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MODEL AIDED DESIGN OF TUNED RUBBER TVD

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Abstract

The rotation of a crankshaft in piston combustion engines results from the acting of tangential forces on the crank, whose value changes with the change of the angle of shaft rotation. This results in torsional vibrations. These vibrations become especially dangerous when the frequency of changes of any components of tangential force is near or equal to the natural frequency of the shaft. It leads to resonant amplification of vibration amplitude and to exceeding the limit values of the angle of shaft torsion. Most often in such cases, various types of torsional vibration dampers are used. In automotive industry, these are usually rubber vibration dampers. Typical torsional vibration damper is an example of a resonant damper, which is designed for the most dangerous resonant frequency of the crankshaft related to the first form of vibration for which the torsional vibrations usually have the greatest amplitude. The design of such a damper involves choosing the inertia moment of the flywheel and the parameters of viscous-elastic element. The article describes the model and the simulation research, which allowed for creating the procedure of designing rubber dampers of torsional vibrations. This procedure can help to reduce the costs of operation tests for the design of optimal torsional vibration damper.

Keywords: *automotive research, simulation, combustion engines, crankshaft, rubber damper, torsional vibration*

1. Introduction

Piston combustion engines are most commonly used drive units in currently manufactured cars. In such engines, rotation of the crankshaft is a consequence of acting of the tangential forces on its cranks. The values of these forces change with the change of the angle of rotation of the shaft, which results in torsional vibrations. The shaft twists periodically between its ends around rotation axis at an angle of rotation. This angle is relatively small; however, because of torsional stiffness of the shaft, the instantaneous values of torsional moments occurring in cranks are much bigger than the value of torque transmitted by a shaft to the flywheel. The frequency of changes of these moments is much bigger, and the external damping in the crankshaft is too small to reduce the amplitude of extorted torsional vibrations. Therefore, this phenomenon, in many cases, may lead to fatigue fracture of the crankshaft and side effect damages of equipment of the combustion engines, which are driven by the crankshaft.

Torsional vibrations of the crankshaft become especially dangerous when the frequency of any of the components of tangential force is near or equal to the natural frequency. It leads to resonance amplification of vibration amplitude and exceeding permissible values of the angle of shaft torsion. In such cases various types of torsional vibration dampers are used. Although such dampers are very rarely a solution planned in early stages of designing drive systems, in most cases they occur to be the best solution in the task of reducing vibrations and noise generated by a combustion engine. They should not be considered in terms of a proverbial “patch” [2, 17].

2. Torsional vibrations of the crankshaft

Within the forces acting in a crank-piston system, rotational motion of the shaft is made by forces tangent to the trajectory of the crank (Fig. 1). Tangential forces T affect the value of torque that can be described by the equation:

$$M(\varphi) = T \cdot r = \frac{(P_g - P_b)}{\cos\beta}\sin(\varphi + \beta)\cdot r. \qquad (1)$$

Variability of the force T (Fig. 2) causes the acceleration in rotational motion of the engine crankshaft, resulting in torsional vibrations, which change with the change of rotational shaft velocity. In order to determine the influence of tangential force T on torsional vibrations, the harmonic analysis is carried out. This analysis is based on the Fourier series expansion:

$$T = T_0 + \sum_{k=1}^{\infty} T_k \sin\left(\frac{k}{2}\varphi + \phi_k\right), \qquad (2)$$

where:
T_0 – the average value of the tangential force,
T_k – amplitude k of this harmonic tangential force,
ϕ_k – phase angle k of this harmonic tangential force.

