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Optimization of Crankshaft using Strength Analysis

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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, to optimize its design, 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 modal analysis of modified design is also done to investigate possibility of resonance.

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

Any physical system can vibrate. The frequencies at which vibration naturally occurs, and the modal shapes which the vibrating system assumes are properties of the system, and can be determined analytically using Modal Analysis. Analysis of vibration modes is a critical component of a design, but is often overlooked. Inherent vibration modes in structural components or mechanical support systems can shorten equipment life, and cause premature or completely unanticipated failure, often resulting in hazardous situations. Detailed modal analysis determines the fundamental vibration mode shapes and corresponding frequencies. This can be relatively simple for basic components of a simple system, and extremely complicated when qualifying a complex mechanical device or a complicated structure exposed to periodic wind loading. These systems require accurate determination of natural frequencies and mode shapes using techniques such as Finite Element Analysis [7].

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.

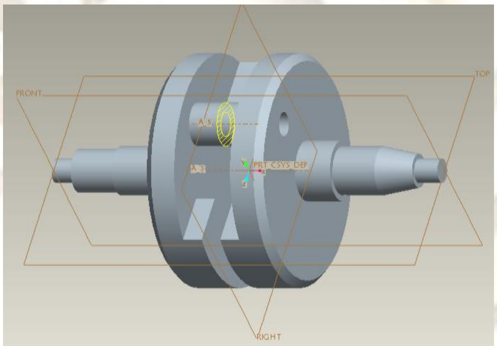

Fig.1 The 3-Dimensional model in PRO/E

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

III. STRESS CALCULATION USING FEM

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.

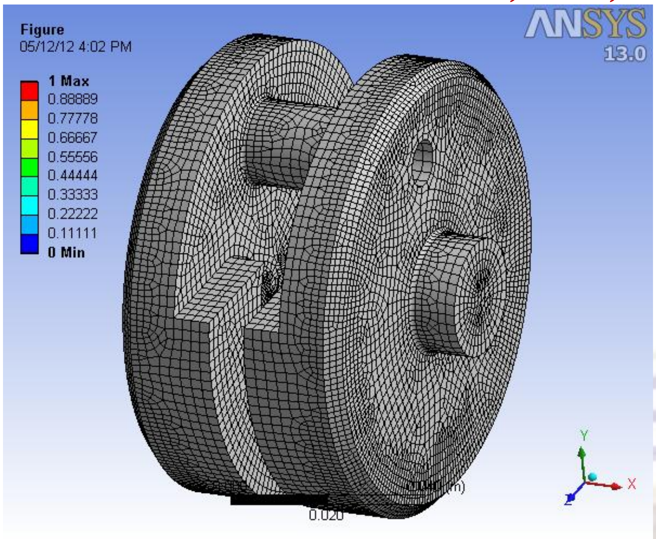

Fig.2 Meshing the model in ASYS

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+011Pa
Poisson's Ratio	0.3
Tensile Strength	7.7e+008 Pa

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.

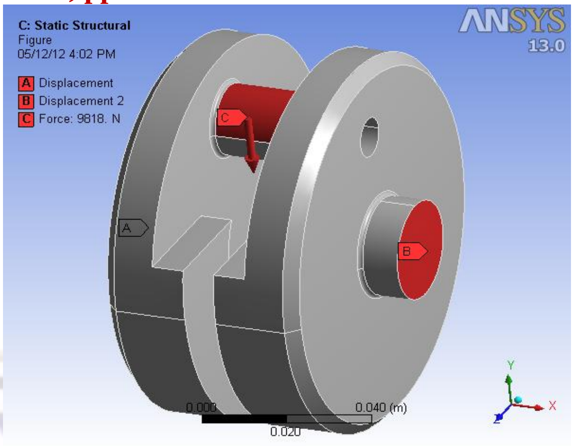

Fig.3 Gas Force applied at the centre of crankpin

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.

