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Numerical investigation of the influence of geometric parameters on crankshaft fatigue failure using finite element analysis

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ABSTRACT

The paper examines how geometric design parameters affect crankshaft fracture due to fatigue using the finite element method. It explores the influence of the fillet radius on the crankpin journal, the shape of the grooves on the splined shaft, and their interaction on the crankshaft's fatigue behavior. Using established material traits and operating load magnitudes, a finite element analysis (FEA) computational model of the crankshaft was created, and stress simulation was performed. Following the obtained results, critical points on the crankshaft were identified, followed by fatigue analysis and lifespan evaluation. Numerical simulations showed that increasing the fillet radius on the crankpin journal significantly reduces stress concentration and boosts the crankshaft's fatigue life. Altering the geometry of the grooves on the splined shaft redistributes stress and moderately extends its lifespan. The best results came from combining both geometric changes, demonstrating the synergistic effect of optimizing design parameters. The findings provide numerical insight into the relationship between geometric features and fatigue behaviour of crankshafts and may support further research on geometry-based design optimization.

Keywords: crankshaft, fatigue failure, finite element analysis.

INTRODUCTION

The crankshaft is a core component of the engine and is continuously subjected to cyclic loading during operation. Such loading conditions make crankshafts particularly susceptible to fatigue damage, which represents one of the most common causes of their failure. Fatigue occurs as a result of continuous, recurrent loading and promotes the breaking mechanism of the crankshaft. Fracture generally occurs in critical areas of the crankshaft, acting as a localized stress concentrator. Those critical areas are zones of cross-sectional discontinuities (key slots, oil ports, etc.), as well as unevenly formed areas such as sharp edges, abrasion, and internal fractures caused by production imperfections [1]. Due to its importance, a great deal of research, primarily in the field of fatigue fracture design strategies, is dedicated to the topic of crankshafts.

A number of studies have focused on the analysis of stress concentration and structural behaviour of crankshafts using experimental and numerical approaches. Guagliano et al. [2] explored the stress concentration factor of the crankshaft of diesel engines based on experimental test programs and FEA using a three-dimensional (3D) model and an associated simplified two-dimensional (2D) model. Chien et al. [3] investigated the alterations in residual stresses induced by the fillet rolling surface strengthening on the fatigue mechanism of a ductile cast iron crankshaft section under bending, using two-dimensional elastic and elastic-plastic FEA in conjunction with a linear elastic fracture mechanics model. Jiao et al. [4] investigated the KL crankshaft of a heavy-duty four-cylinder engine and performed the free-state Eigenvalue analysis of the crankshaft using ANSYS software to mitigate resonance and vibration effects at specific operating frequencies. These studies demonstrate the importance of numerical modelling in understanding stress distribution and dynamic behaviour of crankshaft structures.

Another group of investigations focuses on the influence of manufacturing processes and material-related factors on crankshaft durability. Silva [5] investigates the cause of damage to the journals of two crankshafts of diesel engines after grinding, where it was determined that thermal fatigue cracks caused by an improper grinding process led to their accelerated failure. Aliakbari [6–7] conducted a set of experimental tests on damaged crankshafts, including chemical compositions, microstructural properties, mechanical-related parameters, surface hardness, and fracture morphology. Such studies provide valuable insight into the influence of material properties and manufacturing defects on crack initiation and propagation.

