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Non-Condensable Gases and their Effect on the Dynamic Behavior of Domestic Refrigerators

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Non-condensable Gases and Their Effect on the Dynamic Behavior of Domestic Refrigerators

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ABSTRACT

The aim of this work was to experimentally investigate the effect of non-condensable gases on the thermal acoustic behavior of domestic refrigerators under dynamic conditions. To this end, an acrylic made filter dryer was installed in the system for a proper visualization of the refrigerant flow at the capillary inlet. An accelerometer was also installed at the evaporator inlet to capture vibrations/noise signals, caused by the refrigerant flow pattern at the inlet of the expansion device. During the experiments the energy consumption was measured with the system systematically doped with precise amounts of nitrogen. It has been found that the energy consumption increased up to 25.9%, when a mass fraction of 0.43% of N_2 was added to the system. Moreover, it has been found that the energy consumption increases almost linearly with the N_2 mass fraction. Pull-down tests were also carried out, showing a clear relationship between the N_2 amount and the power and mass flow rate oscillations, soon after the compressor start-up, caused by the unbalance between the compressor and capillary tube mass flow rates. It could also be observed that the expansion noise increased when liquid was admitted in the capillary, due to the increased mass flow rate.

KEYWORDS: domestic refrigerator, non-condensable gases, expansion noise, energy consumption

1. INTRODUCTION

The inherent variability of the evacuation process in the production line can eventually leave some residual air inside the refrigeration circuit. As air is composed mostly by nitrogen and oxygen, gases with a very low boiling point, it does not condense and remains in the vapor phase throughout the whole circuit. The non-condensable gases (NCG) can adversely affect the system operation (Arora, 2012), increasing energy consumption and noise levels, and therefore must be carefully investigated.

The system penalization due to NCG depends on the refrigerating circuit. If a liquid receiver is located at the condenser outlet, a liquid seal is formed and the NCG become trapped inside the condenser. The area occupied by these gases becomes unavailable for heat transfer and the internal heat transfer coefficient is reduced. In order to overcome these drawbacks, the temperature difference between the refrigerant and the air must increase, leading to a higher discharge pressure and compressor power, resulting in a lower coefficient of performance.

In the absence of a liquid receiver at the condenser outlet, however, no trap is formed and the NCG can freely circulate throughout the circuit. A gaseous mixture of refrigerant and NCG can then enter the capillary tube and, since the temperature of the suction line in contact with the capillary tube is not sufficiently low, the bubbles cannot collapse. This effect partially clogs the capillary tube, causing fluctuations in the flow and reducing the average refrigerant mass flow rate. Furthermore, the discharge pressure increases due to the addition of the partial pressure of NCG to the condensing pressure. The combination of these factors leads to a decrease in the system coefficient of performance.

Very few reports were found in the open literature on this topic. Cecchinato *et al.* (2007) studied the effect of NCG in household refrigeration systems by using a very simple device to inject nitrogen into the refrigerating circuit. Tests were carried out with an all-refrigerator and a chest freezer while thermal and electrical parameters were monitored.

The results showed that the system penalization due to the NCG was more related to the clogging effect in the capillary tube than due to a reduction of the heat transfer rate in the condenser. However, the authors did not analyze the flow pattern at the capillary tube inlet and the impact on the expansion noise. Espíndola *et al.* (2016) experimentally investigated the thermal-acoustic behavior of a domestic bottom-mount refrigerator contaminated with non-condensable gases under steady state conditions. The authors developed a more accurate device to inject very small amounts of nitrogen into the refrigerating circuit. An accelerometer was strategically installed at the evaporator inlet in order to capture the vibration on the tube wall caused by the internal flow. Simultaneously, images of the flow at the capillary tube inlet were recorded by a high-speed camera using an acrylic-made filter dryer. It was concluded that for very small amounts of nitrogen, the system performance was not penalized because the capillary tube was probably oversized. Therefore, the admission of the gaseous mixture (NCG + refrigerant fluid in vapor phase) increased the capillary tube restriction, and the system even performed slightly better. However, for higher concentrations of nitrogen, the system performance was negatively affected and large fluctuations in the flow pattern could be visualized, alternating between moments with mostly vapor entering the expansion device and moments with mostly liquid.

