Abstract

This paper describes the results of a study carried out in Renault about the effects of variation of flywheel bending stiffness on powertrain vibrations in the range of acceleration noise (250 Hz and 500 Hz octave bands). The flywheels used covered a wide range of stiffness, from a standard rigid flywheel to a very flexible one.

On the one hand, NVH tests have been carried out on a Diesel common rail injection engine with 4 flywheels of different bending stiffness, from a rigid flywheel to a flexible one. On the other hand, simulations have been carried out by Excite on a whole Diesel powertrain model. Both test and calculation results have shown that the vibrations at powertrain mounting points in the 250 Hz octave band decrease continuously with the reduction of flywheel bending stiffness, and there is a nearly linear relationship between the vibration amplitudes and the first bending frequency of the cranktrain. It has been found that the decrease of engine vibrations in the 250 Hz octave band is mainly due to the reduction of vibration amplification by the cranktrain bending mode. When reducing the flywheel stiffness, the first bending frequency of the cranktrain decreases, and creates less amplification of the main engine excitations in the 250 Hz octave band. The most important vibration reductions are observed in the odd and half engine orders (3, 3.5, 4.5 and 5), known to be relevant for rough noise. From this study, it is recommended to tune the first bending mode of the cranktrain as low as possible, which means to decrease the flywheel stiffness as much as possible, provided that there is no problem of mechanical resistance of the flywheel.

Introduction

The crankshaft bending vibrations are one of the main sources of engine acceleration noises (250 Hz and 500 Hz octave bands). They have also a significant influence on engine noise quality (roughness). The bending modes of a cranktrain (crankshaft + flywheel) depend largely on the bending stiffness of flywheel. By changing the flywheel stiffness, it is possible to shift the cranktrain bending modes in ranges favorable to engine acceleration noise and roughness reduction. The use of the flexible flywheel is an application of this principle.

In order to determine quantitatively the evolution of engine acceleration noise as a function of flywheel bending stiffness and to find the best values, 4 flywheels covering a wide range of stiffness have been prepared. These flywheels of different bending stiffness have been tested on a 4 cylinders Diesel common rail injection engine, with measurements of flywheel dynamic bending mode and of vibrations at powertrain mounting points.

On the other hand, NVH simulations have also been carried out, by using the software Excite, on a whole Diesel powertrain model, permitting to calculate not only the vibrations of the cranktrain (bending, torsion, pumping, etc), but also the vibrations of the whole powertrain at its mounting points.
Method of determination of cranktrain bending modes

In this study, the cranktrain bending modes were obtained by measuring directly the bending angle of the flywheel and then plotting a Campbell diagram (frequency – engine speed). An example is shown in Fig.1 for a crankshaft connected to a flexible flywheel. A cranktrain whose bending frequency is $f$, when it is rotating at speed $\omega$, the bending mode observed in the fixed coordinates system (engine structure) contains 2 frequencies: $(f-\omega/2\pi)$ and $(f+\omega/2\pi)$. When the engine runs from low speed to high speed, the cranktrain bending mode forms a V. At 0 engine speed, there is only one value, which corresponds to the peak of the V. The value of $f$ can change slightly with engine speed due to centrifugal and gyroscopic effects. For simplicity, the cranktrain bending frequencies presented in this article correspond to those at 0 engine speed.

![Fig.1 – Determination of dynamic bending mode of cranktrain](image)

It is to be noted that for a classical cranktrain, there are 2 kinds of bending modes: the mode in the symmetrical plan of the crankshaft and the mode in the perpendicular plan. In general, the mode in the symmetric plan is lower than that in the perpendicular plan. The difference can reach 20-30 Hz with a rigid flywheel. With a flexible flywheel, there is nearly no difference (only one V, as shown in Fig.1).

Measurement results on a 4 cylinders Diesel common rail injection engine

In this study, 4 flywheels covering a wide range of stiffness have been tested: one mono-bloc rigid flywheel, 3 flexible and semi-flexible flywheels. All the four flywheels had the same wheel inertia. For the flexible and semi-flexible flywheels, the wheel inertia was fitted on a disc. The values of the disc thickness were respectively 2.5mm, 4mm and 5.5mm, which gave different flywheel bending stiffness. For the rigid flywheel the equivalent disc thickness was 9.5mm. All the flywheels had been tested on the same engine, at the same running conditions, i.e.: engine speed sweeping at full load from 1000 rpm to 4500 rpm.
• Results of cranktrain bending modes

For each flywheel, the first bending frequency can be determined from the Campbell diagram of flywheel dynamic bending angle, as explained above. The results obtained for all the 4 flywheels are shown in Fig.2. The variation of the 1st bending frequency as a function of flywheel disc thickness is shown in Fig.3. It can be seen that the cranktrain 1st bending frequency increases almost linearly with the increase of the disc thickness until 5-6 mm. Beyond that, the cranktrain 1st bending frequency varies less and less with the flywheel disc thickness, because the cranktrain bending mode is controlled more and more by the stiffness of the crankshaft. Consequently, when the flywheel disc thickness is high (> 6 mm), the variation of flywheel stiffness (thickness) has little impact on the bending mode of the whole cranktrain. However, when the flywheel disc thickness is low enough (< 5 mm), the bending frequency of the whole cranktrain changes a lot with the flywheel thickness.

