POSSIBILITIES OF USING DYNAMIC TORSIONAL VIBRATION DAMPERS WITH SPRINGS IN IC ENGINES FOR ROAD VEHICLES

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1. INTRODUCTION

Internal combustion engines have a changing value of the current torque at the crankshaft within one working cycle (two revolutions of the crankshaft in four-stroke engines for one working cycle) due to their cyclical tape of work and the stochastic nature of the combustion process. Also within the IC engine due specific regulation processes a so called quasi stationary process occurs at constant speed operation [2], which further affects the change of torque at the crankshaft.

In addition to the above described character changes of the crankshaft excitation torque, the crank-gear shave variable moments of inertia of rotating masses and the crankshaft has a variable torsional stiffness by cross-sections [2, 3, 5]. All this results in the presence of significant torsional twisting at some cross-sections of the crankshaft with associated elements around the equilibrium of angular rotation. Occurrence of crankshaft twisting at some engine operating modes, primarily due to torsional vibration, can be of such intensity that can lead to fatigue and fracture of the crankshaft.

When defining the engine operating modes where high value amplitudes of crankshaft twist angle due to torsional vibration are expected, an important role has the machine that is powered by the internal combustion engine (road vehicle, power unit, ship, etc.).

This paper will observe the internal combustion engine without a powered machine, and the possibilities of reducing the amplitudes of twist angle due to torsional vibration at critical operation modes. Any attempt to avoid critical operation modes of the torsional vibration system, shifting the so called resonant crankshaft speeds outside the operation mode by changing the torsional stiffness and/or mass moments of inertia at the current level of development of crankshaft construction is quite unrealistic. The real way is the introduction of devices for forced reduction of critical amplitudes of twist angle due to torsional vibration, so-called torsional vibration dampers.

An overview of torsional vibration dampers which are used in internal combustion engines is best described in [9], from elastic dampers, dynamic absorbers, frictional dampers, hydraulic dampers to centrifugal pendulum which is used in some engines as a torsional vibration damper. Detailed overview of conventional torsional vibration dampers which are

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used for engines for road vehicles, with the basic characteristics of damping elements, is given in [1, 3].

Most commonly used conventional dampers for engines for road vehicles are the so-called viscoelastic dampers (dampers with rubber), who have a number of advantages (efficiency, durability, storage space, etc.). Approach to define the basic characteristics of damping and a way of modeling these dampers is given in [1, 5].

In recent years thanks to new technologies of engine parts manufacturing and new materials, there are also other solutions for torsional vibration damping based on so-called dynamic absorbers. Solutions of these dampers are shown in [4, 6, 8]. The advantages of these dampers are reflected in lower cost, higher reliability and the same or greater efficiency of damping of amplitudes of twist angle due to torsional vibration. Mounting and a way of modelling of the basic features of the new damper designs is described in [4].

In this paper a comparative analysis of amplitudes of twist angle due to torsional vibration with the application of conventional viscoelastic torsional vibration damper and a torsional vibration damper incorporated into the crankshaft counterweight in the form of a dynamic absorber for a same engine taking into account the real dimensions of the crankshaft counterweights is carried out.

2. OBJECT OF RESEARCH

For research purposes of applying a dynamic torsional vibration damper a medium speed, turbo-compressor charged, four-stroke, water cooled, six-cylinder diesel engine for road vehicles was considered. The main characteristics of the engine are given in Table 1.

Table 1 Wheel numeric equations proposed by different researchers

Number of cylinders	6
Piston diameter	D_k= 125 mm
Piston stroke	s = 150 mm
Power (P_e) / speed (n)	184 kW / 2100 rpm
Maximum torque	890 Nm
Firing order	1-5-3-6-2-4
Moment of inertia of masses according to Figure 5 a)	θ_1 = 0,0277 kgm^2, θ_2= θ_4= θ_5= θ_7=0,147 kgm^2, θ_3 = θ_6 = 0,0835 kgm^2, θ_8 = 1,87 kgm^2
Stiffness according to Figure 5 a)	c_1 = 4,32·10^6 Nm/rad, c_2 =...= c_7 = 2,81·10^6 Nm/rad, c_8 = 4,12·10^6 Nm/rad,
External damping coefficient	ν_1=... =ν_6= 7,5 Nms/rad
Internal damping coefficient	ε_1, ε_2, ..., ε_7, computed according to model in [3]

In series mass production the engine is equipped with a viscoelastic torsional vibration damper, shown in figure 1.

