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Popping Noise in Household Refrigerators: Fundamentals and Practical Solutions

Daniel Hartmann
melo@polo.ufsc.br

Claudio Melo

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Daniel HARTMANN, Cláudio MELO*

POLO Research Laboratories for Emerging Technologies in Cooling and Thermophysics
Department of Mechanical Engineering, Federal University of Santa Catarina
88040-900, Florianópolis, SC, Brazil, +55 48 3234 5691
* Corresponding author: melo@polo.ufsc.br

ABSTRACT

An intermittent noise that occurs around 30 seconds after compressor start-up has been disturbing customers of many domestic refrigerator manufacturers over the past 10 years. This noise is very similar to that produced by popcorn popping, being referred to as popping noise. The most probable cause of this strange and unpleasant noise is thought to be CIS (Condensation Induced Shock). The aim of this study was firstly to check the possible occurrence of CIS in a specific household refrigerator and, secondly, to identify ways to avert it without affecting the product performance. To this end some changes were made to the refrigerator: i) the diameter of the filter dryer was altered; ii) the slope of the filter dryer was altered; iii) the compressor mass flow rate was changed and iv) an additional internal heat exchanger was installed. The replacement of the filter dryer made the sound even more unpleasant, while the replacement of the original compressor by another of smaller capacity considerably reduced the noise. The noise was effectively eliminated only by placing the filter dryer horizontally or by installing an additional internal heat exchanger. Visualization studies were also carried out, making it clear that the occurrence of the popping noise is closely related to the flow pattern at the inlet of the capillary tube.

1. INTRODUCTION

For many years household refrigerator manufacturers have investigated an intermittent, high intensity noise that occurs soon after compressor start-up. The noise is very similar to that which occurs when popcorn pops in a metal pan, hence the term *popping noise*. The possible causes of the popping noise remained unknown until the behavior of a specific refrigerator was thoroughly studied (McLevige and Miller, 2001). The authors concluded that the most probable cause for this type of noise is CIS (Condensation Induced Shock) and suggested some alternatives to avert it. Most of the alternatives, however, increased the product energy consumption.

Martinez and Bringhenti (2011) carried out tests with a refrigerator in a semi-anechoic chamber. Figure 1 shows a spectrogram of the measured noise, a graph that condenses information with time on the horizontal axis, frequency on the vertical axis and sound intensity in a color scale. The pops that characterize the popping noise are shown in Figure 1 through the frequency peaks. It can be seen that the noise occurs in a very wide frequency range with mid-high intensities, which means that it as an intense and uncomfortable noise in terms of human perception. Figure 1 shows only the first 10 seconds after the first pop; however, this pattern repeats over 30 to 40 seconds.

1.1 Causes of the problem

The CIS occurs when a vapor bubble in the capillary tube is surrounded by sub-cooled liquid, kept colder than the vapor due to the heat exchanges with the suction line. If the vapor bubbles collapse sufficiently fast in the presence of liquid shockwaves are formed. Figure 2 illustrates this process.

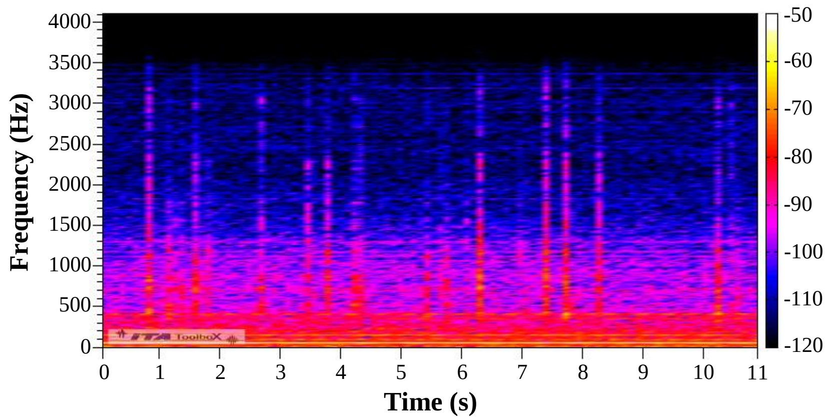


Figure 1: Spectrogram of the popping noise (adapted from Martinez and Bringhenti, 2011)

Soon after compressor start-up there is a substantial amount of liquid refrigerant in the evaporator. The quick and strong pressure drop caused by the compressor start-up moves a considerable amount of cold liquid through the suction line, inducing a temperature drop to the capillary tube. The vapor bubbles inside it quickly lose heat to the suction line, collapse and generate shockwaves. The shockwaves travel in both directions, to the condenser and to the evaporator. According to McLevige and Miller (2001), those that go to the condenser encounter other pockets of vapor and are dissipated; however, those that proceed to the evaporator find no restrictions, excite it and generate noise. Furthermore, vapor entrainment vortices are formed at the capillary inlet. Such vortices are dependent on the liquid level inside the filter dryer, as indicated in Figure 3. The most and least critical situations are shown in Figures 3C and 3D, respectively.

