

INTERDEPENDENCE OF TORSIONAL VIBRATION DAMPER PARAMETERS ON CRANKSHAFT'S TORSIONAL VIBRATIONS

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Abstract

Torsional vibration of crankshaft is a harmful phenomenon in every type of engine. It can easily cause a fatigue failure in engines with relatively long crankshafts and big bore diameters (usually above 90 mm). For these types of engines, the resonance caused by specific low order harmonic of gas force and inertia force of oscillating masses occurs in engine speed range, thus resonant torsional moment of big amplitude is generated. This moment, additionally acting on crankshaft, is harmful and can easily cause its fatigue failure. In order to reduce the resonance effects, the torsional vibration damper (TVD) has to be used. That is the reason why application of torsional vibration dampers is almost a rule in these types of engines. The viscose TVD is the most popular type of vibration damper in heavy duty diesel engine. Moment of inertia of its plunger is the main criterion of damper selection. It is the basic factor, which determines the amount of dissipation work, decreasing the additional torsional moment caused by resonance. Increasing plunger's moment of inertia induce the rise of dissipation work but simultaneously decrease natural frequencies of a crank train, what allows to resonate with low order harmonic of tangential force acting on crankpin. The paper concerns an analysis of influence of viscose TVD parameters on crankshaft's torsional vibration for six cylinder inline heavy duty diesel engine.

Keywords: transport, combustion engines, HD engines, crank drive, torsional vibration

1. Introduction

Torsional vibration of crankshaft presents a significant problem in big bore multi-cylinder engines ($D > 90$ mm). For these types of engines, the resonance caused by specific low order harmonic of gas force and inertia force of oscillating masses occurs in engine speed range, thus resonant torsional moment of big amplitude is generated. This moment, additionally acting on crankshaft, is harmful and can easily cause its fatigue failure. In order to reduce the resonance effects, the torsional vibration damper (TVD) has to be used.

Analysis of torsional vibration can be carry out by different calculation methods. Nevertheless all of them consist of:

- calculation of natural frequencies of crank train,
- determination of different vibration modes,
- Fourier analysis of exciter forces,
- determination of exciter work,
- calculation of damping work, including dissipation work of TVD,
- determination of resonant moment amplitude,
- firing order analysis.

Calculation of natural frequencies of crank train and determination of different modes is usually made by taking advantage of crank train reduction to multi-mass model (Fig. 1). The model consists of elements with specified moments of inertia θ_i and zero-mass elements with determined flexibilities $e_{i,i+1}$. The θ_i parameters take into consideration masses of specified parts of

crankshaft, connecting rod assemblies and piston assemblies, when $e_{i,i+1}$ describes flexibilities of crankpins, main journals, and cranks.

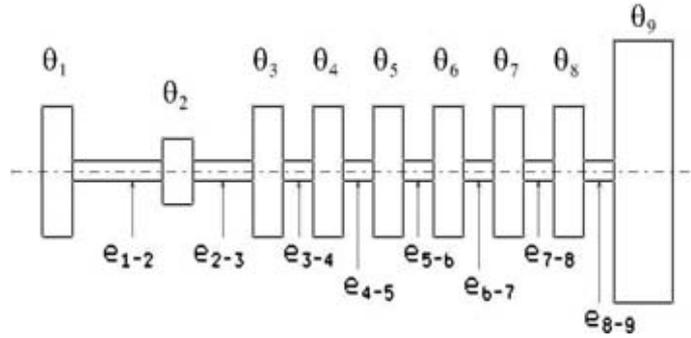


Fig. 1. Multi-mass model of 6 cylinder inline engine

The solution of the physical problem can be obtained by solving system of differential equations provided by Lagrange formula:

$$\frac{d}{d\tau} \left(\frac{\partial E}{\partial \dot{B}_i} \right) + \frac{\partial V}{\partial B_i} = 0, \quad (1)$$

where:

- E - kinetic energy of whole model,
- V - potential energy of whole model,
- B - instantaneous angle deflection of element i ,
- τ - time.

Results of foregoing problem return values of natural frequencies and relative amplitudes of different modes of resonance.

