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Noise Generation in Household Refrigerators: An Experimental Study on Fluid-Borne Noise

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ABSTRACT

The oscillatory pressure disturbance created by the refrigerant flow through pipes and components is one of the main sources of noise in household refrigerators. This type of excitation is transferred to the pipes and the vibratory energy travels through the structure reaching the cabinet from which it radiates as sound energy. In order to obtain a better understanding of this type of noise a household refrigerator was instrumented with thermocouples, absolute and acoustic pressure transducers and accelerometers. Tests were carried out under different operating conditions. It was observed that the compressor is the main source of excitation, affecting practically the entire refrigeration loop. It was also found that the acoustic excitation at the exit of the capillary tube is high enough to induce significant vibrations in the evaporator. In view of the database collected, comments and suggestions concerning the design of household refrigerators are presented and discussed.

1. INTRODUCTION

Noise generation from home appliances is nowadays an increasingly important factor, together with energy consumption and cost. Consumers usually associate a high level of noise or anomalous noises to low quality products, and this increases the service calls and damages a company's image.

For these reasons noise level has become one of the key factors in the design process of household refrigerators. In the past, attention was mostly focused on the compressor design. More recently other types of noise have started to play a significant role in the overall system design. One of these, the so-called fluid-borne noise, is a type of noise created by the refrigerant flow at the pipes, which is the main focus of this study. For the sake of clarity the word "noise" will be used in this text to describe the airborne sound radiated by the refrigerator. The oscillatory phenomena in solid structures will be referred to as vibrations. The oscillatory phenomena in the refrigerant flow will be referred to as acoustic excitation or acoustic pressure.

1.1 Vibro-Acoustic Behavior of Refrigerators

The noise radiated by a refrigerator is basically generated by vibrations of the surfaces in direct contact with the surrounding air. The cabinet walls are efficient sound radiators since their area is relatively large and they are light.

There are two main excitation sources in a refrigerator: (i) rotating and vibrating components (compressor and fan) and (ii) fluid flow (air and refrigerant). The compressor is the main source of oscillatory excitation (Moorhouse, 2005; Carvalho, 2008). The compressor shell radiates noise directly into the ambient air. Moreover, as the compressor is tightly attached to the refrigeration loop and also to the cabinet its vibrations are easily transmitted to the cabinet walls. The compressor also generates pressure pulsations in the refrigerant flow which are transmitted through the suction and discharge pipes (Farstad and Sing, 1990; Carvalho, 2008). Fan vibrations transmitted to the cabinet also represent a significant contribution to the overall noise level of the refrigerator.

The refrigerant flow, by its nature, induces pressure fluctuations (Ffowcs Williams and Hawkings, 1969; Reethoff, 1978; Pierce, 1981). As shown in Figure 1, the pressure fluctuations are firstly applied to the inner side of the tubing. Secondly, the vibratory energy propagates through the tubing and is transmitted to the cabinet through the fasteners. Thirdly, the cabinet radiates this energy as airborne noise (Caetano, 2013).

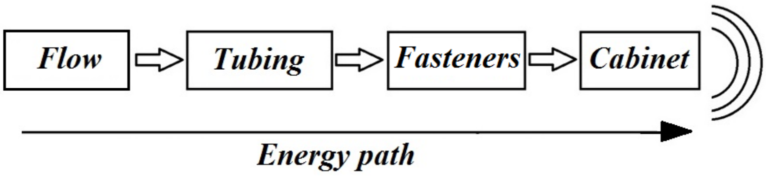


Figure 1: Vibratory energy flow in a refrigerator

Refrigerant-induced noise thus results from a combination of a fluid-acoustic and a vibro-acoustic phenomenon. The former is related to the acoustic excitations generated and propagated in the refrigerant flow and the latter to the tubing vibrations which are transmitted to the cabinet walls.

The main sources of acoustic excitation in the refrigerant flow are turbulence, compressor-induced pulsations, phase change and throttling. Several studies on this subject can be found in the open literature (Van Wijngaarden, 1972; Reethoff, 1978; Baumann, 1984; Singh *et al*., 1999; Zummo, 1999; Lattanzi *et al*., 2001; McLevige and Miller, 2001; Han *et al*., 2009; Han *et al*., 2010; Han *et al*., 2011; Hartmann and Melo, 2013).

