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An Experimental Study of HC-600a Flow Through Variable Expansion Devices for Household Refrigerators

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An Experimental Study of HC-600a Flow Through Small Capacity Variable Expansion Devices

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ABSTRACT

This work focuses on the HC-600a flow through small capacity variable expansion devices. To this end three electronic expansion valves of the same innovative design but with distinct orifices, were firstly tested in a dry nitrogen flow testing apparatus by varying the opening ratio and the operating conditions. Secondly, equivalent capillary tubes were defined based on the measured nitrogen mass flow rates through an updated version of the well-known Kipp Schmidt's equation. Thirdly, simulations were carried out with an in-house validated capillary tube model to estimate the HC-600a mass flow rate, using the length and inner diameter of the equivalent capillary tubes as input data. A test rig was also designed, constructed and used to measure the valve HC-600a mass flow rates under different operation conditions and opening ratios. Comparative analyses between the valve and capillary tube mass flow rates were then carried out under different conditions. In-depth knowledge about the two-phase capillary tube and valve flows were then used to explain the observed differences.

1. INTRODUCTION

Expansion devices are essential in refrigeration systems, both to reduce the refrigerant pressure and to properly feed the evaporator. In domestic appliances such a device is usually a capillary tube, which is nothing but a tube of very small inner diameter and a relatively long length. Despite being simple and cheap, the capillary tube does not adjust to variations in the operating conditions. As the cross-section area is fixed, once sized, the capillary tube does not allow the system to adequately adapt to operating conditions other than the one it was designed for, which may reduce the energetic efficiency of the refrigerator. An alternative would be the use of variable action expansion devices that modulate the free flow area according to the necessity. However, although widely used in commercial and industrial appliances, such devices are not found in the domestic sector mainly due to the small mass flow rates involved. This can be proved by the almost complete lack of correlated subjects in the literature. Although there are several works concerning expansion valves, are rare the ones that focused on domestic refrigeration systems. Are also rare the studies that aimed to investigate the valve behavior when tested with the current dominant fluid, HC-600a (isobutane). Ronzoni *et al.* (2013), for instance, in order to obtain suitable mass flow rates for small capacity systems, adopted a PWM-driven electronic valve designed for light commercial appliances in series with a capillary tube (Thiessen and Klein, 2007), but used HFC-134a as the refrigerant fluid.

In general, the literature review showed that the use of variable expansion devices is in fact justified and that expansion valves can contribute to a reduction in the energy consumption of refrigerators. Lazzarin and Noro (2008), pointed that the performance of a refrigeration system can be strongly affected by low ambient and evaporating temperatures, which would motivate the use of variable action expansion devices. In addition, a new version of the energy consumption standard IEC 62552 (20015) is about to take place and will require tests not only at the usual ambient temperature of 32 °C, as stated in the old standard ISO 15502 (2005), but also at 16 °C, which can penalize systems mounted with capillary tubes. However, among the few works that focused on the effect of the expansion device on the system performance, only Marcinichen and Melo (2006), Björk and Palm (2006) and Boeng and Melo (2012) conducted experiments with household refrigerators. Björk and Palm (206) and Boeng (2012), however, carried their experiments without controlling the superheating degree and at only one compressor speed. Marcinichen and Melo (2006), in turn, in addition to testing at different compressor speeds, kept the superheating control activated, but, as the authors themselves mentioned, they used an inappropriate control logic and a valve with an oversized orifice, which masked eventual performance gains. Regarding the works with focus on the expansion device itself, none tried to analyze the thermodynamic behavior of such component under operating conditions typically found in domestic appliances. All studies focused on commercial and industrial applications, with refrigerant fluids such as R22, R407C and R410A (Shanwei *et al.*, 2005, Park *et al.*, 2007 e Li, 2013, for example), and pressure and mass flow rate ranges completely out of those of small systems.

The physics behind capillary tube flow is relatively dominated. It can be seen, however, that the background about valves for domestic refrigeration is still scarce and that the need for a better understanding of refrigerant flow through such devices is evident. In this context, the present work intends to fill these gaps by carrying out experiments with specific valves focusing on operating conditions typical of household refrigerators. At first, tests were conducted on a dry nitrogen test bench with the purpose of obtaining a preliminary operational envelope of the valves and checking if they can really behave as capillary tubes of domestic systems. Later, the most promising valve was installed at another test bench, now with HC-600a, where the focus was the characterization of the valve in terms of refrigerant mass flow rate.

