Improving the NVH performance of a motorbike powertrain

A CAE investigation of a motorbike showing an unsatisfactory NVH behavior during gear shifts: identification of the root causes and definition of the most effective design modifications

Piaggio Group is the largest European manufacturer of two-wheeled motor vehicles and the world's leaders in its sector. The Group is also a major international player in the commercial vehicle market. The product range includes scooters, motorcycles and mopeds from 50 to 1,400 cc marketed under the Piaggio, Vespa, Aprilia, Moto Guzzi, Gilera, Derbi and Scarabeo brands. The Group also operates in the light transport sector with its Ape, Porter and Quargo (Ape Truck) ranges of three- and four-wheel commercial vehicles.

The quest

The study's motivation was the assessment of the NVH behavior of a motorbike engine driveline, during its working condition, in first gear. More precisely, its behavior was considered in transient conditions during relative rotations between the first and fifth gear of the driven shaft. The study was carried out to investigate the possible causes of an undesired noise observed during road and lab tests and to identify the most effective design changes to remove it.

The solution

Physical evidence

Figure 1 shows the first and fifth gear, in the engaged condition.
The undesired noise was detected when the teeth hit the slot, during quick clutch engagement / disengagement cycles.

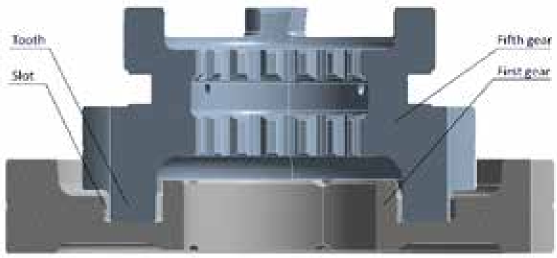

Figure 1 - Physical evidence, Engaged gears

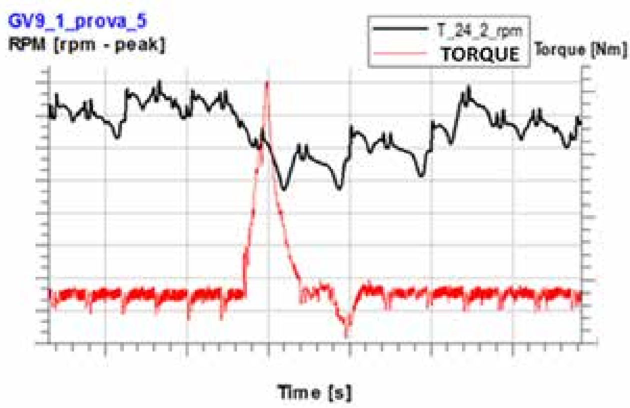

Figure 2 - Physical evidence, Torque transmitted by the driving shaft

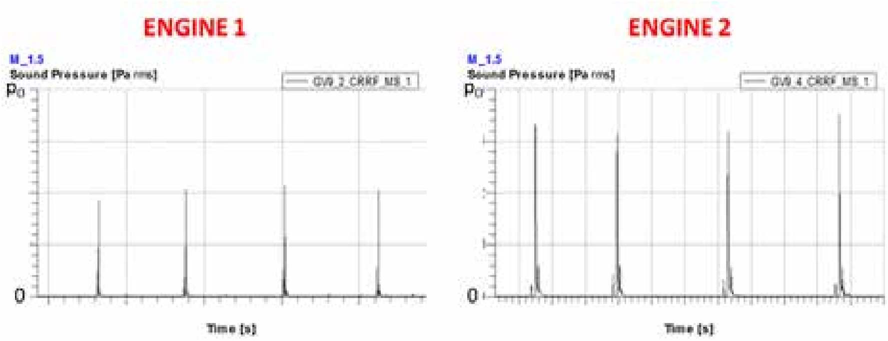

Figure 3 - Physical evidence, Comparison between two engines in terms of acoustic pressure

The phenomenon was measured in terms of acoustic pressure and of torque transmitted by a driving shaft included in the driveline. Figure 2 shows the torque time-history, which was used to validate the CAE model.

The engine under investigation ("Engine 2") was compared with another belonging to the same class, making the tooth repeatedly hit the slot. The acoustic pressure levels were measured, as shown in figure 3.

One of the main differences between the two engines is the driving shaft, which has different geometries in the two cases. Its effect was simulated by an MBS model of the powertrain.

While comparing the two engines, frequency analyses were carried out, which showed that the engine under investigation emitted noise mainly in the [300;5000] Hz.

This circumstance was taken into account to set up an FEM modal analysis of the bodies which form the engine casing; the analysis was carried out to find the cause of the worse acoustic performance of the engine under investigation, when compared with the other engine of the same class ("Engine 1").

