

THE INFLUENCE OF NON-CONDENSABLE GASES ON THE THERMAL-ACOUSTIC BEHAVIOR OF HOUSEHOLD REFRIGERATORS

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Abstract. In refrigeration systems with evaporating pressure below atmospheric pressure, air from the external environment can infiltrate into the circuit through small leaks in the suction line. Additionally, if a problem occurs during the evacuation process on the production line, residual air might be left inside the circuit. This paper reports an experimental study on the influence of non-condensable gases on the thermal-acoustic behavior of a household refrigeration system. Controlled amounts of nitrogen were injected into the system through a purpose-built device. Steady-state energy consumption tests were carried out for each situation. Simultaneously, acceleration signals were monitored by an accelerometer installed at the evaporator inlet and videos of the flow pattern at the capillary tube inlet were recorded. The results show that, with very small amounts of non-condensable gases, the system performance was slightly improved. However, with large amounts, a worse performance was observed combined with large fluctuations in the flow pattern and the acceleration readings. In addition, for all cases it was noted that the sub-cooling degree increased with the amount of non-condensable gases.

Keywords: non-condensable gases, energy consumption, refrigerator

1. INTRODUCTION

In refrigeration systems with evaporating pressure below atmospheric pressure (e.g. refrigerators operating with isobutane), air from the external environment can infiltrate into the circuit through small leaks in the suction line. Additionally, if a problem occurs during the evacuation process on the production line, residual air might be left inside the circuit. Since air is composed mostly of nitrogen, a gas with a very low boiling point, it remains in the vapor phase throughout the refrigerant circuit, which means it does not condense. A non-condensable gas can adversely affect the system operation, reducing its energy efficiency and increasing the noise level generated by the flow.

According to Cecchinato *et al.* (2007), if a liquid receiver is located at the condenser outlet, a liquid seal is formed and all non-condensable gases become trapped inside the condenser. The area occupied by these gases in the condenser is not available for refrigerant heat transfer. In addition, the internal heat transfer coefficient is reduced, since the gas does not condense. In order to compensate for the reduction in the heat exchange area and the heat transfer coefficient, the temperature difference between the refrigerant and the air has to increase. This causes an increase in the discharge pressure and in the power required by the compressor, resulting in a lower coefficient of performance (*COP*).

In the absence of a liquid receiver at the condenser outlet, no trap is formed and the non-condensable gases can freely circulate through the circuit. A mixture of refrigerant and non-condensable gases can then enter the capillary tube and, since the temperature of the suction line in contact with the capillary tube is not sufficiently low, these vapor mixture cannot collapse. Thus, the capillary tube is choked, causing fluctuations in the flow and reducing the refrigerant mass flow rate. Furthermore, the discharge pressure increases due to the addition of the partial pressure of non-condensable gases to the condensation pressure. The combination of these factors reduces the system *COP*.

There are few reports in the open literature on this topic. In fact, the only study found was performed by Cecchinatto *et al.* (2007), who investigated experimentally the effect of non-condensable gases in household refrigeration by injecting small amounts of air into the refrigeration circuit. The doping procedure consisted of varying the air volume in the plunger of a syringe. An all-refrigerator and an upright freezer were tested under steady-state and cyclic conditions. However, the analysis was restricted to thermodynamic parameters. In this context, the aim of the study reported herein was to extend the analysis of the effect of non-condensable gases in household refrigeration by visualizing the flow pattern at the capillary tube inlet and capturing acceleration signals at the evaporator inlet. A variable speed compressor was used to allow operation under different conditions and create an imbalance between the compressor and the capillary tube. Furthermore, an accurate device to inject controlled amounts of nitrogen into the system was developed.

2. EXPERIMENTAL SETUP

2.1 Refrigerator

A bottom-mount frost-free refrigerator was used in the experiments, with a 120-liter freezer located below a 302-liter fresh-food compartment. The refrigeration system has a variable speed compressor and operates with 56 g of isobutane. The evaporator is of the finned-tube type, subject 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 on the refrigerator rear wall.

