Momin Muhammad Zia Muhammad Idris / International Journal of Engineering Research and Applications (IJERA) ISSN: 2248-9622 www.ijera.com
Vol. 3, Issue 1, January -February 2013, pp.1694-1698

Crankshaft Strength Analysis Using Finite Element Method

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ABSTRACT

The crankshaft is an important component of an engine. This paper presents results of strength analysis done on crankshaft of a single cylinder two stroke petrol engine, using PRO/E and ANSYS software. The three dimensional model of crankshaft was developed in PRO/E and imported to ANSYS for strength analysis. This work includes, in analysis, torsion stress which is generally ignored. A calculation method is used to validate the model. The paper also proposes a design modification in the crankshaft to reduce its mass. The analysis of modified design is also done.

Keywords – ANSYS, Crankshaft, Finite Element Method, PRO/E, Strength Analysis

I. INTRODUCTION

In strength analysis, considering loads acting on the component, equivalent stresses are calculated and compared with allowable stresses to check if the dimensions of the component are adequate. Crankshaft is an important and most complex component of an engine. Due to complexity of its structure and loads acting on it, classical calculation method has limitations to be used for strength analysis [1]. Finite Element Method is a numerical calculation method used to analyze such problems. The crankpin fillet and journal fillet are the weakest parts of the crankshaft [1] [2]. Therefore these parts are evaluated for safety.

II. FINITE ELEMENT MODEL

Fig.1 shows the 3-Dimensional model in PRO/E environment. As the crankshaft is of a single cylinder two stroke petrol engines used for two wheelers, it doesn't have a flywheel attached to it, a vibration damper and oil holes, making the modeling even simpler. The dimensions of crankshaft are listed in Table 1.

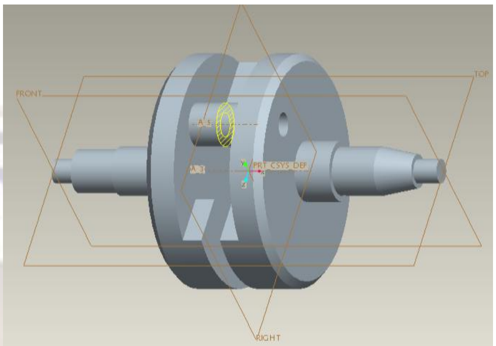

Fig.1 The 3-Dimensional model in PRO/E

Table1. DIMENSIONS OF CRANKSHAFT

Parameter	Value (mm)
Crankpin Outer Diameter	18
Crankpin Inner Diameter	10
Journal Diameter	25
Crankpin Length	50
Journal Length	10
Web Thickness	13

The procedure of using FEM usually consists of following steps. (a) modeling; (b) meshing; (c) determining and imposing loads and boundary conditions; (d) result analysis

A. Meshing

Greater the fineness of the mesh better the accuracy of the results [5]. The Fig. 2 shows the meshed model in ANSYS consisting of 242846 nodes and 67723 elements.

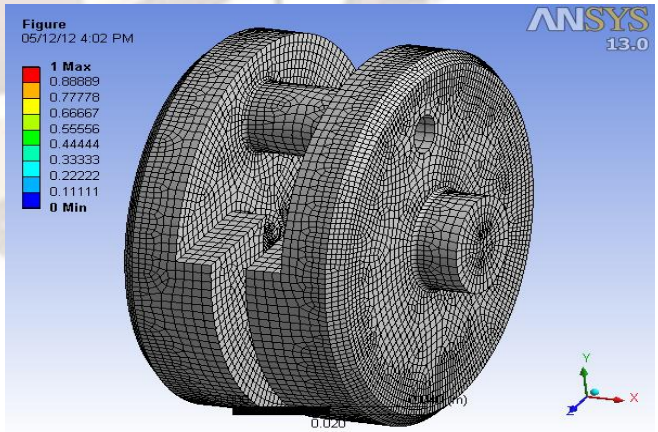

Fig.2 Meshing the model in ANSYS

B. Defining Material Properties

The ANSYS demands for material properties which are defined using module

ENGINERING DATA. The material used for crankshaft is 40Cr4Mo2.The material properties are listed in Table 2.

