CHALLENGES AND BEST PRACTICES DURING THE FINITE ELEMENT ANALYSIS FOR MODAL INVESTIGATION OF DRIVETRAIN COMPONENTS

Thales Sardinha Garcia Souza[1], Felipe Moura Fontes Novo[1],Dr.-Ing. Mauro Moraes de Souza[1], Juliano Savoy[1]

[1]Neumayer Tekfor Automotive Brasil Ltda.

E-mails: thales.souza@neumayer-tekfor.com, felipe.novo@neumayer-tekfor.com, mauro.souza@neumayer-tekfor.com, juliano.savoy@neumayer-tekfor.com

ABSTRACT

Engine downsizing is the use of a smaller engine in a vehicle that provides the power of a larger one. It is the result of car manufacturers attempting to provide more efficient vehicles by adding modern technologies, for instance, turbochargers, direct injection and variable camshaft. The smaller engine is also lighter and provides torque and power with similar performance to a much larger engine. However, the downsizing technique may lead to undesirable vibration effects on the driveline, such as structural damaging, vibration fatigue failure and extra noise. All these issues are related to natural frequencies investigation and they are often determined through the finite element method together with experimental tests during the product development phase.
This work presents the finite element method limitation for natural frequencies determination of automotive components and a possible solution for this issue.

INTRODUCTION

It is common knowledge that torsional compliance and stiffness variation are controlling factors of the vehicle noise. As improvements have been made to vehicle NVH characteristics, some concerns that have been masked by other NVH sources are now requiring attention [1]. Basically, there are three vibration sources that may result in damage or undesirable effects on the transmission and driveline: Ground excitation forces, transmission errors produced by geometric inaccuracies of the powertrain components, and excitation forces provided by the engine combustion. Figure 1 shows the speed variation of a modern six cylinders engine flywheel, which will generate the progressive excitation forces to the transmission [2]. In cases that the rotational speed irregularity at the clutch is unknown, neither by measurement, nor by calculation, a sinusoidal waveform may be used to prescribe the speed irregularity for an initial model.

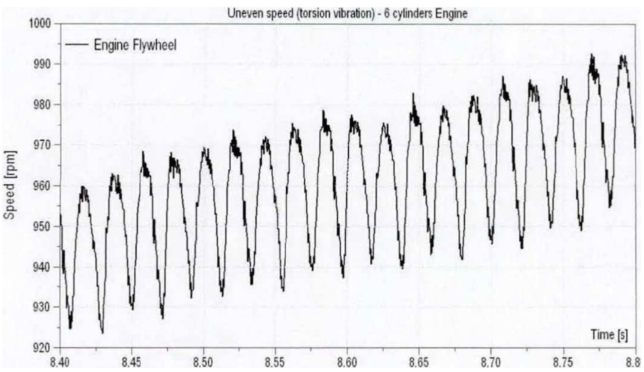

Figure 1 - Torsional vibration of a six cylinder engine [2]

The strong tendency of downsizing for internal combustion engines for passenger and commercial vehicles has been leading to higher amplitude values on the flywheel. Figure 2 presents the increasing torsional acceleration of the flywheel according to the engine evolution. By analyzing the vibration amplitude evolution from Euro 0 to Euro V, one can expect an increasing vibration amplitude for the next engine generation. Therefore, the correct understanding of the consequences of this undesirable effect is essential for the design success of transmission and driveline components.

