Riding experience enhancement through Engine and Driveline dynamics optimization

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Abstract

The aim of this work is to simulate the vehicle drive-line dynamic behavior in order to optimize drivers experience and comfort.

During the early phase of the work, engine and drive-line layouts have been defined and the inertia properties and gear ratios have been estimated. For each working speed condition (WOT and throttled conditions), the boundary conditions and loads have been applied and simplified models have been used to evaluate stiffness and damping of drive-line flexible joints.

Particular attention has been spent to the design of the cam, defining the cam profile in order to provide the driver different driving experiences, depending on torque demand.

Once established the basic dimensioning of the damper, the model virtual validation has been performed for the working conditions of interest, using Ricardo Valdyn software. The model has been used to:

- Frequency Response Analysis of the drive-line;
- Detail design of joints and dampers;
- Comfort Evaluation.

Experimental measures have been performed on prototypes with two different optimized configuration of the cam damper and a good correlation with the simulated results has been achieved.

In conclusion, the present work has been useful to develop a complex model which allows to calculate the resonance frequencies and the system response in terms of angular accelerations and torques transmitted by varying the working conditions.

Therefore it has been possible to modify the system response to minimize torsional vibration and improve driving experience from the earliest stages of design.

Introduction

Market drivers for top class motorbikes in Europe and North America has been remarkably influenced by increased welfare among target customer population in the latest decades; some trends, in particular, are influencing technical decisions about powertrain:

- Aging population of potential riders, being the motorbike an object of desire still in the fifties and the sixties;
- Time availability for medium range tourism;
- Reduced expenditure capabilities of young people in favor of middle-aged population.

This trends are defining a targetable and interesting sector of the market particularly sensible to, and ready to remunerate, the following product characteristics:

- Comfort in medium range journeys at low average speed;
- Sense of power only if and when desired to satisfy the desire to “feel young”;
- Absolute performance has limited importance relatively to the two other requirements.

Even though very simple to state, designing a comfortable powertrain ready to become "nervous" and "breathtaking" when pushing on gas may seem a tough engineering task. This article tries to suggest a way to achieve it with simple and reliable mechanical components, relying on a reliable simulation tool as Valdyn from Ricardo SW.

Engine architecture and target definition

From the introduction it is evident that there are, at least, two different riding conditions: one in which the comfort performances are determinant, in particular, where vibration in the frequency range under 30 Hz are considered detrimental for the rider perception and a second one in which the feeling of engine torque generated vibration is desired to experiment an enthusiastic riding experience. We will call the latter "Performance zone" and the former the "Comfort zone".

The first powertrain characteristic element defining engine related vibration is the engine architecture: this market sector is dominated by V2, V4 and B2 (Boxer 2 Cylinders) engines.

In the figures 1 to 6 the different harmonic composition, for gas and inertia loading, of these architectures are reported: as it can be easily seen many of the spectral contributors of a V4 are intrinsically null, corresponding to market accepted conclusion that motorbikes with V4 engines are a premium choice for riding comfort. On the opposite side of the same perception axis, V2 engines are typical of vehicles, whom the market attributes the palm of best emotional powerfulness to.

This drives us to the position of the technical objective for the development of an optimized driveline: *Comfort level equivalent to a V4 engine in the low to medium loading condition, the "comfort zone", while rough and nervous at high load, the "performance zone".*

Lumped parameters model

In order to analyze the dynamic behavior of the powertrain a very simple lumped parameter model has been defined, where main contributors have been modeled as concentrated masses, stiffnesses and dampers. Gear ratios were changed in relation to the different working conditions. A descriptive image of the model is reported in figure 7, while the corresponding Valdyn model is reported in figure 8. A list of the importance of the different contributors in terms of inertias transferred on the engine crankshaft is given in figure 9.

Variable stiffness damper

In order to have a variable dynamic behavior with torque demand, the best and most logical solution is to have an element whose stiffness is varying with average engine torque, in the end with the performance expected by the driver.

This is achieved with a cam damper mounted on the primary shaft, in which the rotational motion is transferred through the tangential component of the force with which a round tappet is pushed against a cam profile by a preloaded spring.

