Finite Element Approach for Study of Torsional and Bending Effect on Four Cylinder Engine Crankshafts

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Abstract: **In high-speed diesel engine the problem of torsional vibration of the crankshaft has become critical with increase in excitation forces. This results in high torsional vibration amplitudes and hence high stresses .This paper aims at complete FEM analysis of a crankshaft for torsional and bending vibrations & identification of stresses. It is analyzed for natural frequency, rigid body mode shape by ANSYS. . The complete simulation of actual boundary conditions is done for journal bearing support, inertia lumping for reciprocating parts and bearing stiffness. ANSYS-programme, which will convert user input Pressure-Crank angle variation to excitation forces for various orders through FFT. The dynamic responses obtained for displacement and stresses. Finally all results are combined to obtain the variation of Fillet Stress as a function of engine speed and harmonic orders. The critical dynamic response is compared with results obtained experimentally for torsional amplitudes. The parametric optimization has been done to increase the frequency it means the higher the frequency higher the stiffness or strength with optimum weight of the component.**

***Keywords*:** **Torsional vibration, Bending vibration, Modal analysis, Rigid body modes, Static analysis, Critical fillet stresses, , FFT setup.**

I. INTRODUCTION

Crankshaft is one of the most important moving parts in internal combustion engine. Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston into a rotary motion. Crankshaft consists of the shaft parts which revolve in the main bearings, the crankpins to which the big ends of the connected rod are connected the crank arms to webs (also called cheeks) which connect the crankpins and the shaft parts. The crankshaft main- journals rotate in a set of supporting bearings (main bearings) causing the offset road journals to rotate in circular path around the main journal centers.

The complicated geometry of crankshaft and the complex torque applied by cylinders make their analysis difficult. But optimized meshing and accurate simulation of boundary conditions along with ability to apply complex torque provided by various FEM packages have helped the designer to carry torsional vibration analysis with the investigation of critical stresses.FEM enables to find critical locations and quantitative analysis of the stress distribution and deformed shapes under loads. The specific engine crankshaft of a major automobile company (the name is kept confidential) is taken as the model for the analysis.

Objectives: The project aims at detail FEM analysis of crankshaft. The following are the main objectives of the project.

1. Building a 3-D Solid parametric model of crankshaft, Flywheel and pulley in Pro-Engineer wild fire.

2. Meshing the model by Tetrahedral Solid 285 elements in Ansys. MPC 184 element is also used to define the journal bearings.
3. Rigid body modes and Normal modes- we have calculated in free vibration analysis for Crankshaft.
4. Experimental modal analysis carried out using FFT ANALYSER MACHINE to validate our CAE work.
5. Behaviour of torsional modes, bending modes and combined modes of vibration we have studied for all systems.
6. ANSYS program is used to find out the F_t and F_r forces from given P and θ diagram.
7. C programming is used to find out the T and θ diagram from corresponding P and θ diagram.
8. Dynamic response and investigation of critical stresses are found out by incorporating the boundary conditions at the journal bearing positions. And tangential and radial forces on corresponding nodes.
9. Durability analysis has done to check the strength of the crankshaft with different radial and tangential forces.
10. The parametric optimization has been done to increase the frequency it means the higher the frequency higher the stiffness or strength with optimum weight of the component.

***Crankshaft Vibrations*:** In I.C. engines various types of excitation forces exist. These directly or indirectly affect the crankshaft dynamics.

The major types of these vibrations are:

• ***Torsional vibrations***: shows spatial view of a 4-cylinder engine crankshaft. In multi-cylinder engine crankshafts, the crank throws are spatial or out of phase with each other for the balancing purpose. It is also attached with a flywheel and some driven system. The torque is applied to the crankpin by the connecting rod. This torque is of varying nature because of variation in gas pressure and inertia forces. The fluctuating torque at the crankpin causes the twisting and untwisting periodically. Hence the torsional vibrations are induced.

• ***Flexural vibrations***: The lateral periodic motion of crankshaft under the fluctuating forces exerted by connecting rod at crankpin cause bending vibrations of crankshaft. This mode shape generally has many nodes because the bending vibrations are strongly reacted at the bearings.

• ***Axial vibrations***: The torsional vibrations can cause axial vibration in the twisting and untwisting motion. Also radial forces at crankpin cause some axial movement of crankthrow.

• ***Coupled vibrations***: In general, however the various modes of vibration are coupled so that vibrations of one type can't occur without an accompanying vibration of the other type. These are not troublesome if there is considerable spread between the natural frequencies of the modes of vibration involved; i.e. the modes get weakly coupled.

***Influence of Crankshaft Vibrations*:**

The crankshaft vibrations badly affect the working of engine.
The major areas are as follows.

1. The torsional vibrations cause the angular velocities of all the cranks to vary but not in the same proportions. The crank away from the node has maximum effects compared to crank near the node. This affects the balancing.

2. Due to same reason discussed above, stresses of varying intensity are generated in whole length of the crankshaft. These are also fluctuating in nature and hence cause fatigue of crankshaft, reducing its life. The stresses induced are dangerous at fillet or oil-hole locations.

