Approche simplifiée de la prestation vibro-acoustique par hybridation calculs/essais des sources secondaires

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The characterization of the secondary sources of vibration is a staple step to predict and decrease the contribution of the structure-borne sound and, thus, the vibrations into the transport vehicles to reach a good level of vibro-acoustics performance. The OEMs and Tier-1 suppliers have included into their development expensive numerical methodologies allowing to characterize the active (sources) and passive (body). These methodologies are unaffordable to small-size suppliers and manufacturers because of their price.

A simplified approach coupling tests and FEM analysis is thus proposed. This test/FEM analysis hybrid methodology allows to substructure the vibratory transfer from the source to the driver's ears. It consists in isolating the sources of vibration and the elements participating to structure-borne noise (subsystems) on a test bench and to transpose the efforts generated by the active components (sources) to the vehicle. To do so, the subsystem is modeled numerically. The blocked forces of the system on the test bench are computed and readjusted with accelerometric measurements. Analytic formula based on the methodology from the XP 19-701 norm allow to deduce the efforts injected to the vehicle thanks to blocked forces.

This methodology, validated numerically, aims to optimize the designs of the components, the elastomeric mounts and the hosting structure to improve the vibro-acoustics performance.

1 Introduction

In the automotive industry, to predict the vibro-acoustics performance of a vehicle, the OEMs and the Tier-1 suppliers use methodologies mainly based on FEA. To do so, the whole vehicle is modeled. The input data, which are forces injected by the sources to the structure, are the only things gotten from tests or from operational experience. The sources (engine, HVACs…) are fastened on a rigid test bench so that the blocked forces can be quantified. They are then injected into the model to get the trend of the vibratory response of the structure.

Consequently, the methodology that is used requires a lot of operational experience to know the average response of a source, an important amount of time to prepare the models and very high performances computers or servers (huge RAM capacities and number of processors). All those parameters make these methods extremely expensive and unaffordable for small-size suppliers.

The methodology proposed here after is based on measurements of the blocked forces of the source (in its final design) on a test bench, measurements of the inertances of the reception structure (in its final design) and of the test bench (not always perfectly rigid), measurements or FEM characterization of the dynamic stiffness of the mounts and, FEA of the vibratory responses of the interface between the mounts and the reception structure.

Thanks to this method, it is possible to analytically foresee the vibratory behavior of the receptor and, thus, iterate on the design and material of the interface and mounts to reduce the noise at passengers' ears (can be deduced by efforts/acoustic transfer measurements).

This methodology only requiring some low-priced tests and fast calculations on small models needing computers with reasonable hardware configuration, it is way less expensive than the current methods.

To check the robustness of the method, having resonance modes of all the components in the studied frequency range is necessary.

To be sure to have some, all the data were obtained thanks to calculation. Consequently, it is possible to iterate on the design of the components to get the most constraining vibratory responses.

Firstly, the theory on which relies the method is described.

Secondly, the model, CAD designed with Solidworks 2016, meshed with Hypermesh 2017 and computed with Optistruct solver and all the input data are detailed.

All the results that are necessary for the determination of the efforts injected to the receptor are presented.

The results of the direct calculation of the efforts injected to the receptor is then compared to the one gotten analytically (thanks to an Octave script).

Finally, conclusion and perspectives are presented. The realization of a test rig and measurements on a given system is proposed to check the sensitivity of the method to noise from tests.

Is also foreseen to realize a Python program that will allow to load test data, FEM analyses data and realize the analytic calculation in a faster way than with Octave. This will also give the advantage of being less restricted by the file format of the input data.

2 Theoretical approach

As explained in the XP 19-701 norm, the goal is to predict the efforts injected into a passive component and coming from an active one. Here, the difference is that the {active component + elastomeric mount} subsystem is not directly linked to the passive component. Indeed, an interface is placed between them so that the design of the passive and active components does not need to be changed to reach a good vibro-acoustic performance. Indeed, in many cases, changing the design and material of the interface and/or the mounts is less constraining.