2. EXPERIMENTAL WORK

2.1 Valves

The valves studied herein are actually micro electronic valves, as small as the tip of a pencil, that work through what the manufacturer (DunAn Microstaq) calls the electrothermal principle. The body of the valve is formed by three superimposed silicon layers (see Fig. 1). The top layer includes the electric connectors and acts as a cover for the intermediate layer, which contains the thermal actuator and the sliding plate. The thermal actuator is responsible for moving the sliding plate over the outlet orifices located at the bottom layer. This movement occurs through the application of an electric current on the intermediate layer and, due to the variation of its thermal resistivity, the plate dilates or contracts as a function of the generated heat and moves over the orifice. Thus, the orifice opening can be controlled proportionally. Figure 1 shows three ports at the bottom layer. The third one is optional and allows the valve to operate in a 3-way arrangement, with one inlet and two outlets (which is beyond the scope of the present work). Before being installed at a refrigeration system the body of the valve is encapsulated in a metallic housing with two copper tubes, one for the inlet and the other for the outlet of refrigerant fluid.

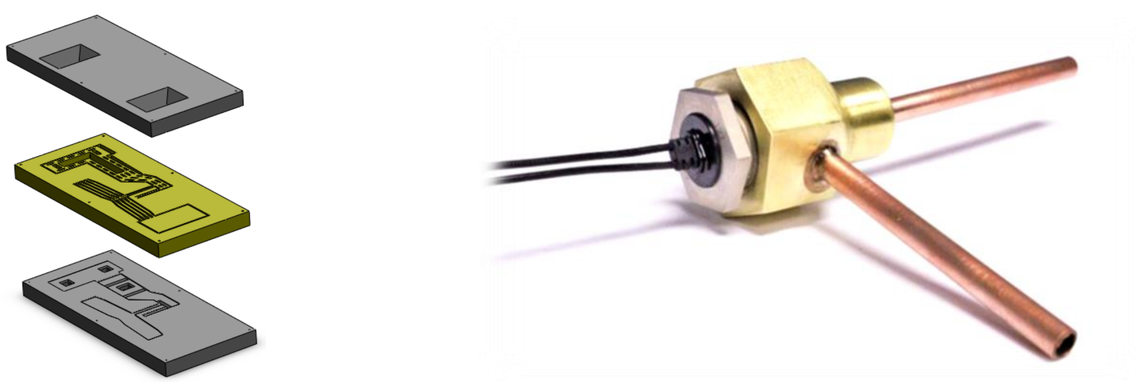

Figure 1: Valve assembly (DunAn Microstaq)

As shown in Fig. 1, the valves have constant-width rectangular orifices. The opening is regulated through the variation of the length covered by the sliding plate. Table 1 shows geometric parameters for each of the three valves used in the present study.

Table 1: Valves dimensions

Valve	Fixed width [mm]	Maximum length [mm]	Maximum area [mm^2]	Maximum hydraulic diameter [mm]
Smallest	0.22	0.120	0.0264	0.155
Intermediate	0.26	0.135	0.0351	0.178
Largest	0.32	0.135	0.0432	0.190

2.2 Nitrogen Flow

When it comes to capillary tubes the inner diameter is the most important parameter, once variations of the order 0.1 mm may have a significant impact on the system performance. Therefore, the measurement of such parameter is extremely important. As the nominal diameter provided by the manufacturer is not always precise, some techniques have been developed throughout the years in order to accurately determine the geometry of the tube. One of the most recent uses electronic microscopy (Silva, 2008 and Lara, 2014). Briefly, the procedure consists of cutting a short sample of the tube, embedding it in a resin mold, rectifying the cross section and recording the image with a microscope. With the aid of image edition and post processing algorithms the diameter can be obtained with a satisfactorily uncertainty. Such technique, however, requires an expensive equipment which forces many to opt for the old method based on the well-known Kipp and Schmidt (1961) equation.

The Kipp and Schmidt equation is an empirical correlation that relates the nitrogen flow rate through the capillary tube with its geometry and the pressure difference between inlet and outlet:

$$D = \left(\frac{VL^{a_2}}{a_1 \sqrt{P^2 - 1}} \right)^{1/a_3} \tag{1}$$

Where V is the nitrogen volumetric flow rate, P is the inlet pressure, and D and L are respectively the inner diameter and length of the tube. The outlet pressure is assumed to be 1 atm. The constants a_1, a_2 and a_3 were obtained firstly by Kipp and Schmidt (1961). Later, Boeng and Melo (2012) extended the valid range of the equation by testing more capillary tubes with diameters between 0.64 and 1.07 mm and lengths of 1, 2 and 3 m. Joining their data with those of Boeng and Melo (2012), Montibeller *et al.* (2016) further extended the application range of such equation by carrying 21 additional tests to measure the nitrogen flow rate through capillary tubes with inner diameters of 0.263 mm and 0.448 mm and lengths of 2, 3 and 4 m. In the present work, the coefficients proposed by Montibeller *et al.* (2016) were adopted since they are more recent and cover a wide range of geometries. Table 2 shows the coefficients proposed by each author.

