Flexible hoses dynamic characterization including amplitude and frequency dependency

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Abstract
Hoses are very common vehicle components. Unfortunately, they provide a pathway for the transfer of structure-borne noise between active components and the interior of the passenger compartment. Their role in total vehicle acoustics becomes even more important in electric cars, where their contribution to the total noise of electromechanical components is no longer masked by the sounds of the internal combustion engine. Although widely used, hoses are difficult to model, especially with regard to their dynamic behavior, which is usually unknown. In this paper, a methodology for experimentally characterizing dynamic properties of hoses is presented. A newly developed test rig allows these properties to be derived directly from bending, shear, compression and torsion stiffness measurements on a short hose sample, as a function of frequency and amplitude of excitation. The experimentally determined parameters are used to generate numerical models of longer hoses. Mechanical transfer functions of the simulated long hoses are validated by experiments.

1 Introduction

In the automotive world, a major evolution is going on. Electric vehicles are catching market share, requiring novel engineering designs and solutions. In this scenario, many subsystems and components need new development either from scratch or as evolution of existing system to be better integrated in the electrified fleet, generating new challenges for the NVH engineers.

Among the many different components to be found in a vehicle, the HVAC systems are greatly interested by the electrification. While with internal combustion engines, HVAC systems had almost the sole purpose of air-conditioning the vehicle; with electrified cars, they need to be functioning also when the vehicle is standing still and when battery fast charging is being used. From the NVH point of view, the problem is further enhanced by the lack of the masking effect of the ICE that in the past was covering the unwanted noise from those electromechanical components

A special role in the whole HVAC acoustic is carried by the different connecting elements. In fact, noise and vibration are transferred from the operating components to the rest of the vehicle, not only by the classical isolating mounts, but also by the piping system that play an important role to the overall vehicle acoustic. In this paper, the focus is on those components, the flexible hoses.

As piping are in general more common in electrified vehicles, the new methodology appear to have a large impact on the NVH development of a new cars, especially in consideration of the preferred simulation approach toward the experimental one. The latter being available only at very advanced levels of development where allowable design changes are very limited. Having these limitations in mind, the possibility of correcting and modifying hoses' routing, after the first design is practically no longer possible. Thus, numerical models must be able to provide sufficiently accurate solutions to avoid undesirable phenomena due to bad placement and characteristics of these components. In the past, this problem has been tacked by other researcher such as [1] which although do not provide a methodology for the simulation of hoses' dynamic behavior in a complete system simulation.

Numerical methodologies usually used in the NVH field, such as blocked forces, dynamic sub-structuring and TPA (in all its declinations), are not directly applicable to components as flexible piping that, since due to their complexity (geometry and material) are realized only when the design is finalized. The design approach must than be fully numerical and the hoses' properties have to be described by a sufficiently accurate virtual model.

Since at the beginning of development, the flexible hoses exist only as material sample, (straight pipes of different lengths), only this can be used for the determination of the dynamic properties of the final product based on the following four dynamic stiffness:

- Tension-Compression
- Bending
- Shear
- Torsion

These four quantities, determined as complex functions, to represent both the dynamic stiffness and damping of the component are obtained independently with dedicated experiments in which both amplitude and frequency of excitation are controlled. As well, the temperature, the pressure (related to the fluid characteristics) and the static pre-load (linked to the mounting tolerances) are taken into consideration.

In this work only the frequency dependency is explained in detail, other parameters effects are studied separately and are not included in this study.

It's worthwhile mentioning that the material of flexible hoses, it is not homogeneous. Especially because of the pressure requirements they must meet, hoses used in vehicle refrigeration systems are composed of layers of very different materials. For this reason, it is not sufficient to determine the Young's (E) and Shear Modulus (G) to obtain a characterization of the material. A full description of the material properties is given when knowing EA, EI, GA and GJ, which dimensionally represent stiffness by hose's length and section dimensions.