Exciter forces, which act on each crank pin, consist of cylinder gas forces and tangential forces acting on specific crank pin caused by oscillating masses. In order to calculate exciter work, above mentioned forces have to be subjected to Fourier analysis. The exciter work can be then expressed by dependence (2):

$$L = \pi \sum_{i=1}^c [M]_i A_i \sin \beta_i, \quad (2)$$

where:

- $[M]$ - amplitude of torque of considered harmonic of gas forces (cylinder i),
- A_i - torsional vibration amplitude of considered crank i ,
- β_i - angular delay of torsion relating to torque harmonic of cylinder i ,
- c - number of cylinders.

In connection with equality of $[M]_i$ values for all cylinders, and additionally defining amplitudes using relative amplitudes as follows:

$$A_i = A_1 \alpha_i, \quad (3)$$

The summation in (2) can be reduced to:

$$\sum_{i=1}^c \alpha_i \sin \beta_i = \sigma \sin \beta, \quad (3)$$

The relation (3) introduces mathematical quantity called resultant relative amplitude which depends on order of considered harmonic and assumed firing order. The exciter work during resonance becomes maximal ($\sin \beta = 1$), and finally it can be determined as:

$$L_{res} = \pi [M] A_1 \sigma, \quad (4)$$

Damping forces during oscillations are mainly caused by piston friction. Other source of damping is crankshaft material, but only cast iron hysteresis has significant influence of damping work. Considering steel crankshafts, damping into material can be neglected. Reminding that piston friction is assumed as semi-fluid, the moment of damping can be expressed as:

$$M_t = \mu \frac{dB}{d\tau}, \quad (5)$$

where:

- M_t - moment of torsional vibration damping,
- μ - damping coefficient,
- B - torsion,
- τ - time.

After mathematical transformations, damping work caused by pistons friction is expressed as:

$$L_{tc} = \pi\mu\Omega A_1^2 \sum_{i=1}^c \alpha_i^2 = \pi\mu\Omega A_1^2 (\alpha^2)_w, \quad (6)$$

where:

- Ω - frequency of crankshaft oscillations,
- A_1 - amplitude of first mass oscillation,
- $(\alpha^2)_w$ - sum of squares of relative amplitudes.

The damping coefficient μ is usually determined by Holtzer formula:

$$\mu = 0,04\theta\Omega, \quad (7)$$

Additional damping is provided by torsional vibration damper. There are many different types of TVD and nevertheless the viscose type is the most popular as efficient and durable. The damping work of this type of TVD can be expressed as:

$$L_T = \frac{\pi \cdot \mu_t \cdot \Omega \cdot A_{ob}^2}{1 + \left(\frac{\mu_t}{\Omega \cdot \theta_p} \right)^2}, \quad (8)$$

where:

- μ_t - viscose damping coefficient into TVD,
- A_{ob} - amplitude of torsion of TVD's housing,
- θ_p - moment of inertia of TVD's plunger.

Optimum damping coefficient for specific plunger's moment of inertia amounts to:

$$\mu_t = \Omega \cdot \theta_p, \quad (9)$$

and can be varied by changing damping fluid viscosity and gap between damper's housing and plunger. Application of TVD significantly reduces resonant amplitudes but simultaneously decreases natural frequencies of crankshaft. The additional moment of inertia applied on front of crankshaft by TVD, assuming that optimum viscose damping coefficient is used, is equal to sum of moment of inertia of damper's housing and half of moment of inertia of damper's plunger.

The possibility of determination of amplitudes during resonance occurs by equate exciter work with damping work:

$$L_{res} = L_{tc} + L_T, \quad (10)$$

The foregoing equation has only one unknown - amplitude of first mass, which thus can be easily calculated. Determination of first mass amplitude leads to defining the rest of amplitudes, twisting angles of specific elements of crankshaft and, using known flexibilities, torsional moments caused by resonance.

Firing order analysis takes into consideration, that different firing orders produces different exciter work which is caused by varying of resultant relative amplitude (3). Optimum firing order can only be found by evaluating full calculations for different firing orders.

2. Analysis of TVD application in six cylinder heavy duty inline engine

The analysis has been performed on designed six cylinder engine crank train of four stroke heavy duty diesel engine with 12 dm³ engine displacement. The determination of natural frequencies without TVD has been calculated taking advantage of multi-mass model analysis and using FEM, what gives possibility to determine the accuracy of results obtained from multi-mass model (Tab. 1).