Table 2: Constants of the Kipp and Schmidt equation

Author	a_1	a_2	a_3
Kipp and Schmidt (1961)	2.500	0.500	2.500
Boeng and Melo (2012)	2.362	0.496	2.657
Montibeller *et al.* (2016)	2.369	0.465	2.574

To ensure that the same capillary tube is being used in different systems around the world, many manufacturers of household refrigerators measure the nitrogen flow rate for a specific pressure difference (ASHRAE 28, 1996) and use the obtained value as a base of comparison. If the flow rate is measured again but in an allegedly different tube and the value is equal to the baseline, it is concluded that both capillary tubes have the same geometry. Additionally, to check if the nominal diameter is close to that provided by the tube manufacturer, designers use the measured flow rate and, for a given length and inlet pressure, obtain a counterproof with the Kipp and Schmidt equation.

In order to check the valves potential and its applicability in small refrigeration systems, a preliminary characterization was carried out by submitting the valves to dry nitrogen flow. To this end, following the recommendations of the ASHRAE 28 (1996) standard, a test bench was specifically constructed to measure the volumetric flow rate as a function of the inlet pressure and the valve opening fraction (see Fig. 2). The test procedure is relatively simple. First, the valve opening is defined. Then, the shut-off valve is opened and nitrogen is released from the cylinder to the circuit. After that, the pressure at the inlet of the EEV is controlled with the aid of a micrometric valve. Finally, the flow rate is measured with a Coriolis meter. As the nitrogen mass inside the cylinder decays with time, the micrometric valve must be slightly adjusted to keep the pressure constant at the inlet of the EEV. Each point is only recorded after steady state conditions are reached.

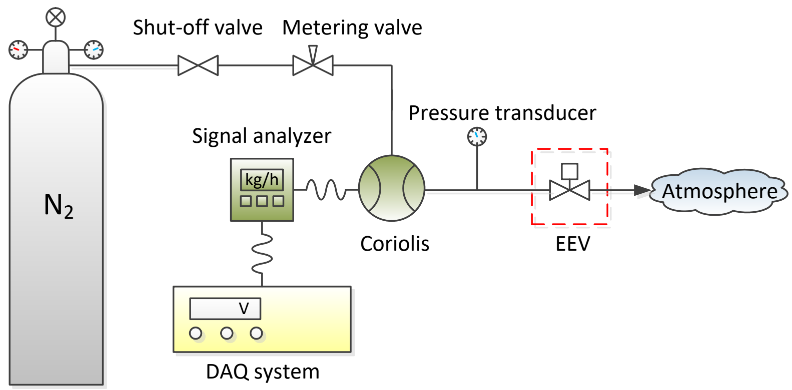

Figure 2: Nitrogen test bench

Therefore, with the volumetric flow rates, it was possible to calculate the geometry of the capillary tubes whose behaviors would be equivalent to those of the EEVs for each opening, that is, the geometry of the capillary tubes that would provide the same flow rate of each valve at the same inlet pressure. To this end, some typical tube lengths were defined and applied in Eq. (1) to obtain the equivalent inner diameters.

With the equivalent geometries, the model for adiabatic capillary tubes proposed by Hermes *et al.* (2010) was used to estimate the HC-600a mass flow rates. Briefly, the model consists of the following explicit equation for $\dot{m}$:

$$\dot{m} = \left\{ \frac{\pi^{2-d}\, 2^{2d-3}}{c} \frac{D^{(5-d)}}{\mu_f^d L} \left[\frac{P_c - P_f}{v_f} + \frac{P_f - P_e}{a} + \frac{b}{a^2} \ln\left(\frac{aP_e + b}{aP_f + b} \right) \right] \right\}^{1/(2-d)} \tag{2}$$

where $c = 0.14$, $d = 0.15$, $a = v_f\,(1 - \kappa)$, $b = v_f\, P_f\;\kappa$ and $\kappa = 1.63.10^5 P_f^{\,-0.72}$. P_c, P_e and P_f are respectively the condensing, evaporating and flashing pressures, and μ_f and v_f are respectively the viscosity and specific volume calculated at P_f.