The figure here below schematically represents the full system that needs to be characterized.

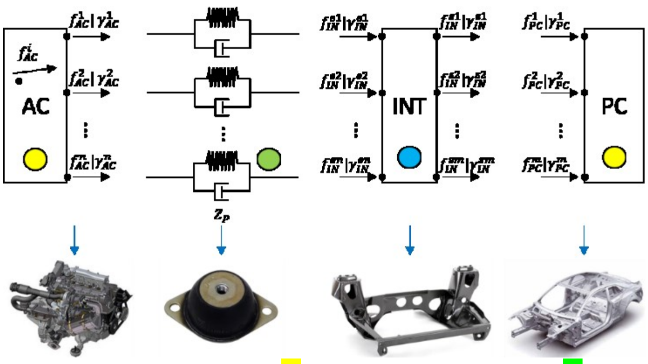

Figure 1 : System – O characterized by tests – O characterized by tests or by characterized and design iterated by calculations – O characterized and design iterated by calculations

The vibratory behavior of the active component is described as follows:

$$Y_{AC} f_{AC} + Y_{AC}^{i} f_{AC}^{i} = \gamma_{AC} \quad (1)$$

Where:

Y_{AC} = matrix of inertances and cross transfers acceleration over force between the points of fixation of the active component

f_{AC} = vector of forces at the points of fixation of the active component

Y_{AC}^{i} = matrix describing the transfers between the source point and the fixation points

f_{AC}^{i} = vector of internal forces

γ_{AC} = vector of accelerations at the fixation points of the active component

In the same way, the vibratory behavior of the passive component can be described as:

$$Y_{PC} f_{PC} = \gamma_{PC} \quad (2)$$

Where:

Y_{PC} = matrix of inertances of the m fixation points and cross transfers acceleration over force between the m fixation points of the passive component

f_{PC} = vector of forces at the points of fixation of the passive component

γ_{AC} = vector of accelerations at the fixation points of the passive component

The interface can also be described as:

$$\begin{bmatrix} Y_{IN}^{ee} & Y_{IN}^{es} \\ Y_{IN}^{se} & Y_{IN}^{ss} \end{bmatrix} \begin{Bmatrix} f_{IN}^{e} \\ f_{IN}^{s} \end{Bmatrix} = \begin{Bmatrix} \gamma_{IN}^{e} \\ \gamma_{IN}^{s} \end{Bmatrix} \quad (3)$$

Where:

Y_{IN}^{ee} = matrix of inertances of the entry points and cross transfers acceleration over force between the n entry points

Y_{IN}^{es} = matrix of cross transfers between the n entry points and the m exit points (acceleration at the entry / force at the exit)

Y_{IN}^{se} = matrix of cross transfers between the m exit points and the n entry points (acceleration at the exit / force at the entry)

Y_{IN}^{ss} = matrix of inertances of the exit points and cross transfers acceleration over force between the m exit points

f_{IN}^{e} = vector of forces at the entry points

f_{IN}^{s} = vector of forces at the exit points

γ_{IN}^{e} = vector of accelerations at the entry points

γ_{IN}^{s} = vector of accelerations at the exit points

At the spring/damper interface, the following equations can be set:

$$\begin{cases} -f_{AC} = Z_P (x_{AC} - x_{IN}^{e}) \\ f_{IN}^{e} = Z_P (x_{AC} - x_{IN}^{e}) \end{cases} \quad (4)$$

Where:

Z_P = matrix of impedances

x_{AC} = vector of displacements at the fixation points of the active component

x_{IN}^{e} = vector of displacements at the entry points of the interface

From (4) we can get the following:

$$\begin{cases} -f_{AC} = \frac{-Z_P}{\omega^2} (\gamma_{AC} - \gamma_{IN}^{e}) \\ f_{IN}^{e} = -\frac{Z_P}{\omega^2} (\gamma_{AC} - \gamma_{IN}^{e}) \end{cases} \quad (5)$$

