# Design And Analysis Of Zero Coupled Compressor Crankcase

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**Abstract:** The Piston compressor is a positive displacement compressor with crankshaft, connecting rod, crankcase, piston etc. as main components. Piston compressor are highly used in oil and gas industries as they demand high pressure air, also chemical, petrochemical and process gas plant use piston compressor for compressing final product gases. This compressor used at CNG stations to refill the automobile CNG tanks. The crankcase is the core structure of the reciprocating compressor, supports and encloses almost all internal components of piston compressor. The main function of the compressor crankcase is to provide sufficient rigidity to the compression mechanism to make it work properly. Another function of the crankcase is to act as a limiter that absorbs any shock that may occur during compressor transport. Crankcase is one of the bulky part in piston compressor and there persist the scope of weight reduction of crankcase. This paper focus on static and harmonic analysis of crankcase . For doing analysis of crankcase along with crankcase crankshaft, connecting rod, crosshead guide, cross head also weight of cylinder and piston are considered. For analysis purpose FEA solver (ANSYS) is used. static analysis is done to check stresses in crankcase, harmonic analysis is done to evaluates vibrations in terms of velocity. After transient dynamic analysis By analysing the results optimization of crankcase will be done for weight reduction.

**Keywords:** Compressor, crankcase, harmonic analysis static analysis,

**Nomenclature**

l = Length of connecting rod
r = Crank radius
$P_{HE}$ = Head end pressure
$F_{GHE}$ = Gas force at head end side
$F_I$ = Inertia force
F = Total force
Q = Force in connecting rod
$F_N$ = Normal force
$F_r$ = Radial force
$F_t$ = Tangential force
T = Torque
Θ= Crank angle