The literature review showed that the effects of NCG on the start-up and on-off cyclic operation of the compressor have not yet been systematically investigated. In this context, the aim of this work was to extend the analysis of Espíndola *et al.* (2016) by evaluating the influence of non-condensable gases on the thermal-acoustic behavior of domestic refrigerators under dynamic conditions. To this end, standardized energy consumption (ISO 15502, 2005) and pull-down tests were conducted with a typical frost-free appliance. Small amounts of nitrogen were injected into the refrigerating circuit with the aid of a purpose-built doping device and the resulting energy consumption was correlated with the contamination level. In addition, an acrylic filter dryer was used and an accelerometer was installed at the evaporator inlet in order to correlate the flow pattern at the capillary tube inlet with the expansion noise. Further details are presented below.

2. EXPERIMENTAL APPROACH

A purpose-built doping device was developed to accurately inject small amounts of nitrogen into the refrigeration circuit (see Figure 1). The injection process comprises the following steps: first, the apparatus must be evacuated and then valve V_1 must be opened to inject nitrogen into the device, until a predefined initial pressure is reached. Then valve V_1 must be closed and after a steady-state condition is achieved the initial pressure and temperature inside the device are recorded (p_i and T_i, respectively). After that, valve V_2 must be opened in order to release nitrogen to the refrigeration system until the desired final pressure is reached. Then, valve V_2 must be closed and the final pressure and temperature are recorded (p_f and T_f, respectively). The mass of nitrogen injected (m_{nit}) is calculated by Equation (1), assuming an ideal gas behavior, where R is the specific gas constant of nitrogen and V_{dev} stands for the inner volume of the device (previously measured by a similar method, Gonçalves, 2004). This device is capable of injecting nitrogen in the order of milligrams with a maximum uncertainty of ±5%.

$$m_{nit} = \left(\frac{p_i}{T_i} - \frac{p_f}{T_f} \right) \frac{V_{dev}}{R} \quad (1)$$

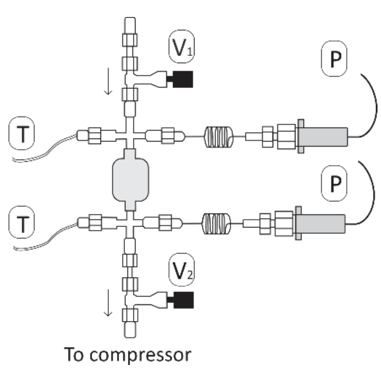

Figure 1: Doping device

A bottom-mount frost-free refrigerator with a 120-liter freezer placed below a 302-liter fresh-food compartment was used in this work (see Figure 2). The refrigeration system has a reciprocating variable-speed compressor and operates with 56 g of isobutane. The evaporator is of the finned-tube type, subjected to forced convection by an axial fan, which blows the total airflow into a plenum, where a damper directs part to the freezer and part to the fresh-food compartment. The condenser is of the wire-and-tube type, with 25 rows, located at the refrigerator rear wall.

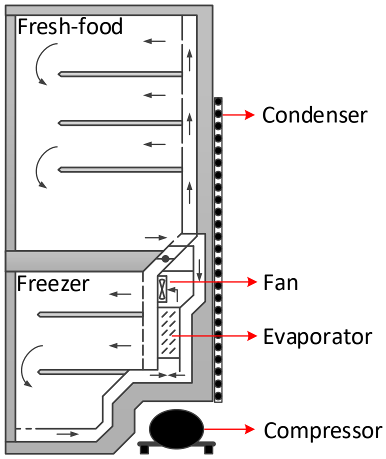

Figure 2: Refrigerator

As one of the objectives of this work was to evaluate the flow pattern at the capillary tube inlet, the original filter dryer was replaced by an acrylic-made replica (see Figure 3), and a high-speed camera was used to record the images. In addition, an accelerometer was strategically installed at the evaporator inlet (see Figure 4). This allowed correlating the expansion noise at the capillary tube outlet with the flow pattern at the capillary tube inlet.