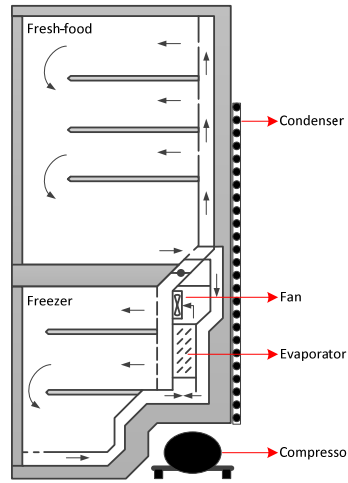


Figure 1. Refrigerator.

As previously stated, the flow pattern at the capillary tube inlet can be strongly affected by the presence of non-condensable gases. In order to visualize this effect, the original filter dryer was replaced by a replica made in acrylic (see Fig. 2), and images were recorded by a high-speed camera, model i-SPEED TR, manufactured by Olympus.

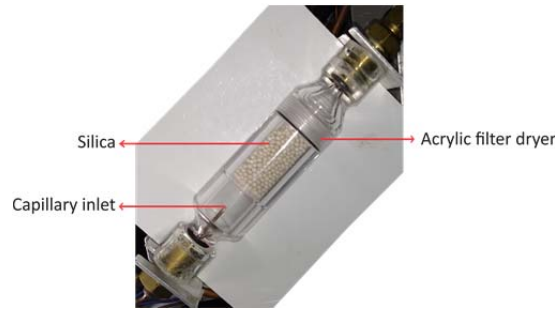


Figure 2. Acrylic filter dryer.

In order to obtain the temperature profile along the condenser, 25 T-type thermocouples (accuracy ± 0.2 °C) were installed on each row of the condenser. Since the presence of non-condensable gases tends to increase the sub-cooling degree at the condenser outlet, it is very important to monitor the whole extension. With regard to the air temperature monitoring, five thermocouples were installed in the fresh-food compartment (two in the cellar compartment and three higher up at the geometric center of the shelves) and three other thermocouples were uniformly placed in the freezer. The electrical measurements were taken with a wattmeter, model WT230 (accuracy $\pm 0.1\%$ of the reading), manufactured by Yokogawa.

To monitor the refrigerant pressure, pressure transducers (accuracy ± 0.01 bar), model PM3B, manufactured by HBM, were installed at the suction and discharge of the compressor, and at the filter dryer inlet. The sub-cooling (*SUB*) was calculated using the following equation:

$$SUB = T_{sat}(P_{dryer}) - T_{dryer} \quad (1)$$

where $T_{sat}(P_{dryer})$ is the refrigerant saturation temperature considering the filter dryer pressure and T_{dryer} is the refrigerant temperature in the filter dryer.

In addition, an accelerometer, model CCLD 4533-B, manufactured by Brüel & Kjær, was strategically installed at the evaporator inlet, in order to monitor the acceleration in the tube caused by flow oscillations at the capillary tube.

2.2 Doping device

A doping device was developed to accurately inject small amounts of nitrogen into the refrigeration circuit (see Fig. 3). The injection process comprises the following steps. First, the apparatus must be evacuated and then the valve V_1 is opened and nitrogen is injected into the device until the predefined pressure is reached. The valve V_1 is then closed and, after a steady-state condition is achieved, the initial pressure and temperature are recorded (p_i and T_i , respectively). In the next step, the valve V_2 is opened in order to release nitrogen to the system until the desired final pressure is reached. The valve V_2 is then closed and the final pressure and temperature are recorded (p_f and T_f , respectively). The nitrogen mass injected (m_{nit}) is calculated from Eq. 3, assuming an ideal gas behavior, where V_{dev} stands for the internal volume of the device (previously measured by a similar method) and R is the specific gas constant of nitrogen.

$$m_{nit} = \left(\frac{p_i}{T_i} - \frac{p_f}{T_f} \right) \frac{V_{dev}}{R} \quad (3)$$

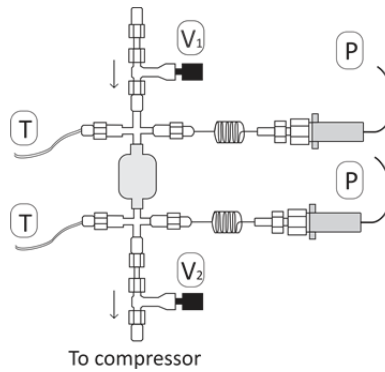


Figure 3. Doping device.