1.2 Currently employed solutions

McLevige and Miller (2001) suggested two ways of averting the popping noise: prevent bubbles from entering the capillary tube or reduce the cooling rate of this device soon after compressor start-up. To this end, refrigerator manufacturers adopted one of the following alternatives: i) a capillary tube-discharge line heat exchanger, ii) a small amount of nitrogen added to the system, and iii) a horizontal filter dryer. The first two alternatives completely eliminate the noise, but compromise the product energy consumption. The third does not affect the energy consumption, but is not effective in all refrigerators.

The capillary tube-discharge line heat exchanger is shown in Figure 4. It is mounted placing the initial portion of the capillary tube in contact with the discharge line. Since the discharge line is the hottest tubing of a refrigerator the use of this heat exchanger decreases the temperature difference between the vapor bubbles entering the capillary tube and the liquid surrounding them. Thus, there is a decrease in the bubble collapsing rate, which avoids the shockwaves and the noise. The effect of this heat exchanger is illustrated in the P-h diagram for HFC-134a in Figure 5, where the dashed lines represent the regular cycle and the solid lines represent the cycle with the above-mentioned heat exchanger. It is worth noting that the specific refrigerating effect and the cooling capacity decrease and the compressor power increases, all with a negative impact on the system performance.

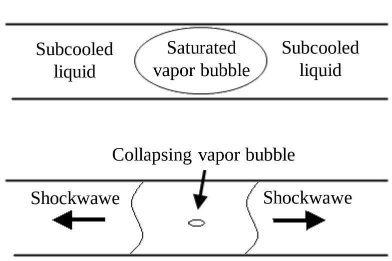


Figure 2: Illustration of CIS (McLevige and Miller, 2001)

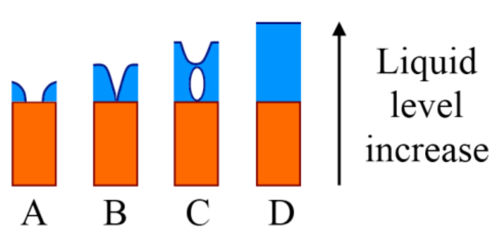


Figure 3: Liquid level variation and vortex formation at the capillary inlet (McLevige and Miller, 2001)

Figure 4: Capillary tube-discharge line heat exchanger

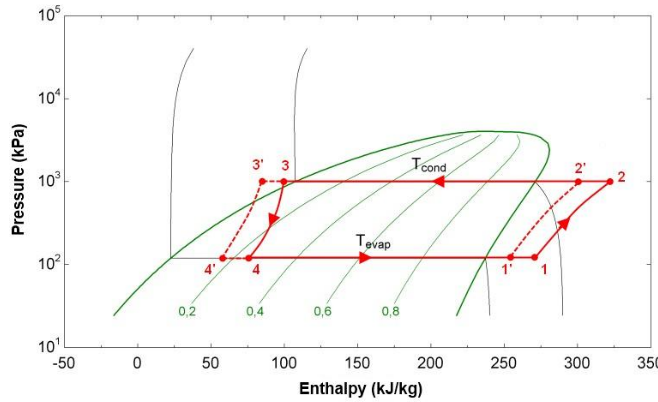


Figure 5: Cycles with (solid line) and without (dashed line) the capillary tube-discharge line heat exchanger

The saturation temperature of nitrogen at 900 kPa (usual condensation pressure at 25^{o}C in HFC-134-a refrigerators) is around -170 °C. Hence, the nitrogen will always remain as superheated vapor when mixed with the refrigerant in the refrigeration loop. After the compressor start-up the first vapor bubbles that enter the capillary carry some amount of nitrogen with them, and will only collapse at very cold temperatures, usually not found in refrigeration systems. The nitrogen, however, accumulates in the condenser and there exerts a partial pressure. The compression ratio is then increased, which increases the compressor power, decreases the cooling capacity and consequently decreases the system Coefficient of Performance (COP).

Placing the filter dryer in the horizontal position, as shown in Figure 6, is an attempt to increase the liquid level at the capillary inlet as fast as possible, in order to avoid the vapor entraining vortices. This solution, however, does not apply to all appliances, being very much dependent on the manufacturing process level, dryer geometry and refrigerant mass flow rate.

Figure 6: Filter dryer in original (left) and horizontal (right) positions

Although McLevige and Miller (2001) have studied this phenomenon in depth, solutions that completely eliminate the popping noise without compromising the energy consumption have not yet been found. The purpose of this investigation is, therefore, to experimentally verify the McLevige and Miller's (2001) findings and also to propose a definitive solution to this problem.