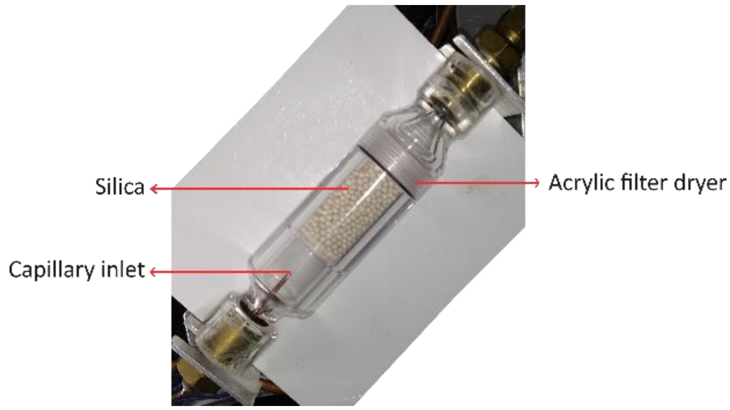

Figure 3: Acrylic filter dryer

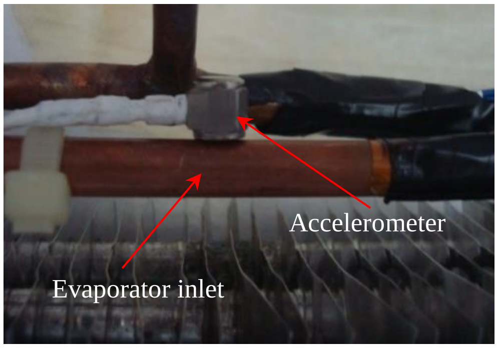

Figure 4: Accelerometer

The refrigerator was fully instrumented. The air-side temperatures were measured by T-type thermocouples (uncertainty of ±0.2 °C), five were placed in the fresh-food compartment (two in the cellar compartment and three in the geometric center of the shelves) and three thermocouples in the freezer compartment. To monitor the refrigerant pressure, absolute pressure transducers (uncertainty of ± 0.02 bar) were installed at the compressor suction and discharge ports. The refrigerant mass flow rate was measured by a Coriolis-type mass flow meter installed at the compressor discharge, with a maximum uncertainty of ±0.1% of the readings. The electrical parameters, such as voltage, current and power were measured by transducers with an uncertainty of ±0.1% of the readings. For the standardized energy consumption tests (ISO 15502, 2005) the freezer compartment was filled with tylose packages according to the storage plan provided by the manufacturer.

The energy consumption tests were carried out with the following nitrogen mass fractions: 0 (baseline), 0.09%, 0.17%, 0.26% and 0.43%. The ambient temperature was kept at 32°C in all tests. Two runs were conducted for each nitrogen contamination, one with the compressor speed constant and equal to 3000 rpm and another with the compressor speed varying automatically according to the inverter original logic. Pull-down tests with the compressor speed at 4500 rpm and the thermostat deactivated were also carried out at the ambient temperatures of 25 and 43°C. The following nitrogen mass fractions were evaluated: 0, 0.22% and 0.43%.

3. RESULTS AND DISCUSSIONS

The results of the energy consumption tests are shown in Tables 1 and 2. It can be noted that the system performance was negatively affected by the presence of non-condensable gases, even at very low concentrations. When the compressor speed was kept at 3000 rpm, the energy consumption increased by 13% for a nitrogen mass fraction of 0.43%, which was related to a reduction in the refrigerator cooling capacity. As already pointed by Espíndola *et al.* (2016), in small refrigeration systems, such as the present appliance, the NCG do not stay trapped in the condenser, but are free to circulate throughout the circuit. Therefore, periodically, a gaseous mixture composed by NCG and refrigerant fluid is admitted into the capillary tube. The boiling point of this mixture is so low that the vapor bubble cannot collapse and passes through the capillary tube in the vapor phase with a high specific volume. This is responsible for partially clogging the capillary tube, reducing the refrigerant mass flow rate, and consequently reducing the system cooling capacity. Thus, the compressor run-time ratio increases to reach the desired temperatures, which explains the increase in the energy consumption.

