CONTRIBUTION IN DEFINING PARAMETERS OF THE EQUIVALENT SYSTEM OF TORSIONAL VIBRATIONS WITH THE SIGNIFICANT INTERNAL DAMPING ELEMENTS

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INTRODUCTION

Internal combustion engine is a cyclic machine that develops its power from associated chemical energy of liquid or gaseous fuel that is released through the combustion process. Because of the cyclic process, the unevenness of fuels quality and the stochastic combustion process that takes place in a very short period of time in the cylinder, internal combustion engine is subjected to uneven mechanical loading. To obtain a more uniform speed of rotation of IC engine it is common to set additionally a flywheel. But even so, there is a variation of the current effective torque which represents the excitation of torsional vibrations of the crankshaft. The occurrence of vibrations is undesirable and it is because of the additional dynamic load of the crankshaft that can lead to its fracture, which is especially pronounced in more powerful IC engines.

Since the torsional vibration system of IC engines, including its additional elements in the form of pumps, additional drives, etc., is very complex, it is very difficult to describe it mathematically in exact from, so it is necessary to set a model that will be a dynamical equivalent to the actual system. The parameters identification phase of the equivalent IC engines torsional vibration system in practice faces a number of difficulties that are associated with certain unknowns. With the aim to mathematically solve the problem it is common in everyday engineering practice to use approximations, which give good enough agreement with experimental results. The fact that a lot of different empirical and half-empirical approaches for parameters identification of the IC engine torsional vibration system are existing, indicates the complexity. Knowing the critical working areas of the IC engine, devices and systems can be designed that enable shifting of critical regime from the working area of the system, or, which is in practice more often, which would provide damping of critical vibration amplitudes.

There is a large number of different solutions for torsional vibration dampers taking benefit of damping properties of different materials. Commonly for the IC engine applications viscous and viscous-elastic dampers are in use. According to their simple design, reliability, diversity and economy in production the viscous – elastic dampers have priority.

Viscous-elastic dampers use, such as the word itself says the viscous-elastic properties of polymers (rubber) for damping torsional vibrations. However, the properties of polymers are not constant and vary greatly with temperature, frequency and vibration amplitudes of

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the system. The existence of such an element in torsional vibration system, despite all its undoubted benefits, brings unknowns that have to be defined, especially in very responsible systems.

The following gives an example of determining the equivalent torsional vibration system with special emphasis on possible ways of presenting system damping as a highly nonlinear component.

1. EQUIVALENT TORSIONAL VIBRATION SYSTEM OF THE IC ENGINE

Due to limited knowledge and often complicated mathematical apparatus that follows original presentation of the torsional vibration system, it is necessary for practical application to develop equivalent systems that are simpler and which also reflect with sufficient accuracy events in the real system.

Often in use is a in line equivalent system where all elements of the vibration system are reduced to a single axis, of course with the condition of equality of potential and kinetic energy of the real and equivalent systems. The in-line equivalent torsional vibration system with torsional vibration damper for a 6 cylinder in-line, water cooled diesel engine is shown in the figure 1. Moments of inertia (Θ) for the cylinders are different because of different contra weights at throws of crankshaft.

