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A THEORETICAL MODEL OF FLOW THROUGH SHORT TUBE ORIFICES

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Abstract. *Modelling of flow through short tube orifices, expansion devices used mainly in air conditioning units, has been predominantly of the semi-empirical type. The limitations of such models are obvious, with their application restricted to the the refrigerant tested, geometry studied and the range of experimental flow conditions. Surprisingly, theoretical models are still scarce in the literature. Moreover, one of the very few of these models still relies on experimental data (exit pressure) for its completeness. The present work describes a fully theoretical model for flow trough short tube orifices. Flow predictions are compared with experimental data showing good agreement.*

Keywords. *short tube restrictors; modeling*

1. Introduction

Short tube restrictors are widely used in both automotive and small residential air conditioning units (Kim and O'Neal, 1995). A length to diameter ratio within 3 and 20 defines short tube restrictors (Aaron and Domanski, 1989). They are also called short-tube orifices, when the length to diameter ratio drops below 3 (ASHRAE, 1994). They can be of the stationary or moveable type (ASHRAE, 1994). Main characteristics are (Aaron and Domanski, 1989): low cost, high reliability, ease of inspection and ease of replacement. Figure (1) shows the geometry of a short tube restrictor of the fixed type (Aaron and Domanski, 1989). Basic dimensions are the length and diameter. If chamfered, angle (typically 45^{o}) and depth of the chamfer are also included. Figure 2 displays the schematic of a moveable short tube restrictor (ASHRAE, 1994), consisting of a longitudinally ribbed piston that can move within its housing. Refrigerant flow to the left forces the piston against its seat, thus offering greatest flow restriction. On the other hand, flow to the right will move the piston away from its seat, allowing a larger flow area and, consequently, considerable lower restriction (ASHRAE, 1994). The possibility of restricting the flow in one direction, and practically allowing full flow in the opposite direction, makes moveable short tube restrictors quite useful for dual mode (cooling-heating) heat pumps, with the elimination of additional check valves (Aaron and Domanski, 1989).

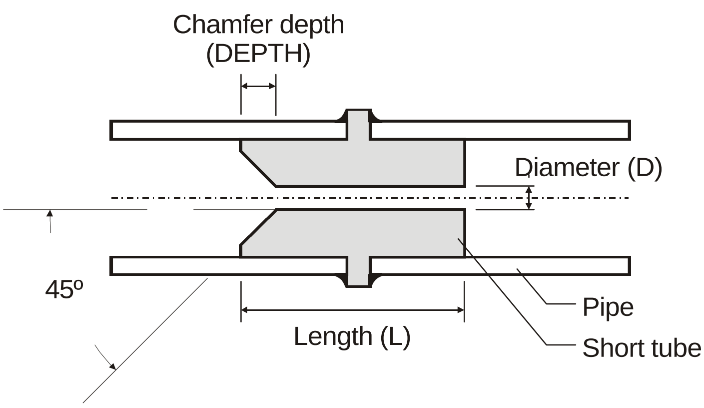

Figure 1 – Schematics of a stationary short tube restrictor (Aaron and Domanski, 1989).

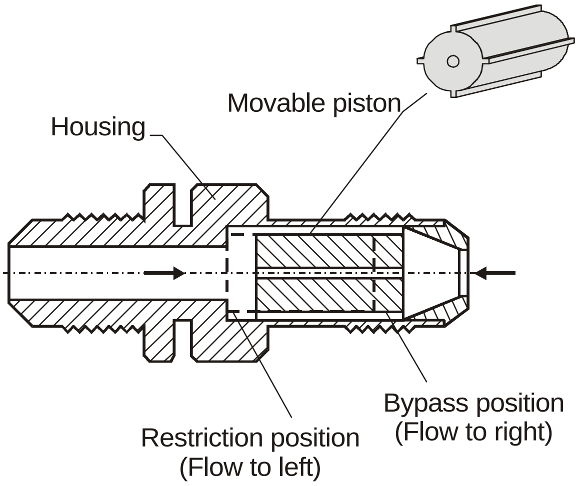

Figure 2 – Schematic of a moveable short tube flow restrictor (ASHRAE, 1994).

2. Physical Phenomenon

A comprehensive experimental investigation, carried out by Aaron and Domanski (1989), provided some insight into the physics of flow through short tube restrictors. Experiments were conducted to obtain performance data, pressure distribution along the tube length and flow visualization. Results have shown a strong dependency of the mass flow rate upon geometry (length, diameter and inlet/exit chamfering) and upstream conditions (subcoooling and pressure).

In figure 3 experimental data from Aaron and Domanski (1989) shows the mass flow dependency upon the downstream pressure. It is evident that the mass flow rate becomes fairly insensitive to the downstream pressure, when it falls below the liquid saturation pressure. It should be emphasized that this condition, i.e. downstream pressure below liquid saturation pressure, is the one most likely to occur in vapor-compression systems (two-phase flow at the exit of the expansion device).