Table 1: Energy consumption tests at 3000 rpm

	Unit	**Baseline**	**Test 1**	**Test 2**	**Test 3**	**Test 4**
Nitrogen mass fraction	%	0.00	0.09	0.17	0.26	0.43
Run-time	%	72.5	78.0	82.2	83.5	92.3
Energy consumption (EC)	kWh/month	47.22	48.62	50.05	50.15	53.38

Table 2: Energy consumption tests with variable compressor speed

	Unit	**Baseline**	**Test 1**	**Test 2**	**Test 3**	**Test 4**
Nitrogen mass fraction	%	0.00	0.09	0.17	0.26	0.43
Average compressor speed	rpm	2100	2400	3000	3300	3900
Run-time ratio	%	75.1	79.6	79.3	79.1	81.9
Energy consumption (EC)	kWh/month	45.74	46.71	49.85	52.29	57.59

When the compressor speed was varying automatically according to the inverter logic, the impact in the energy consumption was even higher. The system energy consumption increased by 25.9% when the nitrogen mass fraction was 0.43%. To better understand this behavior, the test results were plotted in Figure 5. In the baseline tests (no nitrogen), the system energy consumption reduced 3.2% when the compressor speed was variable, due to a lower compressor average speed. However, as the nitrogen level increased, the system cooling capacity progressively decreased due to the clogging effect in the capillary tube. Therefore, the compressor speed automatically increased to overcome this effect. As can be seen in Figure 5, when the nitrogen mass fraction reached 0.17%, the average compressor speed during the test with variable speed was nearly 3000 rpm, and therefore the system energy consumption in both strategies was very close. From this contamination level on, the control logic kept increasing the compressor speed. Thus, a combined effect resulted in a stronger impact in the system energy consumption: (i) the reduced mass flow rate and cooling capacity led to a higher compressor run-time ratio and (ii) the higher average compressor speed led to a higher average compressor power. In both cases, it was observed an almost linear increase in the energy consumption with the nitrogen mass fraction.

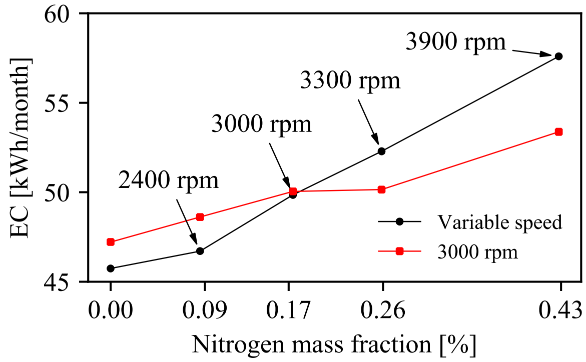

Figure 5: Overall energy consumption as a function of the nitrogen mass fraction

The pull-down tests results are shown in Table 3, where the pull-down time is defined as the time required for the average temperature of the air to reach -18°C in the freezer compartment and 5°C in the fresh-food compartment. As expected, the higher the ambient temperature, the longer the pull-down times. Furthermore, it was noticed that due to the reduction in the cooling capacity the pull-down times increased significantly when the system was contaminated, especially for the freezer compartment. When the nitrogen mass fraction was 0.22%, the freezer pull-down time increased 3.0 and 1.8h for ambient temperatures of 25 and 43°C, respectively. When the nitrogen mass fraction was 0.43%, the impact was more expressive, and the freezer pull-down time increased 8.3 and 9.8h. A similar behavior was found for the fresh-food compartment.

Table 3: Pull-down times

Nitrogen mass fraction [%]	Ambient temperature [°C]	Fresh-food → 5°C [h]	Freezer → -18°C [h]
0.00		2.3	2.7
0.22	25	3.0	5.7
0.43		4.7	11.0
0.00		4.2	5.4
0.22	43	5.0	7.2
0.43		8.0	15.2

In order to better clarify the system behavior on the pull-down tests, the average temperatures in the freezer and fresh-food compartments were plotted for the baseline test and the test with 0.43% of nitrogen mass fraction (see Figures 6 and 7). It can be seen that the slope of the curves and the steady state values are quite different when there is NCG, which indicates the reduction in the cooling capacity. The freezer temperature increased from -22.4°C to -18.7°C and the fresh-food temperature from -7.5°C to -4.0°C when comparing the baseline case to that with a nitrogen mass fraction of 0.43%.

