Estimation of Vibration Response of Reciprocating Engine Crankshaft

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Abstract - *Most of the people hear the word "crankshaft," they imagine it as a main part of the engine, which also consists of parts like slider crank, connecting rod, and piston. Crank shaft is an important part of engine used to transmit reciprocating motion into rotary motion. Crank shaft is designed according to number of cylinders used for fuel combustion. Strokes of cylinders are so adjusted that forces acting is simultaneous on crankshaft. According to studies most engine failures are occurred due to failure of crankshaft. Failures of crankshaft are due to many reasons such as fatigue failure, failures due to vibration, etc. All machines in the world have some sort of vibrations. But most important factor for failure of any machinery is because of vibration. Vibration generated in crankshaft may be responsible for failure of engine. If vibrations generated are more than allowable vibration, engine tends to fail. Vibration monitoring of crankshaft is very difficult. There are many methods to find vibration in crankshaft such as analytical, mathematical, graphical, by use of software i.e. by ANSYS, CATIA, and NASTRAN etc. This software's are used to crosscheck and to find critical observations which are difficult to find by analytical methods in mechanical parts.*

This research is genuine attempt to find out effects of vibration produced inside the engine on crankshaft and its effect on performance of engine. The process used to find out vibration response of reciprocating engine crankshaft is by making graphical investigation of various forces produced in the engine parts and its effect on crankshaft. The stresses generated on crankshaft are calculated using ANSYS which gives detailed information about vibration response of crankshaft

***Key Words*: Crankshaft, Engine, Vibration response, MatLab,**

1. INTRODUCTION

Crankshaft dynamics became the interest to the automobile industry with the rise in importance of noise, vibration & harshness (NVH). Vibration and noises generated by crank transmission during the high speed application is the main problem of dynamic research. The major difficulties in the study of the crankshaft are related with the way that the crankshaft wraps around the bearing, forming a continuous rotation. This effect is called the reciprocating into rotary, together with the impact between crank and crankshafts participates in the creation of the noise and vibrations on the engine and to determine the crankshaft load capacity and its service life.

Crankshaft is extensively used in automobile industries. It is seen that it is used right from small capacity machine to heavy capacity machine. During machine operation the crankshaft receives variable load and thus induces variable tension in it. The linkage member of a crank experiences this varying tension during complete work cycle. Evidences says that due to this varying load condition the crank itself vibrates which provides impact loads on its bearing and thus it enhances the wear and tear of a crank. Hence it is inevitable to see the effect of varying load condition of a specific unit on its performance by estimating its vibration.

2. FORMULATION OF PRESENT WORK

With reference to the literature review and past work done by various researchers, we find that crankshaft plays a very crucial role in energy as well as motion transfer. So, it is required to study its dynamic behavior in working condition. Owing to the complexity and construction of its geometry the analysis of crankshaft is very difficult to carry out by analytical means and therefore we find software based approaches in all the researches to analyze the dynamic behavior of the crankshaft. There are many software like CATIA, HYPERMESH, ANSYS etc. used for analyzing the component against development of stresses, strains etc. and clearly gives an idea about the fatigue life cycle of the component being tested. But still software based approaches finds limitation in giving an exact statement of values of the parameters against which the component is going to work and hence, it is difficult to ascertain the life span of the component or mention the particular cause of its failure.

In fact results obtained from the software relay on the computation done with the help of programs which is fed manually by a programmer. The software only shows results stored in the memory and it has no relation with practical situation.

So, we can say that analytical means is equally important as software based approaches. But the result obtained by analytical means is limited by the number of variables taken into account and in turn to get the exact solution of the problem.

A general procedure for the analysis of the component is given below:

1. Obtaining detailed dimensions of the component to be tested.
2. Creating a model using any of modeling software like Creo.2 parametric, CATIA, and Solid Edge etc.
3. Saving the file of the model created so that it can be imported to analysis software, ANSYS, Unigraphics etc.
4. Opening the file in Analysis software [ANSYS] for performing static analysis [using dynamic forces as input] and model analysis [vibration].
5. Applying meshing condition. This is an important step because difference between exact solution and approximate solution depend upon the type of meshing we choose.
6. Obtaining the value of stress, strain, natural frequency, mode shapes etc.