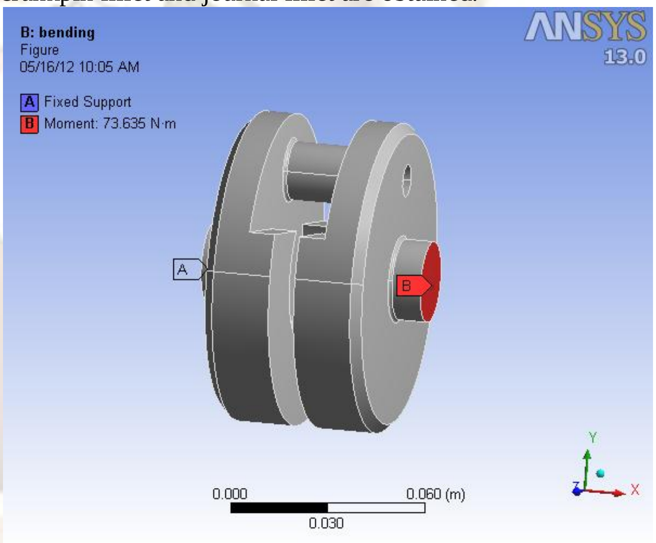

Fig.4 Bending Moment applied at one of the journals

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.

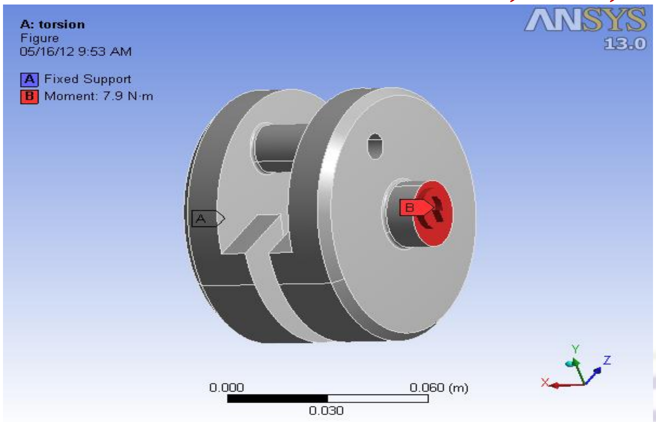

Fig.5 Torque applied at one of the journals

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)

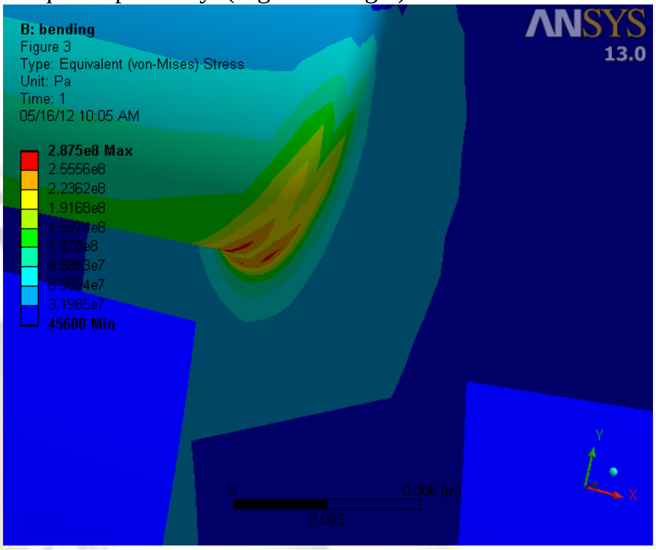

Fig.6 Maximum bending stress in crankpin fillet

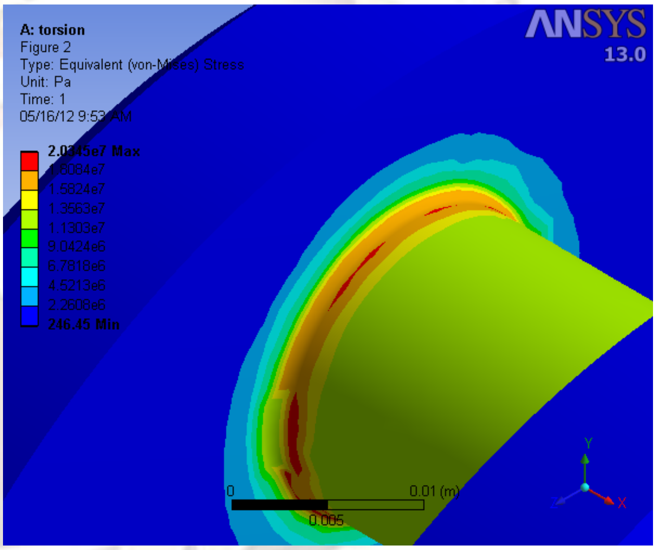

Fig.7 Maximum torsion stress in crankpin fillet

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)