3. Vibratory energy is transmitted to all parts of the structure where it causes structural damage.

4. It induces noisy operation of engine, which is undesirable in passenger cars. It also causes wear of all running parts.

II. SOLID MODELING OF CRANKSHAFT

As a prerequisite to the finite element model is the physical geometry of the part i.e. the suspension link we have created using Prove-Wildfire software.

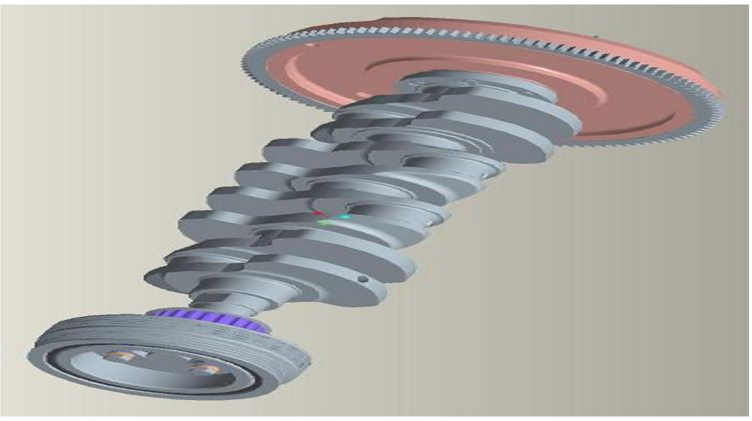

Fig 1. Solid Modeling of Crankshaft

III. MESH GENERATION

Meshing generally falls in two categories depending on the geometry of the element. For a 3D machine element of regular shape, solid meshing is sufficient, but for irregular geometries we have to first use surface meshing and then solid meshing.

We have done surface meshing first by Solid 183 element. One of the important aspect in surface meshing is merging of nodes or technically it can be called node equivalence. Once the node equivalence is confirmed we have to go for free-edge checking which will ensure that there are no free surfaces. Now, after free surface check is done, we go for quality check, the quality check in Ansys is a unique feature as it allows us not only to check for internal and external angles of the mesh element but also facilitates in checking aspect ratio, warpage ratio, skew ratio, and most important the Jacobian matrix. The value of the Jacobian matrix should always lie between 0 and 1. Any other value of the Jacobian matrix renders the element faulty and a new element should be created by deleting the previous one. In our case the value of Jacobian matrix is 0.7

Once assured with a safe and sound surface meshing our next step is to import the model in ANSYS for solid meshing.

The element used for solid meshing is ***10 Node Solid 187 Tetrahedral Element***. The special features of this element are Plasticity, Creep, Swelling, Stress stiffening, Large deflection, Large strain, Birth and death.

MPC184

Multipoint Constraint Elements: Rigid Link, Rigid Beam, Slider, Spherical, Revolute, Universal MPC184 comprises a general class of multipoint constraint elements that implement kinematic constraints using Lagrange multipliers. The elements are loosely classified here as “constraint elements” and “joint elements”. All of these elements are used in situations that require you to impose some kind of constraint to meet certain requirements. The constraint may be as simple as that of identical displacements at a joint. They can also be more complicated, such as those that involve rigid modeling of parts, or kinematic constraints that transmit motion between flexible bodies in a particular way. For example, a structure may consist of some rigid parts and some moving parts connected together by some rotational or sliding connections. The rigid part of the structure may be modeled using the MPC184 Link/Beam elements, while the moving parts may be connected with the MPC184 slider, spherical, revolute, or universal joint element. Since these elements are implemented using Lagrange multipliers, the constraint forces and moments are available for output purposes. This element is used to define the connectivity between the point of application of force and the nodded on the surface of the structural member.

