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Design and Simulation of Cracks in A Four-Cylinder Engine Crankshaft Using Finite Element Method

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Abstract. The translational motion of the piston was transformed into a rotational motion by the crankshaft. The combustion resulted in straight thrust in the piston was passed through the piston rod and pushed the crank pin, resulting in a rotational force on the crankshaft. In other words, the crankshaft received high loading and rotated at high speed in each cycle of work. Therefore, crankshafts should be made of high carbon steel with a high endurance limit. The major cause of failure is thermal fatigue since the contact between journal and bearing surface occurred. The contact may result from two reasons: problems of lubrication and use. Based on the factor of safety calculation, crank 1 and 4 were considered to meet the safety criteria because their positions were close to the support of the crankshaft

1. Introduction

When combustion takes place in a cylinder, the energy generated from gas combustion is extremely high [1]. This energy is transferred to the piston and then passes through the connecting rod to the crankshaft. The crankshaft serves a function to transmit the power released from the process of combustion to the flywheel and then to the gearbox, providing power to run the car's engine, radiator, dynamo, and AC compressor [2].

During its operation, a crankshaft receives inertial force and combustion force generated in the combustion chamber which causes torsional force and bending force. Crankshafts are generally made of case-hardened alloy steel to make them resistant to wear [3]. Poor material selection may lead to a decrease in the performance of crankshaft components since the maximum stress generated in the cylinder is very high [4]. Material damage usually starts from defects on the material surface due to the influence of environmental factors such as corrosion or wear due to interaction with other components [5]. The corrosion rate is dependent much on the solution concentration[6].

One serious failure in the component parts of a four-cylinder car engine is the occurrence of cracks in the crankshaft. Wear is caused by friction between the crank pin and piston position (round bearing), resulting in grooves that create a stress concentration area and that causes the initial cracking of crankshaft, as shown in Figure 1.

Fonte et al. [7] stated that cracks in crankshaft could be caused by poor design and use of alloy steel. Further, Zhiwei and Xiaolei [8] pointed out that crankshaft failure occurs due to partial loss of nitride layers resulting from over-nitriding results. In addition, crankshaft failures can also be caused by resonant vibrations generated by an unbalanced crankshaft [4]. High cycle fatigue due to resonant vibrations causes wear on the crankshaft.

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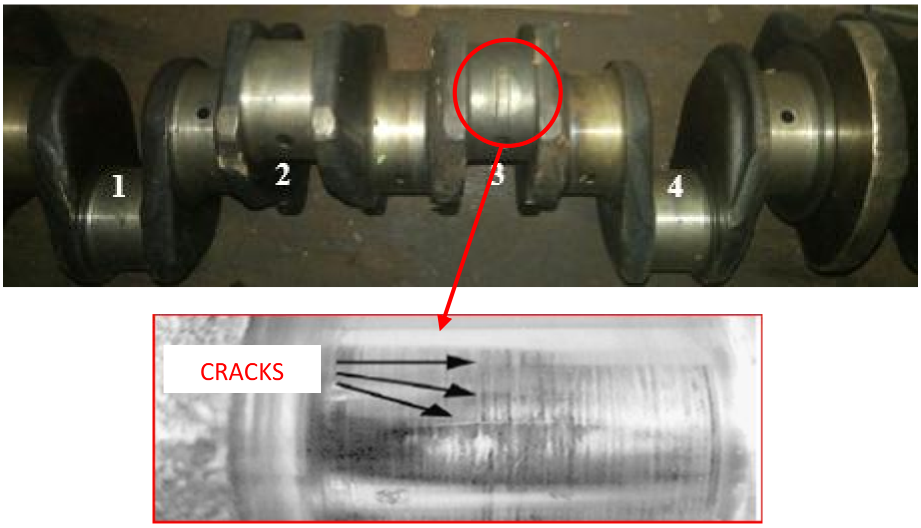

Figure 1. Cracks in crank 3

The above explanation suggests that, in order to extend crankshaft life, effective and efficient production planning is necessary, including analysis of loading and selection of material components. With the aim of investigating the damage in crankshaft due to dynamic loading, it is necessary to conduct a simulation analysis which aims to determine the stress distribution that occurs.