Modelling remarks

The acoustic disturbance is the product of a sequence of causes and effects, roughly described as follows:

- the fifth gear teeth hit the slots, impulsively introducing mechanical energy into the system;
- the engine casing vibrates due to the action of the mechanical input; being the latter impulsive, all the eigenmodes can be excited; in fact, the acoustic emission interval is limited;
- the engine casing's surface velocity field, induced by the above mentioned phenomenon, is the acoustic source for the perceived disturbance, which is of course determined by the propagation through the air too.

This study covers the first two points only, by means of CAE techniques. The first point was tackled with a multibody model of the whole powertrain system, which was used to study the effects of design variants both at a component (driven fifth gear, driving shaft) and at a subsystem level (adoption of a cam coupler).

The second point was tackled with FEM modal analyses, whose results were coupled with the ISO equal loudness curves.

Multibody analysis

The model included the whole powertrain system and a dummy body, rotating around the rear wheel axis, which was used to take into account the inertial properties of the remaining part of the vehicle. The model is shown in figure 4.

On the clutch shaft, the cam coupler is visible. This subsystem was deactivated to simulate the engine's behavior in the measured configuration (see Figure 3, Engine 2); it was subsequently re-activated to measure its NVH effect.

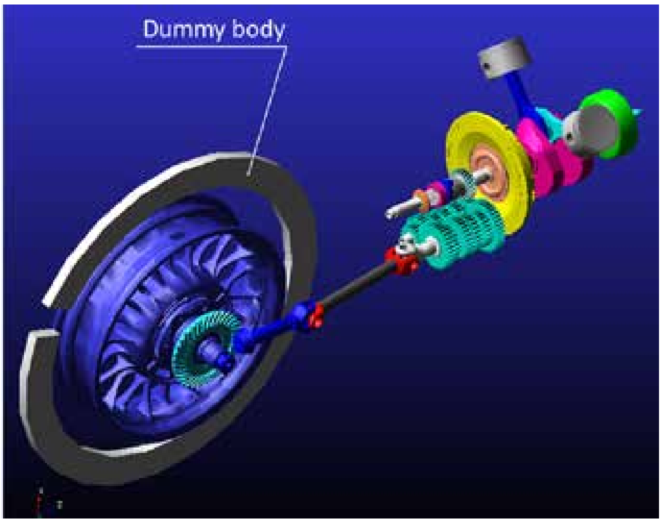

Figure 4 - Modelling remarks. Multibody analysis. The model

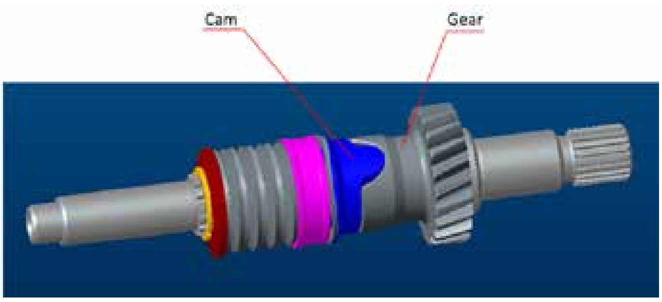

Figure 5 - Modelling remarks. Multibody analysis. Cam coupler. Interacting bodies

Clutch flexible coupler

The clutch includes a flexible coupler: a nonlinear flexible element was introduced into the model to simulate this effect.

Cam coupler

When the cam coupler (see figure 4) was activated, the interaction was modelled between the bodies shown in figure 5.

Rear wheel axle damper

A set of rubber dampers is used on the rear wheel axle.

Tyre

A tyre was introduced between the rear wheel and the dummy body (see figure 4).

Dummy body

The dummy body to simulate the remainder of the vehicle was given equivalent inertia and null mass.

Engagement between driven first and fifth gears

The description of the impacts between the two driven gears was detailed because these events trigger the cause-effect chain which determines the system's NVH behavior (see par. Modelling Remarks).
The multibody code computed the relative rotation between the two gears, starting from the configuration shown in figure 6.

Connection between the crankshaft and the clutch shaft

Figure 7 shows the crankshaft and the clutch shaft of the multibody model (see also figure 4).

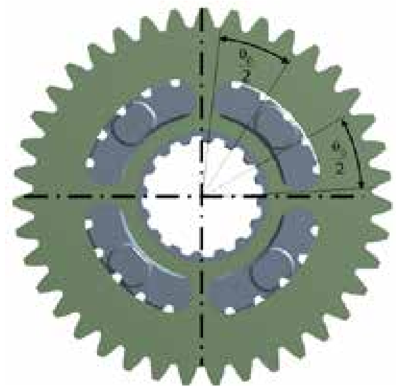

Figure 6: Modelling remarks. Multibody analysis. Engagement between driven first and fifth gears. Initial configuration