In addition, several authors have performed comprehensive failure analyses of crankshafts operating under real service conditions. Zhu et al. [8] investigated localized wear mechanism on one side of the oil port, focusing on the specific crankpin of a straight-six-cylinder engine made out of 42CrMo material, using a combination of experimental analysis and numerical simulations to determine the mechanisms of abrasive, adhesive, and fatigue wear caused by the penetration of grinding particles, friction, increased temperature, and deformation under real operating conditions. Hou et al. [9] conducted an all-inclusive investigation of the most common crankshaft failure modes, including distortion, wear, and fracture mechanisms, and provided relevant findings on environmental characteristics and failures. Similar failure investigations were conducted by Fonte et al. [10,11] for diesel engines used in different applications, while Pandey [12] investigated the premature fatigue failure of tractor diesel engine crankshafts made of 0.45% carbon steel, focusing on the web region of the crankshaft. Fatigue mechanisms were determined through mechanical tests, microstructural analysis, and fractography; the stress levels required for crack initiation and propagation were estimated, and corrective measures such as increasing the fillet radius and careful machining were proposed to prevent failure recurrence. Comparable conclusions were also reported by Espadafor et al. [13], Alfares et al. [14], and Yu et al. [15].

Similar approaches based on fracture and fatigue analysis of mechanically loaded components with pronounced stress concentrations have been reported for additively manufactured polymer and nylon gears [16-18], as well as for powertrain components such as drive shaft housings [19], which confirms the strong impact of geometry-related design parameters on fatigue mechanisms.

While crankshaft fatigue has been extensively studied, previous research has mainly focused on material properties, manufacturing defects, or failure analysis. Considerably less attention has been devoted to the systematic investigation of the influence of geometric parameters and their mutual interactions on fatigue performance.

Accordingly, this study employs the finite element method to numerically investigate the role of selected geometric features on the fatigue behaviour of the crankshaft. Particular attention is given to how the crankpin fillet radius and the geometry of the splined shaft grooves, considered individually and in combination, influence the stress distribution. Through this analysis, the paper provides a clearer insight into the importance of geometric design parameters for improving crankshaft durability.

DEVELOPMENT OF COMPUTATIONAL FEA MODEL

The SolidWorks software is employed to develop the FEA model of the crankshaft. Based on the actual crankshaft (Figure 1a), a CAD model of the crankshaft (Figure 1b) was created, and then the crankshaft material (25CrMo4) with the basic mechanical traits shown in Table 1 was defined. The specified crankshaft material mechanical properties (Table 1) are integrated into the SolidWorks software package, so that they can be simply assigned to the created model, without additional data entry.

With the volumetric model formed, FEA modeling is performed. The first step in the formation of the FEM model of the crankshaft is to define the constraints. The analysed crankshaft is located in the beam mower assembly and is supported by two bearings. The locations where the bearings are mounted are treated as surfaces that are prevented from moving in the radial direction, while rotation is allowed.

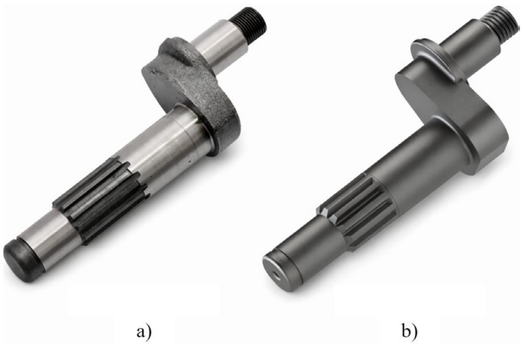

Figure 1. Crankshaft: (a) actual component, (b) 3Dmodel

Table 1. Mechanical properties of crankshaft made of 25CrMo4 [20]

Properties	Unit	Value
Modulus of elasticity	GPa	210
Yield strength	MPa	590
Poisson's ratio	–	0.3
Brinell hardness	HB	217
Shear modulus	GPa	80
Tensile strength	MPa	740
Ductility	%	15
Density	kg/m^3	7850

