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RESIDUAL STRESSES ASSESSMENT IN THE MARINE DIESEL ENGINE CRANKSHAFT 12V38 TYPE

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Abstract

In the paper, the measurement of the marine diesel engine crankshaft residual stresses is presented. The hole drilling strain gauge was chosen because of its simplicity and low cost. Moreover, deflection calculation was made for the load corresponding to its weight and the concentrated force in the chosen bearing conditions. Deflection of the journal and bending rigidity under the concentrated force was calculated as well. In that case, the crankshaft was supported in the neighbouring crank journals. The reduced stresses were calculated according to the Huber – Mises hypothesis. The residual stresses were measured in the marine diesel engine crankshaft type 12V38 after the finishing, with the hole-drilling method. It was found that the residual stresses reach the maximal value of 86 MPa, but for 50% of measuring points, the residual stresses did not exceed the value of 20 MPa. The squeezing stresses were found to be dominant. The measurement results provided important data for further consideration, especially for the correction of the bending deformation of the produced crankshaft.

Keywords: *diesel engine, crankshaft, simulation, residual stress*

1. Introduction

Correct performance of the engine, its durability and often the safety of humans depend on the crankshaft quality ensured on each stage of its production, starting with the material analysis [1] up to its fatigue [2, 3]. This applies especially to the monolithic crankshafts [7] of the high power engines like the marine ones of length ca. 12 m and of weight ca. 25 tons. In that case, even before the machining, internal stresses take place, and if there are no outer forces applied, these stresses remain. It was reported that the residual stresses might lead to the failure of the crankshaft even when there were no material defects and corrosion [6].

There are three categories of the residual stresses: the 1st order stresses that appear in the entire volume of the crankshaft, the 2nd order stresses that take place in the microstructures like grains, and the 3rd order stresses that appear in the crystal lattice. The main role plays the 1st order stresses, which is the subject of the changes during the machining. When the machining allowance is removed, the distribution of the residual stresses is changed because the initial balance is broken, and a new inner stress balance is being formed. The values and the characteristics of the residual stresses distribution depend on the crankshaft form, its dimensions, dimensional proportions, as well as the technology. The forging technology generates large residual stresses in the workpieces, which is a natural consequence of the material squeezing and non-uniform cooling. In some cases, the residual stresses may lead to the deformation of the workpiece or even to the appearance of cracks. The present study is aimed to the analysis of the residual stresses after machining and their impact on the crankshaft shape.

2. Crankshaft description

The investigations were performed with the monolithic marine diesel engine crankshafts type 12V38. Its 3D model is presented in the Fig. 1, and the main features are described in the Tab. 1.

Fig. 1. The 3D model of the monolithic marine diesel engine crankshafts type 12V38

Tab. 1. Main geometrical features of the monolithic marine diesel engine crankshafts type 12V38

Symbol of the crankshaft	12V38
Crankshaft length L	5155 mm
Diameter of the main journal d_g	380 mm
Length of the main journal l_g	180 mm
Diameter of the crank journal d_k	360 mm
Length of the crank journal l_k	302 mm
Crank radius R	237.5 mm
Journals coverage factor $\varepsilon = 0.5 \cdot (d_g + d_k) / R$	1.558
Radius between the main journal and the crank arm R_g	30 (43) mm
Radius between the crank journal and the crank arm R_k	21 (29.5) mm
Material	32CrMo12

Tab. 2. Mechanical characteristics of the crankshaft material

Material	32CrMo12
Tensile strength R_m [MPa]	≥850
Proof strength $R_{p0.2}$ [MPa]	≥630 (R_e)
Percentage elongation A_5 [%]	≥13 (A)
KV [J]	≥20

For the crankshaft, deflection calculation was made for the load corresponding to its weight and the concentrated force in the chosen bearing conditions. Maximal vertical deflection of the crankshaft caused by its weight was $f_G = 0.776$ mm and bending rigidity $k_G = 77.36$ kN/mm. When the load was 10 kN only, the maximal vertical deflection was $f_{10kN} = 0.285$ mm, and bending rigidity $k_{10} = 35.05$ kN/mm. Deflection of the journal under the concentrated force 10 kN was $f_{10kN} = 0.0062$ mm, and bending rigidity $k_{10} = 1613$ kN/mm. In that case, the crankshaft was supported in the neighbouring crank journals. The Fig. 2 presents the calculation model, and the graphical results for the vertical deflections and reduced stresses at different rotational positions.