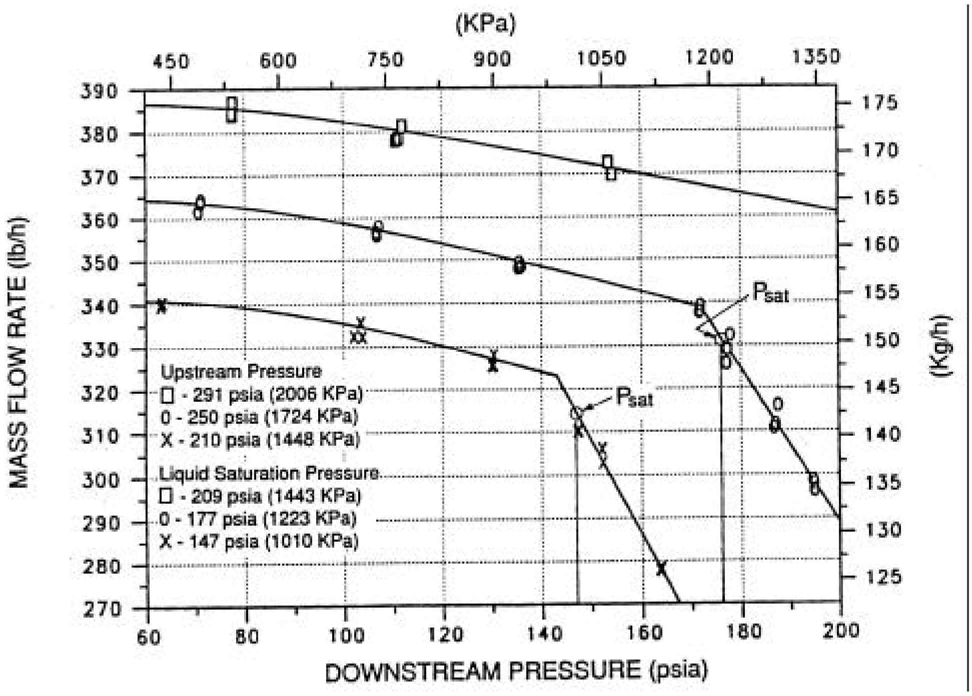

Figure 3 – Mass flow dependency upon downstream pressure (Aaron and Domanski, 1989).

Aaron and Domanski (1989) classify this condition as “nearly” choked flow. Since the mass flow rate does not become strictly constant with further reduction of the downstream pressure, the authors understand that it would be technically incorrect to term the flow simply as choked. Approximately choked flow is desirable in the operation of heat pumps and refrigeration systems. Consider, for example, a decrease in the evaporator load, causing an increase in the evaporator pressure (Aaron and Domanski, 1989). If the restrictor were sensitive to the downstream pressure, such as proportional to the square root of the pressure differential across the restrictor, less refrigerant would be fed to the evaporator, exactly when more mass flow rate would be in demand to meet the increased heat load. The result would be an even higher degree of superheat at the evaporator exit.

3. Semi-empirical flow models

Single-phase flow. When the downstream pressure is greater than the liquid saturation pressure, the single-phase orifice equation (ASME, 1971) applies. It is assumed that the flow is steady, with uniform velocity profile, adiabatic, isothermal, incompressible, performs no external work and that potential effects are negligible.

$$\dot{m}_s = C_s A \sqrt{\frac{2\, r\left(P_{up} - P_{down}\right)}{\left(1 - b^4\right)}} \qquad (1)$$

The above equation may prove useful in transient regime simulation, when condensing and evaporating pressures may be close enough at, for instance, the beginning of a pull-down test.

Two-phase flow. The experimental data from Aaron and Domanski (1989) and derived flow model was used to construct mass flow rate prediction charts. The actual mass flow rate is given by a reference short tube mass flow rate, corrected by three factors.

$$\dot{m} = \dot{m}_{ref}\,\Phi_1 \Phi_2 \Phi_3 \qquad (2)$$

The reference mass flow rate and the three correction factors are provided in charts, which makes this method of mass flow prediction, adopted by AHRAE (1994), of little use for computational simulation purposes.