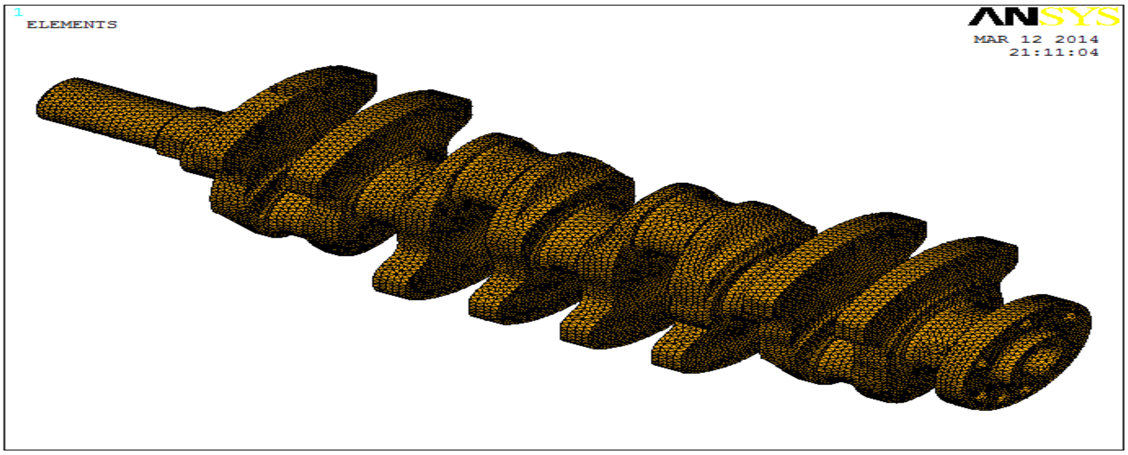

Fig 2. Meshed model of crankshaft

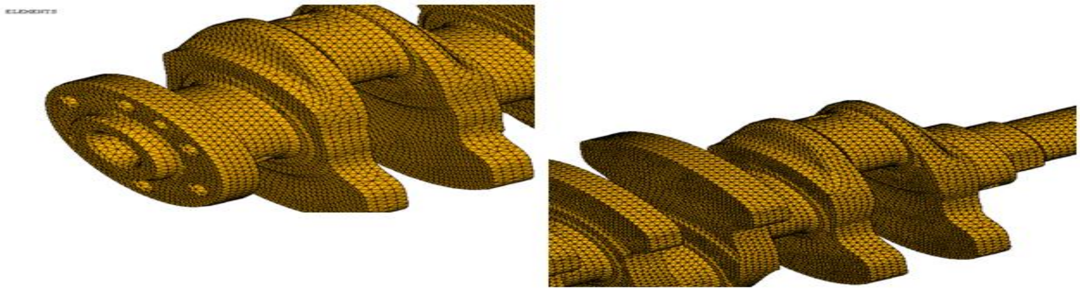

Fig 3. Enlarge view of dense meshed model

IV. EXPERIMENTAL SETUP FOR NATURAL FREQUENCY

The FFT analyzer used is dual channel FFT (Fig.4) analyzer 2900 B from Larson & Davis Company Ltd. It is full function, yet portable, time/frequency signal analyzer. In this experimental analysis, we hang the crankshaft at its C.G. and excitation is given by hammer. Then we get results from FFT, which are given in table 2.

Fig. 4. Experimental setup

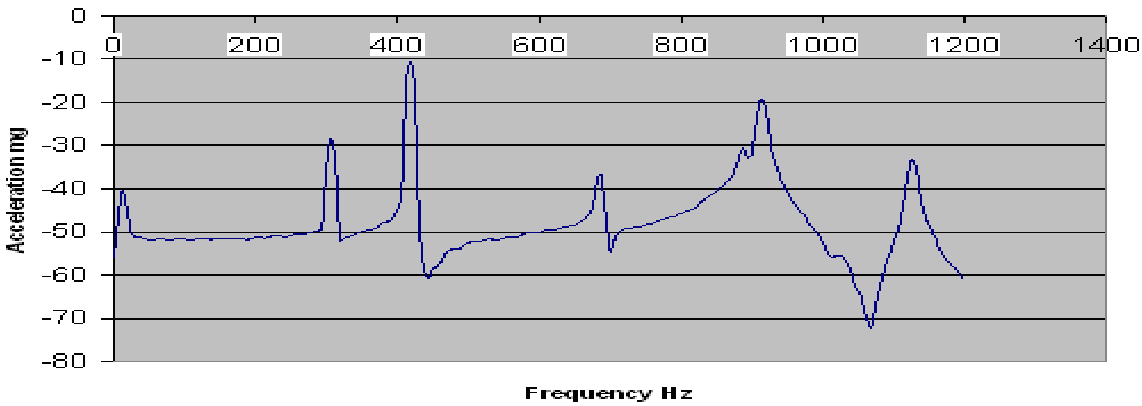

Fig. 5 Frequency Vs Magnitude by Normal Function

From below table we see that Ansys results are agree with the experimental results. So our software results are accurate and these results are used for further analysis.

Table I. Comparison for Crankshaft and Pulley

Mode	ANSYS Results	FFT Results	% Error
7	337.4001	306.25	7 %
8	452.6897	418.75	7.5 %
9	757.2222	687.50	8 %
10	972.9449	918.75	4 %
11	998.1289	1100.00	9 %

V. TORSION AND BENDING MODE ANALYSIS

Keeping in mind the different mode of vibration as discussed in chapter no 2; we are interested in the study of the following systems. The effect of different assembled components on the crankshaft mode shape and natural frequency is our objective.