Table 2 . THE MATERIAL PROPERTIES

Density	7800 kg m^-3
Young's Modulus	2.05e+011
Poisson's Ratio	0.3
Tensile Strength	7.7e+008

C. Loads and Boundary Conditions

Boundary conditions play an important role in FEM. Therefore they must be carefully defined to resemble actual working condition of the component being analyzed. The crankshaft is subjected to three loads namely Gas Force F, Bending Moment M and Torque T. The boundary conditions for these loads are as follows [3].

1. Gas Force F

Gas Force F is calculated using maximum cylinder pressure, 50 bar for petrol engines [4], and bore diameter of engine cylinder. This load is assumed to be acting at the centre of crankpin. Displacements in all three directions (x, y and z) are fully restrained at side face of both journals as shown in Fig.3. From this loading case, maximum compressive stress in the journal fillet is obtained.

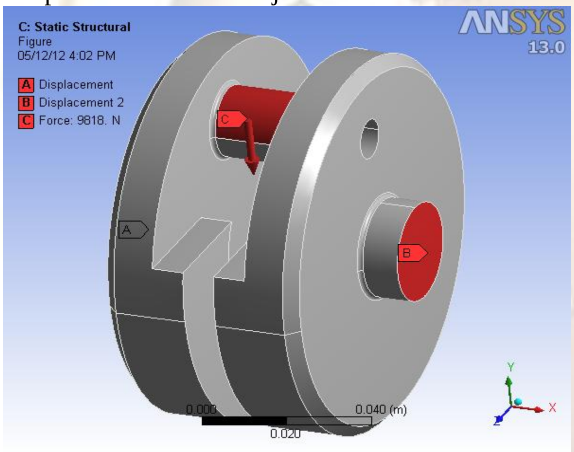

Fig.3 Gas Force applied at the centre of crankpin

2. Bending Moment M

For strength analysis crankshaft is assumed to be a simply supported beam with a point load acting at the centre of crankpin. The maximum Bending Moment M is calculated accordingly. One journal of the crankshaft is kept free (six degree of freedom) and Bending Moment M is applied to this journal as shown in Fig.4. The degrees of freedom at the other journal are fully restrained. From this loading case maximum bending stresses in the crankpin fillet and journal fillet are obtained.

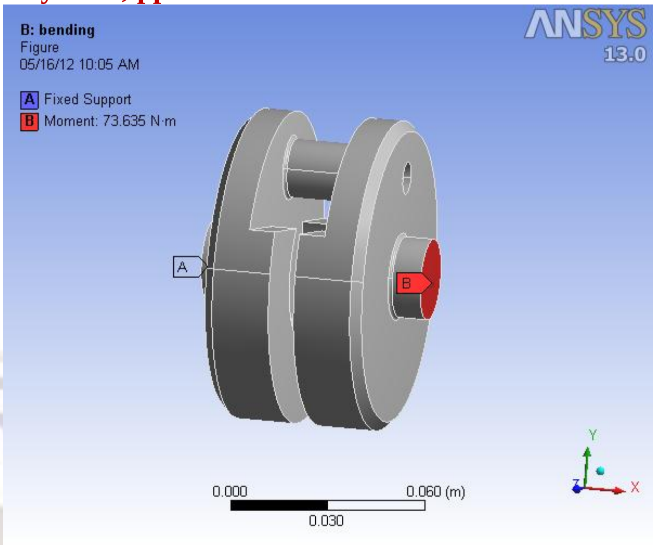

Fig.4 Bending Moment applied at one of the journals

3. Torque T

Maximum Torque T is obtained from manufacturer's engine specifications. One journal of the crankshaft is kept free (six degree of freedom) and Torque T is applied to this journal. The degrees of freedom at the other journal are fully restrained as shown in Fig.5. From this loading case maximum torsion stress in crankpin fillet and journal fillet are obtained.