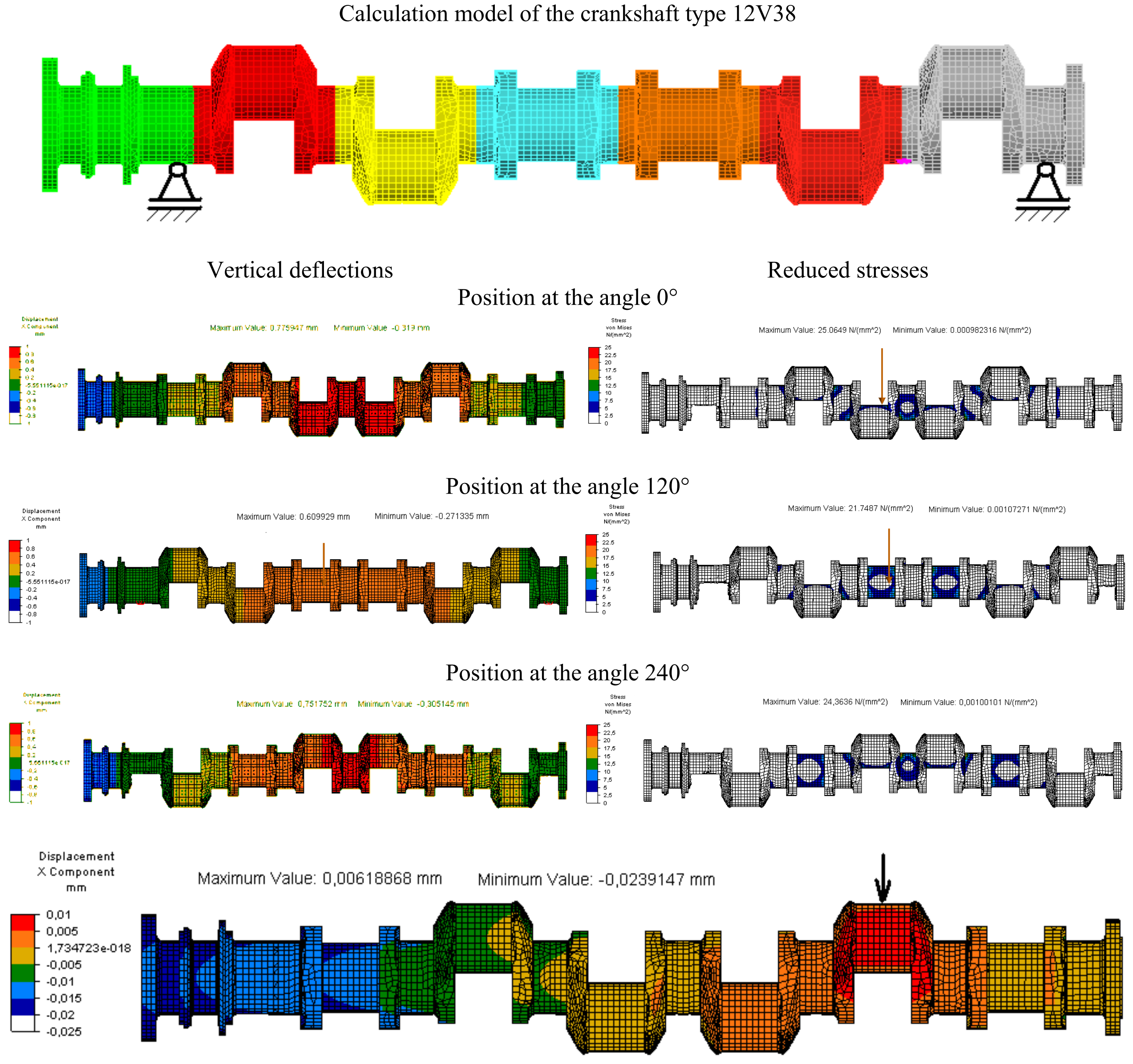

Fig. 2. The deflections in x direction and the reduced stresses according to the hypothesis of Huber – Mises under own weight or under the concentrated load, with chosen bearing points

