Vibroacoustic characterization of flexible hoses for air conditioning systems

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Introduction

One of the most interesting refrigerant fluid for automotive air conditioning systems is the carbon dioxide (i.e. CO_2), a natural inert gas available in nature, no toxic, no inflammable, harmless for the biosphere, compatible with the refrigerant components materials (metal, plastic, rubber) and lubricants. Therefore, this fluid, named R744, in terms of safety, has better characteristics with respect to the traditional refrigerant fluids like the R134a. The main drawback of the R744 is the thermodynamic cycle, see Figure 1, completely different from the R134a one, called "trans-critical cycle" because it works between two isobars below and above the critical temperature, close to 31°C, that is the maximum summer temperature in tempered climate countries. Besides, the discharge line downstream the compressor works at very high pressure, up to 133 bar while the high pressure line of R134a works between 18 – 23 bar. These high pressure levels inside the pipeline can produce both leakage and efficiency loss and high dynamic excitation levels so that an accurate vibrational characterization of the high pressure line is necessary.

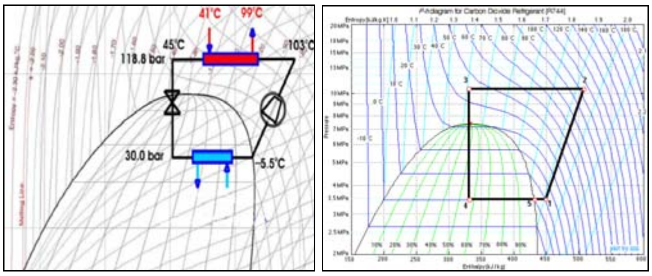


Figure 1: R744 thermodynamic cycle and p-H diagram

For the complete characterization of the fluid inside the high pressure line the fluid-dynamic data have been measured; they are summarized in Table 1.

Point	Temp. (K)	Press. (bar)	Density (kg/m^3)	Velocity (m/s)	Sound speed (m/s)
2	378	105	194	5.34	298
3	319	105	560	3.14	274

Table 1: R744 thermodynamic parameters in the discharge line

A test bench reproducing an R744 refrigeration system and cycle, typically used in car air conditioning, has been realised and used for the dynamic characterization of the high pressure pipe, see Figure 2.

The pipe used in the high pressure line, studied in this paper, has been designed to resist at severe operating conditions: pressure up to 133 bar and temperature up to 450 K. The permeability, which has to be maintained as low as possible, plays the most important role in the hose design so that the pipe, so called Shark F4, has a complex structure. It is made of several layers: an internal corrugated steel pipe to avoid CO_2 leakage, rubber to resist at high temperature, steel braid for pressure maintenance, second rubber layer, see Figure 3.

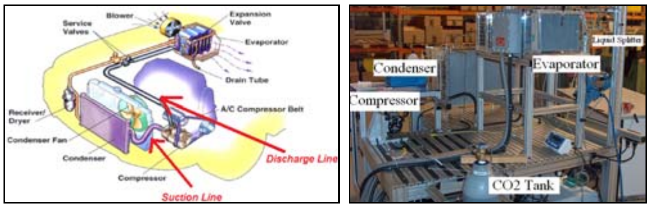


Figure 2: R744 refrigeration system: air conditioning application in a car (left) and experimental test bench (right)

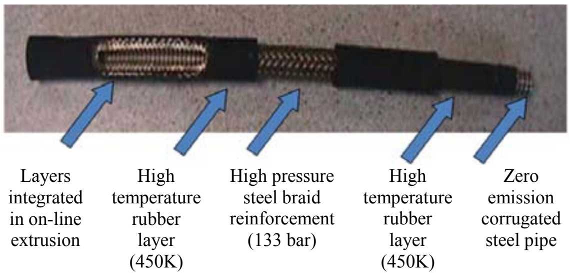


Figure 3: Shark F4 hose layers composition

Dynamic characterization

The dynamic characterization of pipelines conveying fluid is an important issue in many application fields, i.e. pipelines arrays in steam generators, pump discharge lines, reactor systems, etc. Therefore extensive research studies are present in literature focused on modelling flow-induced radial, axial and transverse vibration of pipes [1-2]. In the case of pipelines conveying carbon dioxide in refrigerator systems, the fluid-pipe interaction can deeply influence the dynamic behaviour of the system, the operating pressure of the internal fluid being very high (above 100 bar). The high pressure pipeline have been studied thus extensively in two steps: (i) dynamic analysis of the empty pipe in free-free-conditions, (ii) dynamic analysis of the pipe mounted on the refrigeration system and operating in real working conditions.
The dynamic models of pipes given in literature are based on the theory of beams or cylindrical shells. For the Shark F4 hose the theory of single-span beams has been taken as reference [3], considering in the model a sandwich cylindrical slender beam made of uniform layers of different materials glued together.

