Experimental Validation of a Methodology for Torsional Vibration Analysis in Internal Combustion Engines

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Abstract—*The scope of this paper is the study of the crankshaft torsional vibration phenomenon in internal combustion engines. The formulation, based on state equation solution with system steady state response calculation done by transition state matrix and the convolution integral, will be applied to a six-cylinder Diesel engine for vehicular application manufactured by MWM International Motores. The analysis considers a rubber and a viscous damper assembled to the crankshaft front-end. From the torsional vibrations analysis, it is possible to obtain the dynamic loading on each crankshaft section and these loads can be applied as boundary conditions in a finite element model to predict the safety factor of the component and compare the system behavior with rubber and viscous damping options. By this way, it is possible to emphasize the importance of the torsional vibrations analysis on the structural dimensioning of the crankshafts. The vibration amplitudes results will be compared to measured values for experimental validation of proposed mathematical model.*

Keywords: Torsional Vibrations, Crankshaft, Damper.

I. Introduction

Initially, an analysis considering no torsional vibration damper (TVD) was performed to adjust and calibrate the engine internal damping and the natural frequencies of the system. In the second step, it was considered a rubber TVD, where its power dissipation capability was checked for structural integrity verification. Finally, the calculations were done considering a viscous damper at the system.

Complete torsional vibration analysis (TVA) including the calculation of the vibration amplitudes at the crankshaft front-end, actuating torques at rear and front bolted connections and damper power dissipation will be performed for the both cases. The theoretical results, summarized in [1], will be compared to experimental data.

The technical characteristics of the engine, which crankshaft system will be analyzed, are listed below:

- Ignition timing: 1-5-3-6-2-4;
- Four-stroke cycle;
- Connecting rod length: 207 mm;
- Cylinder bore: 105 mm;
- Piston stroke: 137 mm;
- Oscillating masses: 2.521 kg;
- Maximum torque: 1100 N.m at 1200 rpm;
- Maximum power: 228 kW at 2200 rpm;
- Maximum engine speed: 2550 rpm;

To determine the dynamical and dissipative characteristics of the absorbers and system, some experimental constants must be used [2]. A complete description can be found in references [3] and [4].

From the torsional vibration analysis, it is possible to calculate the dissipated energy at the TVD. The damper thermal load is given by:

$$Q_j = \int_0^t Cr_j \cdot \left(\dot{\theta}_j - \dot{\theta}_3\right)^2 dt \quad ; \text{ j = 1, 2 (double rubber TVD)} \qquad (1)$$

$$Q_1 = \int_0^t Cr_1 \cdot \left(\dot{\theta}_1 - \dot{\theta}_2\right)^2 dt \quad \text{(single viscous TVD)} \qquad (2)$$

The permissible dissipated power for a rubber damper can be calculated as follows:

The mean convection coefficient at the damper external surfaces can be computed according to reference [2], considering the equation bellow:

$$h_c = 7.56 \cdot \left(\frac{\pi \cdot D \cdot n}{60}\right)^{0.8} \quad [\text{W/m}^2\,\text{K}] \qquad (3)$$

Where:
D – diameter for convection coefficient evaluation [m]
n – engine speed [rpm]

Applying a thermal load in a finite element model and considering a thermal conductivity of 0.26 W/m.K for the rubber, it is possible to determine the maximum power that the damper of the figure 1 can dissipate, taking into account that 120°C is the maximum operational temperature for nitrile butadiene rubber (NBR). The thermal analysis results, showed in figure 2, considers a

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heat generation at the rubber of the 1st damper ring as being 582000 W/m³ and 1500000 W/m³ for the 2nd damper ring. The results of the analysis indicate that the permissible damper load is approximate 250 W at each damper ring.