The performance evaluation of expansion valves and capillary tubes also requires experiments with refrigerant fluid, since nitrogen tests are not capable of capturing some phenomenon, like phase change, for example. The main objective of such tests is to obtain the refrigerant mass flow rate as a function of parameters such as the condensing and evaporating pressures and the sub-cooling degree. However, refrigerant tests are considerably more complex because of the occurrence of two-phase flow and the necessity of several control strategies. In this context, an apparatus was specifically constructed, as described below.

2.3 Isobutane Flow

A test bench was designed and assembled to analyze the HC-600a flow through the most promising valve, as illustrated in Fig. 3. The apparatus is capable of automatically controlling the condensing and evaporating pressures, as well as the sub-cooling degree at the valve inlet. In order to do so and to ensure a higher cooling capacity, the test bench was mounted with two parallel variable speed compressors.

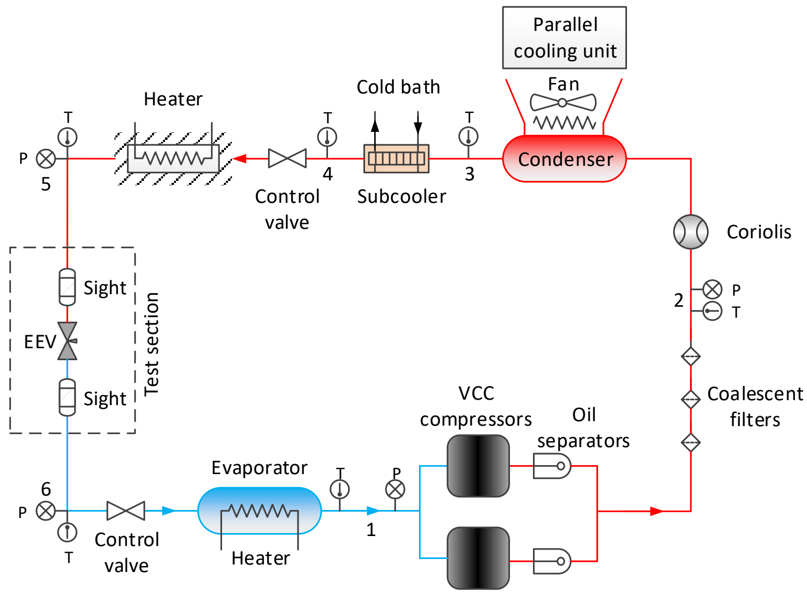

Figure 3: Isobutane test bench

From Fig. 3, it can be noted that after being compressed the refrigerant flows through two oil separators (one for each compressor) and three coalescent filters in series (1-2), to avoid the presence of oil and impurities in the electronic expansion valve (EEV). After filtration, the fluid passes through a Coriolis mass flow meter and then through the condenser, where it changes phase (3). The condenser is a tube-and-fin type and counts with an electrical heater at the air side that, together with a centrifugal fan, is used to control the condensing pressure. The fan operates at a constant speed while the heater has its power modulated by a PI (proportional integrative) controller to maintain the desired pressure. Subsequently, the liquid is sub-cooled (4) in a small brazed-plate heat exchanger subjected to the counter flow of cold water from a thermostatic bath. To promote the sub-cooling fine tuning at the EEV inlet, the fluid is then heated in a helical coil by the action of a PI-driven electrical heater. Between the sub-cooler and the heating coil, a micrometric valve was also used to manually decrease the condensing pressure when necessary. After that, the refrigerant flows through a sight glass and expands in the EEV (5-6). Both pressure and temperature are measured at the inlet and outlet of the EEV. The fluid then enters the evaporator where it changes phase again and becomes superheated. The evaporator is nothing but a long tube that extends from the EEV to the compressors, attached to another electrical heater. This heater is responsible for generating artificial thermal loads to ensure the presence of superheated vapor and to keep the temperature constant at the compressors suction. The evaporating pressure, in turn, is controlled by the compressors speed variation by means of a PI controller based on the pressure signal from a transducer installed at the EEV outlet. Another micrometric valve, installed between the EEV and the evaporator, was also manually operated to elevate the evaporating pressure when needed.

