Characterization of Intercoolers Dynamic Behavior

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ABSTRACT

The work presented in this dissertation, was realized with the support of JDeus&Filhos, S.A., with the objective of studying the dynamic behavior of an intercooler produced by them, for the Alfa Romeo Giulietta. In order to achieve this objective, laboratorial tests were done to define the acceleration matrix from independent forces applied in the several supports and identify the modal parameters of the structure (modal frequencies and damping factors), which were later compared to numerical results from several finite element method analyses provided by JDeus®. The definition of the transmissibility matrix between the supports to a coordinate of interest was also done and confronted with data retrieved from road tests, provided by JDeus®. The experimental setup and the experimental procedure are described as well as the procedure leading to modal identification. The modal parameters obtained from the experimental procedures show that the intercooler studied has no risk of resonance due to vibrations imposed by the normal operation of a car (according to the criteria used by the OEM).

Keywords

Intercooler, Modal Analysis, Modal Identification, Transmissibility

1. INTRODUCTION

In order to better understand the dynamic behaviour of an intercooler, together with the opportunity to give a practical problem for the finalist students to apply their knowledge, JDeus® has proposed the following work. The referred enterprise has provided data from field tests and numerical analysis, which is compared to the results obtained in experimental works.

1.1. PROBLEM

Although the static study continues to be a priority when designing a structure, the dynamic study for the structure has been increasing its importance. Specially the modal analysis, to identify the structure resonances and to verify if the normal service is dangerous to the projected structure. With the evolution of technology more powerful machines are designed, made of lighter materials and less damped, which leads to dynamic solicitations and responses of great magnitude which can have drastic consequences to the structure.

In order to assure the structural integrity, it is necessary to understand the structure dynamic behaviour. This understanding is an important process in modern design. One powerful tool leading to the better understanding of the dynamic behaviour of a structure is the modal analysis.

Another powerful tool that is being developed is the transmissibility concept. This concept is useful, allowing to predict the response in a coordinate where the response isn’t measured either because the access is limited or there is no space for the sensor to measure the response.

1.2. RELATED WORK

In modal analysis, one can find several books and papers which can be useful to understand the problem in hands. Some authors describe the theoretical problem [1, 2, 3, 4, 5] while others explain the theory and the testing in the same work [6, 7, 8, 9, 10].

For cars in particular, Happian-Smith [11] defines the modal analysis process as a necessary tool for the modern design of these vehicles. Specifically, for intercoolers, one should know the materials and its properties used to fabricate these components in order to define the physical modal of the intercooler. ImechE [12] has done a dense characterization of charged air coolers, including differences between several types of air coolers as well as the properties of the materials usually used to produce them.

The transmissibility concept appears related to SDOF (Single Degree of Freedom) systems [2, 8, 13, 14, 15, 16, 17, 18]. The concept was derived for MDOF (Multiple Degree of Freedom) systems [21, 22]. There are two papers by Ribeiro, Maia and Silva where the concept is generalized [19, 20]. Fontul, Ribeiro, Maia and Silva [23, 24] show that the transmissibility concept is valid for harmonic and random solicitations.
When studying vibrations, one should be able to understand them as well as knowing how to ease them reducing the risks of a failure caused by a dynamic solicitation. In order to do that, one should study solutions to the vibration attenuation problem \cite{25, 27, 28, 29}. Fontul \cite{26} presents the concept of smart design in order to avoid the unwanted vibrations and suggest a few solutions for the problem.

In engineering, it is always important to have a good solution so one can at least compare the experimental results. With that in mind, the Finite Element Method present a good approximation of the reality. Although for complex structures it still has some limitations as Cook \cite{30} and Reddy \cite{31} have shown in their books. Noor has done a compilation of the bibliography existent in this subject in \cite{32}.

1.3 PROPOSAL

In this work a characterization of an intercooler’s dynamic behaviour is made, which starts with a modal identification recurring to experimental results. The concept of transmissibility is also applied, where the transmissibility matrix is defined for a coordinate of interest (where there are no forces applied), showing the importance of this tool as well as a practical example of its application.

After the characterization, the results are compared to the external results provided by JDeus® which consist in the modal analysis using finite elements method and the vibration data acquired during road tests.

In the end, a few conclusions are presented as well as several suggestions for future work.

2. APPROACH

2.1. INTERCOOLER

The intercooler is a necessary component in turbocharged vehicles. Its goal is to refrigerate the air flow that comes out of the turbocharger before it enters in the motor. By doing that, the density of the air increases allowing more air to enter the engine, leading to more power.