3. AN APPROACH FOR VIBRATION ANALYSIS OF MODEL

As there different techniques available for performing vibration analysis of a model however, the stepwise procedure of the methodology used in the present research is discussed below:

1. The physical system [engine] is converted into a simplified dynamic model. The dynamic model is a slider crank chain where center of gravity of each component of the engine is assumed to be concentrated at the geometric center.
2. Kinematic analysis is performed graphically for the model under which velocity and accelerations of center of gravity of each links have been estimated.
3. After kinematic analysis of the model, it is considered for dynamic analysis according to law proposed by DE 'Alembert. According to which a dynamic case can be considered as a static case when inertia forces due to acceleration associated with the link is considered on the link.
4. A load Torque on crankshaft of the engine is calculated for mentioned crank rotation.

Finally the calculated load torque is used for estimation of torsional amplitudes of shaft under torsional vibration.

4. Dynamic Analysis of Slider Crank Mechanism

The dynamic behavior of crankshaft of a reciprocating engine has been explained. For the analysis purpose the reciprocating engine is modeled as a simplified slider crank arrangement where cylinder, crank, connecting rod and piston of the reciprocating engine is represented as link 1, link 2, link 3 and link 4 in slider crank arrangement respectively [refer figure 1.1] and their center of gravity is assumed to be concentrated at the center of respective links.

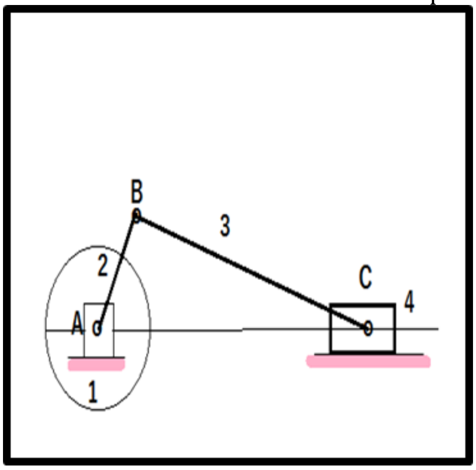

Fig -1: Schematic diagram of slider crank

A. Estimation of forces on links

The analysis is initiated by considering position shown in figure 1.1 as the reference position where angular position of link 2 is 360 with respect to link 1 at this instant. With reference to this position, the crank is rotated in anticlockwise direction with an angular displacement of 360 and therefore dynamic analysis is carried out at 360, 1080, 2520, 3240, 3960, 4680, 6120, and 6840.

Since all the links of the mechanism are moving with some velocity and acceleration, the inertial forces associated with the links cannot be ignored. So, it is required to perform a kinematic analysis of the links at every angular position mentioned above. The table4.1 and table 4.2 shows all the calculated values of velocity and acceleration and a graph has been plotted for the same [shown in figure.1.2 & 2.3]

Table 1: Calculation for velocity

Θ	V2	V3	V4
36	94.24	81.66	84.84
108	94.24	31.41	78.52
252	94.24	34.55	75.38
324	94.24	81.84	84.80
396	94.24	81.66	84.84
468	94.24	31.41	78.52
612	94.24	34.55	75.38
684	94.24	81.84	84.80

