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DYNAMIC NUMERICAL STRESS ANALYSIS OF A CRANKSHAFT OF INTERNAL COMBUSTION ENGINE

Summary. Dynamic numerical stress analysis of a crankshaft subjected to load at selected operational points of a diesel engine is presented in this paper. The calculations and the analyses were carried out for six values of engine rotational speed and for two temperature values of engine structural elements. At each operating point of the engine, the piston-crank system was loaded with maximal gas pressure force, and additionally, inertia forces resulting from rotational speed of the crankshaft were taken into consideration. The analysis was carried out to obtain the distribution of the stress and to indicate critical areas where concentration of the stress may occur. In addition, the analysis was extended to other operational factors, such as the determination of the natural frequency of vibrations and effects of maximal torque on torsion of the crankshaft.

Keywords: crankshaft, FEM, ANSYS, numerical calculations, thermal calculations

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1. INTRODUCTION

In an internal combustion engine equipped with a piston-crank system, exhaust gas pressure is converted into torque. Its direction, turn, and value of the load are changing in time and can result in material fatigue. Phenomenon of material fatigue is caused by cyclic changes in the stress; however, the course of this phenomenon also depends on other factors, e.g., temperature [6]. A significant part of this stress is caused by torque, which is related to the current load on the engine. The value of forces acting on the crankshaft depends on such factors as geometrical parameters of the engine (e.g., crank radius), mass of individual components of the crank-piston system (depending on the material used, among others), technical parameters of the engine, such as mean effective pressure, related to the efficiency of the engine and having an effect on the value of gas forces acting on the piston, and through the connecting rod, on the crankshaft. In the course of engine operation, its crank mechanism is loaded by many forces, the most important from them are:

- combustion gas pressure force,
- inertia force,
- friction force.

The pressure of gases in the cylinder changes, and is the highest at the beginning of the expansion stroke.

The inertia forces are present during the movement of masses in the piston-crank mechanism, and their values are a function of the mass of its components and accelerations of its individual elements.

All masses of the crank mechanism can be grouped into:

- the masses concentrated in the center of the piston pin, and performing reciprocating movement (piston with its pin and rings),
- the masses concentrated in the center of the crankpin of the crankshaft, performing rotational motion (crankshaft),
- the masses of the conrod – performing complex moves (composed of reciprocating and rotational moves). For this reason, masses of the conrod are replaced by a system of two equivalent masses, and it is assumed that part of this mass is concentrated in the center of the piston pin and performs reciprocating movement, and the other part of the mass is concentrated in the center of the crankshaft's journal and performs rotational movement.

The friction forces are considerably restricted by lubrication, and compared to the others, such forces reach small values, and due to this, in the majority of the analyses, these are neglected.

The crankshaft must be robust enough to carry the load acting on it and ensure reliable operation of the internal combustion engine. High-volume production of components as the crankshaft requires its proper design and preliminary testing before putting it into production. In such cases, numerical analyses allow verification of the crankshaft in terms of strength as well as optimization of its shape and dimensions, reducing manufacturing costs.

The Finite Element Method is a type of numerical analysis allowing finding, in an approximate and discrete way, a function describing the behavior of a given system (boundary problem) under certain given conditions by dividing a complex problem (e.g., with complex shape) into a finite number of elements (having a simpler geometry).

In the study [13] its authors performed a stress analysis of the crank system of a diesel engine, taking into consideration flexibility of all elements of the system. Most studies contain a single member treated as deformable only; this approach limits the scope of the strength analysis only to the case of this one single member. The flexibility of all elements ensures the possibility of simultaneous strength analysis of the entire set of interacting elements.

In the study [11] optimization of the crankshaft was performed in range of its geometry and shape without effects on the structure of the cylinder block and cylinder head of the engine. The optimization was carried out mainly on the shape of counterweights, removing parts of material with slight changes in the strength properties simultaneously allowing for a reduction in the weight of the complete crankshaft by 4.37%.

The numerical analysis [10] may also comprise strength analysis of the crankshaft at several points of engine operation, and not only in extreme cases such as the TDC and position for maximum tangential force. In this study, there were also undertaken calculations for three different materials.

The Finite Element Method also allows for predicting areas most susceptible to damage. In the paper [1], the service life of the crankshaft of a six-cylinder diesel engine of a truck was determined, modeling the growth of fatigue cracks with the use of linear fracture mechanics. The results of the calculations were confirmed by experimental tests.