3. TESTING PROCEDURE

In order to evaluate the effect of non-condensable gases on the performance of household refrigerators, energy consumption tests were carried out according to the steady-state methodology proposed by Hermes *et al.* (2013). Reverse leakage tests (Gonçalves *et al.*, 2000) were carried out to determine the thermal conductances of the compartments (UA), parameters which are required to calculate the thermal load and the energy consumption. Further details are shown in the following sections.

3.1 Reverse heat leakage tests

Reverse heat leakage tests were carried out in a climate chamber at 15 °C. PID-driven electrical heaters were uniformly distributed inside the compartments in order to increase their temperatures, thereby creating a heat flux from the internal compartments to the external ambient. During the tests, only the fan was kept running. Six tests were performed, varying the damper positions (open and closed) and the temperatures of the fresh-food and freezer compartments, as shown in Tab. 1.

Table 1. Reverse heat leakage test data.

Damper	Test	T_a [°C]	T_{ff} [°C]	T_{ff} [°C]	W_f [W]	W_{ff} [W]	W_{ff} [W]
Open	1	15.0	54.7	47.7	8.6	42.0	9.4
	2	14.9	60.1	51.9	8.6	49.2	10.7
	3	15.2	59.9	60.0	8.5	24.8	44.4
	4	15.1	57.8	35.0	8.6	24.8	14.5
Closed	5	15.1	60.0	44.8	8.6	25.1	26.0
	6	15.0	39.9	60.0	8.7	5.0	48.0

The thermal conductances of the freezer and the fresh-food compartments (UA_{fz} and UA_{ff}) were calculated by adjusting the equation of the energy balance in the refrigerator, expressed by Eq. (4), based on experimental data shown in Tab. 1. Thermal conductances of 0.66 W/K and 1.03 W/K were obtained for the freezer and fresh-food compartments, respectively.

$$UA_{fz}(T_{fz} - T_a) + UA_{ff}(T_{ff} - T_a) - (W_{f,e} + W_{fz} + W_{ff}) = 0 \quad (4)$$

where T_{fz} is the average temperature of air in the freezer compartment, T_{ff} is the average temperature of the air in the fresh-food compartment, T_a is the ambient temperature, W_{fz} is the heat generated in the freezer, W_{ff} is the heat generated in the fresh-food compartment and $W_{f,e}$ is the fan power.

3.2 Energy consumption tests

The energy consumption tests were carried out according to the steady-state methodology proposed by Hermes *et al.* (2013). In this methodology, the system is kept running under steady-state conditions through PID-driven electrical heaters strategically placed in the fresh-food and freezer compartments, as the excess of cooling capacity is overridden by heat generation inside the refrigerator. During the tests, the thermostat must be deactivated while the damper is fixed at a predefined position.

In order to determine the system energy consumption, the cabinet thermal load (Q_t) must first be calculated, through the following equation:

$$Q_t = UA_{fz}(T_a - T_{fz}) + UA_{ff}(T_a - T_{ff}) + W_{f,e} \quad (5)$$

The cooling capacity (Q_e) is then obtained by taking into account the thermal load and the power dissipated by the electrical heaters:

$$Q_e = Q_t + W_{fz} + W_{ff} \quad (6)$$

When the refrigerator operates according to a cycling pattern, all of the energy transferred into the refrigerated compartments during the whole cycle (time-on plus time-off) must be removed by the cooling system in the time-on cycle. Thus, the compressor runtime (τ), defined as the ratio between the time-on cycle and the total cycle time, can be derived from an energy balance over a whole cycle as follows:

$$Q_e t_{on} = [UA_{fz}(T_a - T_{fz}) + UA_{ff}(T_a - T_{ff})](t_{on} + t_{off}) + W_{f,e} t_{on} \quad (7)$$

resulting in,

$$\tau \equiv \frac{t_{on}}{t_{on} + t_{off}} \cong \frac{Q_t - W_{f,e}}{Q_e - W_{f,e}} \quad (8)$$

Hence, the monthly energy consumption (EC) can be calculated as follows:

$$EC = 0.72\tau(W_k + W_{f,e}) \quad (9)$$

where W_k is the compressor power and the 0.72 coefficient stands for the conversion factor from [W] to [kWh/month].

