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PRELIMINARY SELECTION OF BASIC PARAMETERS OF DIFFERENT TORSIONAL VIBRATION DAMPERS INTENDED FOR USE IN MEDIUM-SPEED DIESEL ENGINES

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Summary

In the development and application of highly dynamic mechanical systems, major problems can arise from usually unwanted accompanying processes, such as torsional vibrations. The internal combustion engine is a typical dynamic system with a high probability of fracture of a system part due to the effects of torsional vibrations. In engineering practice, the IC engine crankshaft fracture due to torsional vibrations is prevented by using additional devices that allow the transfer of critical vibration modes of the crankshaft out of the IC engine operating range, or devices damping the resulting twist angle amplitudes.

This paper presents a possible approach to defining parameters of torsional oscillatory systems of IC engines needed for a preliminary selection of basic parameters of various types of torsional vibration dampers, such as the elastic damper, the balance weight damper and the dual mass flywheel. The proposed physical and mathematical models and methods of defining the input parameters were compared with experimental results.

Key words: *torsional vibration damper, IC engine, balance weight damper, dual mass flywheel*

1. Introduction

Because of their specific way of work (cyclic energy machine), internal combustion (IC) engines produce a variable torque at the crankshaft at all speeds. Variation of the torque is influenced by the engine design (number and position of cylinders) and the maximum cylinder pressure. Taking the variation of the torque at the crankshaft [2, 3, 10, 11] and different crankshaft stiffness at different cross-sections [2, 6, 12] into consideration, the appearance of torsional vibration is inevitable. Besides to torsional vibration, the crankshaft is somewhat less exposed to flex and axial vibrations. If some resonant regimes are in the IC engine operating range, torsional vibrations of the crankshaft pose a risk due to fatigue and possible fracture of the crankshaft. The increase in specific IC engine energy parameters (power, torque, in-cylinder pressure, etc.) due to the reduction in engine dimensions, or the number of cylinders, produces greater unevenness of the torque at the engine crankshaft [9, 11]. The analysis of torsional vibrations requires close attention, especially when the IC engine works at critical resonant speeds. The emergence of critical resonant vibrations in the engine operating range can be overcome in one of the two ways:

– by shifting the resonant regimes outside the IC engine operating range,
– by introducing special devices to reduce the vibration amplitude in the resonant regimes.

The first mentioned method requires increasing the crankshaft stiffness while reducing the moment of inertia of all masses of the entire piston and connecting rod assembly. From the structural point of view, this is practically impossible. The second method, the introduction of special devices called torsional vibration dampers (TVDs) is realistic and extensively used. Torsional vibration dampers used in IC engines differ in structure and working principles [2, 13]. For road motor vehicles, viscous and elastic (rubber) dampers were irreplaceable for a long period of time. They were more acceptable because of their simplicity, cost and easier maintenance.

Recently, there has been a trend of replacing the TVD as a separate unit with different variants of the dynamic absorber built in into the existing engine parts. So, torsional vibration absorbers are today already extensively developed and used as incorporated into the engine flywheel or the crankshaft balance weights [5, 8, 9].

This paper presents one possible method of preliminary selection (evaluation) of damper parameters on the basis of dynamic absorbers, based on the modelling of torsional vibration processes. The final solutions must be experimentally verified.

2. Torsional vibration dampers for road vehicle engines

By far the most commonly used TVD for road vehicle engines is the so-called elastic damper with a rubber elastic element. Such a damper is shown in Figure 1a. The damper is mounted on the crankshaft at the opposite side of the engine flywheel, where the largest twist angle amplitude occurs due to crankshaft vibrations (Fig. 2a).