Tab. 1. Comparison of natural frequencies obtained by multi-mass model analysis and FEM analysis

Mode	Natural frequencies (multi-mass model results)	Natural frequencies (FEM calculation results)	Relative error
	n_{mm}	n_{FEM}	n
	[1/min]	[1/min]	%
1	9954	10623	6.3
2	19726	19953	1.1

Analyses with different TVDs (viscose type) have been performed only on multi-mass model. The moment of inertia of plunger θ_p has been assumed as independent variable (8), keeping the viscose damping coefficient μ_i on its optimum value according to (9). The dependency between moment of inertia of plunger and crankshaft natural frequency of 1st mode is shown in Fig. 2. Application of TVD strongly reduces the natural frequency of first mode of crank train, which maximal speed reaches 1900 rpm, and also moves the node of oscillation to front of crankshaft (Fig. 3).

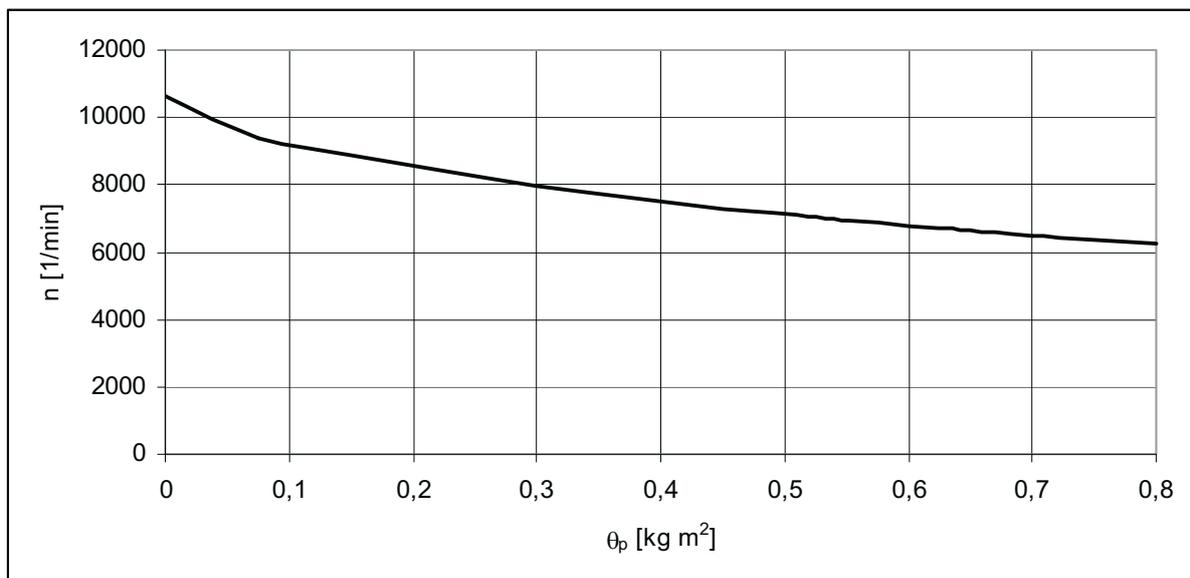


Fig. 2. Crankshaft natural frequency of first mode for different moment of inertia of TVD's plunger

Fourier analysis of exciter forces (tangential forces) has been performed up to six order harmonic ($h = 6$) - Fig. 4. It can be noticed, that low order harmonics has the biggest amplitude and thus the resonance with this harmonics has been assumed as the most dangerous. In order to determine exciter work, the resultant relative amplitudes for different firing orders have been calculated (Fig. 5).