3.3 Acceleration signal and image recordings

After performing a steady-state energy consumption test, with all variables stabilized, data were acquired from the accelerometer for 30 s. Simultaneously, images of the flow at the capillary tube inlet were recorded by the high-speed camera at 180 fps. All of the data collected were then processed. The images were played at a rate of 30 fps, which provided a playback speed six times slower than real time and allowed easier analysis of the flow. The accelerometer data was then analyzed by plotting an acceleration versus time graph, and synchronized with the images recorded. The vibration recorded by the accelerometer at the evaporator inlet could thus be associated with the flow pattern at the capillary tube inlet.

4. RESULTS AND DISCUSSIONS

In all energy consumption tests, the ambient temperature was kept at 32 °C, the fresh-food compartment temperature at 5 °C and the freezer temperature at -18 °C. Tests were carried out at three different compressor speeds: 2500, 3000 and 4000 RPM and the results are shown in Tab. 1 to 3.

Table 2. Results obtained at 2500 RPM.

2500 RPM	Unit	Baseline	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
Nitrogen charge	mg	0	40	80	120	160	200	300
Nitrogen mass fraction	-	0	0.07	0.14	0.21	0.28	0.36	0.53
Compressor power	W	66.9	66.6	63.7	66.6	66.4	65.6	64.3
Cooling capacity	W	106.9	109.2	103.6	107.5	98.7	97.9	90.6
Run-time	%	63.1	61.6	64.4	62.7	68.9	69.0	74.9
Energy consumption	kWh/month	34.79	33.76	33.96	34.36	37.63	37.24	38.48
Discharge pressure	bar	6.17	6.32	6.34	6.64	6.76	6.94	7.23
Suction pressure	bar	0.63	0.61	0.58	0.60	0.60	0.58	0.56
Mass flow rate	kg/h	1.54	1.50	1.37	1.43	1.42	1.37	1.27
Sub-cooling	°C	1.6	7.2	11.3	13.0	14.7	16.3	17.9

Table 3. Results obtained at 3000 RPM.

3000 RPM	Unit	Baseline	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
Nitrogen charge	mg	0	40	80	120	160	200	300
Nitrogen mass fraction	-	0	0.07	0.14	0.21	0.28	0.36	0.53
Compressor power	W	81.1	78.6	75.7	72.6	75.9	74.3	72.9
Cooling capacity	W	121.0	117.5	113.1	109.8	106.4	103.2	101.5
Run-time	%	55.1	56.7	59.3	61.5	63.2	65.1	66.1
Energy consumption	kWh/month	35.99	36.00	36.36	36.41	38.91	38.50	39.15
Discharge pressure	bar	6.45	6.56	6.63	6.73	6.98	7.10	7.41
Suction pressure	bar	0.61	0.58	0.55	0.52	0.56	0.54	0.53
Mass flow rate	kg/h	1.78	1.68	1.56	1.44	1.52	1.46	1.37
Sub-cooling	°C	2.0	7.5	11.6	13.9	15.6	17.1	18.6

Table 4. Results obtained at 4000 RPM.

4000 RPM	Unit	Baseline	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
Nitrogen charge	mg	0	40	80	120	160	200	300
Nitrogen mass fraction	-	0	0.07	0.14	0.21	0.28	0.36	0.53
Compressor power	W	103.7	101.3	95.7	92.0	88.3	89.6	92.1
Cooling capacity	W	136.4	137.3	127.4	115.6	108.7	109.3	110.8
Run-time	%	48.5	48.0	52.1	57.9	61.7	61.4	59.8
Energy consumption	kWh/month	39.53	38.35	39.48	42.31	43.44	43.82	43.60
Discharge pressure	bar	6.81	6.94	6.96	7.00	7.08	7.29	7.77
Suction pressure	bar	0.54	0.53	0.49	0.46	0.43	0.44	0.46
Mass flow rate	kg/h	2.00	1.94	1.77	1.62	1.51	1.53	1.55
Sub-cooling	°C	2.8	8.0	12.2	15.3	16.8	17.9	20.1

As previously discussed, an increase in the nitrogen mass can choke or clog the capillary tube due to the entry of vapor streams. This contributed to flooding the condenser and led to an almost linear increase in the discharge pressure, as observed in Fig. 4a. As a consequence, the evaporator was starved of refrigerant and the suction pressure decreased.

