Analysis & Optimization of Torsional Vibrations in a Four-Stroke Single Cylinder Diesel Engine Crankshaft

Amitpal Singh Punewale, Mr. Tushar Khobragade, Mr. Amit Chaudhari, Prof. N. M. Khandre

***Abstract*— Every material system containing individual mass and stiffness distribution is susceptible to vibrate. These vibrations can be caused either by single impulse load or a periodical load. Analysis of free vibration is essential to determine natural frequencies of material system, responsible for resonance phenomenon, occurred due to equalizing periodical load frequency with natural frequencies and leads to high amplitude of vibration & high fatigue stress into material.**

Torsional vibrations of engines has received considerable attention from researchers and engineers from many years, but influence of each drive train component on engine's torsional vibration has received less attention. In this paper, the influence of individual crankshaft geometrical parameters on torsional vibrations are studied.

Crankshaft torsional vibration is one of the most important factors affecting engine operation and in crankshaft design of IC engines. The crankshaft is subjected to complex loading due to motion of connecting rod, the mass of piston assembly & connecting rod increases resultant moment of inertia, that decrease natural frequency of first and second mode within range of engine rotational speed, resulting in resonance with low order harmonic of tangential force acting on crankpin, which is very undesirable.

Torsional vibrations detoriate the NVH performance of engine and reduces fatigue life of crankshaft. Hence, there is necessity to reduce torsional vibrations. In this study, by employing finite element analysis (FEA) on crankshaft, different methods will be proposed to reduce torsional vibrations and increase fatigue life.

***Index Terms*— Crankshaft, Fatigue, FEA, Torsional Rigidity, Torsional Vibration.**

I. INTRODUCTION

Crankshaft is the main component of internal combustion engine, converting reciprocating motion of piston into rotational motion. Crankshaft vibration is one of the most important factors affecting engine operation. The different vibrations affecting the crankshaft are lateral vibrations and torsional vibrations.

The problem of torsional vibrations are very severe, as they are very difficult to detect and the first symptom of problem is often a broken shaft, gear tooth, or coupling. The difficulty of detecting incipient failures in the field makes thorough torsional vibration analysis as an essential component of the engine design process.

According to *Montazersadgh*. F. H. and Fatemi. A. [1], the crankshaft torsional vibration occurs as each power stroke tends to slightly twist the shaft. When the power stroke subsides, the crankshaft untwists. One would think that something like crankshaft would not twist significantly, but any piece of metal always deflects a bit when force is applied, and in case of IC engines, where large amount of power is generated, these forces can become huge indeed. Torsional vibrations of engines arise due to application of periodic combustion forces in the cylinder and associated inertial forces.

Boysal. A. and Rahnejat. H. [2] shown that crankshaft experiences large number of load cycles resulting from gas combustion (F_g) and inertia forces (F_i), during its service life. These forces acting on the crankshaft causes two types of fluctuating loads on the crankshaft structure i.e. torsional load (F_t) and bending load (F_r) as shown in fig. 1.

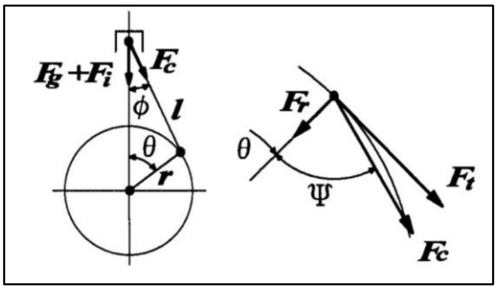

Fig. 1. Forces acting on Crankshaft

According to Feese. T. and Hill. C. [3] traditionally torsional vibrations have been studied only from the standpoint of reducing noise and improving comfort. However, one must also bear in mind that this kind of vibration, and resonance in particular, can generate loads that far exceed the maximum engine torque.

Torsional vibration is a concern in the crankshafts of internal combustion engines because it could break the crankshaft itself, shear-off flywheel or cause driven belts, gears and attached components to fail, especially when frequency of vibration matches the torsional resonant frequency of the crankshaft.

Charles. P., Sinhala. J. K., Gu. F., Lidstone. L., and Ball. A. D. [4] states that engines always have some degree of torsional vibration during operation due to their reciprocating nature. The measurements of torsional vibrations is a difficult task. The encoder signal has been used to construct IAS (Instantaneous Angular Speed) waveform, representing signature of torsional vibration. IAS signal and its Fast Fourier Transform (FFT) analysis are effective for monitoring engines with less than eight cylinders.

