Motorless pilot studies of crankshaft dampers of combustion engines

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Abstract. This article is about motorless studies of crankshaft dampers of combustion engines. An option to reduce the adverse effect of torsional motions of the crankshaft to the working life and effective performance of the engine is the dampers' installation. However, dampers' installation causes a change in the number of motor masses, what in turn results in a change in free frequency. As a consequence, there are changes in the resonant modes of operation, and ones in the type and shape of the constrained oscillations. To ensure the efficient operation of the recently designed construction of dampers, numerous final tests have to be performed, and the calculation itself is iterative, which considerably increases the complexity and a number of calculations. Stand design is suggested for carrying out these researches in a motorless way, while the modes and parameters of the torsional motions of the crankshaft match the operating modes of the combustion engine itself. Changeability of the distance between the motor masses and the type and magnitude of the torques acting on the motor masses makes the suggested stand universal for testing engines of any layout diagram. Torque modulation in amplitude, frequency, and phase is obtained by changing the appropriate electrical settings on the stator windings, creating torque for an equal torsional circuit. This circuit also enables testing only for resonant torque harmonics, which is an unquestionable advantage of the offered stand.

1. Introduction
Design elements of modern tractor engines possess a high degree of thermal and mechanical load, which is defined by the desire of implementers to get the maximum efficiency of their work. They tackle issues of eliminating or rationally decreasing the occurrence of unfavorable operating modes that can additionally load parts and units [1, 2]. These include, for example, the torsional motions of the crankshafts occurring in the operation of the engines.

2. Materials and methods
Considering the design features of motor vehicle engines and their loading modes, it can be asserted that torsional motions are the most dangerous for them. Bending vibrations of the crankshafts are more specific for large-sized and stationary engines, and extension vibrations are in particular of interest in the development of marine engines [3, 4]. In the design of a modern engine, it is essential to take
measures avoiding the development of torsion motions of the crankshafts in the range of possible speed and load modes of operation.

Engineering calculation of torsional motions of crankshafts involves several stages [5]:

1) replacement of a real shaft system with a calculated one which is dynamically equivalent to it. It is called an equivalent torsional system;
2) determination of the free oscillation frequencies of an equivalent torsional system;
3) harmonic analysis of the torque of one or more cylinders performing on a single crank pin;
4) determination of the resonant rotational frequencies and the main (strong) harmonics of the torque;
5) amplitudes calculation of constrained oscillations in resonant modes of operation;
6) Secondary stresses definition due to torsional motions in case of resonance;
7) development of organizational or design measures to reduce the amplitude of torsional motions.

Following the theoretical and experimental researches of torsional motions of the crankshaft and the analysis of their results, it is feasible to make a decision on the need to decline their unfavorable impact on the performance of the power unit. There are following ways to solve this problem:

1) if there are several versions of the sequence of alternating working strokes in the engine, it is required to choose one of them to provide the phase conditions of the exciting harmonics on the cranks, which give the lowest values of the resultant vector of the relative amplitudes of the displacements;
2) changing the system settings, especially the design of the crankshaft, which will result in a change in the frequency of free oscillations and the output of the resonant mode out of the operating range of rotation speeds; an increase in the mass moment of inertia results in a decrease in the frequency, and an increase in the deflection rate causes an uptick in the frequency. However, with a multi-mass plan, this dependence is less pronounced than with a single - or two-mass plan, and for a comparatively small change in the frequency of free oscillations, considerable structural changes in the parts may be necessary, which is not always possible [6];
3) additional mechanisms installed on the crankshaft, are intended purely to decrease the adverse effect of torsional motions and are called dampers [7, 8].

3. Results and discussion
Calculation and determination of dampers settings is a multivariate iterative task, requiring experimental evolution of its results. It is highly time-consuming and costly.

The authors suggest a technique for motorless studies of the designed dampener based on a stand (Figure 1), which enables us to model a different spatial design of the crankshaft and its loading modes [9].
Figure 1. Stand scheme for dampers testing

On the figure: 1 - splined shaft, 2 rotating bearing, 3 – flywheel, 4 – inertial mass, 5 – sensors, 6 – stators, 7 - rotors, 8 – damper, 9 – pulse AC supply system, 10 — digital quantizer, 11 — signal booster, 12 — recording device.