With the above mentioned parameters, experimentally determined for different samples' lengths, it became possible to simulate longer hoses. The derived numerical model will be capable to predict the system frequency and amplitude dependencies like those experimentally observed. Due to the amplitude and frequency dependencies, methods based on modal superposition cannot be used for simulation purposes. Therefore, the adopted solution is based on Direct Frequency Response calculations.

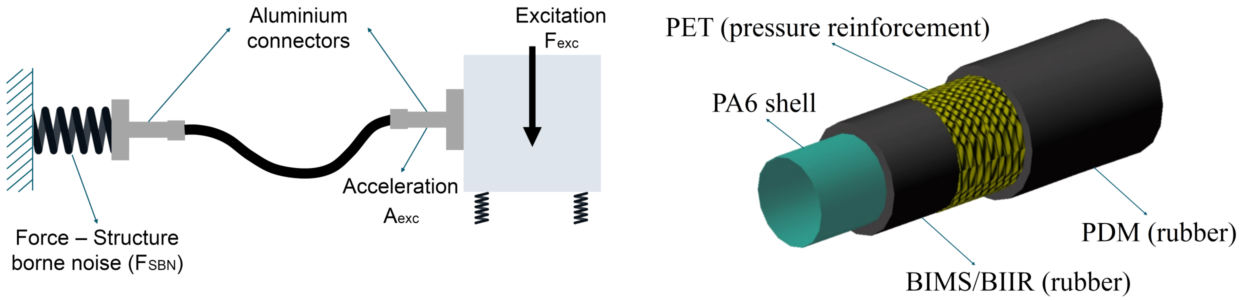
It is worth mentioning that the proposed approach is a step forward to a better industrial design tool for the hoses layout, but it is not meant to be the definitive tool for multi-material and multi-layered flexible components simulation.

2 Technical background

Electromechanical components, as the one represented in figure 1a, are usually mounted in vehicles with isolating elements. By doing so the component is dynamically isolated, and the vibration and dynamic forces transmitted to the rest of the vehicle are minimized. From these components hoses, tubes and cable are used to transfer operational fluids to other components and parts of the vehicles. For assembly and package reasons those flexible components are usually fixed to the chassis or to other parts.

Regarding the refrigerant gas compressor specifically, it has been shown that in the low-frequency range, the pipes themselves transfer a non-negligible if not considerable part of the noise and vibration into the vehicle [2].

Hoses like the ones used for the refrigerant gas of the AC system, must operate with high pressure, both on the suction and pressure line. For this reason a reinforcement layer is used, as it could be seen in figure 1b. The flexible hoses used for fluids and gases are in general constituted of different layers, each made of a different material, making the material far from being homogeneous.



(a) Structure borne noise from electromechanical component - scheme

(b) Hose section view

Figure 1: A/C flexible hoses

3 Methods

In this section the procedure for the experimental characterization of the hoses' characteristics is described and the results are presented. Following the numerical approach used is described and discussed.

3.1 Experimental characterization

For the experimental characterization of the hoses' dynamic stiffness, an on purpose built test-bench is used. The experimental tests are performed on hoses' samples of different lengths (specifically 2cm, 3cm, 4cm and 5cm), under the assumption that for those values, the mass effects of the hose itself can be negligible. With this assumption, the considered hose's specimen can be approximated by a massless ideal spring.

The hose sample is mounted between two sufficiently rigid bodies, which are respectively named *active* and *passive*, as represented in figure 2. On each, 4 tri-axial accelerometers are mounted, taking a lot of care for their positioning and orientation. The active side is excited with the desired excitation, that for the specific case is a constant displacement with a variable frequency sweep. For the measurement a SCADAS acquisition system has been used. Overall 8 accelerometers have been used. 4 of the type 356A15 from PBC, having a sensitivity of $100\frac{mV}{g}$ and other 4 of the type 356B18 from PCB having a sensitivity of $1000\frac{mV}{g}$. For the excitation 2 shaker type TV 51110 from TIRA, having a rated peak force sine of 100 N. The use of shaker for the excitation has been required for controlling the excitation's amplitude.