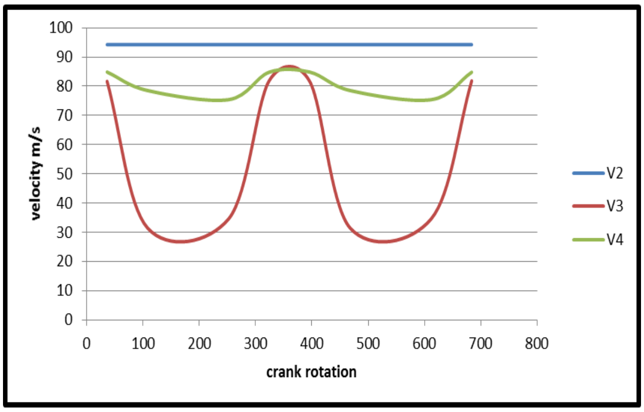

Fig -2: Graph of Crank rotation Vs Velocity

Table 2: Calculations for Acceleration

θ	ag2	ag3	ag4
36	98679.75	193412.3	205253.88
108	98679.75	142098.84	142098.84
252	98679.75	138151.65	126310.08
324	98679.75	193412.3	205253.88
396	98679.75	193412.3	205253.88
468	98679.75	142098.84	142098.84
612	98679.75	138151.65	126310.08
684	98679.75	193412.3	205253.88

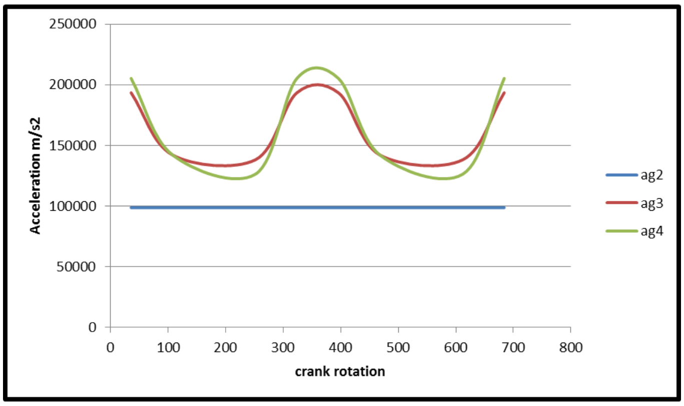

Fig -3: Graph of Crank rotation Vs Acceleration

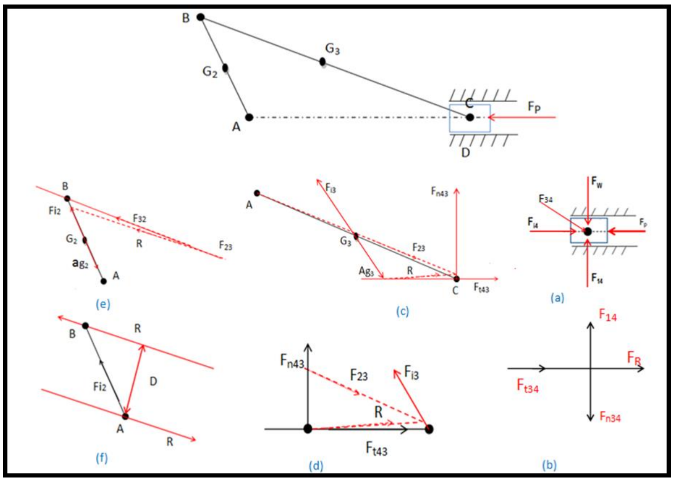

Fig -4: Schematic diagram of slider crank at 108⁰

Figure 4.2 shows a dynamic analysis of slider crank mechanism at θ=108⁰. In this figure we have focused on the forces acting on the connecting rod, crank and slider. These forces are denoted by F_P, F14, Fi4, FW, F34, Fi3, F23, Fi2, where, F_P is a pressure force acting on slider or piston due to power stroke at point C, $F_{14}$is a force due to normal reaction from cylindrical wall, Fi4 is the inertia force due to movement of slider, F_W is a force due to weight of the slider, and F_{34} is a force due to link 3 on the link 4 i.e. slider. But F_{34} force is unknown and the calculation for this force is done by resolving the force F_{34} in two components i.e. $F_{34(t)}$ and $F_{34(n)}$,where $F_{34(t)}$ represents tangential components acting along the surface of the cylinder and $F_{34(n)}$ represents normal components acting perpendicular to the surface of the cylinder.