2. Method

Finite element method (FEM) was used to estimate the stress in the crankshaft. Numerical simulation is, in fact, a cost-effective and economical method of designing a crankshaft. The finite element method is a numerical method used to solve engineering problems and mathematical problems of a physical phenomenon [9]. The finite element analysis offers a way of conducting easy and efficient research on various parameters used with design and manufacturing conditions that are easily evaluated.

2.1. Contruction model

In this study, the crankshaft was designed using software called Autodesk Inventor Professional 2012 with dimensions of a common crankshaft for a four-cylinder engine. The crankshaft is usually made of carbon steel because, during use, it is spun at high speeds and under heavy loads [10]. Moreover, Naik [11] stated that carbon steel is a suitable material for producing crankshafts although it requires additional heat treatment to achieve the desired properties. Its material properties include density = 7870 kg/m3, Young's modulus = 200000 MPa, Poisson's ratio = 0.29, yield strength = 350 MPa, and ultimate tensile strength = 420 MPa.

2.2 Simulation

A simulation was done on the crankshaft to determine its strength using ANSYS Workbench 15. During simulation, the crankshaft was subjected to various loading conditions, such as a force of 4527.08 N, a moment of 130 Nm, an engine speed of 4000 RPM, and a mesh size of 12 mm. The magnitude of force given on the crankshaft simulation depended on the firing order (FO).
Analysis of simulation results showed the stress and deformation of the crankshaft under maximum loading conditions. The analysis results became the basis for conducting crack analysis on crank 3, as the crank had cracked.

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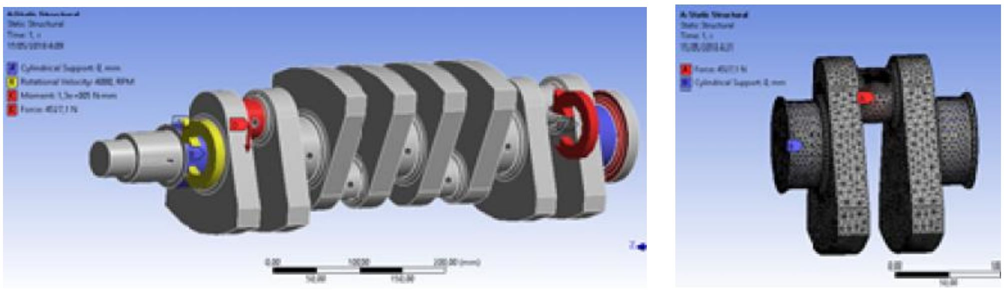
The crack testing on crank 3 was done when a crack occurred at connecting rod journals. The crack was located at Y = 55.358 mm and had dimensions of 8 mm (major radius), 2 mm (minor radius), and 2 mm (large contour radius) with a mesh size of 13 mm. The given force was 4527.08 N.

3. Result and discussion

3.1 Stress and deformation



(a) Loading on each crank (b) Crack test loading

Figure 2. Dynamic simulation on the crankshaft

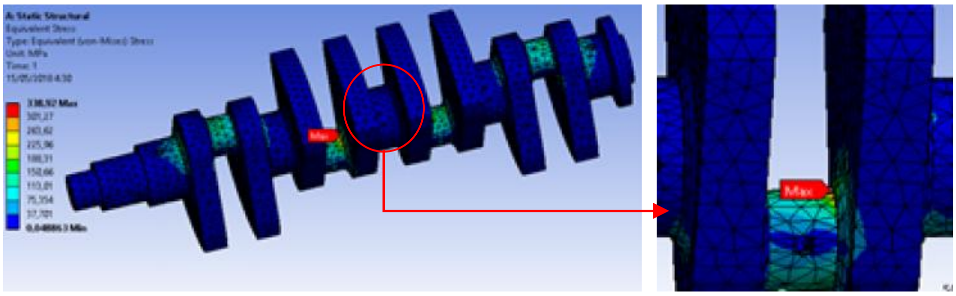


Figure 3. Equivalent (von Mises) stresses on crank 1

Figure 3 shows the result of simulation on crank 1 at maximum stress. The highest maximum equivalent stress (indicated in red) was 338.92 MPa, while the lowest (indicated in blue) was 0.048919MPa. Figure 4 shows the result of simulation on crank 2 subjected to equivalent stress loading. The highest maximum equivalent stress was 370.33 MPa, while the lowest was 0.045612 MPa.

