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Stress and Deformation Analysis of a Crankshaft in Range Extender Engine

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Abstract: Crankshaft is an essential part of an internal-combustion engine. In this paper, 3D Model of the crankshaft was made using solidWorks software. SolidWorks software was used to analyze the point of maximum stress and maximum deformation point of the two materials to be used in production (Ti-8AL-1Mo-1V and 40CrMnMo7). The material characteristic, crankshaft dimension and obstacle of boundary conditions were established as the constrains to the simulation. Finally, the point of maximum stress and maximum deformation point were analyzed. Moreover, static strength and fatigue evaluation of crankshaft were also done and the results obtained are used as valuable reference in the optimization and improvement to the crankshaft design.

Key words: Crankshaft, stress, deformation, range extender, solidworks, material Ti-8AL- 1Mo-1V, material 40CrMnMo7

INTRODUCTION

Crankshaft is one of the moving components and most important part of the internal combustion engine. Its function is to convert back and forth movement of the piston into rotational movement (Meng *et al*., 2010; Gopal *et al*., 2016). Crankshaft strength affects the reliability and life of the internal combustion engine. Crankshaft is supported by main bearings. Moreover, it obtains cyclic bending moment of the connecting rod consequence of the force generated by the combustion in the cylinder. The force produces torque on crankshaft. The torque generates stress and deformation on crankshaft (Fonte *et al*., 2013).

The main cause of failure on crankshaft is the fatigue phenomenon. The fatigue phenomenon on crankshaft is caused by high stress on the crankshaft. The highest stress on the crankshaft is located on the counter weight fillet area, the main journal and lubrication holes (Meng *et al*., 2010; Fonte *et al*., 2015a, b; Pandey, 2003; Fonte and Freitas, 2000; Teng *et al*., 2015).

Two cylinder crankshaft will be used on range extender engine of 999 cc capacity which is applied to hybrid car. The power of range extender engine is about 22 kW, speed of 3200 rpm and torque 66 Nm.

Crankshaft failure caused by several factors which exceeds the capacity of a sudden, the maintenance of which is not good, improper machine operation, fatigue, a condition produced by the cyclic loading stress levels higher than the strength of the material (Alfares *et al*., 2007).

In the past studies the beam and frame models usually were used to calculate stress on crank throw. However, there were only limited number of nodes in the models. With the development of computer engineering, a lot of crank throw design is analyzed using Finite Element Method (FEM) in software simulation to calculate the stress and maximum deformation of the crank throw (Meng *et al*., 2010; Taylor *et al*., 1999; Sun *et al*., 2016). Crankshaft work with harmonic torque combined with cyclic bending stress due to the pressure of the combustion chamber radial force transmitted from the piston and connecting rod as well as the load inertia of the piston and connecting rod (Fonte *et al*., 2015a, b; Becerra *et al*., 2011).

In this study, stress and maximum deformation analysis of crank throw on crankshaft when the crank receives maximum twisting moment is discussed. The analysis was done using SolidWorks simulation software. Crank throw model that created using solid works Software and analyzed using. Solid works simulation software allows finite element method analysis (Glodova *et al*., 2014) for this study. Stress analysis was performed on a set of crank throw because it was assumed that the other crank throws gets the same force and torque. Analysis of stress and deformation of crank throw

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was simulated under restriction of materials mechanical characteristic, crank throw dimension and torque to determine the point of maximum stress and deformation experienced by crank throw. The material used in this study is the Ti-8AL-1Mo-1V and 40CrMnMo7. Analysis of von Mises stress, shear stress and deformation will be used as the basis for choosing a good material.

MATERIALS AND METHODS

Crank throw design and modeling

Crank throw design: Range extender engine specifications for the design of crank throw are tabulated in Table 1. Figure 1 shows the forces acting on the crank throw.

When the crank is at an maximum twisting angle the force on the connecting rod (F_Q) is divided into two forces, the tangential force (F_T) and the radial force (F_R). When maximum twisting moment on crank occurs, the value of the tangential force is also maximum. In the gasoline combustion engine maximum tangential force value occurs when the crank shaft is at an angle of 25-30° (Khurmi and Gupta, 2005).

