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SPECIFYING THE PROCEDURE FOR DESIGNING THE ELEMENTS OF THE CRANKSHAFT SYSTEM FOR A SMALL HIGH-SPEED DIESEL ENGINE

F. Abramchuk
Doctor of Technical Sciences, Professor, Head of Department*
E-mail: fedor.abramchuk@gmail.com
O. Grytsyuk
Doctor of Technical Sciences, Senior Researcher*
E-mail: dthkbd@ukr.net
A. Prokhorenko
Doctor of Technical Sciences, Professor
Department of Internal Combustion Engines
National Technical University
«Kharkiv Polytechnic Institute»
Kyrpychova str., 2, Kharkiv, Ukraine, 61002
E-mail: ap.kharkiv@ukr.net
I. Reveliuk
Lead Engineer
Ltd. Naukovo-vyrobnyche
pidpryyemstvo Dyzel' Hrup
Nauky ave., 60, Kharkiv, Ukraine, 61072
* Department of Internal Combustion Engines
Kharkiv National Automobile
and Highway University
Yaroslava Mudroho str., 25,
Kharkiv, Ukraine, 61002

Розвиток модельного ряду високообертових рядних малолітражних дизелів шляхом розробки шестициліндрової версії на базі вже доведеного чотирьохциліндрового тягне за собою зменшення жорсткості колінчастого вала. Це, в свою чергу, зменшує його надійність у результаті збільшення впливу крутильних коливань. Для оцінки цього впливу була проведена серія розрахунків з визначення амплітуд і дотичних напружень в системі колінчастого вала.

Для зменшення максимальних значень кутових амплітуд і дотичних напружень найбільш доцільним є застосування силіконового демпфера гасіння крутильних коливань. При його проектуванні необхідно враховувати такі конструктивні параметри, як ефективність гасіння та габаритні розміри.

В роботі було проведено розрахункове дослідження крутильних коливань з використанням відомого та досить простого методу розрахунку дійсних амплітуд і механічних напруг крутильних коливань колінчастого валу двигуна внутрішнього згоряння. Цей метод базується на основі чисельного розв'язання високорівневої системи нелінійних диференціальних рівнянь у вигляді моделі в просторі станів. У процесі визначення вихідних даних було запропоновано формулу для визначення жорсткості кривошипа колінчастого вала. Удосконалення відомої формули полягало у введенні коефіцієнту, що наближає значення, отримані за допомогою розрахунків, до отриманих експериментальним шляхом. Однак це уточнення є правомірним лише для колінчастих валів зі схожими параметрами кривошипів.

В процесі даного дослідження були обрані критерії, які використовуються при проектуванні елементів системи колінчастого вала. Це дало змогу, на основі даних критеріїв, визначити для дизеля, що проектується, наступні конструктивні характеристики: масовий момент інерції маховика демпфера, масовий момент інерції корпуса демпфера, а також діаметр та жорсткість носка колінчастого вала.

Проведення даного дослідження допомагає конкретизувати відому методику та створити науково-технічне забезпечення, що може бути використано при проектуванні дизелів зі схожими геометричними розмірами та форсованістю у майбутньому

Ключові слова: високообертовий малолітражний дизель, колінчастий вал, крутильні коливання, силіконовий демпфер, критерії проектування, амплітуда, дотичні напруження

1. Introduction

In order to substantially improve capacity of the adjusted automotive four-cylinder engines, there is a method to increase the number of cylinders. When six cylinders are chosen, the engine can be inline, V-type, or with opposed cylinders. Most global manufacturers prefer a six-cylinder inline engine because it is entirely balanced externally, has a uniform operation of cylinders and a simple design. The only drawback of the inline six-cylinder engine, especially small diesel, is the low rigidity of the crankshaft, resulting in the torsional vibrations.

To absorb torsional vibrations of the crankshaft system, internal friction dampers (made of rubber) and liquid-friction dampers (made of silicone) are applied. Silicone dampers are more reliable in the operation of automobile engines. However, silicone dampers have a limited life cycle and require periodic inspection. Maintenance and replacement of liquid is required to restore the damping properties. Therefore, the dampers must be positioned so that they are accessible for maintenance.

2. Literature review and problem statement

Paper [1] shows that the silicone damper of the six-cylinder diesel engine made by BMW (V_h=2.99 l, D=84 mm, S=90 mm, N_e=190 kW, n=4,000 min^{-1}) is mounted onto the crankshaft front end; it has a sophisticated design, while the casing is at the same time a pulley for the drive of auxiliary assemblies. Such an arrangement of the damper ensures maximum operational performance; there is access in order to service it, and, therefore, it is interesting to methodologically process its design. However, the difference between the designs of the crankshaft and damper does not make it possible to apply the solutions proposed in paper [1]. The crankshaft has four unloading counterweights while the projected crankshaft has twelve of them. Therefore, it is proposed to use the simpler and adjusted design of the damper.