According to Wilson. W. K. [5] A favorable solution of torsional vibration problem is provided by inherent mass,

Manuscript received April 07, 2015.
Amitpal Singh Punewale, Second Year (M.Tech CAD/CAM), Department of Production Engineering, SGGSIE&T, Nanded, India,
Mr. Tushar Khobragade, Deputy General Manager, Greaves Cotton Limited, Aurangabad, India.
Mr. Amit Chaudhari, Manager (FEA), Greaves Cotton Limited, Aurangabad, India.
Prof. N. M. Khandre, Associate Professor, Department of Production Engineering, SGGSIE&T, Nanded, India..

elastic, exciting & damping characteristics of the system, either as originally designed or after simple and inexpensive adjustments of one or more characteristics. In some cases, the modifications may found impracticable or costly or involve sacrifice of performance or other desirable design requirements. In such cases, special devices for reducing torsional vibration are used. This special devices are usually termed as 'tuning or damping devices'.

The primary function of these tuning devices is to assist in producing a favorable adjustment of natural frequency. At the same time, they can be arranged to provide a significant contribution to overall damping of the system. A heavy flywheel situated at the driving end of an engine crankshaft can also be classified as a tuning member, and the natural frequency of the system can be adjusted by changing the flywheel inertia. Therefore, the different methods to reduce torsional vibrations of crankshafts are:

a) Use of special Devices, known as 'Torsional Vibration Damper' (TVD), to reduce vibration amplitude in resonant points.
b) By increasing torsional rigidity of crankshaft, so that resonance points are shifted outside the IC engine operating range.

Among the two methods, this study focus towards the aim of reducing torsional vibration of crankshaft, by improving the torsional rigidity or torsional stiffness of the engine crankshaft.

II. TORSIONAL ANALYSIS

The torsional analysis is used for determining torsional response of system. Ronald L. Eshleman [6] states that the torsional analysis is the analysis of torsional dynamic behavior of a rotating shaft system as a result of forced vibration. A comprehensive torsional analysis has been carried out for complex components, in which various modes of torsional vibrations are analyzed. The complicated combinations of various effects in steady state and transient operating cases are also considered in torsional analysis.

A torsional system can be modeled as mass-elastic system (inertia & stiffness) to predict stresses in each component. The mass elastic properties of such system can be changed by adding flywheel (additional inertia), using as soft coupling (change in stiffness), or by viscous damping (absorbing natural frequency).

As with other forms of vibration, torsional vibration in machines occurs in characteristic patterns called *'Modes'*. The mode shape is dependent on torsional stiffness, polar moment of inertia, and damping of various components in the system. The frequency of vibration at which the mode shape occurs is called *"Natural Frequency"*. When a natural frequency coincides with an excitation frequency, a state of resonance occurs. At resonance, the system damping alone controls the motions and stresses in the system.

The crankshaft of a reciprocating engine or the rotors of a turbine or motor, and all moving parts driven by them, comprise a torsional elastic system. Such a system has several modes of free torsional oscillation. Each mode is characterized by a natural frequency and by a pattern of relative amplitudes of parts of the system when it is oscillating at its natural frequency. The harmonic components of the driving torque excite vibration of the system in its modes.

If the frequency of any harmonic component of the torque is equal to or close to the frequency of any mode of vibration, a condition of resonance occurs and the machine is said to be running at a critical speed. The operation of the system at such critical speeds can be very dangerous, resulting in fracture of the shaft. The number of complete oscillations of the elastic system per unit revolution of the shaft is called an *"Order"* of the operating speed.

III. TORSIONAL STIFFNESS

The first step in analytically determining torsional response is to calculate torsional natural frequencies of the system. As per Wilson. W. K. [7] For simpler geometry, such as, cylindrical shafts, the mass-moment of inertia and torsional stiffness can be calculated using simple formulae.

For more complicated geometries such as crankshafts, the crankshaft can be simplified into several main components, such as, stub shaft, journals, webs, and crankpins. Thereby, a mass-elastic model of the crankshaft is created by lumping inertia at each throw & calculating equivalent torsional stiffness between throws.