2. THERMODYNAMIC CHARACTERIZATION OF THE NOISE

A bottom-mount, frost free, French door refrigerator was the subject of this study. A total of 22- type-T thermocouples were installed along the refrigeration loop. Two absolute pressure transducers were also used, one at the dryer inlet and another at the evaporator inlet. An accelerometer was also used to measure the vibration induced by the refrigerant flow in the internal heat exchanger region, as shown in Figure 7. Equations (1) and (2) were used to estimate the compressor mass flow rate and power, respectively. Both calculations were carried out based on the pressure and temperature measurements at the compressor suction and discharge. The volumetric and overall compressor efficiencies were obtained from the compressor datasheet. The compressor power was measured using a power transducer and the readings were compared to the results provided by Equation (2) in order to validate the mass flow rate estimates.

$$\dot{m} = \frac{V f \eta_V}{v_1} \quad (1)$$

$$\dot{W}_C = \frac{\dot{m}(h_2 - h_1)}{\eta_G} \quad (2)$$

Figure 7: Accelerometer positioned at the top of the freezer liner

Figure 8: Sight glass at the capillary inlet

Visualization studies were also carried out, using a cylindrical sight glass with the same internal volume of the original filter dryer, as illustrated in Figure 8. The flow images at the capillary inlet were taken by a high speed video camera, allowing comparisons with the flow patterns shown in Figure 3.

The internal heat exchanger of this particular refrigerator is in contact with the upper wall of the freezer liner. Since the shockwaves are generated in the internal heat exchanger it is expected that the vibrations are stronger in regions closer to this component. The accelerometer was positioned according to this assumption (see Figure 7). In fact, the popping noise is nothing more than the sound that results from the impact of the heat exchanger on the upper wall of the freezer liner (Martinez and Bringhenti, 2011). It is important to note that all the experiments carried out in this study, except for the energy consumption tests, were performed under the ambient conditions of 25 °C / 50% relative humidity.

Figure 9 shows the fluid temperatures at the compressor suction and at the capillary inlet, the saturation temperature at the filter dryer and the compressor mass flow rate estimated by Equation (1), for the first 100 seconds after compressor start-up. The region bounded by the gray rectangle marks the length of the noise, 29 seconds for the baseline system. Figure 10 shows the accelerometer readings from the moment at which the first pop was captured. Figure 9 also shows that that the noise started 31 seconds after compressor start-up, when the green and red lines cross each other. The intersection of these two lines indicates the moment at which liquid starts to form in the dryer, *i.e.*, the first vapor bubbles enter the capillary.

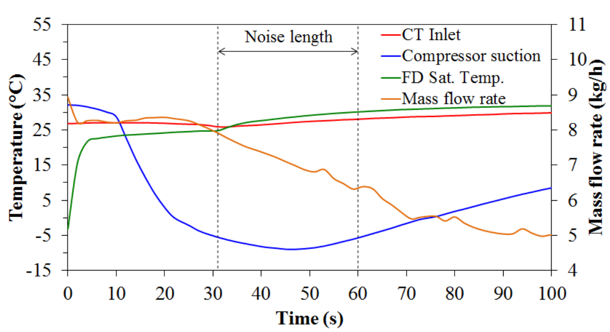


Figure 9: Thermodynamic readings: baseline system

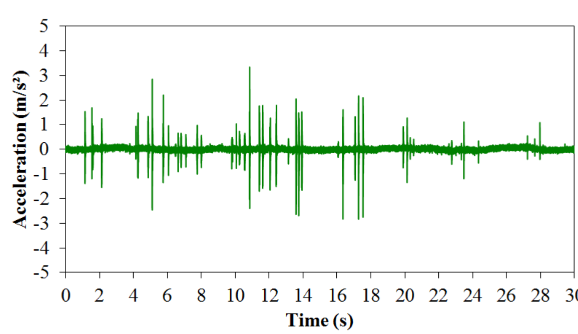
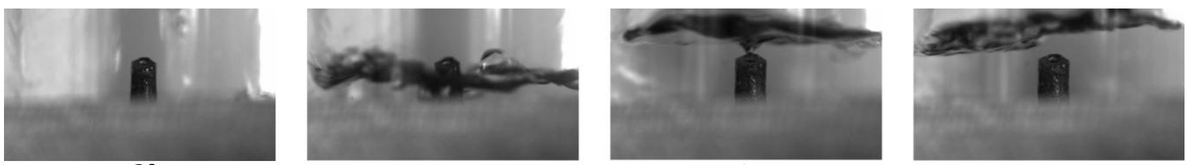