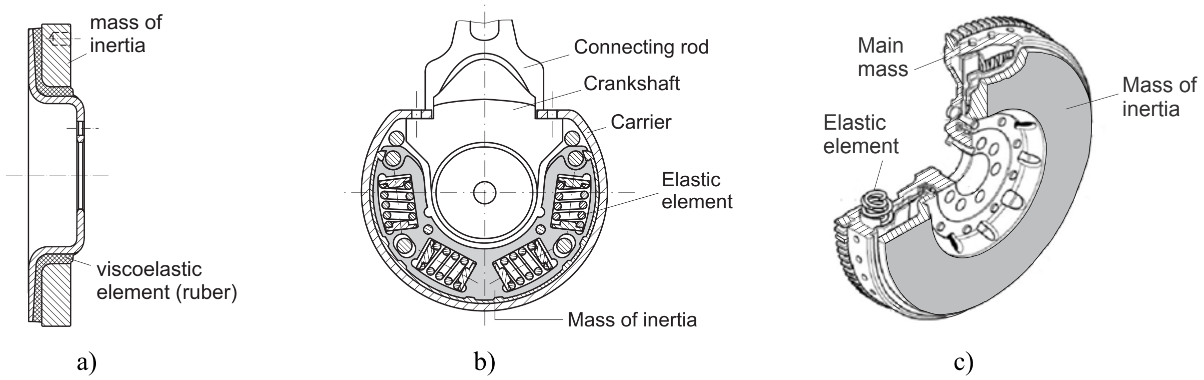

Fig. 1 Three different TVDs for road vehicles IC engines

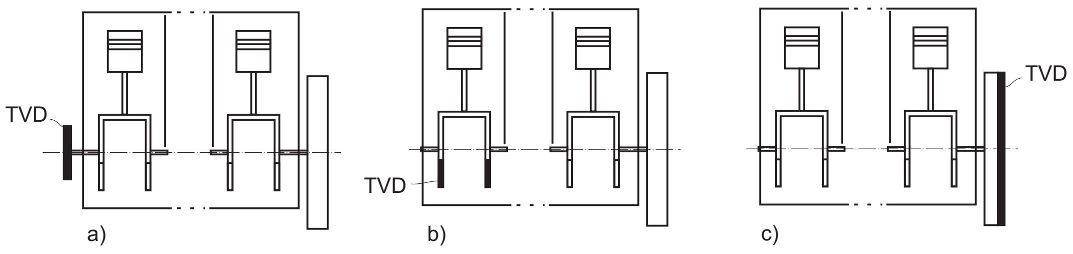

Fig. 2 TVD mounting places in IC engines

Elastic damper with rubber as an elastic element is, in principle, a separate unit mounted on one end of the engine crankshaft. The parameters determining the damper are highly nonlinear (stiffness, damping) and depend on the type of elastic material, its dimensions, the twist angle and the temperature of elastic material, which varies depending on the engine operating mode. The main disadvantage of the elastic damper is its reliability in operation because the elastic element sticks to the metal (inertial mass of the hub). In order to increase reliability and eliminate the TVD as a separate element, vibration dampers based on the principle of dynamic absorbers have been recently developed and incorporated into the existing structural elements of IC engines. Thus the balance weight damper is successfully developed [5] and built into the balance weight of the first crank web, see Figure 2b). If necessary, the balance weight damper can be mounted in more than one crankshaft balance weight. Similarly, the damper mounted in the IC engine flywheel [1, 8, 9], the so-called dual mass flywheel (DMF), shown in Figure 1c), is developed and shown in the IC engine presented in Figure 2c). The DMF has multiple functions: it performs vibration amplitude damping at critical engine speeds, balances the crankshaft angular velocity, provides comfort to the driver when starting the vehicle from standstill, supports the work of vehicle transmission and has impact on reducing engine noise. Because of the multiple functions, and for the purpose of achieving the desired effects, the DMF has a number of different structural designs (swing masses, different damping performances, etc.) [8, 9], that will not be separately analyzed here. Here, the DMF is seen as a torsional vibration damping element.