Similarly forces acting on connecting rod and crank are $F_{43(n)}$,$F_{43(t)}$, F_{i3},F_{23},and F_{32},F_{i2}respectively shown in figure 1.4 (c)and(e) where $F_{43(n)}$ represents normal component acting at point Con link 3, $F_{43(t)}$ represents tangential components of link 4 on link 3 of the surface at point C, F_{i3}represents an inertia force acting through center of gravity of the link 3, F_{23} is a force due to link 2 on the link 3 i.e. crank to connecting rod, F_{32} is a force due to link 3 on link 2 and F_{i2} represents an inertia force acting through center of gravity of the link 2.

Forces F_P, FR, F_{i2}, Fi3, F_{i4} and F_R can be calculated by using the following formulas:

F_P= P x Area, (Area $=\frac{\pi}{4} \times D^2$)

Where, P =Pressure
D =Diameter of cylinder

$F_R=F_P-F_{i4}$

$F_{i2} = m_2 \times a_{g2}$

$F_{i3} = m_3 \times a_{g3}$

$F_{i4} = m_4 \times a_{g4}$

Where, m_2, m_3, m_4 are the mass of crank, connecting rod, and slider.

a_{g2}, a_{g3}, a_{g4} are the acceleration due to gravity.

Further R is resultant force of forces $F_{32}$and F_{i2} which passes through point B [refer figure 1.4 (f)].

The calculated values of all the forces is given in the table 4.3

Table 3 Calculation of forces

[1] Link	[2] Forces (N)
[3] 2	[4] F_{i2} = 12788.87 [5] F_{32}=310000 [6] R=300700
[7] 3	[8] F_{i3} = 36832.01 [9] F_{t43} =300000 [10] F_{n43}=170000 [11] F_{23}=310000
[12] 4	[13] F_{i4}= 308354.48 [14] F_P=1759

Hence the magnitude of the load torque for each position is calculated and given in [refer table 4.3]

T=R × D

Where, D is perpendicular distance of R from point A.

Table 4 Calculation of Torque

Crank rotaton (θ)	Load Torque (KNm)
36	167.4
108	9.62
252	9.03
324	167.4
396	167.4
468	9.62
612	9.03
684	167.4

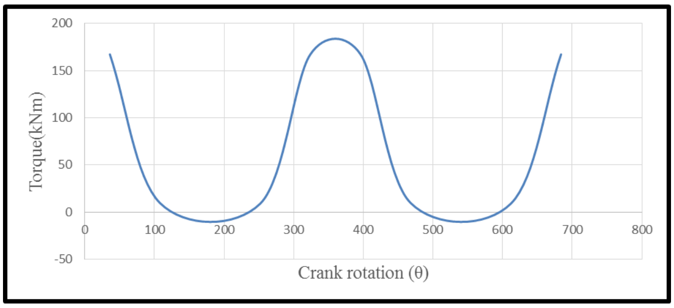

Fig-5 Graph of load torque Vs Crank rotation

5. VIBRATION ANALYSIS OF CRANKSHAFT

The dynamic analysis of a crankshaft has been carried out, where a single cylinder engine is model as a simplified slider crank arrangement which forms the basis for vibration analysis.

Considering slider crank arrangement all the forces propagating through the links has been estimated and ultimately a load torque acting at crankshaft is calculated. This analysis is carried out at various degrees of crank rotation taken at an interval of 36^0. The estimated load torque is shown in figure below:

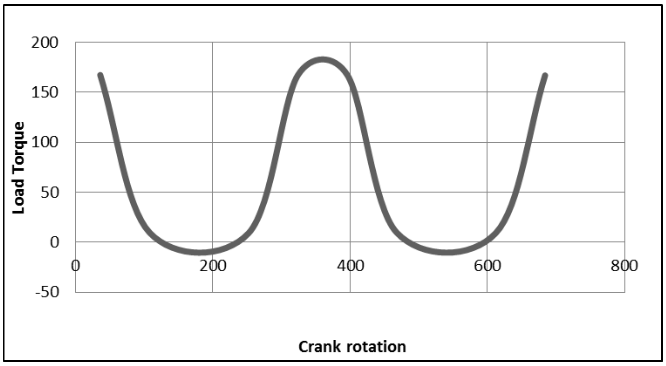

Fig- 6 Graph of load torque vs. crank rotation

In the present research, the crankshaft components of the engine are chosen for analysis purpose. We can see that the crankshaft is responsible for converting reciprocating motions of slider (piston) into rotary motion of the gear box. This component bears a load from the engine (developed inside cylinders) and load from gear box (due to weight of the system). So crankshaft can be assumed to execute torsional vibrations. Now torsional vibrations concerning with crankshaft is discussed in detail in forthcoming article.

6. TORSIONAL VIBRATION OF CRANKSHAFT

The crankshaft can be modeled as a single rotor shaft system for analysis purpose. The piston-connecting rod-crank assembly has been replaced by a rotor of equivalent inertia calculated in the following manner:

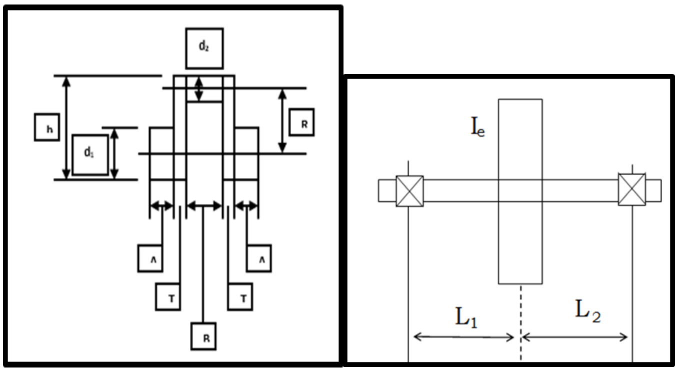

Fig-7 Single cylinder crankshaft

Ie= [(W1/g) + (W2+WP)/2g]*R2 + ½[(W1/g)*R2] + [W1/g] x 2] +2[(W*h*T*ρ)/g)]*{[(h2+W2)/12] + R2}

Where,
R is Crank radius,
W is Width of the crank web,
h is Height of crank web,
T is Thickness of crank web,
W1 is Weight of connecting rod at wrist end,
W2 is Weight of connecting Rod at piston end,
Wp is Weight of piston assembly,
W1 is Weight of crank pin,
ρ is Specific gravity of steel,
Ie is Mass moment of Inertia of piston & connecting rod referred to crank cg

The calculated value equivalent inertia I_o is 753598081.2 kg-mm^2. Now the equivalent inertia, Ie represents a single rotor mounted on the shaft shown in Fig-7.

Equation of motion for torsional vibrations

Considering a rotor of inertia I.e. mounted on the shaft of torsional stiffness Kt [refer figure 5.1 b], the equation of motion for torsional vibration of shaft can be written as:

$$I_0\ddot{\theta} + k_t\theta = T(t) = 167.4\cos\omega t \qquad \text{Eq. 1}$$

$$I_0\frac{d^2\theta}{dt^2} + k_t\theta(t) = 167.4\cos\omega t \qquad \text{Eq.2}$$

$$\frac{d^2\theta}{dt^2} + \frac{k_t}{I_0}\theta(t) = \frac{167.4 \cos\omega t}{I_0}$$ Eq.3