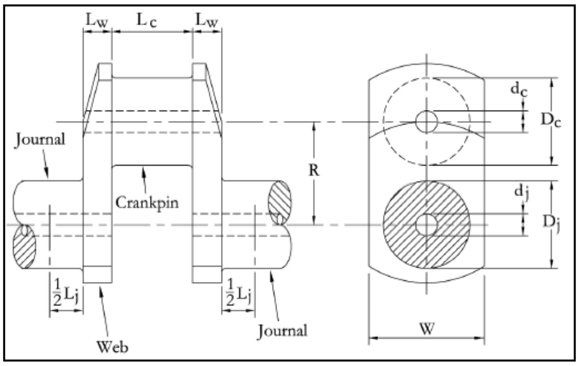

Fig. 2. Simple Crank-Throw

Fig. 2 shows a basic crankshaft throw, consisting of two webs, crankpin & journal. Many equations are available for calculating the torsional stiffness of crankshaft, such as, Ker Wilson, Carter, etc. requiring basic dimensions of the journals, webs, and crankpins as well as shear modulus of the shaft material.

a) CARTER FORMULA:

$$K_t = \frac{G \cdot \pi}{32\left(\dfrac{L_j + 0.8L_w}{D_j^4 - d_j^4} + \dfrac{0.75L_c}{D_c^4 - d_c^4} + \dfrac{1.5R}{L_w \cdot W^3}\right)}$$

b) WILSON FORMULA:

$$K_t = \frac{G \cdot \pi}{32\left(\dfrac{L_j + 0.4D_j}{D_j^4 - d_j^4} + \dfrac{L_c + 0.4D_c}{D_c^4 - d_c^4} + \dfrac{R - 0.2(D_j + D_c)}{L_w \cdot W^3}\right)}$$

Carter's formula is applicable to crankshafts with flexible webs with stiff journals and crankpins as well as for crankshafts with no overlapping of journals & crankpins, while Ker Wilson's formula is better for stiff webs with flexible journals, crankpins and for crankshafts with overlapping of journals & crankpin.

A Finite Element Method (FEM) can also be used to determine torsional stiffness for crankshaft section. Milasinovic. A., Filipovic. I. & Hribernik. A. [8] explained FEA boundary conditions to determine torsional stiffness of single throw crankshaft which simulates engine condition i.e. journal mounted inside bearing.

In their study, an attempt is made to determine torsional stiffness of single throw crankshaft by establishing FEA boundary conditions similar to crankshaft physical fatigue test, i.e. *one end is fixed and torque is applied at other end*. The angular displacement (θ) is calculated and torsional stiffness (K_{t1}) is calculated as,

$$K_{t1} = \left(\frac{T}{\theta}\right)$$

After calculating torsional stiffness, the natural frequency is calculated as,

$$\omega_n = \sqrt{\left(\frac{K_t}{I_p}\right)}$$

Where,

I_p = Mass moment of Inertia, kg-mm^2

ω_n= Natural frequency, rad/s,

$$f_n = \left(\frac{\omega_n}{2\pi}\right)$$

Ukhande. M., Mane. R. and Shegavi. G. [9] carried an extensive analytical study for evaluation of torsional stiffness for various crankshafts by varying different geometric parameters and compared analytical results with FEA results. They found that torsional stiffness is directly proportional to crank pin diameter (D_c), web width (W), web thickness (L_w) & inversely proportional to throw (R) and pin width (L_c).The torsional rigidity evaluated by analytical method and FEA correlates with 85-95 % accuracy. Thus, by changing different geometric parameters in a crankshaft, its torsional rigidity can be changed, which changes natural frequency of crankshaft.

IV. ANALYSIS AND RESULTS

For carrying out Finite Element Analysis (FEA), First, a CAD model of crankshaft geometry is prepared, by PRO-E WILDFIRE 5.0, as shown in fig. 3. The CAD model is developed for crankshaft of *Greaves 435cc Engine*.

During engine working, the force in engine cylinder produces a torque to the crankshaft. The magnitude of this torque can be determined by procedure given by Kimura. J., Kai. R., & Shibata. S. [9]

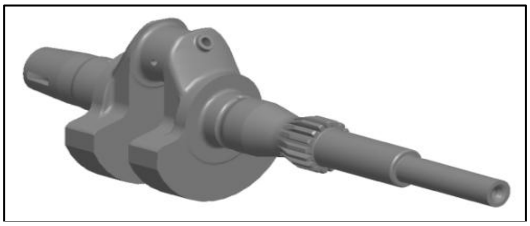

Fig. 3. CAD Model of Crankshaft Geometry

In order to estimate the torsional stiffness, Finite Element Method (FEM) is employed. The analysis is first performed on a simple crank-throw, shown in figure 4.