The design of such damper is shown in figure 10: a round ended tappet (1) is pushed against the cam (2) by the spring (3); one end is linked to the torque source, the engine, while the other end is constrained to the torque user, the entry shaft the clutch. The torque transferred trough the joint depends on the contact angle between the cam and the tappet, being null for null angle, at the deepest point of the cam profile, and increasing, theoretically to infinite, until a 90° angle.

The design limit of such joint is clearly the contact stresses, for which the classical hertzian theory has been considered valid, because of the very low mutual velocities between the cam and the tappet.

The definition of the cam profile in order to obtain the desired dynamic behavior is the main engineering task to achieve the desired objective. In particular , being the working position of the tappet a function of the average torque transmitted through the shaft the cam must be designed in order to have a very sharply increasing stiffness with applied torque and hence an intrinsically strongly non-linear behavior.

Modal analysis

Modal analysis with a preliminary damper was performed in order to understand resonance frequencies and modal shapes, results in term of schematic modal shapes representation and frequencies are reported in figure 11, while exciting engine harmonics, in the form of a Campbell diagram are given in figure 12, the expected vehicle excitation is found by matching the harmonic content of the engine with the excitable modes of the driveline.

Confort area frequencies

Provided that the performance zone is easily defined as the proximity to the full load curve of the engine, it is important now to define the area, which will be considered as belonging to the "Comfort zone".

In order to identify vehicle driving conditions in which comfort is considered of prominent importance discussion was promoted with test drivers and market analysts: different conditions were defined, as reported in figure 13: different vehicle speed and corresponding different inserted gears. Moreover also a slow sweep from 65 km/h to 120 km/h in 3rd gear was also consider to prevent the method not to highlight resonances in the "Comfort zone" not to be caught from a steady state analysis.

Modal shapes and resonance frequencies

Notwithstanding the non-linearities inherently present in the model a modal analysis was made possible, in different working conditions, by the use of the "perturbation" analysis, available in Valdyn, by which a local perturbation is superimposed to the running simulation to define local linear approximation of stiffnesses and hence define a locally linear model on which modal analysis can be performed.

Resulting mode shapes and modal frequencies of the first 4 modes are reported in figure 11: already the 4th one, above 200 Hz and involving the alternator drive is not interesting for the driveline dynamics; it is therefore sufficient to focus on the first three modes. For the understanding of results it must be specified that vehicle and driver inertias are accounted for in the wheel's one.

The following observations can be extracted from modal shapes analysis:

1. First mode: all the driveline is oscillating around the final shaft elasticity, which is endowed with a rubber coaxial damper; as the reaction on the ground and hence the pushing force on the vehicle are acting as a vibration node, this mode is of utmost importance for comfort;
2. Second and third modes are characterized by momentum oscillation between different portions of the powertrain: gearbox vs. engine for the second mode, around 35 Hz, and gearbox vs. wheel for the third one, around 50 Hz. Also in the latest case the final drive is acting as a node but a much lower importance can be attributed to the third mode because of its frequency, sufficiently high not to impact on riding comfort.

Planned interventions

In order to avoid dangerous excitation by high content harmonics in the comfort zone, it was decide to reduce as much as possible the frequency of the first and the second mode without significantly impacting the third one.

The planned intervention was therefore to introduce a very steep change in the stiffness of the cam damper between low and high load condition and a significant reduction of the stiffness of the transmission shaft. Both requirements were particularly demanding for the design point of view: the required cam damper stiffness change was higher than one order of magnitude between the reference torque level and the full load one, while the stiffness of transmission shaft was to be reduced by so much as 80% . The former requirement was faced by a new procedure in the cam profile definition, as described in the following paragraph, while the latter one was achieved by the supplier of the part with a significant re-design of the part.

Results are reported in figure 14 and 15: first mode frequency was reduced to 2 Hz while the second one was brought under the 20 Hz target, avoiding most of low order harmonics resonance.

Criteria for stiffness curve definition and cam profile definition

As said in the previous paragraphs, stiffness of the damper should be increased with engine torque and the extent of the necessary increase, as deducted from the modal analysis, required a very steep variation to be achieved geometrically in a very narrow zone of the cam.

In order to obtain the required change without falling into unfeasible cam geometry, it was decided to drive the cam design from the law of variation of local curvature radii with rotational angle and hence constraining the curvature profile not to fall underneath the minimum feasible negative radius.