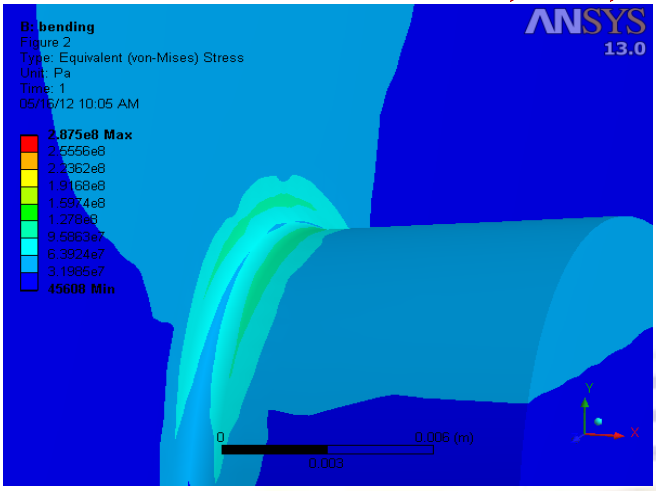

Fig.9 Maximum bending stress in journal fillet

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

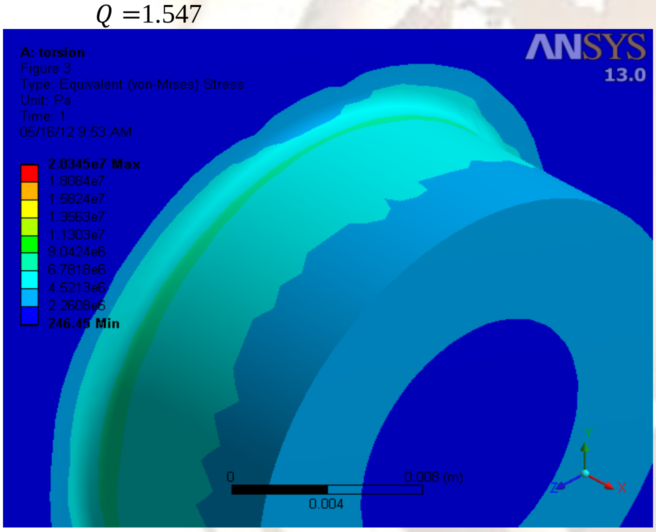

Fig.9 Maximum torsion stress in journal fillet

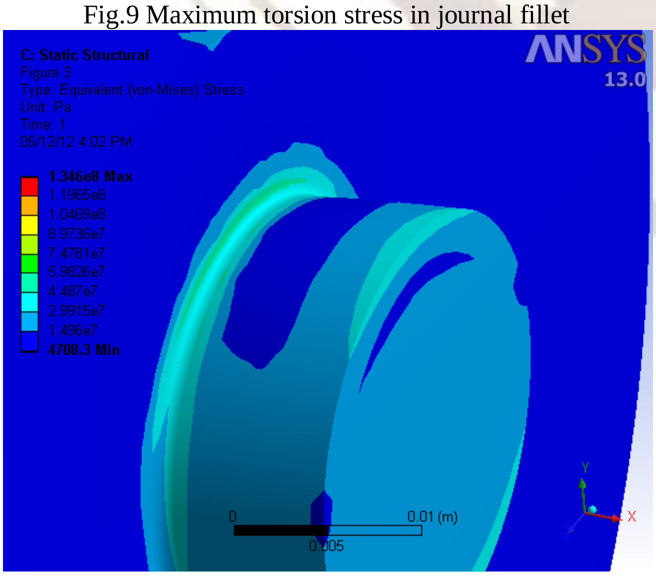

Fig.10 Maximum compressive stress in journal fillet

IV. 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:

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

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

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

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:

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

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

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

The Acceptability Factor is calculated using (1)

Q =1.525

V. 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.

Table3. 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. The original thickness of the web is 13 mm which is reduced in step of 1 mm and acceptability factor was calculated for each thickness value using Finite Element Method.