Figure 10: Accelerometer readings: baseline system

It should be noted that at the very first moment in which the noise was captured the temperature at the compressor suction was around -5 °C. This point of temperature measurement is the hottest along the suction line, in other words, the remainder of this tubing, including the portion that is in contact with the capillary tube, was even colder. Moreover, the compressor mass flow rate at this moment was relatively high, between 8.0 and 6.5 kg/h, which contributed to a high cooling rate of the capillary tube. The subcooling and, consequently, the amount of liquid in the dryer increased over time. However, over the popping period, the liquid level was not sufficiently high to avoid vapor bubbles from entering the capillary. These findings are very similar to those obtained by McLevige and Miller (2001), leading to the conclusion that, in this particular refrigerator, the CIS occurs and generates the popping noise.

Figure 11 shows the liquid level behavior in the dryer a few instants after compressor start-up. There clearly is a resemblance with the illustrations in Figure 3, particularly between the condition shown in Figure 3C and the image taken at 31 seconds. The images also corroborate the behavior predicted by the thermodynamic analysis. Soon after compressor start-up the liquid level increases rapidly until it reaches the edge of the capillary tube. At this moment the capillary tube mass flow rate begins to equal the compressor mass flow rate, and the liquid level exceeds the capillary edge by a few millimeters, reaching a steady condition at 70 seconds. Although there are bubble entraining vortices up to this instant the noise ceased 10 seconds prior to this, most probably because the capillary cooling rate was not high enough to cause CIS.



t = 20 s t = 25 s t = 31 s t = 70 s

Figure 11: Liquid level in the glass sight: baseline system

2.1 Energy consumption of the baseline refrigerator

The main goal of this study is to eliminate the popping noise without compromising the product energy consumption. Therefore, a baseline energy consumption test was carried out for comparative purposes. The test, which provided an energy consumption of 486.4 kWh/year, followed the ANSI/AHAM HRF-1 (2004) standard recommendations leaving aside the effect of the defrost heater.

3. PROPOSED ALTERNATIVES AND RESULTS

As suggested by McLevige and Miller (2001) the filter was firstly placed horizontally, but other possibilities were also explored. In addition to the acoustic and thermodynamic measurements, visualization studies were also carried out in order to better understand the phenomenon investigated.

3.1 Horizontal filter dryer

These test results are shown in Figures 12 and 13. It can be seen that the compressor suction temperature when the filter dryer reached saturation conditions is a bit lower than that of the baseline system (Figure 9). Since the compressor mass flow rate in both tests is virtually the same, there must have been an increase in the capillary cooling rate when the dryer was placed horizontally, which favors the occurrence of the noise. Nevertheless, no noise was detected.

Figure 14 shows images of the sight glass after compressor start-up. It can be seen that the liquid level in the dryer increases gradually until it reaches the upper edge of the capillary and that there is no vortex. In this case vapor bubbles do not enter the capillary tube, which explains the absence of noise. It should also be noted that the capillary tube was kept parallel to the dryer wall. This situation is not always reproduced on a production line, which may induce the appearance of vortices and noise, even though the dryer is apparently in the horizontal position.