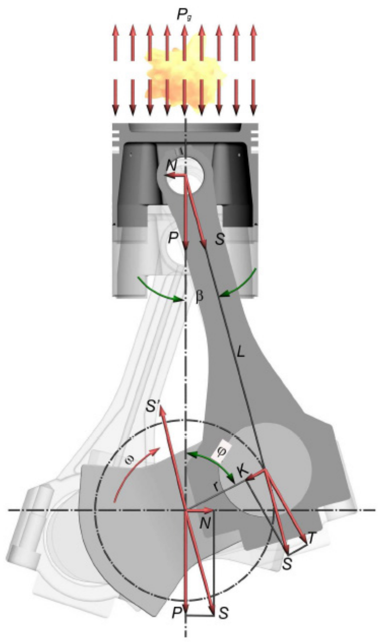

Fig. 1. The distribution of forces in the crank--piston system

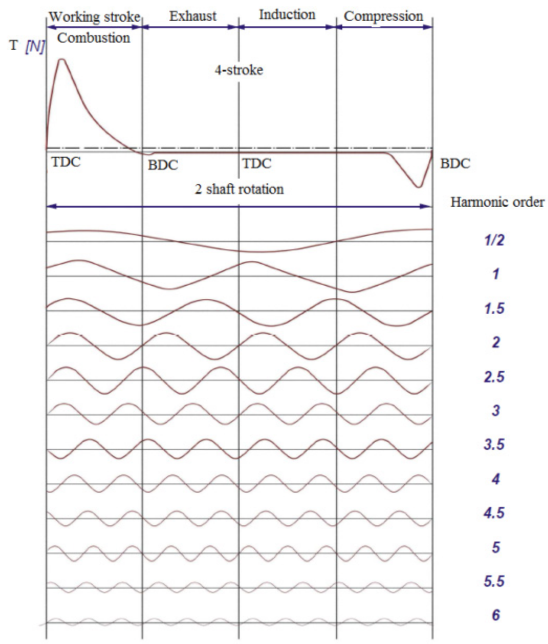

Fig. 2. The plot of tangential force T and its harmonics in the four-stroke engine

As the experiments show [8, 12, 13, 16], T it is enough to find only a certain number of harmonics in the analysis of the harmonic tangential force k. Usually the number of harmonics is in the range of the first 12-18 harmonics because higher harmonics of high frequencies and small amplitudes have no significant effect on torsional vibrations.

Ratio k-of this harmonic to the angular velocity of the engine crankshaft is called the order of harmonic h which for four-stroke engines is $h = k/2$ (Fig. 2).

The vibrations of the crankshaft with the highest amplitude usually occur when the frequency k of this harmonic is equal or close to its natural frequency, usually corresponding to the first mode of vibrations (in resonance conditions).

In a multi-cylinder combustion engine, each group of harmonics excited by one cylinder overlaps the harmonics excited by other cylinders. Therefore, the harmonics of a given order *h* may be in phase. In such a situation, there are so-called "amplified harmonics" called major harmonics. In the case of the engine where ignitions occur at equal intervals, the most dangerous are the critical speeds at which the order of *h* harmonic corresponds to the number of ignitions per one rotation of the crankshaft of the engine. Thus, in the case of four-stroke engine it corresponds to the multiple of the number of cylinders.

Dangers of working of the crankshaft in resonance conditions can be avoided by, inter alia:

– the "shift" of free vibrations frequencies of the system towards higher values, beyond the range of working rotational speed of the engine;
– the use of special devices in the system to reduce the amplitude of the torsional vibration in resonance conditions, so-called torsional vibration dampers.

The first way requires an increase in torsional stiffness of the crankshaft and simultaneous reduction of moments of inertia of the masses in reciprocating motion, which in most cases is practically impossible. Therefore, the second method, i.e. the use of torsional vibration damper, is increasingly used in the task of vibration reduction.

In practice all devices reducing the amplitude of the angle of crankshaft torsion are commonly called "torsional vibration damper", no matter what their construction and working principle are. Over the years, in order to minimize the risk coming from torsional vibrations, there were used different kinds of dampers: pendulum, frictional, spring, viscous, rubber. They differ not only in construction but above all in characteristics of work.