Paper [2] studied torsional vibrations of the crankshaft with twelve unloading counterweights of the six-cylinder in-line diesel engine (V_h=6.21 l, n=2,200 min^{-1}, order 1-5-3-6-2-4), whose capacity is twice larger than the object of research. The authors argue that the computer simulation of dynamic processes is the most effective technique at the beginning of the design process of the engine. Simulation also makes it possible to examine and compare different approaches and structural solutions. A combination of methods for calculating the torsional vibrations with the method of finite elements (ANSYS) allows determining the stresses arising from the torsional vibrations. The difference in the capacity of the engine and the crankshaft rotation frequency produce a different pattern of resonance modes, and that imposes the limitations on the application of research results.

The results reported in paper [3] describe methodological approaches, calculation algorithms for dampers of fluid and internal friction, as well as the estimated and experimental study into dampers of the crankshaft torsional vibrations of petrol engines made at Zavolzhsky Motor Plant (ZMZ) (Russian Federation). The example used is the constructed dependences of amplitudes of the angular vibrations of the crankshaft front end of the engine ZMZ-4062.10 (made in Russian Federation) on the frequency of its rotation for an engine without a damper and an engine with a silicone damper. Results of the reported study into tangential stresses in the elements of the crankshaft does not allow us to apply the results to the crankshaft in the designed engine. This is due to the fact that the diesel engine is exposed to much larger loads than those in the gasoline engine.

Paper [4] studied the impact of various design parameters of the crankshaft on their torsional vibrations. For the sake of convenience, the authors introduced dimensionless complexes that depend on one or more parameters. The proposed dimensionless complexes do not make it possible to determine the structural characteristics required for the design. The result of numerical calculation of torsional vibrations of the crankshaft system in multi-cylinder engine is the determined frequencies of dangerous harmonics. The authors noted a heavy impact of the rotation frequency on the amplitudes and tangential stresses. However, the results of the calculations do not make it possible to determine design parameters for other engines.

Of considerable interest in the diagnosis of ICE is the technique, proposed in [6], for detecting the malfunctions related to the torsional vibrations of the crankshaft, and methods for the detection of engine failures. Authors of a given work, based on the six-cylinder in-line engine (SL90, United Kingdom, V_h=8.8 l, N_e=174 kW), conducted calculations of the torsional crankshaft vibrations, and defined the resonance modes. Significant differences in the dimensions and frequency of the crankshaft rotation does not make it possible to apply the results for the designed diesel engine.

The estimation model for choosing the basic parameters of a silicone damper for a six-cylinder marine inline engine [7] included substantial simplification for its linearization, which is unacceptable for the high-speed diesel engine.

Authors of work [8] reported the results of calculation and experimental research into torsional vibrations of the six-cylinder in-line turbocharged engine and showed results of torsional crankshaft vibrations dampening using a silicone damper. However, the mass moments of inertia, section pliability, and the dimensions of the damper, are not acceptable for the high-speed diesel engine.

Further development of approaches to the design of silicon dampers was presented in paper [9]. The authors analyzed various directions and proposed the simplest and most rational method of calculation. However, attempts to use a given method yield large errors.

Generalization of results of the cited research suggests that existing approaches do not make it possible to solve a scientific and technical task on designing the elements of the crankshaft for a high-speed small diesel engine. Solving it requires the estimation and experimental studies with respect to all peculiarities in the design of the crankshaft.

3. The aim and objectives of the study

The aim of this study is to choose the design criteria for the elements of the crankshaft system for the high-speed small diesel engine. The criteria to be selected would make it possible to develop a design of the silicone damper with the required dimensions, ensuring sufficient rigidity of the crankshaft front end and the efficiency of torsional vibrations dampening.

To achieve the set aim, the following tasks have been solved:

– to select the criteria for the design of structural elements of the crankshaft system;

– to perform calculations on determining the structural characteristics of the elements of the crankshaft system based on the selected criteria;

– to analyze the results obtained in the course of estimation.

4. Preparation of input data for the simulation of torsional crankshaft vibrations

Using the example of designing the diesel 6ChN8.8/8.2 (the series DA15, «Diesel group», Ukraine) of our own design, we devised a sequence of stages to address the challenges of this research.

Performing the calculation in the programming environment AVL EXCITE Designer (version 2011) allowed us to obtain preliminary results for further research in this direction and to determine the appropriate location and type of the applied damper of torsional vibrations.

We subsequently derived the refined values for rigidity of the shaft cranks [10], which enabled a more precise calculation of torsional vibrations of the crankshaft system in the designed engine.

The magnitudes obtained in paper [10] differed from the values derived from various empirical formulae, however, some of them produced fairly similar results. This observation led to the following idea: by taking a formula according to which the obtained values for rigidity of the crank are the closest to those derived experimentally, to determine a correction factor. Introduction of the correction factor would bring the estimation results closer to experimental values. Such an innovative technique might in the future, when designing new engines without the crankshafts implemented in a metal, with similar geometrical dimensions of the crank, help obtain the refined value for rigidity. Note, however, that the refined value of rigidity would hold only to the cranks that have similar geometrical dimensions to the crank of the designed crankshaft (Fig. 1).

