Analysis of the effectiveness of using torsional vibration dampers for crankshafts with adaptive characteristics in internal combustion engines

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Abstract. The research aims to determine the effectiveness of torsional dampers with adaptive characteristics for crankshafts of piston internal combustion engines. The paper deals with the pressing issue of designing a torsional vibration damper for crankshafts of internal combustion engines, which can work efficiently in the entire operating range of speed variance. The standard approach to design the damper with constant characteristics implies one resonant operating mode, which in actual practice is not always justified. This fact decreases the operating efficiency, especially for the forced V-type engines. In order to determine the design parameters of dampers with adaptive characteristics, a multi-parameter optimization problem must be solved, which requires experimental validation. The work considers an approximate method, according to which the main parameters of the designed damper for different operating modes are determined. We obtained the general trend of changing the torsional damper characteristics by performing calculations for different engine operating modes. Based on these data, it is possible to determine the operating modes of the control mechanism. Comparing the efficiency of dampers with constant characteristics under the same modes allows validating the results' adequacy. The scientific novelty lies in the developed method of modes determining torsional vibration damper parameters at changing speed and operating can help design new efficient forced internal combustion engines.

1 Introduction

The pace of movement and the speed of machine-tractor mechanisms should constantly and to a large extent show positive dynamics because it is the increase in the productivity of the return of technical devices that depends on the speed of movement [1].

Designing torsional vibration dampers for crankshafts of machine-tractor engines that work great over the full speed range is a rather difficult task. Even to calculate the crankshaft

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only for strong and fundamental harmonics of the exciting moment, it is necessary to know the required work due to friction torque [2].

Let us consider the opposite spark ignition six-cylinder engine (O6) to analyze these target parameters.

2 Materials and methods

The research aims to consider the issues of designing a torsion vibration damper for crankshafts of internal combustion engines, capable of operating efficiently in the entire operating range of speed dispersion [3].

To achieve the stated aim, a range of the following tasks has been formulated:

- Explore standard approaches to designing dampers with constant characteristics;
- Solve a multi-parameter optimization problem;
- Consider an exemplary method according to which the main parameters of the designed damper are determined for different operating modes.

The study of torsional dampers with adaptive characteristics for crankshafts of piston internal combustion engines was carried out within the basic provisions of physics, conceptual principles of mechanics, vibration theories, mathematical interpretation of factors and model calculations, and methods of designing and testing systems. A specialized MatLab (MATrix LABoratory)/Simulink software package was applied to study physical and mathematical efficiency systems of torsion shock absorbers with adaptive characteristics for crankshafts of reciprocating internal combustion engines.

3 Results

Determination of the vibration damper parameters is the result of torsional vibration analysis, including the following calculations [4]:

1) parameters of the equivalent torsional scheme;
2) natural frequencies of torsional oscillations and corresponding resonant modes of engine operation;
3) harmonic analysis of engine torque in all resonant operation modes;
4) amplitudes of vibrations of motor masses and arising elastic moments;
5) shape of the vibration and required damper installation place;
6) determination of potential for effective damper operation during all modes.

Consider the opposite spark-ignition engine (O6) with the following technical features:

- Nominal power: Nen=170 kW at nn=5800 rpm;
- Cylinder bore: D= 0.095 m;
- Piston stroke: S= 0.073 m;
- Main journal: diameter d_m= 0.055 m and lengthl_m= 0.025 m;
- Rod journal: diameterd_r=0.055 m and lengthl_r= 0.025 m;
- Crank web: thicknessd_w=0.055 m and widthl_w=0.025 m;
- Crank radius: r=0.0366 m;

The calculation is carried out according to the standard method described in [5].

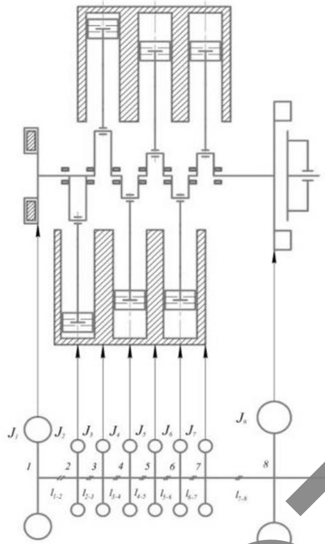

Fig. 1. Scheme of the engine with torsional vibration damper: J_1 – moment of inertia of the torsional vibration damper; J_2, J_3, J_4, J_5, J_6, J_7– motor mass moment of inertia; J_8 – flywheel moment of inertia; $l_{i–i+1}$ – length between i-th and (i+1) stsections of the equivalent torsional scheme.

Source: Compiled by the authors.

The equivalent torsion scheme parameters were determined by the calculation-graphical method, assuming the identity of the sections between the motor masses [14]. Table 1 presented results of the preliminary calculation of the eight-mass torsional scheme, where *J*i is the i-th motor mass moment of inertia, li–i+1 is the length of the section between the i-th and (i+1)st motor masses, Ci–i+1is the stiffness of the section between the i-th and (i+1)st motor masses [5, 6].