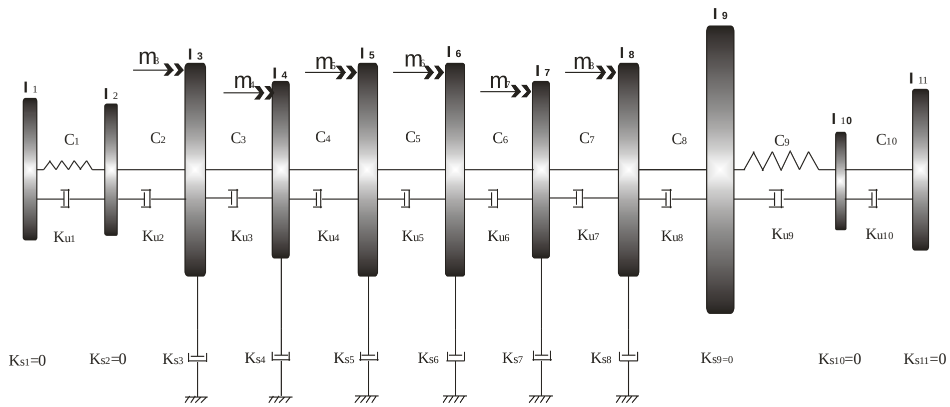

Figure 1: *Equivalent torsional vibration system of a IC engine*

1.1 Parameter determination of the torsional vibration system

After adoption of the equivalent torsional vibration system it is necessary to define its parameters. Determination of a set of parameters of the equivalent torsional vibration systems such as moments of inertia of concentrated masses (Θ), reduced length (l) and torsional stiffness of shaft parts (c), is extensively described in many literature sources, especially in [2, 3, 4].

Furthermore, it is necessary to determine the excitation torque (M) by IC engine cylinders. The most reliable estimation of torque is through measurement, more exactly through measurement of indicated pressure in cylinder of IC engine for the given case. Having in mind great demands for carrying out such measurements, different half empiric methods for estimation of the uneven part of the excitation torque, presented in form of Fourier series, are in use. This methods differs one from another in number of influence parameters taking in to account [6].

For a complete definition of a torsional vibration system it is necessary to determine the parameters that define the damping in the system.
There are different approaches to describe the damping, but basically they are:

- Describing the damping of the entire system based on the known properties of the system or structure, without determination of damping for individual (local) structural elements;
- Describing the damping of the system based on the known properties of certain structural elements of the system; (describing the damping of the system components and based on them damping of the system);
- Describing the damping on hand of describing behaviour of materials using the laws of continuum mechanics (constitutive equations for materials).

In this paper total system damping is separated to internal (K_u) and external (K_s) damping. Internal damping represents the material damping, that highly depends on geometric shape, material properties and vibration frequency. The internal damping (K_u)is estimated on base of using empiric relation presented in [9]:

$$K_u = \frac{c\,\mathrm{M}_1^{1/2}}{a\;\omega\;\sqrt{\left(R_0^3 - R_i^3\right)\dfrac{\pi}{2}}} \qquad (1)$$

where are: c- stiffness, M_1- excitation torque , ω- circular vibration frequency, R_0, R_i- outer and inner half diameter of shaft, a - coefficient that depends on kind of material.

External damping represents resistance of moving parts of the IC engine (pistons, connector rods, crankshaft, etc.) and other auxiliary engine devices and is very difficult for explicit determination. A common approach is to use a combination of empirical recommendations, such as [3, 8, 9], and through confirmation of the selected damping on the basis of calculation the vibration amplitude, which again can be experimentally determined.

A particular problem is the introduction of the contribution of the torsional vibration damper in vibration model. The problem occurs primarily because of the nonlinearity of the damping and its dependence on a number of parameters that are changeable with the IC engine regimes. In figure 2, for purposes of illustration, are shown the changes of stiffness and damping of torsional vibration dampers depending on the relative amplitude of vibration [1], for a concrete torsional vibration damper of an engine (figure 1). Because of great number of quantities that can be encountered in the literature as a measure to define the internal damping of viscous-elastic dampers, within table 1 an overview of commonly used terms for their definition and relationships between these quantities is given.