Mass flow equations, derived from experimental data, are presented next. The original nomenclature was preserved. Aaron and Domanski (1989) started from the orifice equation, with corrections for the flow coefficient and pressure differential.

$$\dot{m} = C_c A_S \sqrt{2\, r\left(P_1 - P_2\right)} \qquad (3)$$

The correlation factor for chamfered inlet short tubes was derived as a function of chamfer depth and the L/D ratio.

$$C_c = 1 + 0.551\left(\frac{L}{D}\right)^{0,5844}\left(\frac{d}{D}\right)^{0,2967} \qquad (4)$$

Note that, with the absence of chamfering, the correction equation reverts to sharp edged entrance flow (Aaron and Domanski, 1989).

$$\frac{P_2}{P_{sat}} = 1 + 12.599\left(SUB\right)^{1.293} - 0.1229\exp\left[-0.017\left(\frac{L}{D}\right)^2\right]0.04753\left(EVAP_{AD}\right)^{0.6192} \qquad (5)$$

where

$$SUB = \frac{T_{sat} - T_{fluid}}{T_{sat}} \qquad (6)$$

$$EVAP_{AD} = \frac{P_{sat} - P_{down}}{P_{sat}} \qquad (7)$$

Care should always be taken when employing an empirical correlation. Its application should be limited to the data range used in its derivation. Limitations of Aaron and Domanski's model (1989) are as follows.

Table 1 – Data range of correlation of Aaron and Domanski (1989).

Refrigerant R-22
5.5 oC < subcooling < 13.9 oC
1448 kPa < upstream pressure < 2006 kPa
207 kPa < downstream pressure < liquid saturation
9.5 mm < length < 25.4 mm
1.09 mm < diameter < 1.7 mm
0 < chamfer depth < 0.508 mm
inlet chamfer depth: 45^o
short tube material: brass

Other semi-empirical models, based on the methodology of Aaron and Domanski (1989), can be found in the literature. For instance, Kuehl and Goldschmidt (1992) tested a number of commercially available short tube restrictors, partially complementing the data from Aaron and Domanski (1989). Tests were performed for refrigerant R-22. Kim and O'Neal (1994a) and Kim et al (1994) conducted measurements with R-12 and R-134a. Both two-phase and subcooled liquid flow conditions, entering the short tube, were considered. A model was developed by empirically correcting the modified orifice equation as a function of a normalized form of upstream pressure, upstream subcooling (or upstream vapor quality), downstream pressure and short tube geometry. A two-phase correction factor included effects of upstream quality and short-tube geometry (Kim et al, 1994). A reference short tube diameter of 1.35 mm was used. Later, Kim and O'Neal (1994b) extended the work of Aaron and Domanski (1989), for refrigerant HCFC-22. Further tests were carried out for non-azeotropic mixtures, R-407c (Payne and O'Neal (1998) and R-410A (Payne and O'Neal, 1999), this time including the effects of the presence of polyolester oil. Motta et al (1999) carried out experiments with mixtures of R-407c and lubricating oil. Compared to the results of Payne and O'Neal (1998), mass flow rates were lower and the oil, of the ester type, presented a lower viscosity. Experimental values were fitted to a function correlating the mass flow ratio between the flow of oil-refrigerant mixture and pure refrigerant with the oil concentration and the degree of subcooling. More recently, Paulucci (2001) performed a series of experiments for flow of R404a through chamfered short tube restrictors of three different diameters (#41, #48 and #51). The same experimental apparatus from Motta et al (1999) was employed. Data was correlated to Aaron and Domanski (1989) equation for mass flow rate, equation (3). The value for chamfered inlet correction factor was calculated from Aaron and Domanski (1999), equation (4). All these models are presented in detail, with equations, coefficients and experimental data range, by Paulucci et al (2001).

The main shortcoming of semi-empirical models, of being limited to the range of the experimental data, could be overcome with the use of a theoretical model. A single general model would be more appropriate, for example, in the study of different refrigerants for a givenrefrigeration(8) system.

4. Theoretical flow model

Hsu and Graham (1976) derived a general form of the critical flow equation, as presented next. The criterion for critical flow is that mass flow rate does not increase with further reduction of the pressure, i.e.

$$\frac{\partial G}{\partial P} = 0 \tag{9}$$

It should be noted, however, that inspection of Figure 3, with the experimental findings of Aaron and Domanski (1989), shows that the assumption embodied in equation (9) is only an approximation.

The mass flux can be related to the liquid and vapor velocity through the vapor quality and void fraction (Hsu and Graham, 1976).

$$(1-x)v_l G = (1-a)u_l \tag{10}$$

$$x v_g G = a u_g \tag{11}$$

Equations (10) and (11) can be combined to provide the void fraction in terms of the slip ratio, k .

$$a = \frac{xv_g}{xv_g + k(1-x)v_l} \tag{12}$$

The slip ratio is defined as:

$$k = \frac{u_g}{u_l} \tag{13}$$

The relation between the two-phase mass flux and the liquid velocity can thus be derived, taking into account the slip ratio:

$$G = \left[\frac{k}{xv_g + k(1-x)v_l}\right]u_l \tag{14}$$

The relation between velocity and pressure is provided by the momentum equation. Neglecting the hydrostatic head and viscous dissipation terms, and assuming one-dimensional steady-state, one has:

$$G\frac{d}{dz}\left[xu_g + (1-x)u_l\right] = -\frac{dP}{dz} \tag{15}$$

By combining equations (9), (14) and (15), Hsu and Graham (1976) obtained the general form of the critical flow equation, at a given z:

$$G_c = \frac{1}{\frac{\partial}{\partial P_e}\left\{\frac{\left[k(1-x)v_f + xv_g\right]\left[xk + (1-x)\right]}{k}\right\}_z} \tag{16}$$