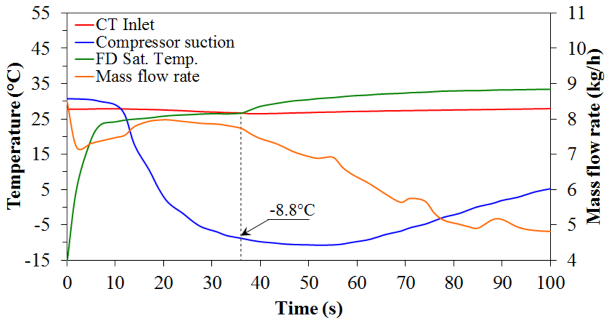


Figure 12: Thermodynamic readings: horizontal dryer

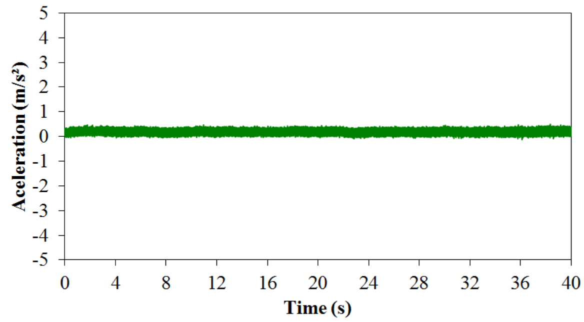


Figure 13: Accelerometer readings: horizontal dryer

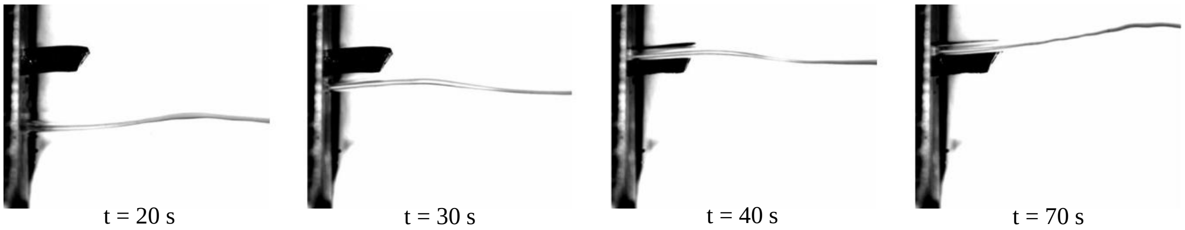


Figure 14: Liquid level in the sight glass: horizontal dryer.

3.2 Filter dryers of different diameters

A narrower dryer would allow the liquid level to increase more rapidly, reducing the interval between the levels shown in Figures 3A and 3D. This would decrease the length of time for bubble entrance into the capillary, reducing the amount of pops. Even more importantly, the level shown in Figure 3D could be reached before the suction line temperature becomes cold enough to cause CIS, *i.e.*, the bubbles that would enter the capillary would not condense sufficiently fast to create shockwaves and generate noise. Figure 15 illustrates this cogitation. On the other hand, a wider dryer would enlarge the time interval for liquid level increase. In this case bubble entraining vortices would be formed after the suction line reached its minimum value, avoiding CIS and noise, as postulated in Figure 16.

The original filter dryer has 20 mm inner diameter. The test results obtained with a narrower dryer (15 mm inner diameter) are shown in Figures 17 and 18. It can be seen that the noise started 38 seconds after compressor start-up, against 31 seconds for the baseline test. With the narrower dryer, however, the noise lasted for 40 seconds and the intensity of the pops remained high throughout this period, against 29 seconds and decreasing intensity observed in the baseline test. Thus, the narrower dryer is not an effective solution to this problem, since it worsened the noise.

The colder suction line was responsible for the high intensity pops. The noise lasted longer due to the liquid level oscillation in the dryer, as illustrated in Figure 21. Such oscillation occurs because the smaller inner diameter accelerates both the increase and decrease of the liquid level. Figure 18 shows periods with and without pop concentration. In the pop concentration period the liquid level is probably low enough to allow vapor bubbles in the capillary (Figure 3C). In the other periods the liquid level is higher, preventing vortices from forming (Figure 3D).

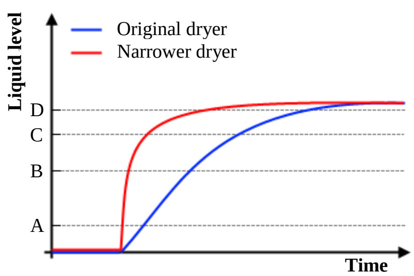


Figure 15: Expected liquid level variation: narrower vs. original dryer

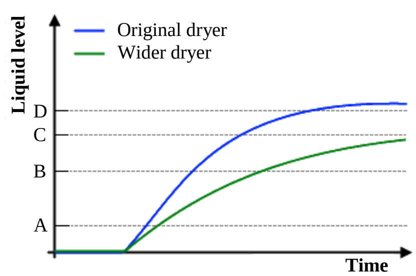


Figure 16: Expected liquid level variation: wider vs. original dryer

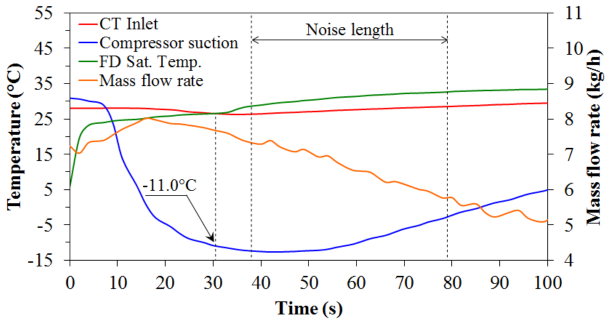


Figure 17: Thermodynamic readings: narrower dryer

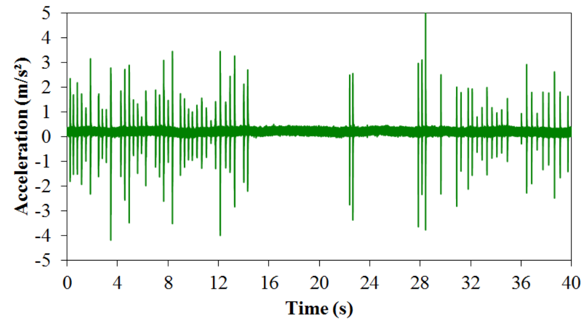


Figure 18: Accelerometer readings: narrower dryer

The results obtained with the wider dryer (25 mm inner diameter) are shown in Figures 19 and 20. Comparing with the baseline system it can again be noted that the suction line became colder and the compressor mass flow rate was slightly higher when the wider dryer was installed. Figure 19 shows that the noise was detected 33 seconds after compressor start-up. Its length decreased only slightly, from 29 to 26 seconds. Finally, both the intensity and the frequency of the pops increased. Therefore, the 25 mm dryer is not an effective solution to the popping noise.