The results of the numerical calculations using the Finite Element Method are implemented for analysis of various types of damage and to identify areas on the crankshaft structure that are weak in terms of strength and which require improvements. The following methodology was adopted in the article [8]: a model of the crankshaft is generated in a computer system destined for advanced computer-aided design (CAD). The Finite Element Analysis is next performed in the ANSYS system under static and dynamic conditions to obtain changes in the stress at key areas. Data of the engine, boundary conditions, and mechanical properties are taken into account as the input data.

The FEM [12] allows examination of an effect of inclusions, e.g., manganese and sulfur, in the structure of the steel of the crankshaft on its strength properties. In the article, the structural model was precisely replicated from the real geometry of a damaged crankshaft and divided into octahedral, tetrahedral elements having the number of nodes equal to ten (ten-node tetrahedral). Analysis of convergence of the mesh showed that values of reduced stress are convergent when the size of the FEM element is 2 mm, and appears in the same areas as the real damages. This will allow for predicting areas of probable damage.

Simulation methods are also useful in the course of measuring of the crankshafts. In the study [9], it was proposed a measurement system developed on the basis of numerical calculations to support the crankshaft with the use of flexible supports. This allowed elimination of effects of reaction forces of the supports and thus elimination of deformations of the crankshaft regardless of any possible deviations, e.g., concentricity of the main journals. Values of the forces were calculated using the Finite Element Method, based on these values, it was found that reaction forces change not only on the supports but also in cases when the angular position of the crankshaft is changing.

The Finite Element Method enables thermal simulations as well. In the article [7], there is presented analysis of displacements of the crankshaft's axis under influence of temperature in a low-speed marine engine. Information about the thermal displacements of the axis of the power transmission system is important when determining the linearity of the crankshaft and its bearings. Thermal displacements of the crankshaft calculated by numerical analysis have shown a higher value than recommended by the manufacturer. The difference is not large (less than 20%), but it may be a source of additional bending torque and shearing force acting between the crankshaft and the cylinder block. The calculations have shown that the manufacturer's assumption about parallel displacement of the axis of the crankshaft is incorrect.

The modeling may include analyses of low temperatures [3]. In the article is presented a method to determine the shrinkages of the main elements of the V6 engine (the crankshaft mainly) with a small displacement for the cold phase at extremely low temperatures. The phenomenon of thermal shrinkage generates mechanical wear of surfaces of rotating elements in components of the engine. Based on calculations using the ANSYS program, in the study it was proposed a suitable mathematical model enabling selection of the fits at extremely low temperatures.

In many cases, the phenomenon of vibrations can have decisive meaning in the proper functioning of the crankshaft [5]. The vibrations may cause the combustion engine to malfunction. The occurrence of phenomena of resonance may cause an increase in amplitude of the vibrations, which may result in damage to the crankshaft and related components. Regardless of the dynamics of the system in which the engine operates, the greatest threat is caused by torsional vibrations of the crankshaft.

In the article is described the influence of selection of computational mesh on results obtained during simulation of the crankshaft of a diesel engine. Presented here, the methodology may be useful for performing calculations for a new crankshaft. In the case of determination of mode of vibrations only, a smaller mesh with a larger element can be used. It should be emphasized, however, that the selection of elements of the mesh and its density must be consistent with the geometric model. The smaller mesh elements, the more accurate results, which has an effect on significant prolongation of the calculations.

The article [2] describes a study on the causes of premature failure of a high-pressure diesel engine crankshaft. It was noted that all crankshafts failed in the same part, namely the first crankpin. Dynamic analysis and finite element modeling were conducted to determine the stress state in the crankpin. The results of the finite element method showed that the crankpin throw was the most susceptible to cracking. The Soderburg diagram for the analyzed crankshaft indicated that the operational point, which signifies the values of mean and alternating stresses in the critical crankpin throw area, was within the safe zone. The results suggest that the failure was induced by overload, with no signs of fatigue. It was recommended to reevaluate the design and production, as well as optimize the crankpin throw rolling process. This recommendation was adopted by the manufacturer, and since then, no further cases of failure have been reported. The article concludes that the analysis of the crankshaft failure helped eliminate fatigue as a cause and highlighted the need for redesigning and reconstructing the project and production processes.

The research [15] focused on analyzing various aspects related to the failure of a diesel engine crankshaft. Initially, a visual inspection of the damage was conducted, revealing signs of material fatigue at the fracture site. Subsequently, material tests confirmed that the mechanical properties of the crankshaft material were within the norm. However, microcracks were found in the area of crack initiation, suggesting that the damage could have been caused by material fatigue. Numerical analysis indicated that the maximum stresses in the crack initiation area were relatively low compared to the material's ultimate strength, suggesting that the fracture could have been due to operational conditions. Moreover, modal analysis showed that the second mode of vibration might be responsible for the damage to the crankshaft in the crack initiation area. Based on these findings, several corrective and improvement actions were proposed, such as increasing the fillet radius on the connecting rods, further material testing, and implementing a rolling process for the connecting rod fillets to prevent similar failures in the future.