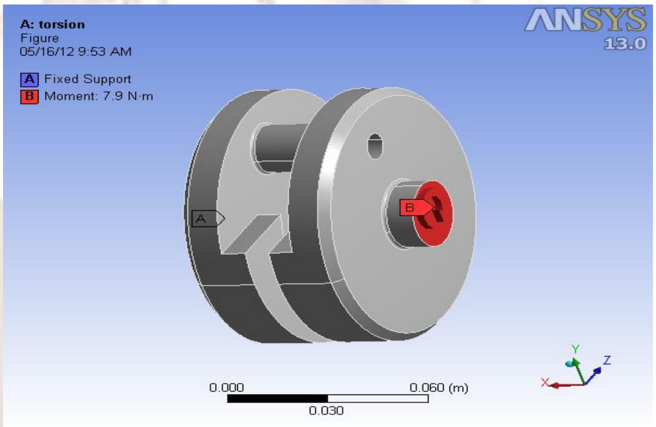

Fig.5 Torque applied at one of the journals

D. Calculation of Equivalent Stress

As the boundary condition in each load case is different, it is impossible to combine them in ANSYS to find equivalent stress. Therefore, stress values obtained from various load cases are used in formulae given in [3] to obtain equivalent stress in crankpin fillet and journal fillet. As the load on the crankshaft is fluctuating, the equivalent stress is to be compared with fatigue strength of crankshaft material. This is done by calculating fatigue strength σDW and acceptability factor Q as given in [3].

Fatigue Strength:

$$\sigma DW = \pm K.(0.42.\sigma B + 39.3)[0.264 + 1.073.D^{-0.2} + \frac{785 - \sigma B}{4900} + \frac{196}{\sigma B}\sqrt{\frac{1}{RH}}]$$

Where

$\sigma B[N/mm^2]$ minimum tensile strength of crankshaft material

K [-] factor for different types of crankshafts without surface treatment. Values greater than 1 are only applicable to fatigue strength in fillet area.

= 1.05 for continuous grain flow forged or drop-forged crankshafts

= 1.0 for free form forged crankshafts (without continuous grain flow)

RH [mm] fillet radius of crankpin or journal

$\sigma DW = \pm 468.24\ N/mm^2$ related to crankpin fillet

$\sigma DW = \pm 413.3\ N/mm^2$ related to journal fillet

Acceptability Factor:

$$Q = \frac{\sigma DW}{\sigma v} \quad (1)$$

Adequate dimensioning of the crankshaft is ensured if the smallest of all acceptability factors satisfies the criteria [3]:

Q ≥ 1.15

1. Equivalent Stress σv and Acceptability Factor Q in Crankpin Fillet

The maximum bending stress and torsion stress in crankpin fillet were obtained from equivalent stress diagrams for the load cases Bending Moment and Torque respectively. (Fig.6 and Fig.7)

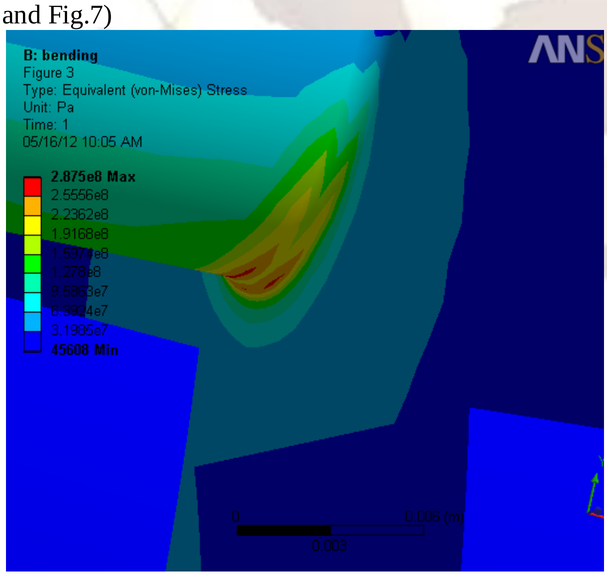

Fig.6 Maximum bending stress in crankpin fillet

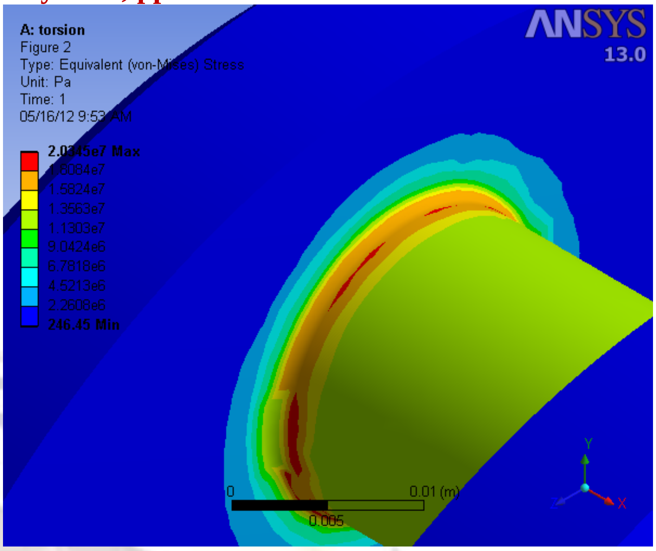

Fig.7 Maximum torsion stress in crankpin fillet

The Equivalent Stress in crankpin fillet is calculated as:

$$\sigma v = \pm\sqrt{\sigma BH^2 + 3 \times \tau H^2} \quad (2)$$

$$= \pm\sqrt{287.5^2 + 3 \times 20.34^2}$$

$\sigma v = \pm 289.65\ N/mm^2$

The Acceptability Factor is calculated using (1)

Q =1.616

2. Equivalent Stress σv and Acceptability Factor Q in Journal Fillet

The maximum bending stress, torsion stress and maximum compressive stress in journal fillet were obtained from equivalent stress diagrams for the load case Bending Moment, Torque and Gas Force respectively. (Fig.8, Fig.9 and Fig.10)

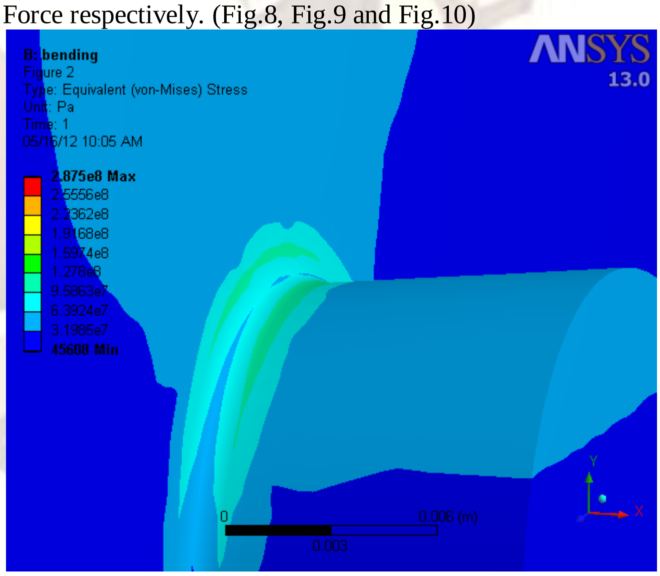

Fig.9 Maximum bending stress in journal fillet

The Equivalent Stress in journal fillet is calculated as:

$$\sigma v = \pm\sqrt{\sigma BG^2 + 3 \times \tau G^2} \quad (3)$$

$$= \pm\sqrt{266.46^2 + 3 \times 9.042^2}$$

$\sigma v = \pm 267.01\ N/mm^2$

The Acceptability Factor is calculated using (1)