Knowing that $\gamma_{INT}^{s} = \gamma_{PC1}$, $f_{PC1} = -f_{INT}^{s}$ and $f_{AC} = -f_{INT}^{e}$ and by combining (1), (2), (3) and (5), we can get :

$$Y_{AC}^{i} f_{AC}^{i} = \left[(Y_{IN}^{e} + Y_{AC} - \omega^2 H_P) Y_{IN}^{se\,-1} (Y_{PC} + Y_{IN}^{ss}) - Y_{INT}^{es} \right] f_{PC} \quad (6)$$

Where:

$H_p = Z_P^{-1}$

Not being able to characterize Y_{AC}^{i} and f_{AC}^{i}, it is necessary to use a second passive component. This will allow to get rid of these two parameters.

That is why a test bench should be used.

In this case, only an active component, the elastomeric mounts and the second passive component (bench) are present. Therefore, no interface is present.

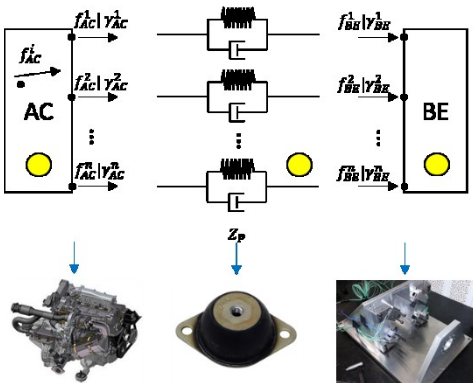

Figure 2 : Active component and mounts on the bench – O characterized by tests

In the XP 19-701 norm, the development is already done. The result is the following:

$$Y_{AC}^{i} f_{AC}^{i} = \left[Y_{AC} + Y_{BE} - \omega^2 H_P \right] f_{BE} \quad (7)$$

Combining (6) and (7) and considering the passive component is a vehicle:

$$\boxed{f_{PC} = \left[(Y_{AC} + Y_{IN}^{ee} - \omega^2 H_P) {Y_{IN}^{se}}^{-1} (Y_{PC} + Y_{IN}^{ss}) - Y_{IN}^{es} \right]^{-1} [Y_{AC} + Y_{BE} - \omega^2 H_P] f_{BE}} \quad (8)$$

3 Validation

To be able to foresee the forces injected to the reception component, the following measurements must be done:

- unit input frequency responses on bench
- forces measurements on bench
- unit input frequency responses on active component with free-free conditions
- unit input frequency responses on passive component

The following calculations also must be done:

- unit input frequency responses and cross-transfers on the interface

At first, to check the method and being able to easily change parameters, all the required data where obtained by FEA.

A model of a fan unit set on a reception component is done and meshed thanks to Hypermesh.

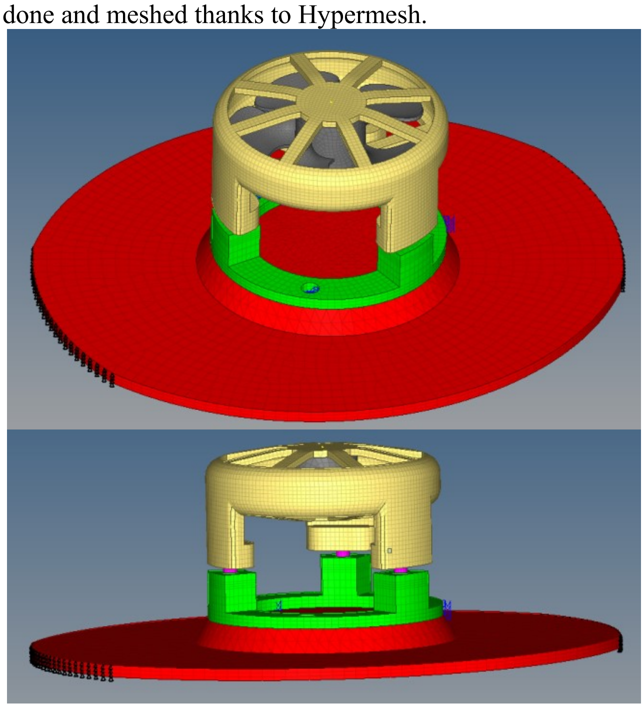

Figure 3 : Meshed fan unit on reception component with interface and elastomeric mounts