The authors [4] proposed a methodology encompassing several key steps in the design of crankshafts. Firstly, it utilizes advanced computer-aided design (CAD) tools, such as CATIA, to precisely develop the crankshaft model. Next, numerical analyses are conducted using ANSYS software, including stress assessment, deformation analysis, and fatigue life prediction. In the subsequent step, researchers perform design optimization aimed at reducing the weight of the crankshaft with minimal impact on its strength and functionality. The entire process is based on advanced computer engineering techniques, allowing for precise analysis and optimization.

In their conclusions, the authors emphasize the significance of integrating ANSYS software in the design and development process of crankshafts. They indicate that this tool enables higher efficiency, reliability, and performance in internal combustion engines. Ultimately, this review suggests that the synergy between ANSYS software and crankshaft development can lead to further innovations in the automotive industry, paving the way for more sustainable and efficient internal combustion engines in the future.

2. OBJECT OF THE ANALYSES AND SCOPE OF THE NUMERICAL CALCULATIONS

The crankshaft of an internal combustion engine plays an important role in converting reciprocating motion of elements of the crank mechanism into rotational motion. The complex state of the stress variable and significant values of the load to which the crankshaft is subjected mean that this component must be designed very carefully. Appropriate stiffness of this component must be assured with possibly small dimensions. These dimensions determine the size of the engine and therefore the possibilities of its installation.

The objective of this study was to determine the stress present in the crankshaft operating at various rotational speeds. Efforts were taken to approach this issue as broadly as possible. Therefore, in addition to the stress, the displacements were determined, both for the load mentioned above and maximum torsional torque; also, modal analysis of the crankshaft was performed.

The crankshaft of a diesel engine to light road traction was analyzed. The engine data is shown in Table 1.

Tab. 1

Technical data of the internal combustion engine

Type of the engine	Compression ignition engine, CR system, turbo supercharger
Layout/number of cylinders	in – line engine / 4
Displacement	1248 ccm
Maximal power output	55.2 kW at 4000 rpm
Maximal torque	190 Nm at 1500 rpm

The calculations and the analyses were performed for six values of engine rotational speed: 1000, 2000, 3000, 3500, 4000, 4500 rpm. They cover the entire operational range of engine speeds and enable assessment of an impact of the load on the stress and the displacements. The calculations were performed for two load cases. In the first one, it was assumed that for each of the considered rotational speeds, the same force acts on the crankshaft, constituting 75% of the maximum force loading the crank. There were considered two temperature values of the engine's structural elements: 323 and 353K. The second case consists of the assumption that the crankshaft is loaded on the cylinder axis by force resulting from the sum of the forces: gaseous and inertia. The gaseous force was determined based on measurement of the pressure in the combustion chamber. The maximum value of this force was taken for each from the rotational speeds. In turn, the value of the inertia force was determined for each rotational speed. Fig. 1 shows the summary value of the force acting in the axis of the cylinder for the second loading case.

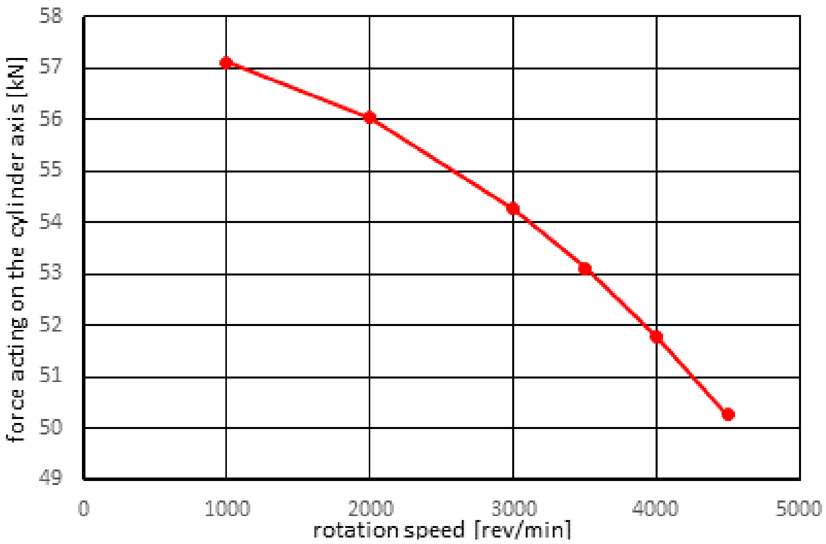

Fig. 1. Change of value of the force acting on
the crankshaft for the second load case of the crankshaft