When a fluid flow passes through the pipe, the dynamic of the system changes, due to the added mass of the fluid itself on the pipe mass. This justifies the decrement of the natural frequencies. If the fluid flows at very high velocity the pipe can buckle and enter into instability, therefore it is very important to understand its dynamics in this condition. For the Shark F4 hose in working conditions the critical velocity for buckling the pipe is 827.3 Hz, while the carbon dioxide operating velocity is 5.34 m/s, and therefore it does not induce instability on the system. Also the natural frequencies of the pipe are not affected by such a low velocity. Another factor influencing the pipe dynamics, when traversed by a fluid flow, is the internal pressure of the fluid [4]. For the Shark F4 in operating conditions the carbon dioxide pressure is very high (105 bar). An internal pressure (*p*) produces an axial force on the pipe F_A (pA_m, with A_m the mean cross-section area), which, in the Shark F4 case, is of 892 N. The tensile load will change the natural frequencies of the pipe even if this is smaller than the axial load required for buckling the pipe. When considering the Shark F4 hose in operating conditions, the edge constraints should be changed because it is mounted on the refrigeration system. In practice, it can be considered as simply supported at both ends. The critical axial load for buckling the pinned-pinned pipe is 2810 N and is larger than the actual axial load due to the internal pressure. In any case this induces a frequency variation, see Table 2.

The Shark F4 hose dynamical behavior has been tested in operating conditions, by a Laser Doppler Vibrometer (LDV), measuring the surface vibration velocity over two lines of 16 points each on the pipe. The modal test has been performed at several compressor speeds between 1000 and 5000 rpm with a step of 500 rpm.

Natural frequency at zero flow velocity and pressure (Hz)	Natural frequency with fluid flow at zero pressure (Hz)	Natural frequency with fluid flow at high pressure (Hz)
130	130	100
519	519	363
1169	1169	801
2078	2078	1413
3246	3246	2200

Table 2: Effect of the internal fluid velocity and pressure on natural frequencies

As reference signal, for the modal analysis, has been used an accelerometer mounted close to one end of the pipe. In order to monitor the pressure field inside the hose, a pressure transducer has been placed in the connection between the Shark F4 hose and the pipeline coming from the compressor. Since the vibration of the test bench was very important in comparison to the hose vibration, because of the high vibration levels induced by the alternative compressor, the pipe has been mounted on a rigid support isolated from the rest of the test bench, see Figure 4.

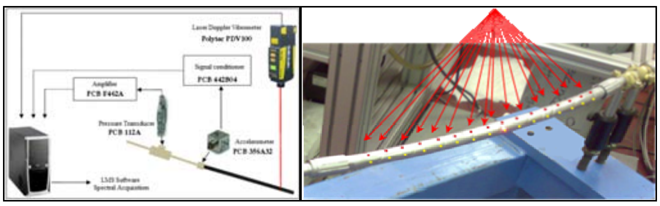


Figure 4: Experimental set-up for the analysis of the Shark F4 hose in operating conditions

The dynamic pressure measured inside the pipe monitored by the pressure transducer is shown in Figure 5. The time history evidences an high component at low frequency of amplitude of about 10 bar (this is the pressure oscillation around the DC pressure of 105 bar that is the carbon dioxide pressure in the high pressure line). The low frequency component is 16 Hz and this represents the acoustic resonance of the fluid inside the entire pipeline, from the compressor to the condenser, whose length is 4.6 m. This frequency is visible in the pressure signal spectrum, together with the second acoustic resonance at 32 Hz and the frequency of the compressor at 50 Hz, it running at 3000 rpm. The higher harmonics of the compressor speed modulated by the first acoustic resonance of the pipeline are also evident.