As the compressor pressure ratio increases the power consumption is also expected to increase. However, due to the capillary tube clogging effect, a strong reduction in the mass flow rate was observed. On comparing the tests with 0.53% of nitrogen to the baseline tests, reductions of 17.5%, 23.0% and 22.5% in the mass flow rate can be observed at 2500, 3000 and 4000 RPM, respectively. Therefore, since the compressor power is dependent on both the pressure ratio and mass flow rate, a reduction in the compressor power was observed as the nitrogen mass fraction increased. On the other hand, as the mass flow rate decreased the cooling capacity also decreased. To compensate for this effect, the compressor has to run for a longer period, which explains the runtime increase.

The energy consumption is dependent on the aforementioned parameters. Figure 4b shows the energy consumption results at all compressor speeds. It can be noted that when the amount of nitrogen was very small the system was not

strongly penalized, in fact, there was even a slight improvement in the performance compared with the case with no nitrogen. Since the capillary tube is a fixed expansion device, it can operate better under some conditions than others. Therefore, depending on the operating conditions, a small amount of nitrogen can slightly increase the capillary tube restriction and increase the system performance. However, when the concentration of nitrogen exceeds acceptable limits, clogging of the capillary tube becomes detrimental. In the tests at 2500 and 3000 RPM, for example, when the nitrogen mass fraction was 0.28%, the energy consumption was almost 8% higher than that of the baseline test. At 4000 RPM, when the nitrogen mass fraction was 0.21%, the energy consumption was 7% higher than that of the baseline test.

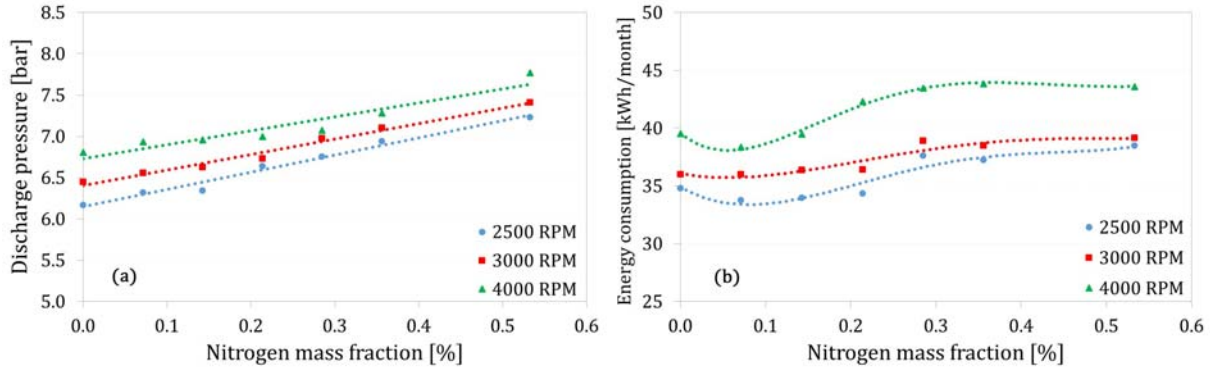


Figure 4. (a) Discharge pressure and (b) energy consumption versus nitrogen mass fraction.

It can be noted that the effect of non-condensable gases varies for different refrigeration system and components. On changing the compressor speed from 2500 to 4000 RPM, for example, the system was more affected at lower concentrations of nitrogen, which is consistent with the above statements. The higher the compressor speed, the lower the restriction of the expansion device must be in order to optimize the performance. Again considering that the capillary tube is a fixed expansion device, it seems to be better sized for 4000 RPM than for lower compressor speeds. Hence, the clogging of the capillary tube was more detrimental in this case. However, when higher concentrations of nitrogen were applied the system was penalized to a much greater extent at all compressor speeds.