Since the dynamic forces transmitted to the passive side are very small, these are calculated trough the dynamic of the passive body, measured with the aforementioned accelerometers, thanks to the Newton relation linking loads and accelerations. With the measured and derived dynamic behavior of the two bodies, the four sought dynamic stiffness are derived.

Before proceeding with further analysis the experimental data are fitted in order to obtain smoother curves which can also be parameterized. For each DoF considered a different range of frequency is used for the fitting, as reported in figure 3.

For the sake of brevity only the experimental results for the 4 cm sample are shown, while the other lengths have a comparable quality.

The dynamic stiffness is derived as complex function and in this way both the stiffness as well as the damping of the hose are determined.

While using very short samples has the advantage of reducing mass effects, and thus avoiding the presence of acoustic modes in the considered frequency range, it has the negative effect of introducing additional boundary effects. Fixing the measured pipes-both on the active and passive side-introduces a constraint error, which is mainly observed when measuring compression and bending, as also reported in the work of Koblar et al. [3] and in that of Goebel [4]. For torsion and shear, on the other hand, this effect does not affect the measurements.

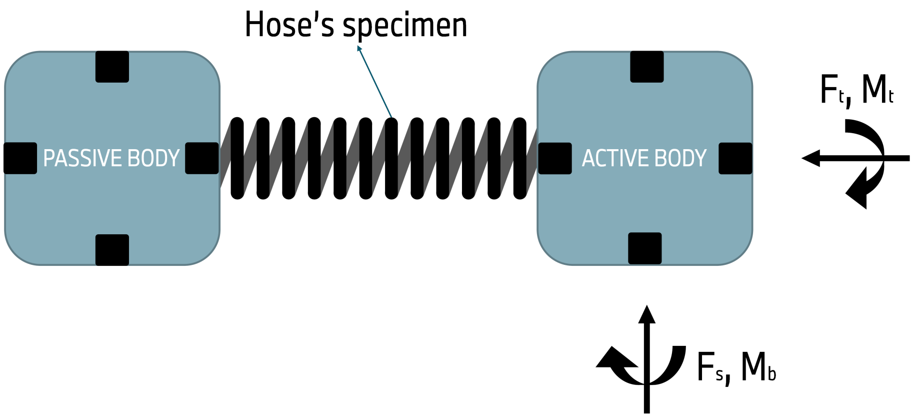
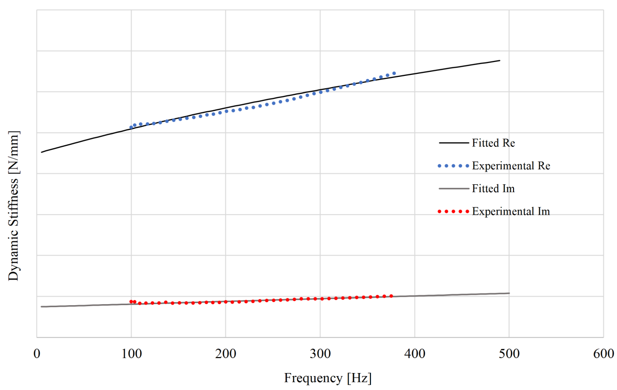
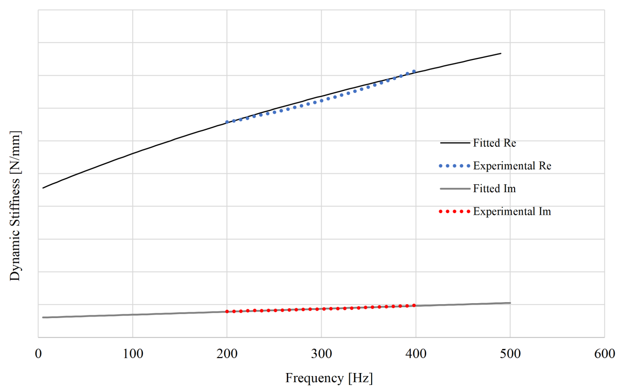