3. Models for calculating amplitudes of torsional vibration system

To analyze and study the TVD influence parameters, it is common to use the so-called physical models of linear torsional vibration systems [2, 7, 10], where all the rotating masses are reduced to the engine crankshaft rotation axis, under the condition of equal potential and kinetic energy of the real system and the physical model.

Applying this principle, Figure 3 shows physical models of torsional vibration systems for engines without a TVD (Fig. 3 a)), with an elastic TVD (Fig. 3b), with a balance weight damper mounted in the balance weight of the first crank web nearest to the first engine cylinder (Fig. 3c) and with a DMF (Fig. 3d).

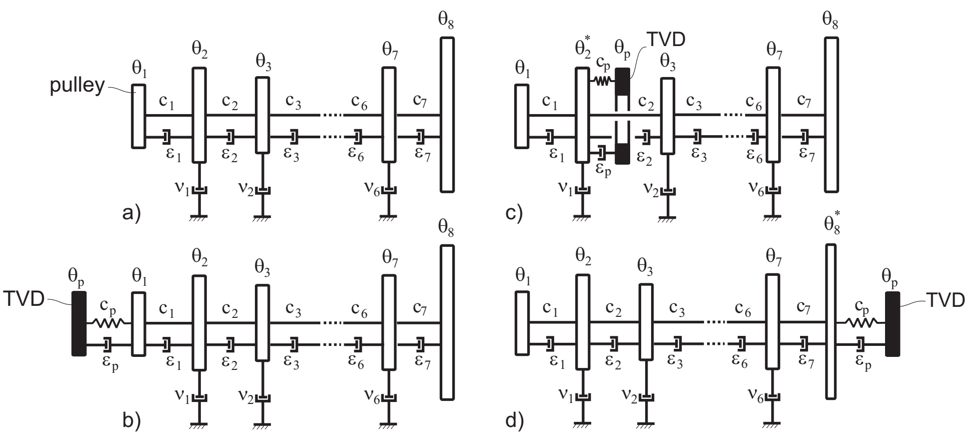

Fig. 3 Physical models of torsional vibration systems with different dampers

The movement of discs in the presented torsional vibration systems can be described mathematically by using the second order Lagrange equations [2, 3]. By simplifying Lagrange's equations, the system of equations describing the vibration of an individual disk around the equilibrium position (Figure 3), can be expressed as:

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

where:

[I] – the moment of inertia of the mass matrix

[K] – the internal and external damping matrix

[C] – the stiffness matrix

{T(t)} – the excitation matrix

$\{\vartheta\}, \{\dot{\vartheta}\}, \{\ddot{\vartheta}\}$ - the twist angle, speed and acceleration matrix.

The general system of vibration equations (1) can be solved in several ways. Common methods are: the central difference method [2] and the method of the harmonic analysis by introducing excitation (*T(t)*) in the form of Fourier's order, where solutions are obtained in the form of harmonic functions [2, 3]. In engineering practice, the most acceptable method is still the last mentioned method, which is used in this paper too. Particularly sensitive input data in this model are determined from the literature sources as follows:

- the moments of inertia of masses (θ_i) of complex elements (connecting rod, crank web) are determined both experimentally and numerically [2, 3, 7, 10],
- the crankshaft stiffness (c_i), which is very sensitive because of the specific support of crank pins at sliding bearings, is calculated by applying the finite element method and specific boundary conditions as described in [12],
- the excitation amplitudes (*T(t)*), for each excitation order, are calculated by using the model described in [10], with corrections of some constants of the model, based on experimental results,
- The internal (ε_i) and external (υ_i) damping coefficients are selected for a particular engine on the basis of the recommendations given in [2, 3, 14], with some corrections based on experimental results.

The parameters for damping and stiffness of the elastic damper elements, shown in Figure 1, are not given within the above mentioned input data for the mathematical model (1).