Currently, there are two most commonly used dampers: viscous and rubber ones. Viscous dampers are used primarily in engines operating at a constant working rotational speed, for example, in the engines of ships, machines and diesel locomotives, etc. Rubber dampers are most widely used in car engines, where the working rotational speed changes in a relatively wide range [4, 6].

3. Rubber torsional vibration damper

Rubber torsional vibration damper is assembled of the flywheel connected to the hub by the means of a flexible ring (Fig. 3). These rings are made not only from natural or synthetic rubber, but also from various kinds of elastomers, silicones, etc. No matter what the kind of the material for elastic ring is used, all dampers of this type are in practice called "rubber" – the principle of their working is the same.

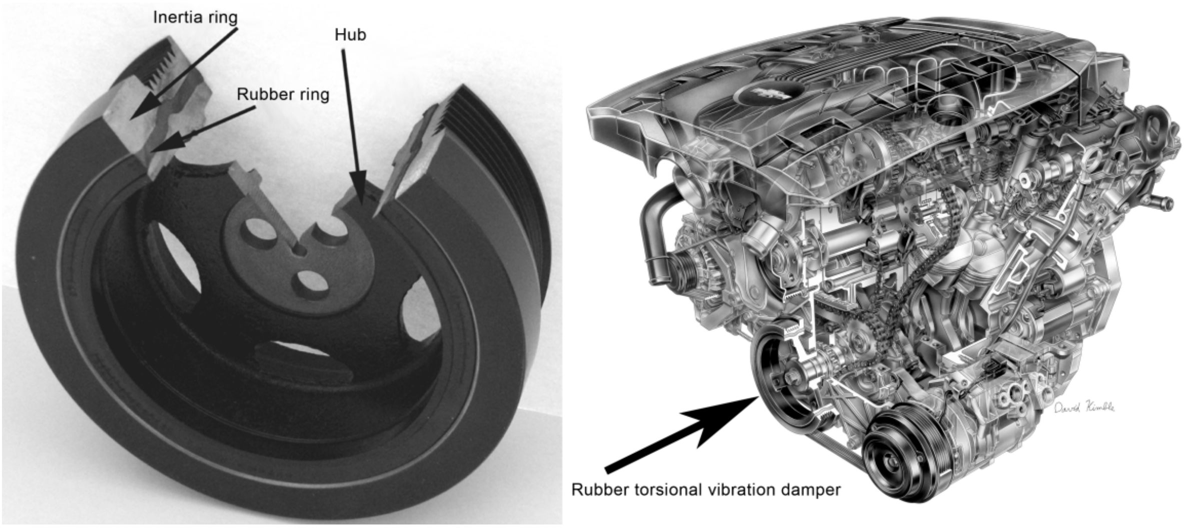

Fig. 3. Construction of rubber torsional vibration damper and common way of its mounting

Rubber torsional vibration damper is in fact a dynamic vibration eliminator, which "introduces" an additional degree of freedom to the system. The effect of this is a change of one of the main resonant frequency of the system for two "new" resonant frequencies. In addition, energy of vibrations is dissipated by the phenomenon of internal friction in the rubber element connecting the hub of the damper with the inertia ring. It works effectively from the moment when the engine is started in relatively wide range of working rotational speeds. It is "tuned" to frequencies of free vibrations of the crankshaft (usually connected with the first mode of torsional vibrations) [1, 3, 9].

4. The model of torsional vibration damper

After analysis of different solutions [5, 7, 8, 10, 16] the following model of torsional vibration damper was developed (Fig. 4).