The so-called expansion noise is created by the throttling of refrigerant passing through the expansion device of refrigerating systems. This process produces a turbulent two-phase compressible flow which can even reach critical (sonic) conditions. At the exit of the expansion device an under-expanded supersonic jet is usually present, which can creates intense acoustic excitation (Lighthill, 1963; Zhang *et al.*, 2002). The capillary tube is a particular type of expansion device where other types of phenomena take place. Lattanzi *et al.* (2001), McLevige and Miller (2001) and Hartmann and Melo (2013) have observed, for example, that the ingestion of vapor bubbles by the capillary tube can lead to an anomalous type of noise, known as a popping noise. Overall, the expansion noise is a rather complex type of fluid-borne noise because the excitation varies according to the operational characteristics of the refrigerating system.

A brief inspection of the literature reveals that most previous studies on fluid-borne noise have been carried out in acoustic chambers, with measurements of the airborne noise and acceleration taken at specifics points of the appliance. This approach is somewhat limited because the end effect of the flow-related phenomenon is being measured, i.e., the fluid-acoustic phenomenon is not distinguished from the vibro-acoustic phenomenon (see Figure 1). Thus, some types of anomalous noise may be detected in one class of product but not in another, even though the working conditions are the same, due to the different vibro-acoustic properties of each refrigerator. Moreover, even distinct samples of the same model can behave differently under the same working conditions as a consequence of differences in the manufacturing process.

Since the distinction between the fluid-acoustic and vibro-acoustic phenomena is crucial in the current study, several acoustic pressure sensors were installed in the refrigeration loop specifically to monitor the acoustic waves in the refrigerant flow. Thus, it was possible to detect the noise source and cross-correlate it with the thermodynamic readings. This approach distinguishes this study from others reported in the literature.

2. EXPERIMENTAL WORK

Measurements were carried out on a typical frost-free, bottom-mount refrigerator, equipped with a variable-speed reciprocating compressor, running with the use of R-600a.

In addition to the acoustic pressure measurements, acceleration measurements were also carried out in an attempt to cross-correlate the accelerations with the acoustic pressure signals. The intention was to verify whether or not the acceleration readings, which are much easier to carry out, can be used as a proxy for acoustic pressure measurement.

2.1 Instrumentation

Figure 2 shows the refrigeration loop and the instrumentation used. T-type thermocouples, with a maximum uncertainty of ± 0.2°C, were employed to measure the temperature at specific points of the refrigerator. Strain gage absolute pressure transducers, with a maximum uncertainty of ± 0.03 bar, were used to measure the static pressures. The acoustic pressure measurements were carried out using piezoelectric transducers, with a sensitivity of 14.5 mV/kPa, resolution of 7 Pa, measurement range of 345 kPa and minimum frequency of 0.5 Hz. A pressure tap, illustrated in Figure 3, was designed and constructed specifically for these transducers. The acceleration measurements were carried out with piezoelectric accelerometers (6.2 mm I.D. and 1.5 g weight) with a sensitivity of 10 mV/g and a frequency range of 0.5 - 20 kHz.