Table II. Natural Frequencies from Ansys

Mode No.	Frequency	Mode
7	408.8705952033	First Bending Mode
11	1054.341510378	First Torsion Mode
14	1832.614484473	First Combined Mode

A) Deformation Plot For 1st Bending Mode of Only Crank Shaft

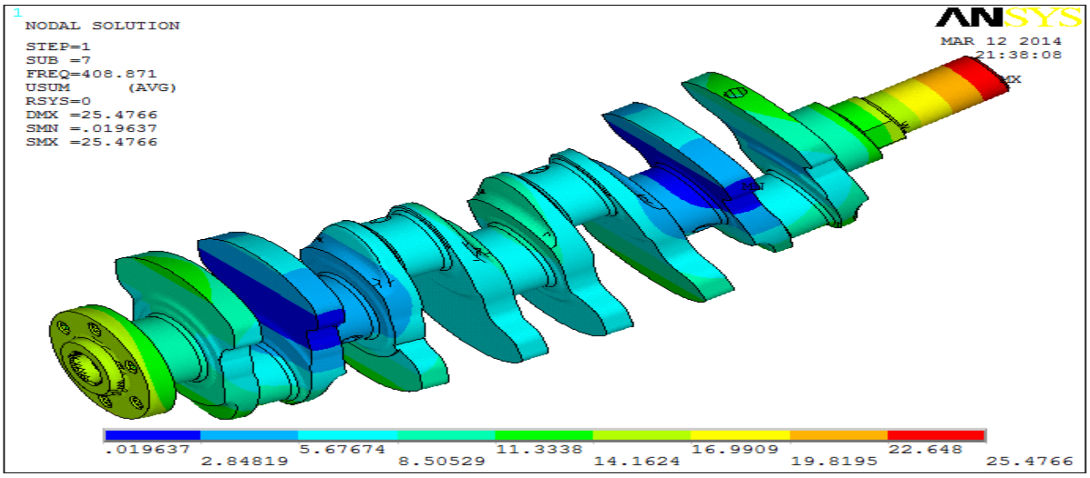

Fig 6. Deformation Plot For 1st Bending Mode of Only Crank Shaft

From above plot we found the maximum deflection is 25.977 mm and minimum deflection is 0.019637 mm.

B) Deformation Plot For 2nd natural frequency Mode of Only Crank Shaft

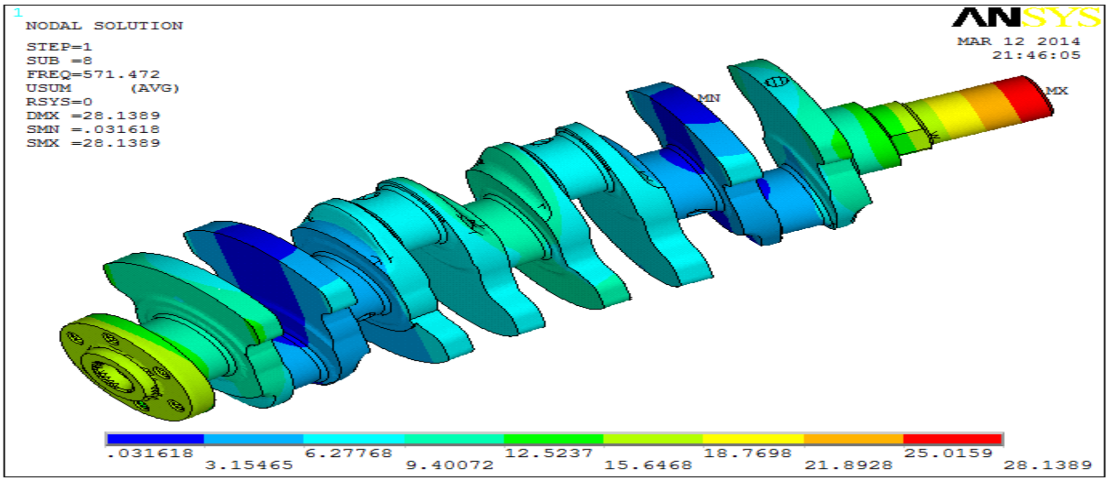

Fig 7. Deformation Plot For 2nd natural frequency Mode

From above plot we found the maximum deflection is 28.1389 mm and minimum deflection is 0.031618 mm.

C) Deformation Plot For 1st Combined Mode of Only Crank Shaft

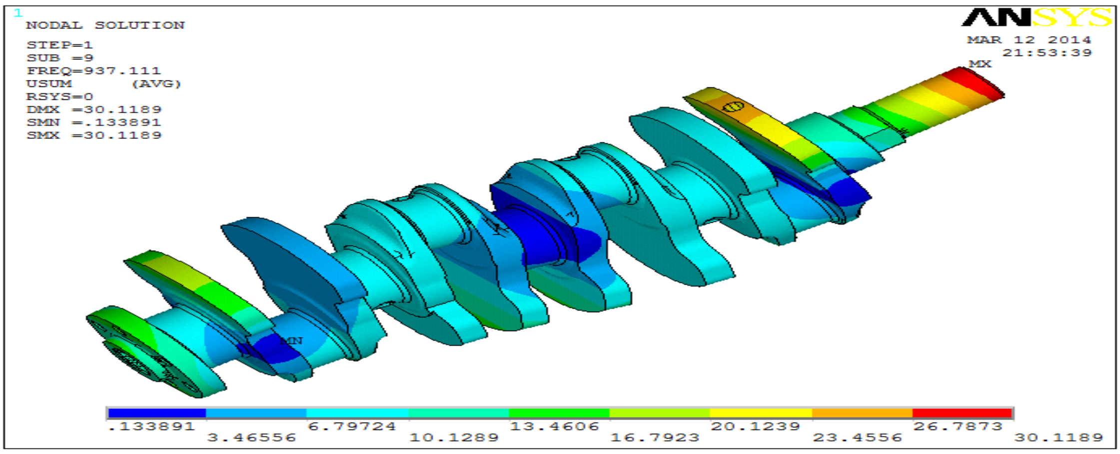

Fig 8. Deformation Plot For 1st Combined Mode

From above plot we found the maximum deflection is 30.1189 mm and minimum deflection is 0.1338 mm.