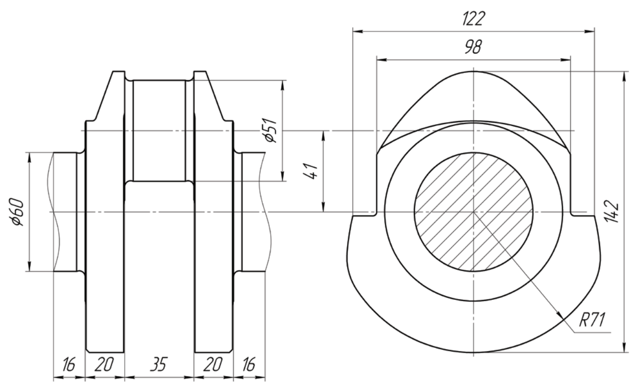

Fig. 1. Geometrical dimensions of the crank in the designed crankshaft

The W. Coeur Wilson formula was chosen [11]:

$$K_t = \frac{G\pi}{32\left[\frac{L_j + 0.4D_j}{D_j^4 - d_j^4} + \frac{L_c + 0.4D_c}{D_c^4 - d_c^4} + \frac{R - 0.2\left(D_j + D_c\right)}{L_w W^3}\right]}, \quad (1)$$

where K_t is the rigidity of the crank, G is the shear modulus, L_j, D_j, d_j are, respectively, the length, outer diameter, and the diameter of the oil opening of the radical neck, L_c, D_c, d_c are, respectively, the length, outer diameter, and the diameter of the oil opening of the crank neck, R is the radius of the crank, L_w is the thickness of the cheek, W is the width of the cheek.

The numerical value for the rigidity of the crank, obtained using a given formula, was K_t=524,767 N·m/rad, which is 1.05 times larger than the corresponding value obtained in paper [10]. Consequently, by adopting the correction factor ζ=0.952, we shall obtain the estimated value for rigidity, which is different by 0.017 % from that derived experimentally. That produces a sufficient accuracy for calculating the torsional vibrations at the preliminary design stage.

As a result, the formula for determining the rigidity of the crank in the crankshaft will take the following form:

$$K_{kr} = K_t \cdot \zeta. \quad (2)$$

Building the 3D models in the programming packages SolidWorks and Autodesk Inventor allowed us to more accurately determine the mass moments of inertia of the reduced masses, taking into consideration different design features (the value of 0.0118 kg·m^2, obtained using the 3D models, compared with 0.0138 kg·m^2 when applying the empirical formula by Zimanenko). Such design features are the oil openings in the cranks, fillets, bevels on the cheeks, splines on the front end of the crankshaft, pulley, and damper casing, etc.

5. Estimation study into torsional crankshaft vibrations

At this stage of the study, we conducted final calculation based on the procedure from [12]. In the course of the calculation, we accepted the circuit of the vibratory system of crankshaft in accordance with Fig. 2.

The initial data to carry out the calculation were taken from Table 1, 2.

When performing the calculation, we took into consideration that the crankshaft at operation is exposed to the forced, connected, and parametric vibrations. Motion of a separate mass, joined with other soft weightless regions, is described by a second-order differential equation [12]. The integration of parametric fluctuations introduces non-linearity to the system, which in turn leads to the natural frequencies of oscillatory system w$_{F0i}$, unstable over time, where i_{max} is the number of oscillating masses.

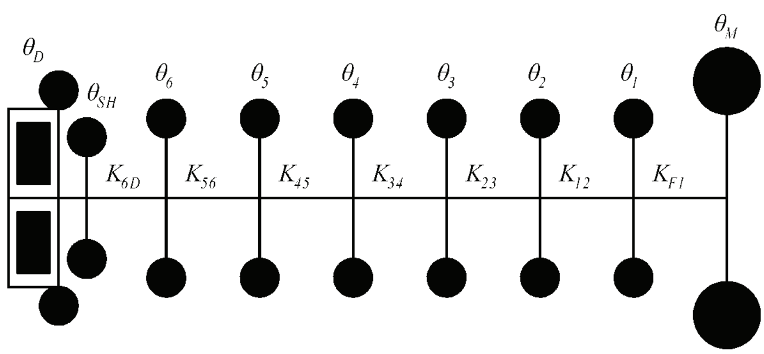

Fig. 2. Schematic of the oscillatory system of the crankshaft: θ_D – mass moment of the damper inertia; θ_{SH} – mass moment of the pulley inertia; $\theta_6...\theta_1$ – mass moments of the cranks inertia; θ_M – mass moment of the flywheel inertia; $K_{6D}...K_{F1}$ – rigidity of sections of the crankshaft

Table 1

Mass moments of inertia of the concentrated masses of the estimated model

θ of damper casing, kg·m^2	θ of damper flywheel, kg·m^2	θ of pulley, kg·m^2	θ of the concentrated masses of cranks, kg·m^2						θ of flywheel, kg·m^2
			1	2	3	4	5	6	
0.026	0.083	0.004	0.012	0.012	0.012	0.012	0.012	0.012	0.745

Table 2

Rigidity of sections of the estimated model

K_{6D}, N·m/rad	K_{56}, N·m/rad	K_{45}, N·m/rad	K_{34}, N·m/rad	K_{23}, N·m/rad	K_{12}, N·m/rad	K_{F1}, N·m/rad
48,303	499,492	499,492	499,492	499,492	499,492	850,145

Results of the performed series of calculations over the entire range of possible rotation frequencies of the engine crankshaft are shown in Fig. 3, 4. Calculations were conducted for an oscillatory system without a damper, as well as for the system with a damper.