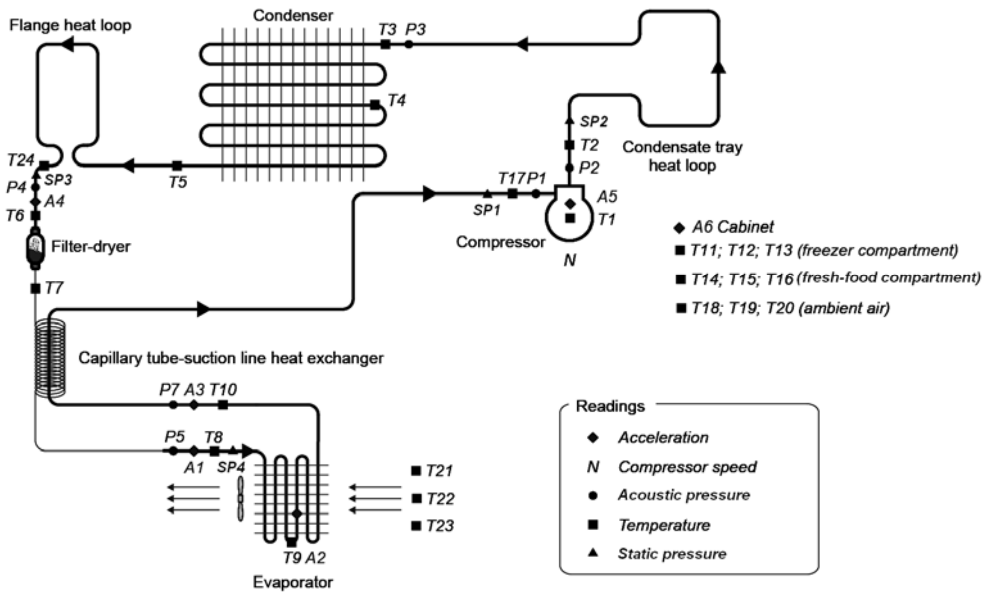
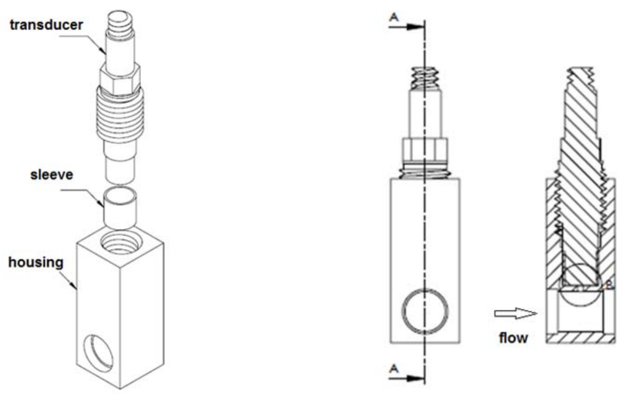


Figure 2: Schematic representation of the refrigeration loop



(a) exploded view (b) cutaway view

Figure 3: Acoustic pressure tap

2.2 Experimental Design

The refrigerator was tested in a room with low background noise and without precise control of the ambient temperature. The tests were carried out at two compressor speeds, 3500 rpm and 2500 rpm, and with varying thermal loads. To this end electric heaters, with a total power of 45W and 90W, were strategically distributed in the fresh-food compartment.

2.3 Data Acquisition Systems

Two data acquisition systems were used in this study, one for the temperature, absolute pressure and compressor speed measurements (DAQ 1), and another for the accelerations and acoustic pressure readings (DAQ 2). A voltage signal supplied by the refrigerator electronic board was used to synchronize the measurements. DAQ 1 is a general-purpose data acquisition system capable of measuring DC voltage signals, with an ADC of 6½ digits. This system scans all thermodynamic signals every 7 seconds. DAQ 2 is a typical signal analyzer with an ADC of 24 bits, maximum sampling frequency of 205 kHz and dynamic range of 115 dB. The sampling frequency was set at 25,600 samples/s, with a high-pass filter of 5 Hz.

3. EXPERIMENTAL RESULTS

The ambient temperature has a considerable effect on the condensing pressure and on the subcooling. As previously mentioned the ambient temperature was not controlled but it remained stable (22 - 26^{o}C) during the experiments.

Figure 4 shows the acoustic pressure and the acceleration values during the on cycle of the refrigerating system. Levels refer to the normal values (2 x 10^{-5} Pa for pressure and 10^{-5} m/s^2 for acceleration) and the start time coincided with the compressor start-up time.

It can be observed that the behavior of the acoustic pressure is fairly constant over time, the highest levels being observed at the compressor discharge point (P2) (153 - 161 dB), the compressor suction point (P1) (145 - 149 dB) and the condenser inlet (P3). In fact, at these points the high-speed superheated vapor flow creates a high level of turbulent excitation and also the effect of the compressor pulsations is high, particularly at the discharge pipe. It can also be seen that the acoustic pressure at the condenser inlet follows closely the acoustic pressure at the compressor discharge point, with an offset of around 5 - 8 dB. This is due to the attenuation properties of the flow, the refrigerant fluid properties and the geometry of the loop between the two measuring points.