Table 4 lists the summary of these iterations which were done to arrive at final acceptable value of web thickness. It is clear from Table4 that web thickness can be reduced to 10 mm still keeping the acceptability factor above the specified limit (iteration No. 4). Although there is still a little margin between calculated value and specified value of acceptability factor, it is better not to reduce web thickness further to be on the safer side. Further reduction in web thickness results in acceptability factor values which are less than specified value (iteration No.5 and 6) indicating that web thickness can't be reduced below 10 mm.

Table4: DESIGN MODIFICATION STEPS

Sr. NO	Web Thickness mm	Area	Equivalent Stress σV (N/mm2)	Acceptability Factor (Q ≥ 1.15)
1	13	Crankpin Fillet	289.65	1.616
		Journal Fillet	267.01	1.547
2	12	Crankpin Fillet	352.08	1.329
		Journal Fillet	322.1	1.283
3	11	Crankpin Fillet	366.45	1.277
		Journal Fillet	333.84	1.238
4	10	Crankpin Fillet	400.76	1.168
		Journal Fillet	355.056	1.164
5	9	Crankpin Fillet	434.53	1.077
		Journal Fillet	361.98	1.141
6	8	Crankpin Fillet	481.7	0.972
		Journal Fillet	363	1.138

VI. MODAL ANALYSIS

In order to determine fundamental mode shapes and corresponding natural frequencies, Modal Analysis of the modified design of crankshaft was done. All six modes of vibration and corresponding natural frequencies were determined. Table 5 lists all six natural frequencies of vibration.

Table5. NATURAL FREQUENCIES OF VIBRATION

MODE	FREQUENCY [HZ]
1	1432.7
2	2332.
3	3215.
4	3598.9
5	3918.9
6	4218.4

First Mode of Vibration

The first mode of vibration is bending vibration in X-direction at natural frequency of 1432.7 Hz. The maximum deformation appears at the bottom of crank WEB, as shown in Figure11.

Second Mode of Vibration

The second mode of vibration is torsional vibration of right web about Y-direction at natural frequency of 2332 Hz. The maximum deformation appears at the sides of crank WEB as shown in Figure12.