As Hsu and Graham (1976) pointed out, the vapor specific volume, the slip ratio and the vapor quality may be functions of the pressure, so that relations have to be described. A number of theories on critical flow are presented by Hsu and Graham (1976). Kim and O'Neal (1995) examined eight of these critical flow models and compared them to experimental data for HCFC22 and HFC134a. The models studied were categorized into three groups:

i. Homogeneous equilibrium models: the mixture is homogeneous in phase composition and the two phases have equal velocities, i.e., the slip ratio is equal to unity - models in this category differ in the way the derivative of vapor quality in respect to pressure is determined;
ii. Homogeneous metastable models: they apply to cases where the flow is homogeneous, k=1, and interfacial mass transfer is restricted, due to insufficient time, $\partial x/\partial P=0$;
iii. Non-homogeneous equilibrium models: they are regarded by Kim and O'Neal (1995) as quite complicated as all three interfacial transports have to be considered without limitations.

Kim and O'Neal (1995) observed that homogeneous metastable (or frozen) models, from Smith (1963) and Wallis (1969), compared most consistently with available experimental data, for a wide pressure range, except for the low outlet quality region (<0.06). The Wallis (1969) model applies no-slip, k=1, and $\partial x/\partial P=0$ assumptions in the critical flow general form equation, (16), yielding:

$$G_c = \left[-\left(x\frac{\partial v_g}{\partial P} + (1-x)\frac{\partial v_f}{\partial P}\right)^{-1}\right]^{1/2} \tag{17}$$

In equation (17), partial derivatives of the specific volumes, in respect to pressure, are refrigerant properties and have to be calculated with a proper numerical method (Kim and O'Neal, 1989).

Smith (1963) applied a homogeneous two-phase pressure-volume relation for an isentropic process. The assumption of $\partial x/\partial P=0$ was also used. The final form of the critical mass flux (Kim and O'Neal, 1995) is:

$$G_c = \left[\frac{xc_{pg} + (1-x)\, c_{pf}}{xc_{vg} + (1-x)\, c_{pf}} \left(\frac{P}{xv_g} \right) \right]^{1/2} \tag{18}$$

Note that the properties and the vapor quality of equations (17) and (18) have to be calculated at exit conditions. There is experimental evidence (Kim and O'Neal, 1995) that choking occurs at the exit plane of the short tube. Kim and O'Neal (1995) assumed homogeneous and adiabatic two-phase flow to apply the energy balance equation along the short tube orifice. For a control volume (Aaron and Domanski, 1989) as depicted in figure 4, one has:

$$\bar{h}_2 - \bar{h}_1 + \frac{G_c^2}{2} \left(\bar{v}_2^2 - \bar{v}_1^2 \right) = 0 \tag{19}$$

where, in the two-phase region, average properties are given by:

$$\bar{h} = h_f + xh_{fg} \tag{20}$$

$$\bar{v} = v_f + xv_{fg} \tag{21}$$

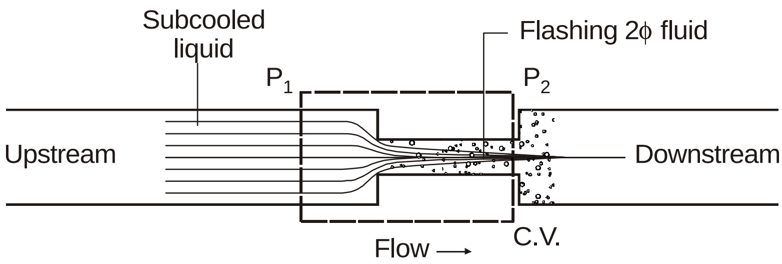

Figure 4 – Control volume for short tube restrictor (Aaron and Domanski, 1989).

The exit vapor quality can be calculated from equations (19) to (21), provided the exit pressure is known. In the only work where theoretical models were employed (Kim and O'Neal, 1995), the exit pressure was experimentally determined, by a linear extrapolation of the measured pressures alongside the short tube. Therefore, to complete the model, the remaining task would be the prediction of the exit pressure. Kim and O'Neal (1995) shows, from experimental pressure distribution with R-134a, that a sharp pressure drop occurs at the inlet section, probably due to acceleration effects, to recover close to P_{sat}, at the tube exit. Aaron and Domanski (1989) also conjecture that, at each subcooling level, from R-22 data, the pressure inside the tube approximates P_{sat} and remains fairly constant. Incidentally, Hsu and Graham (1976) also report, from Henry (1970), that, for a sharp entrance short tube with 3<L/D<12, pressure is essentially constant. Therefore, a first approximation would be $P_e = P_{sat}$.