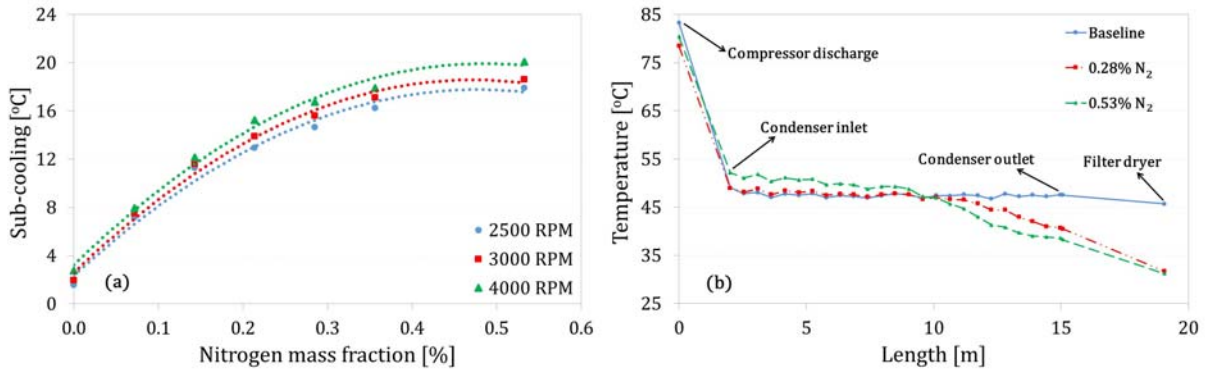


Figure 5. (a) Sub-cooling versus nitrogen mass fraction and (b) condenser temperature profile.

A high sub-cooling degree is a strong evidence of the presence of non-condensable gases, as shown in Fig. 5a. For the tests with 0.53% of nitrogen, for example, sub-cooling degrees of 17.9, 18.6 and 20.9 °C were measured at compressor speeds of 2500, 3000 and 4000 RPM, respectively. Figure 5b shows the temperature profile from the compressor discharge until the filter dryer inlet for three tests at 4000 RPM: baseline and 0.28% and 0.53% of nitrogen. It was observed that as the nitrogen mass fraction increased, the saturation temperature in the condenser increased. It was also noted that as the sub-cooling increases the area of the condenser used for latent heat transfer and the total heat transfer in this component are reduced.

Three different behaviors were observed in the analysis of the flow pattern at the capillary tube inlet. The first is shown in Fig. 7 for the baseline test at 2500 RPM. Even though the system was operating under steady-state conditions, a few fluctuations were observed at the capillary tube inlet. Most of the time, a high liquid level was observed with a vapor vortex entering the capillary tube (Fig. 7a). However, at times the vortex suddenly disappeared (Fig. 7b) and then reappeared within a very short time (Fig. 7c and 7d), causing the acceleration spikes shown in Fig. 6a.

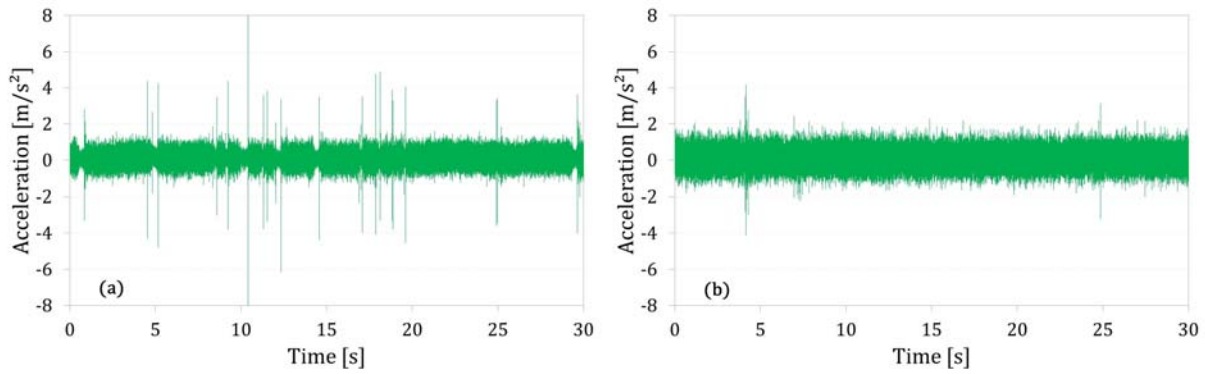


Figure 6. Acceleration data at 2500 RPM: (a) baseline test and (b) 0.07% of nitrogen.

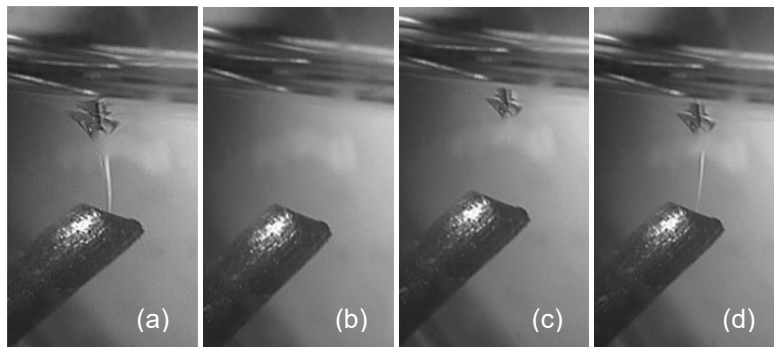


Figure 7. Filter dryer images: baseline test at 2500 RPM.