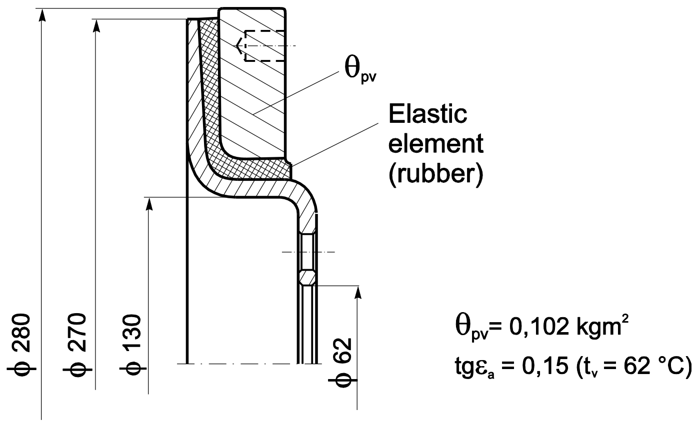

Figure 1 Scheme of a viscoelastic torsional vibration damper

The dynamic torsional vibration damper, with whom the research in this paper is carried out, is set within the crankshaft counterweight at the first joint. The inertial mass is attached to the crankshaft base structure with four coil springs and a stop bar. In Figure 2 the layout of the dynamic torsional vibration damper with the most important coil spring characteristics is shown.

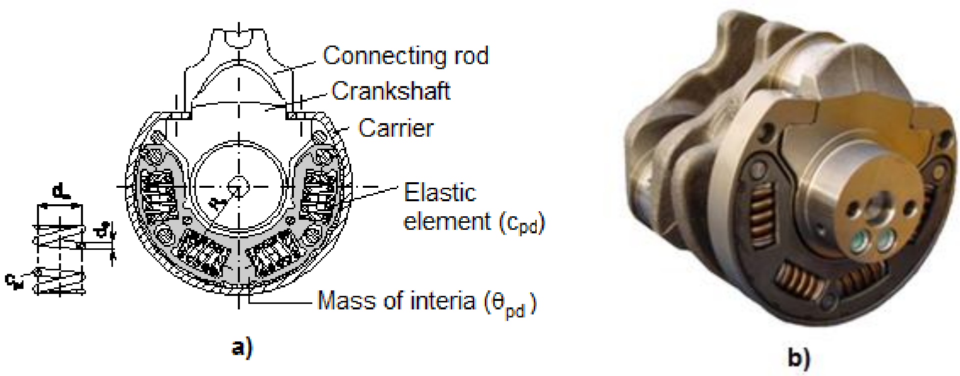

Figure 2 Scheme of the dynamic torsional vibration damper a) and an actual photo b)

The assemble position of the torsional vibration dampers from Figures 1 and 2, are schematically shown in Figure 3.

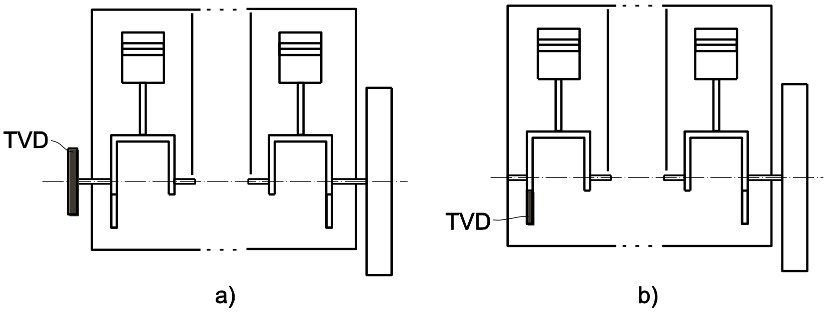

Figure 3 Assemble positions of the viscoelastic a) and dynamic b) torsional vibrations damper at the IC engine