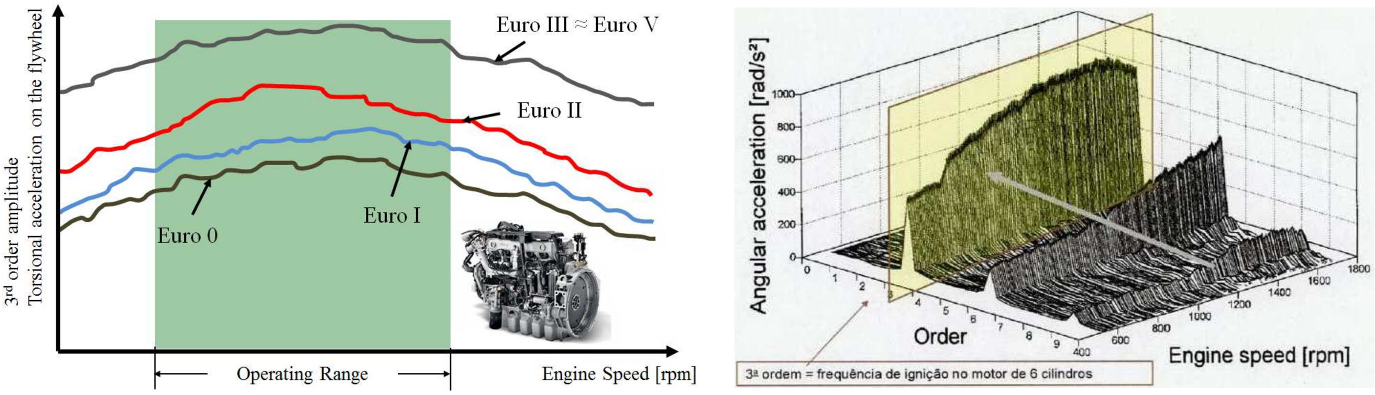

Figure 2 - Torsional vibration of a six cylinder engine [Adapted from 2]

This paper presents the numerical method for the investigation and reduction of vibration issues on two drivetrain components through virtual simulations and experimental practices. The comparison between the expected results with the measured ones was an important part of the development. The use of the finite element method was validated by comparing experimental and virtual results. Necessary changes were made in the model in order to correlate the predicted results with the measured data. The virtual settings at the FEM pre processor played an important role in the predicted natural frequencies response. Also, the filter setting during the validation procedure showed a great influence in the output test value.

CHALLENGES

During the experimental development of a new gear concept for automotive transmissions, a considerable difference was found between the natural frequencies response predicted by the finite element model and the measured data. Figure 3 presents the concept and summarizes the development of the Light Weight Assembled Gear. This gear combines the concept of mass relocation with an exclusive assembly process in order to bring several benefits [3]. By moving the gear body material far away from the neutral line, it was possible to achieve a final product with lower mass and higher stiffness on the axial direction when compared with the state of the art. The stiffness advantage results in a lower axial displacement of the gear teeth leading to a lower angular variation. In addition, the teeth contact area is positively affected. So, a homogeneous contact path and a better stress distribution during alternating loads can be expected, which may reduce or even eliminate the profile correction effort. In other words, the gearwheel performance can be raised and consequently the probability of failure is minimized. This gear was validated in a torque test bench and showed a good stress correlation between the finite element model and the real part. However, a considerable difference was found during the natural frequencies validation process. Although the F.E.

model predicted a natural frequency of 1400 Hz for the first vibration mode, the experimental test resulted in a natural frequency of 1100 Hz.