Force Application: We get data about crank angle (Θ) and corresponding pressure from company. Further we calculate the data of crank angle (Θ) and corresponding torque using „C" programming language.

Output of C program: Below figure shows output of C program.

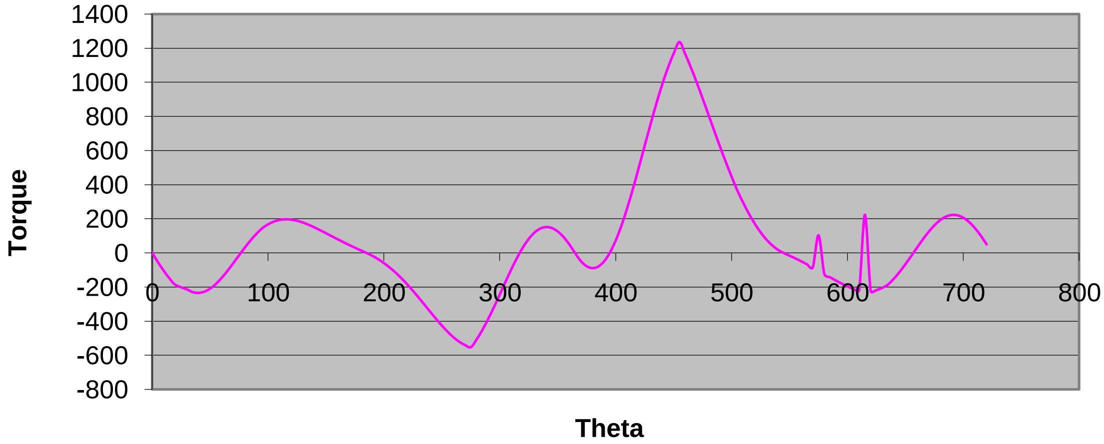

Fig 9. Torque Vs Theta graph.

ANSYS Output table for f_t, f_r torque from macros:

Crank Radius (m) = 0.0450 Connecting Rod Length (m) = 0.1450

Engine Bore Diameter (m) = 0.0831 Piston Area (m^2) = 0.0054

Engine Speed (rpm) = 4300.00 Angular Speed (rad/s) = 450.4762

Mass of Reciprocating Parts (kg) = 0.8860

Table for some of the values of output

Table III. Values of output torque

Crank Angle (Degree)	**Gas Pressure (N/mm^2)**	**Con. rod Force (N)**	**Tangential Force (F_t) (N)**	**Radial Force (F_r) (N)**	**Output Torque (N-m)**
0	0.00000	-10601.700	0.00000	-10601.700	0.000
5	0.00720	-10497.469	-1197.918	-10428.895	-53.906
10	0.01440	-10263.893	-2325.359	-9997.010	-104.641
15	0.02170	-9903.354	-3324.537	-9328.659	-149.604
20	0.02890	-9421.864	-4145.564	-8460.840	-186.550
25	0.03610	-8825.778	-4748.502	-7439.494	-213.682

VI. ANALYSIS FOR RADIAL FORCE

We get radial and tangential force values from ANSYS macro. Further we apply these forces on crankshaft and solve it with ANSYS solver.

To get the effect of bending forces on the crankshaft, we have applied the radial forces F_r on the crankpin positions and the displacement constraints is applied at the multiple journal bearing positions. The study of maximum stress distributions around the fillet which may be stress concentration region is our objective. Fig. shows application of radial force.

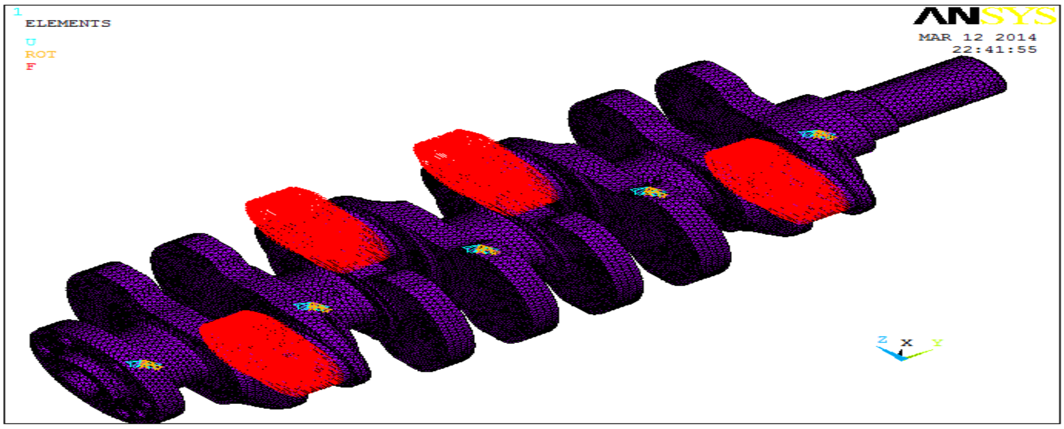

Fig 10. Boundary Conditions For Radial Stress Analysis.