3. MODELS FOR CALCULATING TWIST ANGLES DUE TO TORSIONAL VIBRATIONS

The analysis of the torsional vibration system was performed using the mass-spring type model [3]. The basic data for the given model, which are partially given in Table 1, are explained in detail in [5]. Since the viscoelastic damper has highly nonlinear damping (ε_{pv}) and stiffness (c_{pv}) characteristics, in this work the characteristics of the damper are obtained as a combination of computing and experiments presented in [3, 5] were used.

The values of the dynamic stiffness (c_{pv}) and the dynamic internal damping coefficient (ε_{pv}) of the damper in Figure 1 are given in Figure 4, where the surface temperature of the elastic element (rubber) is t_v=62 °C. This temperature is an average temperature of the elastic element at the external speed characteristic of the engine at which all experiments were performed. In Figure 4 the amplitude of twist angle of the appropriate masses is marked with (A_r).

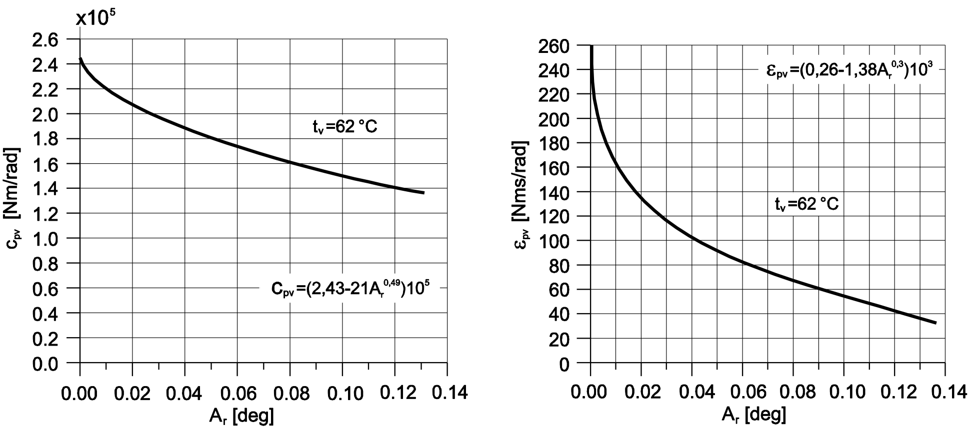

Figure 4 The diagram of torsional stiffness (cpv) and internal damping coefficient (εpv) transition of the elastic element of the viscoelastic damper (Figure 1) as a function of the relative twist angle amplitudes Ar=Ap-A1

The physical torsional vibration system model of the engine with the viscoelastic torsional vibration damper is shown in Figure 5 a) and in Figure 5 b) the physical model of the same engine with the dynamic torsional vibration damper incorporated into the crankshaft counterweight is shown.

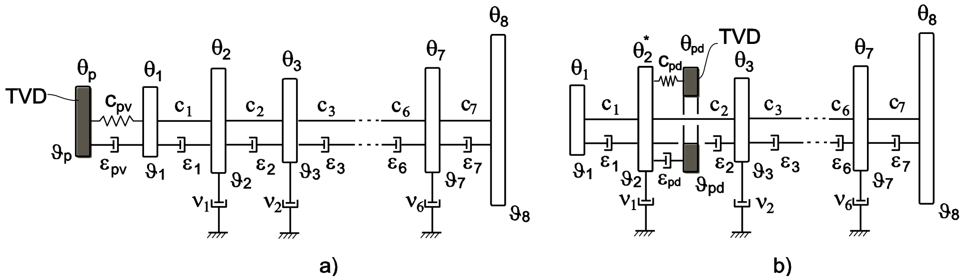

Figure 5 The physical torsional vibration system model with viscoelastic a) and dynamic b) torsional vibration damper