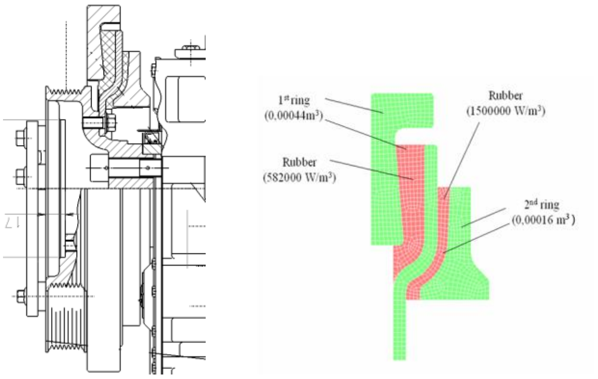

Fig. 1: FEM for thermal analyses

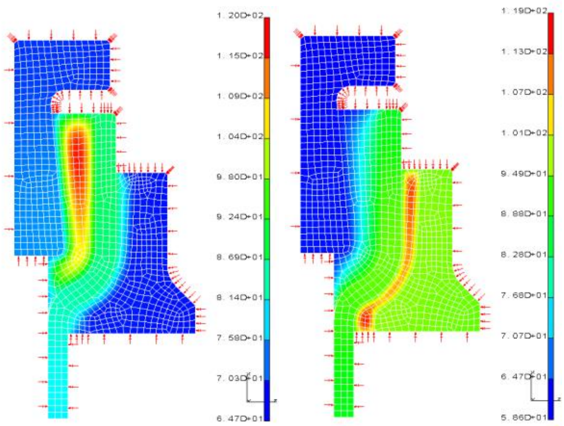

Fig. 2: Results of FE thermal analyses [°C]

Figure 2 indicates the rubber maximum temperatures for the first and second rings individualy.

In case of viscous TVD, the permissible dissipated power can be calculated according to the Iwamoto [2] equation:

$$\dot{Q}_{perm} = f \cdot 105 \cdot am \cdot Ad^{1.3} \cdot \left(\frac{2 \cdot \pi \cdot n}{60}\right)^{0.8} \cdot (t_o - t_{amb}) \qquad (4)$$

Where:
- f = 1.23 for dampers with cooling fins, otherwise f = 1.0
- Ad – reference area of the TVD ring [m²]
- am – damper size factor: 0.0201 ... 0.0303
- n – engine speed [rpm]
- t_o – temperature at TVD surface [°C]
- t_{amb} – ambient temperature [°C]

For rubber TVD, it is also possible to calculate the actuating shear stress and maximum deformation of the rubber. The maximum shear stress shall not exceed 0.3 ... 0.4 MPa. Its calculation can be done, through the relation of the torque between the damper ring and hub and the rubber section modulus under shear:

$$\tau_j = \frac{|\theta_j - \theta_3| \cdot Kt_j}{Wt_j} \quad ; \; j = 1, 2 \text{ (for a double TVD)} \qquad (5)$$

The maximum deformation of the rubber shall not exceed 15 … 20% and its calculation can be done by the following equation, considering that for small angles $tg(\Delta\theta) \cong \Delta\theta$:

$$\varepsilon_j = \frac{\tau_{j\max} \cdot Wt_j}{Kt_j} \cdot \frac{R_j}{e_j} \cdot 100\% \quad ; \; j = 1, 2 \qquad (6)$$

Where:
- Wt – rubber section modulus under shear
- Kt – rubber torsional stiffness
- θ – torsional vibration amplitude
- R – maximum radius of the rubber at TVD
- e – rubber thickness

These permissible parameters are stipulated by TVD manufactures and its reliability is obtained from many tests at dynamometers and vehicles.

II. Results

In this section, we will present the input data of the analyzed system and the results of the calculations that were performed on torsional vibration analysis software developed in MATLAB®. More detailed information about the equivalent systems can be found at references [3] and [4].

• Dynamical characteristics of system without TVD:

Inertia [kg.m²]:

I(1) = 0.0170	(Crankshaft pulley)
I(2) = 0.0090	(Gear train)
I(3) = 0.0467	(1st crank throw and oscillating masses)
I(4) = 0.0327	(2nd crank throw and oscillating masses)
I(5) = 0.0467	(3rd crank throw and oscillating masses)
I(6) = 0.0467	(4th crank throw and oscillating masses)
I(7) = 0.0327	(5th crank throw and oscillating masses)
I(8) = 0.0487	(6th crank throw and oscillating masses)
I(9) = 2.0750	(Flywheel and dynamometer coupling)

Torsional stiffness [N.m/rad]:

Kt(1) = 1106000	Kt(2) = 1631000
Kt(3) = 1253000	Kt(4) = 1253000
Kt(5) = 1678000	Kt(6) = 1253000
Kt(7) = 1253000	Kt(8) = 1976000