A stand for a dampers’ tests consists of a shaft mounted on rotating bearings, on which the rotors are installed, and stators with field windings connected to an electric current source are installed coaxially to them. A special feature of the suggested stand is that the test object — dampener is installed at one end of the shaft. The flywheel with variable inertial mass is placed at the other end of the shaft. The stators are connected to a pulsed AC supply system, and the rotors are equipped with sensors that register torsional motions, which are interacted with digital quantizer, a signal booster, and a recording device. In addition, the rotors and stators are installed with the ability to move along the shaft axis.

On the shaft, at least four of the above-mentioned rotors are mounted and four stators are coaxially mounted. The shaft is splined.

The device consists of a splined shaft 1 mounted on rotating bearings 2. At one end of the splined shaft 1, a flywheel 3 with a variable inertial mass 4 is installed. At least four stators 6 and rotors 7, acting as electric motors, are mounted coaxially on the splined shaft 1, with the possibility of moving along the axis of rotation of the splined shaft 1. The stators 6 are connected to a pulsed AC supply system 9. On the rotor 7 of the electric motor, sensors 5 are installed that register torsional motions, transmitting data about the oscillations to the digital quantizer 10, then the signals go to the signal booster 11 and are transmitted to the recording device 12. Test damper 8 is mounted on the other end of the splined shaft 1.

The device operates in the following way. The crankshaft configuration is selected. On the flywheel 3, installed at one end of the splined shaft 1, mounted on the rotating bearings 2, the selected inertial masses are installed. Stators 6 and rotors 7, acting as electric motors, push along the splined shaft 1 at a chosen distance from each other. Stand for dampers tests 8 is launched twice, with the installed dampers 8 and without it to check the effectiveness of torsional motion dampening on the splined shaft 1 with the selected crankshaft configuration. The stator 6 is supplied with an electric alternating current, alternately, according to the order of operation of the cylinders, using a pulsed AC supply system 9, which creates a rotating magnetic field, creating a torque that carries the rotors 7, working as electric motors. During the rotation of the splined shaft 1, under the influence of torque, torsional motions occur on it. Torsional motions at first, when operating without installing a damper 8, and then with it, are
recorded by sensors 5 installed on the rotors 7 and transmitted to a digital quantizer 10, a signal booster 11, and a recording device 12. When conducting tests on the stand of the damper 8.

Therefore, due to the fact that it is possible to choose any configuration of the crankshaft by moving the stators 6 together with the rotors 7 along the splined shaft 1, the operational capabilities of the stand are expanded, as well as owing to the appropriate selection of the inertial masses 4 on the flywheel 3, the exactitude of the obtained data on the torsional motions of the tested damper is raised.

We will specify the required values of the settings of the torque harmonics on the stators 7 of the suggested stand.

Using the example for the analysis of torsional motions, we will consider an opposed four-stroke six-cylinder engine (Figure 1) with the following parameters:

\[ N_{eN} = 170 \text{ kW}, \quad n_{eN} = 5800 \text{ rpm}, \quad D = 0.095 \text{ m}, \quad S = 0.073 \text{ m}, \quad d_{cr} = 0.045 \text{ m}, \quad l_{cr} = 0.025 \text{ m}, \quad d_{cp} = 0.045 \text{ m}, \quad l_{cp} = 0.026 \text{ m}, \quad e = 0.008 \text{ m}, \quad h = 0.1 \text{ m}, \quad r = 0.0366 \text{ m}, \quad \rho = 7200 \text{ kg/m}^3. \]

![Figure 2. Engine layout diagram](image2.png)

For harmonic analysis, the torque values \( M_{tor} \) for the mode \( n_N = 5800 \text{ rpm} \) are taken (Figure 3).