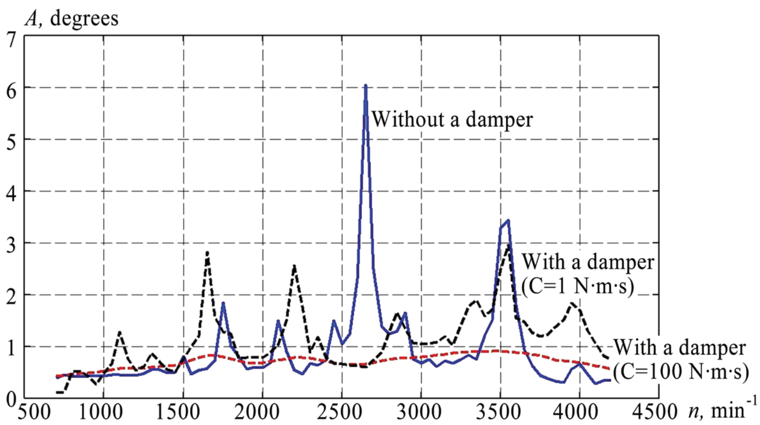

Fig. 3. Dependence diagram of maximum amplitudes of vibrations on the crankshaft rotation frequency

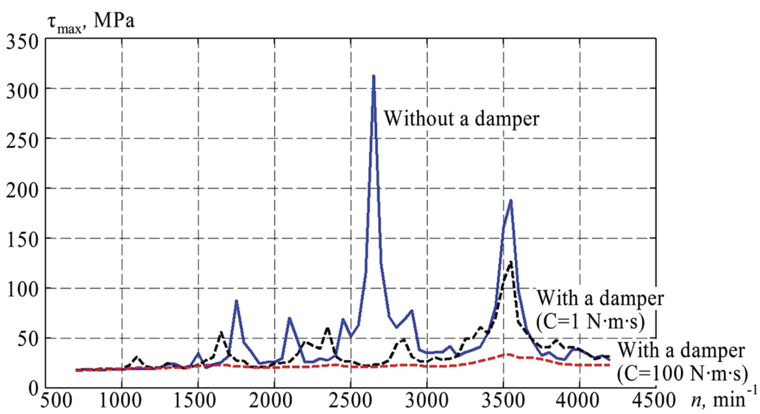

Fig. 4. Dependence diagram of the maximum tangential stresses on the crankshaft rotation frequency

The following variable factors were selected: a mass moment of inertia of the damper casing (θ_{dc}), a mass moment of inertia of the damper flywheel (θ_{df}), and rigidity of the crankshaft front end (K_{cfe}). In this case, K_{cfe} changed due to a change in the diameter of the crankshaft. Values for the variable factors were consolidated in Table 3, whence we took data for the subsequent estimating three-factor experiment. The same Table 3 gives the corresponding normalized values (X_i)) for the varied factors listed above, which is necessary for the universalization of the mathematical notation of the experiment design.

Table 3

Values of variable factors

Experiment No.	θ_{df}, kg·m^2	θ_{dc}, kg·m^2	K_{cfe}, N·m/rad	X_1	X_2	X_3
1	0.083	0.026	200,000	1	1	1
2	0.083	0.026	20,000	1	1	–1
3	0.083	0.022	200,000	1	–1	1
4	0.083	0.022	20,000	1	–1	–1
5	0.067	0.026	200,000	–1	1	1
6	0.067	0.026	20,000	–1	1	–1
7	0.067	0.022	200,000	–1	–1	1
8	0.067	0.022	20,000	–1	–1	–1
9	0.075	0.024	110,000	0	0	0
10	0.083	0.024	110,000	1	0	0
11	0.067	0.024	110,000	–1	0	0
12	0.075	0.026	110,000	0	1	0
13	0.075	0.022	110,000	0	–1	0
14	0.075	0.024	200,000	0	0	1
15	0.075	0.024	20,000	0	0	–1

When choosing the input parameters for the damper involved in the calculations, we primarily took into consideration the dimensions. Since the problem implied that we had to place the damper inside the predefined (rather limited) space, the mass moments of inertia of its components, when performing the calculation, were selected to be maximally permissible based on the structural constraints. The value for dissipative coefficients C were taken to be equal to 1 and 100 (N·m·s), since the expected value of a given coefficient for the examined crankshaft system, according to ref. [3], is (20–40) N·m·s. The range accepted overlaps the expected values for the coefficient of dissipation. It is obvious that the higher the value of the dissipative coefficient, the more efficient the damper. However, at a certain value of this coefficient, there may occur a situation at which it would be impossible to technologically execute the damper with the required dimensions.