To successfully implement the above model describing the torsional vibration movement of the engine crankshaft with its elements, two physical engine models, shown in Figures 3a and 3b, are observed. For them, estimates of the crankshaft twist angles are made, and the results are compared with experimental results. The IC engine is a four-stroke, 6-cylinder turbocharged diesel engine. Basic data about the engine are given in Table 1.

Table 1 Basic data about the IC engine

Number of cylinders	6
Bore	125 mm
Stroke	150 mm
Engine power / engine speed	184 kW/2100 rpm
Maximum engine torque	890 Nm
Firing order	1 – 5 – 3 – 6 – 2 – 4
Moment of inertia of masses, Fig. 3a	θ_1 = 0,0277 kg m^2, $\theta_2 = \theta_4 = \theta_5 = \theta_7$=0,147 kgm^2, $\theta_3 = \theta_6$ = 0,0835 kg m^2, θ_8 = 1,87 kg m^2
Stiffness	$c_1 = 4{,}32\cdot10^6$ Nm/rad, $c_2 = ... = c_7 = 2{,}81\cdot10^6$ Nm/rad, $c_8 = 4{,}12\cdot10^6$ Nm/rad,
Coefficient of external damping	$\upsilon_1 = ... = \upsilon_6$ = 12 Nms/rad
Coefficient of internal damping	$\varepsilon_1, \varepsilon_2, ..., \varepsilon_7$ calculated according to [2]

For the engine model shown in Figure 2a, calculations of torsional vibration amplitudes for the first mode and for the main excitation orders 6, 7.5, 9 and 12 are made. The calculation results are shown in Figure 4.

Along with these results, experimental results are presented with an excellent agreement between experimental and computational results. The former are obtained by the method described in [4, 11]. The agreement of computational and experimental results indicates that the model is properly chosen and that the input data are plausible. Given that the most sensitive element in the modelling process is the TVD, the confirmation of the chosen model for the calculation of torsional vibrations is made on the basis of the experimental results obtained by measuring the engine with an elastic TVD, which has long been in use. The presented model of the engine with a TVD with a rubber elastic element is shown in Figure 3b. The parameters of the damper (ε_p and c_p) are taken as a function of the relative twist angle between the masses (discs) (θ_p and θ_1) and the rubber temperature [3]. The results are shown in Figure 5, where one can see a significant shift in the resonant operation mode of the engine for the first mode of torsional vibration amplitudes and the excitation orders 4, 5 and 6, compared to the corresponding amplitudes for the engine without a TVD at resonant operating mode.

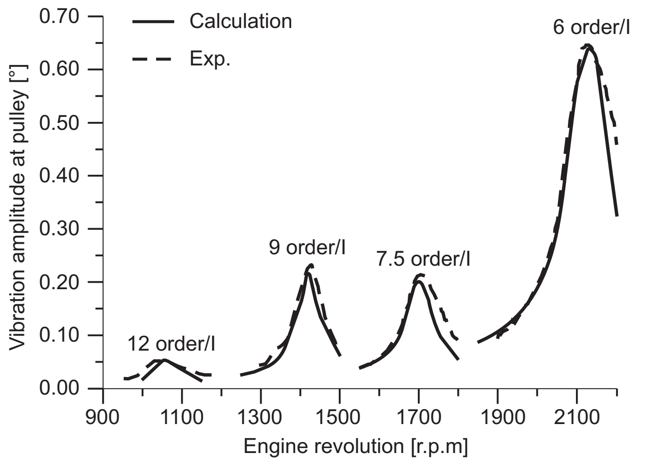

Fig. 4 Diagram of calculated and experimentally obtained twist angles of the torsional vibration amplitudes at the pulley for the engine without a TVD