Absolute damping [N.m.s/rad]:
Ca(1) = 2.0 (1st cylinder) Ca(2) = 2.0 (2nd cylinder)
Ca(3) = 2.0 (3rd cylinder) Ca(4) = 2.0 (4th cylinder)
Ca(5) = 2.0 (5th cylinder) Ca(6) = 2.0 (6th cylinder)

Relative damping:
Engine mean loss factor: d = 0.035
General data considered at the analysis:
Gear train constant torque: 86 N.m
Front-end permissible torque: 2012 N.m
Rear-end permissible torque: 5413 N.m

• Dynamical characteristics considering a viscous TVD:
Inertia [kg.m²]:
I(1) = 0.1520 (TVD ring)
I(2) = 0.0970 (TVD hub and crankshaft pulley)

Torsional stiffness [N.m/rad]:
Kt(1) = calculated according to the methodology of reference [1].

General TVD data:
Kinematic viscosity of the silicone: $\nu = 0.2$ m²/s
Clearance factor: S = 5.0 m³
Damper size factor: am = 0.025
Reference area of the TVD ring: Ad = 0.1396 m²
Silicone film maximum temperature: $t_{SIL} = 115$ °C
TVD maximum temperature: $t_o = 100$ °C
Ambient temperature: $t_{amb} = 51$ °C

• Dynamical characteristics of a double mass rubber TVD:
Inertia [kg.m²]:
I(1) = 0.1230 (TVD 1st ring)
I(2) = 0.0273 (TVD 2nd ring)
I(3) = 0.0440 (TVD hub and crankshaft pulley)

Torsional stiffness [N.m/rad]:
Kt(1) = 70000 (TVD 1st ring)
Kt(2) = 88000 (TVD 2nd ring)

Relative damping:
Rubber loss factor: d = 0.15

General TVD data:
Rubber volume (1st ring): 0.00044 m³
Rubber volume (2nd ring): 0.00016 m³
Section modulus under shear (1st ring): $3.809 \cdot 10^{-3}$ m³
Section modulus under shear (2nd ring): $2.727 \cdot 10^{-3}$ m³

All analysis considered the measured combustion pressure curves to determine the excitation torque at the system.

Figures 3, 5 and 6 present the results of theoretical torsional vibration analysis, considering no TVD assembled to the crankshaft. Comparing the theoretical vibration amplitudes, to the measured ones (figure 4), it is possible to determine the actual damping coefficients of the engine.

The permissible torque was calculated considering the geometric dimensions of the crankshaft front and rear ends.

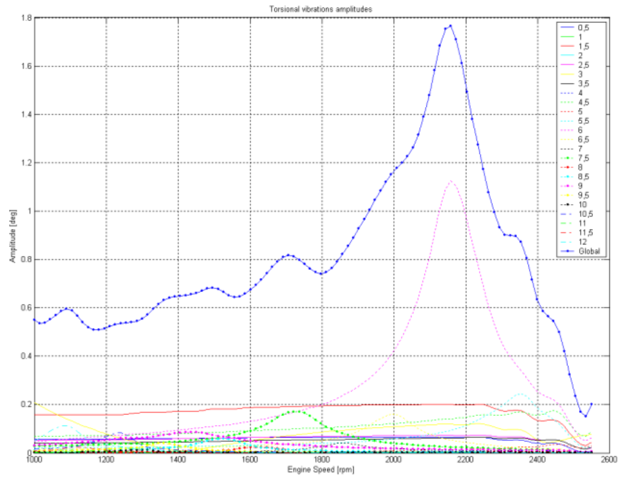

Fig. 3: Calculated torsional vibration amplitudes at crankshaft pulley without TVD

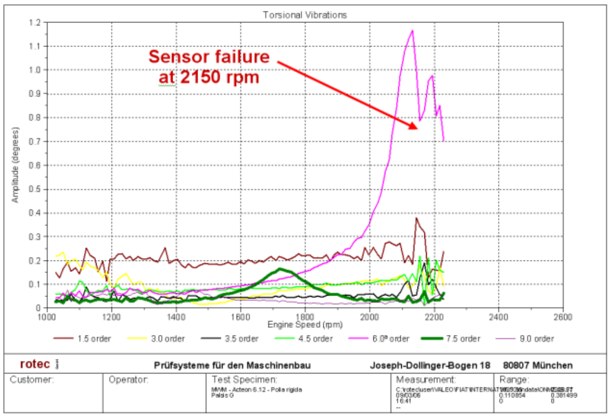

Fig. 4: Measured torsional vibration amplitudes at the crankshaft pulley without TVD