Table 1. Results of the preliminary calculation of the eight-mass torsion scheme.

Moment of inertia J_i, kg·m^2	Reduced length $l_{i–i+1}$, m	Stiffness, $C_{i–i+1}$, N/m
$J_1 = 4.09215·10^{-2}$	–	–
$J_2 = 3.627602·10^{-3}$	$l_{1–2}$= 0.112	$C_{1–2}$ =1.89388·10^5
$J_3 = 3.627602·10^{-3}$	$l_{2–3}$=0.0967	$C_{2–3}$ =3.53993·10^5
$J_4 = 3.627602·10^{-3}$	$l_{3–4}$=0.0967	$C_{3–4}$ =3.53993·10^5
$J_5 = 3.627602·10^{-3}$	$l_{4–5}$=0.0967	$C_{4–5}$ =3.53993·10^5
$J_6 = 3.627602·10^{-3}$	$l_{5–6}$=0.0967	$C_{5–6}$=3.53993·10^5
$J_7 = 3.627602·10^{-3}$	$l_{6–7}$=0.0967	$C_{6–7}$ =3.53993·10^5
$J_8 = 1.041727·10^{-2}$	$l_{7–8}$= 0.0505	$C_{7–8}$ = 6.76655·10^5

Source: Compiled by the authors.

Natural frequencies are determined using the MATLAB computing platform and the Microsoft Excel spreadsheet. Based on the calculation method from [7], the system's natural frequency (ω0) was found as 1284 s–1. We obtained the relative amplitudes (Table 2).

Table 2. Relative amplitudes.

Index number of the *i*-motor mass	1	2	3	4	5	6	7	8
Relative amplitude of the motor mass a_i	1	0.676	0.491	0.298	0.100	–0.099	–0.297	–0.398

Source: Compiled by the authors.

And a single-node vibration mode (Figure 2).

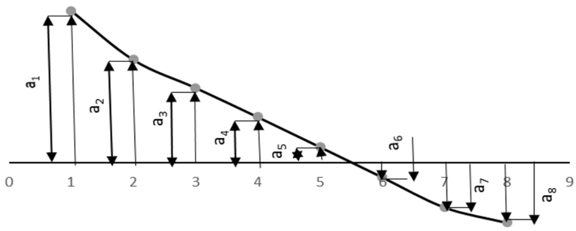

Fig. 2. Single-node vibration mode considering motor masses.
Source: Compiled by the authors.

We performed the harmonic analysis of torque for nominal operating mode n_{eN}= 5,800 rpm according to the standard method (Table 3); for other modes, the harmonic parameter values were calculated as the sum of decomposition of gas and inertial forces [8].

Table 3. Torque harmonic analysis for nominal duty.

k_M	A_k	B_k	M_k	φ_k
0.5	–82.863	–57.248	100.715	55.360
1.0	48.629	125.322	134.427	21.208
1.5	–10.995	–95.504	96.134	6.567
2.0	9.373	–165.810	166.074	–3.236
2.5	10.108	–64.273	65.062	–8.937
3.0	–9.445	–46.893	47.834	11.388
3.5	15.508	–41.563	44.362	–20.462
4.0	–14.784	23.591	27.841	–32.075
4.5	15.685	–25.313	29.778	–31.784
5.0	–13.556	19.321	23.602	–35.054
5.5	12.987	–14.687	19.605	–41.485
6.0	–11.208	11.564	15.979	–43.640

Note: kM – torsional moment; Ak – automatic oscillations; Bk – internal oscillations; Mk – mechanical oscillations; φk – oscillation function.
Source: Compiled by the authors.

Frequency Interference Diagramis used to determine resonant operating modes (Figure 3).

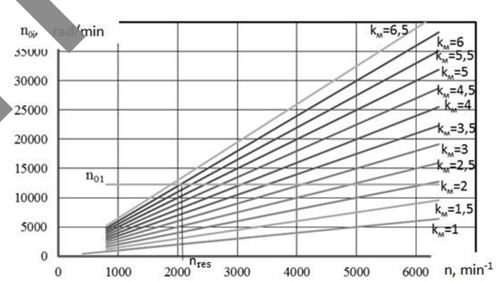

Fig. 3. Frequency Interference Diagram. Source: Compiled by the authors.

Consider the resonant operating modes at nres= 2,045 rpm for kм=6 (for the fundamental harmonic) and nres= 4,089 rpm for km=3 (for strong harmonics).

To determine absolute vibration amplitude of the 1st motor mass, it is necessary [9]:

Calculate RS for k_M=3 and k_M=6 from the following equation:

$$R_{S(k_M=6)} = [[\sum_{i=1}^{N} a_i^* \sin(k_M \Delta\theta_{1-i})]^2 + [\sum_{i=1}^{N} a_i^* \cos(k_M \Delta\theta_{1-i})]^2]^{\frac{1}{2}} = 0.771 \quad (1)$$

where:

a_i^* is the relative amplitude of the i-th motor mass;

$\Delta\theta_{1-i}$ is the phase displacement between the motor masses.