For the analysis of the torsional vibration system model given in Figure 5 a) the next mathematical model was used.

$$[I]\{\ddot{\vartheta}\}+[K]\{\dot{\vartheta}\}+[C]\{\vartheta\}=\{T(t)\} \tag{1}$$

where:

$[I]$ – matrix of the moment of inertia of masses (θ),

$[K]$ –internal and external damping matrix (ε,ν),

$[C]$ – stiffness matrix (c),

$\{T(t)\}$ – excitation matrix and

$\{\vartheta\}$, $\{\dot{\vartheta}\}$, $\{\ddot{\vartheta}\}$ – matrix of angular twist, speed and acceleration due to torsional vibration.

For the physical model shown in Figure 5 b) a corrected version of equation system (1) were used. The correction is reflected in the fact that instead of using one equation for disc (θ_2) and disc (θ_{pd}) of the type.

$$\theta_2\ddot{\vartheta}_2+\nu_1\dot{\vartheta}_2+\varepsilon_1(\dot{\vartheta}_2-\dot{\vartheta}_1)+\varepsilon_2(\dot{\vartheta}_2-\dot{\vartheta}_3)+c_1(\vartheta_2-\vartheta_1)+c_2(\vartheta_2-\vartheta_3)=T_2 \tag{2}$$

two equations were used:

$$\theta_2^*\ddot{\vartheta}_2+\nu_1\dot{\vartheta}_2+\varepsilon_1(\dot{\vartheta}_2-\dot{\vartheta}_1)+\varepsilon_2(\dot{\vartheta}_2-\dot{\vartheta}_3)+\varepsilon_{pd}(\dot{\vartheta}_2-\dot{\vartheta}_{pd})+c_1(\vartheta_2-\vartheta_1)+$$
$$+c_2(\vartheta_2-\vartheta_3)+c_{pd}(\vartheta_2-\vartheta_{pd})=T_2 \tag{3}$$

$$\theta_{pd}\ddot{\vartheta}_{pd}+\varepsilon_{pd}(\dot{\vartheta}_{pd}-\dot{\vartheta}_2)+c_{pd}(\vartheta_{pd}-\vartheta_2)=0$$

Using the mathematical model described with the differential equation system (1), using inputs for the engine with viscoelastic damper from Table 1 and values for (c_{pv}) and

(ε_{pv}) from Figure 4, amplitudes of twist angle of the pulley (A_1) were obtained through computing. Computing results were compared with corresponding experimental values, as shown in Figure 6.

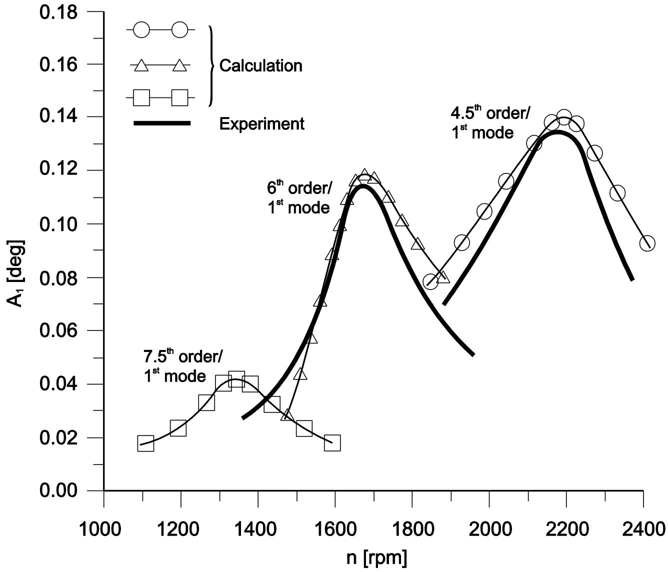

Figure 6 Comparison of the amplitude of twist angle at the engine pulley with viscoelastic torsional vibration damper obtained computationally and experimentally

By analysing comparative values of twist angles due to torsional vibrations shown in Figure 6, obtained computationally and experimentally, it can be concluded that the agreement between the results is satisfactory and that for, for further analysis of the influence of the damper on the torsional vibration of the crankshaft can be carried out through computation.