$$(D^2 + \frac{k_t}{I_0})\theta = \frac{167.4 \cos\omega t}{I_0}$$ Eq.4

$$\therefore (m^2 + \frac{k_t}{I_0}) = 0$$ Eq.5

$$m = \pm\sqrt{\frac{k_t}{I_0}}$$ Eq.6

But,

$$k_t = \frac{\pi}{32} \times d^4 \times (\frac{G}{l_e})$$

$I_0 = 753598081.2$ kg-mm2

$$k_t = \frac{\pi}{32} \times 50^4 \times (\frac{79 \times 10^3}{50})$$ Eq.7

$$k_t = 969475857.9$$

$$\frac{k_t}{I_0} = 1.28$$

$$m = \pm 1.28$$ Eq.8

$$C.F. = A \cos\left(\frac{k_t}{I_0}t\right) + B \sin(\frac{k_t}{I_0}t)$$ Eq.9

$$C.F. = A \cos(1.28t) + B \sin(1.28t)$$

$$P.I. = \frac{1}{D^2 + \frac{k_t}{I_0}^2} \times \frac{167.4 \cos\omega t}{I_0}$$ Eq.10

Replacing, D^2 by $-\omega^2$

$$P.I = \frac{167.4}{I_0} \times \frac{1}{(-\omega^2 + \frac{k_t}{I_0}^2)} \cos\omega t$$

$$P.I = \frac{167.4}{I_0} \cos\omega t \frac{1}{(\frac{k_t}{I_0}^2 - \omega^2)}$$ Eq.11

$$\theta(t) = C.F. + P.I$$

$$\theta(t) = A \cos\left(\frac{k_t}{I_0}t\right) + B \sin(\frac{k_t}{I_0}t) + P.I$$ Eq.12

$$\theta(t) = A\cos(1.28t) + B\sin(1.28t) + 2.22 \times 10^{-7} \cos(2094.36)t \times \frac{1}{(1.28^2 - 2094.36^2)}$$

$$\theta(t) = A\cos(1.28t) + B\sin(1.28t) - 5.06 \times 10^{-14} \cos(2094.36)t$$

Applying Boundary condition,

At t = 0, θ(t)=0

$$\theta(t) = A - 5.06 \times 10^{-14} = 0$$

$$A = 5.06 \times 10^{-14}$$ Eq.13

At t = 0.03,θ(t)=0.628 rad

$$\theta(t) = A\cos(1.28t) + B\sin(1.28t) - 5.06 \times 10^{-14} \cos(2094.36)t$$

$$0.628 = 5.06 \times 10^{-14} \cos(1.28 \times 0.03) + B\sin(1.28 \times 0.03) - 5.06 \times 10^{-14} \cos[(2094.36]) 0.03$$

$$B = 16.35$$ Eq.14

The solution can be written as

$$\theta(t) = 5.06 \times 10^{-14} \cos(1.28t) + 16.35 \sin(1.28t) - 5.06 \times 10^{-14} \cos(2094.36)t$$ Eq.15

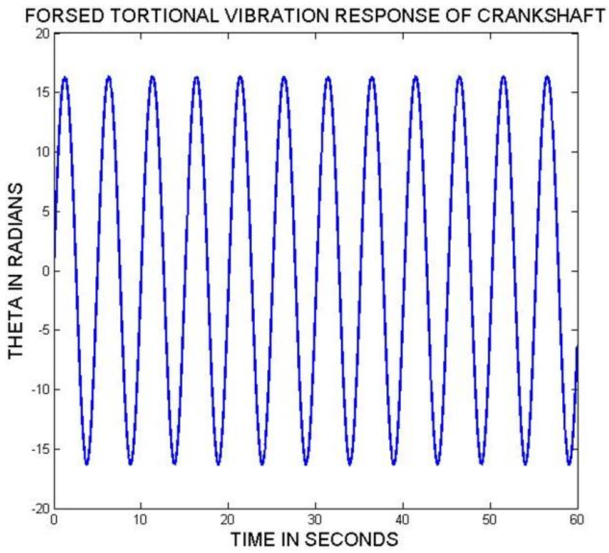

Fig-8 Forced Torsional vibration Response of Crankshaft

7. RESULT AND DISCUSSION

The analysis of slider crank mechanism of an engine is presented through as a paper to find the results of analysis are reproduced here for the sake of explanation. There are in fact two types of graph have been obtained from analysis

1) The load torque variation against crank rotation [fig-5] and

2) The variation of inertia force Vs Crank rotation.