3.1 Inertances and cross-transfers of the bench

As seen before, it is necessary to measure forces injected by the active component on a bench. To do so, the equipment that is required is quite voluminous (force sensors with interfaces). Consequently, one must consider them integrated to the bench. Thus, the measurements of inertances and cross-transfers at the fixation points must be done with force sensors on this latter.

A model of the bench is done and meshed thanks to Hypermesh. The calculations are made thanks to Optistruct code.

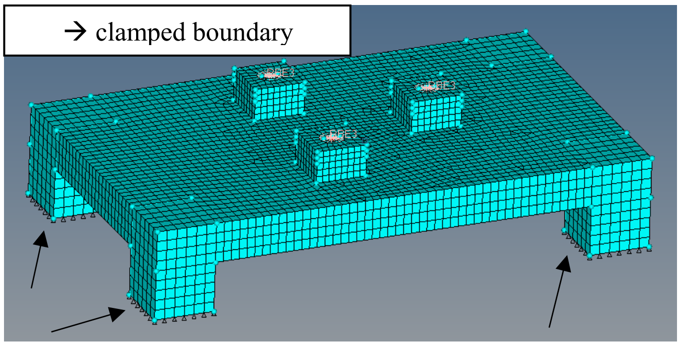

Figure 4 : Meshed bench

A clamped boundary condition is set at the bases of the bench and responses to unit input are computed.

An example of the inertances and cross-transfers on the bench are presented here after:

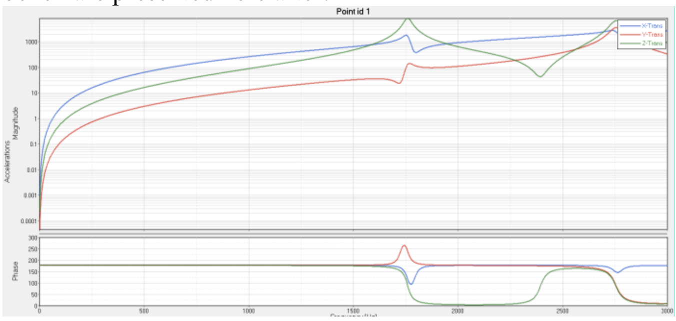

Figure 5 : Example of inertances and cross-transfers on the bench

In many tests cases, for example in blocked forces measurements, a bench that can be considered completely rigid on the frequency range of interest is required.

However, for some large components, designing a rigid bench can be too expensive. Here, the advantage of the method is that the modal effects of the bench can be compensated.

Consequently, one can design a bench, as rigid as possible, but that can present some resonances.

3.2 Forces injected to the bench

The forces injected by the active component with its mounts on the bench are measured in the way that was explained before.

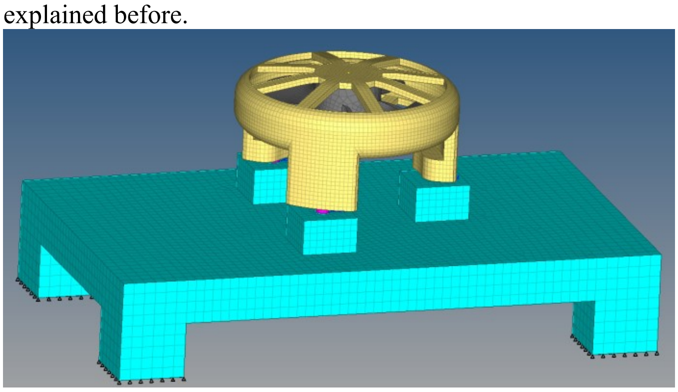

Figure 6 : Meshed fan unit with elastomeric mounts on the bench

Not knowing the real internal forces of a fan unit, they are arbitrary set.