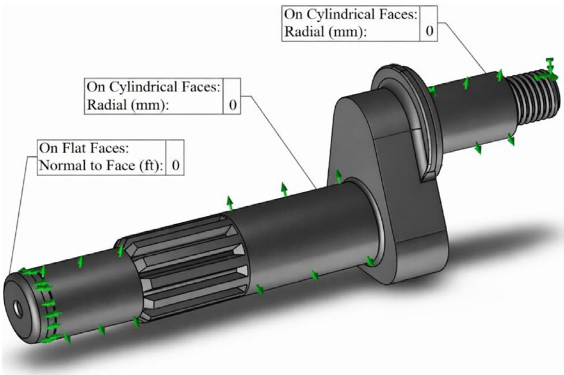

Figure 2. Setting the constraints on the crankshaft

At the location of the retaining ring, restrictions on movement in the axial direction are set. The crankshaft transmits torque via a crown nut, and there is a bearing between the crown nut and the shaft, which enables the rotation of the shaft. In order to simulate real conditions as realistically as possible, it is necessary for the journal on which the bearing and the crown nut are located to be prevented from moving in any direction (referring to the radial direction of the crankpin journal). This simulates the case of the mower blade encountering an obstacle, thus preventing the shaft from rotating and achieving the most unfavourable operating conditions for the shaft. The part of the shaft on which the thread for the crown nut is located is fixed, which is also the last restriction (Figure 2).

After setting the constraints, the next step is to define the crankshaft load. Since it is a 4.5 kW beam mower engine with a power take-off shaft rotating at 800 rpm, and a transmission ratio of 2.0 between the power take-off shaft and crankshaft, it is easy to calculate the value of 26.857 Nm of torque transmitted by the crankshaft. The torque is transmitted at the splined shaft, and it is necessary to mark the surface of each groove, define the torque direction, and spread the torque value equally on each surface of the groove (Figure 3).

The last step in the development of FEA model of the crankshaft is to generate a finite element mesh. SolidWorks provides three types of finite element meshes. Before conducting the analysis, it was necessary to verify the reliability of the mesh, particularly in critical regions such as the crankpin fillet and groove areas where stress concentrations are most pronounced. By comparing several different mesh densities, it was observed that the results stabilized, meaning that further mesh refinement no longer significantly affected the peak stress values. This indicated that the mesh was sufficiently accurate. For the discretization of the crankshaft model, a curvature-based mesh with parabolic tetrahedral finite elements with a minimum size of 0.3 mm and a maximum size of 1.5 mm is selected, total number of finite elements is 181,518 (Figure 4).

Following the FEA modeling of the crankshaft, a stress analysis is conducted for a designated load. The resulting von Mises stress distribution and accompanying displacment values are analyzed (Figures 5 and 6).

As shown in Figure 5, the peak von Mises stress occurs at the crankpin journal, at the transition from the crankpin journal diameter to the gasket seat diameter. A fillet radius of 0.5 mm was made at the mentioned transition place. Such transitions with no fillet radius at all or with a small value of fillet radius represent places of stress concentration, and potentially problematic places when analysing fatigue of the crankshaft.

As presented in Figure 6, the greatest displacement amounting to 0.01399 mm occurs on the grooved part of the crankshaft. It can be