4. ANALYSIS OF THE COMPUTATIONAL RESULTS

Dynamic torsional vibration damper, based to Figure 2 b), is designed and incorporated in the first counterweight on the left crankshaft end (on the opposite side of the engine flywheel).Moment of inertia of the first left crank-gear with counterweight without damper is θ_2=0,147 kgm^2 (Table 1). Based on the analysis of the technical documentation and computation, the inertial mass of the counterweight, which is located on the first crank-gear from the left is 0,064 kgm^2.

Having in mind the shape change of the counterweight (Figure 2), the total inertia moment of the mass of the counterweight is computed and has the value 0,085 kgm^2. From the technical documentation 65% to 75% of the total moment of inertia of the mass belongs to the dynamic damper (inertial mass + springs) which in this particular case is θ_{pd} = 0,055 ÷ 0,065 kgm^2. The value of the moment of inertia of mass θ_2^* (crank gear with a new shape of counterweight without inertial mass) is in the range of 0,104 kgm^2 to 0,115 kgm^2.

Bearing in mind the design of the four coil springs (Figure 2), where in pare of two springs are acting on the inertial mass (θ_{pd}), realistically the dimensions of the springs can be seated as: wire diameter D_0 = 5.5 ÷ 6.3 mm, with a mean diameter of the spring d_m ~ 32 mm. Spring length in the free state is 50 mm. Bearing in mind the dimensions of this springs their individual axial stiffness ranges from 120 N/mm to 160 N/mm.

Taking into account springs in pares in parallel connection, their position relative to the center of rotation of the crankshaft is R_0=93÷95 mm, the torsional stiffness of the springs is within the range 1500 Nm/rad to 2700 Nm /rad. Using the above data, self-developed software for the mathematical model given by equations (1) and (3) an analysis of the changes in the maximum vibration amplitude for various values of the characteristics of the dynamic torsional vibration damper is performed.

In these analyses the coefficient of internal damping ε_{pd}=20 Nms/rad is maintained the same and it is the result of friction between the counterweight and elements of the inertial mass.

In order to obtain optimal characteristic values of the dynamic damper, torsional stiffness of the springs (c_{pd}) and the moment of inertia (θ_{pd}) of the inertial mass were varied. In the analysis the 6th order and I mode of vibration were considered, since the 4.5th order fall out of the operating range of the engine (n_{max}=2100 ^{o}/min). Despite the statements that the 4.5th order lies at the border of the operating range of the engine, the same trends can be observed for any of the main excitation orders. The results are shown in Figures 7 and 8. In these figures maximum values of the amplitude of twist angle of the engine pulley (A_{1max}), where practically the largest twist due to torsional vibrations occurs, are presented.