We noticed in the diagram in Fig. 4 an increase in the tangential stresses in the region of the crankshaft front end leading to exceeding the limit of strength of a material for the designed crankshaft. This was the cause for performing additional calculations in order to choose structural characteristics of the damper and the front end of the crankshaft that involved 3 varying factors.

Fig. 5 shows a Campbell's diagram, according to the established form [13], which makes it possible to determine the resonance modes needed for further calculations. By analyzing the data obtained, it was found that the dangerous harmonics are 3, 4, 5, and 6 for natural frequency I; and 6, 7, 5, and 9 – for II.

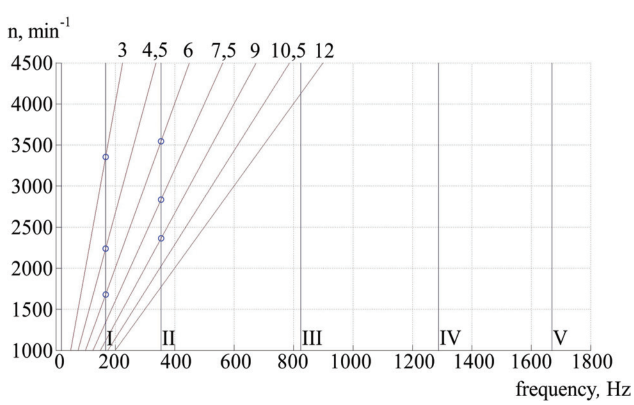

Fig. 5. Frequency diagram of torsional vibrations: 3...12 — harmonics; I...V — natural vibration frequencies

Given this, we can adopt a procedure for the calculation of amplitude and stresses of torsional vibrations not for the entire range of engine speed modes, but only for the resonant rotation frequencies.

These frequencies are derived from formula:

$$n_{rez} = \frac{30\omega_i}{N_g \pi},$$

where ω_i is the natural crankshaft vibration frequency I or II in rad/s, determined from the eigenmatrix of the system, N_g is the ordinal number of harmonic. Then, for example, for the 6-th harmonic of the first natural frequency:

$$n_{rez} = \frac{5}{\pi}\omega_I.$$

Results of calculations based on the proposed experiment design are summarized in Tables 4–7. It should be noted that the value for the coefficient of hydraulic resistance (dissipative coefficient) was taken to be equal to $C=1$ N·m·s. This is explained by that its magnitude affects only a quantitative pattern of the result, but has almost no effect on quality, as demonstrated in the comparison in Tables 4–7. Tables 6 and 7 are based on the results of calculations with a value of dissipative coefficient $C=10$ N·m·s.

The calculation results, obtained according to the proposed design of experiment, were unambiguous, and could be easily analyzed, resulting in the discontinuation of calculating the coefficients of approximating dependences and deriving the regression equations. Thus, an analysis of the results shows that the minimum values for the amplitudes of vibrations and tangential stresses in the elements of the crankshaft are achieved at maximum values of coefficient θ_{df} and minimum values of coefficient θ_{dc}. In addition, the data given in Tables 4–7 demonstrate that increasing the rigidity of the crankshaft front end «drives» certain dangerous modes of engine operation outside the region of the actual rotation frequencies. Given this, we can conclude that the most preferred structural characteristics of the damper and the crankshaft front end are those that were obtained for mode 1 (Table 3).

Fig. 6 shows results of calculations of the selected mode in the form of a distribution field of values for the tangential stresses relative to the value τ_{rec}=25 MPa. These calculations were performed based on the procedure from [12]. τ_{rec} is the level of tangential stresses exceeding which at resonance modes is not recommended [14].

Table 4

Results of calculations of the tangential stresses based on the adopted experiment design with a value of dissipative coefficient $C=1$ N·m·s

No. of entry	Natural frequencies, rad/s		Dangerous harmonics						Dangerous harmonics					
			3	4.5	6	6	7.5	9	3	4.5	6	6	7.5	9
	I	II	Dangerous rotation frequencies, min^{-1}						Tangential stresses, MPa					
1	1,260	3,274	4,010	2,673	2,005	5,211	4,169	3,474	125	125	154	89	48	81
2	793	1,944	2,523	1,682	1,261	3,094	2,475	2,063	27	21	21	138	41	51
3	1,316	3,378	4,190	2,793	2,095	5,376	4,300	3,584	131	127	157	85	62	96
4	857	1,952	2,727	1,818	1,363	3,106	2,485	2,071	26	23	22	125	37	48
5	1,260	3,274	4,010	2,673	2,005	5,211	4,169	3,474	125	125	154	89	48	81
6	793	1,944	2,523	1,682	1,261	3,094	2,475	2,063	27	21	21	138	41	51
7	1,316	3,378	4,190	2,793	2,095	5,376	4,300	3,584	131	127	157	85	62	96
8	857	1,952	2,727	1,818	1,363	3,106	2,485	2,071	26	23	22	125	37	48
9	1,231	2,818	3,919	2,613	1,960	4,484	3,587	2,990	104	99	120	91	36	41
10	1,231	2,818	3,919	2,613	1,960	4,484	3,587	2,990	104	99	120	91	36	41
11	1,231	2,818	3,919	2,613	1,960	4,484	3,587	2,990	104	99	120	91	36	41
12	1,201	2,781	3,822	2,548	1,911	4,426	3,541	2,951	103	97	117	113	36	45
13	1,264	2,861	4,023	2,682	2,012	4,553	3,643	3,036	107	102	123	78	35	41
14	1,287	3,322	4,097	2,732	2,049	5,287	4,230	3,525	127	127	155	86	55	93
15	823	1,947	2,619	1,746	1,309	3,099	2,480	2,066	32	32	23	132	38	49