In contrast, the lowest values are observed at the liquid line (P4) (110 - 115 dB) and at the capillary tube exit (P5). The high-intensity spikes captured by the P4 transducer are a clear indication of excitations in the refrigerant flow. This is probably due to the presence of vapor bubbles in the subcooled refrigerant flow, as previously observed by many authors (Kalman and Mori, 2002; Brennen, 2005).

The acoustic pressure recorded at the capillary exit (P5) (120 - 145 dB) is higher than that recorded in the liquid line (P4). The signal oscillations are also much higher, with amplitudes of up to 25 dB. This behavior reflects the oscillatory nature of the refrigerant flow through capillary tubes (Revellin, 2006; Lattanzi *et al.*, 2001).

The acoustic pressure at the evaporator exit (P7) (132 - 136 dB) is, in general, higher than that measured at the capillary tube exit (P5), but with much lower oscillations. Furthermore, the acoustic pressure at the evaporator exit (P7) is 13 to 15 dB lower than that at the compressor suction point (P1), indicating that the compressor-induced pulsations are attenuated in between these two points.

Figure 4(d) shows that the acceleration values are almost constant over time. A considerable similarity between the A1 and P5 signals (sensors located at the same point) was observed in all tests. This indicates that the acoustic excitation at the exit of the capillary tube is strong enough to be transmitted to the tubing.

It should be mentioned that, with the exception of the P4 signal, spikes were not observed in any other signal. Also, no anomalous noise, as reported in the open literature, was detected during the tests.

In general, the measurements at P4 and P7 showed little difference, in terms of the average values, from one test to another, reflecting the weak influence of the operating conditions. The P5 signal showed a direct correlation with the compressor speed, revealing its dependence on the mass flow rate in the capillary tube. The P1 signal also showed a direct correlation with the compressor speed. In contrast, the P2 signal showed an inverse correlation with the compressor speed. One possible explanation for this finding lies in the fact that compressors have built-in mufflers, which were traditionally designed to give maximum attenuation at the standard speed of 3500 rpm. When the speed is lowered, the efficiency of these mufflers, at least on the discharge side, is reduced, thereby allowing higher pulsations to be transmitted to the discharge line.