The modal data were combined with the equal-loudness curves, contained in the ISO 226:2003 standard.
The term "loudness" is used as an indicator of the perceived acoustic disturbance and its unit is the "phon". The curves in figure 8 show the points with equal loudness as a function of the frequency. For instance, the same loudness is felt listening to a 61 dB sound at 100 Hz and to a 40 dB sound at 1000 Hz.
The 40 phon curve was used to compare Engine #2 and Engine #1 (see par. Physical evidence); its ordinate was named δ and its abscissa ε.
The eignefrequencies have been classified as primary, secondary and negligible based on physical criteria that have been on purpose defined (for istance, a negligible eigenvalue has significant displacement components along neither the y nor the z direction in the standard vehicle reference frame).
The i-th primary eigenfrequency of each analyzed system was named p_i.
The i-th secondary eigenfrequency of each analyzed system was named s_i.
The following loudness indexes were defined:

$$\text{Primary index} \equiv \Gamma_p = \sum_i \frac{1}{\delta(p_i)}$$

$$\text{Secondary index} \equiv \Gamma_s = \sum_i \frac{1}{\delta(p_i)} + \sum_i \frac{1}{\delta(s_i)}$$

It's apparent how lower indexes are associated with better NVH performances in terms of acoustic source (see par. Modelling remarks).

Boundary and initial conditions
Multibody analysis

The system shown in figure 4 was excited applying the gas forces in the combustion chambers as a function of the crank angle.
Another design parameter taken into account was the shape of the driving shaft. The engine under investigation (Engine 2 in par. Physical evidence) has a driving shaft with a hollow cross section; during the experimental tests, it was compared with another engine (Engine 1 in par. Physical evidence), which has a driving shaft with a solid cross section. Thus, the effect of the adoption of a driving shaft with a solid cross section in Engine 2 was simulated. Two values of θ_0 (see figure 6) were taken into account, θ_{LOWER} and θ_{UPPER}: shifting from θ_{UPPER} to θ_{LOWER} had been proposed by the Design Department. Different sets of experimental tests have been considered: the engine speed was ω_1 during the first set and ω_2 during the second set.
The whole simulation set is shown in the table 1.
These simulations were aimed at evaluating the effect on the acoustic disturbance of the following design variants:
circumferential gap between the fifth gear teeth and the first gear slots (see figure 6): simulation #1 vs simulation #2 and simulations #3 vs simulation #4;
adoption of the cam coupler: simulation #3 vs simulation #5;
cross section of the driving shaft: simulation #4 vs simulation #6 and simulation #5 vs simulation #7.

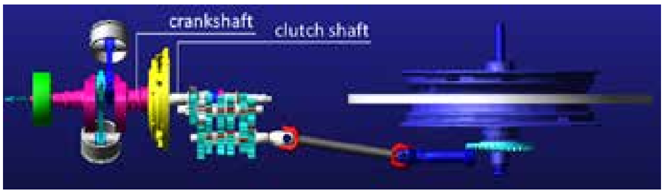

Figure 7 - Modelling remarks. Multibody analysis. Connection between the crankshaft and the clutch shaft. The crankshaft and the clutch shaft

A $\Delta(t)$ function was defined, which managed the behavior of the interface between the crankshaft and the clutch shaft:

$\Delta = 0$ - connected shafts
$\Delta = 1$ - disconnected shafts

Changing Δ between these two levels, it was possible to simulate nearly-impulsive clutch engagements, which were used during the physical tests to trigger the acoustic disturbance (see par. Physical evidence).

FEM analysis

As mentioned in par. Physical evidence, the FEM analysis' objective was to single out the eigenmodes of the engine under investigation ("Engine 2") in the frequency range which contains the significant part of the acoustic emission spectrum. The analysis was carried out to compare Engine 2 with another one ("Engine 1"), which showed a better acoustic performance.

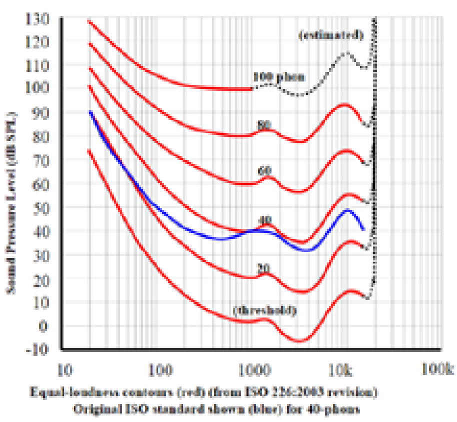

Figure 8 - Modelling remarks. FEM analysis. Equal-loudness curves

Simulation #	$\Theta 0$	Engine speed	Driving shaft's cross section	Cam coupler (see figure 5)
1	Θ_{UPPER}	ω_1	Hollow	Inactive
2	Θ_{LOWER}	ω_1	Hollow	Inactive
3	Θ_{UPPER}	ω_2	Hollow	Inactive
4	Θ_{LOWER}	ω_2	Hollow	Inactive
5	Θ_{UPPER}	ω_2	Hollow	Active
6	Θ_{UPPER}	ω_2	Solid	Inactive
7	ΘU_{PPER}	ω_2	Solid	Active

Table1 - Boundary and initial conditions. Multibody analysis. Simulated configurations

FEM analysis

The modal analyses of the engine casing were carried out in free conditions.