This was achieved through a numerical integration of the profile using Runge-Kutta 4th order forward integration method.

A Valdyn model of the damper was than constructed in order to simulate the dynamic behavior of the part, the modelling strategy allowed to use it both as a separate model to investigate relevant parameters and to include it into the complete driveline model for a synthesis simulation, even though the latter was expected to be very demanding in terms of computational time.

The model was endowed with two LAMINA element representing the cam contact and the sliding degree of freedom of the tappet. Two RACK element were used to bring rotational degrees of freedom from rotational to linear and viceversa. A picture of the model structure and the resulting lamina images are reported in figure 16 and 17.

Results

Synthetic report of the achieved result is shown in figures 18 to 20 for a speed sweep from 65 km/h to 120 km/h , separating the effects of the cam profile modification and the introduction of a reduced stiffness transmission shaft.

The vibration in the comfort zone, shown in gray was reduced by almost one order of magnitude; even though this result was expected to be strongly influenced by the damping coefficients used in the analysis and would need an analytic set-up to be achieved through dedicated measurement campaign, the extent of the improvement was such that significant improvement in driving experience was anyway to be expected.

Experimental results

Extensive measurements on roller test benches qualitatively confirmed the improvements awaited. Vibration levels measured in the comfort zone, compared with previous models, showed significant reductions. Significant improvements were also obtained in the report of test drivers about vehicle drivability and riding experience. Results were so welcome that an analytical comparison test between original driveline and optimized one, though initially planned, was cancelled, considering that test vehicle, in opposite to previous versions, were judged fully satisfactory by test drivers.

Conclusions

The influence of engine dynamics, as defined by its architecture, on riding experience has been discussed and an indication is given on how to comply with comfort requirements without penalties in aggressiveness in performance area.

A lumped parameter model of the driveline has been used to understand dynamic behavior of the driveline under different loading conditions using Valdyn software.

Valdyn has also allowed for the definition of a dynamic model of the damper, considering nonlinear effects like friction and clearances.

A criteria for the definition of the profile of the cam has been established and a mathematical process to achieve it has been proposed.

Finally results were validated by extensive road testing, which confirmed the riding experience as originally desired.

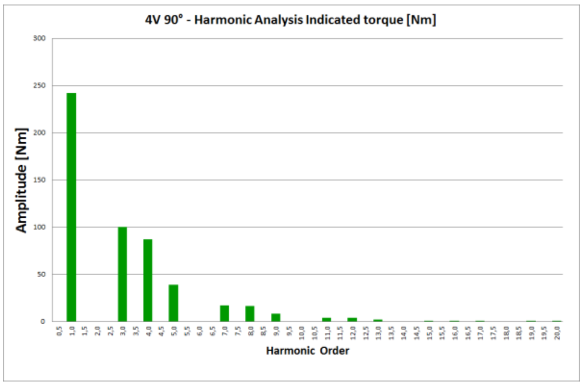

Figure 1 - Harmonic analysis of the indicated torque of a 4V 90° engine

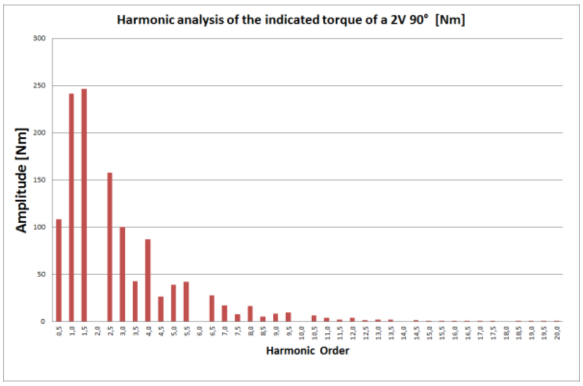

Figure 2 - Harmonic analysis of the indicated torque of a 2V 90° engine

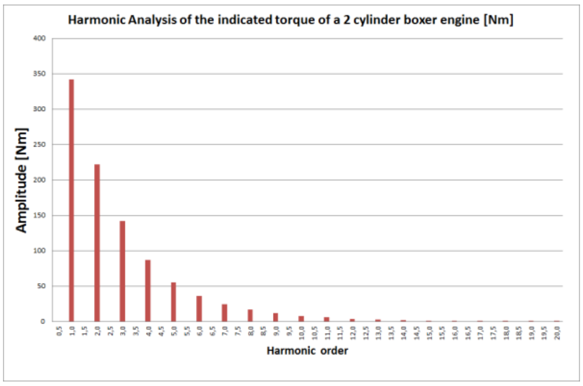

Figure 3 - Harmonic analysis of the indicated torque of a B2 180° crank boxer engine