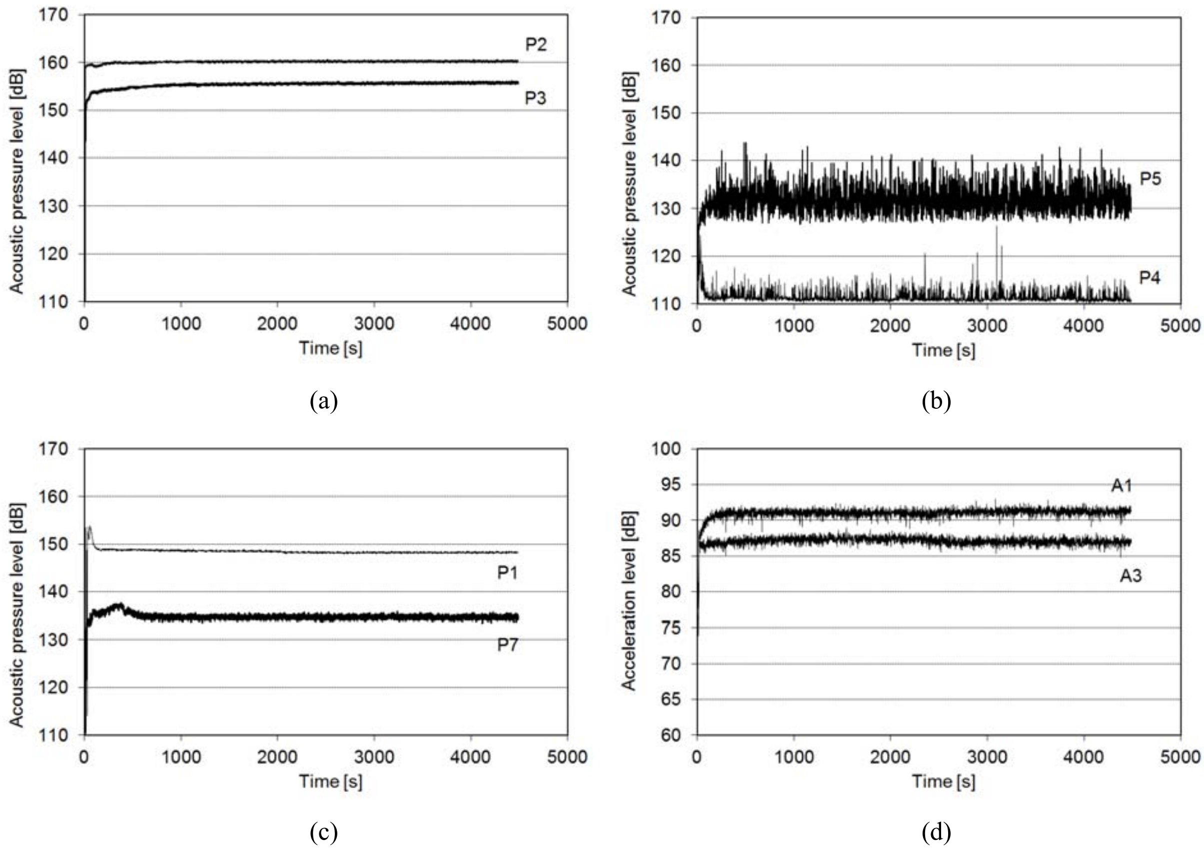


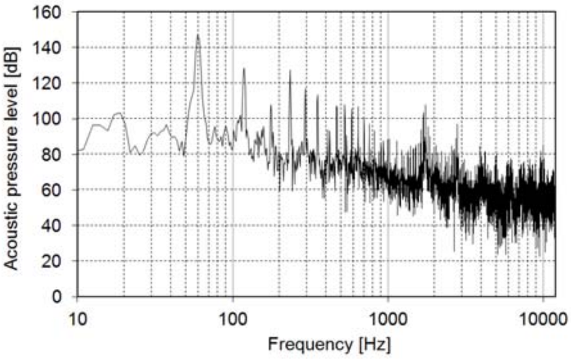
Figure 4: Acoustic pressure and acceleration measurements

Figure 5 shows the acoustic pressure spectral density in narrow frequency bands (1.56 Hz), for a typical test under a compressor speed of 3500 rpm (58.3 Hz approximately). Spectrograms showing the time evolution of the spectral density were also obtained and they verify that all signals are fairly stable over time.

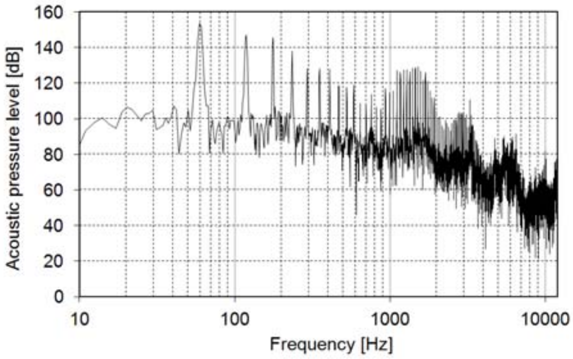
The acoustic pressure signals at P1, P2 and P3 clearly show several harmonics of the compressor speed, indicating the significant effect of the compressor pulsations. Several resonances were observed in the spectra, with small influence of the operating conditions.

Figure 5(d) shows that the acoustic pressure at the liquid line is much lower than that at the condenser inlet, but still with some compressor harmonics, although strongly attenuated. This exemplifies the attenuation effect of two-phase flow in the condenser, as reported in the literature (Van Dijk, 2005).

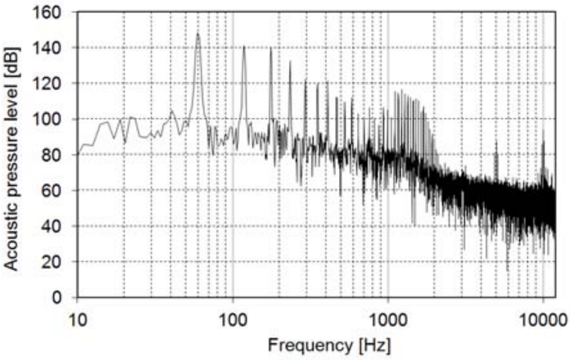
Figures 5(c) and 5(d) show peaks at 5 kHz and 10 kHz, independently of the operating conditions. Based on the bubble oscillation theory (Han *et al*. 2009 and 2010), it can be assumed that these frequencies are related to the natural frequencies of spherical bubbles of approximately 1.79 mm and 0.9 mm, respectively. Since the inner diameter of the condenser tubing is 3.34 mm, it may be possible that these peaks are related to the two-phase condensing flow in the condenser.



