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ABSTRACT

The expansion device is an important component in refrigerating systems. For automotive and residential air conditioners short tube orifices are widely used, because of its low cost, high reliability and easy handling.

In the open literature there are a lot of models for orifices available, but most of them focus on water or commonly used refrigerants like R22, R134a, R410A, but they do not consider R744 (CO_2), i.e. only a few validated models for orifices for CO_2 systems exist.

This paper presents experimental data and a model for predicting the refrigerant flow rate of CO_2 through short tube orifices to be used as expansion device in refrigerant cycles. For this, different tube geometries have been studied, i.e. different tube lengths, bore diameters and the effect of inlet chamfering. The data were analyzed for operating conditions which typically occur in automotive air-conditioning systems. For the inlet conditions the pressure was varied in a range of 75 to 130 bar and the temperature in a range of 15 to 40 °C, i.e. all inlet conditions were single phase.

The investigations show that for all operating conditions choked flow occurs, if the outlet condition from the orifice is in the two phase region. That means that the refrigerant mass flow rate is independent of the outlet pressure, but strongly depending on the inlet pressure. Of course, the mass flow rate increases with increasing upstream pressure. The inlet temperature plays an important role for the mass flow which increases with decreasing temperature (at constant inlet pressure). The effect of inlet chamfering turned out to be in a range of about 5 to 10 %.

1. INTRODUCTION

One important component of a refrigerant circuit is the expansion device. A rather simple solution is an orifice tube: It is robust and reliable, easy to install and to replace as well as cheap compared to a controlled expansion valve. Due to its advantages the application of fixed orifices seems to be of interest for automotive air-conditioning system with CO_2 (possibly with a bypass which damps the pressure peaks), although a control of high pressure would have advantages for operating the system in an optimal operating point regarding the COP and the cooling capacity (Robin *et al.* 2003). Of course it is of special interest how great the impact of the high pressure control is on the system behavior. This can either be investigated by experiments or by means of simulations. The latter requires a reliable equation for the correlation of the mass flow through the orifice.

The single-phase flow through orifices is often used for the measurement of a fluid flow rate, since it can be described reliably by equations. In contrast to a single-phase flow a two-phase flow is much more complex and no "universal" equation can be found. In the open literature there are several models available for orifices, but most of them focus on water or "common" refrigerants like R22 (Kim and O'Neal, 1994b), R134a (e.g. Singh *et al.*, 2001), R410A (e.g. Kim *et al.*, 2005); but they do not consider R744 (CO_2). So far only a few validated models for orifices for CO_2 can be found. Therefore, several experimental investigations were conducted to develop a model for the flow of CO_2 through fixed orifices at conditions which typically occur during the operation of an automotive air-conditioning system.

2. EXPERIMENTAL SETUP

The experimental investigations were conducted with a test rig which was originally designed for the analysis of the entire refrigerant cycle of automotive air-conditioning systems. Therefore the range of the conditions which could be investigated was limited, but the greatest part of operating conditions of an automotive air-conditioning system can be covered. The influence of the oil fraction was not considered, because it was assumed that the oil separation rate is such high, that the oil flow has a negligible influence. However, it could not be ruled out, that the oil has an influence in some operating points, if the compressor has a high oil discharge. Further information about the test rig can be found in Martin *et al.* (2005).

The principal design of the tube orifices is shown in Figure 1. The tube orifices were made of brass with a length of 10 mm and 20 mm. The investigated bore diameters were 0.8 mm and 1.0 mm and the outer diameter was 20 mm. The holes have been eroded with an inaccuracy smaller than 0.01 mm. To implement the orifices into the test rig they have been soldered into a copper tube with a inner diameter of 10 mm. To determine the influence of a non sharp-edged inlet the orifices were provided with a chamfer at the inlet. We have to mark here, that this chamfer was not manufactured with high precision. The intention was just to see the order of magnitude of the influence of the inlet chamfering.