A) Plot For Stress Due To Radial Forces

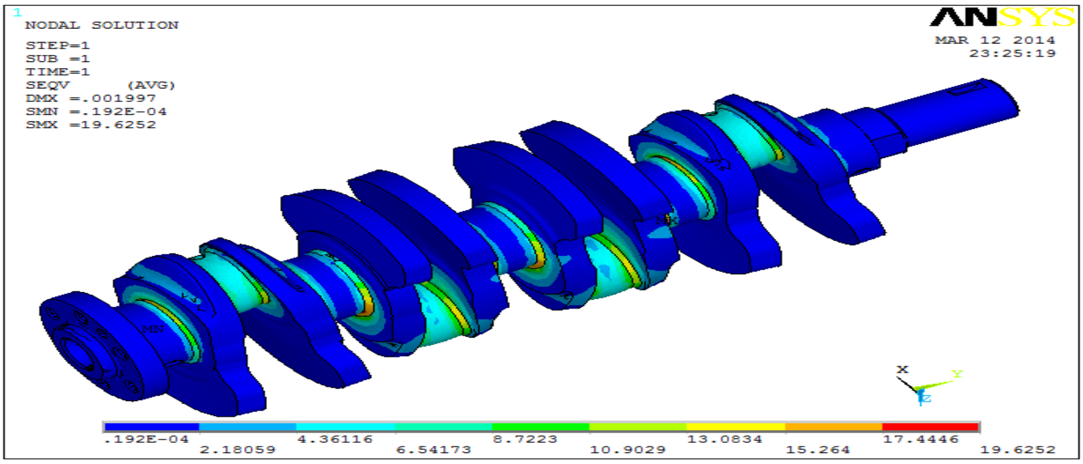

Fig 11. Stress Due To Radial Forces

From above plot we found the maximum von mises stress value is 19.62 N/mm^2 and minimum stress value is 0.19E-4 N/mm^2.

Above figure shows stress concentration at fillet area at flywheel end crankpin position.

VII. ANALYSIS FOR TANGENTIAL FORCE

We get tangential force values from ANSYS, and further we apply these forces on crankshaft.

To get the effect of torsional forces on the crankshaft, we have applied the tangential forces F_t on the crankpin positions and the displacement constraints is applied at the end of one side journal bearing positions to see the effect of toque developed by tangential forces at different lobes of crankshaft. The study of maximum stress distributions around the crankpin positions, which may be stress concentration region is our objective.

Following figure shows tangential force application on crankshaft.

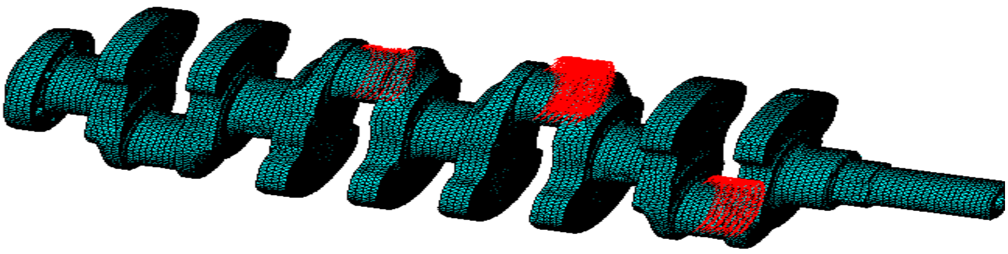

Fig 12. Application of tangential force.

A) Plot for Stress Due To Tangential Forces

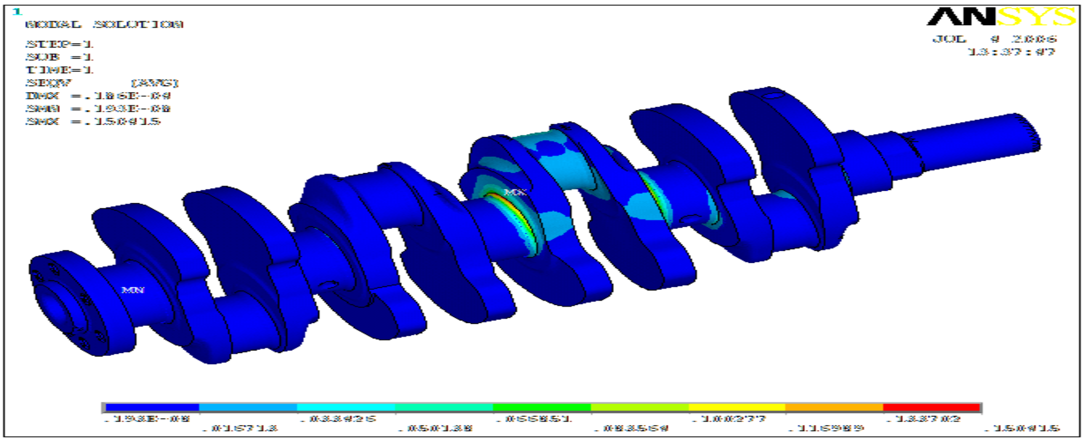

Fig 13. Stress Due To Tangential Forces

From above plot we found the maximum von mises stress value is 20.464 N/mm^2 and minimum stress value is 0.19E-4 N/mm^2