The pops had higher intensity because the suction line was colder. It was also noted that the pops were concentrated in a very specific region (Figure 20). Hence, it is inferred that due to the higher inner diameter the liquid level may have increased gradually until it reached the condition of Figure 3C, which occurred 33 seconds after compressor start-up, remaining at this level for 26 seconds. Therefore, more intense vortices were formed over a higher period of time and more bubbles entered the capillary. It is probable that after this period the liquid level reached condition D of Figure 3, characterizing the end of the pops, as illustrated in Figure 22.

3.3 Reduction of the compressor mass flow rate

Another approach to eliminating the noise is to reduce the cooling rate of the capillary tube. This can be achieved by decreasing the compressor mass flow rate. The original compressor was thus replaced by a smaller capacity one. These test results are shown in Figures 23 and 24, where it can be seen that the compressor mass flow rate was reduced by approximately 10% over the first 100 seconds. Furthermore, the suction line temperature dropped considerably, with its minimum value falling from -10 °C to -20 °C, probably due to a lower mass flow rate through the capillary.

Although the suction line was much colder in this case it can be noted that the popping appeared only 42 seconds after compressor start-up. It should be mentioned that only two high intensity pops were detected when the suction line reached its minimum temperature. This implies that, apart from decreasing the cooling rate of the capillary tube, the compressor change modified the liquid level behavior in the dryer. Thus, it can be assumed that in a hypothetical situation where the suction line is not so cold the noise would be eliminated by the compressor change. Of course, the compressor replacement modifies the behavior of the entire refrigeration system. However, the idea was to demonstrate the requirement to lower the compressor mass flow rate during the initial moments of the system operation, rather than the need to substitute the compressor.