Discussions on load variation Vs crank rotation

If one carefully observes the graph then he may find that the graph is divided into different phase i.e. the graph changes from the positive loop to negative loop and it follows a function of cosine series. Here it needs to be observed that the torque changes its sign from positive to negative.

Discussion on variation of Inertia forces Vs Crank rotation

Table 5 Values of Inertia Forces

θ	Fi2	Fi3	Fi4
36	12788.87	50132.47	4454000.9
108	12788.87	36832.01	308354.48
252	12788.87	35808.90	274092.87
324	12788.87	50132.47	4454000.9
396	12788.87	50132.47	4454000.9
468	12788.87	36832.01	308354.48
612	12788.87	35808.90	274092.87
684	12788.87	50132.47	4454000.9

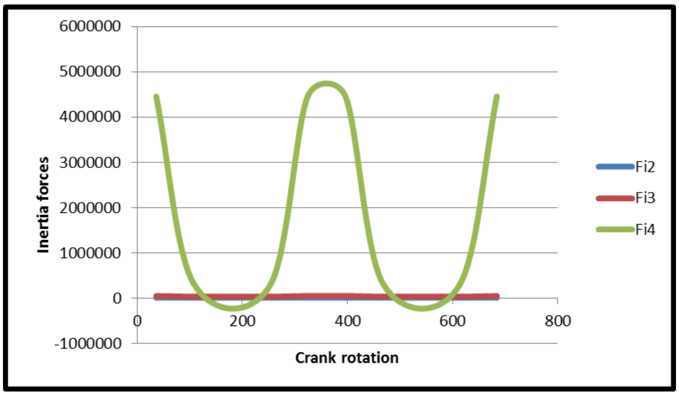

Fig-9 Graph of Load torque Vs Inertia forces

It is observed from the graph plotted above, inertia forces of link 3 and link 4 does not show as much variation as observed in Inertia forces of link4. The variation of inertia force of slider can be shown by a cosine function which is identical to the variation shown by load torque with respect to crank rotation.

8. CONCLUSION

1. It can be concluded that out of four types of vibrations like torsional, flexural, axial, coupled which affect the crankshaft, torsional vibrations are the most dangerous which can break the crankshaft,
2. There are two different load sources in an engine; inertia and combustion, these two load source cause both bending and torsional load of the crankshaft.
3. It can be concluded that modal analysis gives the relation of frequency and the vibration characteristics of the crankshaft. The results of modal analysis are extremely important as the resonance frequency or the frequency at which the effects of vibration are maximum is provided. The modal analysis also provides the starting point for harmonic and transient dynamic analysis where, the details of these mode shapes with their frequencies are useful.
4. The maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. So this area prunes to appear the bending fatigue crack. We can forecast the possibility of mutual interference between the crankshaft and other parts. The resonance vibration of system can be avoided effectively by appropriate structure design the results provide a theoretical basis to optimize the design and fatigue life calculation.
5. Critical locations on the crankshaft geometry are all located on the fillet areas because of high stress gradients in these locations which result in high stress concentration factors.

REFERENCES

[1] BasavarajTalikoti, Dr.S.N.Kurbet, Dr.V.V.Kuppast, AevindM.Yadwad; “Modal Analysis of a 2-cylinder crankshaft using ANSYS”, International Journal of Engineering Research and application December 2015.

[2] Basavaraj S. Talikoti, Dr. S. N. Kurbet, Dr. V. V. Kuppast; *“A review on vibration analysis of crankshaft of internal combustion engine”,* International Research Journal of Engineering and Technology (IRJET), Volume: 02 Issue: 09 | Dec-2015

[3] C.M.Balamurugan,R.Krishnaraj,Dr.M.Sakthive.l,K.Kanthave.l,Deepan Marudachalam M.G.,R.Palani.,2011;“Computer Aided Modeling and Optimization of Crankshaft” International Journal of Scientific & Engineering Research Volume 2, Issue 8 ISSN 2229-5518

[4] Farzin H. Montazersadgh and Ali Fatem2007;“Dynamic Load and Stress Analysis of a Crankshaft,” SAE Technical Paper No. 010258, Society of Automotive Engineers.