In service, the intercooler is subjected to forces in its supports. The sources of those forces are the engine and the movement of the vehicle. Generally, those forces have frequencies between 0 and 100 Hz due to the operating engine. Sometimes that can have great magnitude, for example when the car steps over an obstacle in the road.

As for the tests required by the car manufactures, they are usually fatigue tests, where the input is time or number of cycles. Some manufactures require modal detection tests between 5 and 200 Hz, were the component cannot have any mode below 60 Hz \cite{33}.

2.2 THEORETICAL APPROACH

Being the goal the dynamic characterization of the structure, one possible approach to achieve it is the definition of a mobility matrix of the structure. The mobility matrix allows to calculate the response of the structure knowing the excitation force. It takes the form of the equation (1):

\[
\ddot{\mathbf{x}} = [\mathbf{H}] \mathbf{f}
\]  

Where \(\ddot{\mathbf{x}}\) is the acceleration, \([\mathbf{H}]\) is the accelerance matrix and \(\mathbf{f}\) is the force applied to the system. The accelerance matrix, as well as the receptance and the mobility matrices allow the computation of the responses of a system due to applied forces. The receptance gives a force-displacement relation, the mobility gives a force-velocity relation and the accelerance gives a force-acceleration relation.

The accelerance matrix is symmetric, that property is important not only to reduce the number of tests but also to conclude that it is equal to excite in a coordinate \(i\) and measure in coordinate \(j\) or to excite in a coordinate \(j\) and measure in coordinate \(i\).

When computing the accelerance matrix in a frequency range, there comes a problem due to the modes that are off the limits of the range. These modes influence the modes within the frequency range. One way to consider the influence of the mode out of range is to introduce the concept of residual.

Consider the general FRF (Frequency Response Function):

\[
\alpha_{jk}(\omega) = \sum_{r=1}^{N} A_{jk}^r \frac{\omega_r^2}{\omega_r^2 - \omega^2 - i \eta_r \omega_r}
\]  

Where \(r\) is the mode, \(\omega_r^2\) is the angular frequency of the mode \(r\), \(\eta_r\) is the hysteretic damping factor of the mode \(r\), \(A_{jk}^r\) is a complex constant associated to the mode \(r\) and \(N \rightarrow \infty\) in real cases. Without losing generality, one can write:
\[\alpha_{jk}(\omega) = \sum_{r=1}^{m_1} \frac{1}{\omega_r^2 - \omega^2} + \sum_{r=m_1+1}^{m_2} \frac{\bar{A}_{jr}}{\omega_r^2 - \omega^2} + \sum_{r=m_2+1}^{N} \frac{\bar{A}_{jr}}{\omega_r^2 - \omega^2}\]

(3)

Where \(m_1\) and \(m_2\) are the lower and upper limit of the frequency range in study. The first term of the equation (3) acts like a mass contribution within the frequency range. As for the third term, it acts like a rigid contribution. With that in mind, one can approximate equation (3), introducing the so called residuals. Equation (3) then takes the form of equation (4)

\[\alpha_{jk} \cong - \frac{1}{\omega^2 M^R_{jk}} + \sum_{r=m_1+1}^{m_2} \frac{\bar{A}_{jr}}{\omega_r^2 - \omega^2} + \frac{1}{K^R_{jk}}\]

(4)

Where \(M^R_{jk}\) e \(K^R_{jk}\) are respectively the mass and the stiffness residuals.

Knowing that for different entries of the mobility matrix, the angular frequencies and damping factors related to mode \(r\) are equal (as they are global constants of the structure) and that the only variation is in magnitude due to \(r \bar{A}_{jk}\), Ribeiro [34] defined the CFR (Characteristic Frequency Response) as:

\[\beta(\omega^2) = \frac{1}{\omega_r^2 - \omega^2 + i \eta_r \omega_r^2}\]

(5)

Which depends only on the global constants of the structure. As the numerator is unitary, the inverse method can be applied, leading to:

\[\frac{1}{\beta(\omega^2)} = \omega_r^2 - \omega^2 + i \eta_r \omega_r^2\]

(6)

This definition is very helpful in terms of modal identification. The software used in this work, BETALab [35], uses the CFR for the modal identification.

As referred in the section 1.2, the transmissibility concept has already been derived and proven. Its main goal is to predict the response in a coordinate of interest, where the response is unknown, using the known responses in other coordinates. The equation (7) represents the mathematical concept of transmissibility:

\[\{x_U\} = [T_{AKU}]\{x_K\}\]

(7)

Where \(x_U\) are the unknown responses, \(x_K\) are the known responses and \([T_{AKU}]\) is the transmissibility matrix.