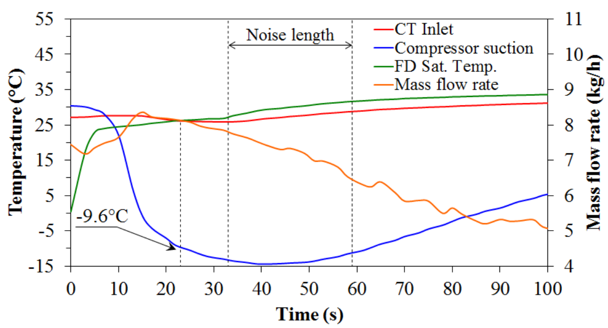


Figure 19: Thermodynamic readings: wider dryer

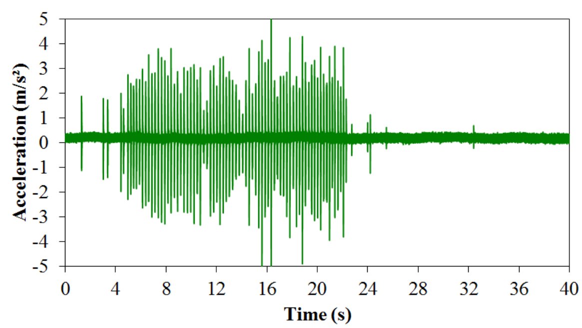


Figure 20: Accelerometer readings: wider dryer

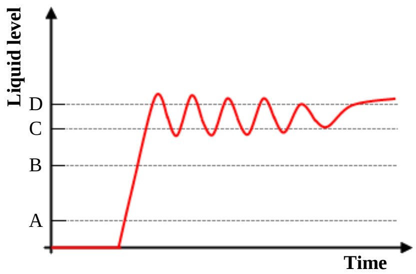


Figure 21: Liquid level variation: narrower dryer

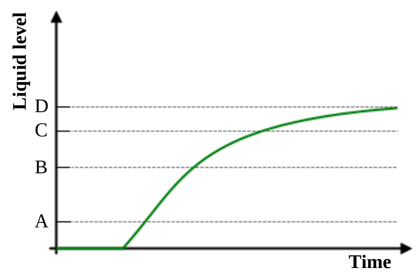


Figure 22: Liquid level variation: wider dryer

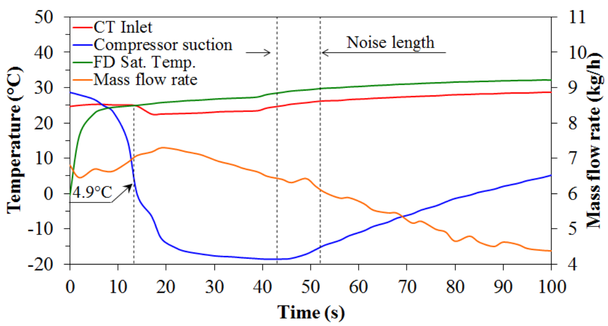


Figure 23: Thermodynamic readings: smaller capacity compressor

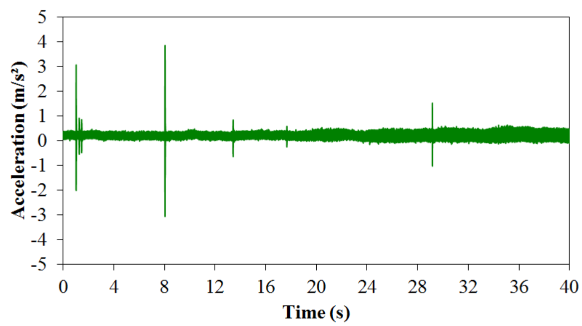


Figure 24: Accelerometer readings: smaller capacity compressor

3.4 Additional internal heat exchanger

The transition time between conditions A and D of Figure 3 can be reduced by supplying the dryer with more liquid. Since the suction line reaches temperatures as low as -10 °C in the first minute of compressor operation it can be placed in contact with the condenser outlet tubing to produce more liquid, as illustrated in Figure 25.

The thermodynamic impact of this alternative is not fully known. On the one hand, the subcooling is increased, increasing the specific refrigerating effect and the cooling capacity. On the other hand, the compressor suction temperature is also increased, decreasing the compressor volumetric efficiency and mass flow rate and increasing the specific compression work. The success of this new configuration relies on a balance between the negative and positive effects on the system performance. It is hard to conjecture which effect will prevail, although it is expected that because the working fluid is HFC-R134a – a fluid whose density varies moderately with temperature – the overall effect will be positive.

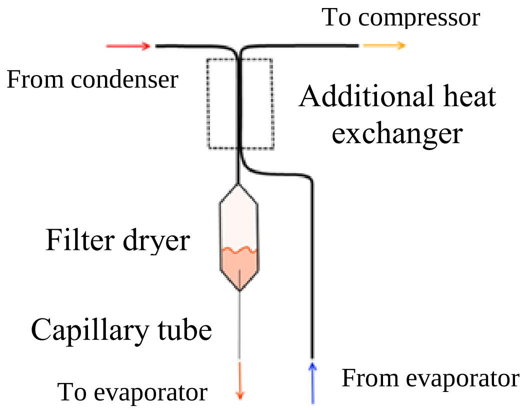


Figure 25: Additional internal heat exchanger

The test results are shown in Figures 26 and 27. An additional thermocouple was installed on the suction line, where it exits the thermal insulation of the refrigerator, before the additional heat exchanger. This thermocouple provided a representative value of the suction line temperature. It can be observed that the subcooling significantly increased 37 seconds after compressor start-up, when the additional heat exchanger began to positively affect the system performance. It can also be noted that the compressor volumetric efficiency and mass flow rate over the first 100 seconds of compressor operation were practically unaffected by the additional heat exchanger. The fluid temperature at the capillary inlet decreased and the compressor suction temperature increased, as expected. In other words, bubbles entered the capillary with a lower temperature and lost heat to a warmer tubing. Since the temperature difference decreased, the cooling rate of the capillary tube also decreased, reducing the possibility of the popping noise to occur.

Figure 28 shows the liquid level variation in the sight glass over time. Vortices were observed at the capillary inlet 20 seconds after compressor start-up. However, at this moment, the suction line was not cold enough to cause CIS, as can be deduced from Figure 26. The suction line temperature decreased over time, providing a high liquid formation in the sight glass, eliminating vortices. Several small bubbles are formed between 20 and 40 seconds after compressor start-up. Some of them enter the capillary but do not cause CIS due to their small size. After 40 seconds the suction line temperature increases and the effect of the additional heat exchanger is reduced. The liquid level decreases, reaching the capillary edge again. Vortices are formed again, but now the suction line temperature is not cold enough to cause CIS.

3.5 Energy consumption of the refrigerator with the additional heat exchanger

An energy consumption test was carried out with the additional heat exchanger in place to check its effect on the refrigerator performance, again according to the ANSI/AHAM HRF-1 (2004) standard. A value of 488.8 kWh/year was obtained, which is only 0.5% higher than the baseline value. Since this difference is of the same order of the magnitude of the measurement uncertainty, it may be concluded that the additional internal heat exchanger is the most promising alternative to avert the popping noise without compromising the product performance.