Table 1. *Correlation between damping quantities*

	K_u	δ	ξ	D	Λ
Damping coefficient K_u	K_u	$2I\delta$	$2\,I\omega_0\xi$	$I\omega_0 D$	$\frac{I\,\omega_d\Lambda}{\pi}$
Suppression coefficient δ	$\frac{K_u}{2I}$	δ	$\omega_0\xi$	$\omega_0\frac{D}{2}$	$\frac{\omega_d\Lambda}{2\pi}$
Damping of Lehr ξ	$\frac{K_u}{2\sqrt{cI}}=\frac{K_u}{2I\,\omega_0}$	$\frac{\delta}{\omega_0}$	ξ	$\frac{D}{2}$	$\frac{\Lambda}{2\pi}\frac{\omega_d}{\omega_0}$
Loss factor D	$\frac{K_u}{\sqrt{cI}}=\frac{K_u}{I\,\omega_0}$	$\frac{2\delta}{\omega_0}$	2ξ	D	$\frac{\Lambda}{\pi}\frac{\omega_d}{\omega_0}$

Table 1 *Correlation between damping quantities (continues from previous page)*

	Ku	δ	ξ	D	Λ
Logarithmic decrement Λ	$\frac{\pi K_u}{I\,\omega_d}$	$\frac{2\pi\delta}{\omega_d}$	$\frac{2\pi\xi}{\sqrt{1-\xi^2}}$	$\frac{\pi D}{\sqrt{1-\frac{D^2}{4}}}$	Λ
Band wide of frequency response $\Delta\eta=\frac{\Delta\Omega}{\omega_0}$	$\frac{K_u}{\sqrt{cI}}=\frac{K_u}{I\,\omega_0}$	$\Delta f=\frac{\delta}{\pi}$ $\Delta\Omega=2\delta$	2ξ	D	$\frac{\Lambda}{\pi}\frac{\omega_d}{\omega_0}$
Dynamic amplification factor (η = 1) M_r	$\frac{\sqrt{cI}}{K}$	$\frac{\omega_0}{2\delta}$	$\frac{1}{2\xi}$	$\frac{1}{D}$	$\frac{\pi}{\Lambda}\frac{\omega_0}{\omega_d}$
Damping ratio $\Psi=\frac{W_D}{W_{pot}}$	$\frac{2\pi\Omega K_u}{c}$	$\frac{4\pi I\,\Omega\delta}{c}$	$4\pi\eta\xi$	$2\pi\eta D$	$2\eta\Lambda\frac{\omega_d}{\omega_0}$
tg ε_a Loss angle	$\frac{\Omega K_u}{c}$	$\frac{2I\,\Omega\delta}{c}$	$2\eta\xi$	ηD	$\frac{\eta\Lambda}{\pi}\frac{\omega_d}{\omega_0}$

In the table 1 significance of tag is as follows:

$\omega_0=\sqrt{\frac{c}{I}}$ – Natural frequency of the system without damping

$\omega_d = \omega_0 \sqrt{1-\xi^2}$ - Natural frequency of the system with damping

Ω - Frequency of excitation

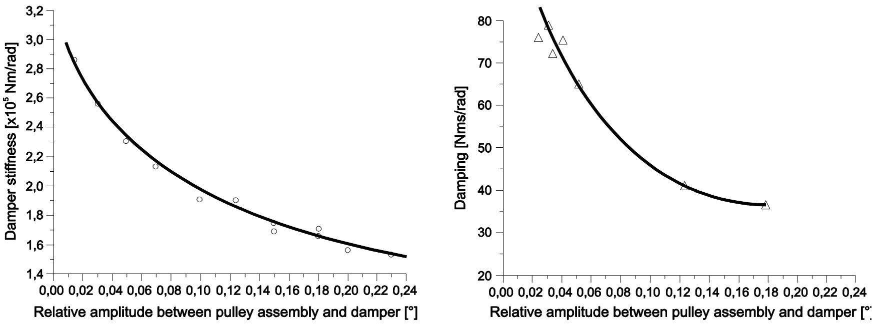

Figure 2: *Change of stiffness and damping of torsional vibration damper, for a concrete example, as function of relative vibration amplitude*

In the mass production of torsional vibration dampers for IC engines, as a measure of quality assurance, measuring of characteristic quantities of the damper is done. Common are measurements for estimation of the dampers nominal frequencies at prescribed temperature conditions. This measurement should confirm that the previously established parameters, like dynamic stiffness (c_d) and loss angle ($tg\varepsilon_a$), after production of multiple dampers are kept at the required tolerances.