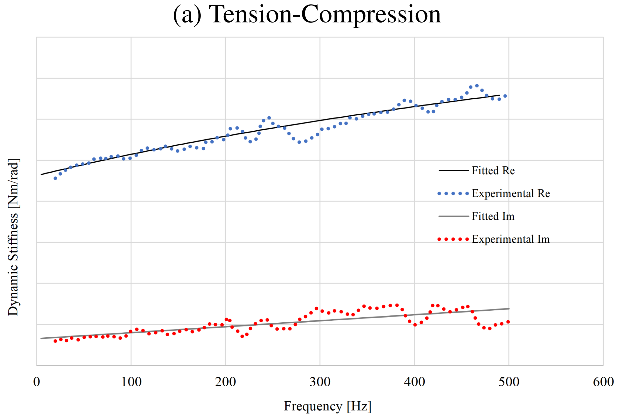
Figure 2: Parameter identification from short hose's specimen



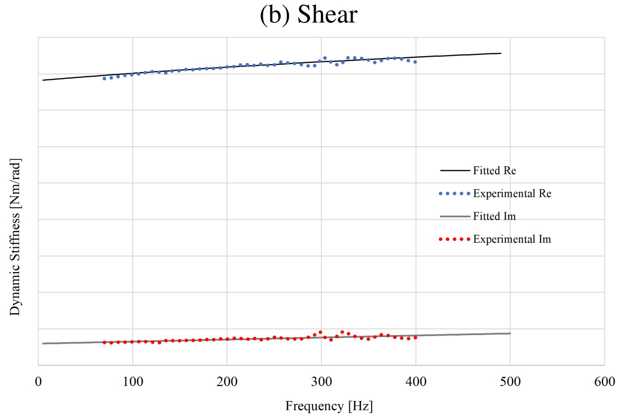
(a) Tension-Compression



(b) Shear



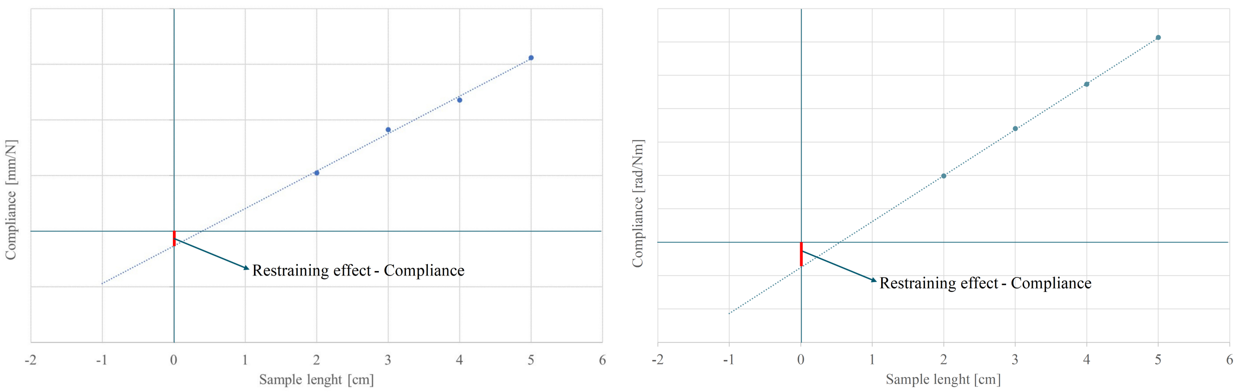
(c) Bending



(d) Torsion

Figure 3: Dynamic stiffness values of a 4 cm sample - Experimental and fitted data

In order to correct for this effect, which is dependent on the length of the test considered, 5 different samples of respectively: 2cm, 3cm, 4cm and 5cm are measured. In figure 4 the restrain effects on the tension-compression and bending are represented. This analysis is performed considering the compliance and not the dynamic stiffness. For each of the two DoF considered the restrain effect reduce measured compliance.