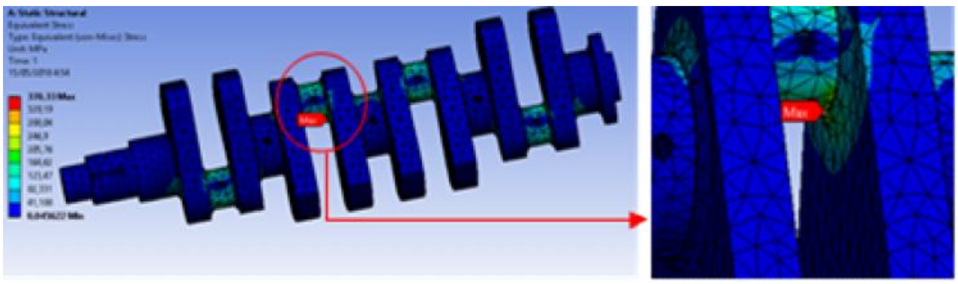


Figure 4. Equivalent (von Mises) stresses on crank 2

The simulation on crank 3 subjected to equivalent stress (see Figure 5) showed that the highest maximum equivalent stress was 370.16 MPa, while the lowest was 0.045693 MPa. Figure 6 shows the result of simulation on crank 4 subjected to equivalent stress loading. The highest maximum equivalent stress found was 341.36 MPa, while the lowest was 0.048738 MPa.

Figure 7 shows the total deformation of each crank. The highest total deformation of (a) crank 1 was 4.3327 x 10^5 mm. The highest total deformation (b) crank 2 was 4.3444 x 10^5 mm. The highest total deformation of (c) crank 3 was 4.3444 x 10^5 mm. The highest total deformation of (d) crank 4 was 4.3326 x 10^5 mm.

The maximum stress and deformation always occurred in crank 3 and 4 because the positions of which were far from the crankshaft supports; unlike crank 1 which was near the pulley and crank 4 which was near the flywheel. This is in line with Naik [11] stating that failure occurs in the cranks located at the centre and subjected to maximum bending moment. In fact, maximum bending moment and stress in the cylinder are the contributing factors to failure.

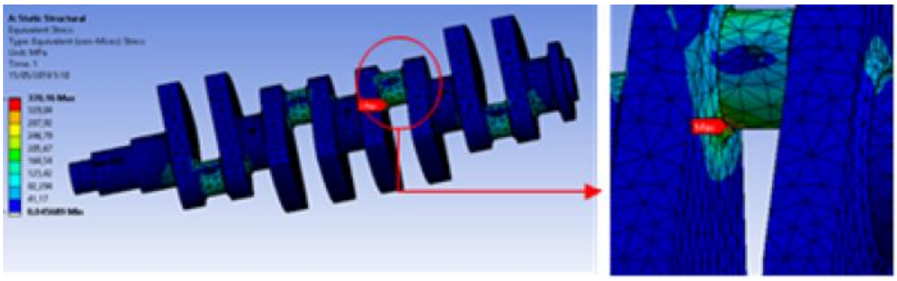


Figure 5. Equivalent (von Mises) stresses on crank 3

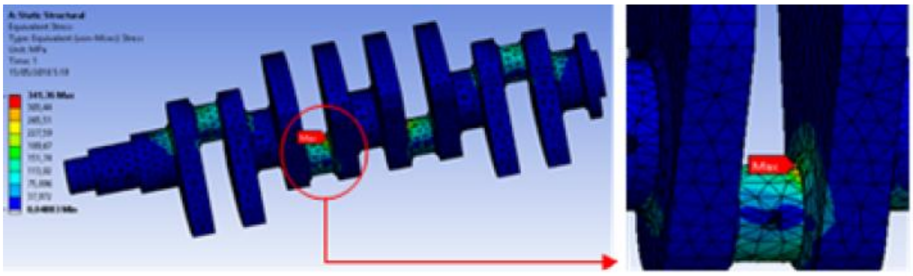
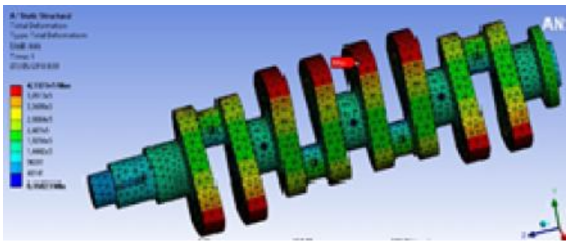
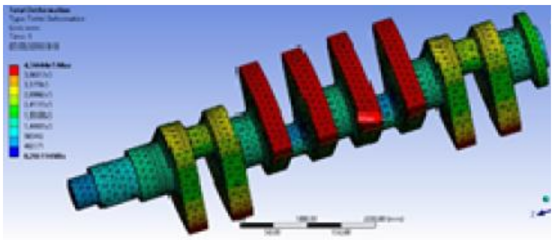