VIII. PARAMETRIC OPTIMISATION

To study the effect of different dimensional parameters on stiffness and frequency of the crank shaft. The parametric optimization has been done to increase the frequency it means the higher the frequency higher the stiffness or strength with optimum weight of the component.

In Ansys the Optimization begins with building a parametric model of the initial design and creating an analysis file.

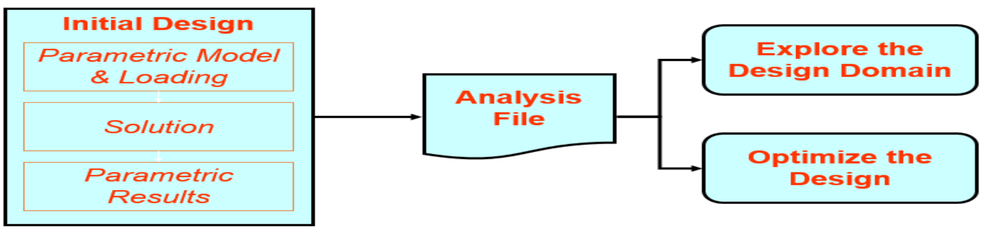

Fig. 14 Steps of optimization in Ansys

There are four main steps (assuming that the analysis file is available):

1. Identify the analysis file
2. Identify optimization variables - DVs, SVs, and objective function

3. Run the optimization
4. Review results

The APDL programme has generated for input and parametric model and for optimisation solution steps

```
/PREP7
!Generate Volume1
D1=35 !Parameter1
D2=16 !Parameter2
L1=55 !Parameter3
L2=52 !Parameter4
C1=1.5
K,1,0,D2/2,,
K,2,0,D1/2-C1,,
K,3,C1,D1/2,,
K,4,L1,D1/2,,
K,5,L1,0,,
K,6,L2,0,,
K,7,L2,D2/2,,
L,1,2
L,2,3
!
!
/solu
!
/post1
!
!
!!!!!!!!!!!!!!!!! The Programme for optimization solution!!!!!!!!!!!!!!!!!!!!!!!!!!!!!!!!!!!
/INPUT,'macro2','txt',
/OPT
SAVE
OPANL,'macro2','txt',' '
OPVAR,D1,DV,30,37, ,
OPVAR,D2,DV,12,17, ,
OPVAR,D4,DV,40,60, ,
OPVAR,L4,DV,20,24, ,
OPVAR,FREQ1,SV,0,350, ,
OPVAR,FREQ2,SV,0,550, ,
OPVAR,DFREQ,OBJ, , , ,
OPDATA, , ,
OPLOOP,TOP,IGNO,SCAL
OPPRNT,OFF
OPKEEP,ON
OPTYPE,SUBP
OPSUBP,5,3,
OPEQN,0,0,0,0,0,
OPEXE
! REVIEW RESULTS
oplist                  ! List all design sets
plvaropt,D1,DFREQ   ! Graph angles vs. set number
plvaropt,D1            ! Graph mid x-loc vs. set number
plvaropt,D2           ! Graph mid y-loc vs. set number
plvaropt,FREQ1          ! Graph max hoop stress vs. set number
```

```
plvaropt,FREQ2                          ! Graph frequency vs. set number
plvaropt,D1,D2,FREQ1,FREQ2,D4,L4,DFREQ          ! Graph total volume

Finish
```

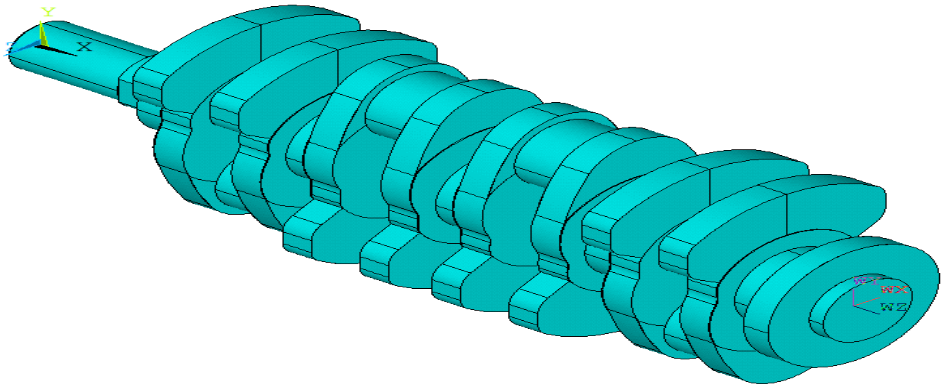

Fig. 15 Parametric model generated by APDL programme

```
OPLIST  Command
File

LIST OPTIMIZATION SETS FROM SET    6 TO SET    6 AND SHOW
ONLY OPTIMIZATION PARAMETERS

                          SET   6
                        (FEASIBLE)
FREQ1     (SV)           311.71
FREQ2     (SV)           507.38
D1        (DV)           33.708
D2        (DV)           16.947
D4        (DV)           52.633
L4        (DV)           20.604
DFREQ     (OBJ)          507.38
```