Experiments were performed within the following operating ranges, totalizing 83 tests:

• <u>Condensing temperature:</u> 25, 35 and 45 ºC.
• <u>Evaporating temperature:</u> -28 and -10 ºC.
• <u>Sub-cooling:</u> 5, 10 and 15 ºC.
• <u>Orifice opening ratio:</u> 25, 50, 75 and 95 %.

The choice of the condensing and evaporating temperatures was based on the temperature levels typically observed in the internal and external environments of the refrigerator. The range of condensing temperatures is in line with the ambient temperature range used in domestic refrigerator tests: 16 to 32 ºC (IEC 62552, 2015). Normally, the difference between ambient temperature and condensation is close to 10 °C, depending on the type of condenser used, which explains the values from 25 to 45 ° C adopted here. The explanation of the evaporating temperature range is analogous. For refrigerators with two or more compartments, which have a dedicated freezer, the internal air temperature can vary between 5 and -18 ° C (ISO 15502, 2005). As for the condenser, the difference between the temperature of the refrigerant and the air is close to 10 ° C, hence the range from -28 to -10 ° C. Additional tests were also performed for evaporating temperatures higher than -10 °C in order to check for choked flow conditions.

3. RESULTS

The nitrogen tests carried by Montibeller *et al.* (2016), who tested old versions of the valves analyzed herein, were repeated with the inlet pressure from 4 to 15 bar, which covers not only HC-600a applications but also HFC-134a. The orifice opening ratio of each valve was altered within the limits imposed by the manufacturer: 2 to 95%. Figure 4 shows the volumetric flow rate as a function of the opening ratio and inlet pressure for the intermediate and the largest valve. It can be noted that the intermediate valve is practically insensible to opening ratios lower than 30%. The largest valve, on the other hand, presented a much wider operating range, including opening ratios of less than 30%.

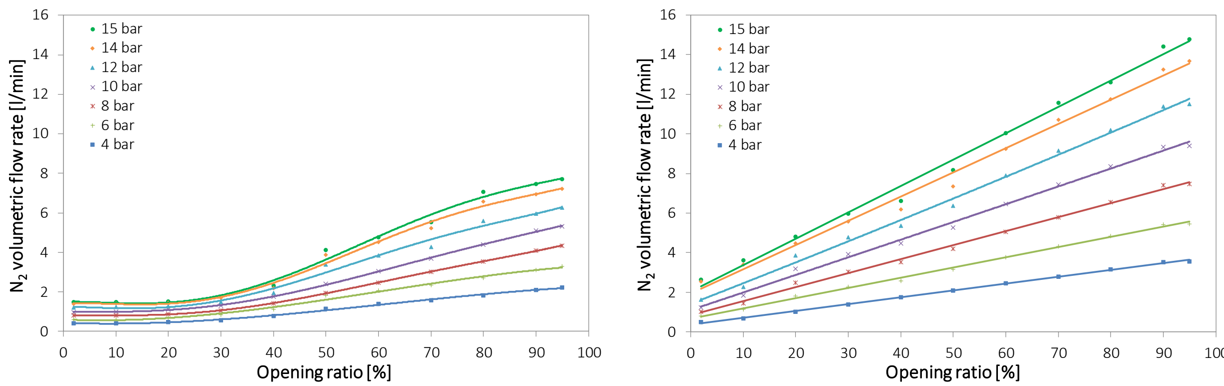

Figure 4: Nitrogen volumetric flow rate vs. opening fraction: intermediate valve (left) and largest valve (right)

The valves can be equivalent to several capillary tubes. For each opening ratio, depending on the selected length, it is possible to compose countless geometries of capillary tubes. Based on the previous results and Eq. (1), the inner diameter of the equivalent capillary tubes was calculated for the lengths of 1, 2 and 3 m. Figure 5 compares the valves for a length of 3 m, considering an inlet pressure of 6 bar, which are typical values for domestic appliances operating with HC-600a (Espíndola *et al.*, 2016 e Colombo *et al.*, 2016).