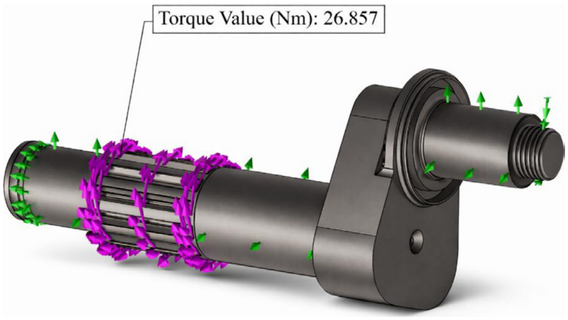

Figure 3. Setting the load on the crankshaft

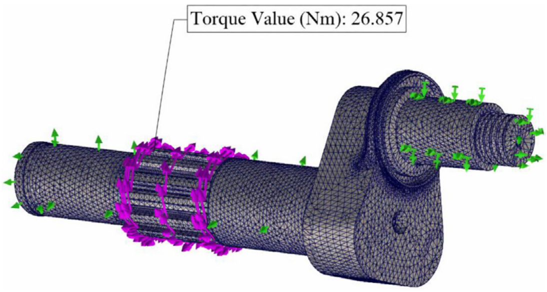

Figure 4. Setting a finite element mesh on a crankshaft

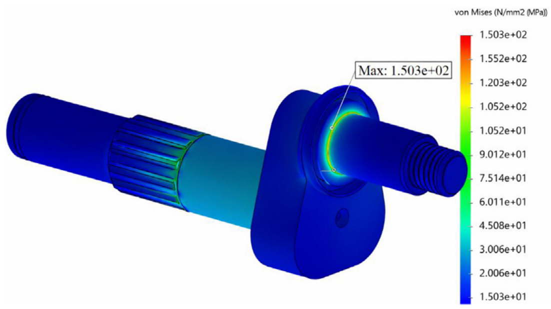

Figure 5. Values and distribution of von Mises stress

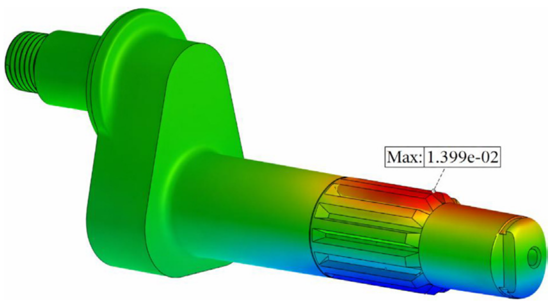

Figure 6. Values and distribution of displacement

noticed that the displacements are of a very small value, so in the analysis of geometrical effect on the fatigue life of the crankshaft, displacement will not be observed, and only the focus will be on the von Mises stresses.

FATIGUE LIFE PREDICTION

Predicting the service life of mechanical elements within the range of operational conditions is crucial for any reliable application and economicaly sound design [21]. In general sense, the data upon stress and service lifespan, directly linked to the operating load, leverage a reliable fatigue life built on linear representation of incremental damage [22]. The alternating stress apropos the number of cycles to failure can be represented via the S-N curve, used to fairly evaluate the fatigue life at specific operating load [23]. Therefore, designation of the S-N curves proves particularly useful within the context of acquiring relevant data on fatigue life, or, more specifically, fatigue damage under different service conditions.

For the engine crankshafts, fatigue failure life is predominantly influenced by the bending stress, showcasing as more severe compared to the accompanying torsional stress, which is often neglected [24, 25]. Based on crankshaft dynamics, the bending stress is obtained according to Equation 1:

$$\sigma_{BN} = K_e K_S \frac{(F_{Rmax} - F_{Rmin})L_1}{2W_{eq}} \tag{1}$$

where: K_e is the correction coefficient, K_S is the empirical factor of stress concentration, generally corresponding to the dimensions and proportions, L_1 is the bending length, and W_{eq} is the resisting moment of inertia.

Calculation of the crankshaft journal force within the specified operating conditions can be achieved using the engine operating models and dynamic formulae. Xu et al. [26] state in their research that the crankshaft journal force shows obvious non-linear behaviour under uncertain operating conditions.

Because the crankshaft load is asymmetric, fatigue life cannot be determined directly from the S-N curve. Accordingly, the Goodman curve must be used to transform the stress amplitude, as shown in Equation 2 [27–29]:

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_b} = 1 \tag{2}$$

The Goodman relation is based on the assumption of a linear interaction between mean and alternating stress relative to the material's ultimate tensile strength. Due to its simplicity and the fact that it provides safe, conservative estimates,