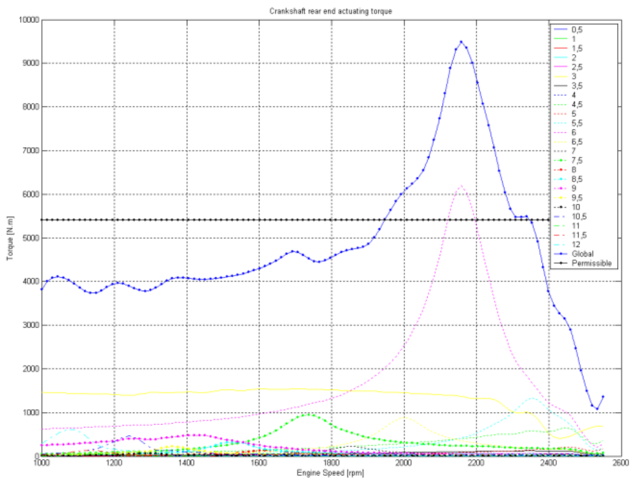

Fig. 5: Torque between the flywheel and the 6th cylinder without TVD

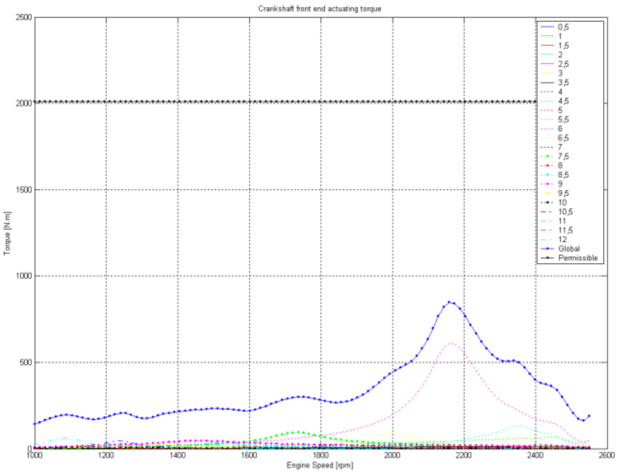

Fig. 6: Torque between the crankshaft pulley and the gear train without TVD

The next conditions consider the viscous damper to reduce the torsional vibration amplitudes and these results are based on data presented previously.

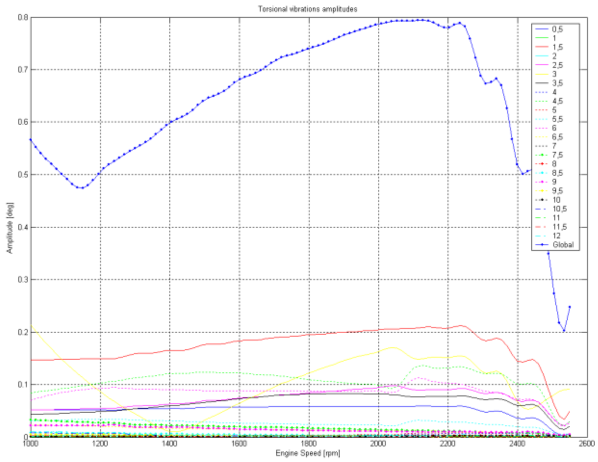

Fig. 7: Calculated torsional vibration amplitudes at crankshaft pulley with viscous TVD

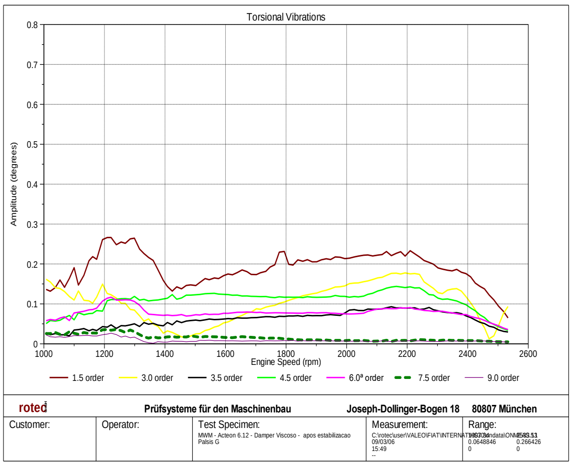

Fig. 8: Measured torsional vibration amplitudes at the crankshaft pulley with viscous TVD