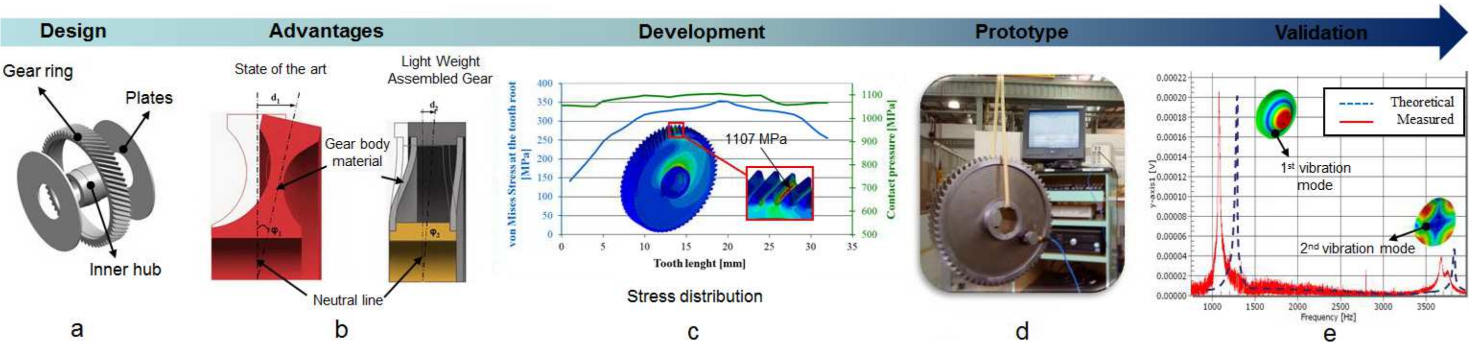

Figure 3 - a) Light Weight Assembled Gear concept; b) Stiffness advantage due to mass relocation; c) Stress distribution; d) Experimental procedure for natural frequencies investigation; c) Comparison between the output test result and the predicted FEM model [Adapted from 3]

At that time, it was not necessary to explain the difference between the output responses of the first vibration mode of the gear. The goal was only to approve the product for the next experimental procedures and both virtual model and prototype test qualified the component.

However, there are further developments of drivetrain components that present multi contact parts. The high complexity level of these projects combined with a short development time prevent, in some cases, the experimental practices. In order to guarantee the design success of those parts, it is necessary to create a method to explain the difference and to correlate the gear model presented in Figure 3. After that, it will be possible to use this method in other components design in order to reduce the development time without jeopardizing the reliability of the project.

The finite element method limitation is given by the contact settings during the modeling phase, which does not take into account parameters such as the coefficient of friction, interferences or gaps in the calculation of natural frequencies and vibration modes. Moreover, the commercial software packages as well as the analytical models cannot determine the vibration response of systems that present components which can get or lose contact according to the excitation force. Figure 4 shows a typical propeller shaft joint that presents multi contact parts that may get or lose contact according to its position.

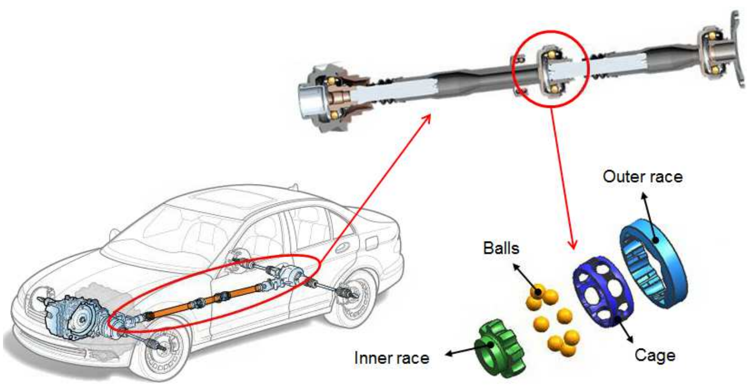

Figure 4 - Propeller shaft joint: According to the excitation force parts can get or lose contact

In order to overcome these situations, design engineers usually adopt simplifications of their systems. The next section shall describe the best practice to determine the natural frequency and vibration modes of transmission and driveline components during the virtual and experimental development.

BEST PRACTICES

In the development of finite element models of transmission and driveline components, the most difficult aspects for natural frequencies investigation are the assumptions regarding component connectivities. These connections usually are the main cause for the divergence between the virtual model prediction and the real behavior of the tested part. It is common knowledge that design engineers may face considerable issues during this stage of the development and the correlation between these two models is essential to the design success of any product.

Design engineers usually start their investigation refining the virtual model. In cases that manual changes are too time consuming due to the trial and error procedure, it is highly recommended proper correlation software packages.

The SDRC CORDS correlation program uses design sensitivity and optimization methods, specifying geometrical changes, in order to identify model updates that minimize the difference in the test and analysis frequencies [4]. After these changes, the engineer has the responsibility to decide whether they are within the manufacturing process tolerance of the part.