However, close inspection of Aaron and Domanski's (1989) data, as reproduced for one single data point in Figure 5, shows that the exit pressure may be far below P_{sat}. For instance, in the example of Figure 5, for a subcooling of 10^{o}F, the measured exit pressure is 165 psia (1137.6 kPa), which does not compare well with a saturation pressure of 218.5 psia (1506.5 kPa). A better approximation is to calculate the pressure drop, due to the contraction of the flow area (sudden flow acceleration) at the entrance and due to friction alongside the short tube. For the evaluation of the inlet pressure drop (sudden contraction originated by the connection between the condenser discharge line and the short tube restrictor), the following equation, for single-phase incompressible flow through an abrupt change of section (Chisholm, 1983), is used:

$$\Delta P_k = K \frac{G^2 v_l}{2} \tag{22}$$

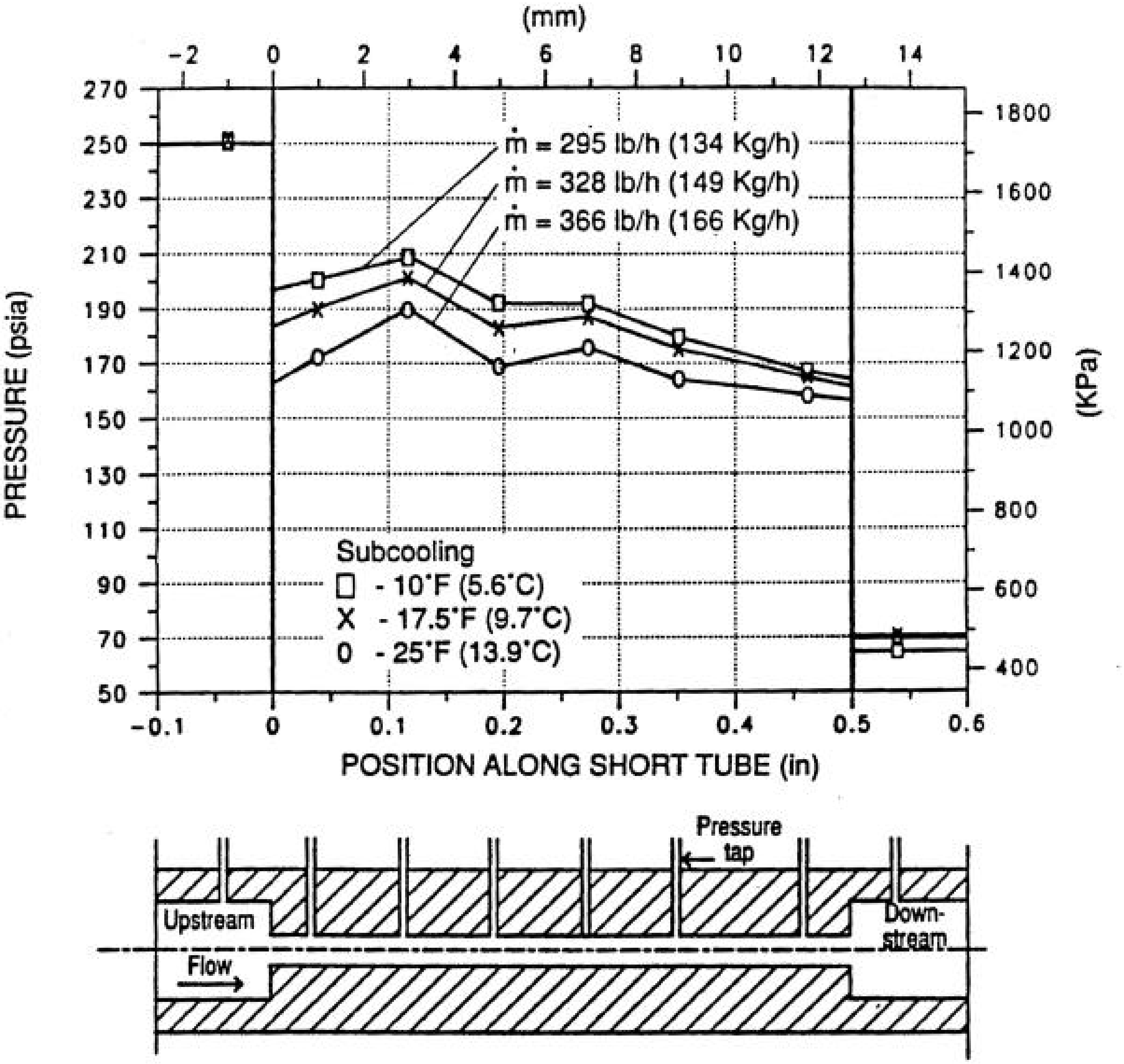

Figure 5 – Pressure distribution along the short tube restrictor, from Aaron and Domanski (1989).