The F_Q is influenced by the amount of force generated by the cylinder due to combustion (F_P). The amount of force generated from the combustion chamber was calculated using Eq. 1:

$$F_p = \frac{\pi}{4} \times D^2 \times P \qquad (1)$$

Where:
P = The pressure in the combustion chamber
D = Piston diameter

The amount of pressure was 55 bar. When main bearing journal experienced a torque due to tangential force acting on the crankpin Eq. 2 was used to calculate the value of the torque:

$$T_C = H_{TI} \times r \qquad (2)$$

Where:
T_c = The value of the torque experienced by the main journals bearing (Nm)
H_{TI} = A tangential force that occurs in the crankpin
r = Distance between the centres of the main journals bearing with crankpin centre (mm)

So, the moment of the torque experienced by crank throw can be calculated using the following Eq. 3:

$$\sum Te = \sqrt{MC^2 + TC^2} \qquad (3)$$

Table 1: Specifications of range extender engine

Parameters	Units
Capacity	999 cc
Number of cylinders	2
Bore×stoke	86×86 mm
Maximum power	22 kw @3200 rpm
Maximum gas pressure	55 bar

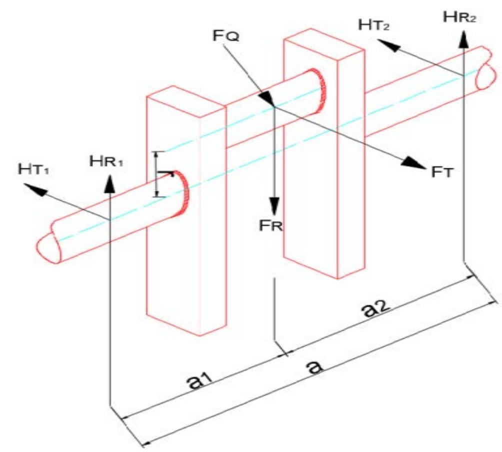

Fig. 1: Force distribution of crank throw when the crank is at an maximum twisting angle

After obtaining the value of the moment of torque, diameter of crankpin can be calculated using the following Eq. 4:

$$dc = \sqrt[3]{\frac{Te}{\tau x \pi / 16}} \qquad (4)$$

where, τ shear stress value which is limited to 35N mm^{-2} to the crank throw (Taylor *et al*., 1999). The von Mises stress induced in the crankpin is:

$$\text{von mises stress} \sqrt{(Kb + Mc)^2 + \frac{3}{4}(Kt \times Te)^2} \qquad (5)$$

where, K_b is combined shock and fatigue factor for bending with value of 2, K_t is combined shock and fatigue factor for torsion, value for K_t is 1.5 (Brahmbhatt and Choubey, 2012).

Crank throw modeling: Crank throw main dimensions shown in Table 2. Dimension of the crank throw affects the value of stress, deformation that occur in crank. In this simulation, dimension is set as a boundary condition in the simulation process. Crank throw was created using solidworks Software. Figure 2 shows a 3D Model of crank throw. Mesh model crank throw was made using solidWorks software. It has 379948 elements and 543283 nodes. The meshing dimension influences the number of elements, nodes and analysis result. Figure 3 shows the mesh model crank throw.

Table 2: Primary dimensions of crank counterweight

Dimensions	Units (mm)
Pin diameter	40
Pin axial length	32
Main journal diameter	45
Crank cheek thickness	10
Crank cheek height	142

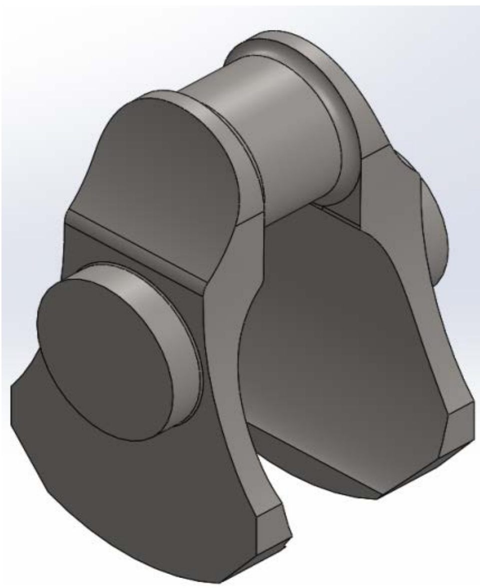

Fig. 2: The 3D Model crank throw

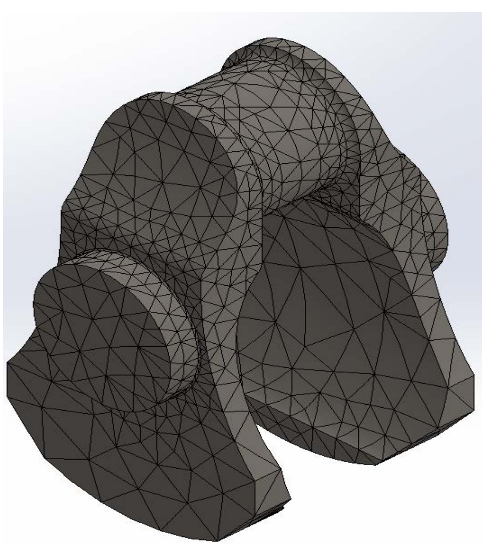

Fig. 3: Mesh model of crank throw

RESULTS AND DISCUSSION

In this study, maximum von Mises, maximum shear stress and maximum deformation of crank throw is analyzed. The mechanical properties of the material are used as the physical parameters to analyze maximum stress and maximum deformation. In this study two materials were selected for the analysis. Materials were selected based on the yield strength, density, poission's ratio. Those said paremeters will be used as standard for real crank shaft production. The mechanical properties of the two materials were taken from solidWorks material database. The material properties are shown in Table 3.

