Experimental evaluation of the shrink-fitted joints in the assembled crankshafts

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Abstract. The paper presents the results of investigations on the quality and loadability of the large size crankshafts of the assembled type. Especially in the marine engines, the shrink-fitted joints have to bear large shock loads in order to ensure the safety of the people aboard. Methodologically, however, it is difficult to perform exact load tests on the ready crankshafts, because of their large sizes and eventual destruction after test. Thus, the models of the joints were made to check the performance of the shrink-fittings of different tightness. First, the inner stresses were measured with ultrasonic devices. Second, the laboratory equipment was developed to measure the critical torque that can break the joint, and the models underwent the tests. The critical force $F_{cr}$ was measured for each shrink-fitted coupling of the pivot with crank, and the critical value of static torque $M_{cr}$ was calculated. The test results showed that the relative tightness of 2.02 per mil provided the maximal reliability of the assembled crankshaft with the critical torque $M_{cr_{\text{max}}} = 35 \text{kNm}$. Increase of tightness up to 2.1 per mil led to the drastic fall of the critical torque down to ca. 26 kNm. The main conclusion was that the critical torque is not higher for relative tightness $W_w$ larger than 2.2 ‰. Thus, in practice, there is no point to make more tight joints, because their reliability does not improve.

Keywords: shrink-fit, internal stresses, crankshaft, critical torque.

1. Introduction

There are two main technologies for the crankshaft fabrication, the monolithic type, and the assembled or built-up type made out of the separated elements [1]. In case of the large-size crankshafts manufactured for the marine engines, the monolithic technology poses many problems, so the assembled crankshafts are preferable. In Poland, only the company CELSA Huta “Ostrowiec S.A.” is producing this type of the crankshafts, and the researches are performed in cooperation with the Kazimierz Pulaski University in Radom in order to improve their reliability. The investigations presented below were performed in frames of this cooperation.

Typical metrological supervision of the process [2] is applied in the production system of the crankshaft. However, the additional reliability study must be made because the failure of such a responsible marine engine detail like crankshaft is very dangerous. Especially during the operation in the storm, it may lead to the life threat of the shipboard personnel and passengers, as well as to the destruction of the ship. That is why the proper assembly of the crankshaft is of a key importance to avoid catastrophic failure of the engine.

In mechanical components, the residual stresses and their distribution may cause the failure at a load level significantly lower than expected [3], which is true for the shrink-fitted couplings. Moreover, in the assembled crankshafts, the hub must be axially positioned on the shaft and must resist the moments and forces generated by misalignments [4]. Hence, the statistical approach is rather ineffective in that case, despite it is a crucial tool in many industrial systems [5].
There are many publications proposing various methods of the shrink fit couplings analysis in different applications [6-8]. Fonte et al. reported the failure case where the crack had initiated approximately 7.5 mm inside the shrink fit crevice [9]. Rade et al. analyzed mathematically the consequences on the press-fit contact pressure of the application of a bending moment to the shaft [10]. Bertini and Santus performed fretting fatigue tests on shrink-fit specimens [11]. McMillan et al. described theoretical analysis and experimental researches on the increasing torque with recurrent slip in shrink fittings [12]. Their findings were aimed to optimize interference-fitted assemblies, particularly in applications where the shaft is to be reused. In the present researches, the empirical measurements of the critical torque applied to the shrink fitted joint are under consideration in order to evaluate the increase of interference tightness in the context of its reliability.

2. Characterization of the examined crankshaft models

The large sized assembled crankshafts are the ones designed for the high power internal combustion engines, especially marine engines, where one cylinder power exceeds 1000 kW. In the Fig. 1, the examples of such a crankshafts are presented. The first one (Fig. 1, left) produced by MAN company weights 225 tons and measures 22 m. The second one (Fig. 2, right) made by B&W Diesel company is of 313 tons and 22.5 m long. The maximal radius of each crankshaft is ca. 1500 mm.

The mass and dimensions of that range, together with the complicated 3D shape, makes it almost impossible to fabricate the monolithic forging of such a crankshaft. That is why they are manufactured as forged pivots and cranks separately, later joined together using the shrink fit method shown in the Fig. 2.