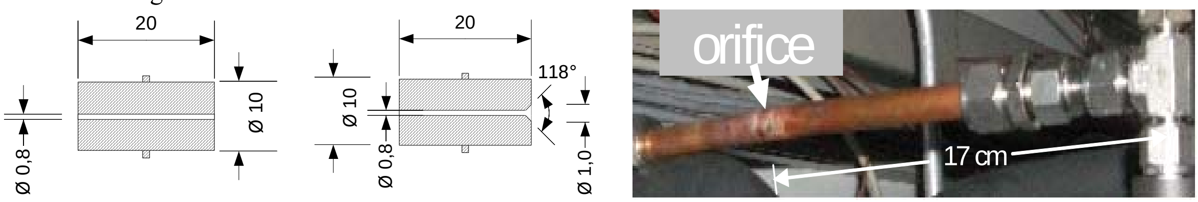

Figure 1: Design of the tube orifice, without and with chamfer at inlet (left) and implementation into test rig (right)

The test matrix was chosen according to the operating conditions of an automotive air-conditioning system. For illustration the inlet and outlet conditions are shown in a pressure/enthalpy-diagram as well as the inlet conditions in a pressure/temperature-diagram in Figure 2.

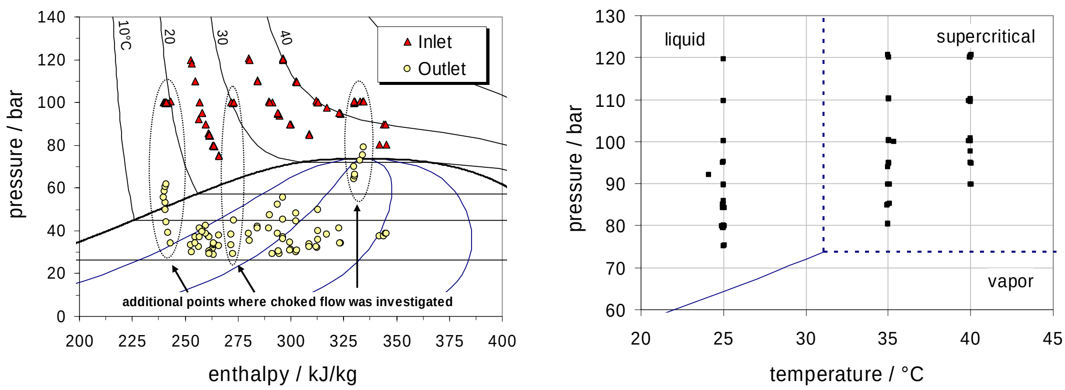

Figure 2: Investigated conditions in pressure/enthalpy diagram (left) and
in pressure/temperature diagram (right)

All inlet conditions were in the single-phase area and the outlet was always in the two-phase region, except for the test series at which choked flow was investigated in detail. The orifice inlet temperatures have been varied between 25°C and 40°C and the orifice inlet pressures between 75 bar and 120 bar. These experimental investigations are shown in chapter 4. Additionally to this some test series with constant inlet conditions and varying outlet pressures were conducted, to determine whether the flow is choked or not (see chapter 3). These tests were carried out with the tube orifice with the larger bore diameter of 1.0 mm.

3. INVESTIGATIONS OF CHOKED FLOW CONDITIONS

According to VDI-WA (1997) at constant inlet conditions the mass flow rate of a fluid through an orifice increases continuously with decreasing backpressure (see Figure 3). But if the backpressure is lower than the so called critical pressure p_{crit} (mark this is not the critical pressure of the fluid), the mass flow rate remains on a constant level, i.e. the flow then is choked. This maximum mass flow rate is called the critical mass flow rate $\dot{m}_{crit}$.

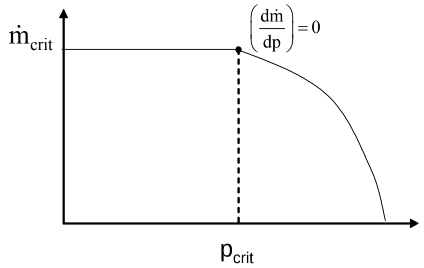

Figure 3: Trend of mass flow rate depending on backpressure (VDI-WA, 1997)

Since two phase flow is relatively complex no reliable equation to determine the critical mass flow rate exists which can be used for modeling an orifice that is used in a refrigerant cycle as a device to expand a fluid from high to low pressure. Investigation of Kim et al (1994a) with the refrigerant R134a show that the mass flow rate is nearly independent from the outlet pressure as long as the outlet pressure is below the saturation pressure corresponding to the inlet temperature, i.e. that the flow is choked if the outlet is in the two-phase region. However no extensive experimental investigations with CO2 as fluid can be found in open literature. Therefore some tests were conducted to evaluate whether the flow is choked within the investigated operating range or not. Note, in CO2 air conditioning systems the inlet pressure is commonly supercritical and thus the greatest part of the pressure loss occurs in the single-phase region.