The multi body system software GTDYN is used for the calculation of excitation forces acting on a complete transmission unit. This software contains predefined elements for shafts, gears (including the gear meshes), roller and slider bearings [1]. The calculation considers torsional degrees of freedom as well as all degrees of freedom for bending and translation (radial and longitudinal) motion of shafts and gears. Thus, the influence of the bending deformation of shafts on actual backlashes in the tooth meshes is included in the simulation calculation.

There are cases where both software may be used to deal with natural frequencies investigation. However, a still valid process is the matching of the experimental test and virtual analysis. This procedure will be presented through Case 1 and Case 2.

Case 1 – Light Weight Assembled Gear

Correlation processes have been developed to quantify the correlation results through the use of Modal Assurance Criteria matrices, or modal orthogonality criteria. However, the most meaningful and valid process is the overlay of test and analysis frequency response functions, and minimizing the error between them [4]. This process was used during the correlation procedure of the Light Weight Assembled Gear through several finite element models attempting to correlate the virtual response with the experimental result shown in Figure 3(e).

- 1st F.E. Model: In this model all parts were modelled with coarse tetrahedral elements. The contact between plates and outer/inner parts was modelled through a multi point constrain tool. The total degrees of freedom amounts 572,225.

- 2nd F.E. Model: The gear ring and the inner hub were modelled with regular mesh size. A refined mesh with hexahedral elements was used on the plates. The contact between parts was modelled through a multi point constrain feature. In this model the total degrees of freedom amounts 1,367,316.

- 3rd F.E. Model: The gear ring was modelled with a regular mesh. The plates and the inner hub were designed with refined hexahedral elements. The contact between parts was modelled through node equivalency. The total degrees of freedom amounts 2,952,144. The better accuracy of this model does not depends exclusively of the refinement process itself, but to the higher amount of degrees of freedom on the direction of maximum amplitude defined by previous F.E. analysis.

None of the described models could correlate the predicted natural frequencies of the gear with the test result shown in Figure 3(e). So, geometric differences may be expected between the virtual model and the real manufactured part. The manufactured gear was measured with a 3D device, which showed slightly warped plates due to the assembly process. This difference modifies the gear stiffness in the axial direction and therefore leads to different natural frequencies results. The warped plates were incorporated in the virtual model and a fourth simulation was performed with similar setting as the third finite element model.

Figure 5 presents the comparison of all results predicted by the simulation and the measured acceleration during the experimental procedure. One can see an accurate comparison over the frequency range of interest between the fourth virtual model prediction and the validated part.

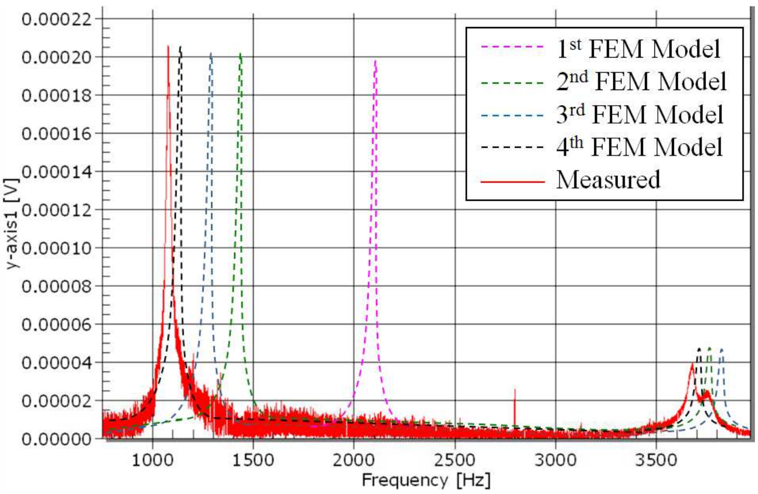

Figure 5 - Correlation of experimental and predicted results

It is important to understand the experimental specifications that made this correlation possible in order to avoid incoherent results in further developments.