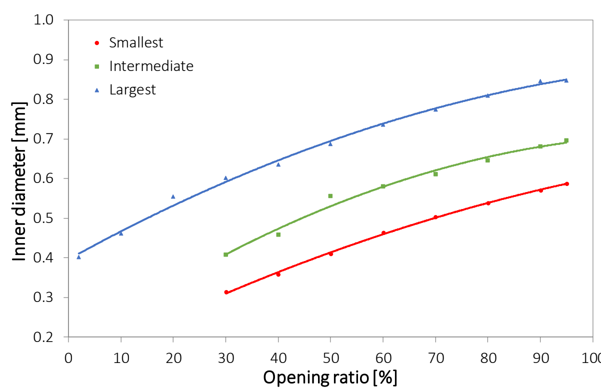

Figure 5: Equivalent capillary tubes

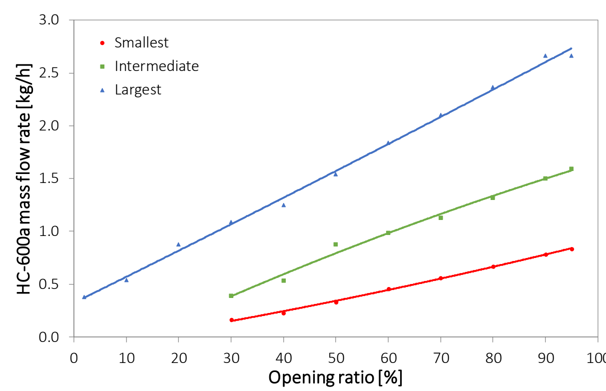

Figure 6: HC-600a mass flow rate predictions

Note that the behavior of the intermediate valve is similar to that of the smallest valve, whereas the largest valve is equivalent to capillary tubes up to 0.9 mm in diameter, depending on the operating conditions. For example, for a 90% opening ratio, the smallest and intermediate valves respectively produce a mass flow equivalent to 0.57 and 0.66 mm capillary tubes, while the largest valve has flow rates equal to those of capillary tubes with 0.85 mm inner diameter. Hence, one can affirm, at least preliminarily, that the valves in question have the potential to be applied in domestic refrigeration systems, since the range of capillary tubes found matches the geometries observed in practice (Melo *et al.*, 2002).

In order to provide a more realistic character to the previous evaluation, an additional analysis was performed by calculating the HC-600a mass flow rate with Eq. (2) based on the geometries of the equivalent capillary tubes previously obtained. Figure 6 shows the results for condensing and evaporating pressures respectively of 6 and 0.5 bar and a sub-cooling degree of 10 °C. If the mass flow rate values are multiplied by an enthalpy difference, the system cooling capacity can also be estimated. Defining then the outlet enthalpy based on the evaporating pressure assuming the compressor suction temperature equal to 32.2 °C (Jähnig *et al.*, 2000) and the inlet enthalpy based on the sub-cooling liquid at the condenser outlet, the cooling capacity can be calculated in a similar manner to what is done in compressor calorimetry. The conversion of the results presented in Fig. 6 reveals that the smallest valve can provide relatively low capacities, between 20 and 70 W, being attractive for some household refrigerators, specially the most efficient ones. The intermediate valve can provide capacities up to 100 W, but only for opening ratios of more than 80%. The largest valve, in turn, practically covers the application range of the previous valves generating cooling capacities between 20 and 180 W for opening ratios between 2 and 95%. According to Ronzoni *et al.* (2013) and Hermes *et al.* (2013), several domestic refrigerators operate with a cooling capacity close to 100 W. Therefore, it can be said that the largest valve is the most promising, since it covers a wide range of capacities and can be continuously controlled throughout its entire opening range.

Nevertheless, it is important to emphasize that the nitrogen results are preliminary and only deal with an estimate of the valves real behavior. Thus, tests with refrigerant fluid were conducted with the most promising valve in an attempt not only to validate such results but also to elucidate the valve behavior. Figure 7 presents the HC-600a mass flow rate as a function of the opening ratio and the condensing temperature for a constant evaporating temperature of -28 °C. By tracing a vertical line along a given opening ratio the points for different sub-cooling degrees (5, 10 and 15°C) can be noted. It is observed that such points are very close to each other, indicating the valve is little affected by the sub-cooling degree at the inlet. This can be explained by the weak dependence of the mass flow rate on the fluid density (as suggests the orifice equation) and by the small variation of density for the temperature range analyzed. It should be noted that the points for the evaporating temperature of -10 °C were not shown because the mass flow values were very similar to those of -28°C, since the flow was choked. Figure 8 shows such behavior for an opening ratio of 50%, a condensing temperature of 45 °C and a sub-cooling of 10 °C. It is observed that the mass flow rate remained practically constant with the variation of the evaporating temperature between -10 and -35 °C.