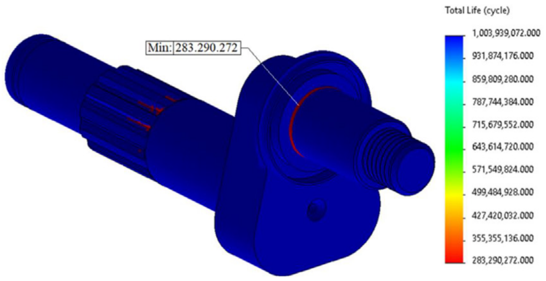

Figure 7. The number of cycles to crack initiation

it remains a standard choice in engineering practice. Therefore, the resulting equivalent stress, σ_e, is yield, as shown in Equation 3:

$$\sigma_e = \frac{\sigma_a}{\left(1 - \frac{\sigma_m}{\sigma_b}\right)} \tag{3}$$

where: σ_b is the ultimate strength, σ_m is the mean stress, $\sigma_m = \frac{\sigma_{dmax} + \sigma_{dmin}}{2}$, and σ_a is the calculated stress amplitude, $\sigma_a = \frac{\sigma_{dmax} - \sigma_{dmin}}{2}$.

Once operating stresses are determined, the fatigue life for each loading phase is estimated using the material's S-N curve. Following the cumulative fatigue damage theory, damage at each stage is predicted. The S-N curve is defined by the Basquin equation [23,30,31], expressed by Equation 4:

$$\sigma_e = \sigma_f \left(2N_f\right)^b \tag{4}$$

where: b is the fatigue-related exponent, σ_f is the fatigue strength coefficient, and N_f is the number of cycles in which fatigue failure occurs.

This study employs 25CrMo4 crankshaft material, with the basic mechanical characteristics presented in Table 1, while the S-N curve can be seen in reference [32].

RESULTS

After the fatigue analysis of the base model of the crankshaft, the results are presented in Figure 7. The figure shows the crack occurrence probability at the point of peak stress value. These are exactly the fillet radius on the crankpin journal and the grooves of the splined shaft. The number of cycles to crack initiation in these places, for the base model of the crankshaft, amounts to 283,290,272 cycles.

Based on these results and recognized critical points on the crankshaft, steps were taken to increase the crankshaft lifespan. The first change to the base model, with a goal to extend the fatigue life of the crankshaft, is to increase the fillet radius on the crankpin journal from 0.5 mm up to 2.0 mm, with a 0.5 mm step (Figure 8). A radius of 2.0 mm is the maximum radius that can be made. The results of stress analysis at two critical points are presented in Figure 9. The number of cycles to crack initiation for this case amounts to 335,610,688.

Another modification to extend the lifespan of the crankshaft is a modification of the grooves of the splined shaft, i.e., rounding at the groove root. Rounding at the groove root is performed with a radius of 0.2 mm, 0.5 mm, and 0.8 mm. Their impact on the fatigue life of the crankshaft is shown in Figure 10. Larger radii are not

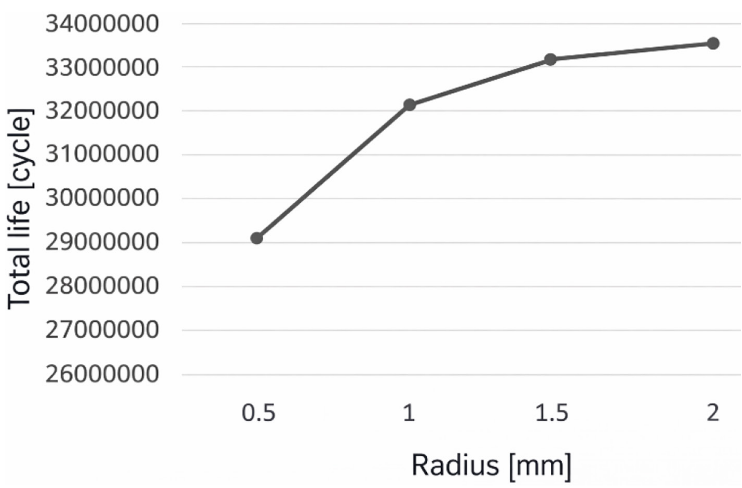

Figure 8. The influence of the fillet radius modification on the number of cycles