Fig. 16 List of the best feasible optimization set

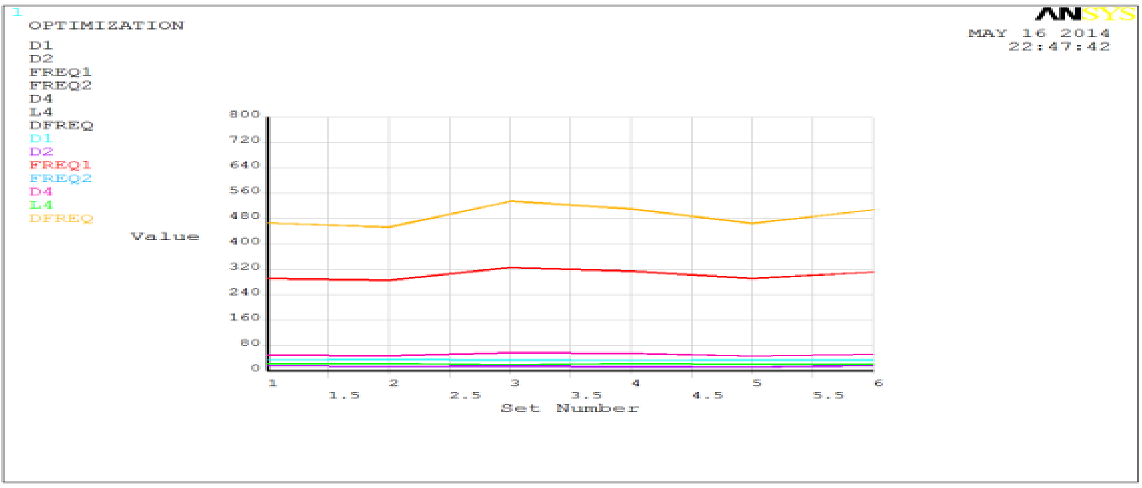

Fig. 17 The parameters variation with optimization

The below is the optimum mass and volume after the parametric optimization

```
SUMMATION OF ALL SELECTED VOLUMES
    TOTAL VOLUME = 0.16826E+07 mm3
    TOTAL MASS   = 0.13209E-01 in Tones
    CENTER OF MASS:  XC= 234.12      YC= 0.0000      ZC= 0.0000

                       *** MOMENTS OF INERTIA ***

       ABOUT ORIGIN      ABOUT CENTER OF MASS     PRINCIPAL
   IXX =    21.775             21.775             21.775
   IYY =    857.70             133.71             133.71
   IZZ =    868.89             144.90             144.90
   IXY =  -0.24706E-04       0.44850E-05
   IYZ =  -0.91886E-07      -0.91885E-07
   IZX =  -0.19755E-05       0.22381E-04
```

Fig 18. Optimum mass and volumez

The best feasible set is set number 6 which indicates the best optimum dimensions i.e. D1, D2, D4 and L4. The Figure 11 indicates there is not much effect of dimension D1 on frequency.

IX. DICUSSION AND CONCLUSION

1. Experimental validation of crankshaft is done using FFT analyser. The values of FFT output and Modal frequency calculated from ANSYS and Hypermesh validates the results as both are matching.
2. The mode shape calculation of different systems of component is done using modal analysis. The effect of flywheel and pulley shows the natural frequency is decreasing due to additional masses at the end of crankshaft.
3. The bending mode, torsion mode and combined mode frequency values calculated by Ansys shows the actual working condition deformation during high speed rotation rpm value of crankshaft.
4. As the stress concentration area for radial forces is fillet and for tangential force it is crankpin positions. Both the stresses plot study shows the critical stress region.
5. We conclude our results and discussion by viewing displacement plot for different mode shapes that the natural frequency is affected or decreased due to the addition of pulley and flywheel. So the resonance condition is more critical for the combined assembly. The Vonmises stress plot concludes that during maximum rpm or maximum torque condition failure will take place near the fillet area.

X. FUTURE SCOPE

1. As we calculated a_n, and b_n values, further it is possible to do harmonic analysis and optimize the crankshaft again for dynamic conditions.
2. It is prove that the meshed model and simulated boundary conditions are accurate hence mashed model can be used for type of other analysis like thermal loading, fatigue analysis, mechanism analysis etc.
3. Variation of cylinder pressure as a function of engine rpm can be included and their effect on the induced stresses can be studied.

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