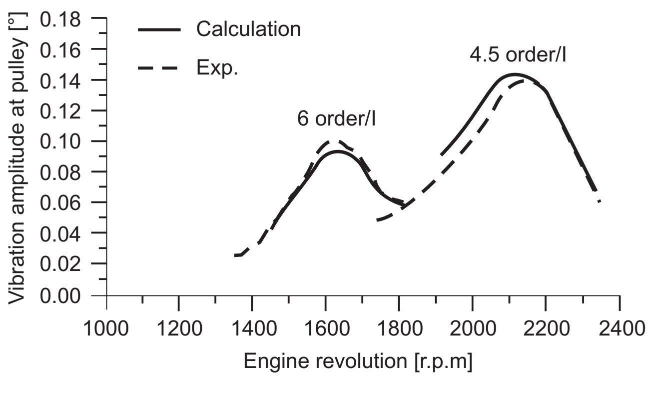

Fig. 5 Diagram of calculated and experimentally obtained twist angles of the torsional vibration amplitudes at the pulley for the engine with.an elastic TVD

The values of the torsional vibration amplitude at the end of the engine crankshaft opposite to the flywheel are below 0.15°. The authors' experience is that these values are, in terms of mechanical load acting on the IC engine crankshaft, within permitted limits. Considering the nonlinear characteristics of rubber in particular, the agreement between the calculated and experimental results is regarded as satisfactory, which means that the chosen model of the torsional vibration system is acceptable for the analysis of torsional vibration parameters of the system with any damper.

4. Analysis of results

4.1. Balance weight damper

The main part of the research deals with the choice of parameters of the torsional vibration damper to be mounted on the engine crankshaft. The damper shown in Figure 1b represents a kind of dynamic absorber built in the balance weight and mounted on the first crank web of the engine crankshaft opposite to the flywheel. The place of mounting the TVD is shown in Figure 2b. The damper can be mounted in one or two balance weights, depending on the required inertial mass of the damper. The main condition to be fulfilled is the equality of mass moments of inertia of the disks $\theta_2=\theta_2^*+\theta_p$, so that the balance of inertial forces and corresponding moments of the complete engine can be maintained. This solution has fully eliminated the TVD as a separate entity, and the balance weight damper should provide similar damping effects. The basic parameters of the new damper are: moment of inertia of the damper (θ_p), torsional stiffness (c_p) obtained from axial stiffness of springs distributed along the balance weight perimeter and internal damping defined by the internal damping coefficient (ε_p). The internal damping coefficient (ε_p) depends on the sliding bearings in which the inertial masses are moving (Fig. 1b). Quantities that define the previously described damper in its character are:

$$c_p = \text{const}, \quad \theta_p = \text{const}, \quad \varepsilon_p = f(\dot{\vartheta}) \tag{2}$$

When selecting the damper parameters, the model uses relative quantities c_p/c_2 and θ_p/θ_2. The internal damping coefficient (ε_p), which depends on the type of bearings and complete sliding surface, is considered a constant value. Its impact is commented on later.

The physical model of the torsional vibration system with a balance weight damper is shown in Figure 3c). It is necessary to modify the mathematical model described by the general equation (1) so it can be used for calculations with the balance weight damper. So, the vibration equations of mass (θ_2) as they are used in the conventional engine model (Fig. 3a)

$$\theta_2\ddot{\vartheta}_2 + \nu_1\dot{\vartheta}_2 + \varepsilon_1(\dot{\vartheta}_2 - \dot{\vartheta}_1) + \varepsilon_2(\dot{\vartheta}_2 - \dot{\vartheta}_3) + c_1(\vartheta_2 - \vartheta_1) + c_2(\vartheta_2 - \vartheta_3) = T_2, \tag{3}$$

are replaced for the model with a balance weight damper (Fig. 3c) by two equations of the following form:

$$\begin{aligned} &\theta_2^*\ddot{\vartheta}_2 + \nu_1\dot{\vartheta}_2 + \varepsilon_1(\dot{\vartheta}_2 - \dot{\vartheta}_1) + \varepsilon_2(\dot{\vartheta}_2 - \dot{\vartheta}_3) + \varepsilon_p(\dot{\vartheta}_2 - \dot{\vartheta}_p) + c_1(\vartheta_2 - \vartheta_1) + \\ &+ c_2(\vartheta_2 - \vartheta_3) + c_p(\vartheta_2 - \vartheta_p) = T_2, \end{aligned} \tag{3a}$$

$$\theta_p\ddot{\vartheta}_p + \varepsilon_p(\dot{\vartheta}_p - \dot{\vartheta}_2) + c_p(\vartheta_p - \vartheta_2) = 0. \tag{3b}$$