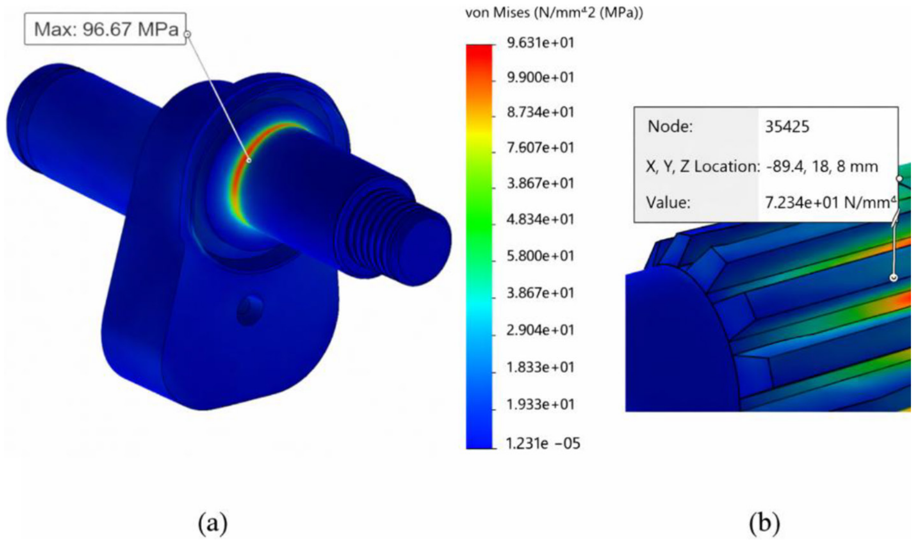

Figure 9. Von Mises stresses for an increase of the fillet radius on the crankpin journal (a) fillet radius on the crankpin journal, (b) grooves of the splined shaft

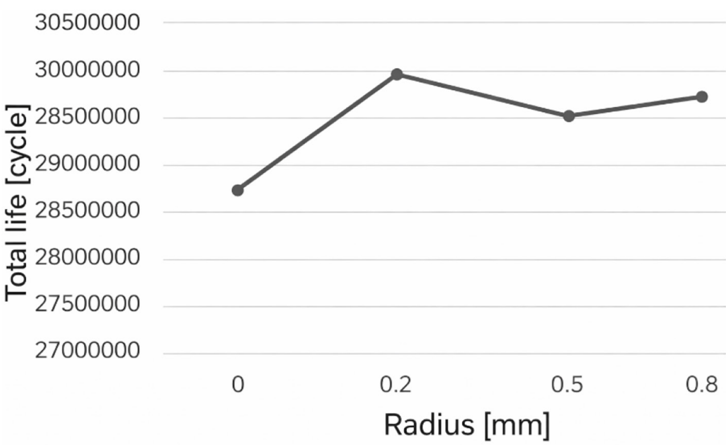

Figure 10. The influence of geometrical modification of the grooves of the splined shaft on the number of cycles

possible due to the distance to the groove root. The results of the stress analysis at two critical points are presented in Figure 11. The number of cycles at which a crack occurs for this case is 300,641,920.

The third modification for extending the fatigue life of the crankshaft is to combine the previous two modifications, i.e., simultaneous modification of the fillet radius on the crankpin journal and modification of the grooves of the splined shaft. Combining was done by taking certain diameter ratios during the analysis, while monitoring their influence on the fatigue life of the crankshaft. The ratio of the radius 2.0/0.2 showed the best performance. The fatigue life of the crankshaft for this case is 355,417,728 cycles (Figure 12).

The results of the structural and fatigue analysis of the based model crankshaft design, as well as all geometrically modified crankshafts, are shown in Table 2.