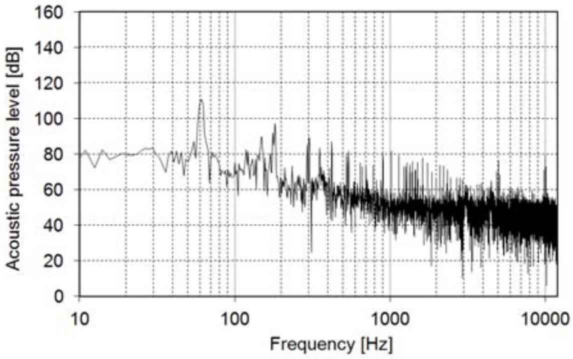
(a) P1



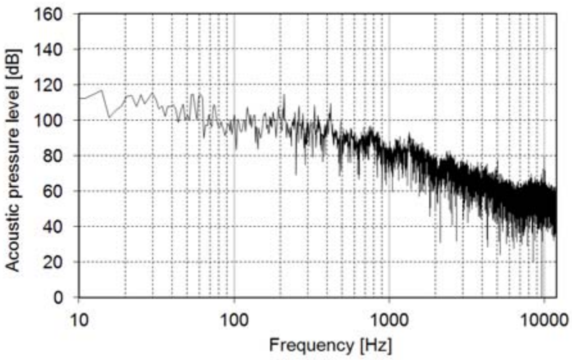
(b) P2



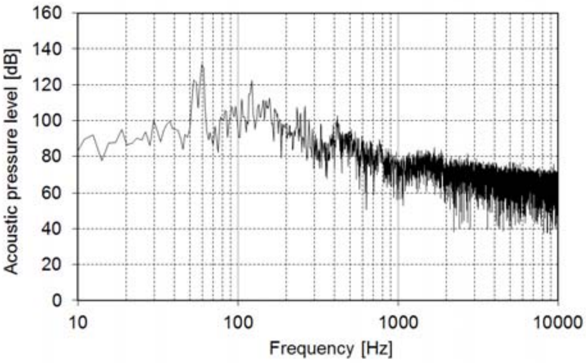
(c) P3



(d) P4



(e) P5



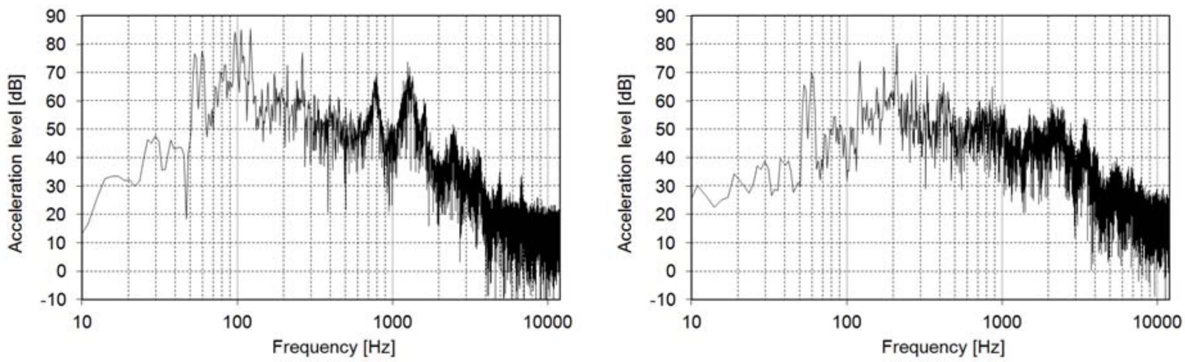
(f) P7

Figure 5: Acoustic pressure spectral density

Figure 5(e) shows that the spectral density at the capillary exit resembles a typical jet noise spectrum (Zhang *et al.*, 2002). The spectrograms for this signal show that the oscillations are clearly visible and stronger below 4 kHz. Unlike the P4 signal, spikes are not observed. This confirms that the oscillations shown in Figure 4(b) are a consequence of the oscillatory nature of the capillary flow, while the spikes are an isolated excitation phenomenon.