[5] H. D. Desai; “Computer Aided Kinematic and Dynamic Analysis of a Horizontal Slider Crank Mechanism Used For Single-Cylinder Four Stroke Internal Combustion Engine”, Proceedings of the World Congress on Engineering 2009 Vol II WCE 2009, July 1 - 3, 2009, London, U.K.

[6] JianMeng, Yongqiliu;*“Finite Element analysis of diesel engine using finite element method”,* Published Online August 2011 in MECS.

[7] JianMeng., Yongqi Liu., Ruixiang Liu.,2011;“Finite Element Analysis of 4-Cylinder Diesel Crankshaft”, I.J. Image, Graphics and Signal Processing, 5, 22-29

[8] JaiminBrahmbhatt, Prof. Abhishekchoubey ;“Design and Analysis of Crankshaft for Single Cylinder 4-Stroke Diesel”, International Journal of Advanced Engineering Research and Studies E-ISSN2249–8974.

[9] K. Thriveni, Dr. B. Jaya Chandraiah, "Modal Analysis of A Single Cylinder 4-StrokeEngine Crankshaft",International Journal of Scientific and Research Publications, Vol.3, Issue 12, December 2013.

[10] Prof. A.M.Patil and Prof. R..G.Desavale;“Theoretical and Experiment Analysis of Torsional and bending effect on four Cylinders Engine Crankshaft by Using Finite element approach” International Journal of Engineering Research Volume no.2 Issue No.6,pp:379-385, Oct.2013

[11] R.B.Sharma&Vikas Sharma; “Dynamic & Static Analyses of Model of Engine Crankshaft”, International Journal of Engineering research and application ,June2013

[12] R.L.Norton;“Kinematics and Dynamics of Machinery”,First Edition in SI Unit,Tata McGraw Hill Education Private Limited ,New Delhi.

[13] R.J Deshbhratar, Y.R Suple;*“Analysis andoptimization of Crankshaft using FEM”,* International Journal of ModernEngineeringResearch, vol-2, issue-5,ISSN:2249-6645, pages:3086-3088, Sept-Oct 2012

[14] S. Bhagya Lakshmi, Sudheer Kumar V, Ch. Nagaraju,“Dynamic Analysis of Honda Engine Crank Shaft”,International Journal of Engineering and Innovative Technology (IJEIT) Volume 2, Issue 1, July 2012

[15] S.S.Rattan;" Theory Of Machines", Third Edition, , Tata McGraw Hill Education Private Limited ,New Delhi

[16] SingiresuS.Rao; "Mechanical Vibration", Fifth Edition, Copyright 2011,2004 Pearson Education, Inc., publishing as Prentice Hall.

[17] Sanjay B Chikalthankar, V M Nandedkar,Surender Kumar Kaundal, "Finite elementanalysis approach for stress analysis of crankshaft under dynamic loading",International Journal Of Scientific &Engineering Research, Vol. 4, Issue 2,February-2013.

[18] VijaykumarKhasnis, ManojUkhande, GirishTilekar, Rajesh Mane, GirishShegavi; "Crankshaft design optimization to improve dynamic balancing and fatigue strength",International Journal ofAutomotive Engineering, Society ofAutomotive Engineers of Japan, 2015, PP.59-66.

[19] Yu Ding and Xiaobo Li.,2011, " Crankshaft Strength Analysis of a Diesel Engine Using Finite Element Method"; Asia-Pacific Power and Energy Engineering Conference

[20] Zissimos P. Mourelatos, "A crankshaft system model for structural dynamic Analysis of Internal Combustion Engines", Vehicle Analysis and Dynamics Lab ,Elsevier Science Ltd., June 2001

[21] KewalPujara*,"Vibration and noise control",Dhanpatrai*& sons, 1992