## I.INTRODUCTION

A compressor is a device for increasing the pressure of a fluid due to working energy. Compressors are often one of the most important and expensive systems in production and deserve special attention. Natural gas pipelines, petrochemical plants, refineries and many other industries rely on this type of equipment. Various types of compressors are found in almost every industrial facility. Reciprocating compressors is a positive displacement having the advantages comparatively low price, easy maintenance, suitable for high pressure. The crankcase is the core structure of the reciprocating compressor, supports and encloses almost all internal components of piston compressor. The main function of the compressor crankcase is to provide sufficient rigidity to the compression mechanism to make it work properly.Crankcase is one of the bulky part in reciprocating compressor and there persist the scope of weight reduction of crankcase. For doing this analytical load calculation is done for various crank angle also FEA solve ( ANSYS ) is used for analysis aspect. [1] represents the load acting on the compressor. Limiting the load on the rod affects the number of stages because the total load on the rod is related to the differential pressure in the cylinder. increasing the number of stages will obviously reduce the pressure drop in each stage. [2] discusses about two different crankcase structure analysis methods by which crankcase can be analyzed. The first method is conventional quasi-static modeling, which will not be described in detail in this work. The second method involves determining the dynamic loads generated by the torsion, bending and axial vibration of the crankshaft on the crankcase.[3] represents finite element analysis with use of FEA solver ANSYS. CAD model of crankcase of a single cylinder diesel engine is used.. The main unknown factor in this structural analysis is displacement, and other quantities, such as deformation, stress and reaction forces, are derived from the displacement of the nodes.In this paper formulae of calculation of forces are given and how to apply forces on crankshaft. Maximum principal stress theory is useful for brittle material. The theory of maximum principal stress indicates that damage occurs when the maximum principal stress occurring in the body exceeds the uniaxial ultimate tensile/compressive strength (or yield strength) of the material.[4] Discussed the crankcase failure event found by TVS Automotive Co., Ltd. when testing the scooter engine strength.To address the issue FEM techniques were applied to investigate the stress fields in the region of crack formation. The authors here considered unit static load acting in the stress fields due to unavailability of direct methods to evaluate actual dynamic load value. [5] describes vibration in mechanical drive. Every mechanical drive experiences vibrations due to improper design, imbalance masses, misalignment, looseness etc. In case of rotating machinery the mass imbalance causes vibration and chances of failure due to that vibration. Reciprocating compressor consists of three main mechanical components piston, connecting rod and crankshaft which are in motion. The counterweight on the rotor-crankshaft is designed to balance the eccentric mass of the slider-crank mechanism. However excessive vibration levels can be observed. The modal analysis of different components can be performed and determine the fundamental frequency of them. [6] discusses vibrations in reciprocating compressor. Vibration is an important criterion for assessing the safety, integrity and efficiency of compressor installation. Several international standards have been developed for this purpose. Existing standards are considered to be non-specific and do not distinguish between vibration levels of different parts of the reciprocating compressor system, such as foundations, frames, cylinders, pulsation dampers, and piping.For that reason the “EFRC Guidelines for Vibrations in Reciprocating Compressor Systems” were developed. As per this standard Acceptable limit for vibration for foundation bolt is 8.00 mm/s rms value. [7] this present The influence of modeling on the inherent mechanical frequency, the influence of inertial load on the structural vibration, and the influence of the crank drive mechanism damping on the speed fluctuation are proposed to ensure safe operation and improve the reliability of the reciprocating compressor.In this paper it is shown, that conventional way of modelling is not sufficient. The main dynamic loads of a reciprocating compressor are: vibrational forces due to pressure pulsations, inertia and gas forces, and additional forces caused by torsional vibration of the drive.[8] describes modal analysis. Modal analysis is a technique used to study the dynamic properties of structures with vibrational excitation. Modal analysis can be used to determine the eigenfrequency, mode shape and mode vector of the structure. Modal analysis allows the structure to avoid resonance or vibration at specific frequencies and to let engineers understand how the structure reacts to different types of dynamic loads. [9] Compressors are used in manufacturing household refrigerators, air-conditioners and heat pumps etc. Most of the household refrigerators The main parts include a cylinder, piston, connecting rod, crankshaft, cylinder head and valves. In this paper author did optimization using FEA solver. The crankcase is the central structure of the reciprocating compressor, and supports practically all the internal components of the compressor.[10] discusses The forces that cause the reciprocating compressor to vibrate come from different sources. These include unbalanced reciprocating forces and moments, piping pulsations and cylinder extensions. The vertical force acting through the guide rails seems to contribute to strong vibrations. Considering the effects of all forces in the design calculations, a reliable and efficient reciprocating compressor unit will be produced. In the design phase, the vertical force in the compressor cylinder nozzle must be considered.

## II. MATHEMATICAL EQUATIONS

$$\sin\varnothing = \sin\Theta / (l/r) \qquad (1)$$

$$\text{Stroke} = r\,(1-\cos\Theta) + l\,(1-\cos\varnothing) \qquad (2)$$

$$F_{GHE} = \pi/4 * D^2 * (P_{HE} - \text{Pambient}) \qquad (3)$$

$$F_{GCE} = -\pi/4 * (D^2 - d^2) * (P_{CE} - P_{ambient}) \qquad (4)$$

$$F_I = -(M_R * \acute{\omega}^2 * r/981) * [\cos\Theta + \{\cos 2\Theta/(l/r)\}]$$

$$F = F_{GHE} + F_I \qquad (6)$$

$$Q = F/\cos\varnothing \qquad (7)$$

$$F_N = Q * \sin\varnothing \qquad (8)$$

$$F_r = Q * \cos(\Theta+\varnothing) \qquad (9)$$

$$F_t = Q * \sin(\Theta+\varnothing) \qquad (10)$$

$$T = F_t * r \qquad (11)$$

## III.STATIC ANALYSIS

Static analysis is done on crankcase in ANSYS software to know the stresses coming on crankcase by doing this analysis we ensure the scope of optimization of crankcase. For doing this analysis forces on connecting rod is divided in two component i.e tangential component and radial component which is done on both side i.e low pressure side and high pressure side. Also gas forces on low pressure and high pressure side is considered. Dead weight of crosshead of both side is applied on their C.G. crankcase material is FG 260 having ultimate stress 260 MPa.