Finally, Figure 5(f) shows that the harmonics of the compressor pulsations are still present, although attenuated. Again, no spikes are apparent in the spectrograms.

Figure 6 shows the acceleration spectral density measurements at the inlet (A1) and outlet (A3) of the evaporator. When compared to the P5 signal, the A1 signal shows close similarity in terms of the spectral density, except for the frequency range of 10 – 100 Hz, where the tubing provides significant attenuations. This similarity is also observed between signals A3 and P7.



(a) A1 (b) A3

Figure 6: Acceleration spectral density

In summary, the results of the analysis of the spectral density and the spectrograms are as follows. The acoustic pressure and the accelerations signals are very stable, as verified through a time domain analysis. The pressure pulsations produced by the compressor are the main source of excitation. Acoustic excitation generated by the flow itself is more significant above 500 Hz. The acoustic excitation is higher at the evaporator exit and at the compressor suction and discharge pipes because of the high-speed superheated vapor flow. At the condenser exit the acoustic excitation is lower because of the low-speed flow. The high-intensity spikes observed in the liquid line are due to the presence of vapor bubbles at the entrance of the capillary tube. A signal of relatively high intensity and with an oscillatory nature, characteristic of a turbulent jet noise, can be noted at the inlet of the evaporator. The acceleration and the acoustic pressure measurements at the evaporator inlet are in line with the results of other studies.

The similarity between the time evolution of A1 and P5 global level signals shows that the former can be used as an indicator of the latter at that specific location (exit of the capillary tube). The direct readings of those sensors confirm this conclusion, particularly when anomalous excitations take place. However, it must be kept in mind that the spectral analysis has shown that the tubing provides attenuation at different frequencies, particularly at the low values.

4. RECOMMENDATIONS FOR REFRIGERATOR DESIGN

The fluid-acoustic phenomenon can be controlled by introducing modifications at the system level. Han *et al*. (2010, 2011) have demonstrated that two-phase intermittent and transient flow patterns generate high levels of acoustic excitation. This type of flow pattern can be changed by using smaller I.D. tubing, although this will require a complete redesigning of the heat exchangers.

The compressor-induced pressure pulsations are an important excitation source. Acoustic filters installed in the suction and discharge pipes, close to the compressor, may reduce such excitations (Troshin *et al.*, 2008). These filters should be designed bearing the following aims in mind: low frequency (less than 1 kHz), low pressure drop and easy lubricating oil return to the compressor crankcase.

The acoustic pressure at the exit of the capillary tube is transmitted to the tubes and is strong enough to induce high vibration levels in the evaporator, as shown by Caetano (2013). Thus, a 4 – 5 kHz acoustic filter is recommended for this location.

It should be noted that the control of the transmission paths in household refrigerators merits special attention (Carvalho, 2008). A poor refrigerator design, even with a very small degree of acoustic excitation, may cause problems related to unacceptable noise.

5. RECOMMENDATIONS FOR FURTHER WORK

- The ambient temperature should be controlled and varied since it affects the condensing pressure, the subcooling and the amount of vapor bubbles at the inlet of the capillary tube.

- A transparent filter-dryer should be designed, constructed and installed in the refrigeration loop in order to monitor the refrigerant flow pattern at the inlet of the capillary tube.

- Tests with different refrigerant charges should be carried out in order to understand the influence of the charge on the filter-dryer liquid level.

- A visualization study at the capillary exit, although difficult to perform, is highly recommended to check whether or not an oscillating jet flow is present in this region.

6. CONCLUDING REMARKS

An experimental study was carried out focused on the refrigerant-induced noise in household refrigerators. The acoustic pressure was measured along the refrigeration loop in order to gain a better understanding of the fluid-acoustic phenomenon. Global levels of acoustic pressure and the associated spectral densities were also obtained.

The pressure pulsations induced by the compressor were found to be the major source of excitation in the refrigerant flow, followed by the high-intensity, turbulent and oscillatory jet at the exit of the capillary tube.

Some recommendations for further studies as well as for refrigerator design have been presented and discussed. This paper covers only the initial part of a broad research program involving a very comprehensive experimental and theoretical study on fluid-borne noise in household refrigerators.

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