Q =1.547

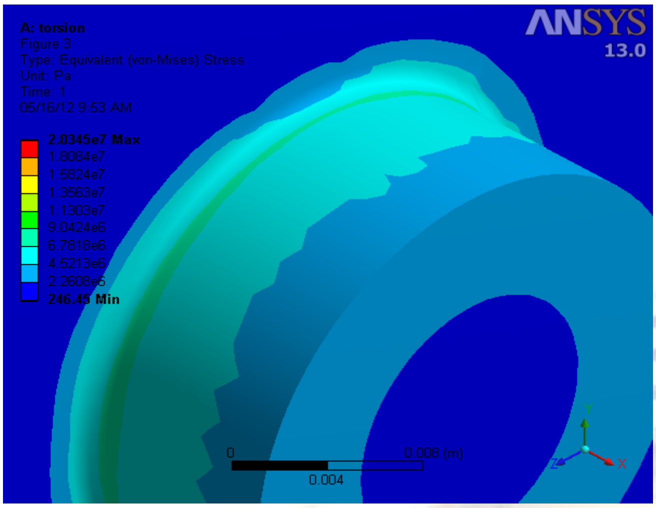

Fig.9 Maximum torsion stress in journal fillet

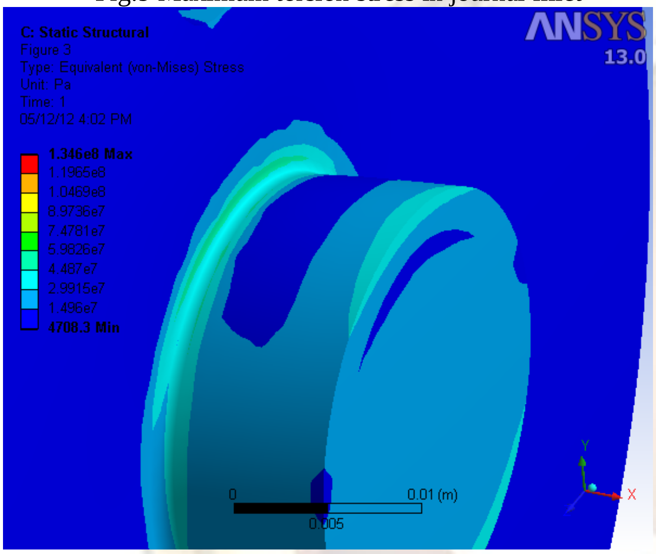

Fig.10 Maximum compressive stress in journal fillet

III. MODEL VALIDATION

Alternatively, a classical calculation method given in [3] was used to validate the model. The equivalent stress and acceptability factor were calculated and compared with values obtained from Finite Element Method described earlier.

1. Equivalent Stress σv and Acceptability Factor Q in Crankpin Fillet

The Equivalent Stress in crankpin fillet is calculated as:

$$\sigma v = \pm\sqrt{\sigma BH^2 + 3 \times \tau H^2} \qquad (4)$$

$$= \pm\sqrt{301^2 + 3 \times 14.83^2}$$

$$\sigma v = \pm 468.24\ N/mm^2$$

The Acceptability Factor is calculated using (1)

Q =1.55

2. Equivalent Stress σv and Acceptability Factor Q in Journal Fillet

The Equivalent Stress in journal fillet is calculated as:

$$\sigma v = \pm\sqrt{\sigma BG^2 + 3 \times \tau G^2} \qquad (5)$$

$$= \pm\sqrt{270.74^2 + 3 \times 5.018^2}$$

$$\sigma v = \pm 270.88\ N/mm^2$$

The Acceptability Factor is calculated using (1)

Q =1.525

IV. RESULT ANALYSIS

The stress concentration is high in crankpin fillet and journal fillet. The values of equivalent stress and acceptability factor obtained from FEM and classical calculation method were almost equal for both crankpin fillet as well as journal fillet. Therefore it is concluded that it is safe to consider stress values obtained from FEM for strength analysis. The results obtained from both the methods are listed in Table 3.

Table 3. RESULT ANALYSIS

Area	**Parameter**	**By FEM**	**By Calculation**
Crankpin Fillet	Equivalent Stress σv	289.65 N/mm^2	302.09 N/mm^2
	Acceptability Factor Q	1.616	1.55
Journal Fillet	Equivalent Stress σv	267.01 N/mm^2	270.88 N/mm^2
	Acceptability Factor Q	1.547	1.525

The large difference between the specified value of Acceptability Factor, Q ≥ 1.15, and its calculated value proved that crankshaft is over dimensioned. Therefore a scope for the improvement in the design was investigated. Web thickness was reduced from 13 mm to 9 mm. Then modified design of crankshaft was again analyzed using FEM. The results of this analysis are listed in Table 4.

Table 4.THE EQUIVALENT STRESS AND ACCEPTIBILITY FACTOR IN MODIFIED CRANKSHAFT

Area	**Parameter**	**Value**
Crankpin Fillet	Equivalent Stress σv	392.42 N/mm^2
	Acceptability Factor Q	1.193
Journal Fillet	Equivalent Stress σv	314.036 N/mm^2
	Acceptability Factor Q	1.316

V. CONCLUSION

Strength Analysis is a powerful tool to check adequacy of crankshaft dimensions and find scope for design modification.

The Strength Analysis of crankshaft of a single cylinder two stroke petrol engine was done and presented in this paper. Based on Result Analysis, a design modification is proposed. The torsion stress was also included in the analysis. It is found that weakest areas in crankshaft are crankpin fillet and journal fillet. The reduction in mass obtained by design modification is 38%. A dynamic analysis is required to be done for the modified design to study its vibration characteristics.

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