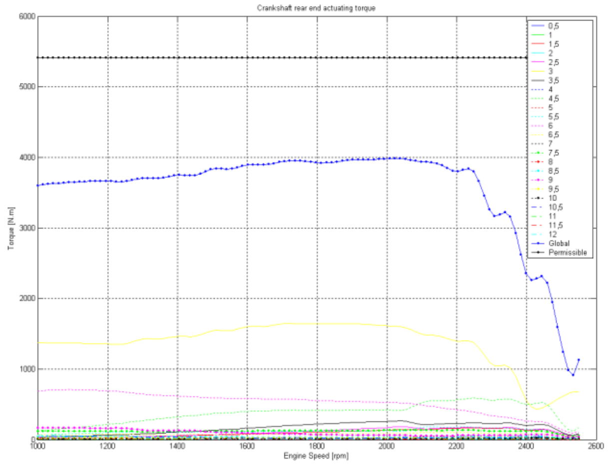

Fig. 9: Torque between the flywheel and the 6th cylinder with viscous TVD

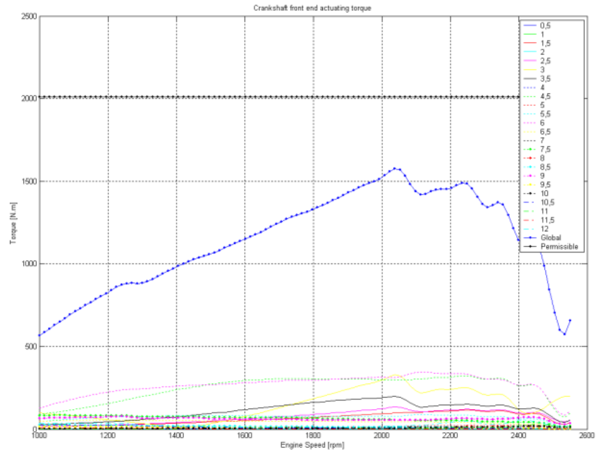

Fig. 10: Torque between the crankshaft pulley and the gear train with viscous TVD

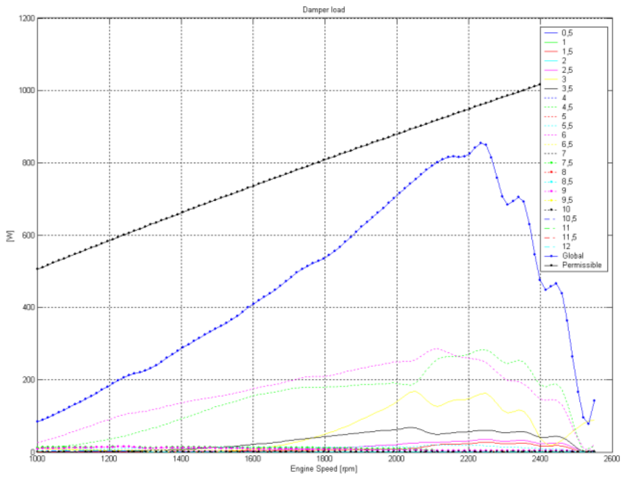

Fig. 11: Viscous damper load

Finally, it will be presented the torsional vibrations analysis considering the double mass rubber damper:

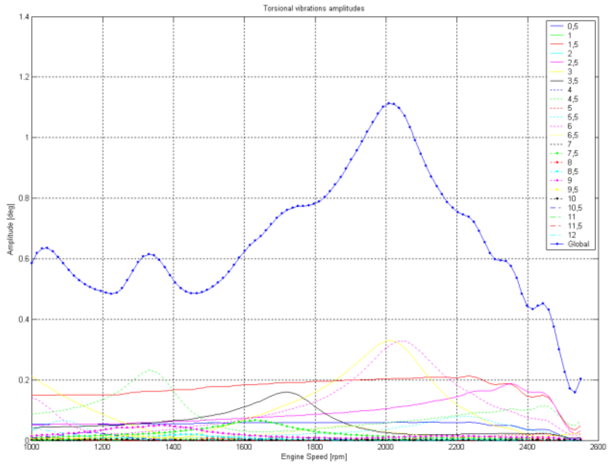

Fig. 12: Calculated torsional vibration amplitudes at crankshaft pulley with rubber TVD