The crankshaft was also subjected to the modal analysis. In combustion engines, there are occurring periodically variable forces caused by changing pressure in the cylinder and inertia forces. They result in torsional, flexural, and longitudinal vibrations of the crankshaft and elements connected to the crankshaft. In extreme cases, these vibrations can lead to resonance, and the stress generated by the vibrations can cause damage to components of the engine or their premature wear. Normally, analysis of torsional vibrations of the crankshaft is carried out because such vibrations create the greatest risk in case of the resonance. In order to estimate the natural vibrations of the crankshaft, computational methods are used, which are based on the reduction of the real vibrating system to a simpler equivalent system and use of one or more analytical methods. However, the analysis of the torsional vibrations only does not provide a complete picture of the issue, especially in the case of multi-cranked crankshafts, where resonance may occur between the main journals. To obtain a more comprehensive analysis, the Finite Element Method is used.

The reduction of the real system, in the case of analytical methods, consists in dividing the system into smaller parts and, next, determining their inertia and torsional flexibility of the sections connecting adjacent parts. In order to simplify the analysis, the division is often limited to individual crank mechanisms, flywheels and auxiliary drives, main drives, vehicle drives, or power take-off drives.

Simpler analysis methods include methods for reducing the real crankshaft to a single-mass, dual-mass, or three-mass equivalent system. To perform analysis of the natural vibrations of the tested crankshaft after usage of the reduction, the method of successive approximations, known as the Holzer method, is also used. In this article, only the Finite Element Method was used.

Use of the Finite Element Method to the issue of determination of natural frequency of the crankshaft enables determination of all modes of vibrations, not only torsional modes. The frequency analysis was performed using the ANSYS package.

The following physical model supported in the main bearings and being able to rotate freely was analyzed.

Supporting conditions of the crankshaft in the main bearings were modeled, allowing the main journals to move in direction of longitudinal axis.

After carrying out the analyses, the properties of the steel from which the crankshaft was made were determined: yield strength = 350 MPa, and ultimate tensile strength = 420 MPa. Additionally, the following parameters were adopted for calculations: density 7870 kg/m^3, Young's modulus = 210000 MPa, Poisson's ratio = 0.3.

3. CONSTRUCTION OF THE NUMERICAL MODEL

The solid model of the crankshaft was created in the NX program. The model prepared in this way and written in the format appropriate for the NX program (.stp) was exported to the ANSYS WORKBENCH software. The calculations were prepared in the Transient Structural module used for dynamic analysis of the system under the influence of time-varying loads. The mesh was created using the Uniform function, which does not correct the shape of the mesh because of dimensions and curvature of geometry. The mesh consists of elements with 1 mm size; number of the nodes for a complete crankshaft amounts to 6845909, and the number of the elements amounts to 1608812. Selection of an appropriate size of the FEM element involves a compromise between accuracy of the solution, available computational resources, and numerical stability. Typically, smaller FEM elements lead to more accurate results because they better model complicated shapes and gradients of areas. The size of the elements was taken based on the previous experience of the authors [14], striving to obtain accurate results of calculations because preliminary earlier calculations, based on elements of various sizes, have indicated the need for local thickening of the mesh up to 1 mm. In most cases the element size is a compromise between the accuracy and time of the calculations. The mesh generated on the model of the crankshaft is shown in Fig. 2.

Fig. 2. Meshed model of the crankshaft

The boundary conditions, i.e., the force loading the crankshaft and its support, were assumed in the following way: the crankshaft was fixed using the cylindrical constraints in the crankshaft main bearing journals, leaving the possibility of rotation of the crankshaft, while the loading force was imposed on the surface of the crankpin in the angular range from 0° to 120°. The boundary conditions as applied in the analyzed model are presented in Fig. 3.