The structural analysis of the base model of the crankshaft showed the critical places where fatigue fracture initiation occurs. The highest values of von Mises stress were recorded in the zone of the fillet radius of the crankpin journal, where, due to the change in the cross-section, there is a pronounced concentration of stress. The maximum stress in that zone is 150.30 MPa, while the estimated number of cycles until the appearance of the first fatigue crack is 283,290,272 cycles. The reason for this behavior is the sharp stress gradient at the transition zone, triggered by the tight radius. Such

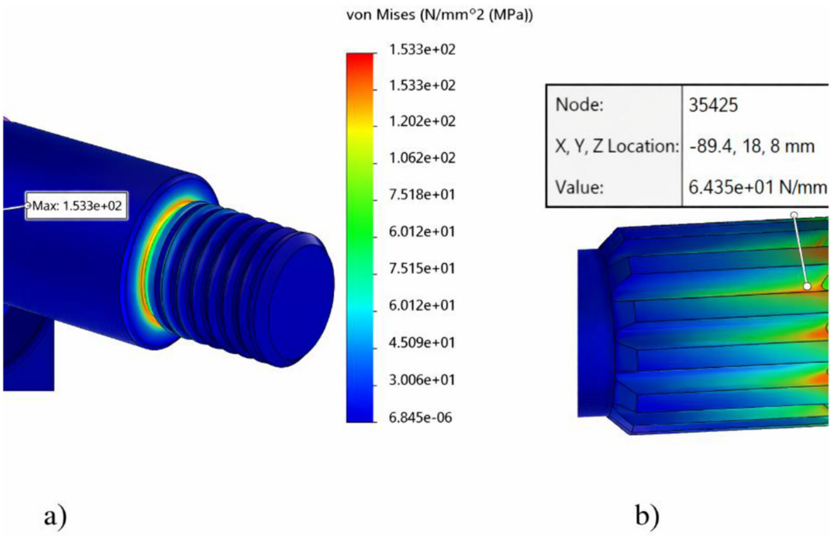

Figure 11. Von Mises stresses the modifications of the grooves of the splined shaft, (a) fillet radius on the crankpin journal, (b) grooves of the splined shaft

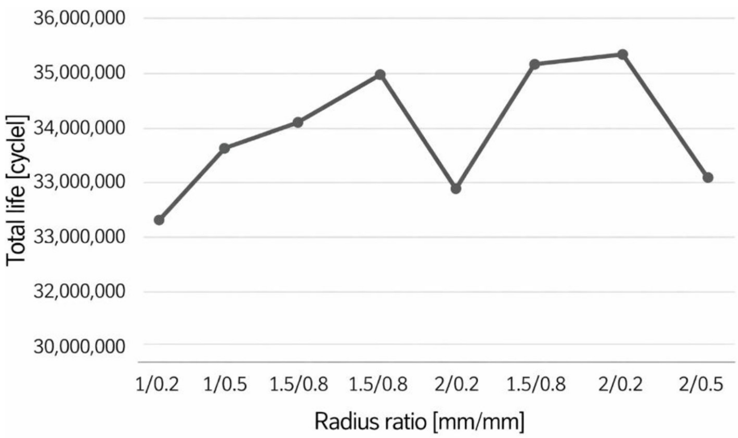

Figure 12. The influence of the interaction of fillet radius and splined shaft groove geometrical modification on the number of cycles

Table 2. Results of structural analysis and fatigue analysis

Crankshaft	von Mises stresses (MPa) Location 1	von Mises stresses (MPa) Location 2	Number of cycles (-)
Base model	150.30	70.48	283,290,272
Modification of the fillet radius of the crankpin journal	96.67 (-36%)	72.34 (+2.6%)	335,610,688 (+18.5%)
Modification of the geometry of the groves of the splined shaft	153.30 (+2%)	64.35 (-8.7%)	300,641,920 (+6.1%)
Combination of modification of the crankpin journal and the groves of the splined shaft	91.40 (-39%)	67.64 (-4%)	355,417,728 (+25%)

Note: values in parentheses represent percentage improvements (stress reduction or life extension) relative to the Base model. Negative percentages indicate a reduction in stress, while positive percentages in the "Number of cycles" column indicate an increase in durability.

geometric jumps function as local stress intensifiers, which greatly raises the likelihood of fatigue cracks. This occurs because fatigue life is extremely sensitive to maximum stress levels in these localized areas.