The mathematical equations of the modified mathematical model are solved with varying relationships c_p/c_2=0.02÷0.1 and θ_p/θ_2^*=0.15÷0.55, which is, from the design point of view, a possibly realistic change of the parameters c_p and θ_p. The value of the internal damping coefficient is kept constant, $\varepsilon_p = 20$ Nms / rad.

The results of the maximum twist angle of the torsional vibration amplitudes for the first mode and 6th order ($A_{6/I}$) are shown in Figure 6, with Figure 6a showing the dependence of $A_{6/I}=f\ (c_p/c_2)$ on θ_p/θ_2^* as a parameter, while Figure 6b shows the dependence of $A_{6/I}=f\ (\theta_p/\theta_2^*)$ on c_p/c_2 as a parameter.

As a comparative analysis of the results of the twist angle amplitudes at the first mode and 6th order for the pulley of the engine without a damper (Fig. 4) and with an elastic damper (Fig. 5), Figure 6 clearly shows that with certain numbers of combinations of the relations c_p/c_2 and θ_p/θ_2^* the same amplitude effect as in case of the engine with a conventional elastic TVD ($A_{6/I} \approx 0.1°$) can be achieved. This applies to the amplitude $A_{4.5/I}$ too. The position of the resonant vibration conditions both for the first mode and 6th excitation order is different from the one presented in Figure 5 and it significantly depends on the relationships c_p/c_2 and θ_p/θ_2^*.

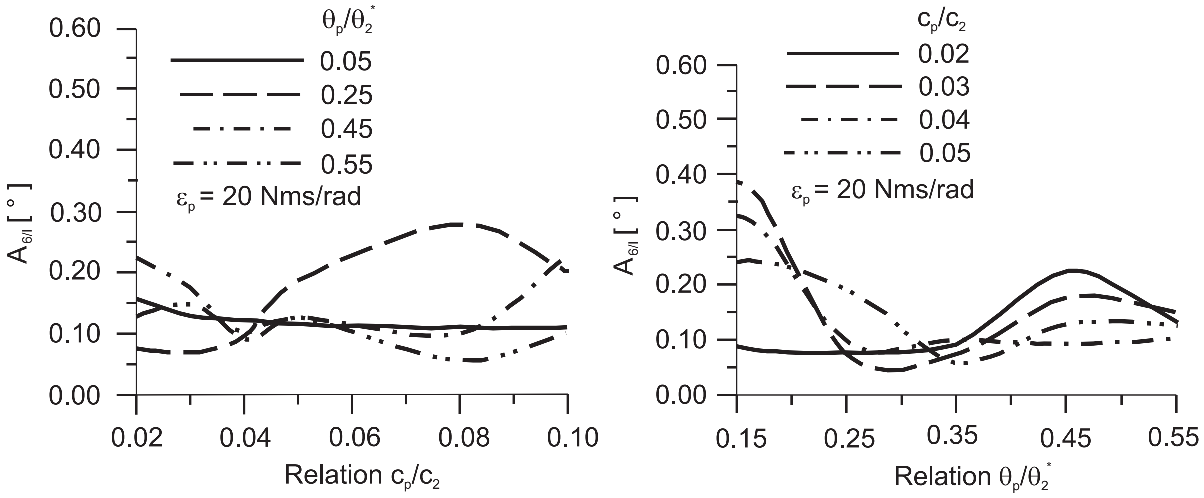

Fig. 6 Diagram of twist angles of the torsional vibration amplitudes ($A_{6/I}$) at the engine pulley (θ_1) depending on the relative stiffness and relative moment of inertia of the inertial mass of the balance weight damper

The effect of the relationship c_p/c_2 on the position of the resonant vibration conditions is shown in Figure 7. The analysis also showed that the internal damping coefficient (ε_p) of the damper minimally affects the position of the resonant vibration.