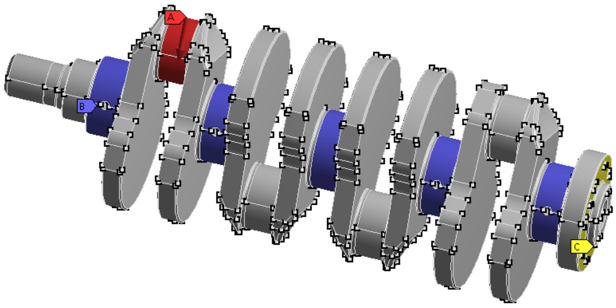

Fig. 3. Boundary conditions applied to the crankshaft

4. ANALYSIS OF RESULTS OF THE NUMERICAL CALCULATIONS

In the first step, numerical analyses were performed applying load on the crankshaft with equal force, constituting approximately 75% of the maximal load on the crank. As expected, the highest values of the stress and the deformations occur in the crankpin. Examples of calculation results are shown in Fig. 4 (stress) and in Fig. 5 (deformations). The deformations of the crankshaft fragments remain in the linear range, and for better illustration, they are presented on scale of 1500:1.

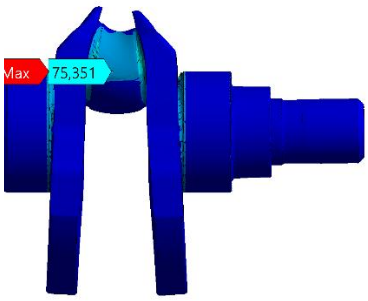

Fig. 4. Stress in the loaded crank for 1000 rpm

In the Fig. 6 are shown calculated values of the displacements and the stress for temperature of 323K. The calculated values of the displacements of the crankshaft operating at this temperature vary from 11.4 µm at 1000 rpm, to 10.47 µm at 4500 rpm, and therefore, by just over 8%.

The stress values for the temperature of 323K change within a bigger range, i.e., from 83.96 MPa to 67.236 MPa, and hence, by almost 20%.

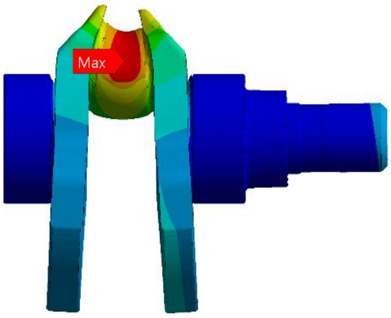

Fig. 5. Total deformations for 1000 rpm (in scale 1500:1)

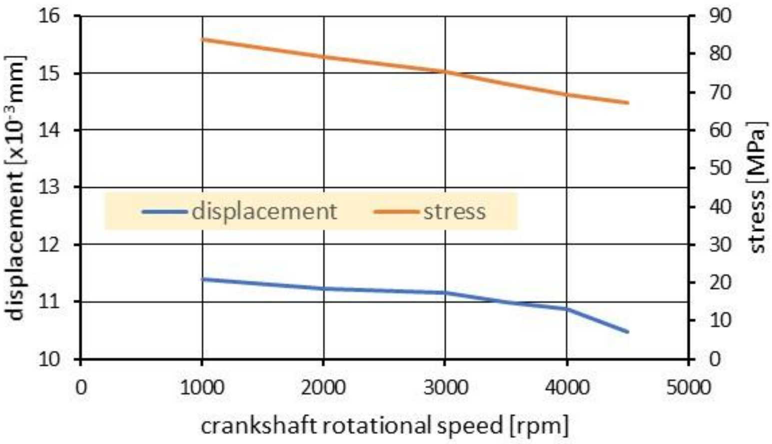

Fig. 6. Displacements (blue color line) and stress (brown color line) of the crankpin calculated for the temperature 323K

A similar character of the courses can be observed for temperature of 353K (Fig. 7). However, in this case the displacement values are higher - they change from 33.9 µm to 31 µm, i.e., by 8.5%. The stress calculated for the temperature of 353K shows a slightly higher values and changes from 84.1 MPa to 68.1 MPa, i.e. by 19%.

In the successive step, calculations were made for loading of the crankshaft with a force acting in the axis of the cylinder. The results of the calculations for the loaded crankpin are presented in Fig. 8÷10.

And thus, together with increasing rotational speed, for temperature of 293K, the displacement values decreased from 15.3 µm to 13.2 µm, i.e., by 13.6%. However, the stress decreased only by 4.5% (from 75.4 MPa to 71.9 MPa, Fig. 8).

Together with increasing rotational speed, for the temperature of 323K, the displacements value decreased slightly, from 13.4 µm to 12.9 µm, i.e., by 3.7%. There was a significant change in the stress values - from 86.4 MPa to 69.9 MPa, i.e., by 19% (Fig. 11).

In turn, for the temperature of 353K, increasing the rotational speed results in decreasing the displacements value from 36.9 µm to 33.9 µm (by 8%), and the stress value to decrease from 87.3 MPa to 70.2 MPa, i.e., by 19, 7% (Fig. 10).

In the course of the calculations, the values of the stress and displacements at the free end of the crankshaft had taken negligible values and were not subjected to the analyses.