i.**forces considered for static analysis**

a) HP side(1) radial force…………..6888.2 N
b) HP side(1) tangential force……. 5003.9N
c) HP side(2) radial force………… 6888.2 N
d) HP side(2) tangential force……. 5003.9N
e) LP side radial force……………..15239 N
d) LP side tangential force……….. 11287N
e) HP side dead weight…………… 2632.8N
f) HP side gas force………………. 1807.9N
g) LP side dead weight…………….3302.1N
h) LP side gas force………………..2256.5N
i) Belt force……………………….. 3804.5N
j)Flywheel weight………………… 1225N

**ii.Flow chart of static analysis**

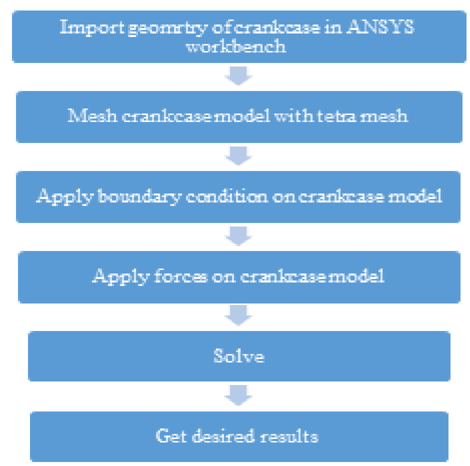


***Fig1. Flow chart***

**iii.analysis setup and geometry of crankcase in ANSYS**

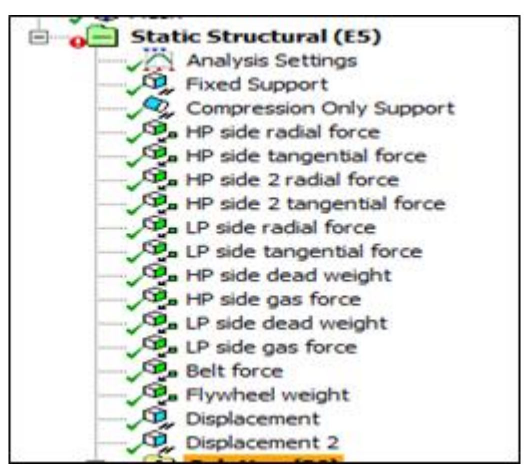


*Fig.2* ***Applied forces***

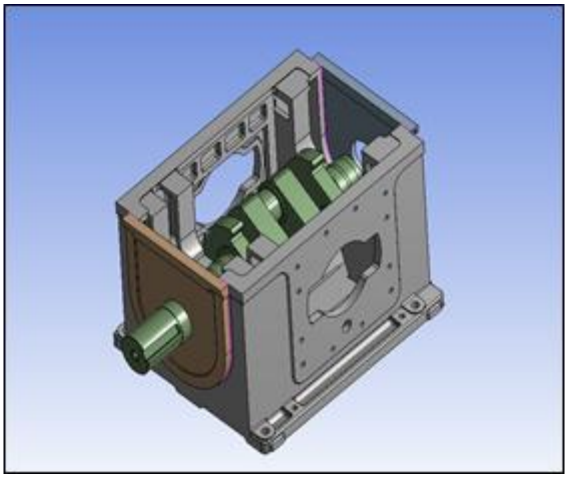

*Fig.3 Crankcase geometry in ANSYS*

Geometry of crankcase and crankshaft is imported in ANSYS and radial forces, tangential forces are applied on crankshaft along with belt tension, dead weight, gas forces of both side i.e. high pressure and low pressure side of compressor. For doing the analysis meshing is done in ANSYS. Material for crankcase is FG 260 which is brittle hence evaluating results we considered maximum principle stresses. For meshing solid 187 element is used which has 10 nodes.