There are two ways of calculating the transmissibility matrix: one that is based on the experimental curves as shown in equation (8); and the other that relies on the FRF of the structure, as shown in equation (9).

\[[T_{AKU}] = \begin{bmatrix} \{x_U^{(1)}\} \{x_U^{(2)}\} \ldots \{x_U^{(K)}\} \end{bmatrix} \begin{bmatrix} \{x_K^{(1)}\} \{x_K^{(2)}\} \ldots \{x_K^{(K)}\} \end{bmatrix}^{-1}\]

(8)

\[[T_{AKU}] = [H_{UA}][H_{KA}]^{-1}\]

(9)

3. EXPERIMENTAL TESTS

The experimental tests were divided in three steps: first the measure of the responses of the structure due to a controlled excitation with a 0-500Hz frequency range(decided with JDEUS); second the identification of the curves obtained in the first step; and finally, the regeneration of those curves.

The structure was excited in four different points and in the three direction in each point. The responses were measure also in all four points for each position of the exciter, which means that there were 144 measurements (4x3x4x3). Twelve more measures were made in order to compute the transmissibility matrix.

Figure 1 shows the set of coordinates used, being 1, 2, 3 and 4 to compute the accelerance matrix (in which the three directions were considered) and 5 the unknown coordinate (only direction \(y\) of the global referential was considered) for the computation of the transmissibility matrix. In figure 1 there is also the global referential used in this work.
The laboratorial equipment used in order to realize these test was:

- Data Acquisition Equipment Brüel & Kjaer Type 3560-D;
- Vibration exciter Brüel & Kjaer Type 4809 (frequency range of 10 Hz - 20 kHz);
- Power amplifier Brüel & Kjaer Type 2712;
- 3 charge amplifiers Brüel & Kjaer Type 2635 (frequency range of 2 Hz - 10 kHz);
- Accelerometer Brüel & Kjaer Type 4326 A (frequency range of 1 Hz – 8 kHz);
- Force Transducer PCB PIEZOTRONICS 208C01 (frequency range of 0,1 Hz - 36 kHz);
- Computer containing software Brüel & Kjaer Pulse® vers.6.1.5.65;
- Cables and connectors;
- Metallic structures;
- Fishing line;

And the experimental setup was as shown in figure 2:

The following step was to identify the curves using BETAlab [34]. For doing so, the data from the experimental test were uploaded to the software, but before they needed to be modified in order to be compatible with the software and converted to the global referential.

All the 144 curves were plotted and there was a problem that the global variables weren’t totally coherent in all the curves. Knowing that the accelerance matrix is symmetric, the symmetric entries were compared and the one
that better represented the structure’s behavior was chosen. This process allowed to reduce the number of curves to identify from 144 to 78 (upper diagonal of the matrix).

After that, all the 78 curves referring to the accelerance matrix were uploaded to the software and the identification started. The goals of the identification are: computation of the global modal constants of the structure and the constants referring to each mode; computation the residuals for each curve; cleaning the noise existent in experimental curves;

Figure 3 shows the workbench of BETAlab [34]. In the graph one can see the regenerated curve in red and the experimental curve in white. The goal is to regenerate the curves as good as possible so the data treatment doesn’t affect the final results.

4. RESULTS

The modal identification resulted in four modes that are presented in table 1:

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
<th>Damping Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>224,7</td>
<td>0,0121</td>
</tr>
<tr>
<td>2</td>
<td>280,2</td>
<td>0,0092</td>
</tr>
<tr>
<td>3</td>
<td>375,4</td>
<td>0,0054</td>
</tr>
<tr>
<td>4</td>
<td>421,6</td>
<td>0,0060</td>
</tr>
</tbody>
</table>

After the regeneration of the curves it was possible to compute the accelerance matrix using the FRFs.

The transmissibility matrix was computed using both equation (8) and (9). For doing the computation using the FRFs it was needed to identify and regenerate the experimental curves of the unknown coordinate. A graphic comparison of the transmissibility matrix obtained by the two method mentioned above is shown in figure 4.
The matrix obtained by the second method is cleaner and it is not affected by the non-coincidence of the experimental curves in terms of the global modal constants. The differences between them are caused by the approximation of the curves in terms of the global constants and in the fact that local modes were not contemplated in the regeneration of the experimental curves.

By observing the graphic referring to the transmissibility matrix obtained by the FRFs of the structure, one can see several peaks. Those peaks do not necessarily mean that the response is going to be 6 or 7 times greater in the unknown coordinate. In fact, when analyzing the transmissibility matrix, one should consider the amplitude and the phase of all the contributions, not only amplitude.