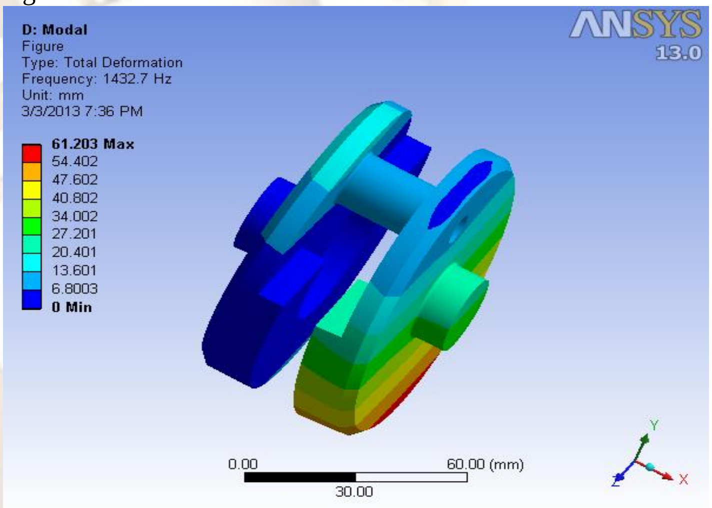

Figure 11 First mode of vibration

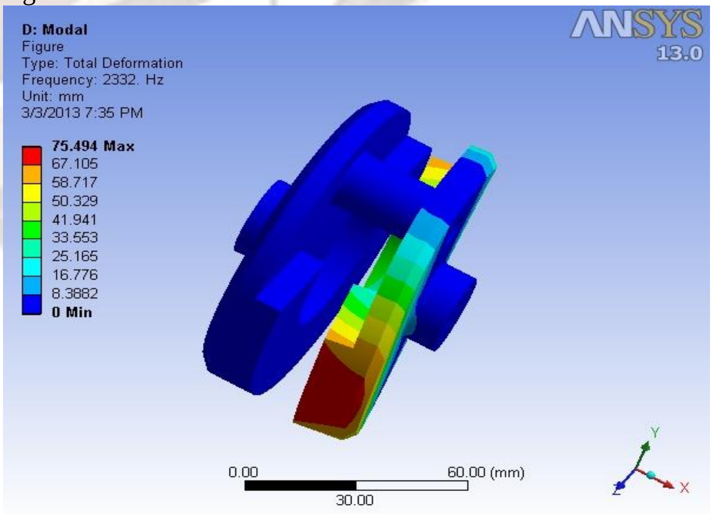

Figure 12 Second mode of vibration

Third Mode of Vibration

The third mode of vibration is torsional vibration about Y-direction at natural frequency of 3215 Hz. The maximum deformation appears at the sides of left crank WEB as shown in Figure 13.

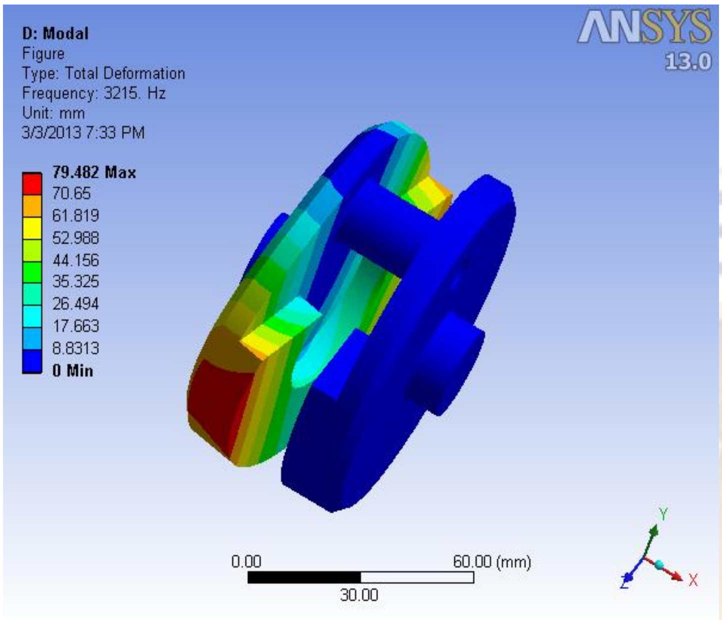

Figure 13 Third mode of vibration

Fourth Mode of Vibration

The fourth mode of vibration is torsion vibration about Z-direction at natural frequency of 3598.9 Hz. The maximum deformation appears at the edges of crank WEB as shown in Figure 14.

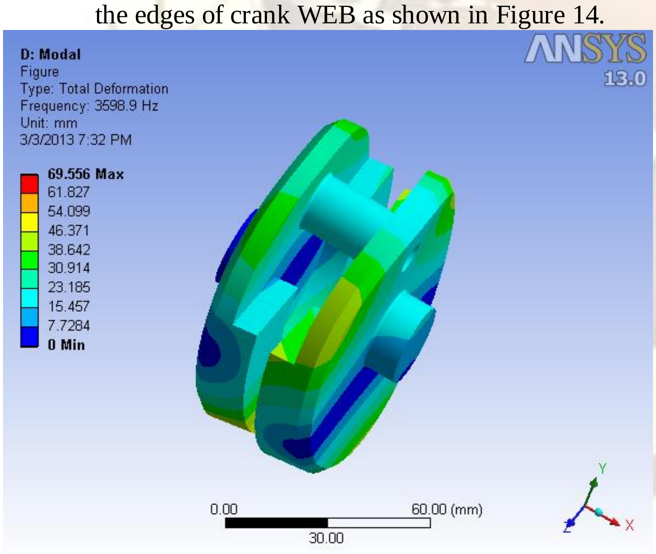

Figure 14 Fourth mode of vibration

Fifth Mode of Vibration

The fifth mode of vibration is torsion vibration about X-direction at natural frequency of 3918.9 Hz. The maximum deformation appears at the edges of crank WEB as shown in Figure 15.