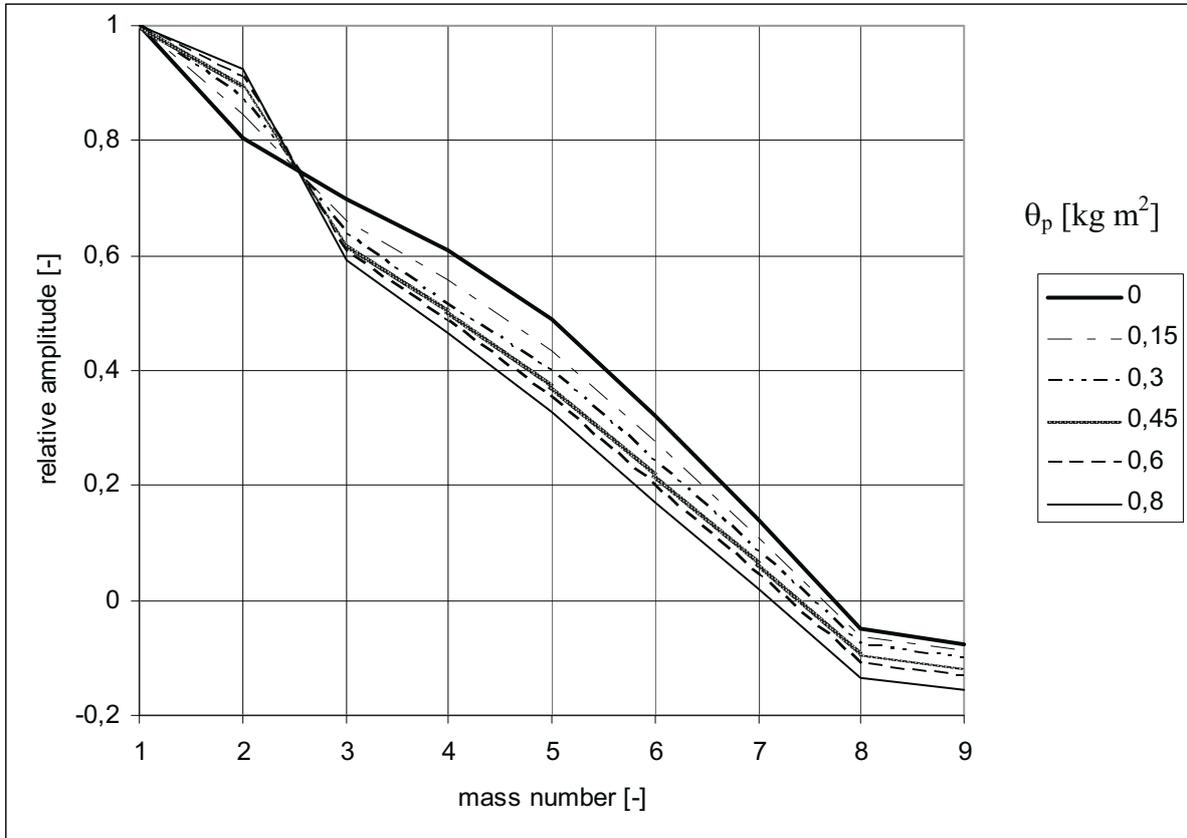


Fig. 3. 1st mode of torsional vibration as a function of TVD parameter

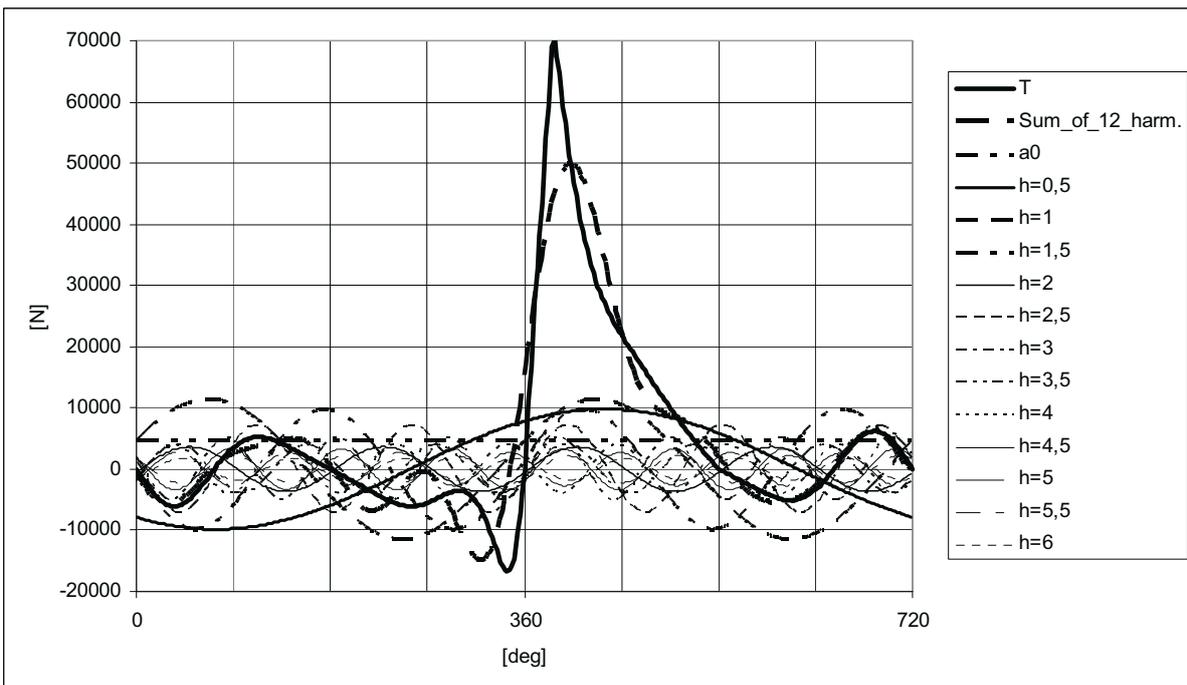


Fig. 4. Fourier analysis of exciter force (T)