(a) Restraining effect - Tension

(b) Restraining effect - Bending

Figure 4: Restrain effect on the short samples for the tension-compression and bending DoFs

While tension-compression, torsion and bending can be completely isolated and exactly measured on the test bench, this is not valid for the shear. In this case the determined stiffness, due to the restraining conditions, is composed of both a shear and a bending component. It is in this case better to talk of a cross stiffness c_c, which in fact includes the effect of both bending (c_{cB}) and shear (c_{sS}) stiffness, which are acting in series, and can be written as in equation 1, where the inverse of the stiffness represent the compliance of the system.

$$\frac{1}{c_{cross}} = \frac{1}{c_{cB}} + \frac{1}{c_{cS}} \tag{1}$$

where the contribution to the overall cross-compliance due to bending compliance can be written as:

$$\frac{1}{c_{cB}} = \frac{l^3}{12EI} \tag{2}$$

and the contribution to the overall cross-compliance due to the shear-compliance only can be written as:

$$\frac{1}{c_{cS}} = \frac{l}{Ak_sG} \tag{3}$$

where A is the area of the hose's section, k_s is the Timoshenko shear coefficient and G is the shear module and l is the length of the measured beam. This component represent the actual value which it is want to determine.

At his point, recalling the definition of the bending stiffness, which can be written as:

$$\frac{1}{c_B} = \frac{l}{EI} \tag{4}$$

where, E is the Young modulus, I is the second moment of area and l is the length of the beam. Substituting equation 4 into equation2, the contribution due to the bending-compliance can be rewritten as:

$$\frac{1}{c_{cB}} = \frac{1}{c_B}\frac{l^2}{12} \tag{5}$$

Substituting equation 5 into equation 1, and solving for $\frac{1}{c_{cS}}$ it is possible to obtain the desired shear compliance, from which the bending compliance has been subtracted, resulting in:

$$\frac{1}{c_{cS}} = \frac{1}{c_{c_{cross}}} - \frac{l^2}{c_B 12} \tag{6}$$

From the calculated and corrected dynamic stiffness values in the considered four DoFs it is possible to determine the equivalent material properties of the hose. Namely for each excitation and dynamic stiffness estimation, the Young's (E) and Shear (G) modulus are determined. Due to the nature of the hose, which is constituted of different layers of different materials, the estimated material properties do not fit in a classical homogeneous material model. In this latter case the material properties should, in fact, not violate the expression: $G = \frac{E}{2(1+\nu)}$. As reported in figure 5, the E modules derived from the tension-compression measurement and the G module derived from the torsion measurement on the same sample violate the above mentioned formula. The identified G module from the torsion has an higher value than the E module derived from the tension- compression test over the complete range of frequency. The reason for that is most likely to be searched in the PET-reinforced fiber used to make the hose compliant with high operating pressures.

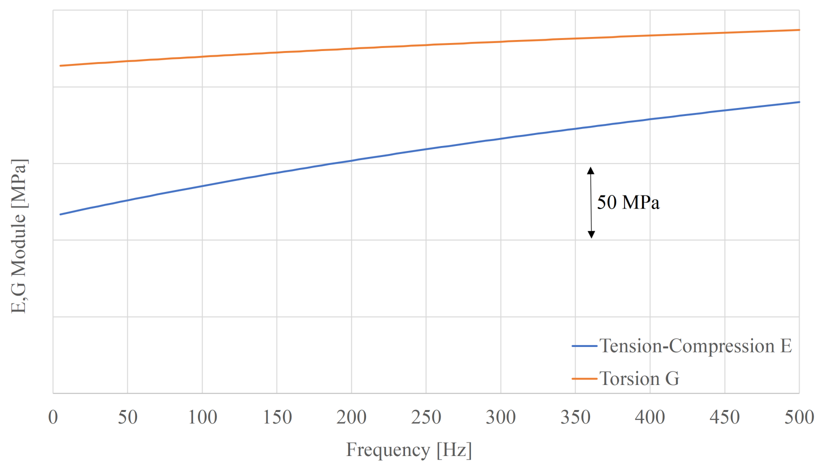


Figure 5: Young's and shear module for a 4 cm hose extracted from Tension compression and torsion excitation respectively