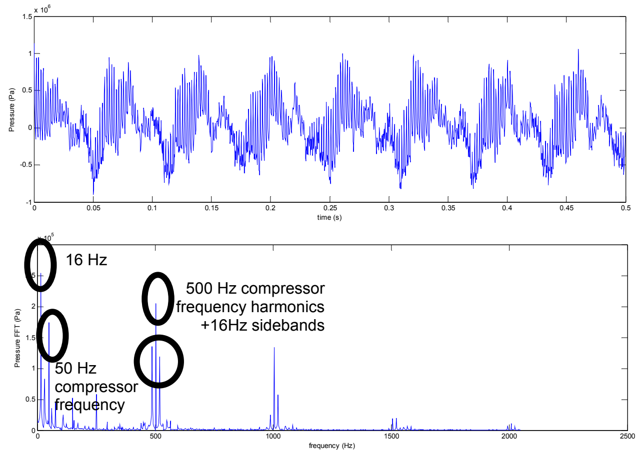
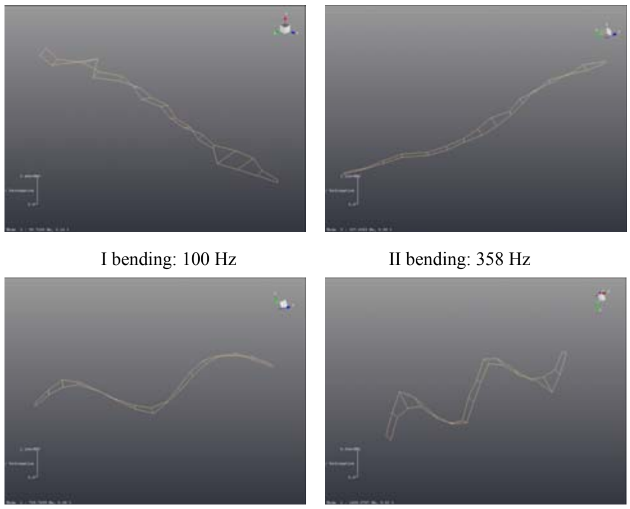


Figure 5: Dynamic pressure measured inside the pipeline: time history (top), FFT (bottom)

The natural frequencies and operational deflection shapes obtained by the modal analysis of the acquired data are reported in Figure 6.



I bending: 100 Hz

II bending: 358 Hz

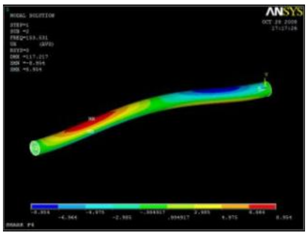
III bending: 760 Hz

IV bending: 1400 Hz

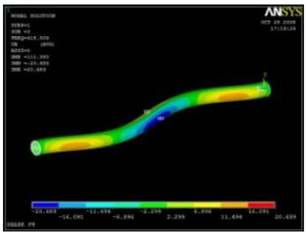
Figure 6: Shark F4 pipe natural frequencies and operational deflection shapes in operating conditions

A numerical model, based on Finite Element Method (FEM), of the Shark F4 hose has been also realized. After having performed an updating of the model for the empty and free-free pipe, the model has been generalized for the pipe mounted on the refrigeration system and excited via wave pressure, produced by the fluid forced into the hose by the compressor. The natural frequencies and mode shapes calculated by the FE model are given in Figure 7.

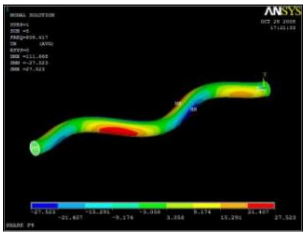
The comparison between analytical and experimental natural frequencies is given in Table 3 where it can be noticed that the maximum discrepancy is of 5.3% for the third bending mode. In the same table the natural frequency differences between experimental and numerical models for the first four bending modes are summarized.



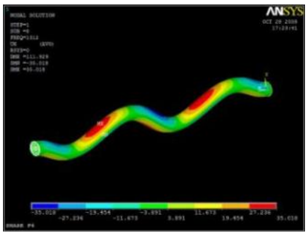
I bending: 103 Hz



II bending: 367 Hz



III bending: 808 Hz



IV bending: 1312 Hz

Figure 7: Shark F4 pipe numerical natural frequencies and mode shapes in operating conditions

Analytical natural frequencies (Hz)	Numerical natural frequencies (Hz)	Measured natural frequencies (Hz)	NFD A-M (%)	NFD N-M (%)
100	103	100	0	3.0
363	367	358	1.5	2.5
801	808	760	5.3	6.3
1413	1312	1400	0.9	6.3

Table 3: Comparison between analytical, numerical and experimental natural frequencies

Acoustic characterization

The acoustic characterization of the high pressure pipe, mounted on the refrigeration test bench, has been performed experimentally by mapping the acoustic intensity emission over a closed surface surrounding the Shark F4 hose, according to the standards UNI-ISO 3744 and ISO 9614-1. This kind of measurement is very difficult being the hose emission comparable, and even low, with respect to the other bench component emission, first the compressor. To increase the measurement accuracy the hose has been isolated from the rest of the bench, i.e. it has been placed outdoor far about 5 m from the wall of the laboratory (Fig. 8). In this configuration the noise floor was 39 dB (very low if only the compressor SPL was 83 dB). The intensity radiated by the hose has been measured over a cubic grid of 96 points. The acquisition time for each position was 25 s.