The most undesirable bearing of the crankshaft is the edge support in its outer main journals. In that case, the maximal deflection was $f_G = 0.99$ mm, bending rigidity was $k_G = 60.64$ kN/mm, and the reduced stresses calculated according to the Huber – Mises hypothesis [4] was $\sigma_{zr} = 25.06$ MPa.

3. Measurement method

In order to determine the residual stresses in the material, various release methods are applied, where some part of the material is removed and the deformation is measured, e.g. with the tensometers. Due to its simplicity and low cost, the hole drilling strain gauge method is one of the most popular techniques to determine residual stresses [5]. After some material is removed, apart from the release of the residual stresses, the concentrator of the stresses appear, as well as the deformations caused by the drilling.

In the elasticity theory, the state of stress is described by the formulas. The equations represent the impact of the round hole on the stress distribution in the plate, which is subject of stretching in the axis, x with pressure σ, as follows:

$$\sigma_r = \frac{\sigma}{2}\left(1-\frac{a^2}{r^2}\right)+\frac{\sigma}{2}\left(1-\frac{4a^2}{r^2}+\frac{3a^4}{r^4}\right)\cos 2\theta,$$
$$\sigma_\theta = \frac{\sigma}{2}\left(1+\frac{a^2}{r^2}\right)-\frac{\sigma}{2}\left(1+\frac{3a^4}{r^4}\right)\cos 2\theta, \quad (1)$$
$$\tau_{r\theta} = -\frac{\sigma}{2}\left(1+\frac{2a^2}{r^2}-\frac{3a^4}{r^4}\right)\sin 2\theta,$$

where (Fig. 3):
θ – angle,
σ_θ, $\tau_{r\theta}$ – components of the stress,
a – radius of the hole,
r – distance from the hole centre.

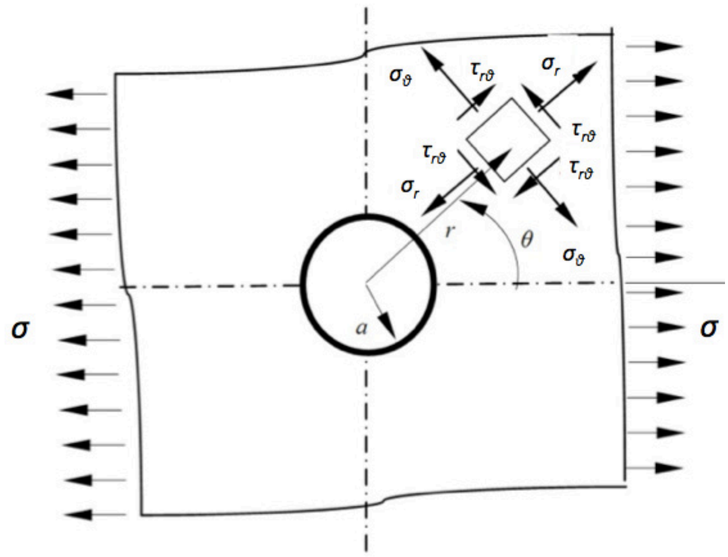

Fig. 3. The example of the influence of the number of thermal shocks on the deformation of the piston sample

If the radial deformations are measured with a tensometer before and after drilling, then the deformation will be equal to the difference between the plate with a hole ε_r and the one without it ε_r^0. Thus, it can be written:

$$\varepsilon_r^{tens} = \varepsilon_r - \varepsilon_r^0 = -\frac{\sigma_1+\sigma_2}{2E}\cdot(1+\nu)\cdot\frac{a^2}{r^2}+\frac{\sigma_1-\sigma_2}{2E}\cdot\left[(1+\nu)\cdot\frac{3a^4}{r^4}-\frac{4a^2}{r^2}\right]\cos 2\theta, \quad (2)$$

where (Fig. 3):
ε_r^{tens} – radial deformations indicated by the tensometer,
ε_r – radial deformations,
ε_r^0 – radial deformations of the plate without a hole,
σ_1, σ_2 – axial and radial stresses, respectively,
ν – Poisson's ratio,
E – modulus of elasticity.