![Figure. 1. Examples of the large size assembled crankshafts [13]: MAN – 225 tons, 22 m long (left), and B&W – 313 tons, 22.5 m (right).](image)

![Figure. 2. Shrink fitting of the pivots with cranks: view (left) [13], assembling stages (right).](image)
It is obvious, that this type of joints is the weakest point in the ready crankshaft. The research problem of the shrink fitted coupling reliability is posed by the fact that the contemporary engines generate increasing exploitation loads on the crankshafts. To ensure the required torque load withstanding, the engineers are forced to apply larger diameters and increased tightness of fitting. Nowadays, the interferences in the range of 2-2.5 per mil are often applied.

The crankshaft is a subject of both degradation and shock failure processes, which are difficult to be modeled [14]. The question rises, however, on the limitation of the ability to receive and bear the torque. A sudden overload of the crank may break the shrink fittings of the assembled crankshaft, and to block the engine. In case of marine engines, when the failure takes place during the storm, such damage makes it impossible to disassemble the engine and repair it ensuring the safe return to the port. In that case, people are in danger.

Thus, the performed investigations were aimed to find out the maximal interference (tightness) ensuring the maximal torque bearing, with further tightness increase giving no effect on the crankshaft reliability. Despite the numerous research projects performed throughout the World, e.g. [15-17], there is still lack of a convincing model of the stresses distribution in the shrink fitted coupling bound with the effective loadability of the joint. The problem has not been solved even by the international Standard M53 [18], which has been worked out as an agreement between the constructors, manufacturers and classification societies.

Thus, it was assumed that there is a maximal tightness ensuring the critical static torque load, and the further increase of the tightness does not increase substantially the critical torque value. In order to check the assumption, the series of the experimental investigations were performed. Because of the large dimensions of the crankshafts, however, the models in scale 1:5 were fabricated, so the laboratory measurement became possible. Fig. 3 presents the examined models.

The shrink fitted couplings were made considering the real technical conditions kept in the Celsa production lines. Each crank was heated from the bottom side around its orifice up to the temperature 350°C, and after the pivot was inserted the coupling was left to cool down in the air. The surfaces in contact were dried before the operation and the fat removed, but no substances increasing friction were applied.

After the measurement of the roughness and form deviations, the couplings were matched according to the fit tolerances shown in the Table 1.
Figure. 3. Models of the crankshaft shrink-fitted couplings: a) cranks, b) rings, c) pivots, d) the elements made out of steel S34MnV.

Table 1. Interference fit of the pivots with rings and cranks assuring the assumed tightness

<table>
<thead>
<tr>
<th>Ring openings</th>
<th>Feature</th>
<th>Diameter $D_{oo}$ [mm]</th>
<th>Pivots</th>
<th>Feature</th>
<th>Diameter $D_{cz}$ [mm]</th>
<th>Tightness $W_1$ [mm]</th>
<th>relative $W_{w1}$ [%]</th>
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3. Internal stresses measurement

The shrink-fitted elements generate internal stresses, which was measured using the ultrasonic measurement device Debro in the laboratory of the Institute of Fundamental Technological Research (Polish Academy of Sciences). The device is shown in the Fig. 4a, while the Fig. 4b presents the positioning of the axes numbered from 1 to 8, where the measurements were performed. The measured value was the ultrasonic wave passing time, polarized along the marked axis and perpendicular to it. In the Fig. 5, the results of measurement at the distance 5 mm from the border between the elements are presented. These examples correspond with two couplings of the crank-pivot type and one of the ring-pivot.
The probing points were assigned along each axis. Close to the contact border between two details they were set more densely (5 mm), while farther along the axes the step was 10 mm. The first measurement was made at the distance 5 mm from the border between the coupled details. For each crank-pivot and ring-pivot couple, the measurement was made before they were joined together and afterwards. The measurement results should be treated as the stresses averaged along the material thickness. In case they were performed close to the border between the details, they meant average along the joint. Similar measurements along the axes marked in the similar way, were performed for the ring-pivot couplings.

The presented measurement results reveal substantial irregularity of the internal stresses distribution for both crank-pivot couplings and ring-pivot ones. It is noteworthy, that the stresses close to the borders between the coupled elements are always of positive values. This result may be attributed to the fact that only the averaged stretching stresses were measured along the material thickness, both in radial and circumferential directions.