The aforementioned amplitude dependency appears to be much less important than the frequency in the overall variation of the dynamic stiffness as it can be seen in the figure 6 where the resulting dynamic stiffness in the tension-compression cases has been obtained exciting the system with a full order of magnitude difference.

3.2 Computation method

The fact that the mechanical characteristics of the considered hoses are frequency dependent means that for each individual frequency there are different system matrices $[C]$ and $[K]$.

Furthermore, in deciding what type of solution to use, methodologies involving the use of modal superposition cannot be considered. In fact, the solution of modal analysis does not involve modeling frequency-dependent materials in standard simulation tools.

The ideal solution when dealing with materials that exhibit this type of characteristic is to use Direct Frequency Response. In this type of solution, the system matrices are inverted directly for each considered frequency line. This approach, impractical when dealing with very large numerical models due to the demanding calculation power, is applicable if the models are small.

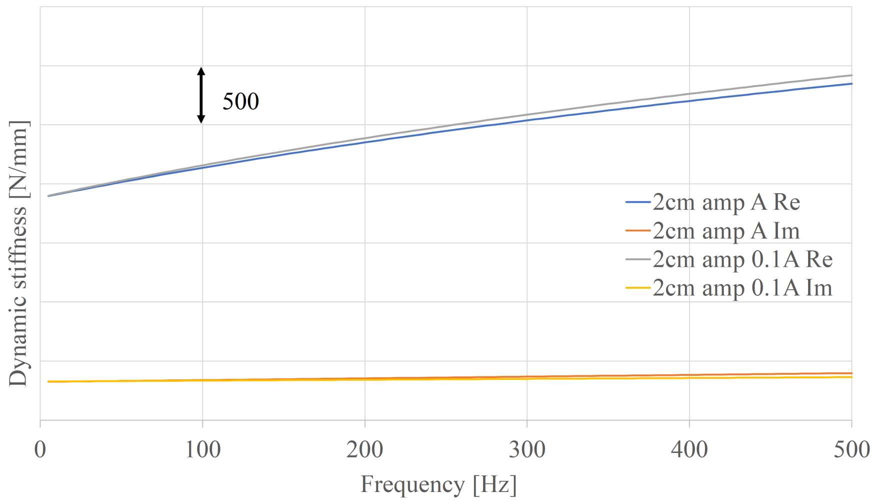


Figure 6: Excitation amplitude dependency on dynamic stiffness - 2cm sample

$$-[M]\omega^2 + j[C]\omega + [K] = F \tag{7}$$

with:

- $C = C(\omega, a, T)$
- $K = K(\omega, a, T)$

To speed up the results' computation, and to minimize the impact of the direct frequency response approach, it has been decided to model the hoses' dynamic with 1D elements in Nastran. The length of each element is restricted to $1cm$ only.

To compensate for the non homogeneous material properties, and include the four elaborated stiffness, four 1D non standard beam elements are used in parallel

The first three elements are modeled as shown in the table 1. This modeling is possible using beam elements with user-defined characteristics; by doing so, compression, torsion, and bending-shear characteristics can be decoupled. Regarding the latter stiffness, it is unfortunately not possible to decouple bending and shear numerically. This, must therefore be modeled by a single element. To each of the beam is then associated a specific material, which describes the experimentally observed behavior and results.

The fourth and final element is used to model the mass of the tube, which is not included in the other elements.