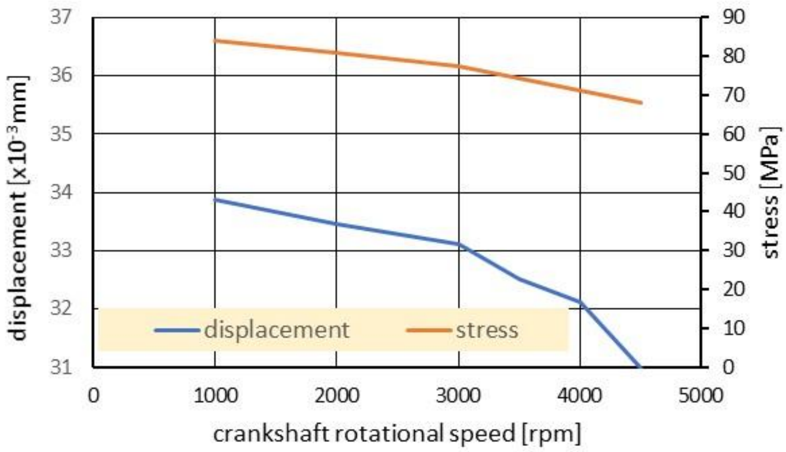

Fig. 7. Displacements (blue color line) and stress (brown color line) of the crankpin calculated for the temperature 353K

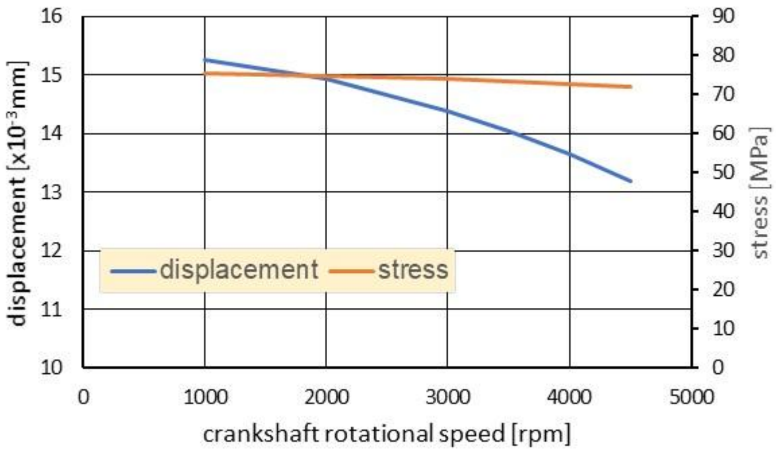

Fig. 8. Displacements (blue color line) and stress (brown color line) of the crankpin calculated for the temperature 293K

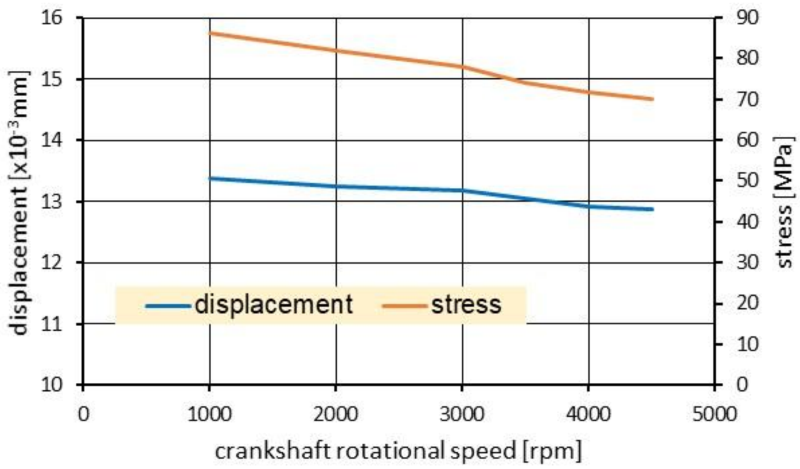

Fig. 9. Displacements (blue color line) and stress (brown color line) of the crankpin calculated for the temperature 323K