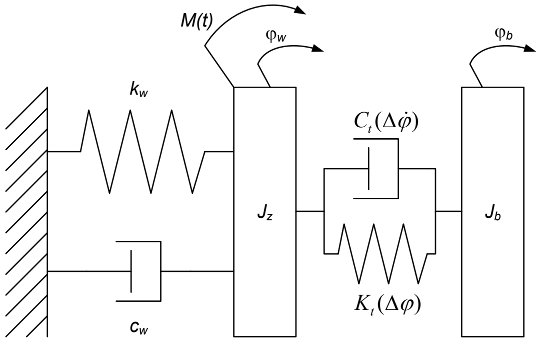

Fig. 4. The developed model of torsional vibration damper

This model is universal. It can be used while modelling viscous and rubber vibration damper and adequately defining the functions describing stiffness and damping (e.g. dynamic stiffness or a rubber damper and damping with internal friction in a rubber damper and shearing the viscous-elastic liquid in a viscous damper).

This model is described with a system of differential equations:

$$J_z \cdot \ddot{\varphi}_w + k_w \cdot \varphi_w + c_w \cdot \dot{\varphi}_w + K_t(\Delta\varphi) \cdot (\varphi_w - \varphi_b) + C_t(\Delta\dot{\varphi}) \cdot (\dot{\varphi}_w - \dot{\varphi}_b) = M(t), \tag{3}$$

$$J_b \cdot \ddot{\varphi}_b + K_t(\Delta\varphi) \cdot (\varphi_b - \varphi_w) + C_t(\Delta\dot{\varphi}) \cdot (\dot{\varphi}_b - \dot{\varphi}_w) = 0 \tag{4}$$

$$\Delta\varphi = \left|\varphi_w - \varphi_b\right|, \Delta\dot{\varphi} = \left|\dot{\varphi}_w - \dot{\varphi}_b\right| \tag{5}$$

where:

$K_t(\Delta\varphi)$ – function describing stiffness of torsional vibration damper,

$C_t(\Delta\dot{\varphi})$ – function describing damping of torsional vibration damper,

J_b – moment of inertia of the flywheel,

J_z – reduced moment of inertia of the crank system,

k_w – reduced stiffness of crank-piston system,

c_w – internal damping in a crank-piston system,

φ_w – angular displacement of the shaft,

φ_b – angular displacement of inertia ring.

5. Simulation research of the model of the torsional vibration damper

Rubber damper of torsional vibrations is an example of resonance damper, which is "tuned" (designed) to the most dangerous resonance frequency of the crankshaft connected with the first mode of vibrations for which torsional vibrations usually have the highest amplitude. In fact, it is a dynamic vibration eliminator. It introduces an additional degree of freedom. The effect of this is a change of one of the main resonant frequency of the system for two "new" resonant frequencies. That is the reason why despite nonlinear viscous-elastic characteristics the studies with the use of simplified linear models may be carried out in a certain approximation [11-14]

In order to start the research the following parameters of the model were chosen J_z, k_w, c_w. They were chosen in such a way so that resonant vibrations of the crank-piston system were simulated. It was done in a way so that the frequency characteristics of the system with one degree of freedom imitated typical frequency characteristics of torsional vibrations of the crank-piston system.

The next step was to choose the moment of inertia of the flywheel J_b. In most cases, it is recommended to choose the flywheel at approx. 1/20 value of reduced moment of inertia J_z.

After tuning the damper to the resonant frequency of simulations, there were performed simulations for different values of the moment of inertia J_b. It was done in order to examine the impact of its changes on the vibration amplitude (Fig. 5).