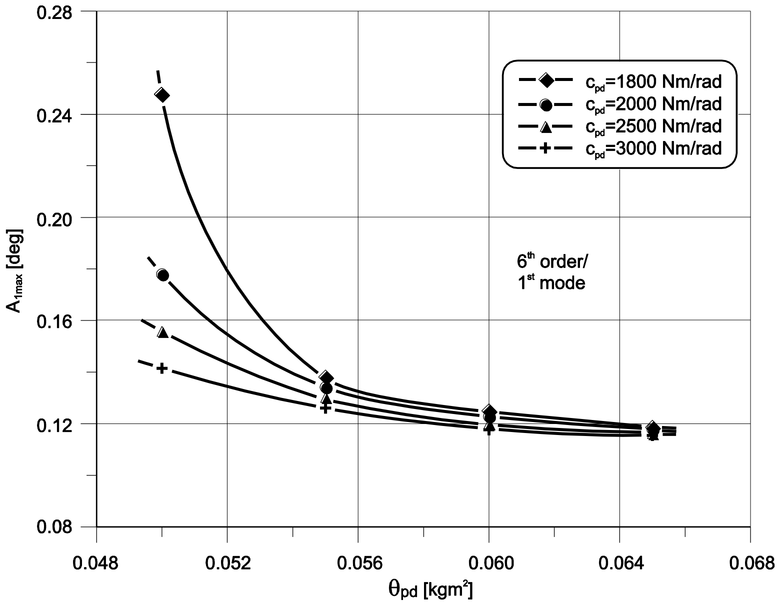

Figure 7 The diagram of maximum amplitudes of twist angle of the engine pulley as a function of the moment of inertia of the inertial mass and torsional spring stiffness of the damper as a parameter

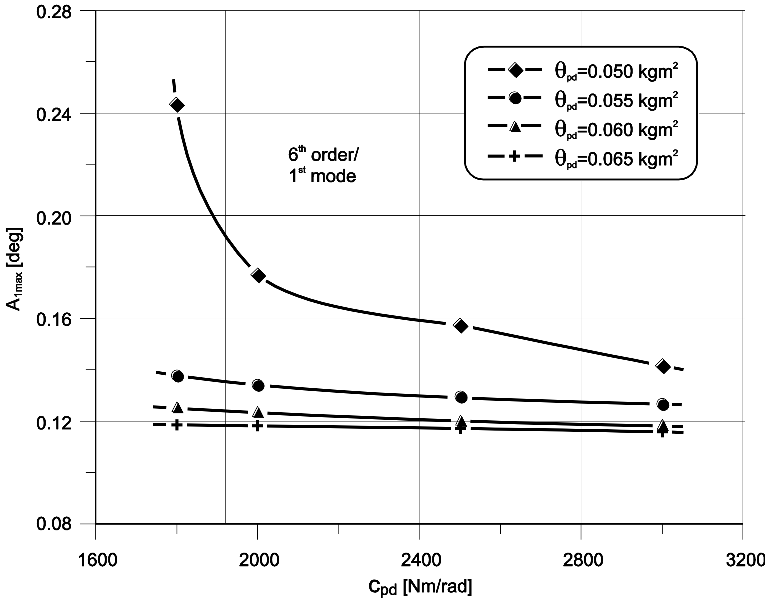

Figure 8 The diagram of maximum amplitudes of twist angle of the engine pulley as a function of the torsional spring stiffness of the damper and moment of inertia of the inertial mass as a parameter

Figures 7 and 8 indicate that minimum reasonable values of the maximum amplitude of the twist angle of the engine pulley of the particular engine with a dynamic damper are obtained for moment of inertia of masses (θ_{pd}) greater then 0,055 kgm^2 and for torsional stiffness of the springs of the damper (CPD) greater than 2000 Nm/rad. These results show that it is possible to incorporate a dynamic damper into a dimensionally realistic crankshaft counterweight to gain the same or even better effectiveness than using a viscoelastic torsional vibration damper.

A comparison example of values of amplitudes of twist angle of the engine pulley with a viscoelastic and dynamic torsional vibration damper is shown in Figure 9.

In this particular example a dynamic damper with characteristics θ_{pd}=0,060 kgm^2 and c_{pd}=2500 Nm/rad is taken. At the same diagram (Figure 9) amplitudes of the twist angle of the engine pulley (A_1) for the same engine without a torsional vibration damper is given.