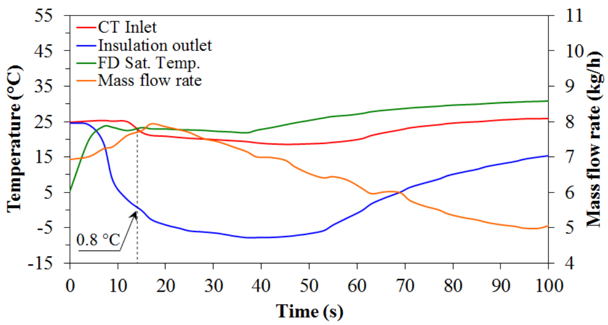


Figure 26: Thermodynamic readings: additional internal heat exchanger

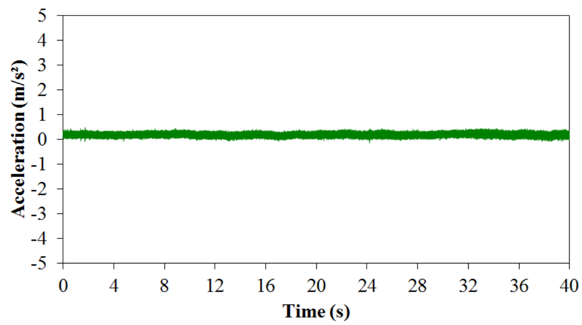


Figure 27: Accelerometer readings: additional internal heat exchanger

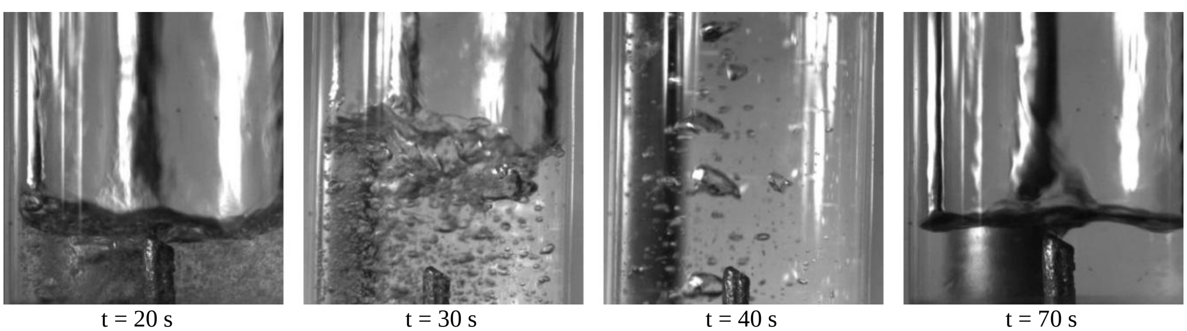


Figure 28: Liquid level in the sight glass with the additional internal heat exchanger

4. CONCLUSIONS

In this work the popping noise phenomenon was studied experimentally from a thermodynamic point of view.

- It was confirmed that the noise is caused by the presence of vapor bubbles in the capillary and by the cooling rate provided by the internal heat exchanger soon after compressor start-up.
- It was shown that the noise is completely eliminated when the dryer is positioned horizontally. It was also shown that this is mainly due to the absence of bubble-entraining vortices at the capillary inlet. This solution, however, is not applicable to all appliances.
- Filter dryers of different diameters were used in an attempt to prevent bubbles from entering the capillary. When a narrower dryer was used, the liquid level inside it oscillated considerably, enabling vapor bubbles to collapse in the capillary over a wide time interval, worsening the noise. The wider dryer made the liquid level increase more slowly, increasing the frequency of the pops, worsening the noise in a different manner.
- The capillary cooling rate was decreased through a reduction in the compressor mass flow rate, which decreased the suction line temperature. The frequency and intensity of the pops were greatly reduced, suggesting that the cooling rate, and also the popping noise, is much more dependent on the compressor mass flow rate than on the suction line temperature.
- Finally, an additional internal heat exchanger was used to increase the liquid supply to the dryer and thus eliminate bubble entrance into the capillary. Among the proposed alternatives this was the most promising. Vortices were observed at the capillary edge, but only when the suction line was not cold enough to cause CIS. When the suction line reached its minimum value the liquid level was well above the capillary edge. Additionally, the energy consumption of the appliance was not changed when applying the additional internal heat exchanger.

NOMENCLATURE

f	Compressor speed	Hz
h	Specific enthalpy	kJ/kg
$\dot{m}$	Mass flow	kg/h
V	Compressor displacement	m³
v	Specific volume	m³/kg
$\dot{W}$	Compressor power	W
η	Efficiency	Dimensionless

Subscripts

1,2,3,4	Points in Figure 5
V	Volumetric
G	Overall

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