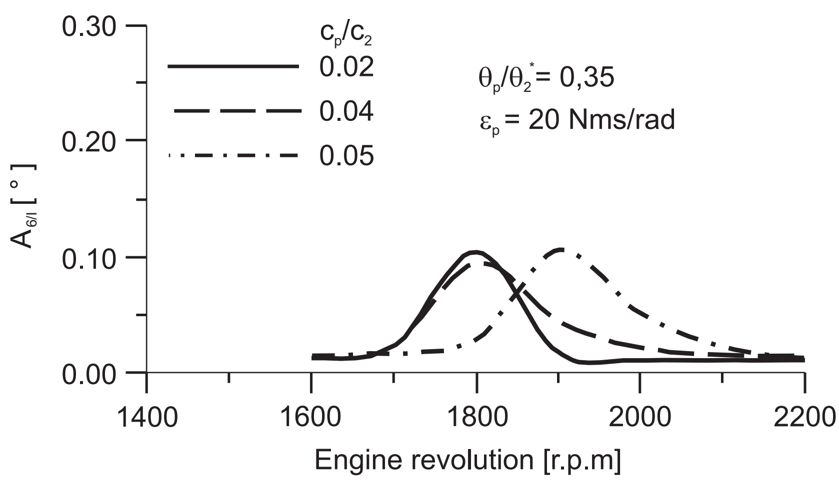

Fig. 7 Diagram of twist angles of the torsional vibration amplitudes ($A_{6/I}$) at the motor pulley (θ_I)

The influence of the damper damping effect (ε_p) on the vibration amplitude ($A_{6/I}$) is positive, but it is not specifically analyzed here.

4.2. Dual mass flywheel

The design of a dual mass flywheel is shown in Figure 1c. In addition to torsional vibration amplitude damping, the DMF achieves other effects too that are suitable for the transmission of the vehicle and driver comfort. This paper analyzes the role of the DMF in the context of the torsional vibration amplitude damping, primarily at the engine pulley (θ_1). Figure 2c shows the mounting place of the DMF on the IC engine crankshaft. The physical model of the torsional vibration system with a DMF is given in Figure 3d. The mathematical model is the same as for the physical models shown in Figures 3a and 3b. The requirement to analyze the influence of stiffness (c_p) and the moment of inertia (θ_p) of the DMF is to respect the equality of the moments of inertia of one mass (θ_8) and the DMF (θ_8^* and θ_p), $\theta_8 = \theta_8^* + \theta_p$. In this way, the degree of rotation unevenness of the engine crankshaft is kept. The values of stiffness (c_p) and the inertial mass moment of inertia (θ_p) are varied as relative quantities c_p/c_2=0.1÷0.25 and θ_p/θ_8^*=0.2÷0.7. In this analysis, the internal damping coefficient (ε_p), which depends on the design of the sliding elements of the springs and the inertial part of the DMF, is kept constant. The calculation results of the torsional vibration twist angle amplitudes for the first mode and 6th excitation order ($A_{6/I}$) at the engine pulley are shown in Figure 8.