Table 1: Modeling with 4 parallel beams

DoF	A $[mm^2]$ Area	$I_{xx,yy}[mm^4]$ Area moment of inertia	$J[mm^4]$ Torsional Stiffness	K_1, K_2 Shear Stiffness factor
Tension	x	0	0	0
Bending and Shear	0	x	0	x
Torsion	0	0	x	0

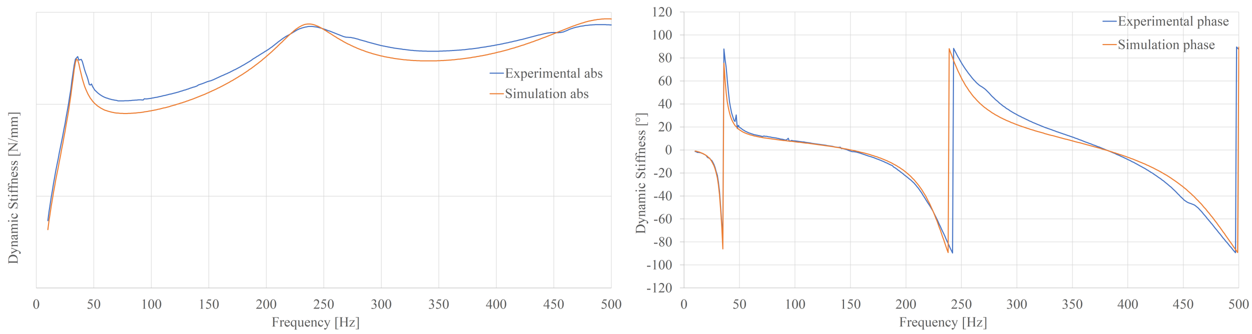
3.3 Results

By using the experimentally determined dynamic hoses' properties in combination with the proposed modeling approach, it is possible to simulate the dynamic behavior of hoses of different lengths. For this study, the validation of the methodology has been conducted on a straight suction hose with the length of 80cm. The realised numerical model consists of 80 elements' group, each with a length of 1 cm and, as previously described, combining 4 beam elements, between each couple of nodes.

Similar to what was done for the properties identification of the short hoses' samples, two extra bodies are used in the tests. One attached to the active side and one to the passive side of the system, similarly to what shown in figure 2. The difficulties related to the measure of small forces and moments remain the same also when measuring long hoses, but the two extra masses allow for accurate accelerations measurements at both ends of the hose. The properties of the masses and inertia information of the two terminal bodies are well known, as well as the position of the accelerometers used to evaluate their dynamics.

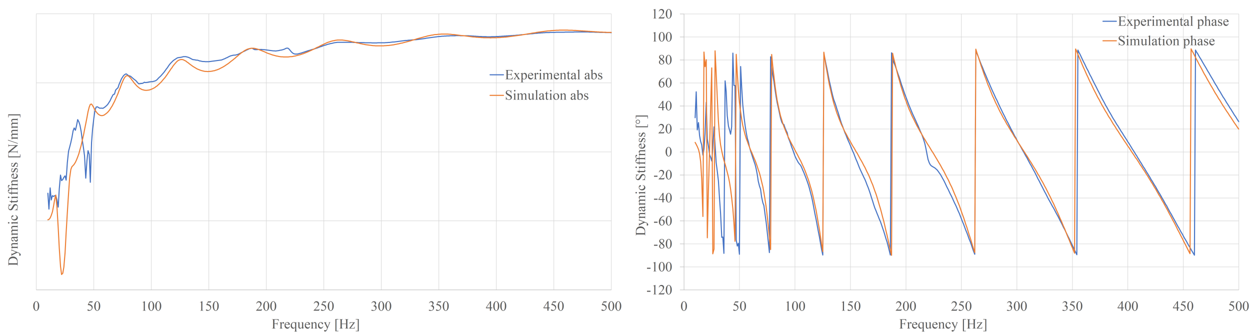
The excitation used to compute the transfer function is again a constant displacement over variable frequency. Specifically with a value of 0.005 mm between 10 Hz and to 500 Hz both for the axial and orthogonal excitation. For the transfer function of torsional nature, a constant rotation of 1° between 10 and 500 Hz has been used.