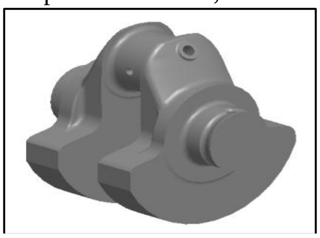

Fig. 4. Crank-Throw of Crankshaft

First, the torsional stiffness of simple crank-throw is determined using FEA. The boundary conditions, shown in figure 5, i.e. *one end is fixed and torque is applied at the other end*. A unit torque, i.e. 1 N-mm, is applied.

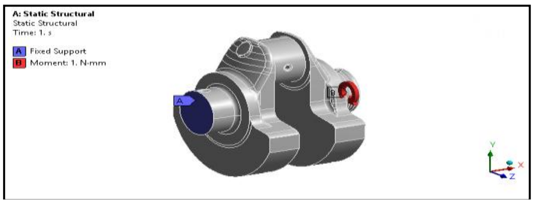

Fig. 5. Boundary Conditions For Crank-Throw

The torsional stiffness is also evaluated using analytical formulas given by Carter & Wilson and compared with FEA results

V. FEA RESULTS:

The value of Angular Displacement *(θ)* is,

$$\theta = \left(2.9948 \cdot 10^{-7}\right) deg$$

Therefore, the value of torsional stiffness (K_t) is given as,

$$K_t = \frac{T}{\theta} = \left(3.339 \cdot 10^6\right) \frac{N \cdot mm}{deg}$$

VI. CARTER FORMULA:

Using carter formula, torsional stiffness (K_{tc}) is given,

$$K_{tc} = \frac{G \cdot \pi}{32\left(\frac{L_j + 0.8L_w}{D_j^4 - d_j^4} + \frac{0.75L_c}{D_c^4 - d_c^4} + \frac{1.5R}{L_w \cdot W^3}\right)} = \left(3.962 \cdot 10^6\right) \frac{N \cdot mm}{deg}$$

Percentage error between FEA result & Carter formula,

$$percentage\_error = \left(\frac{K_{tc} - K_t}{K_t}\right) \cdot 100 = 15.711\%$$

VII. WILSON FORMULA:

Using wilson formula, torsional stiffness (K_w) is given,

$$K_t = \frac{G \cdot \pi}{32\left(\frac{L_j + 0.4D_j}{D_j^4 - d_j^4} + \frac{L_c + 0.4D_c}{D_c^4 - d_c^4} + \frac{R - 0.2(D_j + D_c)}{L_w \cdot W^3}\right)} = \left(3.285 \cdot 10^6\right) \frac{N \cdot mm}{deg}$$

Percentage error between FEA result & Wilson formula,

$$percentage\_error = \left(\frac{K_t - K_w}{K_t}\right) \cdot 100 = 1.617\%$$

As there is an overlapping of crankpin & journal in the crank-throw, the Wilson formula should be employed for accurate results. Also, the results of FEA matches closely with Wilson's formula results.

The torsional natural frequency of crank-throw, shown in fig. 4, determined by analytical method is as follows:

$$I_p = \left(4.609 \cdot 10^{-3}\right) kg \cdot m^2$$

$$K_t = \left(3.285 \cdot 10^6\right) \frac{N \cdot mm}{deg} = \left(1.882 \cdot 10^5\right) \frac{N \cdot m}{rad}$$

The fundamental natural frequency of torsional vibration is,

$$\omega_n = \sqrt{\left(\frac{K_t}{I_p}\right)} = (6.39) \frac{rad}{s}$$

Thus, the natural frequency of torsional vibration is given as,

$$f_{n1} = \left(\frac{\omega_n}{2\pi}\right) = (1017.077) Hz$$

The torsional natural frequency of same crank-throw determined by FEA is shown in fig. 6.

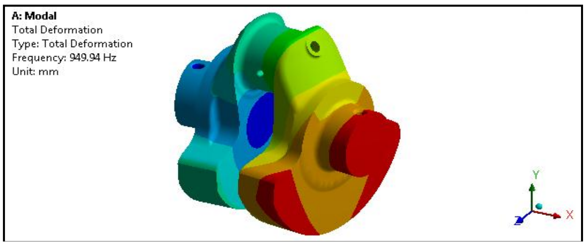

Fig. 6. Torsional Natural Frequency of Crank-Throw

Depending upon the results at different mode shapes, the torsional mode is identified and the associated frequency of is the torsional natural frequency. By FEA method, the torsional natural frequency (f_{n2}) is found to be,

$$f_{n2} = (949.94)\,Hz$$

Percentage error among analytical method & FEA method is,

$$percentage\_error = \left(\frac{f_{n1} - f_{n2}}{f_{n1}}\right) \cdot 100 = 6.601\%$$

The use of analytical method for calculating torsional natural frequency is limited to simple geometrical shape components, for complex shape components, analytical method is not feasible.