The second behavior is exemplified by the test with 0.07% of nitrogen at 2500 RPM. This case was very similar to the baseline, and there was predominantly liquid with a vapor vortex. The flow, in turn, was more constant and fluctuations were barely seen. Thus, a strong reduction in the acceleration spikes was registered (see Fig. 6b). A small amount of nitrogen optimized the capillary tube flow, which was not perfectly sized for this operating condition, as previously discussed.

The third case, represented by the test with 0.53% of nitrogen at 2500 RPM, occurred when the concentration of nitrogen exceeded acceptable limits. An intermittent acceleration pattern was registered, as shown in Fig. 8. Figure 9 shows the moments of 10 and 25 s, where it was observed that the liquid level was very low and there was predominantly vapor entering the capillary tube. This partially clogged the capillary tube and consequently reduced the acceleration. Higher accelerations, however, were captured when the liquid level was higher, as observed at 3 and 15 s. Therefore, a high concentration of nitrogen caused an imbalance between the mass flow displaced by the compressor and the mass flow entering the capillary tube, which reduced the refrigerator performance.

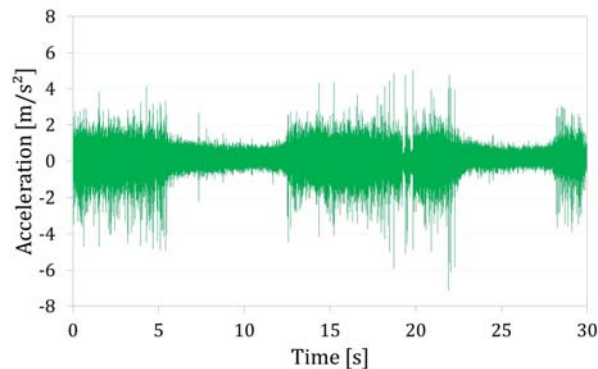


Figure 8. Acceleration data: 0.53% of nitrogen at 2500 RPM.

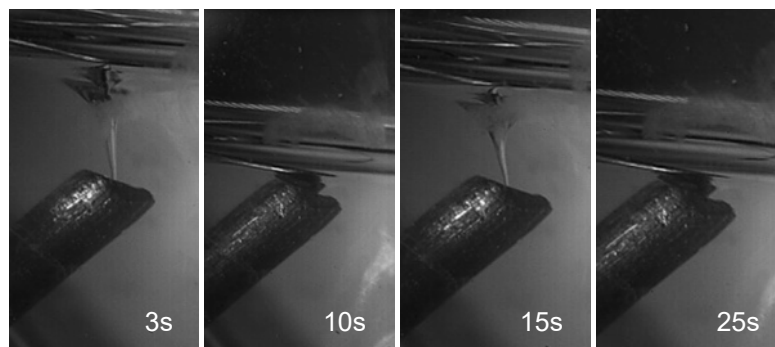


Figure 9. Filter dryer images: 0.53% of nitrogen at 2500 RPM.

5. CONCLUSIONS

In this study, the effect of non-condensable gases in household refrigerators was investigated through steady-state energy consumption tests by inserting different amounts of nitrogen in the refrigeration circuit. Additionally, the vibration on the tube caused by the internal flow was monitored by an accelerometer strategically installed at the evaporator inlet. Simultaneously, images of the flow at the capillary tube inlet were recorded by a high-speed camera using a filter dryer built in acrylic. In some cases, when the nitrogen mass fraction was very small, the system was not strongly penalized and even developed a slightly better performance. At these operating points, the capillary tube was probably not perfectly sized, so the entrance of a vapor mixture increased its restriction, and the system behaved better. However, when the concentration of nitrogen was too high, the system performance was negatively affected, and large fluctuations in the flow pattern could be visualized, alternating between moments with the predominance of vapor entering the expansion device and moments with the predominance of liquid. Finally, a higher sub-cooling degree than usual was found to be strong evidence of the presence of non-condensable gases.

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