Fig.2 – Campbell diagram of flywheel bending angle in engine running conditions

Fig.3 – Cranktrain bending frequency as a function of flywheel disc thickness
• **Results of vibrations at powertrain mounting points**

Vibration measurements had been carried out at different powertrain mounting points. For each flywheel, the vibration values in 250 Hz and 500 Hz octave bands had been plotted as a function of engine speed (from 1000 to 4500 rpm). For simplicity, each curve is presented in this article by only one value which corresponds to that at 3000 rpm after a logarithmic regression of the curve. Results obtained in this way are presented in Fig.4 and Fig.5 respectively for vibrations in 250 Hz and 500 Hz octave bands.

Concerning the vibrations in 500 Hz octave band, the reduction of flywheel bending stiffness is rather beneficial as a whole, but the impact is very limited.

However, concerning the vibrations in 250 Hz octave band, it can be seen that they decrease significantly and progressively with the reduction of flywheel bending stiffness at all the powertrain mounting points and in all the directions. The most important vibration decreases are observed at gearbox side mounting (SMO1) which is close to the flywheel, and particularly in the engine vertical (z) and longitudinal (y) directions (reduction up to 10 dB).

![250 Hz octave band, values at 3000 rpm](image1)

![250 Hz octave band, values at 3000 rpm](image2)

*Fig.4 – Vibration values in 250 Hz octave band at powertrain mounting points (SMO1: gearbox side mounting, SMO2: engine side mounting)*
Fig.5 – Vibration values in 500 Hz octave band at powertrain mounting points (SMO1: gearbox side mounting, SMO2: engine side mounting)

- **Vibration reduction as a function of cranktrain bending frequency**

If we plot the values (in dB) of vibration reduction in 250 Hz octave band (reduction with respect to the value obtained with the rigid flywheel) as a function of cranktrain 1st bending frequency, it can be seen that there is a nearly linear relationship. This is particularly true for vibrations at gearbox side mounting point (SMO1), as shown in Fig.6. Obviously, the cranktrain bending mode has predominant impacts on the 250 Hz octave band vibrations at gearbox side mounting point.
Results of vibration simulation by Excite

In order to understand better how the flywheel stiffness affects the engine mounting vibrations, vibration simulations have also been carried out, by using the software Excite (AVL), on a whole Diesel powertrain model. The cranktrain in the model contained a crankshaft, a flywheel and a torsion damper pulley. The crankshaft was connected to the main bearings by non-linear joints.

The flywheel bending stiffness was changed by changing the flywheel disc thickness in the calculation model. 4 flywheel disc thickness had been chosen in the Excite model, which covered a large range of cranktrain 1st bending frequency (from 100 Hz to 220 Hz). For each flywheel, not only vibrations of the cranktrain (bending, torsion and pumping modes, vibrations in the main bearings, etc), but also the vibrations of the whole powertrain at its mounting points have been calculated at different engine running conditions, i.e.: engine speed sweeping at full load from 1000 rpm to 4500 rpm, with a step of 100 rpm.

- **Vibrations at powertrain mounting points**

As a whole, the vibration results in 250 Hz and 500 Hz octave bands obtained by simulation were quite similar to those obtained by measurement. The vibrations in 250 Hz octave band decreased significantly and progressively with the reduction of flywheel bending stiffness at all the powertrain mounting points and in all the directions, and the most important vibration decreases (up to 10 dB) were observed at gearbox side mounting point (SMO1). However, the flywheel bending stiffness had no significant impact for vibrations in 500 Hz octave band.

- **Vibration reduction as a function of cranktrain bending frequency**

For direct comparison with measurement results, the calculated values of vibration reduction in 250 Hz octave band (reduction with respect to the value obtained with the rigid flywheel) were plotted as a function of cranktrain 1st bending frequency. Fig.7 shows the results obtained at gearbox side mounting point. By comparison between Fig.6 and Fig.7, it can be seen that the results obtained by Excite simulation are almost the same than those obtained by measurement qualitatively and quantitatively. In both cases, we have the following conclusions:
- There is an almost linear relationship between the reduction of 250 Hz octave band vibrations (values in dB) and the decrease of cranktrain 1st bending frequency (in Hz).

- The most important vibration reductions are observed in engine vertical (z) and longitudinal directions (y) at gearbox side mounting point (SMO1).