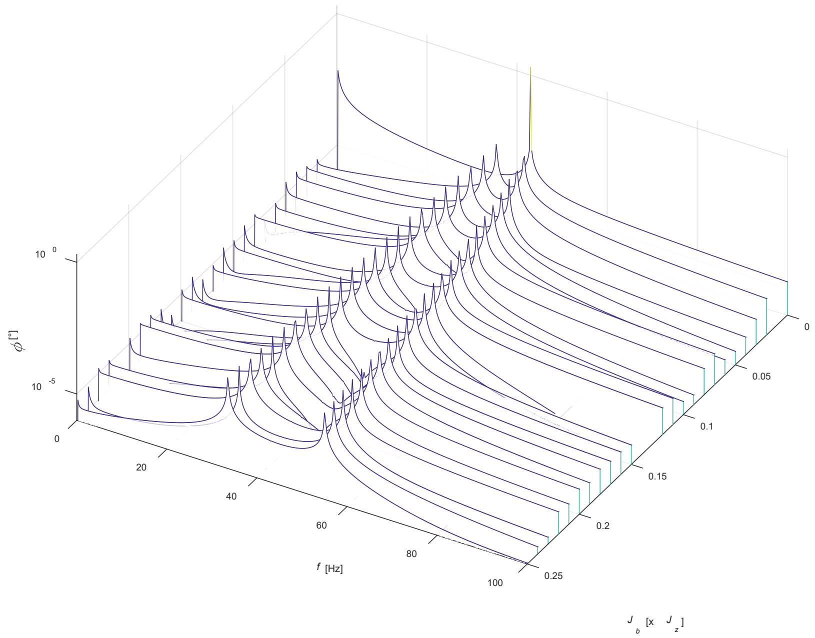

Fig. 5. Sample results of simulation of working of torsional vibration damper with different moments of inertia of the flywheel J_b

The studies showed that increasing the moment of inertia of the flywheel from a certain value J_b only slightly decreases the vibration amplitudes in "new" resonant areas. Frequencies of this resonance are becoming more distant from resonant frequency to which the damper was tuned. This is how a wider range of the band, where damper works effectively, is obtained. This effect is positive, however, the bigger moment of inertia of the flywheel J_b the more energy the damper "takes" from the engine. Another reason against excessive increasing moment of inertia of the flywheel J_b thus, increasing its geometric dimensions, is usually limited space in engine chamber, which prevents from assembling damper with large dimensions.

The aim of further simulations was to study the influence of damping on the effectiveness of working of torsional vibration damper. The research was conducted for a fixed value of moment of inertia of the flywheel J_b the parameter c_t specifying damping in the damper was a subject to changes.

The studies showed that the increase of damping reduces amplitudes for "new" resonant frequencies. At the same time, the effectiveness of vibration damping with the frequency, to which the damper was tuned, decreases. Like in the case of moment of inertia of the flywheel J_b damping parameter cannot be freely increased c_t due to technical limitations. In the case of rubber torsional vibration dampers, designers are constrained by the fact that damping with internal friction in the elastic element is usually too high.

Further simulations were carried out to investigate the influence of losing the initial properties of stiffness for different moments of inertia of the flywheel J_b (unchangeable during the simulation) by changing the value of the parameter describing its stiffness k_t (Fig. 6).