The analysis is done on complete crankshaft by FEA, as analytical method cannot be employed for complex geometry. The boundary conditions employed in the analysis are same as earlier *i.e. one side is fixed, and torque is applied on other side*, as shown in fig. 7.

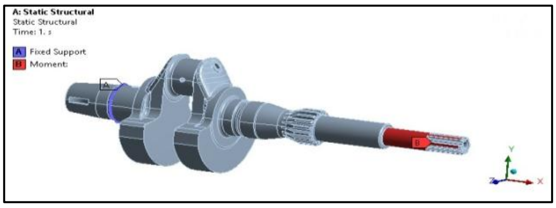

Fig. 7. Boundary Conditions for Crankshaft

The calculated torsional stiffness (K_t) will be equal to the applied torque (T) divided by the angle of twist or angular displacement (θ) at the free end. The value of angular displacement (θ) is found to be,

$$\theta = (0.33096)\,deg$$

The torsional stiffness of crankshaft is,

$$K_t = \frac{T}{\theta} = (395671.504)\frac{N \cdot mm}{deg}$$

The torsional natural frequency of crankshaft determined by FEA is shown in fig. 8.

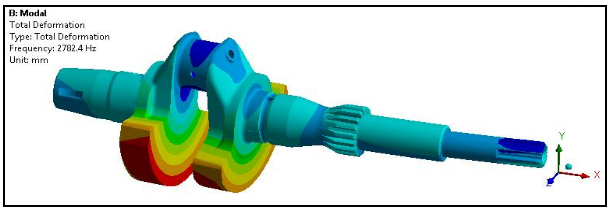

Fig. 8. Torsional Natural Frequency of Crankshaft

The torsional natural frequency (f_n) is found to be,

$$f_n = (2782.4)\,Hz$$

As said earlier, the study focus towards the aim of reducing torsional vibration, by improving torsional stiffness of the engine crankshaft. This is done by changing the different geometric parameters of the crankshaft.

Referring to Ukhande. M., Mane. R. and Shegavi. G. [9], torsional stiffness is directly proportional to pin diameter (D_c), web width (W), web thickness (L_w) & inversely proportional to throw (R) and pin width (L_c). Thus, by changing any of the parameters, corresponding change in torsional stiffness can be achieved. The different changes made in the crankshaft geometry are as follows:

1) Crankpin Diameter:

In this case, diameter of crankpin is increased by 10%, in order to find resultant effect on torsional stiffness. The boundary conditions were similar to the original model.

Using FEA, torsional stiffness of crankshaft prior increasing crankpin diameter was,

$$K_t = \frac{T}{\theta} = (395671.504)\frac{N \cdot mm}{deg}$$

After 10% increase in D_c, modified torsional stiffness (K_{t1}) was found to be,

$$K_{t1} = \frac{T}{\theta_1} = (397629.858)\frac{N \cdot mm}{deg}$$

Where,

$$\theta_1 = (0.32933)\,deg$$

The percentage increase in torsional rigidity (I) is,

$$I = \left(\frac{K_{t1} - K_t}{K_t}\right) \cdot 100 = 0.495\%$$

The natural frequency of the modified crankshaft can be evaluated similar to original crankshaft, as shown in fig. 9.

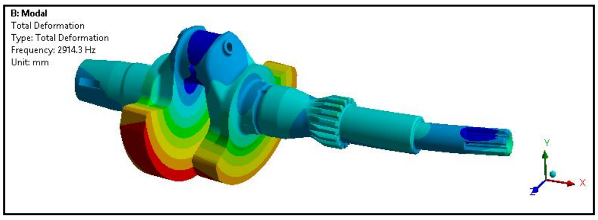

Fig. 9. Torsional Natural Frequency of Crankshaft for first case

The torsional natural frequency (f_{n1}) is found to be,

$$f_{n1} = (2914.3)\,Hz$$

Percentage increase in natural frequency is,

$$I = \left(\frac{f_{n1} - f_n}{f_n}\right) \cdot 100 = 4.74\%$$

Thus, Increase in torsional stiffness is about 0.5%, whereas increase in natural frequency is about 5%. However, increasing crankpin diameter calls for change in design of other components, such as, connecting-rod, bearings, etc. resulting in high cost. Hence, solution is not optimal.

