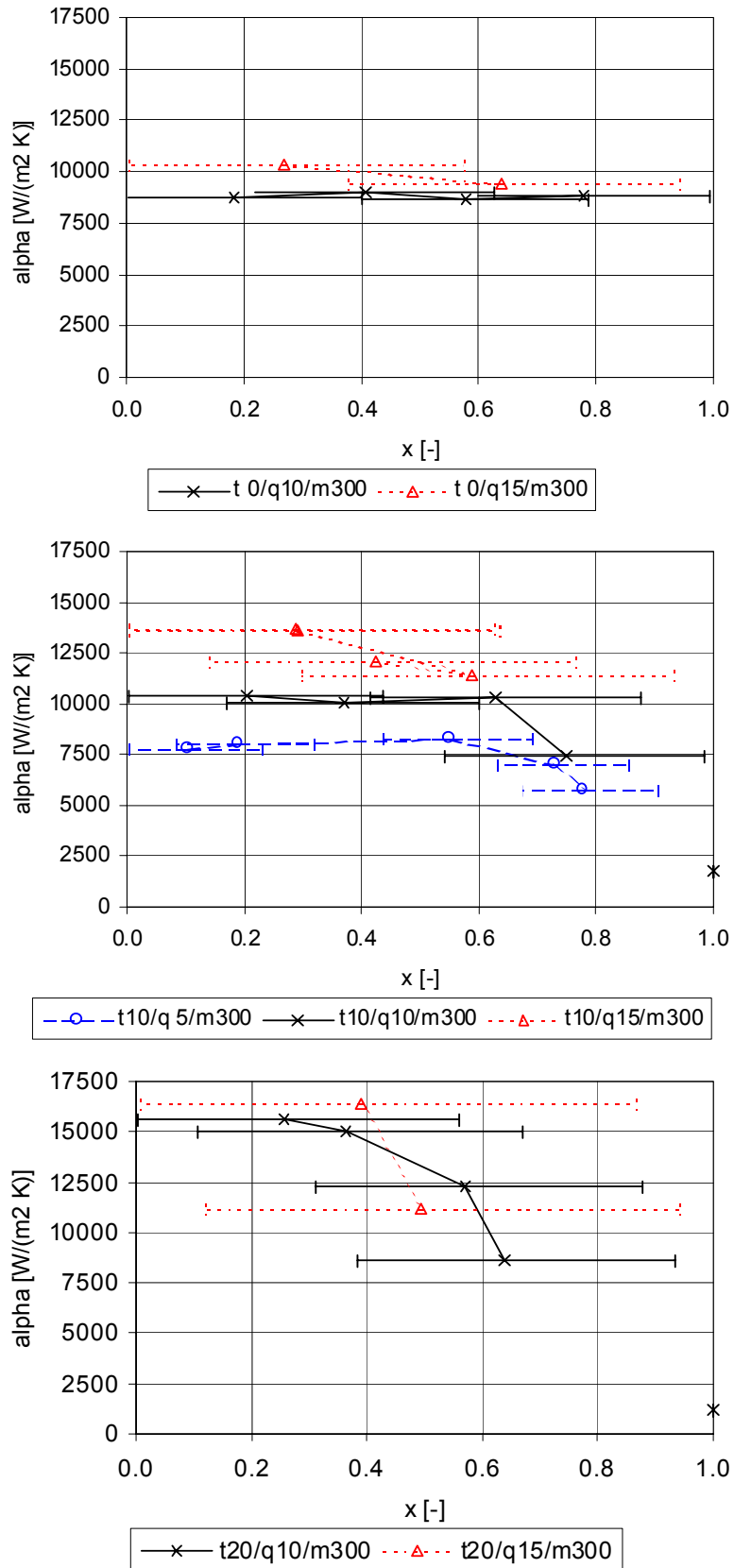


Figure 4: Influence of heat flux on the heat transfer coefficient at different evaporation temperatures