The pressure drop coefficient, K, is calculated using a correlation from Chisholm (1983):

$$K = \frac{1}{C_k^{\ 2}} - b^2 - 2\left(\frac{1}{C_k} - 1\right) \tag{23}$$

$$b = \frac{A_2}{A_1} \tag{24}$$

The ratio between the “vena contracta” and short tube section areas, defined as the contraction coefficient, is determined by Chisholm (1983)

:

$$C_k = \frac{A_0}{A_2} = \frac{1}{0.639\left(1-b\right)^{0.5}+1} \tag{25}$$

Pressure drop with homogeneous two-phase flow was assumed for the tube length.

$$\Delta P_f = \Delta P_{flo} j_{flo}^{2} \tag{26}$$

where

$$\Delta P_{flo} = \frac{l_l (1-x) G^2 v_l L}{2D} \tag{27}$$

The two-phase flow multiplier, j_{flo}^{2}, and the friction factor, l_l, are provided by Chisholm (1983) and Churchill (1977), respectively.

$$j_{flo}^{2} = 1 + \left(\frac{v_v}{v_l} - 1\right)x \tag{28}$$

$$l_l = 8\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + \frac{1}{\left(A+B\right)^{3/2}}\right]^{1/12} \tag{29}$$

$$\text{where } A = \left[2.457\ln\left\{\frac{1}{\left(7/\mathrm{Re}\right)^{0.9} + 0.27\,e/D}\right\}\right]^{16}, \; B = \left[\frac{37530}{\mathrm{Re}}\right]^{16} \text{ and } \quad \mathrm{Re} = \left[\frac{GD}{m}\right] \tag{30}$$

The exit pressure is then calculated from:

$$P_e = P_{up} - \left(\Delta P_k + \Delta P_f\right) \tag{31}$$

The mathematical model thus results in a system on non-linear algebraic equations, (17) or (18), (19) and (31), where exit pressure and vapor quality, P_e and x, and the critical mass flux, G_c, are the unknowns. Refrigerant properties, including those partial derivatives of Wallis (1969) model, were derived from REFPROP (McLinden et al., 1998).

Finally, the critical mass flow rate is calculated from equation (32). It will be the mass flow rate provided by the short tube orifice for given geomery, refrigerant and upstream flow conditions.

$$\dot{m} = A G_c \tag{32}$$

5. Results

Preliminary results were obtained for refrigerant R-22 flowing through short tube orifices with sharp-edged entrance. Predicted values were compared with experimental data from Aaron and Domanski (1989), as shown in Figures 6 to 8.

Initially, the assumption of making the exit pressure directly related to the saturation pressure of the liquid was assessed. Figures 6a and 6b, with results for P_e/P_{sat} equal to 0.9 and 0.7, clearly show the inefficacy of this assumption.