According to recommendations in [10], the damping coefficient ξ can be computed by the following formula 2:

$$\xi = \bar{\xi} F_p r^2 i = 0.1 \cdot 106 \cdot 0.007082 \cdot \pi \cdot 0.03662 = 5.697 \tag{2}$$

where:

F_p is the piston area; i isthe number of engine cylinders.

Using the formula 3, we determine the absolute vibration amplitude of the first motor mass for the absolute value of the torque:

$$A_1^* = \frac{M_k \sqrt{\left[\sum_{i=1}^{N} a_i^* \sin\left(k_M \Delta\theta_{1-i}\right)\right]^2 + \left[\sum_{i=1}^{N} a_i^* \cos\left(k_M \Delta\theta_{1-i}\right)\right]^2}}{\xi k_M \omega_{ICE} \sum_{i=1}^{N} (a_i)^2}, \tag{3}$$

The calculated values of the absolute amplitudes of motor masses for harmonics $k_M = 3$ and $k_M = 6$ are shown in Table 4.

Table 4. Estimated values of absolute amplitudes of motor masses.

i	a_i	A_1^*, rad	
		k_M =3	k_M =6
1	1	0.002456	0.000821
2	0.676	0.0016603	0.000555
3	0.491	0.0012059	0.0004031
4	0.298	0.0007319	0.0002447
5	0.100	0.0002456	8.21E–05
6	–0.099	–0.0002431	–8.128E–05
7	–0.297	–0.0007294	–0.0002438
8	–0.398	–0.0009775	–0.0003268

Source: Compiled by the authors.

Condition of effective operation of torsional vibration damper is equality of work of friction moments created by it to operation and work of the disturbing moment. Due to external and internal resistance, resonance amplitudes increase only until a balance is established between the energy supplied to the system by disturbing harmonics over one vibration period and the energy consumed to overcome the resistance during the same period. Therefore, it follows that the work of the resonating harmonic with the motor index k_M should be approximately equal to the work of the viscous friction forces generated inside the torsional damper body.

Supposing there is the following condition [11]. Work of resonating harmonic was determined by the following equation (formula 4):

$$L_{dist} \cong L_{fr} \tag{4}$$

where:

L_{dist} is the resonant harmonic work.

Work of the disturbing moment was given by (formula 5):

$$L_{dist} = \pi \times M_к \times A_1^* \times R_s, \tag{5}$$

where:

$M_к$ is the amplitude of k-th torque harmonic for resonant mode;

A_1^* is the absolute amplitude of the first motor mass;
Rs is the belt transmission.
Calculated values for work of the disturbing moment are shown in Table 5.

Table 5. Estimated values of the work of the alarming moment.

k_M	3	6
L_{dist}, J	0.28448093	0.031790271

Source: Compiled by the authors.

Work done by viscous forces can be expressed as, (formula 6) [5]:

$$L_{fr} = F_{fr} \cdot \Delta\varphi, \tag{6}$$

where:
$\Delta\varphi = A_{1\,max}$;
$F_{fr} = V_{av} S \mu / z$ is the viscous force;
$V_{av} = \frac{30 \cdot \Delta\varphi}{\pi \cdot n} R_{av}$ is the velocity of the fluidbetween the friction surfaces; z is the thickness of the layer;
S is the area;
μ =13200 Pa·s is the dynamic viscosity of the silicone;
R_{av} is the average friction radius.

Thus, the work of the friction forces depends on the area of viscous friction and the amplitude of the vibrations of the motor masses. For the torsional damper's effective operation in the full speed range and fundamental and strong harmonics of the disturbing moments, the work value of friction force should be more than these parameters. This fact can be attained by changing torsional dampers' friction surfaces with variable characteristics [12].

4 Discussion

The efficiency issues of torsional dampers with adaptive characteristics for crankshafts of piston internal combustion engines are quite debatable. According to N. D. Chin, the standard approach to designing dampers with constant characteristics implies one resonant mode of operation. D.C. Nguyen also believes that the damper design on the machine shaft should be based on maximum stability. In our opinion, the use of a standard approach to designing dampers with constant characteristics in real practice is not always justified. We believe that to determine the design parameters of dampers with adaptive characteristics, it is necessary to solve a multi-parameter optimization problem that requires experimental validation. Thus, the work of the friction forces depends on the area of viscous friction and the amplitude of the vibrations of the motor masses. For the torsional damper's effective operation in the full speed range and fundamental and strong harmonics of the disturbing moments, the work value of friction force should be more than these parameters. This fact can be attained by changing torsional dampers' friction surfaces with variable characteristics [13].

5 Conclusion

The torsional dampers' design should be based on the crankshaft calculations for fundamental and strong harmonics for all engine speed modes. Simultaneously, the vibration dampers' design parameters should provide effective suppression of disturbing moments.

We obtained a general trend in changing the characteristics of the torsion damper by performing calculations for various engine operating modes. Based on these data, it is possible to determine the operating modes of the control mechanism, which is the practical significance of the research.

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