Reducing the noise from the cavity by tuning the compressor’s oil level

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Abstract. This paper proposes a unified solution for noise of two different versions of the same compressor. First, the problem was identified as a high noise in 1kHz third-octave band from the version using R134a refrigerant and 1.25kHz band from R600a version. The first hypothesis was the noise coming from the motor through shifting frequency, which would explain the different frequency ranges for different refrigerants. However, this was debunked experimentally, and results are shown in this paper. The other hypothesis was the noise coming from the cavity. This was proved to be the problem's root cause and this paper shows the simulated and experimental results which corroborates the inference. Therefore, we propose a single solution for both versions by using an optimized amount of oil, to control the cavity volume.

1. Problem Definition and Literature Review
The object of study is a battery driven compressor that weights approximately 1.5kg, and measures 130mm wide, 110mm long and 100mm tall. Hence, its cavity volume is very small, measuring approximately 320 cm³. This compressor’s family has 2 different versions, one dedicated to run with R134a and the other with R600a, however, the same 1.4cc version was tested for both types of refrigerants, due to availability of parts.

With R134a, the noise problem was identified in between 800Hz and 1kHz 1/3rd octave bands, and with R600a, between 1kHz and 1.25kHz.

These problematic regions were also sensitive to the compressor speed. While running on 2300 RPM (hereinafter referred as “low speed”), the effect was less harsh. On the other hand, the compressors showed a higher noise while running with 4500 RPM (hereinafter “high speed”).

The results shown in Figure 1 (a) and (b) are from running noise measurements with evaporator and condenser temperatures at -10°C and +55°C respectively, following CECOMAF standards. The noise was measured according to ISO-3743 for reverberant rooms, and the results are displayed in Sound Power Level (normalized for data protection).
Similar problems were investigated by [1] and [2]. They approached the characterization of cavity resonances and identified the changes in oil level as significant to change the cavity’s modal density in low frequencies. However, they focused on changes of shell design, which can have a high cost, depending on the maturity of the project.

On the other hand, the work of [3] aware of noise problems that could occur through transmission of vibration. Therefore, we also point out the importance of optimizing the oil level considering all different aspects of noise generation.

2. Cavity Hypothesis

In the meantime, we carried several simulations of the cavity acoustics, as an attempt to determine a correlation with noise from critical bands. ANSYS® 2019 R2 was used, the acoustic mesh consisted in both cavity and suction muffler, with elements of type FLUID221 with edges of 1mm length, hence the mesh had approx. 180 thousand elements and 292 thousand nodes. The element FLUID221, according to ANSYS® Reference [4], is a higher order 10-node solid element that with quadratic shape functions.

An unitary surface velocity was prescribed at the suction port on the cylinder head, and an average pressure was calculated from all surfaces that had direct contact with the shell (hereinafter referred as “cavity bounds”).

![Figure 1. Critical bands at High Speed (a) and Low Speed (b), with R134a shown in full line and R600a in dashed line.](image)

![Figure 2. Finite element model used on Simulations.](image)
To obtain the proper cavity’s gas-filled geometry, we used a cutting plane (identified on Fig. 2 as a black dashed line) and neglected the oil-filled volume. Afterwards, to simulate changes in oil level, we simply moved this cutting plane to the desired position.

With approximately 15ml of oil, the cavity response had an interestingly similarity to the noise of tested samples. The results below represents the simulation versus maximum and average narrow band values of all speeds, from all samples.

![Graphs showing narrowband SWL versus pressure on Cavity Bounds for R134a and R600a](image)

**Figure 3.** Narrowband SWL versus pressure on Cavity Bounds for R134a (a) and R600a (b)

Therefore, because of this outstanding similarity, we started to investigate further the cavity-oil relationship.

It appears the mechanism for noise generation was highly connected to a specific “cluster of modes”. This cluster was formed by 2 main modes, which high pressure regions are located on the shell, aligned with the front of the piston (see Figure 4 below). Since this is a rather flat surface, the pressure pulsation could excite the shell and, finally, generate noise.

![Examples of problematic cavity mode at the front and side](image)

**Figure 4.** Example of problematic cavity mode at the front of the piston (a) and on the side (b).

Finally, we performed a heuristic optimization and, when comfortable with the results, we validated the findings with another round of measurements.
3. Heuristic Optimization

After seeing the high sensitivity to oil level, we saw a strong potential for tuning it. Thus, we performed a heuristic optimization by varying the oil level, gas temperature and compressor inclination.

Simulations with oil level variation consisted in a fixed temperature for the refrigerant of 60°C, a desired frequency range of 640Hz to 1.4kHz, spaced in 76 solution intervals. We chose this coarse frequency resolution to avoid long calculation times.

Within this temperature, R134a has a speed of sound $c = 170.67 \text{ m/s}$ and density $\rho = 2.15 \text{ kg.m}^{-3}$, and R600a has $c = 225.42 \text{ m/s}$ and $\rho = 1.23 \text{ kg.m}^{-3}$.

For the geometry variation, we started with 15ml of oil, which was arbitrarily chosen as the reference plane of cut. Then, by displacing this plane upwards 2mm, 5mm and 7mm to respectively achieve 23ml, 37ml and 47ml of oil. The results of average pressure on the cavity bounds are shown below for R134a and R600a.

![Figure 5. Average pressure on Cavity-Shell interface with oil variation for R134a (a) and R600a (b)](image)

We can observe from the results a clear separation of the cluster of modes. The mode from higher frequency even gets filtered away by the suction muffler, as expected.

These results also show how this small compressor has a sensitive cavity; thus, further simulation was carried out to ensure safe margins before investing resources on experiments.