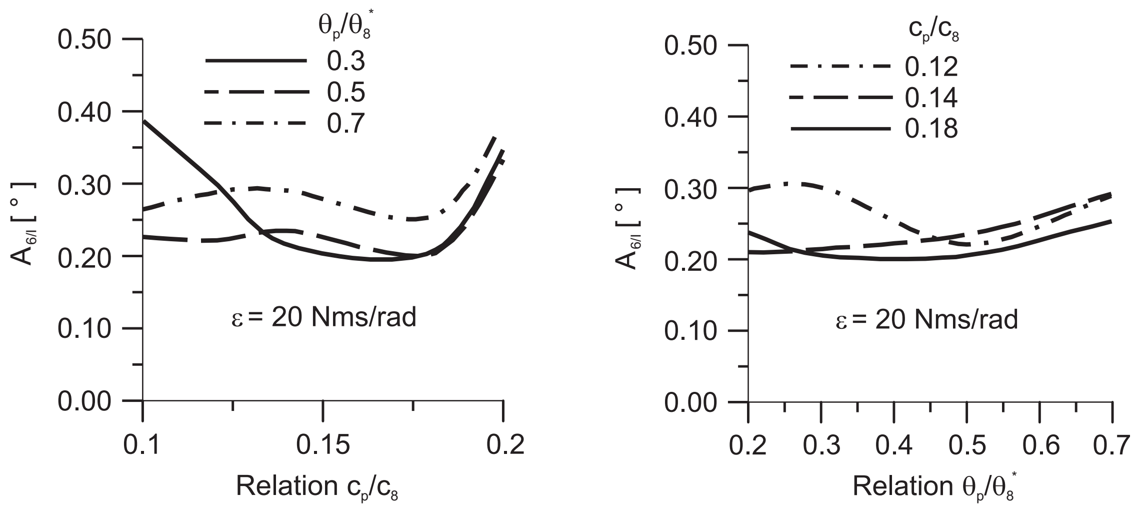

Fig. 8 Diagram of the torsional vibration twist angle amplitudes ($A_{6/I}$) at the engine pulley

In Figure 8a, the dependence of $A_{6/I}=f\,(c_p/c_2)$ on the ratio θ_p/θ_8^* as a parameter is shown, while in Figure 8b the dependence of $A_{6/I}=f\,(\theta_p/\theta_8^*)$ on the ratio c_p/c_2 as a parameter is shown.

The results shown in Figure 8 do not have the effect of the torsional vibration twist angle damping as this is the case with using the TVD shown in Figures 1a and 1b. This is expected due to the mounting position of the DMF (Fig. 2c) and its multiple functions. The increase in the torsional vibration amplitude damping effect can be achieved by introducing greater internal damping (ε_p) and adding dangle masses.

5. Conclusions

The paper presents a possible approach to a preliminary selection of basic parameters of different types of conventional and new designs of torsional vibration dampers, and the effect of the parameters selected in this way on the critical torsional vibration twist angle amplitude at the pulley of medium speed diesel engine for road vehicles.

The choice of models (physical and mathematical) and the proposed methods of defining the input data on the torsional vibration system are convenient for engineering practice considering good agreement between the calculated and experimental results for the critical amplitude of the twist angles. These results are compared between engines without any significant internal damping and those showing significant internal damping (engine with an elastic TVD).

The calculation results for the balance weight torsional vibration damper show that it can fully replace the elastic torsional vibration damper. Taking the design possibilities into account and based on the torsional vibration twist angle amplitudes ($A_{6/I}$), the inertial mass moment of inertia (θ_p) and the corresponding spring stiffness (c_p) can be chosen. Depending on the selected parameters (θ_p and c_p), balance weight dampers are mounted on one or two crank webs opposite to the engine flywheel.

The effect of damping the critical twist angle amplitude in the case of using a DMF is slightly smaller than in cases in which other dampers presented in this paper are used. In this case, however, the twist angle amplitudes of the engine pulley are reduced by more than 3 times. If additional structural details are introduced, the DMF can successfully replace the conventional torsional vibration damper.

The presented method of choosing torsional vibration damper parameters is regarded as satisfactory for the so-called preliminary selection, especially if engine parameters (moment of inertia, stiffness, damping and excitation) are not precisely defined. The final TVD design has to be confirmed experimentally.

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