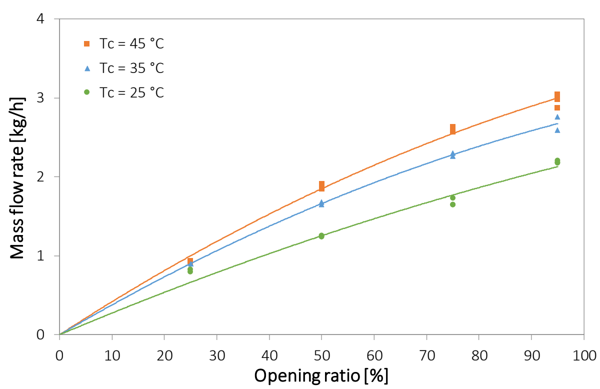

Figure 7: Measured HC-600a mass flow rate

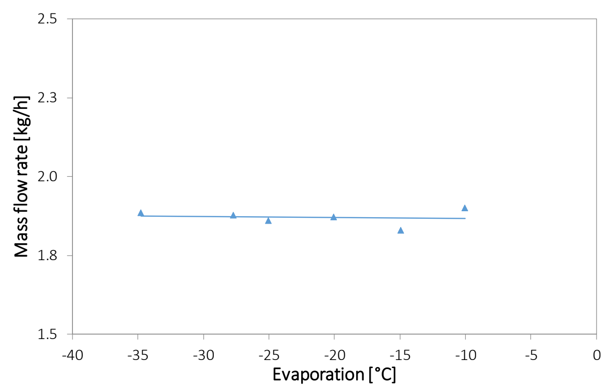

Figure 8: Evaporation effect on the mass flow rate

Based on the experimental values of the HC-600a mass flow, the estimates obtained from the conversion of nitrogen flow into equivalent capillary tubes can be validated. The results of the validation are shown in Fig. 9. It should be noted that the comparison takes into account only the inlet pressure and opening ratio combinations tested with HC-600a that are within the range of the tests with nitrogen. It is observed that the vast majority of the data is concentrated within a range of ±20% error, and that this range is due in part to the differences between the expansion devices, both with respect to the physics of the flow and the constructive characteristics aspects of each.

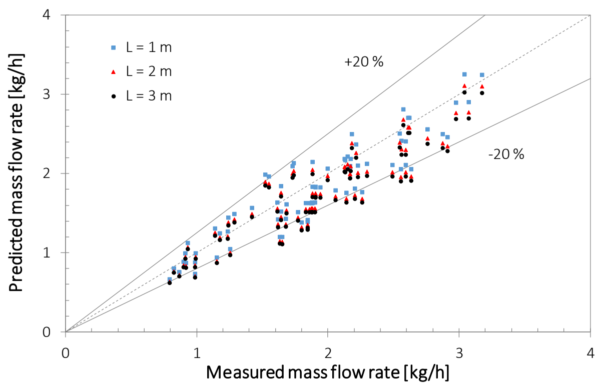

Figure 9: Predicted vs. measured HC-600a mass flow

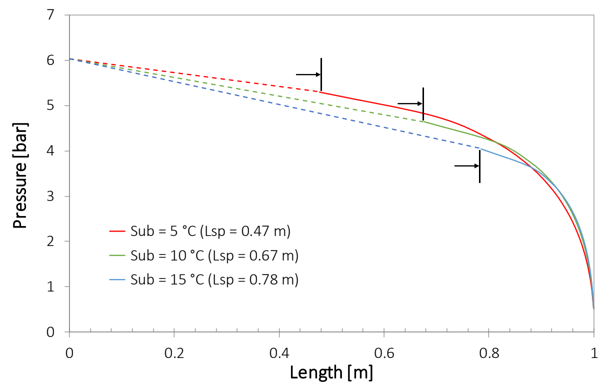

Figure 10: Sub-cooling effect on pressure profile

In general, larger errors were verified for lower degrees of sub-cooling. Capillary tubes are more sensitive to sub-cooling, as can be shown by the mathematical model and the data of Melo *et al.* (1999), but the valve is not. In capillary tubes, the effects of friction are dominant in relation to those of acceleration, since their lengths are some orders of magnitude higher than those of valves. As a result, the flash point moves inside the tube, causing the portion of two-phase fluid to be larger or smaller because of the degree of sub-cooling at the inlet. Figure 10 shows results generated with the model to illustrate the effect of sub-cooling on the pressure profile along a tube of 1 m in length and inner diameter of 0.55 mm, for an inlet pressure of 6.04 bar and outlet of 0.515 bar. Note the variation of the length occupied by single phase fluid (L_{sp}) and, consequently, the change of the flash point. Pressure drop in valves, in turn, occurs almost locally, differently from that observed in capillary tubes. In valves the friction effect is insignificant, given the millimetric passage length, and is masked by the preponderant acceleration effect. When fluid enters the valve, the inlet loss caused by abrupt section change and friction within the valve body is insufficient for flashing to occur inside the valve. This leads to the formation of a jet of metastable superheated liquid of conical shape at the exit of the valve and the phase change occurs on its surface (Simões and Bullard, 2003). The flashing then occurs from the throat of the valve. An evaporation wave is formed and the flow becomes supersonic. The two-phase refrigerant is accelerated and encounters a shock wave, responsible for raising the pressure so that it matches the evaporation pressure of the cycle (Liu *et al.*, 2008).