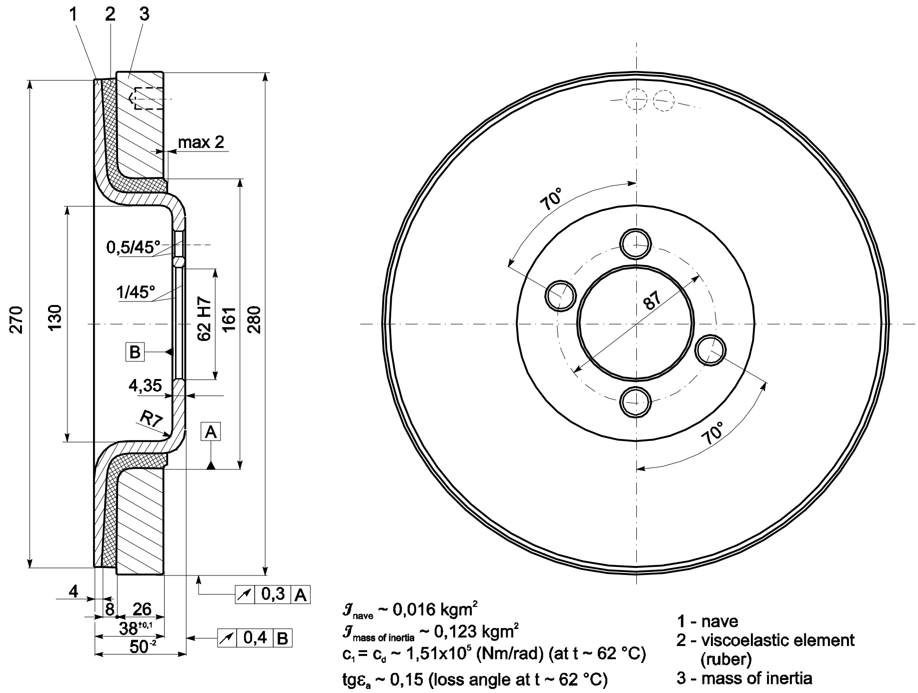

Figure 3: *Viscous-elastic torsional vibration damper*

Figure 3 shows a drawing of a particular torsional vibration damper with the basic parameters provided by the manufacturer.

2. SOLVING AND VERIFICATION OF EQUIVALENT TORSIONAL VIBRATION SYSTEM

After identifying all the necessary parameters of the system, it is necessary to establish a system of equations describing the considered processes.

Torsional vibrations of the individual disks presented in the model (figure 1) can be described trough system of equations (2) in matrix notation:

$$[\mathscr{J}]\{\overset{\circ\circ}{\vartheta}\}+[K_u]\{\Delta\overset{\circ}{\vartheta}\}+[K_s]\{\overset{\circ}{\vartheta}\}+[C]\{\Delta\vartheta\}=\{M\} \qquad (2)$$

where are: ϑ twisting angle, $\overset{\circ}{\vartheta}$ and $\overset{\circ\circ}{\vartheta}$ derivation of twisting angle, $\Delta\overset{\circ}{\vartheta}$ difference of twisting velocity of neighbouring disks, $\Delta\vartheta$ difference of twisting amplitudes of neighbouring disks. Knowledge of all parameters from system of equations leads to solve twisting amplitudes of the pulley of the IC engine. The pulley assembly is the most common location for measurement of vibration amplitudes. The obtained system of equation is solved by MatLab.

For the purpose of verifying torsional vibration model IC engine with following data is analysed:

- Natural aspirated, in line 6 cylinder diesel engine with firing order 1-5-3-6-2-4;
- diameter/stroke of piston 123 mm/140 mm; compression ratio ε = 16,5;
- max power/engine speed: 143 kW/2200 rpm;
- torsional vibration damper on the specific engine is taken from figure 3. In one example the parameters of the torsional vibration damper are taken from figure 3 (c_1=1,51 10^5 Nm/rad, K_{u1}=2,51 Nms/rad ($tg\varepsilon_a$=0,15)), and in the second example parameters are estimated according to experimental analyses and calculations.