2) Journal Diameter:

In this case, diameter of journal is increased by 10%, in order to find resultant effect on torsional stiffness. The boundary conditions were similar to the original model.

Using FEA, torsional stiffness of crankshaft prior increasing journal diameter was,

$$K_t = \frac{T}{\theta} = (395671.504)\frac{N \cdot mm}{deg}$$

After 10% increase in journal diameter (D_j), modified torsional stiffness (K_{t1}) is,

$$K_{t1} = \frac{T}{\theta_1} = (417961.256)\frac{N \cdot mm}{deg}$$

Where,

$$\theta_1 = (0.31331)\,deg$$

The percentage increase in torsional stiffness (I) is,

$$I = \left(\frac{K_{t1} - K_t}{K_t} \right) \cdot 100 = 5.633\%$$

The natural frequency in this case can be evaluated as,

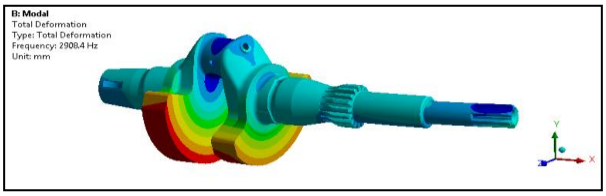

Fig. 10. Torsional Natural Frequency of Crankshaft for second case

The torsional natural frequency (f_{n1}) is found to be,

$$f_{n1} = (2908.4)\, Hz$$

Percentage increase in natural frequency is given by,

$$I = \left(\frac{f_{n1} - f_n}{f_n} \right) \cdot 100 = 4.52\%$$

Hence, by increasing journal diameter (D_j) by 10%, an increase of about 6% is achieved in torsional stiffness and about 4.5% in torsional natural frequency.

3) Throw:

In this case, throw (R) distance is increased by 1.5 mm. The change is applied to crankshaft, and boundary conditions are similar to previous case.

The torsional stiffness of the original crankshaft was,

$$K_t = \frac{T}{\theta} = (395671.504) \frac{N \cdot mm}{deg}$$

After suggested change, torsional stiffness is found to be,

$$K_{t1} = \frac{T}{\theta_1} = (395623.68) \frac{N \cdot mm}{deg}$$

Where,

$$\theta_1 = (0.331)\, deg$$

The percentage increase in torsional stiffness (I) is,

$$I = \left(\frac{K_{t1} - K_t}{K_t} \right) \cdot 100 = -0.012\%$$

The Negative (-) sign specifies decrease in torsional stiffness. The torsional natural frequency can be evaluated as shown in fig. 11.

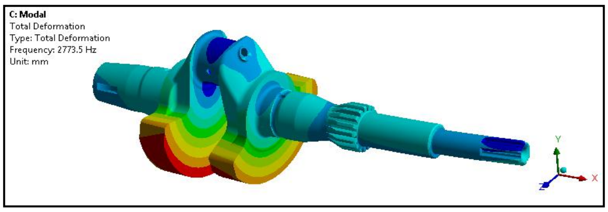

Fig. 11. Torsional Natural Frequency of Crankshaft for third case

The torsional natural frequency (f_{n1}) is found to be,

$$f_{n1} = (2773.5)\, Hz$$

The percentage increase in natural frequency is,

$$I = \left(\frac{f_{n1} - f_n}{f_n} \right) \cdot 100 = -0.32\%$$

The Negative (-) sign specifies decrease in torsional natural frequency. Hence, by increasing throw (R) distance in crankshaft by 1.5 mm, torsional stiffness is decreased by about 0.01%, whereas, torsional natural frequency gets reduced by 0.32%. Thus, increasing throw (R) distance adversely affects torsional vibrations, but it calls for change in design of connecting-rod.

4) Face-inclination:

The various geometrical parameters of crankshaft, like, crankpin diameter (D_c), journal diameter (D_j), length of crankpin (L_c), throw (R), web thickness (L_w), & web width (W), affects its torsional stiffness.

But changing any one of the parameter affects design of one or several adjoining components. Thus, there is a need to optimize torsional vibrations in engine crankshaft without changing other components. Hence, a new geometrical parameter, Face-inclination (FI), is considered to optimize torsional vibrations, as shown in figure 12.

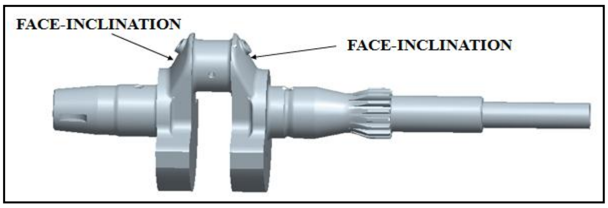

Fig. 12. Face-Inclination (FI) Area of Crankshaft

Addition of material in FI area results in increased torsional stiffness, without affecting design of other components. The amount of material added in FI area is 144 gm, as shown in fig. 13.