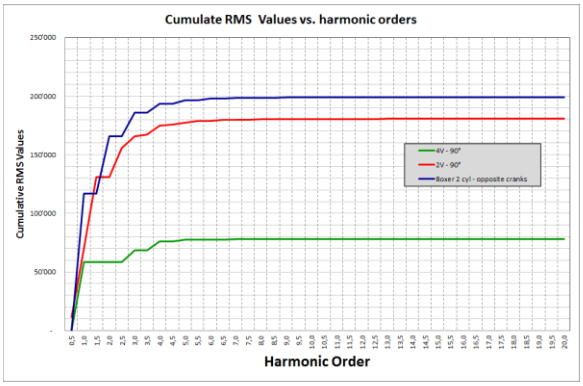

Figure 4 - Cumulate RMS values vs. harmonic order

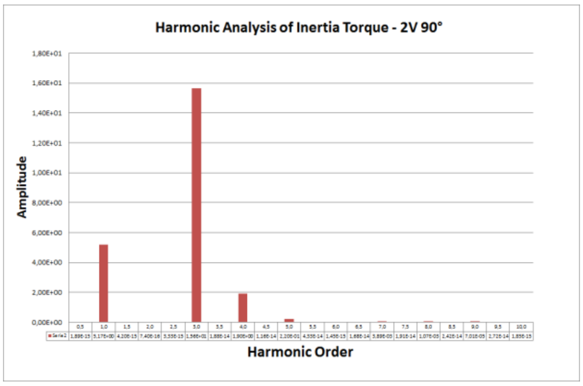

Figure 5 - Harmonic Analysis of Inertia Torque - 2V/4V 90°

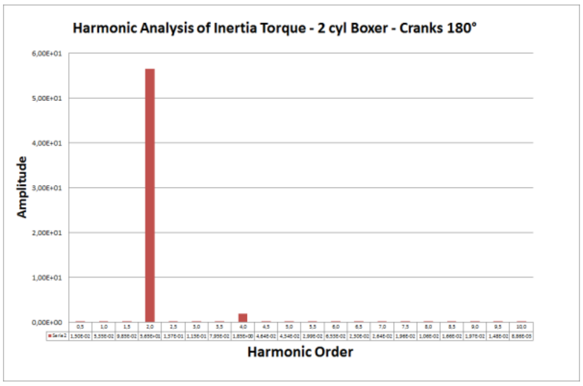

Figure 6 - Harmonic Analysis of Inertia Torque - 2 cyl Boxer 180°

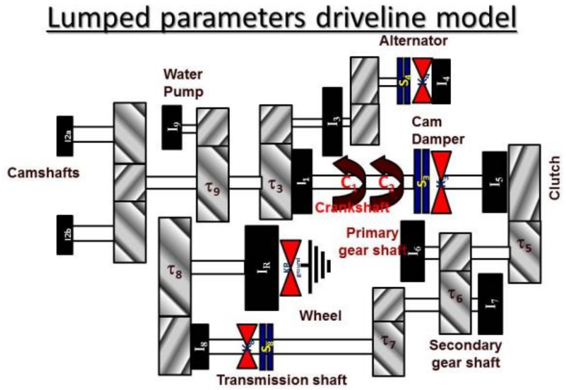

Figure 7 - Lumped parameters Driveline model

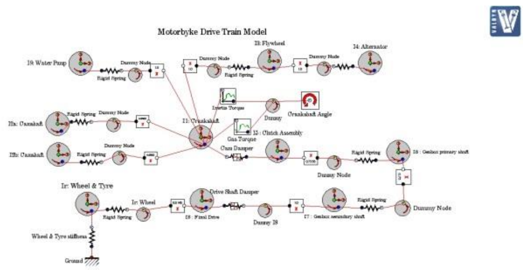

Figure 8 - Valdyn model of the driveline

		I_1	I_2	I_3	I_4	I_5	I_6	I_7	I_8	I_9
		Crankshaft	Camshaft	Flywheel	Alternator	Clutch assembly	Primary Gear shaft	Secondary gear shaft	Wheel	Water pump
Inserted Gear	1	39%	0%	15%	4%	10%	21%	0%	9%	1%
	2	35%	0%	14%	4%	9%	19%	0%	18%	1%
	3	31%	0%	12%	3%	8%	16%	1%	28%	1%
	4	27%	0%	10%	3%	7%	14%	1%	37%	1%