It was also observed that the scattering slightly decreased when the length of the equivalent capillary tube used to estimate the HC-600a mass flow rate was shorter. In fact, the shorter the capillary tube length the more like a valve it becomes. However, the errors do not decrease continuously with the reduction of the length. For lengths less than 1 m, the trend reverses and the errors raise again, since both Kipp and Schmidt equation and capillary tube mathematical model are out of the validity range.

Boeng (2012) also adopted a similar approach to find which capillary tubes would be equivalent to the pair valve-capillary tube in series that was used during his experiments. The refrigerator capillary tube had an inner diameter of 0.83 mm and a length of 3.32 m, while the valve had an orifice of 3.25 mm with 12 opening settings. In order to convert the arrangement into an equivalent capillary tube, tests with dry nitrogen were performed and the Kipp and Schmidt equation was applied. Subsequently, steady state energy consumption tests (Hermes *et al.*, 2013) were conducted with the refrigerator and the pressure and mass flow rate measurements were applied in a mathematical model to estimate again the inner diameter of the equivalent capillary tube. The diameter estimates based on nitrogen tests were then compared to those based on HC-600a tests and errors within a ±3% band were observed, which correspond to approximately 7% in mass flow rate (see Eq. 2). This range is reasonably narrower than that observed in the present work and the explanation is analogous to that presented above. Boeng (2012) had a capillary tube together with a valve and this was significantly large. Thus, the capillary tube effect in the arrangement was dominant and the conversion was practically between capillary tube and capillary tube, not between valve and capillary tube.

Finally, it is also expected that the errors are associated with uncertainty of the calculation procedure. Both Kipp and Schmidt and capillary tube model are not perfect. The model of Hermes *et al.* (2010) was validated within a ±10% error band. The Kipp and Schmidt equation, in turn, even with the constants refitted by Montibeller *et al.* (2016), presents only 59% of the points with deviations between ±5% and 79% between ±10%.

4. CONCLUSIONS

The literature review showed that there are very few studies with focus on variable expansion devices specifically manufactured for household refrigerators. Most of the available valves are sized for commercial and industrial refrigeration systems and present orifices larger than 1 mm (Ye *et al.*, 2007). The capillary tubes of domestic refrigerators, on the other hand, have an inner diameter between 0.55 and 0.85 mm (Hermes *et al.*, 2008). To meet the requirements of a domestic appliance the valve orifice should then have an orifice smaller than that, considering that its passage length is way too short. To fill this gap, this work focused on an experimental analysis of three electronic valves with hydraulic diameters smaller than 0.19 mm that were firstly tested with dry nitrogen. With the obtained volumetric flow rate and the Kipp and Schmidt equation the equivalent capillary tube geometries were defined and used to estimate the HC-600a mass flow rate with the aid of a mathematical model. To check these estimates and to characterize the HC-600a flow through the most promising valve, a test bench able to reach condensing temperatures as low as 25 °C was specifically built. Results showed that the largest orifice EEV is capable of supplying the pressure differences typically found in real appliances, without a capillary tube in series. Also, it was found that the valve is almost insensible to sub-cooling variations and that the flow is choked under several practical conditions. When comparing the nitrogen-based mass flow rate estimates with the values obtained with the HC-600a test bench, a 20% error band was found, which can be considered good if the idea is to obtain a preliminary estimative of the valve application envelope. However, this analysis cannot completely replace the conduction of tests with refrigerant fluid and this is related to the differences between the capillary tube and valve physics. The friction effect is much more intense in a capillary tube than in a valve, for instance. Due to its millimetric length, the valve acts like a restricted orifice and the pressure drop happens almost locally. In this case, the acceleration effect is dominant. Also, the uncertainties associated with both Kipp and Schmidt equation and the capillary tube mathematical model contributed to increase the error.

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