Table 3: The properties of selected materials

Properties	Ti-8Al-1Mo-1V	40CrMnMo7
Shear modulus	46000 MPa	79000 MPa
Mass density	4370 Kg m^{-3}	7800 kg m^{-3}
Tensile strength	937 MPa	992 MPa
Yield strength	910 Mpa	821 MPa

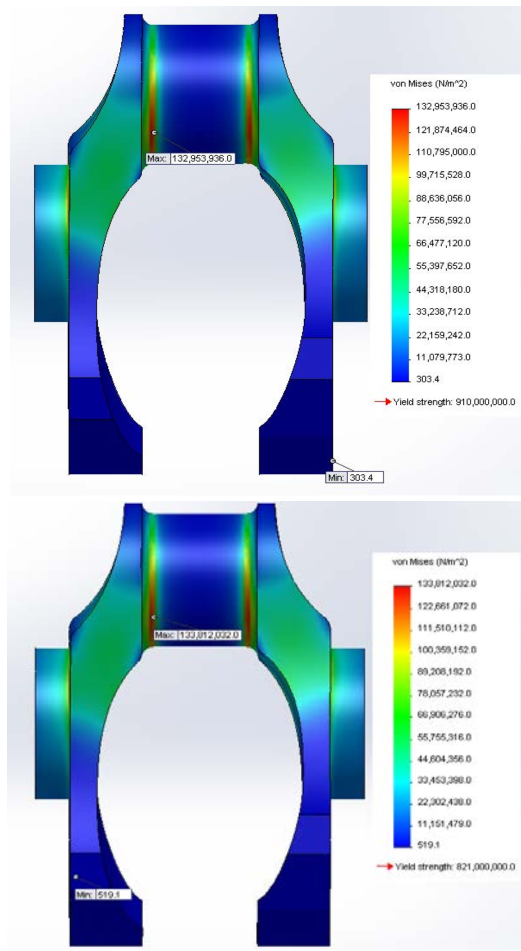

Fig. 4: Result of von Mises on the crank counterweight: a) material Ti-8Al-1Mo-1V; b) material 40CrMnMo

Stress analysis: Analysis stress was performed to find out maximum stress point where maximum strain happens on the crank throw. Von Mises stress that occurs on crank throw is shown in Fig. 4.

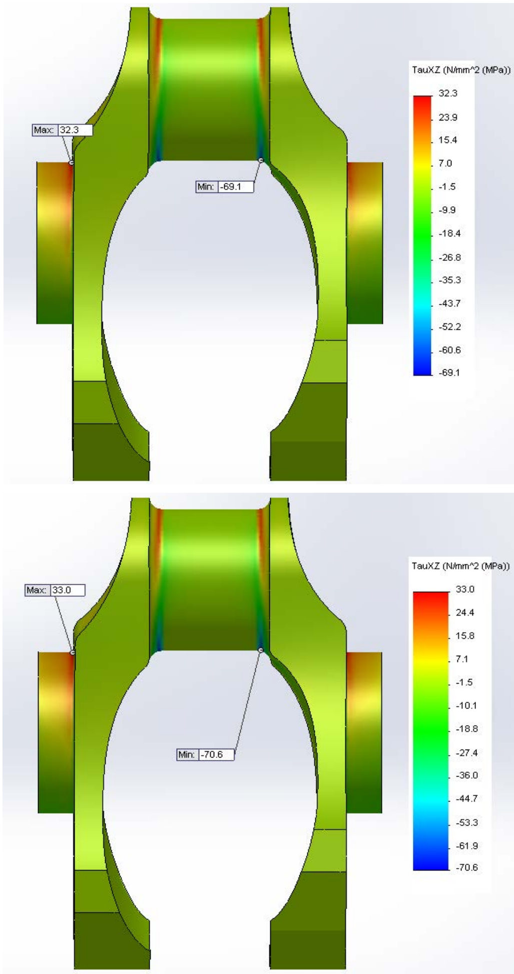

Fig. 5: Result of shear stress on the crank throw: a) material Ti-8Al-1Mo-1V; b) material 40CrMnMo7

The maximum von Mises stress lies in the crank throw fillet, that is between the counterweight and the crankpin of crank counterweight. The maximum von Mises stress of both materials are under each material yield strength. Material 40CrMnMo7 has a yield strength of 910 N/mm^2 and the material Ti-8AL-1Mo-1V has a yield strength of 821 N mm^{-2}. Minimum von Mises point is located at the end of the ballast crank throw. The maximum shear stress occurs in crank throw is shown in Fig. 5.