	5	24%	0%	9%	3%	6%	13%	1%	43%	1%
	6	23%	0%	9%	2%	6%	12%	1%	47%	1%

Figure 9 - Inertia contributions - Referred to crankshaft axis

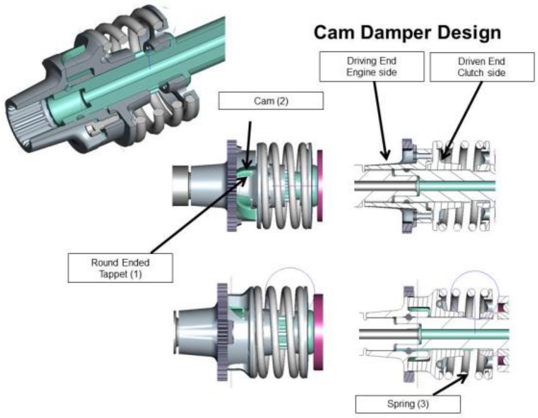

Figure 10 - Damper Design

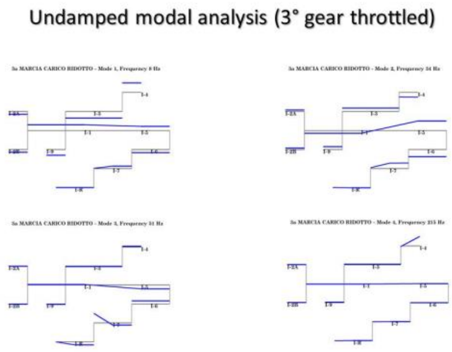

Figure 11 - Undamped modal Analysis

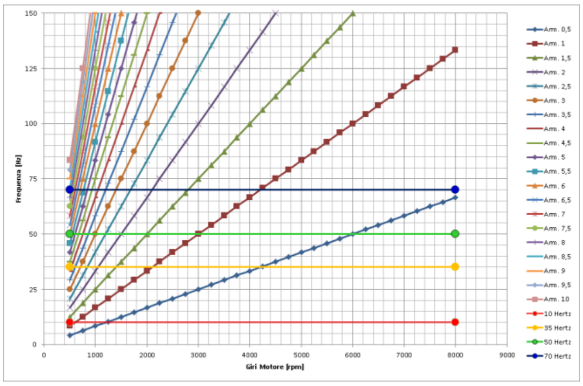

Figure 12 - Campbell diagram with relevant frequencies

Use	Gear	Vehicle Speed	Engine Load
		[km/h]	
Highway Constant speed	6ª	140,0	Intermediate
Highway Acceleration	6ª	140,0	Full
Extraurban Road Constant Speed	5ª	85,0	Intermediate
Extraurban Road Acceleration	5ª	85,0	Full
Extraurban Road Constant Speed	4ª	57,0	Intermediate
Extraurban Road Acceleration	4ª	57,0	Full
Urban Road Acceleration	3ª	57,0	Intermediate
Urban Road Constant Speed	3ª	57,0	Low