Differently for the short sample, the long hose presents flexible modes in the considered frequency range. To compare results between simulation and experimental data, the dynamic stiffness is used, considering the forces per length and the moment per radians of the measured tube.



(a) Magnitude (b) Phase

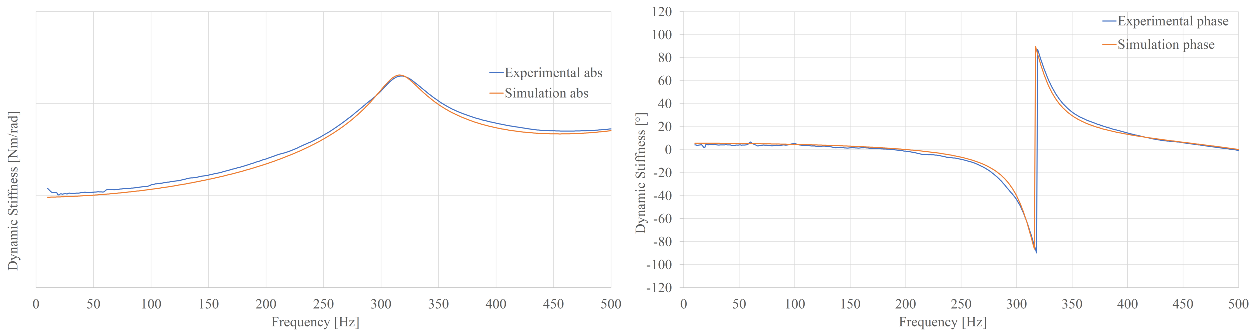
Figure 7: Torsion/Compression - Comparison between simulation and experimental measurement on a 80 cm long straight hose



(a) Magnitude (b) Phase

Figure 8: Orthogonal - Comparison between simulation and experimental measurement on a 80 cm long straight hose

The results reported in in figures 7, 8 and figure 9, show the trends of the dynamic stiffness for the experimental and numerically obtained results. The difference between the curves is limited, both for the magnitude and phase information. We apologise for the omitted Y scale values that are covered by the company secrecy policy. Some discrepancies are still visible in the low frequency range (between 10 and 50 Hz) for the orthogonal excitation, probably due to the coupled bending-shear coupled effects. The matter is under investigation at the time of writing this paper. The 80 cm long hose exhibits in the axial and rotational excitation



(a) Magnitude (b) Phase

Figure 9: Torsion - Comparison between simulation and experimental measurement on a 80 cm long straight hose

respectively 3 and 1 flexible mode in the frequency range of interest. In the orthogonal direction the number of detectable modes is 6, this is compatible with the lower stiffness in this direction if compared with the axial and torsional cases.

4 Conclusion

The paper presents a combined experimental and numerical method to characterize the load transferring capabilities of hoses used in the HVAC system of electrical powered vehicle. These elements are becoming critical, being capable to transfer loads and modify the car acoustic performance. The proposed methodology allows to determine the mechanical properties of the hoses, starting from very short elements measurements to arrive at prediction of the dynamic behaviour of long elements. The measurements allow to derive 4 dynamic stiffness EA, EI, GA and GJ values, for an unitary length element and from those, derive the four FE elements characteristics used to model the complete hose. The non homogeneity of the hose is well captured by the proposed multi elements approach, as the results reported in FIG8 display, that could not be replicated by any standard homogeneous material numerical model. Further test are in progress for more complex and more industrial hoses' layout. The next validation in fact will comprehend pre-formed and curved hoses and the effect of temperature. This latter point is particular difficult to handle due to the high gradient between the inside temperature of the cooling fluid and the outside temperature of the engine compartment.

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