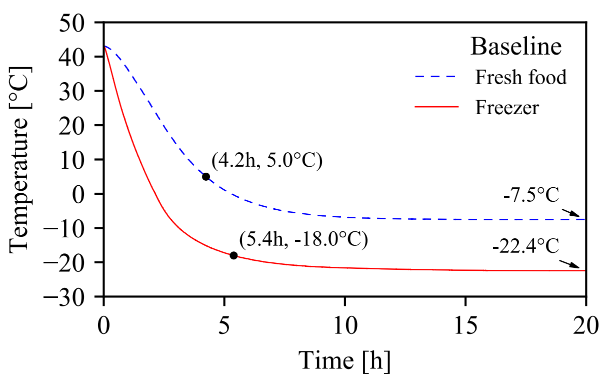

Figure 6: Baseline pull-down test at 43°C

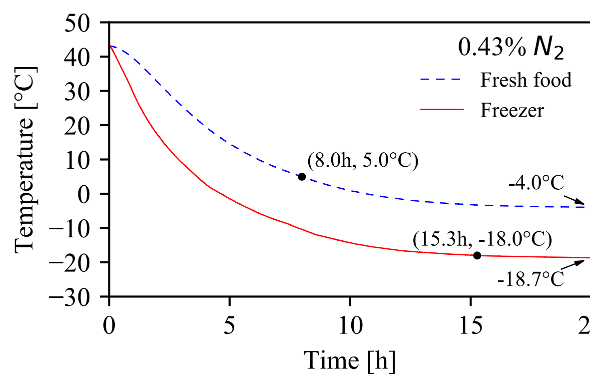

Figure 7: Pull-down test at 43°C with 0.43% of nitrogen mass fraction

Figure 8 shows the compressor power during the pull-down tests at 43°C for the three cases evaluated. It can be noted that after reaching steady-state conditions, the compressor power is lower for higher amounts of nitrogen. The explanation comes from the compressor power definition, which is the product of the specific work of compression and the refrigerant mass flow rate. The former tends to increase because the pressure ratio increases. Whereas, the latter tends to decrease due to the partial clogging effect in the capillary tube. This combined effect ended up slightly reducing the compressor power. However, as already seen in the energy consumption tests, the reduced compressor power did not lead to a reduction in the energy consumption, because the lower mass flow rate also contributed to a lower system cooling capacity, which increased the compressor run-time ratio.

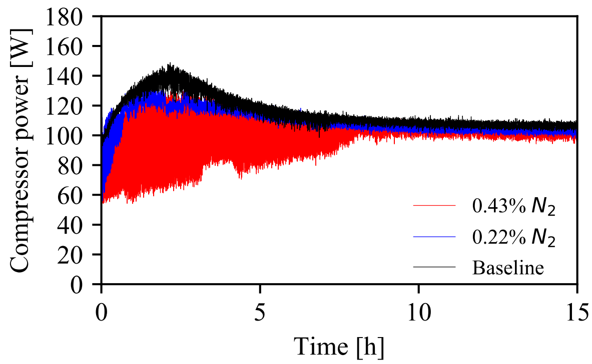

Figure 8: Compressor power during the pull-down tests at 43°C

An intriguing observation in Figure 8 is the large oscillations in the compressor power when nitrogen was added to the system, especially soon after the start-up. According to Hermes (2006), during the compressor start-up most of the refrigerant fluid accumulates at the high-pressure side of the circuit. As the refrigerant starts to change phase at the condenser, the mass flow rate through the capillary tube increases until it matches the compressor mass flow rate. This usually happens quickly in a non-contaminated system (Hermes and Melo, 1999). However, in the presence of NCG the system behaves quite differently. When the capillary tube gets partially clogged, the fluid migration is momentarily affected and the unbalance between expansion device and compressor mass flow rates is intensified. Since the compressor is still running and the amount of refrigerant at the evaporator is low, the suction pressure drops suddenly, as shown in Figure 9. As a consequence, the refrigerant specific volume at the compressor suction increases and the mass flow rate decreases (see Figure 10). Thus, the compressor power is also significantly reduced (see Figure 11). After a few seconds, however, the NCG bubbles move on and the capillary tube gets unclogged. Hence, the fluid starts to migrate to the evaporator and the suction pressure rises again. As shown in Figure 8, this intermittent behavior can last hours. The higher the contamination level the more expressive the effects on the compressor power. Figures 9 to 11 show the results for a nitrogen mass fraction of 0.43%, where oscillations from 0.23 and 0.44 bar in the suction pressure, from 0.8 to 1.8 kg/h in the mass flow rate and from 72 to 114 W in the compressor power were observed.