Table 5

Results of calculation of the amplitudes based on the adopted experiment design with a value of dissipative coefficient $C=1$ N·m·s

No. of entry	Natural frequencies, rad/s		Dangerous harmonics						Dangerous harmonics					
			3	4.5	6	6	7.5	9	3	4.5	6	6	7.5	9
	I	II	Dangerous rotation frequencies, min^{-1}						Amplitudes, degrees					
1	1,260	3,274	4,010	2,673	2,005	5,211	4,169	3,474	2.5	3.5	4.1	1.5	1.3	1.6
2	793	1,944	2,523	1,682	1,261	3,094	2,475	2,063	1.6	1.6	1.0	3.2	1.4	0.8
3	1,316	3,378	4,190	2,793	2,095	5,376	4,300	3,584	2.6	3.5	4.3	1.4	1.6	2.4
4	857	1,952	2,727	1,818	1,363	3,106	2,485	2,071	1.2	1.5	1.5	3.0	1.0	1.1
5	1,260	3,274	4,010	2,673	2,005	5,211	4,169	3,474	2.5	3.5	4.4	1.5	1.3	1.6
6	793	1,944	2,523	1,682	1,261	3,094	2,475	2,063	1.6	1.6	1.5	3.2	1.4	0.8
7	1,316	3,378	4,190	2,793	2,095	5,376	4,300	3,584	2.6	3.5	4.3	1.4	1.6	2.4
8	857	1,952	2,727	1,818	1,363	3,106	2,485	2,071	1.5	1.5	1.5	3.0	1.0	1.1
9	1,231	2,818	3,919	2,613	1,960	4,484	3,587	2,990	2.5	3.3	3.7	2.0	1.1	0.9
10	1,231	2,818	3,919	2,613	1,960	4,484	3,587	2,990	2.5	3.3	3.7	2.0	1.1	0.9
11	1,231	2,818	3,919	2,613	1,960	4,484	3,587	2,990	2.5	3.3	4.0	2.0	1.1	0.9
12	1,201	2,781	3,822	2,548	1,911	4,426	3,541	2,951	2.5	3.3	3.7	2.1	1.1	0.9
13	1,264	2,861	4,023	2,682	2,012	4,553	3,643	3,036	2.5	3.3	4.0	1.6	1.1	0.8
14	1,287	3,322	4,097	2,732	2,049	5,287	4,230	3,525	2.6	3.5	4.3	1.5	1.5	2.1
15	823	1,947	2,619	1,746	1,309	3,099	2,480	2,066	1.5	1.6	1.5	3.4	1.0	1.1

Table 6

Results of calculations of the tangential stresses based on the adopted experiment design with a value of dissipative coefficient $C=10$ N·m·s

No. of entry	Natural frequencies, rad/s		Dangerous harmonics						Dangerous harmonics					
			3	4.5	6	6	7.5	9	3	4.5	6	6	7.5	9
	I	II	Dangerous rotation frequencies, min^{-1}						Tangential stresses, MPa					
1	1,260	3,275	4,012	2,675	2,006	5,212	4,169	3,474	61	53	59	30	35	36
2	795	1,937	2,532	1,688	1,266	3,083	2,466	2,055	22	20	19	57	25	29
3	1,317	3,378	4,193	2,795	2,096	5,377	4,301	3,584	64	55	62	30	43	42
4	861	1,941	2,742	1,828	1,371	3,090	2,472	2,060	25	20	19	51	25	28
5	1,260	3,275	4,012	2,675	2,006	5,212	4,169	3,474	61	53	59	30	35	36
6	795	1,937	2,532	1,688	1,266	3,083	2,466	2,055	22	20	19	57	25	29
7	1,317	3,378	4,193	2,795	2,096	5,377	4,301	3,584	64	55	62	30	43	42
8	861	1,941	2,742	1,828	1,371	3,090	2,472	2,060	25	20	19	51	25	28
9	1,233	2,814	3,926	2,617	1,963	4,479	3,583	2,986	38	33	38	28	29	23
10	1,233	2,814	3,926	2,617	1,963	4,479	3,583	2,986	38	33	38	28	29	23
11	1,233	2,814	3,926	2,617	1,963	4,479	3,583	2,986	38	33	38	28	29	23
12	1,203	2,778	3,828	2,552	1,914	4,421	3,537	2,947	37	32	37	30	28	22
13	1,266	2,857	4,031	2,687	2,015	4,548	3,638	3,032	39	34	39	28	29	22
14	1,288	3,323	4,100	2,733	2,050	5,288	4,230	3,525	62	54	61	30	39	38
15	826	1,939	2,630	1,754	1,315	3,086	2,469	2,057	21	20	19	54	25	29