Several parameters were considered during the validation process of the Light Weight Assembled Gear. The acquisition frequency was determined through the Nyquist theory, which states that the recording rate of any experimental test must be at least twice as high as the predicted frequency to be analyzed (it is recommended a recording rate 3 to 4 times higher than the expected frequencies) [5]. This approach ensured that the signal would not be distorted by the aliasing phenomena. Afterwards, an anti-aliasing filter was automatically used by the data acquisition equipment in order to guarantee that no signal out of the interested frequency range was acquired, which could lead to a misinterpretation of the experimental results. Figure 6 shows why does the aliasing phenomenon may invalidate a test result. Considering three input signals recorded at 5 Hz, the signal of 1 Hz is the only one to be properly described. The other signals will be misinterpreted as a frequency of 1 Hz because there are not enough data to describe them properly. This will prevent the engineer to evaluate the real frequency response of the system. Therefore, Nyquist recommendation of the proper recording rate must always be followed in order to avoid aliasing.

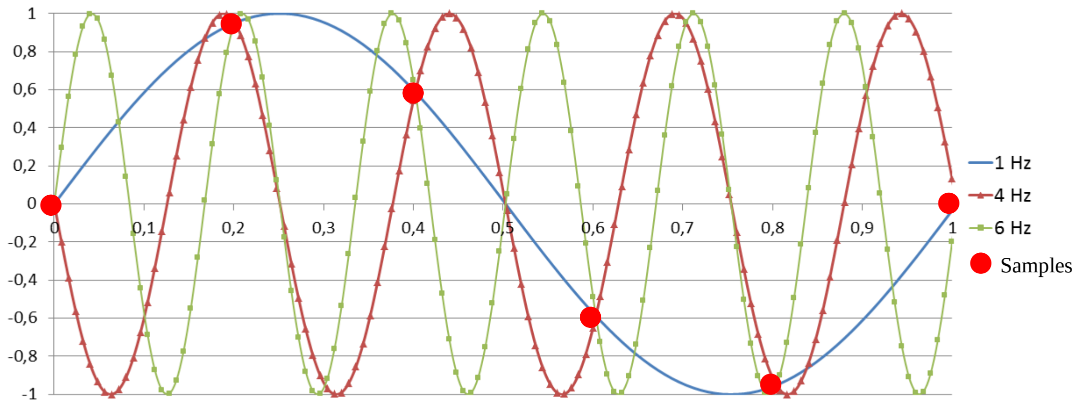

Figure 6 - Aliasing phenomenon issue [Adapted from 6]

The Fast Fourier Transform was also incorporated in the data acquisition software in order to convert a time domain signal into a frequency domain response. All these experimental efforts combined with the virtual model refinement led to the successful correlation practice shown in Figure 5.

Case 2 – Propeller shaft joint

As shown in Figure 4, the propeller shaft joint is a typical case that presents multi contact bodies that may get or lose contact according to its position. In this component, the inner race is mounted through a spline connection to the input shaft and the cage between the inner and outer races guides the balls, with a certain gap, in order to ensure the kinematic of the constant velocity joint. The balls are in friction contact with the outer race, which is connected to the output shaft through a press fit area. This type of connections raises numerous questions. Furthermore, it complicates the testing. This type of loose fit between components will cause rattling and poor quality of the frequency responses.

When it comes to the virtual analysis, the mathematical model requires contact simplifications in order to be analyzed by the finite element method. Due to the FEM limitation for calculating contacts in modal analysis, an alternative to model the CV joint is to replace the balls of the joint by a spring and mass system, which connects the inner and the outer races with the cage without using contact features such as interferences or coefficient of friction.

This simplification is illustrated in Figure 7. The point mass is represented by the circle and the springs, which connects the point mass to the other components, are represented by the dashed lines.