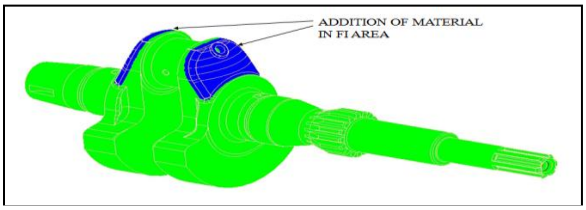

Fig. 13. Material Addition in Face-Inclination (FI) Area

The torsional stiffness in this case is evaluated using same boundary conditions. The torsional stiffness of original crankshaft prior to addition of material in face-inclination (FI) area was,

$$K_t = \frac{T}{\theta} = (395671.504) \frac{N \cdot mm}{deg}$$

The torsional stiffness in this case is found to be,

$$K_{t1} = \frac{T}{\theta_1} = (455847.951) \frac{N \cdot mm}{deg}$$

Where, $\theta_1 = (0.28727)\, deg$

The percentage increase in torsional rigidity (I) is,

$$I = \left(\frac{K_{t1} - K_t}{K_t} \right) \cdot 100 = 15.209\%$$

The torsional natural frequency can be evaluated as,

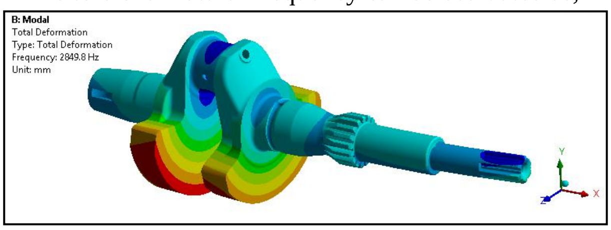

Fig. 14. Torsional Natural Frequency of Crankshaft for fourth case

The torsional natural frequency (f_{n1}) is found to be,

$$f_{n1} = (2849.8)\, Hz$$

The percentage increase in natural frequency is,

$$I = \left(\frac{f_{n1} - f_n}{f_n} \right) \cdot 100 = 2.42\%$$

Thus, a material addition of 144 gm. in face-inclination (FI) area results in increased torsional stiffness of about 15% and about 2.5% increase in natural frequency. Also, this change does not affect design of any adjoining components. Hence, this change overcomes drawbacks of previous cases and is most optimal, compared to other cases.

VIII. FATIGUE LIFE

Another objective of the study involves increasing fatigue life of the original crankshaft. First, the fatigue life of existing crankshaft is evaluated by FEM, as shown in fig. 15.

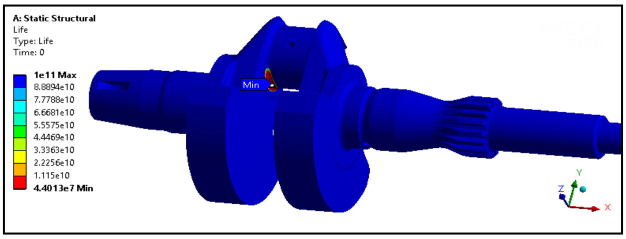

Fig. 15. Fatigue Life of original crankshaft

The fatigue life of the original crankshaft model is found to be,

$$N_1 = \left(4.401 \cdot 10^7\right) cycles$$

The fatigue life of the most optimal change (face-inclination) employed in crankshaft geometry for increasing torsional stiffness is also evaluated by FEM. The results of the fatigue life analysis are shown in fig. 16.

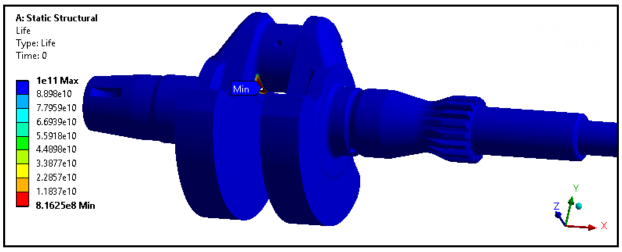

Fig. 16. Fatigue Life of modified design

The fatigue life of the suggested modification is,

$$N_2 = \left(8.162 \cdot 10^8\right) cycles$$

Thus, Addition of material in Face-inclination area results in an increased torsional stiffness. Also, the fatigue life of crankshaft increases by this change.