Figure 4 shows some results of the tube orifice with a bore diameter of 1.0 mm where the inlet conditions (p_{in} = 100 bar and three different temperatures) were kept constant and the outlet pressure of the orifice (backpressure) was varied. It can clearly be seen that for the series with 100 bar and 30°C the mass flow is choked over the entire range.

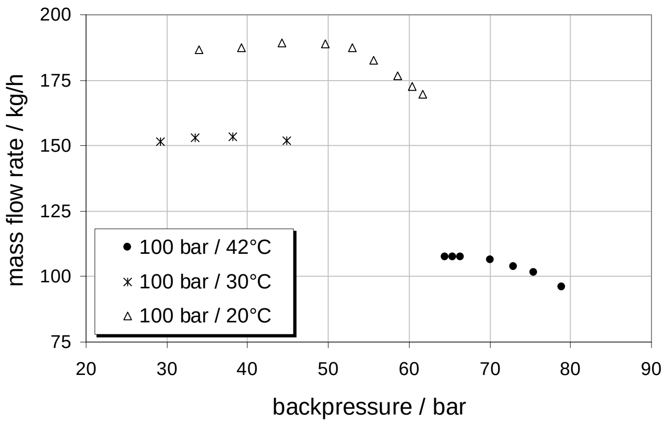

Figure 4: Dependence of mass flow rate on backpressure for tube orifice (D_{orf} = 1.0 mm)

For the test series with 100 bar and 20°C at the orifice inlet choked flow occurs if the pressure is below about 50 bar. This is approximately the saturation pressure of the inlet temperature; this would mean that the flow is choked if the state at the orifice outlet is into the two-phase region (compare Figure 2). This fact would also explain why the flow is choked for the test series at 30°C at the orifice inlet, since all outlet conditions were far in the two-phase region for this tests (compare Figure 2).

For an inlet temperature of 42°C no saturation pressure exists, since the temperature is above the critical temperature of CO_2 ($t_{crit,CO2} \approx 31°C$). The experimental results show that the mass flow is also choked for several conditions. The critical pressure p_{crit} at which choking occurs is in the range of 70 bar.

As a conclusion of the investigations of choked flow it can be stated that for all operating conditions which have been investigated for the application of the tube orifice and will be discussed in the next chapter, choked flow occurs, i.e. that the mass flow rate is independent from the orifice outlet pressure (backpressure). Therefore the backpressure was not considered within the following investigations.

4. EXPERIMENTAL INVESTIGATIONS

In order to investigate the different parameters which influence the mass flow rate several test series were conducted. Within theses test series the orifice inlet temperature was kept constant and the inlet pressure was varied. The outlet pressure adjusted itself due to the cycle behavior, but as shown before, the backpressure has no significant influence on the mass flow rate for the investigated conditions.

Figure 5 shows the dependence of the refrigerant mass flow from the high side pressure for different inlet temperatures and bore diameters. It can be seen that with a constant inlet temperature the mass flow rate increases with increasing high pressure. This is not surprising since the pressure difference and the density increase. If the inlet pressure is constant and the temperature is reduced, the mass flow rate rises due to the increasing density of the fluid. When comparing the tube orifices with the different bore diameters it can be noticed that the inclination of the connecting curves is not so high for the orifice with a diameter of 0.8 mm (Figure 5-right) compared to the orifice with a diameter of 1.0 mm (Figure 5-left). This means that the influence of the high pressure on the mass flow rate decreases at smaller bore diameters.