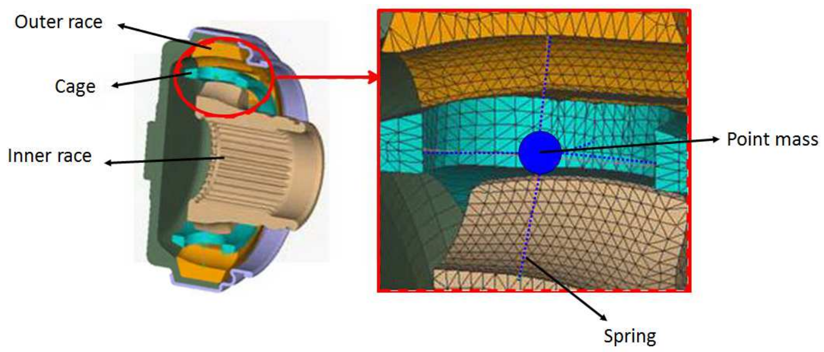

Figure 7 - Balls simplification: The joint is now constrained as in a spring mass system [Adapted from 7]

Not only the weight of the point mass must be the same of the original ball, but also it´s position has to match the ball’s center of mass. This simplification can guarantee the invariability of the component´s inertia.

The contact correlation process in this joint model can be done through adjustments on the spring stiffness used as a simplification of the balls. This approach aims to match the equivalent stiffness of the real contact. Basically, there are two ways to perform this correlation process. The first one automatically modifies the contact stiffness through a proper correlation software, which may suggest possible geometric changes, as described in the Best Practices section. The second option is to manually adjust the spring stiffness, which can be time consuming for complex systems.

A good technique to minimize the manual effort is to run a simplified model of the contact region before the complete F.E. model. Figure 8 shows the structural calculation steps in order to determine the ball mean stiffness. A pre displacement is applied on the upper plate while the lower plate is fully constrained. The mean stiffness is calculated according to the plate displacement and according to the reaction force caused by the ball´s compression. This result will be used as an initial value for the spring stiffness, which may be adjusted during the contact correlation process.

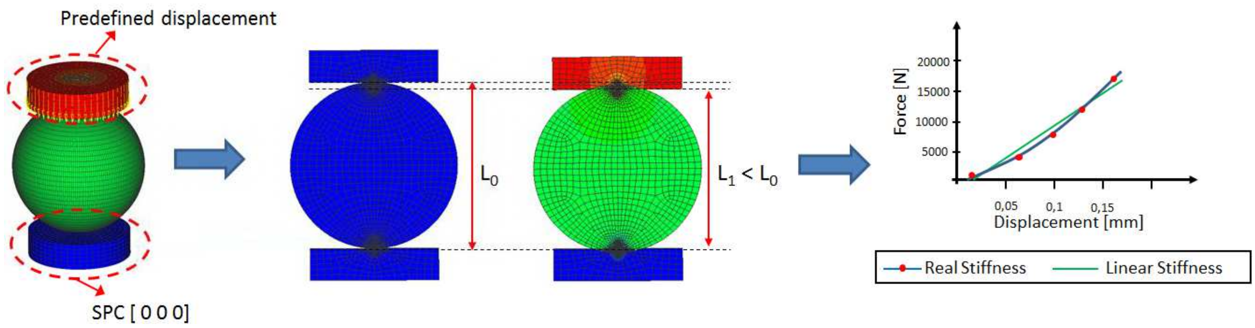

Figure 8 - Definition of the ball stiffness

Besides the design of the joint, the propeller shaft is a very important part to be considered in the driveline design. Due to its length and relative low stiffness, the first vibration modes may be within the working range of the vehicle. Therefore, these systems have to be carefully modeled in order to prevent redesign processes.

The same contact correlation procedure used for the CV joint can be used for the propeller shaft connections with splines and other joints. Figure 9 presents part of the driveline system of a all wheel drive vehicle that may presents these kind of loose fit connections.