When the stresses are axial only ($\sigma_2 = 0$) and the tensometer is coaxial with the axis 1-1, it could be written as follows:

$$\varepsilon_r^{tens} = \varepsilon_r - \varepsilon_r^0 = \frac{\sigma_1}{2E}\cdot(1+\nu)\cdot\left[\frac{3a^4}{r^4}-\frac{a^2}{r^2}\cdot\left(1+\frac{4}{1+\nu}\right)\right] \quad (3)$$

or:

$$\sigma_1 = \frac{2E \cdot \varepsilon_r^{tens}}{(1+\nu)\cdot\left[\frac{3a^4}{r^4} - \frac{a^2}{r^2}\cdot\left(1+\frac{4}{1+\nu}\right)\right]}. \tag{4}$$

4. Results and discussion

The residual stresses were measured in the marine diesel engine crankshaft type 12V38 with the hole-drilling method, in the factory of Celsa in Ostrowiec Świętokrzyski (Poland). The boundary conditions are presented in the Fig. 4 and 5. Tab. 3 contains main characteristics of the material and measurement data, and Fig. 6 presents the example of the tensometer distribution on the crankshaft.

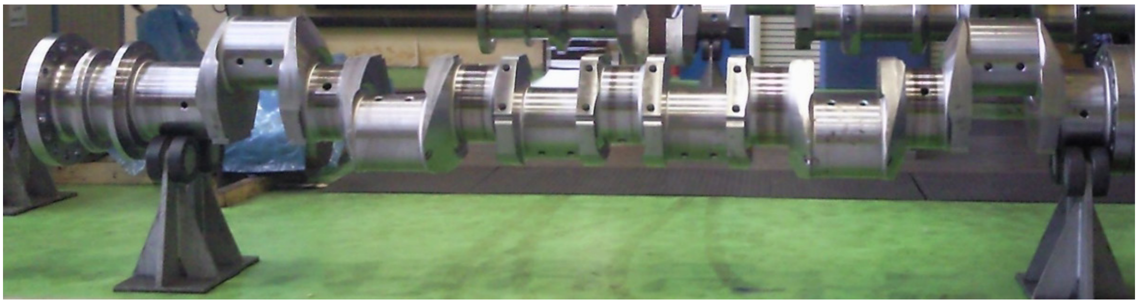

Fig. 4. The crankshaft during residual stress measurement

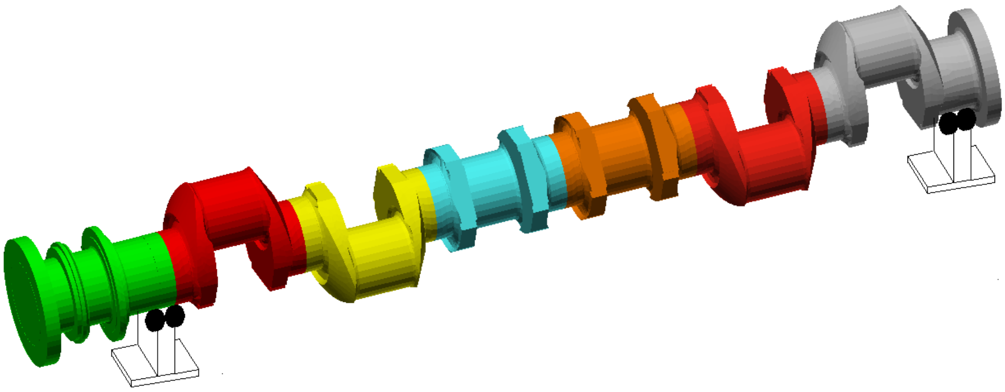

Fig. 5. The boundary conditions of the residual stress measurement

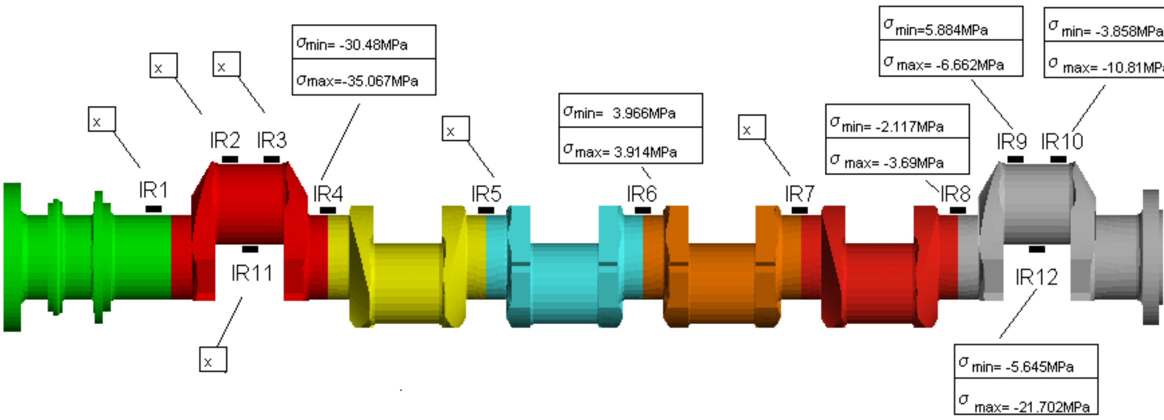

Fig. 6. Distribution of the tensometers on the crankshaft in the plane 1, with the respective main stresses values

Tab. 3. Characteristics of the material and measurement data

Material constants	Tensometric constants
Poisson's ratio $v = 0.3$ Young's modulus $E = 215000$ MPa	Strain gauge probe diameter $D = 5.26316$ mm Depth of drilling $Z = 2$ mm Ratio $Z/D = 0.378$

It was found that the residual stresses in the crankshaft type 12V38 after the finishing reach the maximal value of 86 MPa. For a half out of 20 measuring points, the residual stresses did not exceed the value of 20 MPa. The squeezing stresses were found to be dominant.