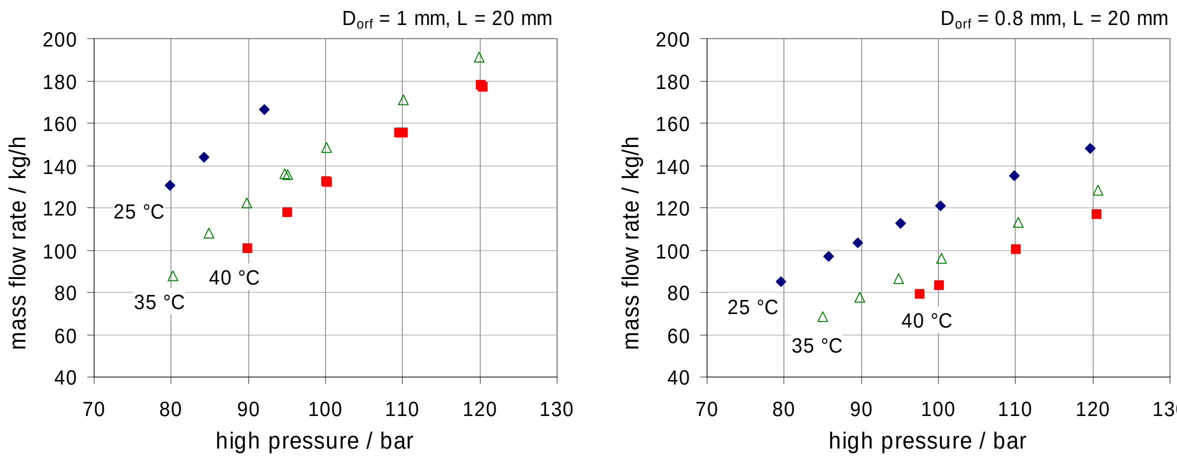

Figure 5: Influence of inlet temperature and pressure on mass flow rate
for bore diameters D_{orf} = 1.0 mm (left) and: D_{orf} = 0.8 mm (right)

The influence of the tube orifice length on the mass flow rate is shown in Figure 6 for a length of 20 and 10 mm. These tests were conducted with the orifice with a bore diameter of 1.0 mm at two different inlet temperatures and varying high pressure. It shows up here, that the mass flow rate of the orifice tube with a length of 20 mm is only slightly lower than for the tube orifice with a length of 10 mm. Hence it can be concluded that the length of the tube orifice plays an minor role compared to other parameters within the investigated operating conditions.

Figure 7 shows the effect of a chamfer at the inlet of the tube orifice in comparison to a tube orifice with a sharp-edged inlet. These tests were conducted with the tube orifice with a length of 10 mm and at an inlet temperature of 25°C. The impact of inlet chamfering on the mass flow rate is not negligible: For the tube orifice with a bore diameter of 0.8 mm the mass flow rate rises by about 5 % for all conditions, while it even increases by 10 % for the tube orifice with a bore diameter of 1.0 mm.