Figure 13 - Reference conditions

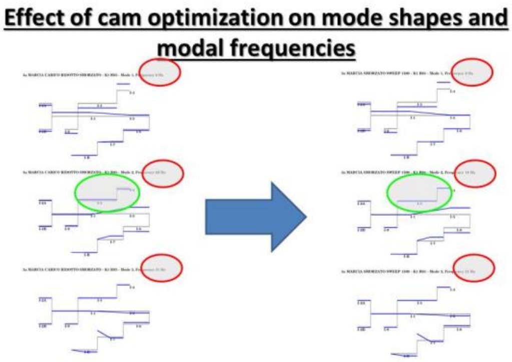

Figure 14 - Effect of cam profile optimization on vibration modes

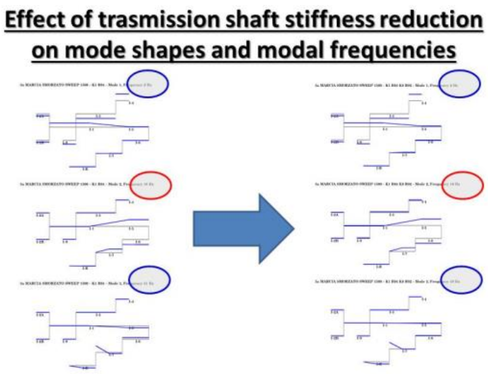

Figure 15 - Effect of transmission shaft reduction on vibration modes

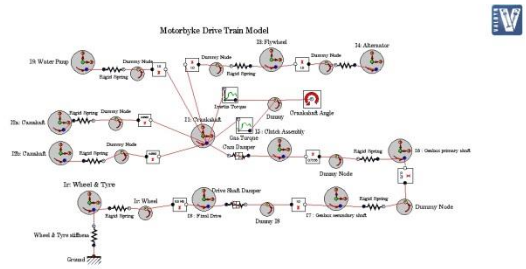

Figure 16 - Driveline Valdyn Model

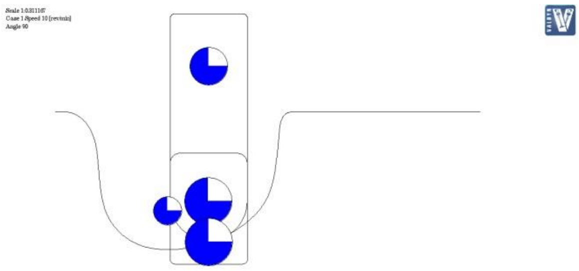

Figure 17 - Valdyn LAMINA elements modelling the damper

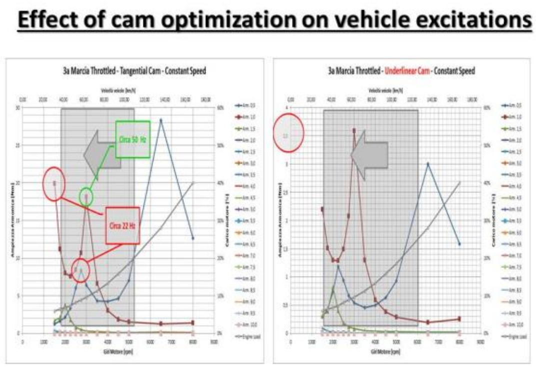

Figure 18 . Effect of cam profile optimization on vehicle excitation - Amplified Y scale

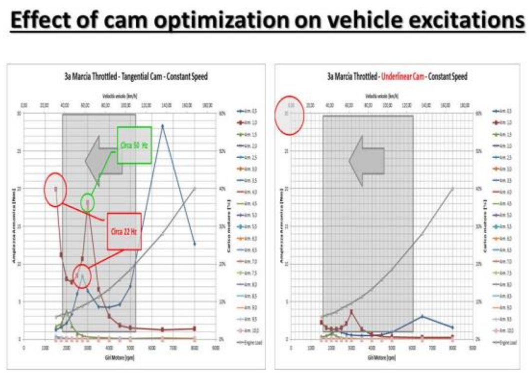

Figure 19 - Effect of cam profile optimization on vehicle excitation - Equivalent Y scales

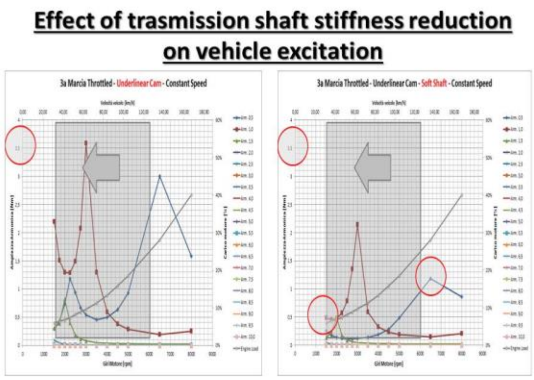

Figure 20 - Effect of transmission shaft reduction on vehicle excitation