**iv.Mesh model and load model of crankcase**

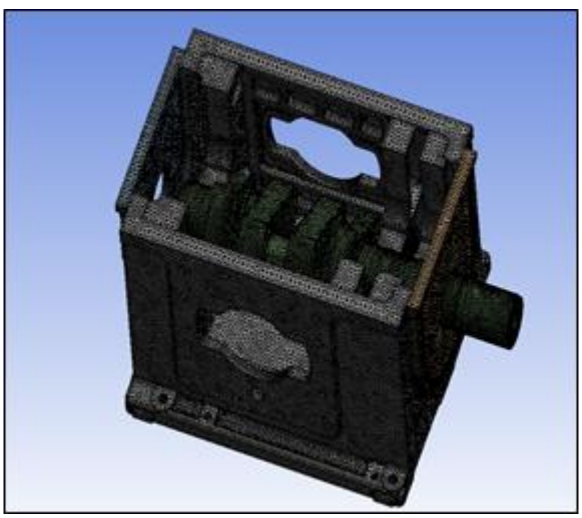

*Fig.4 Mesh model of crankcase*

To get better results fine meshing is done. meshing size is decided according to computation time and processing power available.

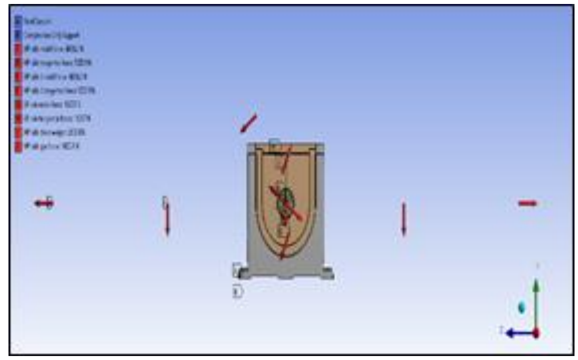

*Fig.5 Load model of crankcase for static analysis*

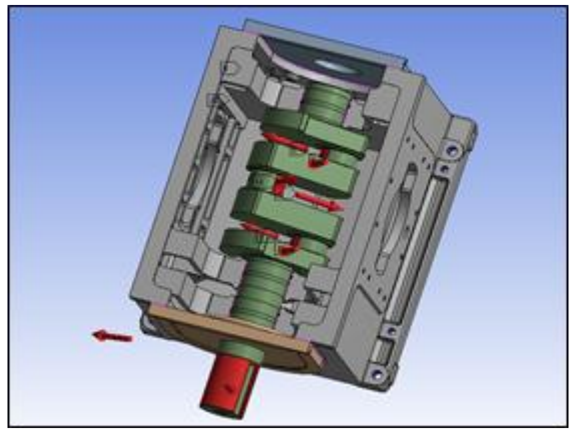

*Fig.6 loads on crankshafts*

**v. Result of static analysis**

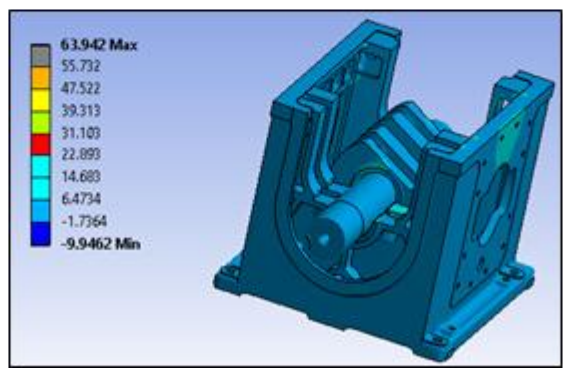


*Fig.7 Maximum principle stress contour*

Maximum principle stress in crankcase by applying forces mentioned above is 10 MPa which is less than ultimate stress value for crankcase material ( FG 260 )

## IV. HARMONIC ANALYSIS

Harmonic analysis is a branch of science mathematics concerned with the representation of function or signal as a superposition of basic waves. It has become a vast subject with application in areas as diverse as signal processing, tidal analysis, neuroscience. It is also used in various field of mechanical engineering to analyse vibrations. In reciprocating compressors gas forces and inertia forces are main caused of vibrations which are repetitive in nature.