Three isolated loads towards the X, Y and Z axes are set on top of the fan unit.

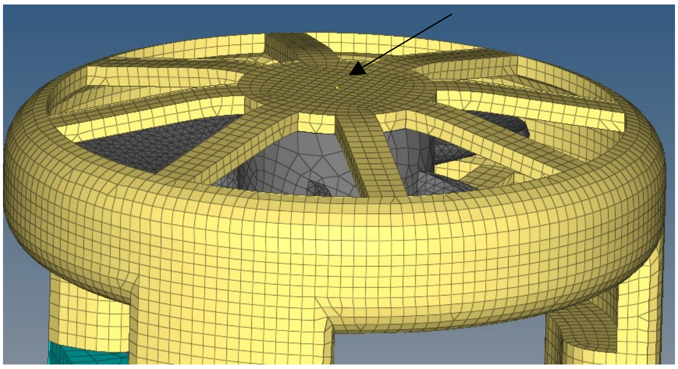

Figure 7 : Isolated forces on active component

The reacting forces at the fixation points are then calculated.

An example of the results is presented here below:

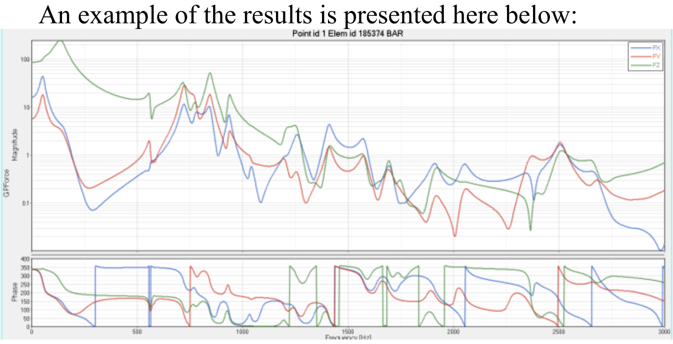

Figure 8 : Example of measured forces injected to the bench

3.3 Inertances and cross-transfers of the active component

As for the bench, the inertances and cross-transfers at the fixation points of the active component are computed.

For this calculation, a free-free condition is used.

An example of the results is presented here below:

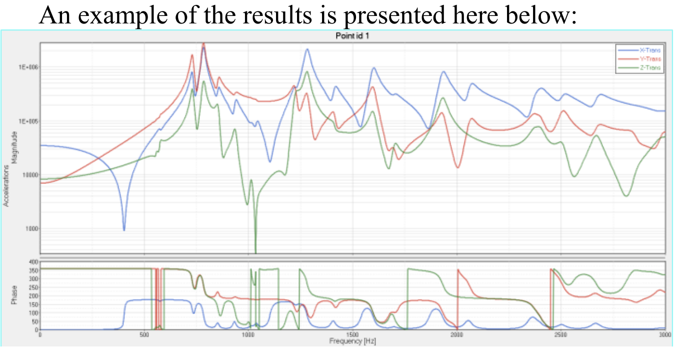

Figure 9 : Example of inertances and cross-transfers on the active component

3.4 Inertances and cross-transfers of the reception component

A reception component is meshed. A clamped boundary condition is set on two portions of the lateral face and, as for the bench and the active component, the inertances and cross-transfers at the fixation points are calculated.

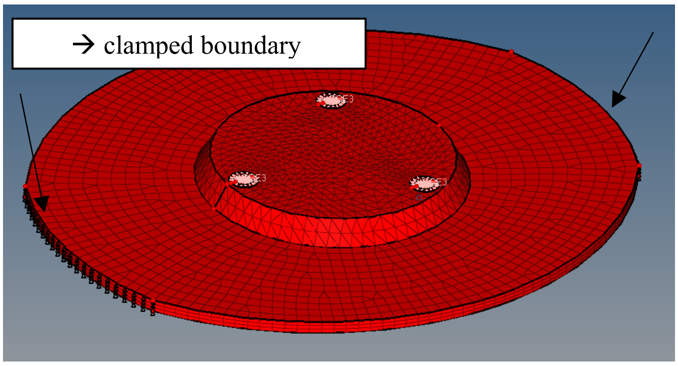

Figure 10 : Meshed reception component

An example of the results is shown here after:

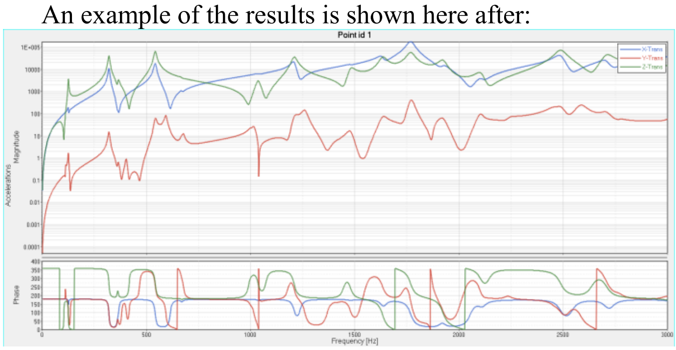

Figure 11 : Example of inertances and cross-transfers on the bench

3.5 Dynamic stiffnesses of elastomeric mounts

To obtain the H_p matrix, the dynamic stiffnesses of the mounts must be known.

A clamped boundary condition is set at the bottom of it.

A unit effort is injected on top of it and its displacement is measured.

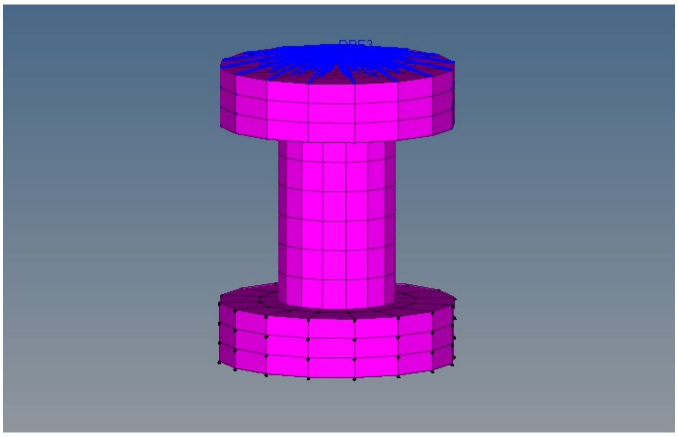

Figure 12 : Meshed fan elastomeric mounts

An example of the results is presented here after:

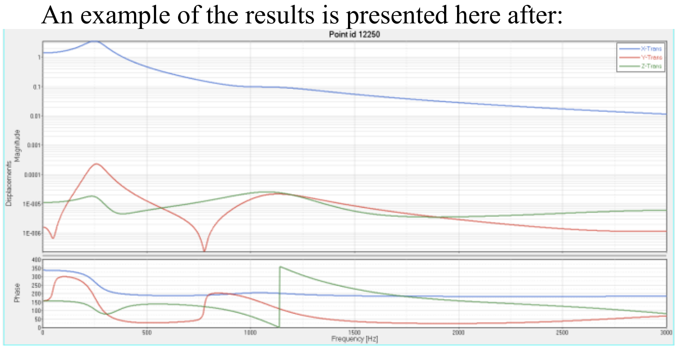

Figure 13 : Example of dynamic stiffnesses of the mounts

3.6 Inertances and cross-transfers of the interface

As explained previously, the aim of this method is to predict the forces injected by an existing active component to an existing passive one with an interface between them.

Thus, the material and geometry of this interface can be adjusted to reduce the vibrations of the reception component without using burdensome calculations of a complex full model.

The interface is modeled and the inertances and cross-transfers between active component fixation points and passive component fastening points are calculated.