By analyzing the results obtained by calculation and their comparison with experimental results, figure 4, it can be concluded that the approach of establishing an equivalent inline torsional vibration system, and the way of determining the parameters of the system, are good enough and acceptable for engineering practice, where the results obtained could be useful for evaluating the efficiency of torsional vibration dampers [1, 7].

Figure 4 presents comparative calculated and experimental results as a result of observation of significant excitation orders and 2nd mode vibrations of the previously described IC engine with a torsional vibration damper. Far better agreement of calculated and experimental results is in the case of knowing the real (nonlinear) characteristics of the elastic element of torsional vibration damper.

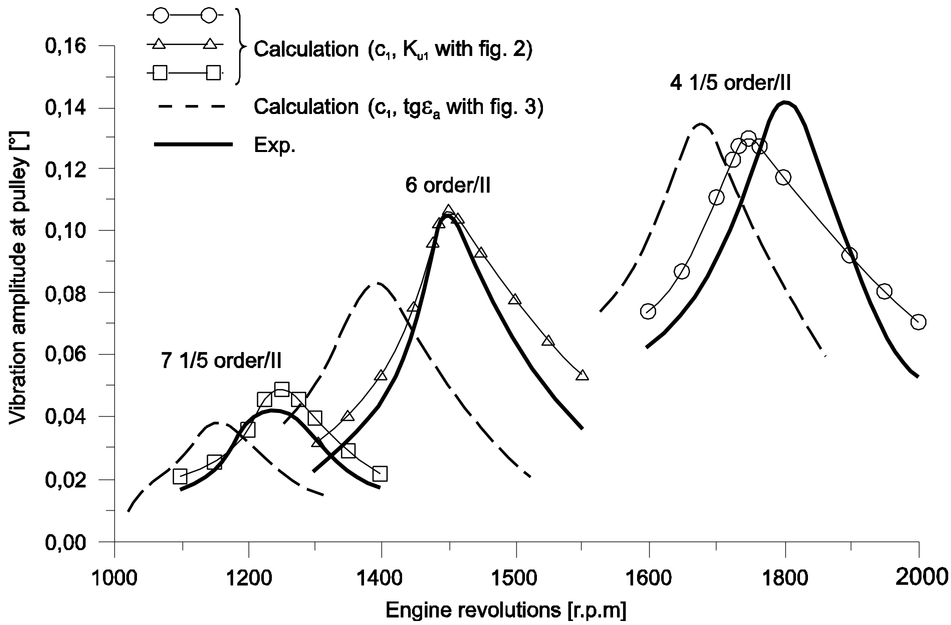

Figure 4: *Vibration amplitude at engine pulley, as a function of engine revolution, for interesting excitation orders and II mode vibrations*

Based on results achieved in this manner, values of dynamic amplitudes of twisting tension on individual part of the engine crankshaft can be calculated which is helpful in determining the dynamic endurance of the engine crankshaft.

CONCLUSIONS

Complicated torsional vibration systems of IC engines, for the purposes of engineering practices, can be effectively replaced by the so-called in line equivalent torsional vibration systems. The calculation results of torsional vibration of certain parts of the system, obtained by the proposed method of calculation, agree quite well in character and the absolute values with the real-measured values. The biggest problem in determining the individual parameters of equivalent torsional vibration systems is the nonlinear damping of torsional vibration damper. Based on comparisons of calculated and experimental results it is confirmed that the entire process of analysis of torsional vibrations, from physical model, ways of determining the input parameters, mathematical models to numerical solutions for the everyday practice of analyzing the influence of certain parameters is largely satisfactory, with provided knowledge of the nonlinear characteristics of elastic element of torsional vibration damper.

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