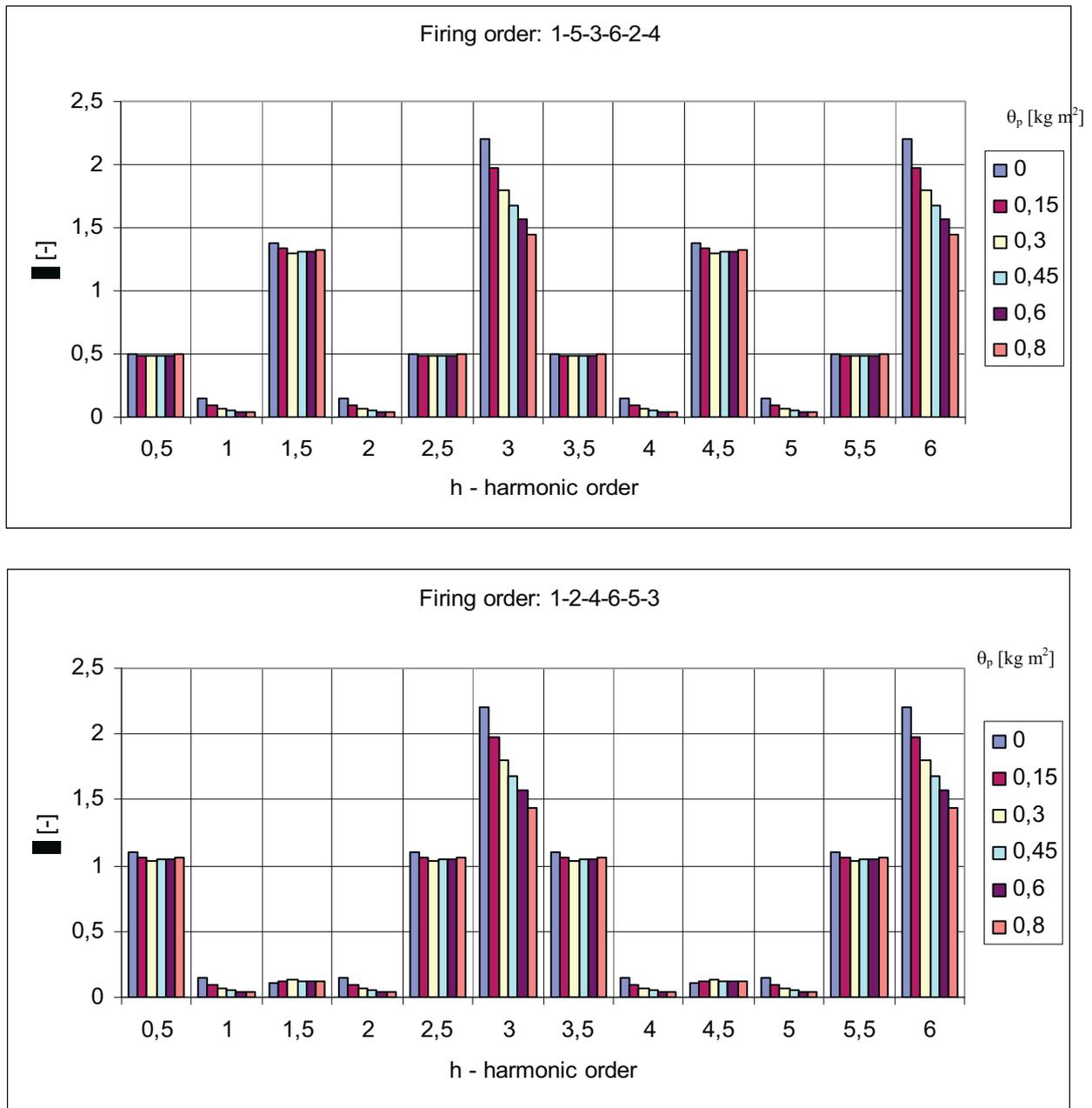


Fig. 5. Resultant relative amplitudes of 1st mode of oscillations as a function of TVD parameters for two possible firing orders

Analysing natural frequencies of crankshaft (Fig. 2) we can notice, that resonance occurs with 5th and higher orders of harmonic for small TVDs, and reach range of 3.5th and higher orders of harmonic for the biggest damper. Considering dependencies in Fig. 5 we can decide, that until 3.5 order harmonic didn't resonate and the 1-2-4-6-5-3 firing order would be an optimal solution. Application the TVD with $\theta_p > 0.625 \text{ kg m}^2$ causes foregoing harmonic to resonate and thus the optimal firing order would be 1-5-3-6-2-4. In