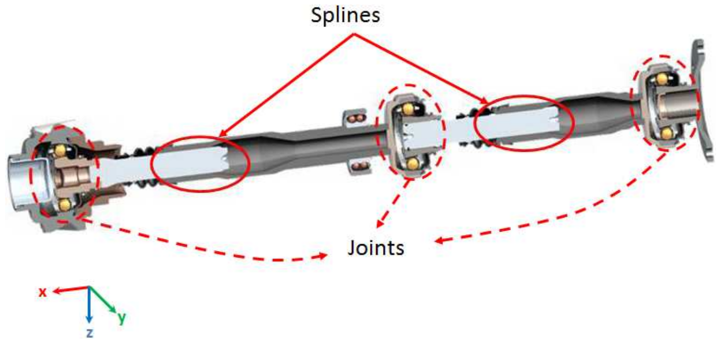

Figure 9 - Possible loose fit connections on the driveline

By using the methodology of contacts simplification, it was possible to calculate the natural frequencies and the vibration modes of the driveline. Figure 10 presents the first and the second vibration modes of the system. One can notice that the contact simplification did not prevent the natural frequencies investigation for the bending mode of each section of the shaft.

The finite element analysis resulted in natural frequencies out of the operating range of the shaft. The operating frequency considered safe is usually determined through empirical basis and it depends on the drivetrain architecture for each vehicle. On average, car manufactures suggest that the first bending frequency should be at least 1.4 times higher than the maximum propeller shaft rotation. Thus, considering a maximum angular speed of 6500 rpm, the system presented in Figure 10 is considered safe.

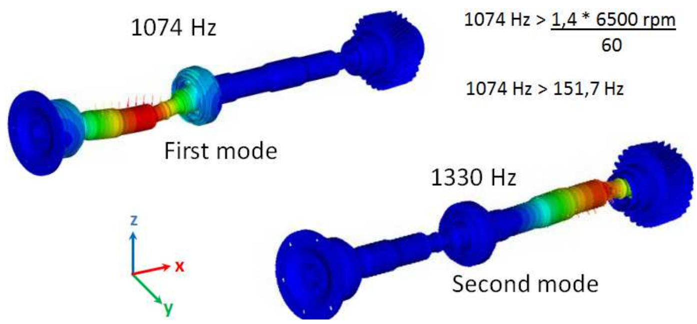

Figure 10 - Natural frequencies and vibration modes of the driveline system [Adapted from 7]

The validation procedure of such system can become quite complex to perform due to the connections between shaft and joints, which may cause unpredictable effects. So, during the validation process of the driveline system, there is the possibility of using dental cement in order to avoid rattling in the component connections [5]. Obviously, these cement bodies must be incorporated in the F.E. model through rigid elements. However, the real automotive system does present these gaps. Although the expected experimental result will not simulate all aspects of the driveline condition in a vehicle, this process can be adopted as a good practice to correlate the mathematical model in order to be a useful guide for further developments.

CONCLUSION

Engine downsizing will continue bringing vibration challenges for transmission and driveline systems. The correct understanding of the consequences of this effect and the proper numerical investigation process is essential for the design success of transmission and driveline components.

The correlation procedure depends on several parameters such as the system architecture, the component design, the correct use of the finite element method and the proper experimental practice. All these tools do not dismiss prepared engineers and their common sense.

It is often necessary to adopt simplifications in order to perform a virtual analysis in an attempt to describe the real phenomenon. To connect different parts with spring and point masses seems to be a good solution to overcome the finite element method limitation of not accounting interferences, gaps and coefficient of friction in the calculation of natural frequencies and vibration modes of multi contact systems.

The good accuracy of the validation process shown in Case 1 does not only depends on the higher amount of degrees of freedom of the virtual model, but also of the right experimental procedures.

The contact correlation process presented through Case 2 disregarded the coefficient of friction between balls and other parts of the CV joint. A damping effect in the validation process may be expected in systems where the coefficient of friction is meaningful for the natural frequencies response.

The process of overlaying the test with the virtual analysis is still a meaningful and valid procedure to correlate frequency responses. In specific cases, this effort can be reduced with proper correlation software packages. However, design studies are only valuable if the engineer can have the confidence that the model is properly representing the actual system.

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