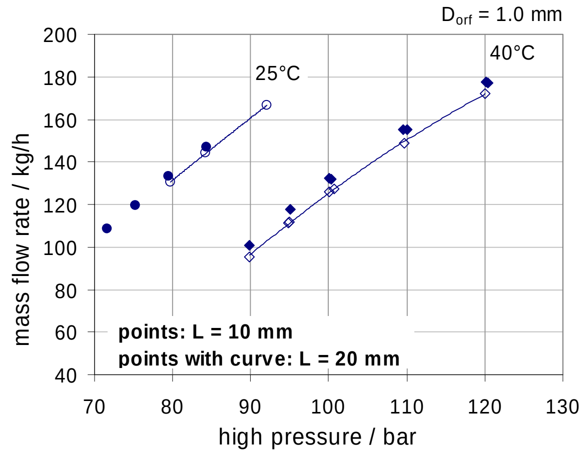

Figure 6: Influence of tube orifice length on mass flow rate

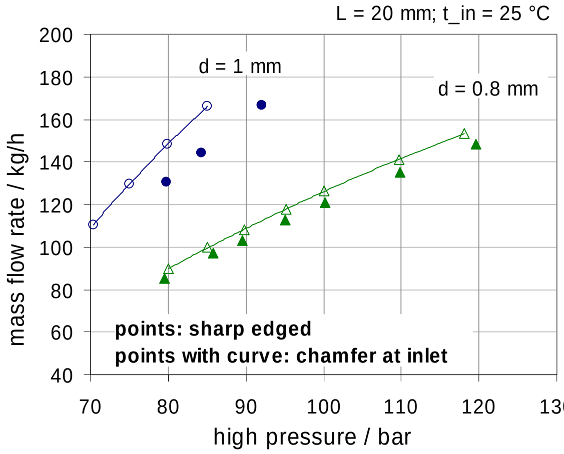

Figure 7: Influence of inlet chamfering on mass flow rate

5. EMPIRICAL MODEL

Within this project it has been tried to apply several models for the determination of the mass flow rate through the tube orifice from literature (e.g. Moody taken from VDI-WA, 1997), but with limited success, because of uncertainties of the predicted critical mass flow rate. Therefore, a model with dimensionless numbers was chosen. This basic approach was also chosen by Kim *et al.* (2005) and Payne *et al.* (2004) to model short tube orifices.

The parameters influencing the mass flow rate were selected and determined based on the experiments shown in the previous chapter. All these parameters are shown in Table 1. They could be described by the main dimensions length, mass, time and temperature. With these parameters five dimensionless numbers can be created (see Table 2).

Table 1: Variables used to produce non-dimensional groups

		SI – Unit
Mass flow rate	$\dot{m}$	$kg \cdot s^{-1}$
Inlet pressure	p_1	$kg \cdot m^{-1} \cdot s^{-2}$
Inlet temperature	T_1	K
Bore diameter of orifice	D_{orf}	m
Length of tube orifice	L	m
Inlet density	ρ_1	$kg \cdot m^{-3}$
Critical pressure of CO_2	$p_{C,CO2}$	$kg \cdot m^{-1} \cdot s^{-2}$
Critical temperature of CO_2	$T_{C,CO2}$	K
Diameter of chamfer	D_{cha}	m

The equation for calculating the mass flow rate should have the following form:

$$\dot{m} = F(p_1, T_1, D_{orf}, \rho_1, p_{C,CO2}, T_{C,CO2}, D_{cha}) \qquad \text{(Eq 1)}$$

Table 2: dimensionless numbers used for the mass flow model

$\pi_1 = \dfrac{\dot{m}}{D_{orf}^{\ 2} \cdot \sqrt{\rho_1 \cdot p_1}}$	$\pi_2 = \dfrac{L}{D_{orf}}$	$\pi_3 = \dfrac{p_1}{p_{C,CO2}}$	$\pi_4 = \dfrac{T_1}{T_{C,CO2}}$	$\pi_5 = \dfrac{D_{orf}}{D_{cha}}$

With the dimensionless numbers from Table 2 equation 1 can be formulated in the following form:

$$\pi_1 = a_1 \cdot \pi_2^{a_2} \cdot \pi_3^{a_3} \cdot \pi_4^{a_4} \cdot \pi_5^{a_5} \qquad \text{(Eq 2)}$$

Therefore the equation for the mass flow rate can be written as:

$$\dot{m} = D_{orf}^{\ 2} \cdot \sqrt{\rho_1 \cdot p_1} \cdot a_1 \cdot \left(\frac{L}{D_{orfl}}\right)^{a_2} \cdot \left(\frac{p_1}{p_{C,CO2}}\right)^{a_3} \cdot \left(\frac{T_1}{T_{C,CO2}}\right)^{a_4} \left(\frac{D_{cha}}{D_{orf}}\right)^{a_5} \qquad \text{(Eq 3)}$$

The results of the experimental investigations described in the previous chapter were used to determine the unknown constants a_1 to a_5. These parameters were optimized to reach a minimal deviation between measurement and calculation. Table 3 shows the set of parameters which yielded the best results for the mass flow rate.

Table 3: Empirically determined parameters for short tube orifice

a_1	a_2	a_3	a_4	a_5
0.41	-0.015	0.9	-5	0.36

Figure 8 shows a comparison between the results of the measurement and the calculation for all tests presented in the previous chapter. It can be seen that the deviation between measurement and calculation is within an interval of ± 5 % for most of the points. The mean deviation is about 2 % and the maximum difference is 6.4 %.