Table 7

Results of calculation of the amplitudes based on the adopted experiment design with a value of dissipative coefficient $C = 10$ N·m·s

No. of entry	Natural frequencies, rad/s		Dangerous harmonics						Dangerous harmonics					
			3	4.5	6	6	7.5	9	3	4.5	6	6	7.5	9
	I	II	Dangerous rotation frequencies, min^{-1}						Amplitudes, degrees					
1	1,260	3,275	4,012	2,675	2,006	5,212	4,169	3,474	1.4	1.5	1.7	0.6	0.8	1.0
2	795	1,937	2,532	1,688	1,266	3,083	2,466	2,055	0.6	0.6	0.6	1.0	0.6	0.7
3	1,317	3,378	4,193	2,795	2,096	5,377	4,301	3,584	1.4	1.5	1.7	0.6	1.0	1.2
4	861	1,941	2,742	1,828	1,371	3,090	2,472	2,060	0.6	0.6	0.6	0.9	0.7	0.7
5	1,260	3,275	4,012	2,675	2,006	5,212	4,169	3,474	1.4	1.5	1.7	0.6	0.8	1.0
6	795	1,937	2,532	1,688	1,266	3,083	2,466	2,055	0.6	0.6	0.6	1.0	0.6	0.7
7	1,317	3,378	4,193	2,795	2,096	5,377	4,301	3,584	1.4	1.5	1.7	0.6	1.0	1.2
8	861	1,941	2,742	1,828	1,371	3,090	2,472	2,060	0.6	0.6	0.6	0.9	0.7	0.7
9	1,233	2,814	3,926	2,617	1,963	4,479	3,583	2,986	0.9	1.1	1.2	0.9	0.9	0.6
10	1,233	2,814	3,926	2,617	1,963	4,479	3,583	2,986	0.9	1.1	1.2	0.9	0.9	0.6
11	1,233	2,814	3,926	2,617	1,963	4,479	3,583	2,986	0.9	1.1	1.2	0.9	0.9	0.6
12	1,203	2,778	3,828	2,552	1,914	4,421	3,537	2,947	0.9	1.1	1.2	0.9	1.0	0.6
13	1,266	2,857	4,031	2,687	2,015	4,548	3,638	3,032	0.9	1.1	1.2	0.8	0.9	0.6
14	1,288	3,323	4,100	2,733	2,050	5,288	4,230	3,525	1.4	1.5	1.7	0.6	0.9	1.2
15	826	1,939	2,630	1,754	1,315	3,086	2,469	2,057	0.6	0.6	0.6	0.9	0.7	0.7

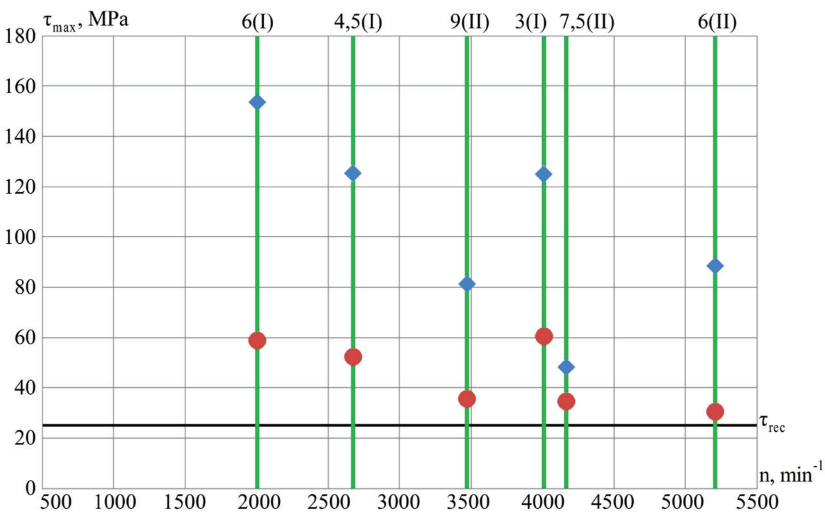

Fig. 6. Distribution field of the estimated values of tangential stresses with the selected characteristics of elements from the crankshaft system: ◆ — $C = 1$ N·m·s; ● — $C = 10$ N·m·s

6. Discussion of results of studying the torsional vibrations of the crankshaft system of a high-speed small diesel engine

When designing the high-speed small inline six-cylinder diesel engine 6ChN8.8/8.2 we encountered a problem related to the lack of a selection procedure for the structural characteristics of the crankshaft system elements, adapted particularly for a small high-speed internal combustion engine. One of the stages in solving this task was the stage of calculating the torsional vibrations of the crankshaft system in the designed diesel engine.