The next round of simulations took into consideration the effects of temperature variation over a fixed amount of oil. Hence, we considered 23ml of oil (reference plane +2mm), and we simulated for 50°C, 70°C and 90°C. The used refrigerant properties can be seen in the table below.

| Gas    | Temp. [°C] | $c$ [m/s] | $\rho$ [kg.m$^{-3}$] |
|--------|------------|-----------|----------------------|
| R134a  | 50         | 168.04    | 2.22                 |
|        | 70         | 173.25    | 2.09                 |
|        | 90         | 178.25    | 1.97                 |
| R600a  | 50         | 222.07    | 1.27                 |
|        | 70         | 228.7     | 1.19                 |
|        | 90         | 235.11    | 1.12                 |

Table 1. Refrigerant properties for Temperature Variation.

The average pressure on cavity-shell interface are shown on the figures below. The major difference is a shifting of the resonances towards higher frequencies, as expected. Also, no new resonances interactions between muffler and cavity appeared, allowing us to conclude this design is robust from the temperature variation point of view.
Finally, out of curiosity, we also simulated the effects of inclination on the cavity acoustics. Since the effects for R134a and R600a were known to be similar, we only simulated with the former. The used geometry can be seen in the Figure 7 below, and results in the next Figure 8.

**Figure 6.** Average pressure on Cavity-Shell interface with temperature variation for R134a (a) and R600a (b).

**Figure 7.** Geometry with oil level at 10° of inclination.

**Figure 8.** Average Pressure on Cavity-Shell interface with inclination variation for R134a.
The same separation of the clustered modes was observed, suggesting a good palliative if we are unable to re-assess the oil level, e.g., if the compressor is already assembled in the appliance and kit rework is unfeasible.

4. Experimental Validation
The experimental validation consisted in several noise measurements according to ISO-3743, with the compressor running filled with R600a at -10°C/+55°C $T_{\text{evap}}/T_{\text{cond}}$ CECOMAF conditions.

We started with the lowest oil level, approximately 48ml which corresponds to 53g. To control the oil amount by its mass is preferable because it allows easy checking between measurements by using a simple scale. Hence, after each measurement, the compressor was weighted, and its oil level was increased by steps of approximately 2.5g. The results are displayed in Sound Power Level, but it was omitted due to data protection.

Results with compressor running at 2300 RPM, 3000 RPM, 4000 RPM and 4500 RPM are shown respectively in Figures 9 (a), (b), (c) and (d).

**Figure 9.** SWL of compressor at -10°C/+55°C CECOMAF conditions with R600a, running speed of 2300 (a), 3000 (b), 4000 (c) and 4500 RPM (d), filled with different oil levels.

We could clearly observe the appearance of a sweet spot for oil level within the cavity bands. With oil increase, the problem in cavity bands would decrease. For better visualization, we took an average of all speeds and normalized each band by its maximum, resulting on the surface plot of Figure 10.
Another interesting conclusion can be drawn by observing this plot above: Even with the decrease of noise coming from the cavity bands, the 4k and 5kHz bands showed a significant increase. This may have happened because of direct transmission of vibration through the oil if it touches the stator and rotor at the same time.

![Figure 10](image1.png)

**Figure 10.** Average differences from maximum SWL from each critical band, from all speeds. Compressor running with R600a, -10°C/+55°C CECOMAF conditions with different oil levels.

Further investigation with an acrylic shell showed the amount of 58g of oil was already dangerously close to the stator, and any inclination could make the oil surface to touch the moving parts and transmit vibration directly.

However, since this phenomenon was observed later and requires a totally different approach for understanding, it will not be discussed in the present paper.

**Figure 11.** See-through acrylic shell shows 58g of oil (dashed line) is already dangerously close to stator (full line).

5. **Conclusion**
This paper covered the issue of a single compressor having distinct high noise bands for distinct refrigerants.

The first hypothesis was the noise being influenced by the electric motor through frequency shifts with different loads. But this was quickly debunked experimentally. The test consisted in maintaining
the same pressure on the evaporator and ensuring the same power consumption, so the high noise should come from the same frequency ranges. However, we observed the same problematic noise bands, even with these specific conditions.

The second hypothesis was the cavity acoustics. Simulations showed strong similarities between a cluster of modes and sound power level measurements. Simulations with oil variation, temperature variation and inclination also increased the strength of this statement.

Finally, the observations made from the simulations were confirmed experimentally by testing a compressor with R600a refrigerant, varying its oil level in steps of 2.5g, measuring its running noise with different speeds.

The correlation of oil increase versus cavity noise decrease was confirmed, but another harmful effect was observed at 4kHz and 5kHz bands. This was due to the oil level touching moving parts and transmitting the vibration directly to the shell. This effect was also pointed out by [3], however there are few studies available.

We recommend further studies regarding the direct transmission of vibration through oil, and its effect on the total noise from the compressor.

References
[1] Bucciarelli, M.; Giusto, F.; Cossalter, V.; Lio, M. Da; and Gardonio, P., "Modal Analysis of a Compressor Shell and Cavity for Emitted Noise Reduction" (1992). International Compressor Engineering Conference. Paper 922. https://docs.lib.purdue.edu/iccc/922
[2] Albrizio, F.; Genoni, C.; Bianchi, V.; and Frontini, G., "Noise Reduction of Hermetic Compressor by Identification of the Gas Cavity Resonance" (1990). International Compressor Engineering Conference. Paper 753. https://docs.lib.purdue.edu/iccc/753
[3] Soedel, Werner, 1936, Sound and vibrations of positive displacement compressors / by Werner Soedel. p. cm. ISBN 0-8493-7049-3
[4] Ansys® Mechanical APDL 2019 R2, help system, Chapter 7: Element Library, FLUID221, ANSYS, Inc.

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