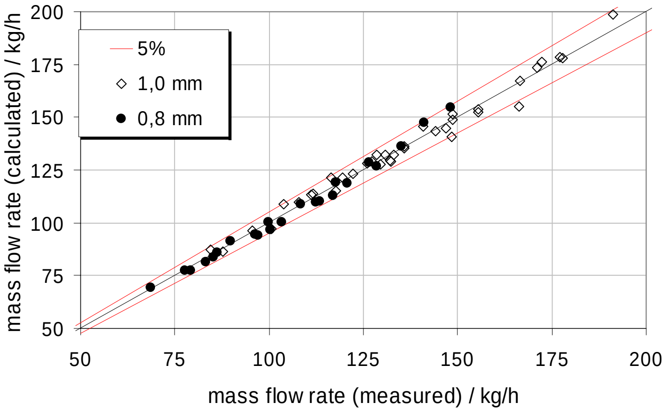

Figure 8: Comparison of measured and calculated mass flow rate

SUMMARY AND OUTLOOK

Not every detail of the refrigerant CO_2 has been investigated extensively up to now. Since the expansion device is an important component of a refrigerant cycle it is interesting to have a closer look at short tube orifices, which are widely spread in connection with “conventional” refrigerants.

First of all it was checked at which conditions the fluid flow through the tube orifice is choked. Experimental investigations have shown that for a wide range of operating conditions choked flow occurs. That means that the refrigerant mass flow is independent of the outlet pressure and that CO_2 behaves similar to other refrigerants, because choked flow occurs if the outlet condition of the orifice is in the two phase region.

Further experimental data were presented as a function of major operating parameters and short tube geometries. From these investigations it can be concluded that the mass flow rate is strongly dependent on the upstream pressure, upstream temperature as well as the bore diameter and less dependent on the tube orifice length. The mass flow rate increases with increasing upstream pressure at a constant inlet temperature and decreases with increasing temperature if the pressure remains constant. The effect of an inlet chamfering turned out to be in a range of about 5 to 10 %.

Based on the experimental investigations an empirical model with five independent variables for predicting the mass flow rate has been developed. The model was designed to cover flow with single-phase inlet and two-phase outlet for the refrigerant CO_2. For the investigated conditions approximately 98 % of the measured data were within an interval of ± 5 % of the models prediction. Possibly the number of variables could be reduced, e.g. if the influence of the tube orifice length is neglected, what seems to be appropriate for a certain range of the ratio of tube length to bore diameter. Another variable that could be discarded is the parameter for the influence of the inlet chamfering, if only sharp-edged inlets are of interest.

Future activities shall focus on further investigations of the choked flow, e.g. the influence of the orifice diameter on the critical pressure. Furthermore different geometries of orifices for other applications, i.e. mass flow rates, pressures and temperatures shall be analyzed by experiments and described by empirical models.

REFERENCES

Martin K., Lang G., Rieberer R. 2005, Mobile HVAC-System with CO_2 as refrigerant – Simulations and Measurements, *Proc. VTMS7*, Toronto, SAE paper 2005-01-2023.

Kim Y., O'Neal D. L. 1994a A semi-empirical model of two phase flow of R134a through short tube orifices, *Experimental & Thermal Fluid Science*, 1994; 9:426-435

Kim Y., O'Neal D. L. 1994b, Two-phase flow of R22 through short-tube orifices, *ASHRAE Transactions*, Vol. 100, No. 1, pp. 323-334

Kim Y., Payne V., Choi J., Domanski P. 2005, Mass Flow Rate of R-410A through Short Tubes Working near the Critical Point, *International Journal of Refrigeration,* Vol. 28, No4, pp. 547-553

Payne W. V., O’Neal D. L. 2004, A Mass Flowrate Correlation for Refrigerants and Refrigerant Mixtures Flowing through Short Tubes, *International Journal of Heating Ventilating Air Conditioning and Refrigerating Research*, Vol. 10, No. 1

Robin J.-J, Winterstein R., Blauth, Ch. 2003, Expansionsorgane im A/C - Kältekreislauf mit R744 (CO_2), DKV Tagung 2003

Singh G., Hrnjak P., Bullard C. 2001, Flow of Refrigerant 134a through Orifice Tubes, *ASHRAE Transactions,* Vol. 7, No. 3 pp. 245-262

VDI-WA97, 1997, VDI-Wärmeatlas, Kapitel Lgc, 8. Auflage, Springer Verlag, Heidelberg

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