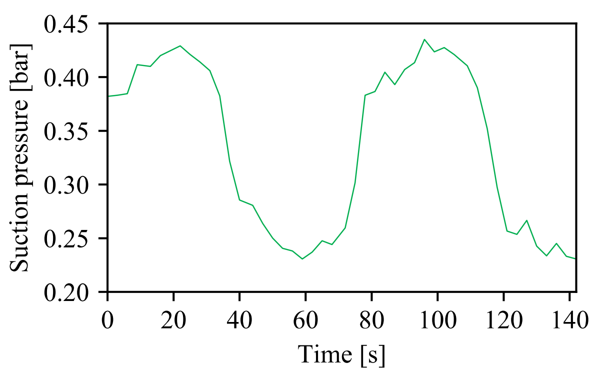

Figure 9: Suction pressure soon after compressor start-up

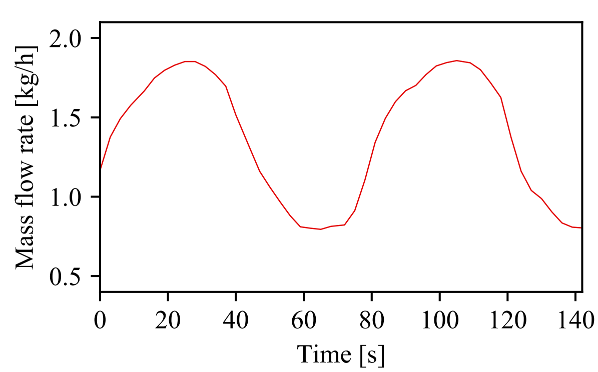

Figure 10: Mass flow rate soon after compressor start-up

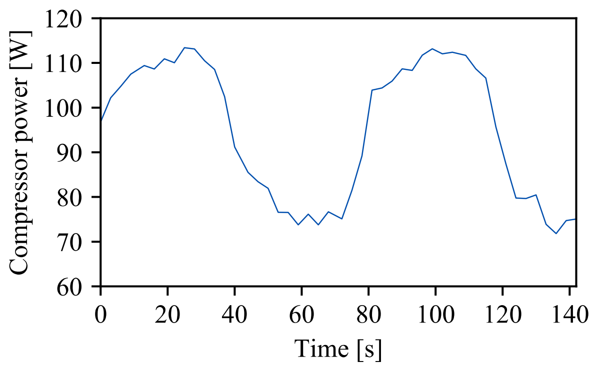

Figure 11: Compressor power soon after compressor start-up

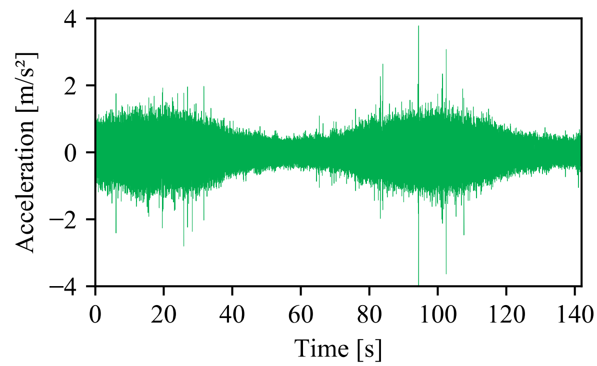

Figure 12: Accelerometer readings soon after compressor start-up

In order to provide more experimental evidence and corroborate the previous thermodynamic analysis, the flow pattern at the capillary tube inlet was registered and synchronized with the expansion noise measured at the evaporator inlet. Figure 12 shows the accelerometer readings and Figure 13 presents images of the refrigerant flow at the capillary tube inlet (access the video at https://youtu.be/8z1J2tVZjnY). It can be seen that the expansion noise also followed an intermittent pattern. At 10 and 100s the capillary tube was admitting mostly liquid, which led to a higher mass flow rate and also a more intense expansion noise. On the other hand, at 60 and 130s, the capillary tube was receiving mostly refrigerant vapor and NCG, which reduced the mass flow rate and the expansion noise.