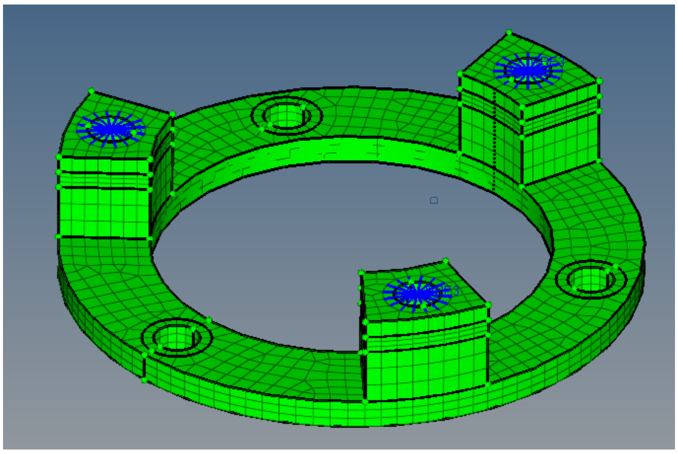

Figure 14 : Meshed interface

An example of the obtained results is shown here:

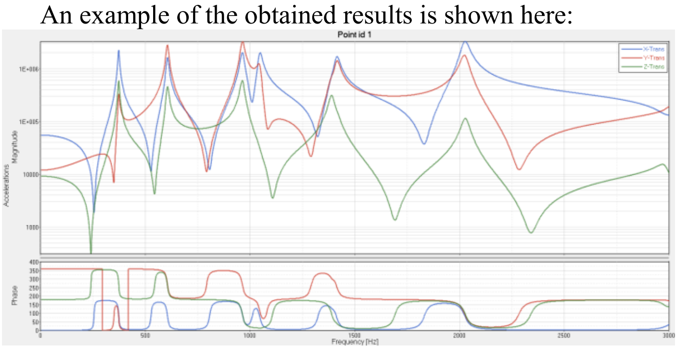

Figure 15 : Example of inertances and cross-transfers on the interface

3.7 Forces injected to the passive component

The forces injected by the active component to the passive one are also calculated to compare the results with those from the proposed method.

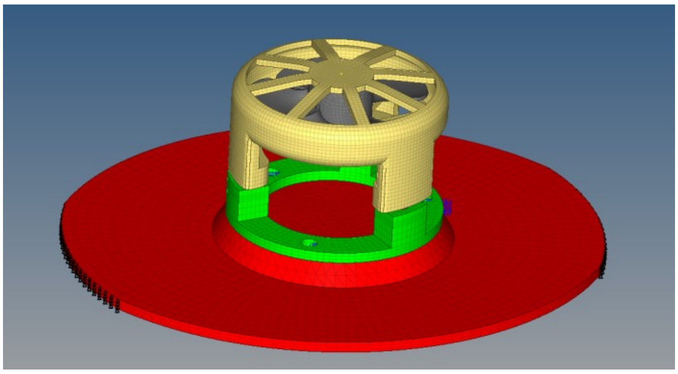

Figure 16 : Meshed whole system

The forces at the fixation points of the interface to the passive component are computed.

Here below is presented an example of the results:

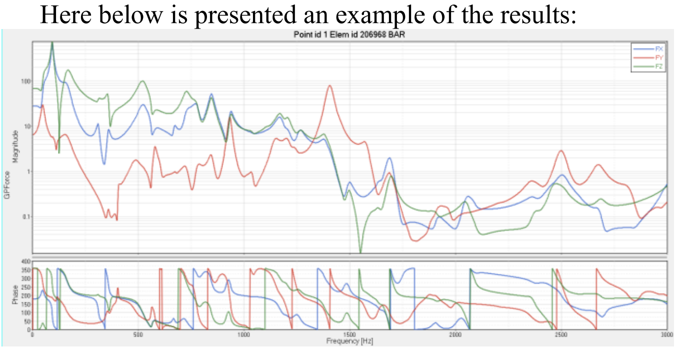

Figure 17 : Example of measured forces injected to the passive component

3.8 Comparison direct forces / recomposed forces

Thanks to an Octave routine, the forces are recomposed using the equation (8).