In order to extend the fatigue life of the crankshaft, certain geometric modifications were made.

The first analysed modification was related to the increase of the fillet radius of the crankpin journal. The results showed that this modification leads to a significant reduction of the maximum von Mises stresses at that location, from 150.30 MPa to 96.67 MPa, which represents a reduction of 36%. At the same time, the number of cycles to fatigue failure increased from 283,290,272 to 335,610,688 cycles, or by about 18.5%. The improvement is a direct result of the smoother transition provided by a larger fillet radius. This larger radius eases stress concentration by spreading the load across a wider area and reducing the local stress gradient. By lowering the stress intensity at the notch root, we delay crack initiation and substantially boost fatigue endurance.

The second modification analysed was related to the change in the geometry of the grooves of the splined shaft, more precisely, the change in the geometry of the rounding in the groove root. In this case, a slight increase in the maximum stress at location 1 was recorded, from 150.30 MPa to 153.30 MPa, while at the same time, there was a decrease in the stress at location 2 from 70.48 MPa to 64.35 MPa, which represents a decrease of 8.7%. This redistribution of stress resulted in an increase in the number of cycles to fatigue failure to 300,641,920, which represents an increase of approximately 6% compared to the base model. Unlike the fillet radius modification, the effect of groove geometry is more about stress redistribution than a direct reduction of the overall maximum stress. Rounding the groove root eases local stress concentration in that specific area, but it also shifts how the load is transferred through the shaft, potentially leading to a slight stress increase at neighboring critical points. This interaction explains both the modest improvement in fatigue life and the non-uniform variation in stress values.

The most favourable results were obtained by combining modifications of the fillet radius of the crankpin journal and the geometry of the grooves of the splined shaft. In this case, the maximum von Mises stresses at location 1 were reduced to 91.40 MPa, which represents a reduction of over 39% compared to the base model. As a result of such a pronounced reduction in stress concentration, the number of cycles to fatigue failure increased to 355,417,728, which represents an increase of about 25%. The improved performance of the combined configuration stems from simultaneous stress relief across multiple critical zones. While the increased fillet radius effectively trims the dominant stress peak at the crankpin journal, the modified groove geometry ensures a more favorable redistribution of loads along the splined shaft. This synergy results in a more uniform stress field and gentler gradients, which delays crack initiation and bolsters fatigue resistance.

CONCLUSIONS

Based on the structural and fatigue analysis, it can be concluded that the geometric design parameters have a decisive influence on the stress distribution and the fatigue life of the crankshaft. The importance of proper shaping of the structures in critical transition zones is particularly emphasized, because in this way a significant extension of the fatigue life can be achieved without changing the material or increasing the mass of the structure. The finite element method has proven to be a reliable and efficient tool for identifying critical zones and optimizing the geometry of the crankshaft.

A detailed analysis of the obtained results determined the following:

- the base model of the crankshaft is characterized by a pronounced concentration of stress in the zone of the fillet radius of the crankpin journal, which represents a critical place for the initiation of fatigue fracture and limits the life of the shaft.
- increasing the fillet radius of the crankpin journal leads to a significant reduction of the stress concentration in the critical zone, which results in a noticeable extension of the fatigue life of the crankshaft,
- modification of the geometry of the grooves of the splined shaft affects the redistribution of stress, reducing the stress in certain critical areas, which results in a moderate increase in the fatigue life of the shaft
- the combined optimization of the fillet radius of the crankpin journal and the geometry of the grooves of the splined shaft gives the best results, because the synergistic action results in the most favourable stress distribution and the increase in the number of cycles to failure compared to the base model.

The results of this research provide numerical insight into the relationship between geometric design parameters and the fatigue behaviour of crankshafts, and may support further research on geometry-based design optimization.

Future work may include experimental validation of the presented numerical results, as well as the consideration of more complex loading scenarios and additional geometric parameters, in order to achieve a more comprehensive understanding of crankshaft fatigue behaviour.

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