I. CONCLUSION

In this way, the phenomenon of torsional vibrations and its influence on engine performance, particularly on engine crankshaft is studied. The study can be concluded as follows:

Firstly, a procedure to determine the torsional stiffness of the given components is prepared. The torsional stiffness of the crank throw (fig. 4) is determined by analytical & FEA method and their comparison is carried out. The results of analytical & FEA method are in good agreement. The same procedure is adopted to determine torsional stiffness of existing complete crankshaft (fig. 7) through FEA method, as the analytical method is not feasible, for evaluation of torsional stiffness of crankshaft.

Secondly, another procedure for evaluation of the torsional natural frequency is prepared. The procedure is first applied to crank throw (fig. 4) and comparison is made between analytical & FEA results, the results also matches closely. Similarly, the procedure is applied to crankshaft, for determining torsional natural frequency.

Then, the different parameters affecting torsional stiffness are identified, and their effect on torsional stiffness is studied to reduce the torsional vibrations in engine crankshaft, followed by suggestion of most optimal modification. The change in fatigue life produced by most optimal modification of torsional stiffness, is compared with original one.

ACKNOWLEDGMENT

The Author wish to acknowledge the support of Greaves Cotton Limited, during the study.

REFERENCES

[1] Montazersadgh. F. H., Fatemi. A, "Optimization of a Forged Steel Crankshaft Subject to Dynamic Loading", SAE Paper No. 2008-01-0432.

[2] Boysal. A., Rahnejat. H., "Torsional Vibration Analysis of a Multi-Body Single cylinder Internal Combustion Engine Model", Department of Mechanical & Manufacturing Engineering, University of Bradford, Bradford, West Yorkshire, UK, 1997, 21:481-493, August.

[3] Feese. T., Hill. C., "Prevention of Torsional Vibration Problems in Reciprocating Machinery", Proceedings of the Thirty-Eight Turbo machinery Symposium, 2009.

[4] Charles. P., Sinhala. J. K., Gu. F., Lidstone. L., Ball. A. D., "Detecting Crankshaft Torsional Vibration of Diesel Engines for Combustion Related Diagnosis", Journal Of Sound & Vibration, 321 (2009) 1171–1185.

[5] Wilson. W. K., "Practical Solution of the Torsional Vibration Problems, Devices For Measuring Vibration Amplitude", Volume No.4, Chapman & Hall Publication, 1972.

[6] Ronald L. Eshleman, "Chapter 38. Torsional Vibration in Reciprocating & Rotating Machines", Harris Shock & Vibration Handbook, Harris. C. M. & Piersol. A.G." Fifth Edition, McGraw Hill Publication.

[7] Wilson. W. K., "Practical Solution of the Torsional Vibration Problems, Frequency Calculations" Volume No. 1, Chapman & Hall Publication, 1946.

[8] Milasinovic. A., Filipovic. I. & Hribernik. A., "Contribution to the Definition of the Torsional Stiffness of Crankshaft of a Diesel Engine used in Heavy-Duty Vehicles", International Journal of Automobile Engineering, Vol. 223, page 921-929, 2009.

[9] Ukhande. M, Mane. R and Shegavi. G, "Optimization of Crankshaft Torsional Rigidity for Fatigue Strength Improvement Using CAE", CDFD Engineering. Bharat Forge Ltd. Pune, India, SAE Paper No.2012-01-0404.

[10] Kimura. J., Kai. R., & Shibata. S., "Six-cylinder Inline Turbo-charged Diesel Engine Crankshaft Torsional Vibration Characteristics" , SAE Paper No. 2001-01-2719.

Amitpal Singh Punewale is the student in the final semester of the Master of Technology (Mechanical - CAD/CAM) in SGGSIE&T, Nanded, India, having research interests in CAD/CAE field.

Mr. Tushar Khobragade is the Deputy General Manager of Greaves Cotton Limited, Aurangabad, India, completed his Master of Engineering (Mechanical Engineering) from BITS Pilani, having more than 10 years of industrial experience.

Mr. Amit Chaudhari is the Manager (FEA), in Greaves Cotton Limited, Aurangabad, India, completed his Master of Technology (Design Engineering) from IIT Chennai, having industrial experience of 7 years.

Prof. N. M. Khandre is the Associate Professor in Department of Production Engineering in SGGSIE&T, Nanded, India, completed his Master of Engineering (Mechanical - CAD/CAM) from SGGSIE&T, Nanded, having teaching experience of about 20 years.