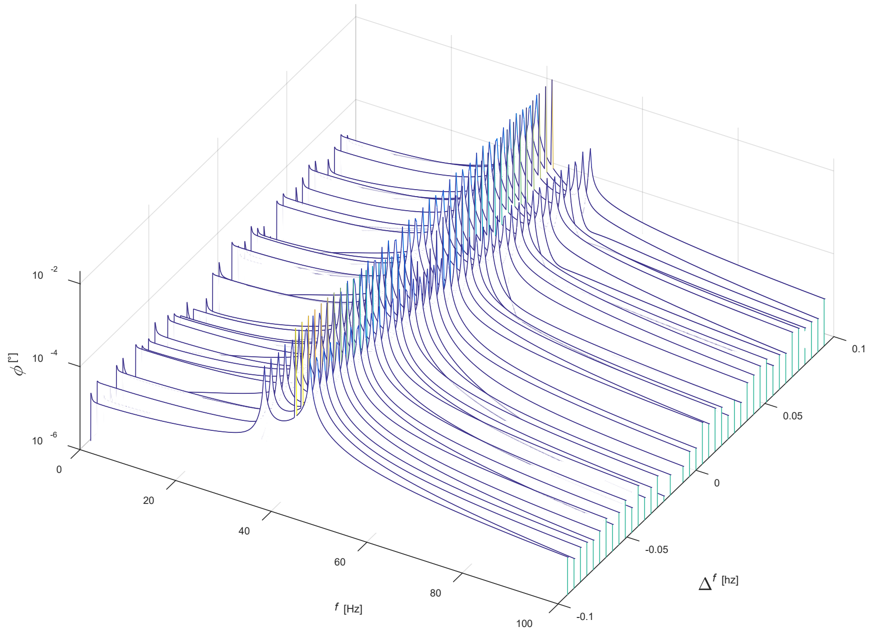

Fig. 6. Exemplary simulation results of working of the torsional vibration damper with different stiffness k_t with constant moment of inertia of the flywheel $J_{b1} = 0.05J_z$

During the analysis of obtained results it was pointed out that torsional vibration damper with a smaller moment of inertia of the flywheel, J_{b2} is characterized with a bigger sensitivity to the change of the stiffness parameter k_t. It results in a smaller effectiveness of vibrations damping in the resonant range. The damper should be theoretically tuned to this range. Therefore, in order to choose optimal moment of inertia of the flywheel J_b, after tests describing its influence on the amplitude of vibrations in resonant area for which a designed damper is "tuned", it is necessary to analyse the sensitivity to the change of the stiffness parameter k_t.

6. Methodology of selection of rubber torsional vibration damper

Torsional vibration damper is chosen or designed individually for each type of combustion engine (drive system). In automotive combustion engines, rubber vibration dampers are currently most widely used solution. Engine manufacturers often use the fact that the "new" engine, which should have the damper tuned, is not a new construction. It is only a modification of a well-known solution. That is why; the manufacturers choose the damper from a set of ready solutions by trial and errors. Such a procedure requires a verification of a correct selection as a result of operation research. It is time consuming and expensive. A significant reduction in the cost of operation tests in terms of a damper optimization can be obtained using simulation methods in the first place. Therefore, there was prepared a 7-point procedure of designing an optimum rubber damper of torsional vibrations for a particular type of the engine based on the research on mathematical model. Of course, it is not possible to eliminate experimental studies in the process of damper design. However, their number may be limited to a minimum.

The procedure of designing rubber damper of vibrations.

1) Determination of frequency characteristics of a crank-piston system on the basis of experimental studies and defining frequency of free vibrations and identifying the frequency for which the amplitude of torsional vibrations should be damped.
2) Construction of an equivalent model of the piston-crank system and identification of its parameters: J_z, k_w, c_w.
3) Construction of an equivalent model of the crank-piston system with a damper of vibrations and determining the range of values of optimum moment of inertia of the flywheel on the basis of the numerical simulations J_b.
4) Determination of the range of optimum values of a damper coefficient c_t of a designed damper on the basis of the numerical simulations of the model from point 3. During this stage, it is important to include physical limitations resulting from practical possibility to solve this task.
5) Determination of the range of optimum values of stiffness coefficient k_t of designed damper (damper should most effectively damp the vibrations with a frequency specified in point 1 of the procedure) on the basis of simulation research of the model from point 3.
6) Simulation research on the model (from point 3),with the parameters J_z, k_w, c_w, J_b, c_t, k_t selected in points 2, 3, 4 and 5, which will lead to defining the sensitivity of a designed damper to small changes of stiffness parameter k_t. In the case of unsatisfactory results, it is important to come back to point number 3 of the procedure, choose a new moment of inertia from the range of values specified in this point J_b and repeat the steps.
7) The construction of a physical model designed according to guidelines of a rubber torsional vibration damper and its verification on the test stand.

7. Conclusions

The research allowed for making a mathematical model allowing for the choice of parameters of torsional vibration damper of the crankshaft. There was proposed a procedure of choice of such a damper. We are able to obtain the best parameters of a damper from simulation research and look for materials, which meet these requirements. On the other hand, we can accept the parameters of available materials and check their influence on shaft vibrations. Such a procedure may reduce time of damper selection and costs. However, it is important to remember that the selected damper should be verified on a real object [8, 12, 13, 15].

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