A practical task that involves the estimation of torsional vibrations has three aspects: determining the resonance frequencies of crankshaft rotation, determining the absolute amplitudes of forced vibrations of the shaft ends, numerical estimation of the emerging tangential mechanical torsional stresses in cranks.

At present, the calculation of torsional vibrations employs and widely uses known specialized methods based on the consideration of equivalent (or reduced to them) oscillatory systems. That implies considerable theoretical simplifications, reductions, and assumptions, however these methods of calculation have remained complicated and cumbersome, which hinders the execution of the optimization design tasks based on them. Earlier calculations in line with the procedure by Krylov State Research Centre prove this fact. There are also fully-fledged software packages, such as AVL EXCITE Designer, which make it possible to solve a given task.

The data shown in Fig. 6 demonstrate that even at $C = 10$ N·m·s the values for tangential stresses closely approach the level of the recommended values for tangential stresses. It is obvious that while ensuring the value of the dissipative coefficient within the specified range of (20–40) N·m·s, the tangential stresses will be below the maximum permissible ones. Fig. 6 also shows that the most dangerous is the resonant harmonic mode of the third order along natural frequency I. At the stage of adjusting the damper, on order to experimentally determine the damping coefficient, this very mode will be employed to adjust the experimental setup for testing the effectiveness of silicone dampers [15]. In the course of our study we established that for a small high-speed diesel engine the determining conditions for designing the crankshaft system are the overall dimensions and the mass of elements of the crank mechanism. Therefore, it is suggested using, as the criteria for the design of crankshaft system elements, along with dynamic loads, a mass moment of the casing inertia and the damper flywheel, as well as the diameter and rigidity of the crankshaft front end.

The tested fairly simple method [12] for the calculation of real amplitudes and mechanical stresses of torsional vibrations in the crankshaft of the internal combustion engine based on the numerical solution to the high-level system of nonlinear differential equations in the form of a model in the state space has shown its effectiveness and capability to compete both with specialized procedures and programming packages for the calculation of torsional vibrations.

By comparing results, obtained using a given method, with the results of calculating the torsional vibrations in the programming complex AVL EXCITE Designer (version 2011), it is demonstrated that the frequencies of resonance modes for a single-node shape of vibrations are slightly different ($n = 1{,}600$ min^{-1} and $n = 2{,}460$ min^{-1} for calculations employing AVL EXCITE Designer compared to $n = 1{,}630$ min^{-1} and $n = 2{,}280$ min^{-1} based on the calculations in line with a given method). The same differences are observed for a two-node shape of vibrations. Apparently, such differences are explained by the fact that when entering the source data to the programming complex AVL EXCITE Designer, one does not take into consideration different design features of the crankshaft, specifically fillets, bevels on the cheeks, etc. When calculating in line with the method from ref. [12], the initial data included the mass moment of the crank inertia, determined using a 3D simulation while the value for the rigidity of the crank was determined experimentally.

It is obvious that a given method could be successfully used to solve the optimization problems on selecting and substantiating the design parameters for the dampers of torsional vibrations. However, it is worth noting that a single computation cycle at a modern four-core computer takes several hours, which is a drawback no matter how small it might seem.

The results of calculations that we obtained based on the proposed procedure, and the structural characteristics, selected on the basis of these results, testify to the correctness of the chosen approach.

It should be noted that a given procedure, although tested using a high-speed small diesel, could be successfully applied in order to choose design characteristics for the elements of the crankshaft and engines with a different dimensionality and forcing, to be supplemented with certain refinements.

Thus, we consider that there is a need to accurately determine the rigidity of the shaft crank. However, this strategy could be implemented only if there is a prototype of the designed engine with similar crank size, albeit the number of cranks differs from the required one. In the absence of the prototype, this strategy cannot be implemented and thus we shall have to apply empirical formulae, which would naturally reduce the accuracy of calculations.

7. Conclusions

1. In order to design elements for the crankshaft system of a small high-speed diesel engine with acceptable stresses from torsional vibrations, there arose the need to specify the design procedure. Refining the procedure implied the introduction of criteria for designing the elements of the crankshaft system in a small high-speed diesel engine. These criteria are the overall dimensions and the mass of the damper, dynamic loads that occur in the crankshaft system at engine operation, as well as the dimensions of other elements of this system.

2. Employing the proposed criteria made it possible to implement the following structural characteristics for the diesel 6ChN8.8/8.2: mass moment of inertia of the damper flywheel $\theta_{df}=0.083$ kg·m^2, mass moment of inertia of the damper casing $\theta_{dc}=0.026$ kg·m^2, diameter of the crankshaft tip $d_{cfe}=0.042$ m, and rigidity of the crankshaft front end $K_{cfe}=200{,}000$ N·m/rad.

3. The chosen structural characteristics make it possible to solve the further task on designing the damper to absorb torsional vibrations in the crankshaft system of the diesel engine 6ChN8.8/8.2.

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