![Figure 3. Torque change graph \( M_{tor} \)](image3.png)
The values $M_k$ and $\varphi_k$ are found which are obtained by Cauchy integrals

\[ M_k = \sqrt{A_k^2 + B_k^2}, \quad (1) \]

\[ \varphi_k = \arctan \left( \frac{A_k}{B_k} \right). \quad (2) \]

Cauchy integrals are replaced by their final sums

\[ A_k = \frac{2}{N} \int_0^N y_i \cos(k \cdot x_i), \quad (3) \]

\[ B_k = \frac{2}{N} \int_0^N y_i \sin(k \cdot x_i), \quad (4) \]

where $N=72$ - the number of sections, taken from the dynamic analysis; $y_i = M_{tor,i}; k=1,2,3,...,30$ - the order of a harmonically changing moment; $x_i = i \cdot \frac{2\pi}{N}, i=1,...,72$.

The results of the values $A_k$, $B_k$, $M_k$ and $\varphi_k$ are given in Table 1.

| $k$ | $A_k$   | $B_k$   | $M_k$   | $\varphi_k$ |
|-----|---------|---------|---------|-------------|
| 1   | -82.863 | -57.248 | 100.715 | 55.360      |
| 2   | 48.629  | 125.322 | 134.427 | 21.208      |
| 3   | -10.995 | -95.504 | 96.134  | 6.567       |
| 4   | 9.373   | -165.810| 166.074 | -3.236      |
| 5   | 10.108  | -64.273 | 65.062  | -8.937      |
| 6   | -9.445  | -46.893 | 47.834  | 11.388      |
| 7   | 15.508  | -41.563 | 44.362  | -20.462     |
| 8   | -14.784 | 23.591  | 27.841  | -32.075     |
| 9   | 15.685  | -25.313 | 29.778  | -31.784     |
| 10  | -13.556 | 19.321  | 23.602  | -35.054     |
| 11  | 12.987  | -14.687 | 19.605  | -41.485     |
| 12  | -11.208 | 11.564  | 15.979  | -43.640     |
| 13  | 11.079  | -8.576  | 14.011  | -52.259     |
| 14  | -9.025  | 6.397   | 11.063  | -54.670     |
| 15  | 9.034   | -4.624  | 10.149  | -62.894     |
| 16  | -7.073  | 3.359   | 7.830   | -64.595     |
| 17  | 7.243   | -2.409  | 7.633   | -71.603     |
| 18  | -5.594  | 1.706   | 5.848   | -73.039     |
| 19  | 5.937   | -1.096  | 6.037   | -79.540     |
| 20  | -4.418  | 0.692   | 4.472   | -81.094     |
| 21  | 4.874   | -0.353  | 4.887   | -85.855     |
| 22  | -3.517  | 0.128   | 3.520   | -87.923     |
| 23  | 4.028   | 0.094   | 4.029   | 88.658      |
| 24  | -2.746  | -0.118  | 2.749   | 87.540      |
| 25  | 3.431   | 0.227   | 3.438   | 86.223      |
| 26  | -2.229  | -0.258  | 2.444   | 83.394      |
| 27  | 2.963   | 0.281   | 2.976   | 84.585      |
Motor harmonics are calculated

\[ M_{mh}(t) = M_{at} + \sum_{k=0.5}^{15} M_k \cdot \sin(k_{m} \cdot \varphi + \varphi_{km}), \]  

(5)

Where \( M_{at} = \frac{1}{N} \sum_{i=0}^{N} y_i \) — average torque, acting on the crank; \( k_{m} = 0.5; 1; 1.5; \ldots; 15 \) — motor harmonic index.

Let us plot the change in the torque \( M_{tor} \) and motor harmonics \( M_{mh}(t) \) from the rotation angle of the crankshaft (Figure 4).

Figure 4. Graphs of torque changes.

Values of the amplitudes and phase shifts of the motor harmonics obtained are the initial data for the planning of the electrical settings of the suggested stand.

4. Conclusion
The suggested scheme of motorless tests and smoothing of dampers gives the possibility to considerably downsize both time and expenses on formation of perspective schemes of the designed structures. Geometric and electrical values of the stand are specified depending on the required power parameters and the layout design of the engine. As for the operating characteristics, they are set by the order of operation of the cylinders and the flow type of the engine torque.

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