These forces varies with angle of rotation of crankshaft. Vibration in reciprocating compressor is mainly due to the periodic unbalance forces. Vibrations are mainly high in BOP ( balance opposite piston ) compressor as compared to zero couple compressor. Compressors are checked at different positions for vibration and depending upon this vibration values compressor gets its quality test certificate. Locations for vibration check includes. Vibration is an important criterion for assessing the safety, integrity and efficiency of compressor installation. Several international standards have been developed for this purpose. Existing standards are considered to be non-specific and do not distinguish between vibration levels of different parts of the reciprocating compressor system, such as foundations, frames, cylinders, pulsation dampers, and piping.For that reason the “EFRC Guidelines for Vibrations in Reciprocating Compressor Systems” were developed. In this case compressor is running with 886 RPM i.e. 14.86 Hz. we considered velocity in mm/s for vibration measurement. As per EFRC standard, for foundation The vibration levels given are valid for rigid mounted compressor systems. This means that the compressor must be mounted directly to the concrete foundation. If the compressor is mounted on a skid, the skid must be stiff enough and directly mounted to the concrete foundation. Isolated mounted foundations e.g. concrete block on springs and skids on anti-vibration mounts (AVM’s) are an exception and the vibration levels for such systems should be agreed upon with the customer. For this analysis meshing is done with solid 187 element which has 10 nodes.

### i. flow chart of harmonic analysis

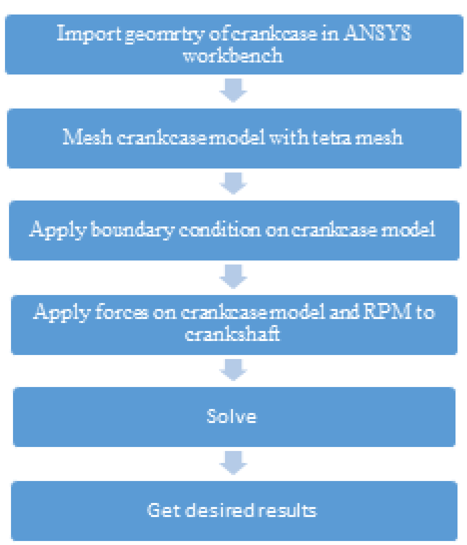


***Fig.8 flow chart***

### ii.forces considered for static analysis

a) HP side(1) radial force…………..6888.2 N
b) HP side(1) tangential force……. 5003.9N
c) HP side(2) radial force…………. 6888.2 N
d) HP side(2) tangential force……. 5003.9N
e) LP side radial force……………..15239 N
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f) HP side gas force………………. 1807.9N
g) LP side dead weight…………….3302.1N
h) LP side gas force………………..2256.5N
i)RPM ……………………………..886
j) Belt force……………………… 3804.5N

### iii.analysis setup and support in ANSYS

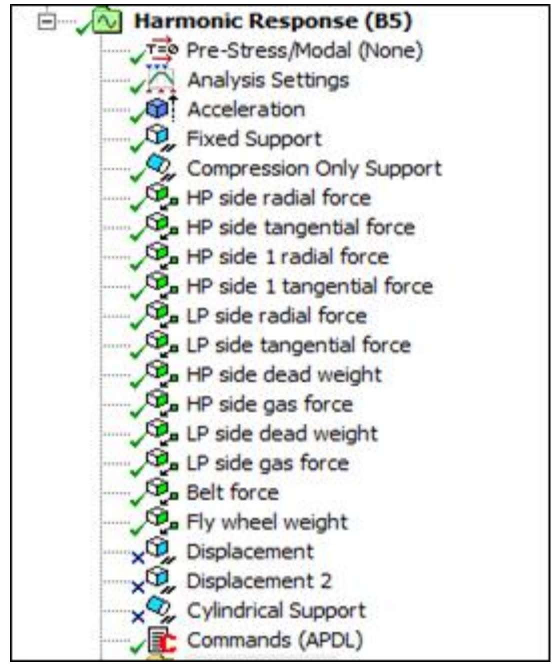


***Fig.9 Applied forces***

All forces mentioned are applied on crankshaft also fixed support is given at bolting position at bottom of crankcase. Compression only support is given at crankcase base to simulate actual condition compression only support allow crankcase to move upward.