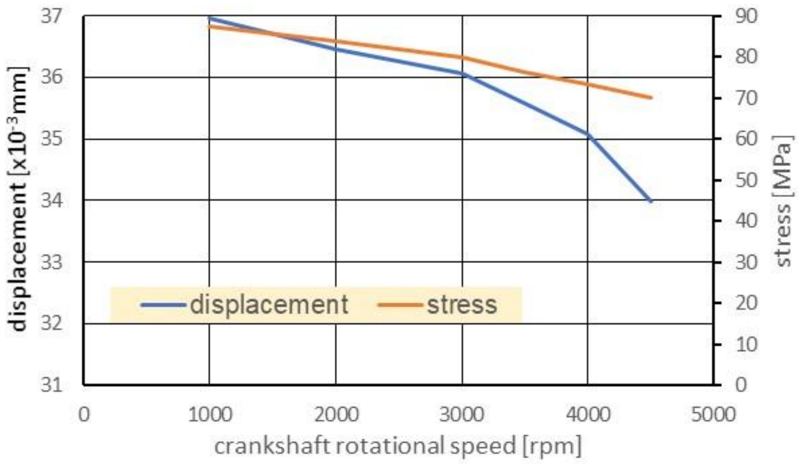

Fig. 10. Displacements (blue color line) and stress (brown color line) of the crankpin calculated for the temperature 353K

The results of the modal analysis are presented in Table 2. The modes of vibrations were classified as a bending, torsional and longitudinal ones, based on the dominant direction of the displacement along the axis (X), or in transverse directions (Y, Z), or also in the direction of the angle of rotation of the crankshaft. In the case of the crankshaft, it is possible to talk only about the dominance of a specific form of vibrations over the others, due to the complex, spatial nature of the vibrations, especially in the case of the modes with a higher frequency.

Tab. 2

Natural frequencies and modes of vibration

Mode	Frequency [Hz]	Modes of the vibrations
1	3918.4	Bending
2	3918.6	Bending
3	3922.8	Bending
4	3922.9	Bending
5	3933.3	Bending
6	3934.3	Bending
7	3946.8	Bending
8	3947	Bending
9	3950.6	Bending
10	4261.6	Torsional
11	4264.7	Torsional
12	4266.2	Torsional
13	4277.8	Torsional
14	4278.3	Torsional
15	4292.1	Torsional
16	4292.4	Torsional
17	4297.5	Torsional
18	6006.5	Longitudinal
19	6059.3	Longitudinal
20	6087.9	Longitudinal

Mode	Frequency [Hz]	Modes of the vibrations
21	6115.9	Longitudinal
22	6442	Bending
23	6442.3	Bending
24	6443.4	Bending
25	6444.9	Torsional
26	6445.2	Torsional
27	6447.9	Torsional
28	6448.2	Torsional
29	6449.8	Torsional
30	6728.4	Longitudinal

As can be noticed, all natural frequencies of the existing types of vibrations lie outside the range of useful rotational speeds of the analyzed crankshaft.

5. CONCLUSIONS

Performed analyses constitute an attempt at a complex method of proceeding, which enables determination of the stress and displacements of the crankshaft of a modern, high-speed diesel engine. The analyses have allowed determination of a role of calculated values in assessing critical stress-strain of the material value and to identify areas that should be designed with a particular care. The analyses have confirmed that selection of the material is very important because it determines usage of components having dimensions specified during the designing stage. After carrying out calculations, it should be stated that analyzed crankshaft of the engine was designed so that excessive stress values will not occur. Stiffness, in spite of small overlapping of the crankshaft main bearing journal and the crankpin, was ensured by using the appropriate thickness of walls. However, special attention should be paid to transitions from the crankpin to the crank arm. During the calculations, increased stress values were found in these areas. In the opinion of the authors, after performing analysis of many cases, increased stress values that appear in these areas result from peculiarities of the FEM method. Hence, the need for completing calculations performed so far with analyses that would take into account their special character.

The analysis of the shaft calculation results shows that both displacements and stresses decrease with increasing rotational speed. These changes are caused by the unloading effect of the centrifugal force associated with the crankshaft, which reduces the total value of the loading force.

The presented stress values are reduced stresses and do not exceed the permissible values specified for the crankshaft material.

Because displacement values decrease with increasing crankshaft speed, the smallest journal-to-bearing clearance values can be expected at low rotational speeds and high crankshaft material temperatures. Please be advised that bearing housings may also experience deformation as a result of elevated temperatures. The authors performed additional analyzes of the crank bearing clearances and found that there was no risk of bearing sticking in the considered range of rotational speeds and temperatures.

From analysis of results of the calculations of the crankshaft loaded with constant force is seen that temperature has a small effect on values of the stress, but has important effect on values of the displacements, which practically increase almost three times when temperature changes from 323K to 353K. At a given operating temperature of the crankshaft, a similar range of the changes in both the displacements and the stress takes place (Fig. 11).