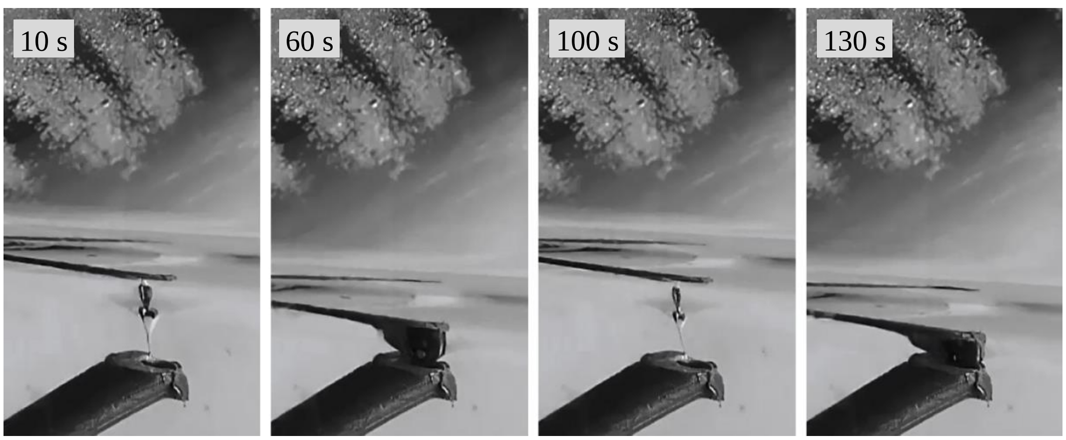

Figure 13: Visualization of the refrigerant flow at the capillary tube

5. CONCLUSIONS

This work addressed an investigation on the effect of non-condensable gases in the dynamic behavior of household refrigerators. Energy consumption tests were carried out according to the ISO 15502 (2005) standard and in each test different mass fractions of nitrogen were added to the refrigeration circuit by a purpose-built doping device. Two test runs were performed for each mass fraction, one with the compressor speed constant at 3000 rpm and another with the compressor speed varying automatically according to the inverter original logic. The results showed that the system performance was significantly affected by presence of non-condensable gases. For a N_2 fraction of 0.43%, for instance, the refrigerator overall energy consumption increased by 13% and 25.9% in the tests with constant and variable compressor speed, respectively, when compared to the baseline case. The energy consumption was higher in the case of variable speed due to the capillary tube partial clogging, which reduced the cooling capacity and increased the run-time ratio, and due to the inverter logic, which automatically increased the compressor speed in order to compensate for the lack of capacity. Pull-down tests were also carried out and it was verified that the time required to reach the desired air temperatures increased by up to 10 h when the nitrogen mass fraction was 0.43%, also due to the reduction in the system cooling capacity. Strong oscillations related to an unbalance between the compressor and capillary tube mass flow rates were observed in the suction pressure, the compressor mass flow rate and the compressor power, especially during the start-up transient. As the amount of refrigerant is still very low at the evaporator during the start-up, when the capillary tube clogs, the compressor evacuate the remaining fluid and the suction pressure decreases suddenly. However, when the capillary tube unclogs, the refrigerant migration to the evaporator is resumed and both suction pressure and mass flow rate rise again. It was found that this intermittent pattern can last for hours. Finally, the same pattern was observed when the flow images at the capillary tube inlet were synchronized with the accelerometer readings. As the capillary tube clogs, the mass flow rate decreases and so does the expansion noise. On the other hand, when there is mostly liquid at the capillary tube inlet, the mass flow rate increases and the expansion noise elevates.

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