The possibility of predicting the acoustic emission via Boundary Element Codes (BEM) allows avoiding this kind of complicated measurement. Therefore a BEM model has been realized using the propagation of the experimental surface vibration previously measured, via dynamic characterization. In practise, vibration velocities measured on the hose surface have been applied as boundary condition into the acoustic model. The acoustic propagation has been calculated on a 96 points cubic surface exactly alike the one made experimentally.

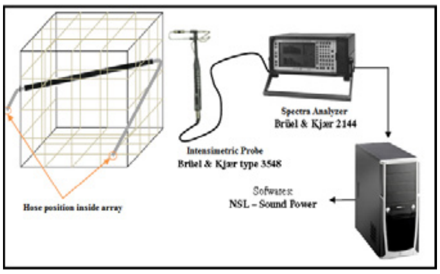


Figure 8: Measurement set-up

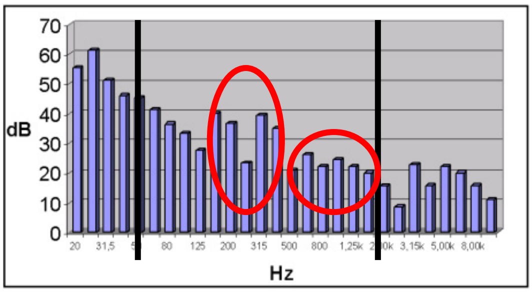


Figure 9: Experimental power spectrum at 1500 rpm

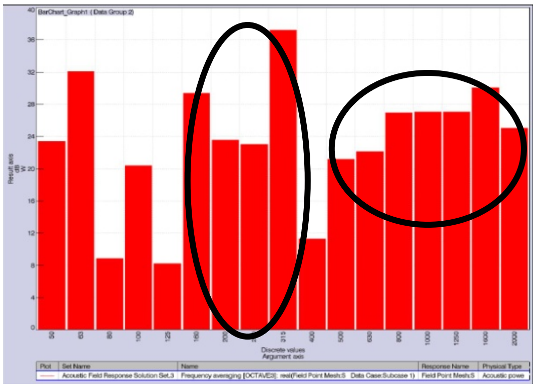


Figure 10: Numerical power spectrum at 1500 rpm

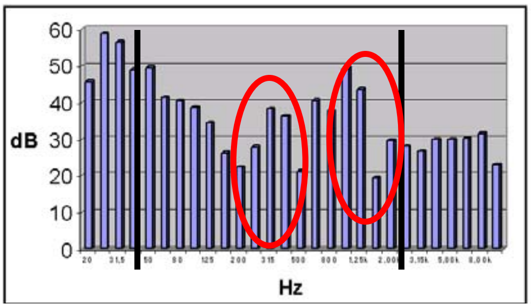


Figure 11: Experimental power spectrum at 3500 rpm

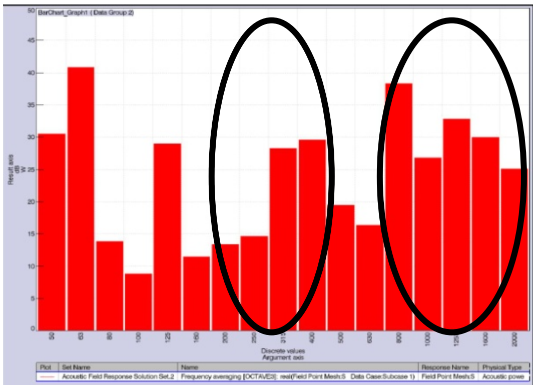


Figure 12: Numerical power spectrum at 3500 rpm

Figures 9, 10, 11 and 12 show the experimental and numerical hose power spectrum at two compressor speeds: 1500 and 3500 rpm. Comparing the Figures 9 - 10 and 11 – 12, it can be observed that the power spectrum trend, in the frequency range of 50 – 2000Hz, is really similar. The main difference is visible at the minima where the calculated emitted power is lower than the measured one, since the model does not take into account the noise floor.