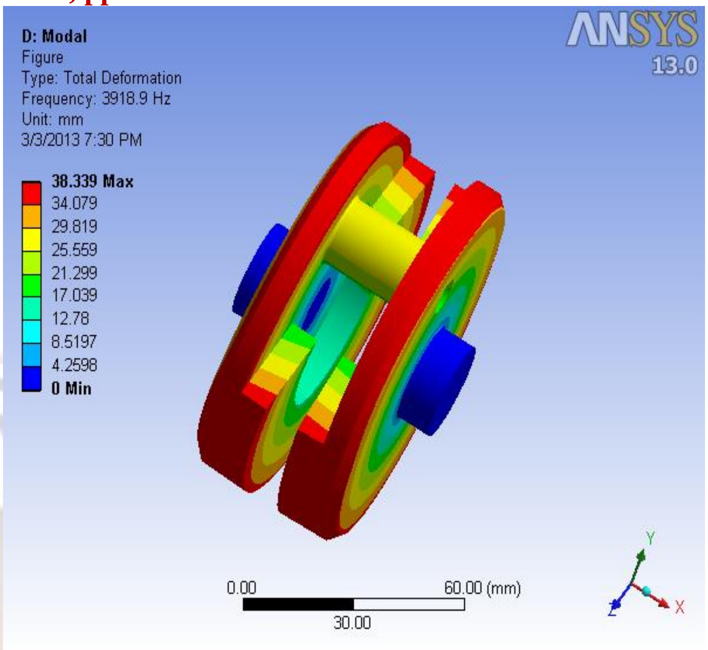

Figure 15 Fifth mode of vibration

Sixth Mode of Vibration

The sixth mode of vibration is bending vibration in Z-direction at natural frequency of 3598.9 Hz. The maximum deformation appears at the edges of crank WEB as shown in Figure 16.

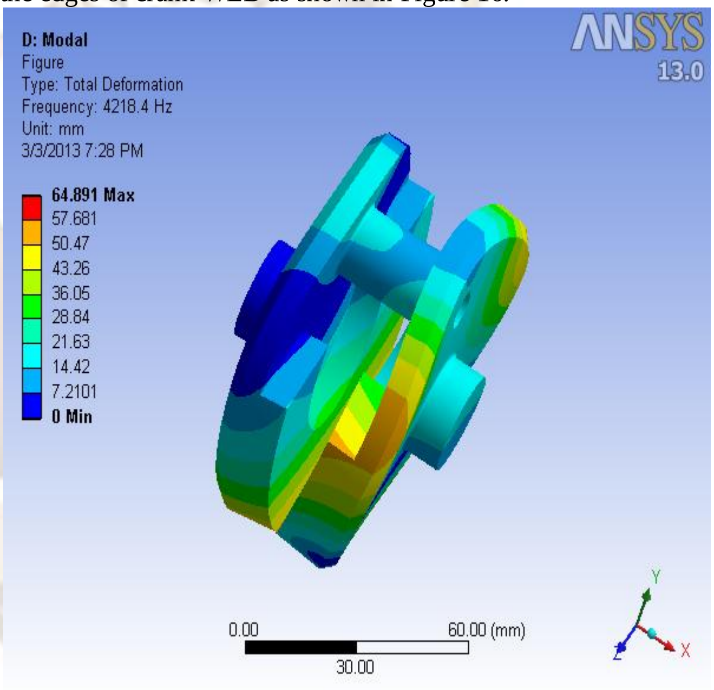

Figure 16 Sixth mode of vibration

VII. CONCLUSION

1. Strength Analysis is a powerful tool to check adequacy of crankshaft dimensions and find scope for design modification.
2. It is found that weakest areas in crankshaft are crankpin fillet and journal fillet. Hence these areas

must be evaluated for safety.

3. This project includes torsion stresses in analysis. But it is found that the torsion stresses are negligible as compared to bending stresses. Hence they can be ignored while doing strength analysis of crankshaft.

4. The crankshaft was found to be over dimensioned. Therefore web thickness was reduced from 13 mm to 10 mm. The reduction in mass obtained by this design modification is:
Mass Reduction = 1.9725 kg - 1.6375 kg = 0.335 kg
Percentage Mass Reduction = 16.98 %

5. The lowest value of natural frequency is 1432.7 Hz while the highest value is 4218.4 Hz (Table 5). When the engine is running at its maximum speed of 6000 rpm, the driving frequency is merely 100 Hz. As the lowest natural frequency is far higher than driving frequency, possibility of resonance is rare.

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