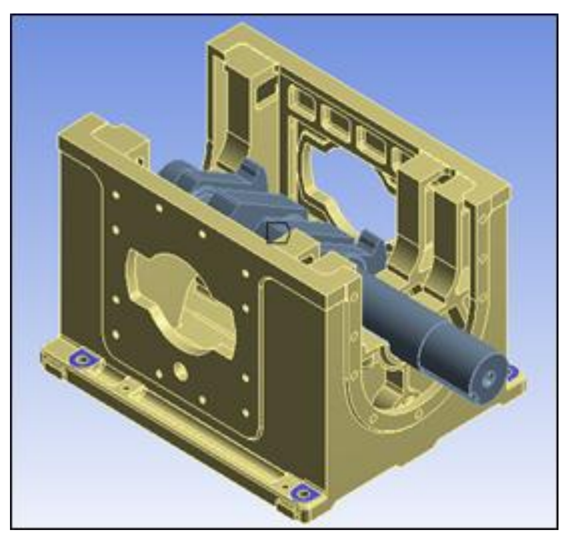

***Fig.8 Fixed support***

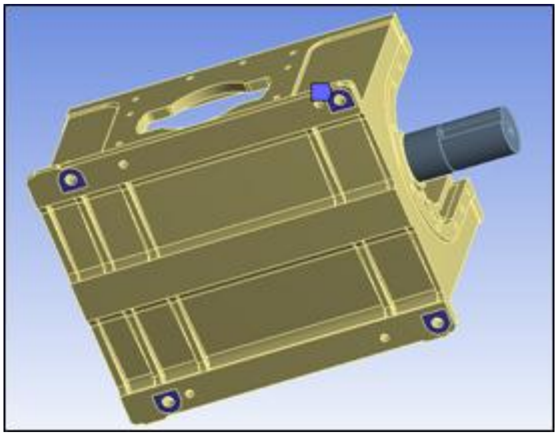

***Fig.10 Compression only support***

**iv. Mesh model and load model of crankcase**

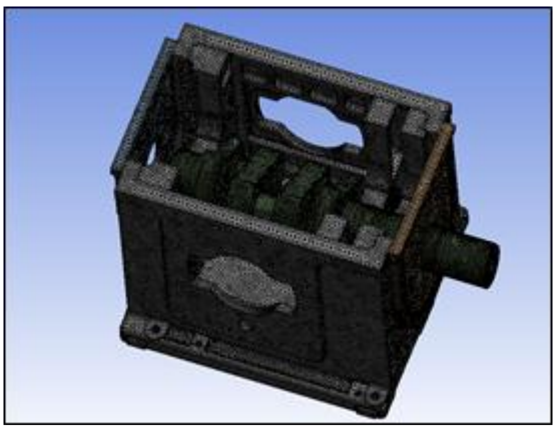

***Fig.11 Mesh model***

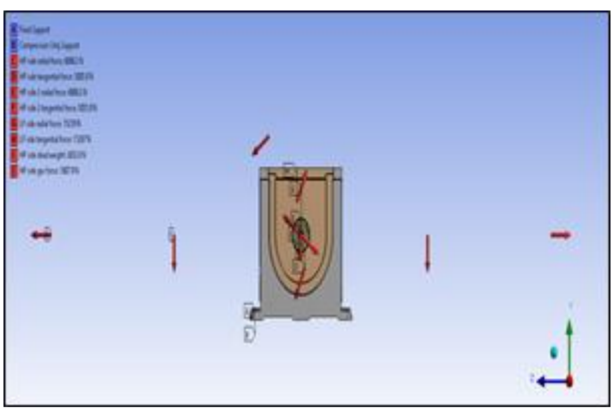

***Fig.12 load model of crankcase***

**v. Location of result**

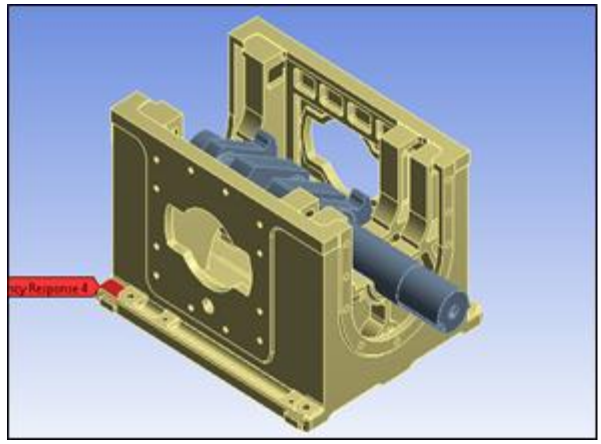

***Fig.13 Result location***

**vi. Result of harmonic analysis**

The harmonic analysis result is taken at surface show in fig 12. vibrations are measured in term of velocity. In this case compressor RPM is 886 i.e. 14.86 Hz. For harmonic analysis result form 0 Hz to 100 Hz are taken with 10 Hz interval and graph is plotted which is shown in fig 13

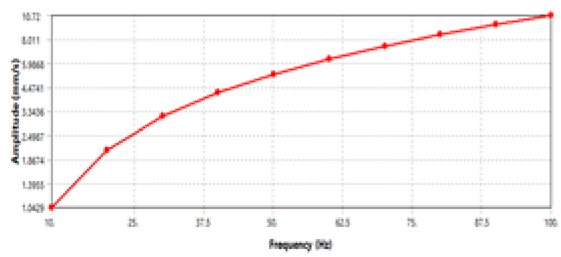


***Fig.14 Amplitude graph***

## V. TRANSIENT ANALYSIS

In this analysis along with crankcase, crosshead guide, crosshead, connecting rod, crankshaft is taken to carry transient analysis. Crankshaft is rotating at 886 PRM. Crankcase is fixed at bolting location also crosshead guide of both stages that is stage-I and stage-II is connected to crankcase with bonded contact. At low pressure side forces of each $10^0$ crank-angle are applied on crosshead . At high pressure side forces of