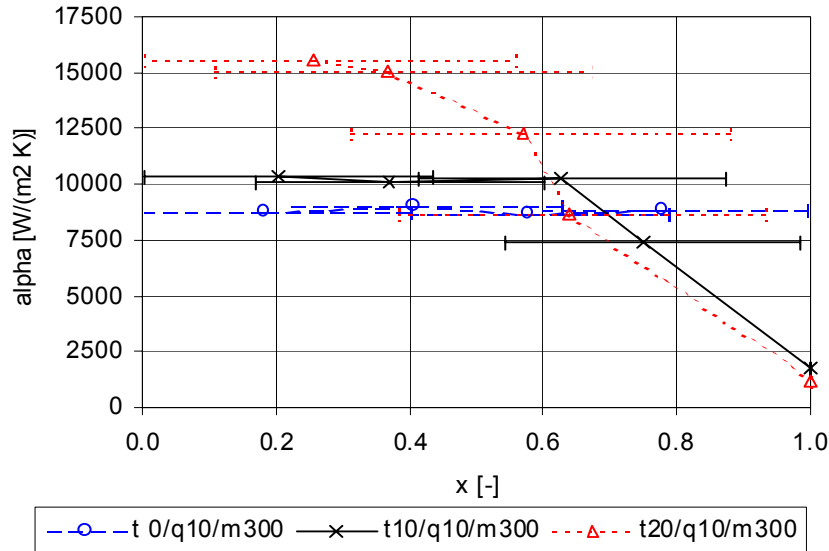


Figure 5: Influence of evaporation temperature on the heat transfer coefficient.

Table 1: Deviation between measured and calculated heat transfer coefficients

Model	σ_{avg} (%)	σ_{mean} (%)
Slipcevic (1987)	60	68
Kandlikar (1990)	49	55
Kattan (1996)	91	92
Shah (1982)	4	61
Gungor and Winterton (1986)	125	56
Gungor and Winterton (1987)	14	117

deviations in the range between 37 and 57 %. In the following figures, some of the measured data are compared with values calculated according to Thome (1997).

The influence of mass flux on the pressure drop at a saturation temperature of 10°C can be seen in Figure 6. While the measurements at 200 kg/(m²s) showed a moderate pressure drop for all vapour fractions, the pressure drop at 400 kg/(m²s) increased significantly with vapour fraction. As one can see, the calculated values did not correlate very well to the experimental data; especially at low vapour fractions the deviation was high.

Figure 7 shows the influence of saturation temperature on the pressure drop. The mass flux was constant at 300 kg/(m²s) during these tests. With decreasing saturation temperature, the vapour volume increases, and this is the main reason for the remarkably higher pressure drop at 0°C compared to that at 20°C. Regarding the deviation between measurement and calculation, one can see that the model underestimated the pressure drop in case of low temperatures.

4. CONCLUSION

In the transcritical cycle, the working fluid is cooled at supercritical pressure, where the influence of the critical point on the properties is quite large, and it evaporates at temperatures close to the critical temperature, where the high saturation pressure results in high vapour densities. Furthermore, the surface tension of CO₂ is very low. These factors lead to conditions in CO₂ equipment differing considerably from those of conventional refrigerants.

In evaporation heat transfer, the effect of the low density ratio between liquid and vapour may significantly affect the two-phase flow. In case of high mass fluxes, the experiments showed a strongly decreasing heat transfer

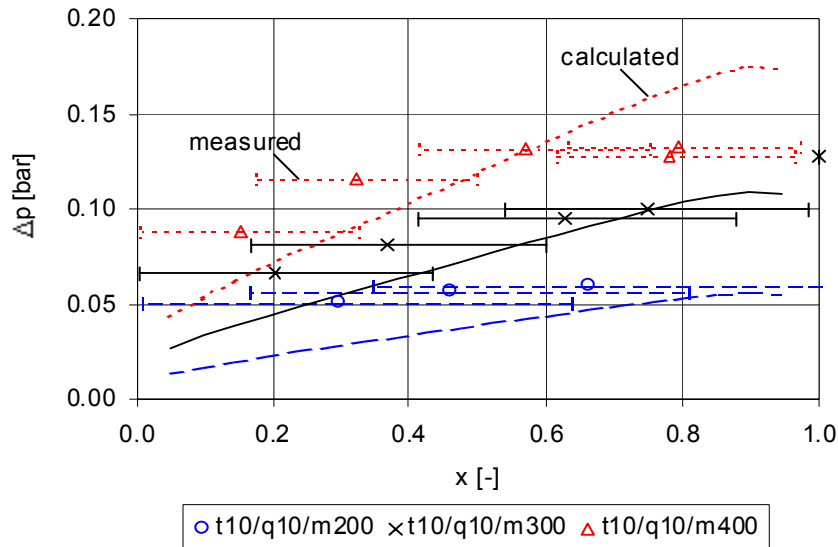


Figure 6: Influence of mass flux on the two-phase pressure drop ($\Delta l = 503$ mm)