- At SMO1, a decrease of 40 Hz in the cranktrain bending frequency permits to have a reduction of almost 3 dB for vibrations in 250 Hz octave band in z and y directions. In engine transverse direction (x), a decrease of 40 Hz in the cranktrain bending frequency permits to have a reduction of almost 2 dB for vibrations in 250 Hz octave band.

Fig.7 – Vibration reduction in 250 Hz octave band at gearbox side mounting point as a function of cranktrain bending frequency (calculation results)

- Analysis of cranktrain excitations in the main bearings

The above results have shown that the flywheel bending stiffness has predominant influence on the 250 Hz octave band vibrations at gearbox side mounting point. To understand how the flywheel impacts the vibrations at different powertrain mounting points, it is interesting to analyse the crankshaft excitation forces in the mains bearings. This had been carried out easily by Excite simulation.

By analyzing the excitation forces in all the main bearings, it had been found that the influence of flywheel bending stiffness was important mainly in the bearing No.1 (the closest to the flywheel). In the 4 other bearings, the influences of flywheel bending stiffness were much smaller. This explains why the flywheel bending stiffness affects much more the vibrations levels at gearbox side mounting point (SMO1), because SMO1 is much closer to the main bearing No.1 than the other powertrain mounting points.

In Fig.8 are shown the Campbell diagrams of cranktrain excitation forces in the main bearing No.1 in vertical direction, calculated respectively with 2 different flywheels: a flexible flywheel (100 Hz) and a semi-flexible or a semi-rigid flywheel (170 Hz). By comparing these diagrams, it can be found that:
in the 250 Hz octave frequency range (from 175 Hz to 350 Hz), the flexible flywheel gives much lower excitation forces in the bearing No.1 than the more rigid flywheel.

with the flexible flywheel, the most significant reductions of excitation forces are observed in the odd and half engine orders, i.e.: order 3, order 3.5, order 4.5, order 5 and order 5.5. In other words, the rigid flywheel amplifies mainly the odd and half engine orders. This phenomenon can be explained by the fact that the main bearing No.1 is particularly sensitive to the amplification, by flywheel bending mode, of excitation forces created by the combustion of cylinder No.1, which happens once every 2 crankshaft revolutions in a 4 cylinders engine.

**Fig.8 – Comparison of excitation forces in main bearing No.1 – vertical direction (calculation results)**

- **Analysis of vibrations at gearbox side mounting point (SMO1)**

In Fig.9 are shown the Campbell diagrams of vibrations at SMO1-vertical direction, calculated with the flexible flywheel and the semi-rigid flywheel. By comparing these diagrams, the following phenomena can be observed:

- in each Campbell diagram, the cranktrain 1st bending mode can be observed clearly in the form of a V, which is formed by the vibration peaks created by the crossing of the cranktrain bending mode with different engine orders.

- with the flexible flywheel of 100 Hz, the cranktrain bending mode V is located completely outside the 250 Hz octave band, so nearly no vibration amplification in this frequency range. With the semi-rigid flywheel of 170 Hz, half of cranktrain bending mode V is located inside the 250 Hz octave band. With the rigid flywheel of 220 Hz, most of the cranktrain bending mode V is located inside the 250 Hz octave band, so creating a lot of vibration amplifications in this frequency range. It is obvious that the decrease of engine vibrations in 250 Hz octave band is principally due to the reduction of vibration amplification by the cranktrain bending mode.

- the most important vibration reductions are observed in the odd and half engine orders (3, 3.5, 4.5, 5 ). This phenomenon is coherent with the results of excitation forces observed in the main bearings: the amplification by flywheel bending mode
affects particularly the combustion forces of cylinder No.1 (the closest to the flywheel), and much less the combustion forces of the other cylinders.

**Fig.9 – Comparison of vibrations at SMO1 – displacement in vertical direction (calculation results)**

**Conclusions**

- The reduction of flywheel bending stiffness decreases the vibrations in 250 Hz octave band at all the powertrain mounting points and in all the directions, without deterioration of vibrations in 500 Hz octave band. The most important vibration decreases are observed at gearbox side mounting point (SMO1) in engine vertical (z) and longitudinal directions (y).

- There is an almost linear relationship between the decrease of 250 Hz octave band vibrations (values in dB) and the reduction of cranktrain 1st bending frequency. Therefore, for obtaining the lowest vibrations in 250 Hz octave band, it is recommended to tune the cranktrain 1st bending frequency as low as possible, which means to decrease the flywheel stiffness as much as possible, provided that there is no problem of mechanical resistance of the flywheel.

- The decreases of vibrations at powertrain mounting points observed with flexible flywheel are mainly due to the reduction of vibration amplification by the cranktrain bending mode. This amplification increases the excitation forces in the main bearings, principally in the one the closest to the flywheel.

- The most important vibration reductions are observed in the odd and half engine orders (3, 3.5, 4.5, 5). It is known that engine rough noise is in large part due to these odd and half orders. Consequently, a flexible flywheel not only reduces the engine vibration level in 250 Hz octave band, but also improve engine rough noise.