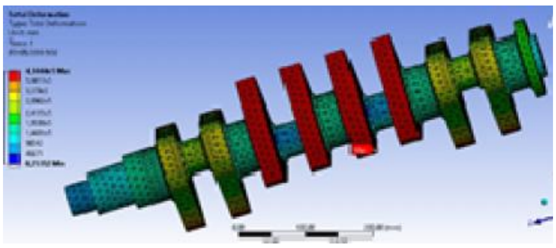
Figure 6. Equivalent (von Mises) stresses on crank 4



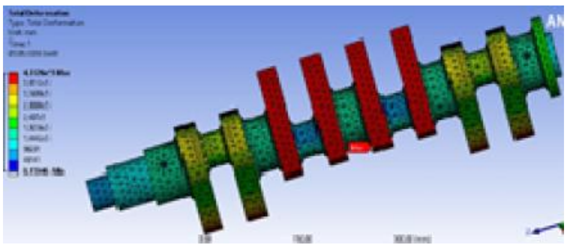
(a) Crank 1



(b) Crank 2



(c) Crank 3



(d) Crank 4

Figure 7. Total deformation of each crank

3.2 Crack testing
The crack occurring in crank 3 was examined based on the value of J-Integral and SIFS (K1). J-Integral value is the strain energy release rate of a crack body per unit. The maximum J-Integral value was 0.68696 mJ/mm^2, while the minimum J-Integral value was -0.44977 mJ/mm^2. Moreover, the SIFS (K1) value was used to determine the stress intensity factor (K) of a material with a certain geometry shape under elastic loading. The maximum value of SIFS (K1) was -0.0041005 MPa.mm$^{0.5}$, while the minimum value was -0.33756 MPa.mm$^{0.5}$.

According to Naik [11], the major cause of failure is thermal fatigue since the contact between journal and bearing surface occurred. The contact may result from two reasons: problems of lubrication and use. Constant use of components results in wear and tear especially in the range of 58,000 km. Also, a lack of lubricating oil leads to direct contour between big end and crank pin [11]. When the small cracks simultaneously spread except the main cracks, the existing elastic energy for the growth from the main cracks certainly would be reduced, particularly in view of the creation of the larger crack surface; thus reducing the general level of crack growth. In some cases, it locally can cause a sudden crack. In addition, the orientation of the ferrite laths in matrix can influence the crack path [12].

3.3 Safety factor
Safety factor is used to perform evaluation so that the safety of the engine components is ensured even though the dimensions used are minimum [13]. The safety factor in this study was based on the maximum equivalent stress. The use of the maximum von Mises stress aimed to identify the combined stresses, i.e. the principal stress (x, y, and z axes) and the maximum stress.The factor of safety calculation showed that the safety factor of crank 1 was 1.03, crank 2 and 3 was 0.94, and crank 4 was 1.02. Crank 1 and 4 were considered to meet the safety criteria proposed by Vidosic [14] because their positions were close to the supports. Crank 1 was adjacent to the pulley and crank 4 adjacent to the flywheel.

4. Conclusions
The crankshaft simulation analysis was done based on the loading specified in each crank angle according to FO. The maximum crankshaft stress and deformation always occurred in crank 2 and 3 because the positions of which were far from the support of the crankshaft; unlike crank 1 which was near the pulley and crank 4 which was near the flywheel. The major cause of failure is thermal fatigue since the contact between journal and bearing surface occurred. The contact may result from two reasons: problems of lubrication and use. Based on the factor of safety calculation, crank 1 and 4 were considered to meet the safety criteria.

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