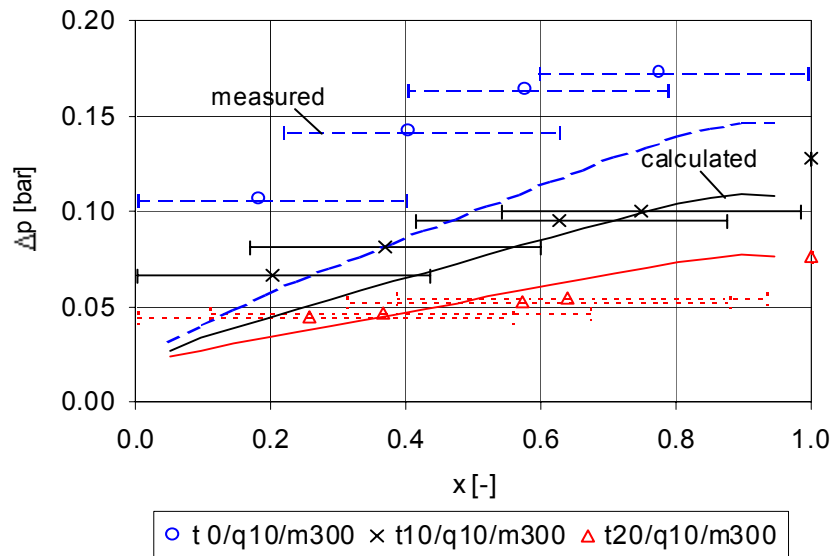


Figure 7: Influence of saturation temperature on the two-phase pressure drop ($\Delta l = 503$ mm)

coefficient from a certain vapour fraction upwards. In the temperature region investigated, the ‘critical’ mass flux was about $300 \text{ kg}/(\text{m}^2\text{s})$, with some variation depending on the saturation temperature and the heat flux. A comparison with a calculation model for dry-out showed that this drop could be expected. The calculated values for the critical vapour fraction corresponded well to the observations. This very important fact has to be considered when designing evaporators.