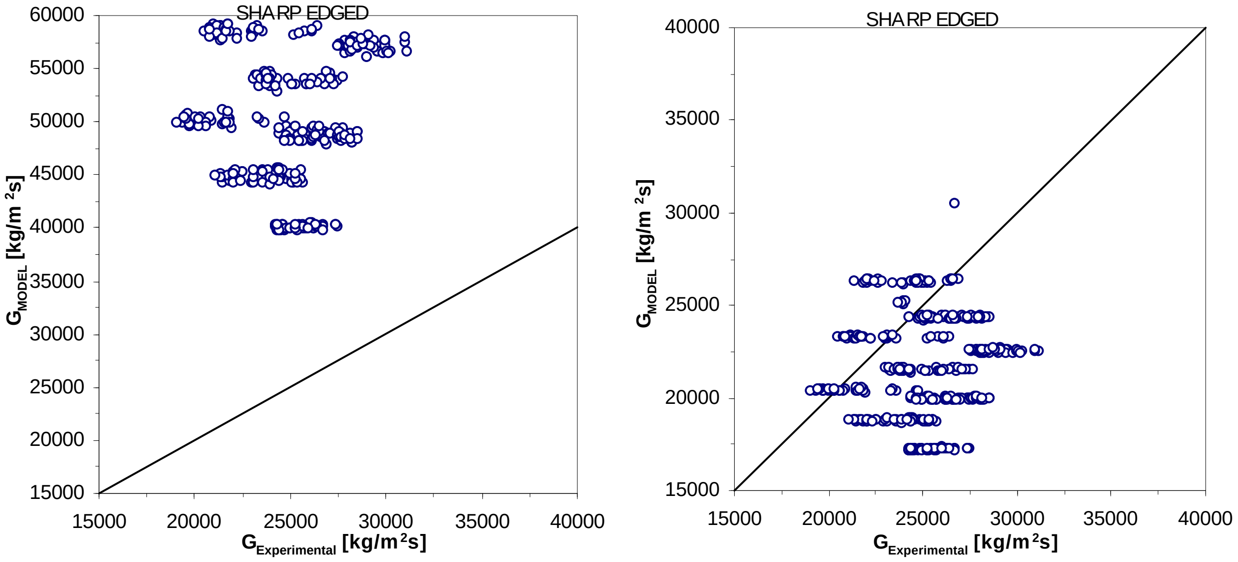

Figure 6 - Comparison between predicted and experimental values. Experimental data from Aaron and Domanski (1989) for R-22 flowing through short tube orifices with sharp-edged entrance. Relative roughness: e/D=0.001 and (a) P_e/P_{sat}=0.9 and (b) P_e/P_{sat}=0.7.

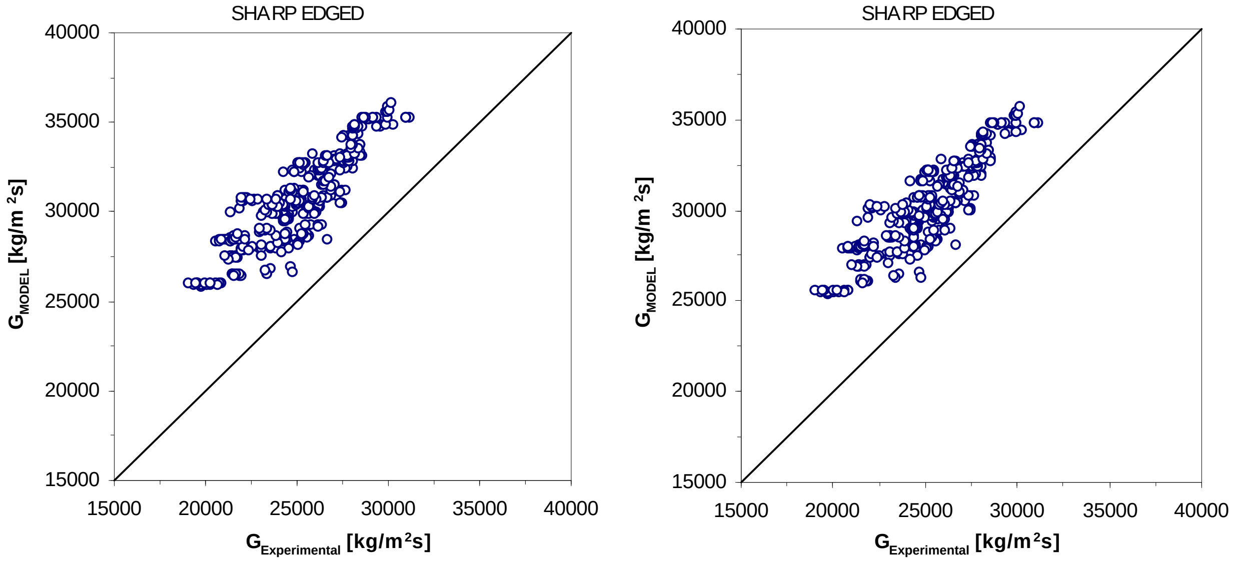

Figure 7 - Comparison between predicted and experimental values. Experimental data from Aaron and Domanski (1989) for R-22 flowing through short tube orifices with sharp-edged entrance. Wallis' (1969) critical flow model. (a) Smooth tube (e/D=0) and (b) Relative roughness: e/D=0.0001.

Predictions were then made with the full model, i.e., calculating exit pressure, vapor quality and mass velocity. Option was made for the Wallis (1969) critical flow model. Figures 7 and 8 depict the comparison between predicted and experimental values for smooth tube and different values of relative roughness. As, at the time of writing, no information was available on the roughness of the restrictors used by Aaron and Domanski (1989), no conclusive comparison can still be made. However, in general, it can be seen that the mass flow rate is still overpredicted.