The performed experiments confirmed also that the depth of the drilled hole has very small impact on the values of stresses measured on the plate surface, compared to the calculated ones. In the range of the hole depth from 1 up to 7 mm, the calculated residual stresses on the surface at the distance $r = 2.5$ mm and hole radius $a = 1$ mm vary between 0.866 MPa and 1.016 MPa. This means, that dependent on the drilled hole depth, the error vary from 1.6% up to 13.4%. The smallest error below 2.7% is obtained for the depths between 1.5 and 3 mm (i.e. $1.5a$-$3a$).

The stresses indicated by the tensometers as differences between the initial and final deformations vary between -0.27 MPa and -0.35 MPa. This means that the real residual stresses in the plate, of values +1 MPa, after release are registered as an average –0.31 MPa, which is of opposite sign, and 1/3 of the true value. Therefore, it is necessary to recalculate the obtained indication according to the equation (4).

5. Conclusions

Since the crankshaft is one of the most responsible parts of the engine, its reliability should be examined thoroughly. The residual stresses may cause the failure, so it is crucial to assess them in an adequate manner. The study involved both digital model and the hole drilling strain gauge method. It was proved that for the most unfavourable bearing conditions, the vertical deflection of the 12V38 type crankshaft was maximally $f_G = 0.99$ mm. Its bending rigidity was $k_G = 60.64$ kN/mm, and the reduced stresses calculated according to the Huber – Mises hypothesis was $\sigma_{zr} = 25.06$ MPa.

Next, it was found experimentally that the residual stresses in the crankshaft 12V38 after machining are not exceeding the value of 86 MPa, while 50% of the points had residual stresses 20 MPa and lower. The dominant were the squeezing stresses.

It was found also, that the reduced residual stresses after thermal treatment are below 14 MPa, but they are responsible for the bending deformations of the crankshaft. Thus, based on the residual stress measurement, it is possible to calculate the bending of the crankshaft axis, and to propose the additional treatment to compensate the deformation.

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