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5. REFERENCES

- Bredesen, A.M., Hafner, A., Pettersen, J., Neksa, P. and Aflekt, K., 1997, Heat transfer and pressure drop for in-tube evaporation of CO₂, *In: R. Radermacher, Proceedings, International Conference on Heat Transfer Issues in 'Natural' Refrigerants*, IIF-IIR, College Park, Maryland, USA, p. 1-15.
- Friedel, L., 1979, Improved friction pressure drop correlations for horizontal and vertical two phase pipe flow, *In: Proceedings, European Two Phase Flow Group Meeting*, Ispra, Italy.
- Fuchs, P.H., 1975, Trykkfall og varmeovergang ved strømming av fordampende væske i horisontale rør og bend, Doctoral thesis, Department of Refrigeration Engineering, Norwegian Institute of Technology, NTH, Trondheim, Norway.
- Gungor, K.E. and Winterton, R.H.S., 1986, A general correlation for flow boiling in tubes and annuli, *International Journal of Heat and Mass Transfer*, vol. 29, no. 3: p. 351-358.
- Gungor, K.E. and Winterton, R.H.S., 1987, Simplified general correlation for saturated flow boiling and comparisons of correlations with data, *Chemical Engineering Research & Design*, vol. 65, no. 2: p. 148-156.
- Hafner, A., Pettersen, J., Skaugen, G. and Neksa, P., 1998, An automobile HVAC system with CO₂ as the refrigerant, *In: Proceedings, Natural Working Fluids '98 -- IIR Gustav Lorentzen Conference*, IIF-IIR, Oslo, Norway, p. 335-345.
- Lorentzen, G. and Pettersen, J., 1992, New Possibilities for Non-CFC Refrigeration, *Proc. IIR International Symposium on Refrigeration, Energy and Environment*, Trondheim, Norway, June 22-24, p.147-163..
- Neksa, P., Rekstad, H., Zakeri, G.R. and Schiefloe, P.A., 1998, CO₂-heat pump water heater: characteristics, system design and experimental results, *International Journal of Refrigeration*, vol. 21, no. 3: p. 172-179.
- Neraas, B.O., 1993, Condensation of Hydrocarbon Mixtures in Coil-Wound LNG Heat Exchangers. Tube-Side Heat Transfer and Pressure Drop, Doctoral thesis, Department of Refrigeration Engineering, Norwegian Institute of Technology, NTH, Trondheim, Norway.
- Patil, A.S. and Manzione, J.A., 1999, US Army CO₂ Development Program, *In: 20th International Congress of Refrigeration*, IIF-IIR, Sydney, Australia.
- Pettersen, J., Aarli, R., Neksa, P., Skaugen, G. and Aflekt, K., 1997, A comparative evaluation of CO₂ and R-22 residential air-conditioning systems in a Japanese climate, *In: IEA/IIR Workshop on CO₂ Technologies in Refrigeration, Heat Pump and Air Conditioning Systems*, Trondheim, Norway.
- Pettersen, J., Rieberer, R. and Munkejord, S.T., 2000a, Heat transfer and pressure drop characteristics of supercritical carbon dioxide in microchannel tubes under cooling, *4th IIR Gustav Lorentzen Conference on Natural Working Fluids*, Purdue, IN, USA, July
- Pettersen, J., Rieberer, R. and Munkejord, S.T., 2000b, Heat transfer and pressure drop for flow of supercritical and subcritical CO₂ in microchannel tubes, Technical report, TR A5127, SINTEF Energy Research, Refrigeration and Air Conditioning, Trondheim, Norway.
- Rieberer, R., 1998, CO₂ as Working Fluid for Heat Pumps, Doctoral thesis, Institute of Thermal Engineering, Graz University of Technology, Austria
- Schmidt, E.L., Klöcker, K., Flacke, N. and Steimle, F., 1998, Applying the transcritical CO₂ process to a drying heat pump, *International Journal of Refrigeration*, vol. 21 Perm CO2 1-3, no. 3: p. 202-211.
- Shah, M.M., 1982, Chart correlation for saturated boiling heat transfer: Equations and further study, *ASHRAE Transactions*, vol. 88, Part 1: p. 185-196.
- Thome, J.R., 1997, Refrigeration Heat Transfer, Course notes, GRETh/CEA, Grenoble, France.
- VDI, 1994, VDI-Wärmeatlas, Düsseldorf, Germany, VDI-Verlag.

CARACTÉRISTIQUES DES TRANSFERTS THERMIQUES ET PERTES DE CHARGE POUR LE DIOXIDE DE CARBONE DANS LES TUBES MICROCONDUITS SOUS EVAPORATION.

Cet article fournit les données en transfert thermique et pertes de charge pour l'évaporation du dioxyde de carbone (CO₂, R-744) dans un tube en aluminium extrudé (MPE) plat et à multi-conduits. Le tube dispose de 25 conduits circulaires de 0.79 mm de diamètre intérieur. Les données sont présentées pour les variables suivantes : flux massique de 200 à 600 kg/(m²s), température d'évaporation de 0 à 20 °C et flux thermique de 5,000 à 20,000 W/m². Le coefficient global de transfert thermique a été mesuré pour un transfert thermique de l'eau vers le CO₂, et le coefficient de transfert thermique côté CO₂ a été déterminé par l'utilisation d'une technique modifiée de traçage de Wilson et le calibrage d'une équation pour le coefficient de transfert thermique côté eau. Les coefficients mesurés de transfert thermique d'évaporation du CO₂ sont compris entre 5 et 15 kW/(m²K). Pour une certaine fraction de vapeur supérieure, le coefficient de transfert thermique chute de manière significative. Cet effet est plus prononcé pour des flux massiques et des températures d'évaporation élevés. Un article d'accompagnement (Pettersen et al., 2000a) présente les résultats des expériences de transfert thermique et perte de charge pour des pressions super-critiques.