each $10^0$ crank-angle are applied on crosshead. Main force generated by reciprocating compressor is gas forces and inertia forces in this case compressor is two stage double acting in such compressor each cylinder of each stage is double acting that is if suction is happing in head end side at same time compression is happing at crank end side. This phenomenon is common in each stag (Stage-I and stage-II).Head-end side compressor takes place simultaneously at both stage that is when In stage-I compression is happing at head end at same time compression is happing in second stage at head end, at the same time in both stage at crank end suction happens. In this analysis except crankcase all other components are kept rigid and crankcase as flexible. For this analysis meshing is done with solid 187 element which has 10 nodes.

**i. flow chart of transient analysis**

**ii .Mesh model and load model of crankcase**

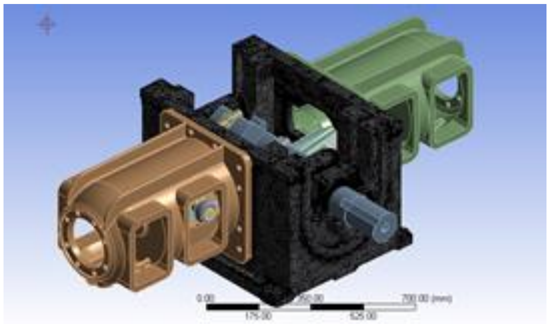

***Fig.15 Mesh model of crankcase***

For meshing curvature algorithm is used which gives capability to capture small edges and corner more efficiently than other algorithm available in ANSYS. Face meshing is given on all faces which is contact with each other as well as in relative motion element size of face meshing is 5 mm.