The maximum shear stress in the material Ti-8AL-1Mo-1V is 32.3 N mm^{-2} and the maximum shear stress in the material 40CrMnMo7 is 33.0 N mm^{-2}. The point of maximum crank throw shear stress of both materials are at the same location which is, the fillet located between the counterweight and the main journal bearing.

When compared, there is little difference of the stress values in theory and simulation. However, the difference

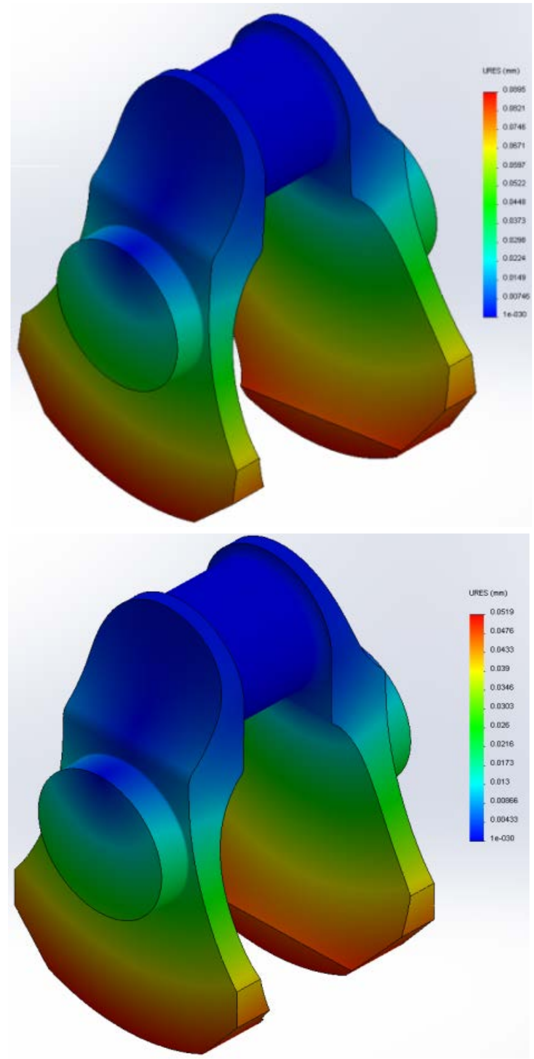

Fig. 6: Result of deformation distribution on the crank counterweight: a) Material Ti-8Al-1Mo-1V; b) Material 40CrMnMo7

Table 4: Comparison of theorical simulation of stress on the materials

Stress (N mm^{-2})	Theory	Ti-8Al-1Mo-1V	40CrMnMo7
Von Mises	132.52	132.954	133.8
Shear stress	32.4	32.3	33.0

is not significant. Table 4 shows the comparison of theoretical simulation of stress on both selected materials. Table 4 Comparison of theorical simulation of stress on the materials

Deformation analysis: Deformation analysis was performed to evaluate the maximum deformation position on crank throw. Distribution of deformation under maximum stress is shown in Fig. 6. The maximum deformation value under maximum stress of the crank throw for 8Al-1Mo-1V material is 0.0895 mm. The maximum deformation value of the crank throw for 40CrMnMo7 material is 0.0519 mm.

When the cylinder is burned, maximum deformation of the crank throw is located at the end of the counterweight. The value of the maximum deformation to 8Al-1Mo-1V material is greater than the maximum deformation value to 40CrMnMo7 material, by 0.0376 mm difference.

CONCLUSION

In this study, the design of crank throw which is part of a two-cylinder crankshaft and is used on a two-cylinders range extender engine with capacity of 999 cc was created. Crank throw designed was done using two different materials to test their physical strength and determine suitability for use in real production. In both of these materials, the point of maximum stress on von Mises lies at the same location. This points lies on fillet between crank pin and counterweight. The point of maximum shear stress and the point of maximum deformation occur at the same location. The point of maximum shear stress in the fillet is located between the main journals and counterweight. The point of maximum deformation occurs at the end of both ends of the counterweight. In this study, the performance of Ti-8AL-1Mo-1V material is better than 40CrMnMo7 material as material crankshaft. Because, its von Mises stress and shear stress are smaller than that of 40CrMnMo7 material. Beside that, mass density of Ti-8Al-1Mo-1V material is lighter than 40CrMnMo7 material. In the future, the results of this study will serve as guidelines for optimizing the design and calculation of fatigue of crank shaft.

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