5. EXTERNAL DATA COMPARISON

5.1. FINITE ELEMENT METHOD

As referred in section 1, JDeus® provided data to be confronted with the experimental results. The data provided was from two different categories. The first one to be confronted with the laboratorial results was the numerical results using the software ANSYS® vers.17.0 to determine the global modal constants. Three simulations were performed in order to achieve a consistency between the laboratorial experiments and the numerical calculations.

From this simulation the frequency of the modes obtained were: 309.17Hz and 362.5Hz. Although the values are quite distant from the observed in the laboratory experiments, the first mode has the same shape as the first mode verified in laboratory at 224.677 Hz. To visualize this mode in laboratory, it was used a stroboscopic light and a plastic pen. The second mode observed in the numerical simulation wasn’t the same as the second mode observed in the laboratory (for a frequency of 280.239 Hz). The differences in the frequency values might be explained by the rubber joint that exists in the intercooler and that was not modelled in this simulation due to its complexity. The introduction of the rubber joint would implicate a great number of elements and the analysis would not be possible in useful time with the available resources.
In the second simulation, it was requested to approximate the joint using a spring with similar characteristics to the joint dynamic behavior. It is common practice in structural analysis and proved to be a good approximation of the real behavior of a structure in those conditions. The simulation revealed to be too complex for the computation of any result. Once more, the structural model and common practice were not a good approximation of the reality when it comes to dynamic behavior.

For the third and last simulation, it was requested to simulate the laboratorial conditions. The nylon support was added to the structure as well as gravity. The reason for this request is because ANSYS® uses the model in free-free conditions when performing modal analysis. This new approximation requested a structural analysis to evaluate the effect of gravity in the coupled structure. After, it was performed the modal analysis and the results are shown in figure 6.

![Figure 6 - Modal shapes for the 3rd simulation](image)

Once again, the modes are the same in terms of shape. But this time, the natural frequencies are further away from the experimental. The first mode has a frequency of 317.64 Hz and the second mode has a frequency of 397.05 Hz. The reason why the frequencies are bigger in this case are the rigid constraints added to the first simulation. Once more, the structural model was not validated to perform modal analysis.

### 5.2. FIELD DATA

JDeus® provided data from two different road tests: one of normal cruising in the highway; and one in an alternative road. Before performing the road tests, an intercooler was instrumented with several sensors in order to measure all the variables of interest (strain, pressure, temperature and acceleration or force). The accelerometers (PCB PIEZOTRONICS MODEL HT356A15) were positioned the same way as in the laboratorial tests, so the results could be compared. The position of the five tri-axial accelerometers is presented in figure 7.

![Figure 7 - Position of the accelerometers in road tests](image)

That data was acquired in function of time, so a FFT (Fast Fourier Transform) was made using MATLAB® so the data could be compared with the laboratorial results (measured in function of the frequency).

Using the concept of transmissibility and equation (7) it was possible to compute the response of the unknown coordinate (in this case the z direction of the accelerometer 5) for both road tests. The response predicted by the transmissibility matrix was compared with the real response measured during the tests as shown in figure 8 and 9.
The differences between response predicted (blue curve) and response verified (orange curve) rely on the fact that the conditions of the intercooler in service are very different from the ones tested in the laboratory. For example, the temperature can go up to 170º and the internal pressure up to 3.2 bar (absolute). The properties of the materials of the intercooler are related to the temperature and pressure, so as the properties change, so does the structure in study. One other reason for the difference in the predicted and real responses is that the intercooler was instrumented with a lot more sensors for road tests, which has a great influence on the mass of the intercooler and also in its dynamic behavior. One last possible cause is the fact that the tubes that conduct the compressed air into the intercooler can excite the structure and that excitation was not considered in laboratory.

6. CONCLUSION

An experimental characterization of an intercooler dynamic behaviour was made. For this purpose, experimental tests were performed and the accelerance matrix and transmissibility matrix between the supports and a coordinate of interest were computed. Those results were compared with numerical and filed data provided by JDeus®. The main conclusions drawn throughout this work were: the great complexity of the structure was a problem when it comes to treatment of the results and some approximations had to be made; there are no resonances observed in frequencies below 200 Hz, as required by the manufactures; the existing finite element
model for structural analysis is not a good approximation for modal analysis; the instrumentation of the intercooler with sensors to measure other variables has changed the dynamic behavior of the intercooler; the variations in temperature and pressure while in service change the properties of the materials, changing the structure as well as its dynamic behaviour; it is probable that the tubes that conduct the compressed air into the intercooler excite the structure; The future works should involve a numerical model capable of reproduce the laboratorial results for ambient temperature, so it would be possible to simulate the results in service conditions. The road tests should be done with the dynamic characterization being then only goal, in order to minimize the alterations to the structure due to the addiction of the sensors.

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