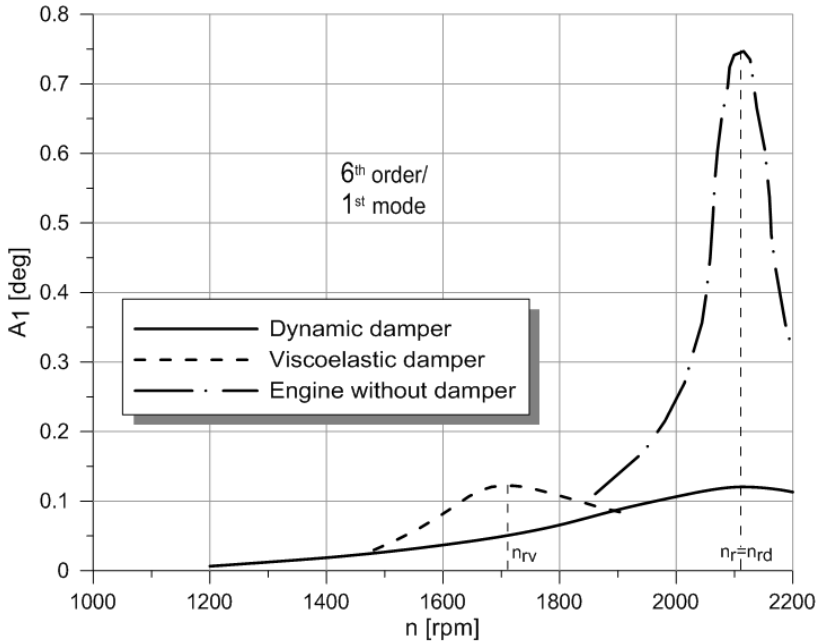

Figure 9 The comparative diagram of maximum amplitude of twist angle of the engine pulley (A1) for an engine with a viscoelastic and dynamic torsional vibration damper

Comparative results in Figure 9 show that it is possible to get the same or even less amplitudes of twist angle due to torsional vibrations using a dynamic torsional vibration damper as using a viscoelastic torsional vibration damper. It should also be noted that in addition to a satisfactory level of absolute values of the maximum twist angles due to torsional vibration in the application of both types of dampers, the maximum value of the amplitudes are at different engine speeds. Applying the viscoelastic torsional vibration damper on the engine the resonant vibration mode moves to the lower speeds of the engine (n_{rv}~1710 ^{o}/min) relative to the resonant vibration mode of the engine without a damper (n_r~2120 ^{o}/min). Unlike the viscoelastic damper, which significantly shifts the resonant mode, the dynamic damper practically does not affect the engine resonant mode (n_{rd}~n_r). This analysis is given only for the 6th excitationorder and I mode of vibration, which are assessed as the most important parameters to check the operation safety of the crankshaft of a six-cylinder engine. Similar conclusions are obtained for other major (9th; 12th; ...) and strong (4.5th, 7.5th, ...) excitation orders for a six-cylinder engine, noting that in this case the most dangerous 6th order, considering the value of the amplitude of the twist angle of the crankshaft due to torsional vibration, is in the operation range of the engine.

The foregoing discussion and analysis can be done for any engine type, but it should be taken into account the proper definition of the parameters of the dynamic torsional vibration damper, with respect of the major excitation orders according to the number of cylinders of the engine.

5. CONCLUSION

Through the analysis of parameters of a torsional vibration system of an IC engine with two different torsional vibration dampers it can be concluded:

- The same or even better damping effects of amplitudes of crankshaft twist angel due to torsional vibration can be obtained with incorporating a dynamic damper in the crankshaft counterweight instead of using so-called conventional viscoelastic dampers.
- The engine resonant mode doesn't change with using the dynamic torsional vibration damper in relation to the resonant mode of the engine without a torsional vibration damper. In addition, conventional viscoelastic torsional vibration dampers significantly reduce the value of the resonant engine speed relative to the resonant mode of the engine without the damper.
- Bearing in mind the real design and dimensions of the crankshaft with counterweights for medium speed engines for road vehicles, there is enough space for incorporating a dynamic torsional vibration damper, with an optimum moment of inertia of the inertial mass and stiffness of the spring damper, into one counterweight. In the case of smaller engines and insufficient space to incorporate the dynamic damper with optimal parameters, two counterweight on the same crankshaft knee can be used.

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