addition, resonance of 3.5 order harmonic causes significant increasing of exciter work. Thus maximal resonant moment rises up in comparison with smaller dampers. The dependency between maximal amplitude of resonant moment of crankshaft and TVD parameters is shown in Fig. 6. There is no reason to apply TVD with higher moment of inertia of plunger then 0.625 kg m^2 because the amplitude of resonant moment rapidly increases. Figure 7 shows the torsional moment in crankshaft after 5th cylinder in resonance with 6th order harmonic which occurs under 1120 rpm engine speed.

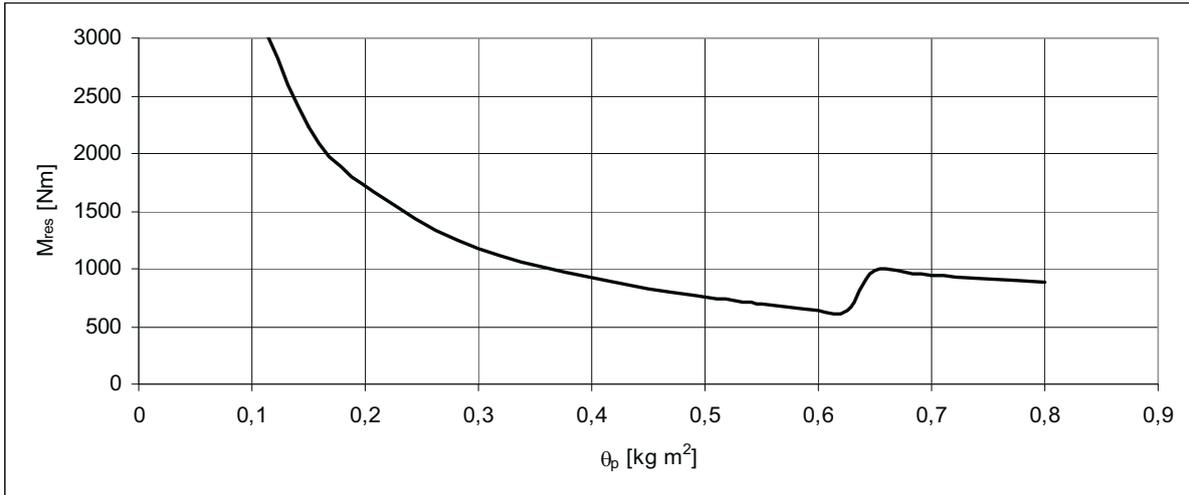


Fig. 6. Maximal amplitude of resonant moment of crankshaft as a function of TVD parameter