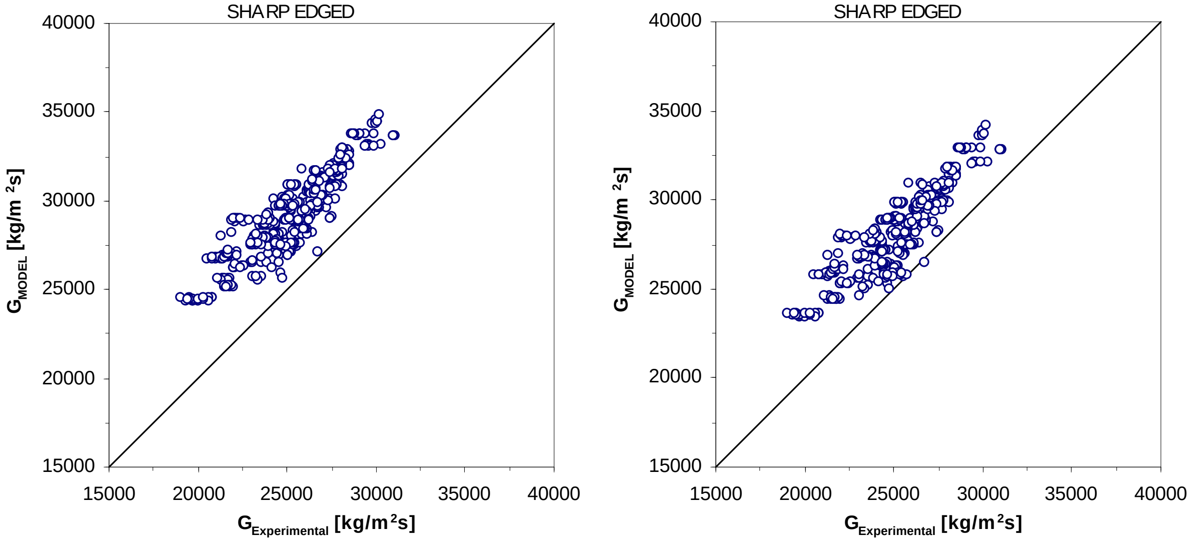

Figure 8 - Comparison between predicted and experimental values. Experimental data from Aaron and Domanski (1989) for R-22 flowing through short tube orifices with sharp-edged entrance. Wallis' (1969) critical flow model. Relative roughness, *e/D* : (a) 0.001 and (b) 0.003.

6. Concluding remarks

Flow models for fixed area expanding devices are extremely useful for the prediction of their performance as well as for their sizing. A literature survey on the modeling of short tube restrictors evidences the absence of theoretical models. In the modeling of flow through short tube restrictors two landmarks can be identified: the works of Aaron and Domanski (1989), who developed a trend-setting semi-empirical model, and of Kim and O'Neal (1995), who first first employed two-phase critical flow models that have been available in the literature for other applications, notably Nuclear Engineering. Their model could not be considered as totally theoretical as a key parameter, the exit pressure, was taken from experimental data. In the present work, the initiative of Kim and O'Neal (1995), in pursuing theoretical models for short tube restrictors, was pushed foward, leading to a model that is fully general. The mathematical model here established can be applied for the flow of any pure refrigerant through sharp-edged short tube restrictors of any given geometry. Preliminary results have shown an overprediction of values, if compared with experimental data. Futher studies should be carried out to improve agreement, to add more refrigerants in the experimental data-base and to include the flow of refrigerant-oil mixtures.

7. Acknowledgements

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8. Nomenclature

A	area (m^2)
A	constant in Curchill's equation for friction factor
A_s	short tube cross sectional area (m^2)
B	constant in Curchill's equation for friction factor
c_p	specific heat at constant pressure (Jkg^{-1}K^{-1})
c_v	specific heat at constant volume (Jkg^{-1}K^{-1})
C_c	correlation factor for chamfered inlet short tubes
C_k	contraction coefficient

C_s	coefficient of discharge for single-phase orifice equation
D	short tube cross sectional diameter (m)
D_{ref}	reference short tube diameter, 1.35mm (m)
$EVAP_{AD}$	non-dimensional effect of downstream pressure, Aaron and Domanski (1989)
G	mass flux ($kg.s^{-1}m^{-2}$)
G_c	critical mass flux ($kg.s^{-1}m^{-2}$)
h	specific enthalpy (kJ/kg)
k	slip ratio
K	pressure drop coefficient
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (Pa)
P_{sat}	upstream refrigerant liquid saturation pressure referenced to subcooled liquid temperature (Pa)
P_1	upstream or condenser pressure (Pa)
P_2	lower pressure that governs the flow through orifices or short tubes (Pa)
R	mass flow ratio between oil-refrigerant mixture and pure refrigerant flows
Re	Reynolds number
SUB	non-dimensional effect of subcooling, Aaron and Domanski (1989)
u	velocity (m/s)
u	specific internal energy (kJ/kg)
v	specific volume (m^3/kg)
x	vapor quality
z	position (m)

Greek Symbols

a	void fraction
β	ratio of short tube diameter to upstream tube diameter
δ	chamfer depth (mm)
ε	tube roughness
l_l	friction factor
Φ_1	correction factor for short tube geometry
Φ_2	correction factor for subcooling
Φ_3	correction factor for inlet chamfering
j_{flo}^2	two-phase flow multiplier

Subscripts

b	bubble point
c	critical point for refrigerant property
$down$	downstream
e	exit
f	friction
k	contraction
l	liquid
lv	liquid-vapor
ref	reference
s	single-phase
up	upstream
v	vapor
0	vena contracta
1	upstream line
2	short tube

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