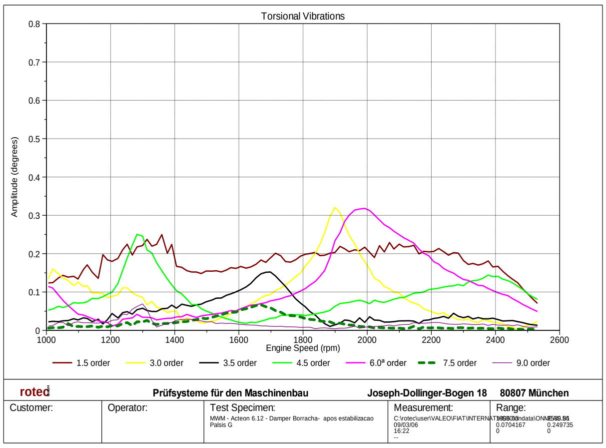

Fig. 13: Measured torsional vibration amplitudes at the crankshaft pulley with rubber TVD

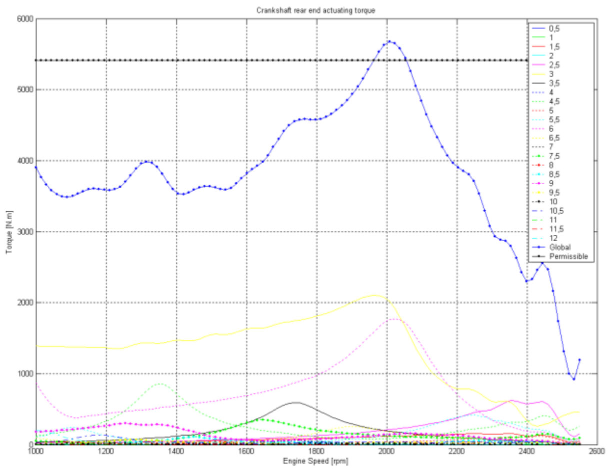

Fig. 14: Torque between the flywheel and the 6th cylinder with rubber TVD

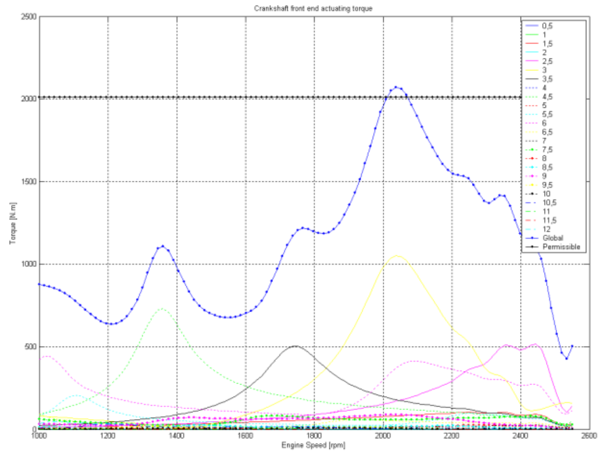

Fig. 15: Torque between the crankshaft pulley and the gear train with rubber TVD

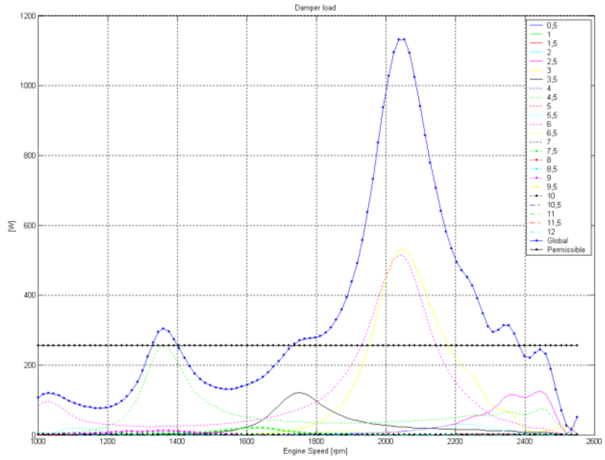

Fig. 16: Rubber damper load (1st ring)