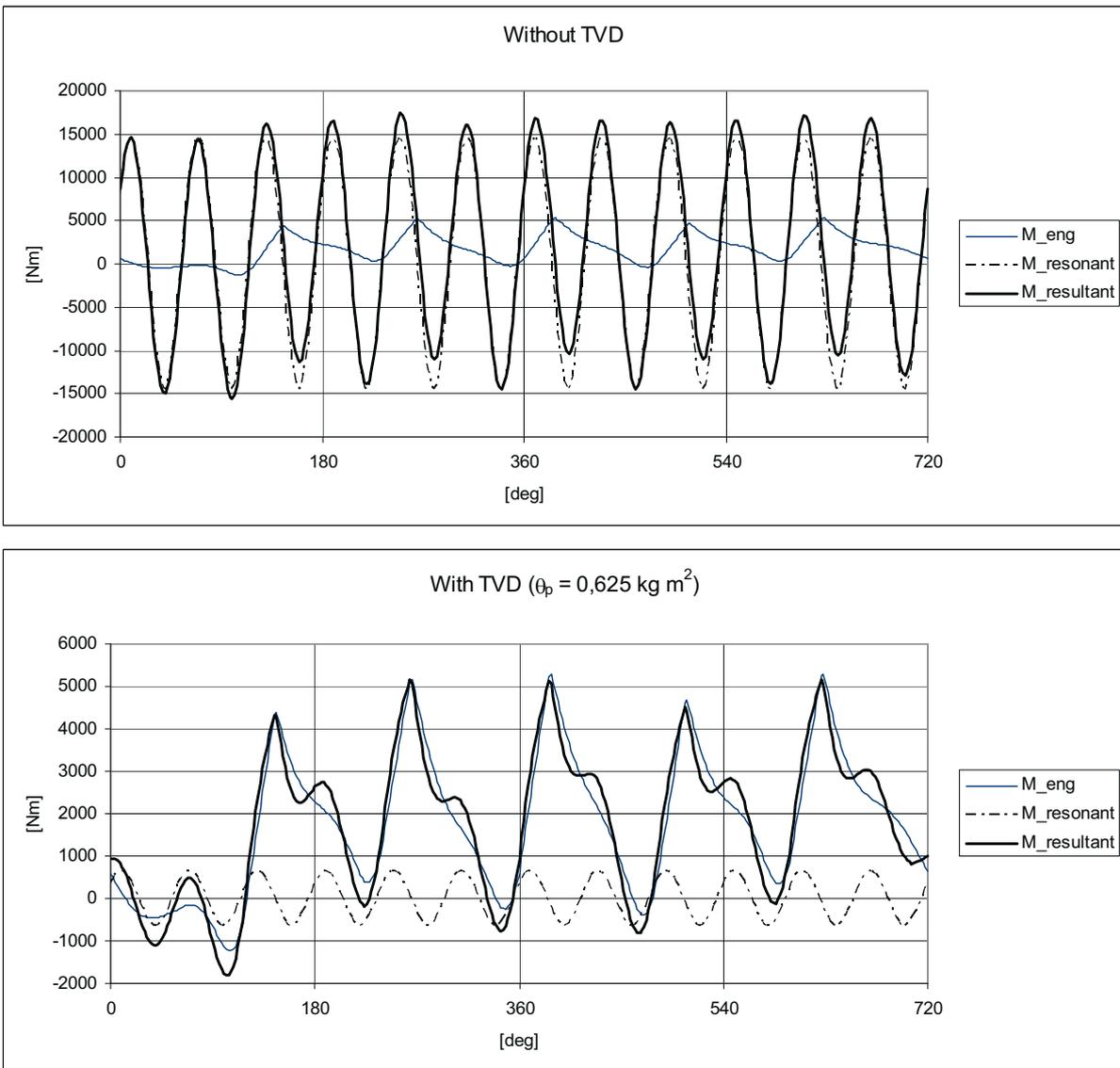


Fig. 7. Torsional moment after between 5th and 6th cylinder caused by tangential forces (M_{eng}), resonance and superposition of this loads in crankshaft with and without TVD.

In order to realize the damper efficiency, the figure also shows the torsional moment during resonance with the same harmonic without TVD.

Conclusions

- 1) Torsional vibrations are especially harmful for multi-cylinder big bore engines,
- 2) Application of viscose type TVD highly reduces the resonant effects,
- 3) Increasing moment of inertia of damper's plunger changes natural frequencies and modes of crankshaft,
- 4) There is optimal size of damper's plunger which gives the lowest values of amplitude of resonant moment.

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