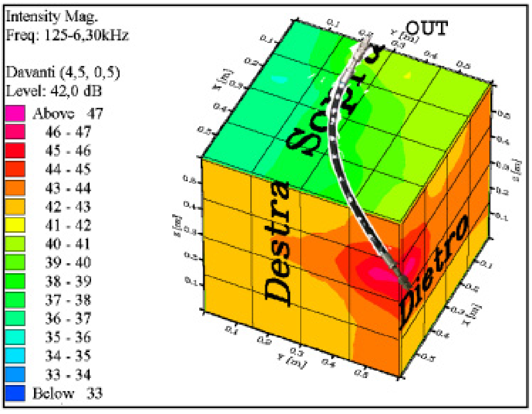


Figure 13: Experimental acoustic emission at 1500 rpm

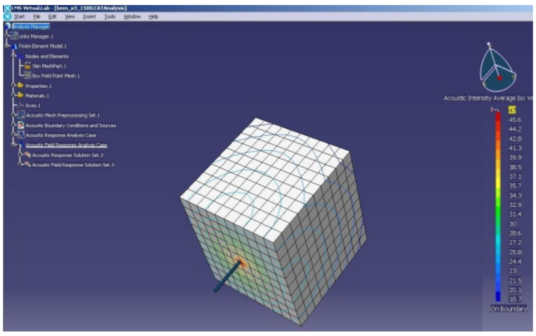


Figure 14: Numerical acoustic emission at 1500 rpm

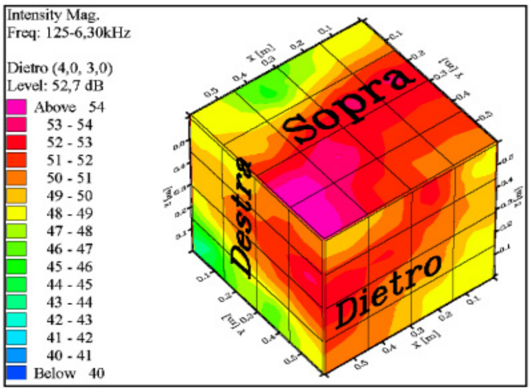


Figure 15: Experimental acoustic emission at 3500 rpm

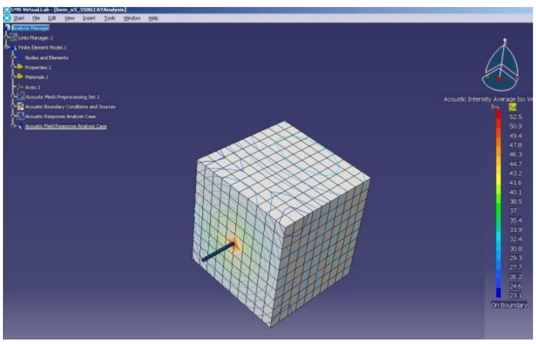


Figure 16: Numerical acoustic emission at 3500 rpm

Figures 13, 14 and 15, 16 show the experimental and numerical acoustic intensity calculated on the measurement surface respectively at 1500 rpm and 3500 rpm. Again, the acoustic intensity distribution and maximum level are the same but the calculated minima are lower than the measured ones because of the noise floor.

Conclusions

This work illustrates the complete vibro-acosutic characterization of the high pressure pipeline of an automotive air conditioning system working with R744 fluid instead of the conventional R134a. The high pressure levels reached with the different thermodynamic cycle within the discharge line requires an accurate characterization of the hose connecting compressor and condenser in terms of vibration and acoustic emission. The pipe has first been studied from the dynamic point of view by performing experimental tests and building an accurate numerical FE model. The acoustic field radiated by the pipe has been measured with a sound intensity probe in order to map the acoustic intensity on a 3D surface surrounding the pipe, according to the standard UNI ISO 3744 and UNI 9614-1. The acoustic test, however, presents severe limitations and inaccuracy due to the difficulty to isolate the pipe from the rest of the test bench, the compressor, in primis. Therefore the by-pass of this kind of test is required, and for this reason, the numerical calculation of the radiated field by BEM can be a good solution. The BE method has been applied to the pipe using as boundary conditions the structural velocities measured by the LDV on its surface. Therefore the final vibro-acosutic model can be considered as an hybrid tool to predict the acoustic emission of the pipe. Having realized also a dynamic FE model of the hose, this can be used as boundary conditions for the BEM and therefore a complete numerical model describing the vibro-acoustics of the system can be built.

References

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