Results

Multibody analysis - Comparison with test data

The comparison was made to validate the model. It was made in terms of the torque transmitted by the driving shaft (see par. Physical evidence, figure 2) using the powertrain configuration #4 (see table 1), which was used during the experimental tests. In figure 9 the red curves show the torque time-histories.

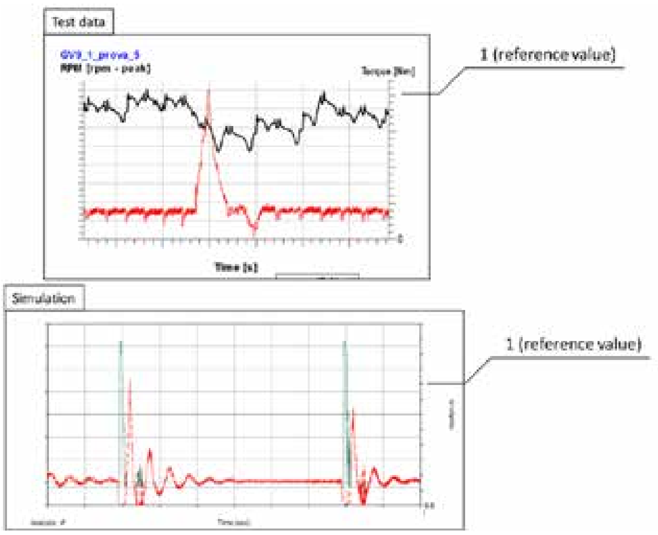

Figure 9 - Results. Multibody analysis. Comparison with the test data. Torque time-histories

Figure 9 shows that the maximum torque value in the test data is comparable with the one coming from the simulation model, which was therefore considered able to represent the physical phenomena the investigation had to deal with.

Comparison index

The simulated configurations (see table 1) were compared in terms of the RMS value η of the torque transmitted by the driving shaft.
As a result of the simulations, the most effective variants were the reduced tooth/slot gap and the simultaneous adoption of the solid cross section and of the cam coupler.
The better driveline performance with a drive shaft's solid cross section is in accordance with the test data shown in figure 3.

FEM analysis

Figure 10 shows the SPL values on the 40 phon curve at the primary and secondary eigenfrequencies for both engines which were taken into account (see par. FEM analysis of the Solution section); as mentioned, the engine under investigation is named "Engine 2" and it was compared with a similar one, named "Engine 1".

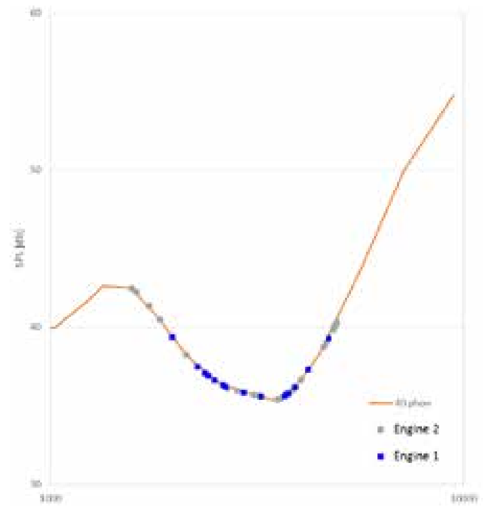

Figure 40 - Results. FEM analysis. Equal loudness curve

The overall loudness index is shown in the table 2 for each engine.

Index [1/dB]			
Secondary index		Primary index	
Engine 2	Engine 1	Engine 2	Engine 1
0.606	0.408	0.335	0.247

Table 2 - Results. FEM analysis. Overall loudness index

When the mechanical excitation (tooth/slot impacts) is the same, Engine 2 is therefore louder, as shown by the test data (see figure 3).

Conclusions

A motorbike engine had shown unsatisfactory NVH behaviour during gear shifts.
The impacts between gears was singled out as the cause.
A CAE campaign was carried out to assess the effectiveness of design variants devised to reduce the mechanical excitation due to the impacts between the gears and to analyse the behaviour of the engine casing, whose vibrations are the acoustic source.
The CAE results showed that following design variants are the most effective in terms of mechanical excitation for the powertrain system:

- reduction of the circumferential gap between the gears' teeth and the slots;
- adoption of different driving shaft's cross section and of a cam coupler.

The behaviour of the engine casing as an acoustic source was quantified by a loudness index related to the casing's eigenmodes. A comparison between different engines based on that index matched test data.

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