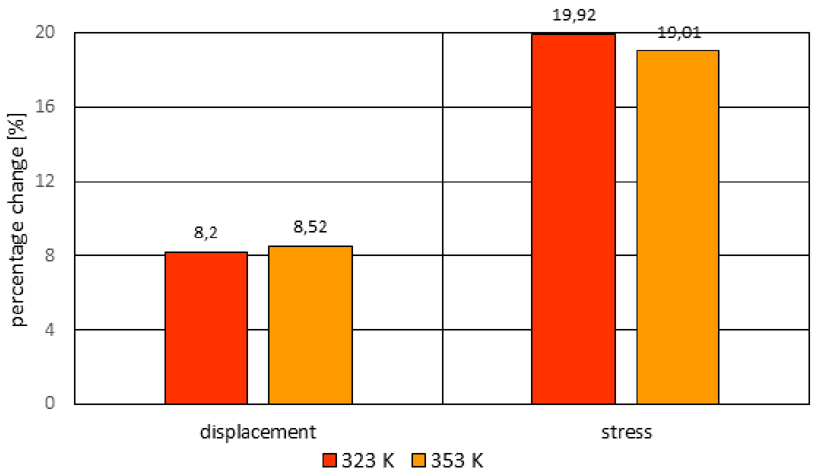

Fig. 11. Percentage change of the displacements and stress calculated for various values of temperature of the crankpin loaded with constant force acting on axis of cylinder

Analyzing the results of the crankshaft calculations loaded with force acting on axis of the cylinder, it can be noticed that the stress increases with increasing temperature, while the most significant is increase in the temperature in the range from 293 to 323K. Changes in the displacements are ambiguous: after a slight decrease in temperature range of 293K÷323K, there occurs a significant increase in the range of 323K÷353K. At a given operating temperature of the crankshaft, the changes both in the displacements and the stress are different (Fig. 12), resulting from change in the crankshaft load by a force acting on the cylinder axis (Fig. 1).

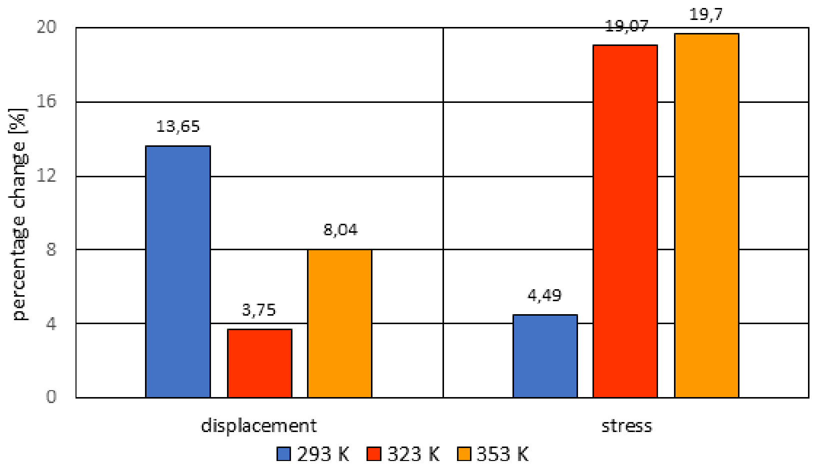

Fig. 12. Percentage change of the displacements and stress calculated for various values of temperature of crankpin loaded with a force acting on the cylinder axis

Using of the Finite Element Method enables a comprehensive approach to the issue of natural vibrations of the crankshaft because it allows for the determination of all possible modes of vibrations, not only these of a torsional nature, as this takes place in the case of analytical methods. Unfortunately, such an approach involves significant labor required to prepare a computational model properly. However, owing to it, it is possible to obtain a more complete picture of the crankshaft's behavior during vibrations, what is particularly important for more complex modes of vibrations with higher frequencies. The crankshaft, due to its high stiffness and method of its supporting, exhibits a dangerous phenomenon of resonance at the level of 3918 Hz, which corresponds to rotational speed of 235080 rpm, impossible to be achieved in operating conditions of the internal combustion engine.

Numerical analyses using the finite element method have proven useful for designing and analyzing structural components, such as crankshafts. These calculations provide valuable information about the behavior of crankshafts under various operating conditions and temperatures. This information helps to optimize the shape and dimensions of crankshafts, improving efficiency and reducing manufacturing costs.

These numerical calculations have proven useful in designing and analyzing structural elements, such as crankshafts. Despite the accuracy of the FEM method, each engine is tested intensively in the prototyping phase for mechanical and thermal durability. However, one should always consider changes in the material resulting from the machining process in relation to the idealized computer model. This issue particularly concerns elements exposed to friction and thermal loads. These processes enable the final evaluation of the product.

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