Figure. 4. Measurement of the internal stresses with the ultrasonic method: a) measuring device, b) measurement axes for the crank and pivot.
Figure 5. The distribution of the radial and circumferential stresses at the distance 5 mm from the border between the elements: a,b) crank-pivot with tightness 2.02‰, c,d) crank-pivot with tightness 2.25‰, e,f) ring-pivot with tightness 2.25‰.

4. Measurement of the critical torque

The critical torque $M_{cr}$ is the torque, which breaks the shrink fitting of the pivot and crank. Fig. 6 presents the device ZD-40 constructed in order to determine the critical torque for the examined joints. The lever of known length was attached to the pivot while the crank remained fixed. The force applied to the end of the lever was measured until the coupling joint was broken.
The device ZD-40 consisted of the support (1) with the holder (2) where the examined crank (3) is fixed. The force applied to the lever (4) through the roller (5) was increased up to the moment when the pivot starts to move in the fixed crank. In order to measure the rotational angle of the pivot in the crank, the dial indicator (6) was fixed to the rod attached to the pivot.

The critical force $F_{cr}$ was measured for each shrink-fitted coupling of the pivot with crank. Then, knowing the lever length $l = 0.2$ m, the critical value of static torque $M_{cr}$ was calculated. The results are presented in the Fig. 7 as a graph of the critical torque $M_{cr}$ versus the relative tightness $W_w$.

The main graph corresponds with the results obtained for the crank-pivot couplings, and the additional one corresponds with the ring-pivot couplings. For the latter, the number of experiments was smaller, just to compare the trend. In fact, the crank-pivot couplings reveal different characteristics, even though in principle the trend is similar. The photo of the surface destruction of the disassembled elements is presented in the Fig. 8.
Figure. 8. The destruction of the surfaces after the coupled crank and pivot are disassembled by the critical torque $M_{cr}$.

In the case of the examined couplings, the relative tightness of 2.02 per mil provided the maximal reliability of the assembled crankshaft with the critical torque $M_{cr,\text{max}} = 35 \text{ kNm}$. Increase of tightness up to 2.1 per mil led to the drastic fall of the critical torque down to ca. 26 kNm, which was just 75% of the maximal value $M_{cr,\text{max}}$. Further tightness increase up to 2.2 per mil provided some improvement of the reliability with $M_{cr} = 30 \text{ kNm}$, which was still ca. 83% of the maximal value of $M_{cr,\text{max}}$, with no further improvement.

The measurement results proved that the assumption was right, and the further increase of tightness does not improve the reliability of the shrink-fitted coupling. The phenomenon may be explained by the irregularities of the internal stresses that lead to the non-uniformity of the adjacent surfaces contact. Fig. 9 presents the example of the crank deformations simulation, which shows that increase of the tightening is in fact undesirable for the shrink fit. The destruction of the surfaces of the disassembled elements seen in the Fig. 8 seems to confirm this explanation.

Figure. 9. The modeled deformations of the shrink-fitted crank.

5. Conclusions

The presented analysis led to the important findings. First of all, the measured distribution of the radial and circumferential stresses confirmed non-uniformity of the residual stresses in the shrink-fitted couplings, especially in case of crank-pivot joint. As a result, the fitting may appear weaker than it was assumed in theoretical calculations.
Secondly, the results of the critical torque measurement revealed no improvement of the shrink-fitted joint loadability for the relative tightness above 2.02 per mil. It has to be considered at the project stage, that the excessive increase of the tightness of shrink-fitted crank and pivot in the assembled crankshafts does not improve its strength.

Even though the experiments were performed on the smaller models and exact values may differ for the large size assembled crankshaft, the general trend must be the same. It was confirmed by the ring-pivot couplings characteristics, that the tightness ca. 2.0 per mil provided the best reliability of the shrink-fitted crank-pivot coupling.

To explain the phenomenon in detail, some further measurements should be done. The temperature distribution during the cooling of the shrink-fitted coupling may shed some light on the shrinking process, especially in terms of internal tensions and the adjacent surfaces contact.

References


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