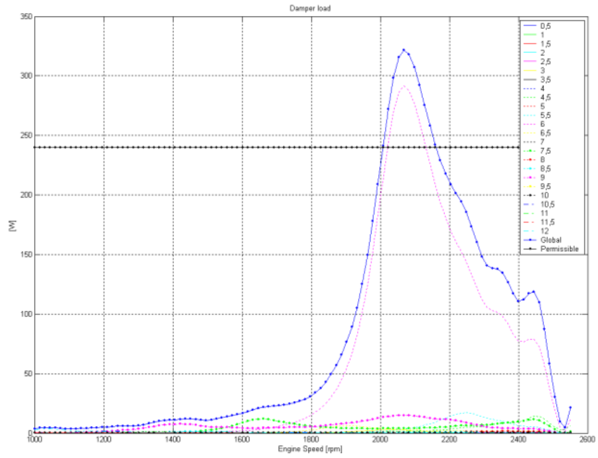

Fig. 17: Rubber damper load (2nd ring)

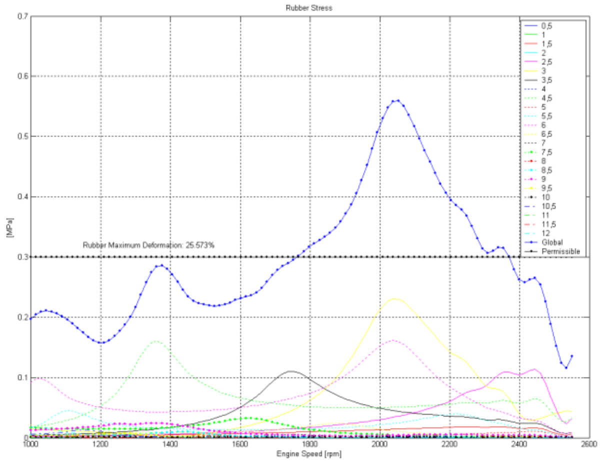

Fig. 18: Rubber shear stress (1st ring)

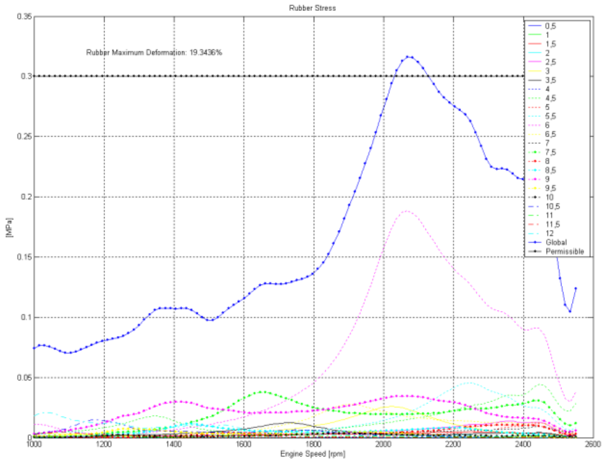

Fig. 19: Rubber shear stress (2nd ring)

III. Conclusions

Analyzing the obtained results and comparing them to the measured ones it is possible to conclude that the proposed methodology for TVA presents similar results and the validorhypotheses adopted for the equivalent model determination are valid.

According to design criteria, regarding noise level and structural integrity, the maximum recommended vibration amplitudes, per order, at the crankshaft front-end are in the range of 0.20° ... 0.25° for in-line six cylinders engines. Considering the presented results for double mass rubber damper, it is possible to note that the 3rd order / 1st mode (3/I) and 6th order / 2nd mode (6/II) have amplitudes higher than 0.30°. The maximum dissipate power is close to 1100W at the 1st TVD ring and 325W for the second one. For this type of component, the permissible damper load is about 250W continuously and taking into account the figures 14 and 15, we can verify that the actuating torques at front and rear-end connections are higher the permissible limit. Shear stress and rubber deformation are also above the recommended limits.

By this way, the rubber TVD is not recommended for the considered engine and only viscous damper is suitable for the mentioned application regarding the mentioned criteria. The next figure shows a structural failure of the rubber TVD occurred at a dynamometer test considering the system critical engine speed, i.e., approx. 2100 rpm.

Fig. 20: Structural failure at 1st TVD ring due to overload

The inclusion of the axial and flexural vibrations at the proposed model can be considered as a next step for crankshaft torsional vibration analysis, taking into account that in some particular cases, the axial vibrations cannot be disregarded from the system.

References

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