They are compared with the forces obtained from direct calculation.

The results are the followings:

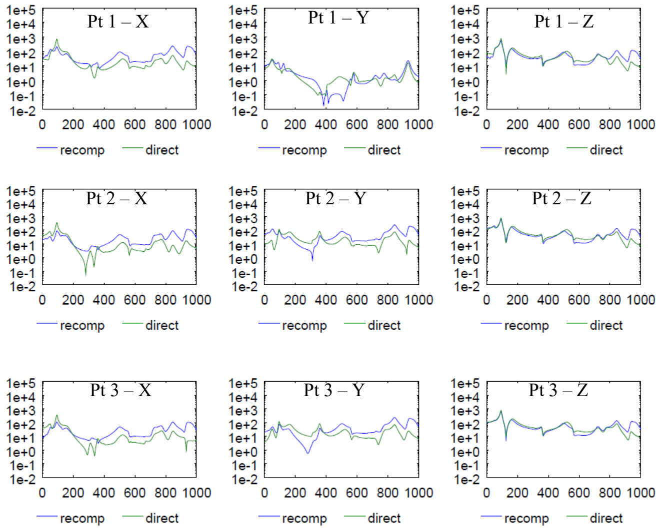

Figure 18 : Comparison between recomposed forces (---) and direct calculation forces (---)

One can notice that the results from the re-composition on the Z axis perfectly match the results from direct calculations except around 1000 Hz.

On the X and Y axes, some slight differences are observed. In this method, the moments are not considered. Indeed, it is difficult to measure them.

Other calculations taking them into account were performed and all the recomposed forces were perfectly matching the efforts from direct calculation.

Consequently, it was confirmed that theses slight differences were due to this approximation.

However, one can see that, despite this hypothesis, the estimation of forces with this method gives solid results.

4 Conclusion and perspectives

A methodology based on vibratory responses measurements of an active component, a passive receptor and elastomeric mounts in their final design and calculations on an interface was developed from the “XP 19-701” norm.

Its aim is to predict forces injected in the passive component and iterate on the design of the interface and the elastomeric mounts to reduce the efforts transmitted to the structure and limit the risks of acoustic emergences.

This combination of tests data and computation allows to get away from complex FEM model that require expensive computer configurations and huge amounts of calculation time. Thus, it allows to reduce the cost of vibro-acoustic performance studies.

A theoretical development was carried out and an Octave routine aiming to deduce the forces injected to the receptor thanks to the tests and calculations data was done.

To be able to iterate on the designs and easily act on various parameters, all the data that should normally be measured (blocked forces, inertances and cross-transfers on active component…) were obtained by calculations.

It also allowed to get the key parameters necessary to carry out the tests in an appropriate way. Indeed, it is important to respect some variables for this method works properly.

To check the consistency of the results of the methodology, the forces injected to the reception structure were gotten from direct calculations.

Thus, these results and those from the re-composition were compared.

It was noticed that, towards the Z direction, the forces vs frequency curves from the re-composition were perfectly matching those from the direct calculation before 1000 Hz.

On the X and Y direction, slight differences were noticed.

The methodology being partially based on tests, moments are not considered because of the complexity of measuring them. By doing another calculation integrating them into the re-composition, the forces vs frequency curves towards X and Y directions were perfectly matching the ones from direct calculations.

Thus, one could conclude that these differences come from this hypothesis. Despite this, the curves share the same trends and the results are very close.

To check the robustness of this methodology, particularly to noise and tests uncertainties (direction of impact of ping tests not perfectly vertical or horizontal, positioning of the accelerometers…), this method should be tested on real components. This task still has to be performed.

References

[1] Moulet, M.H., *XP 19-701*

Definitions/Abbreviations

FEM	Finite Element Method
FEA	Finite Element Analysis
CAD	Computer Aided Design
AC	Active Component
PC	Passive Component
INT	Interface