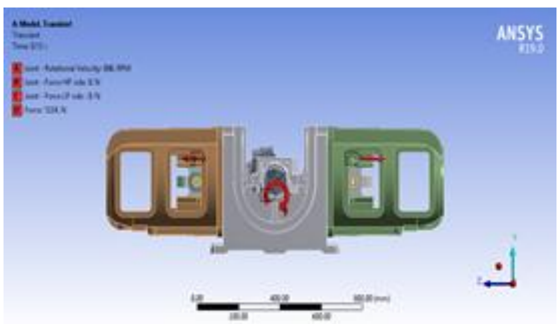

***Fig.16 Load model of crankcase***

**iii.Result of transient analysis**

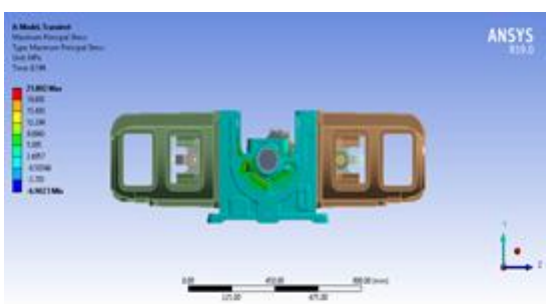

***Fig.17 Maximum principle stresses***

Crankcase material is FG 260 as it is brittle material maximum principle stress are evaluated in ANSYS software. Stress contour of maximum principle stress are shown in fig… Which shows stresses are in between 15 MPa to 19 MPa. Stress coming in crankcase shows that there is scope for optimization in terms of weight. Existing weight of crankcase is 341.29 kg.

## VI. TRANSIENT ANALYSIS OF OPTIMIZED CRANKCASE

Existing crankcase have wall thickness of 14 mm to optimize crankcase for weight reduction wall thickness is reduced to 10 mm. By doing this wall thickness reduction crankcase weight is 305.64 Kgs which is 36.6 Kg less than existing crankcase Replacing the geometry in previous Transient analysis and keeping all other factor such as forces that act on cross head of low pressure side, forces that act on cross head of high pressure side, meshing parameter, flywheel weight, joint velocity. Replaced geometry consist of crankcase with thickness 10 mm. For this analysis meshing is done with solid 187 element which has 10 nodes.

**i. flow chart of transient analysis**

Transient analysis is done after replacing the precious crankcase geometry with new geometry having wall thickness of 10 mm. transient analysis is performed with same load steps and boundary condition as previous.

**i.Result of transient analysis**

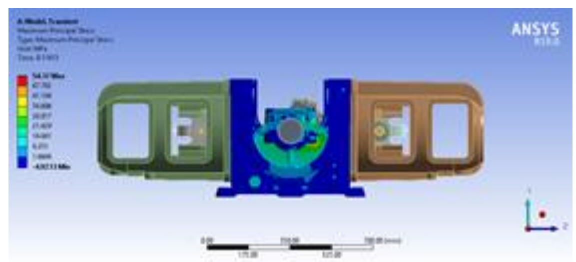

***Fig.18 Maximum principle stresses***

By solving new crankcase geometry maximum principle stress are evaluated. Maximum principle stress value for optimized crankcase is 30 MPa to 40 MPa.

## VII. HARMONIC ANALYSIS OF OPTIMIZED CRANKCASE

Harmonic analysis is done on optimized crankcase. . In this case compressor is running with 886 RPM i.e. 14.86 Hz. we considered velocity in mm/s for vibration measurement.

**i. forces considered for static analysis**

a) HP side(1) radial force…………..6888.2 N
b) HP side(1) tangential force……. 5003.9N
c) HP side(2) radial force…………. 6888.2 N
d) HP side(2) tangential force……. 5003.9N
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h) LP side gas force………………..2256.5N
i)RPM ……………………………..886
j) Belt force……………………… 3804.5N

**ii. Result of harmonic analysis**

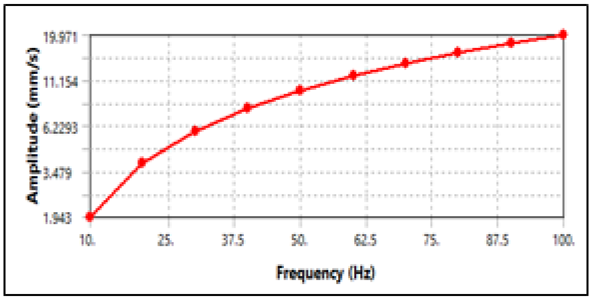


***Fig.19 Amplitude graph***

## V.CONCLUSION

Static analysis is performed on crankcase and maximum principle stresses are evaluated as crankcase material is FG 260 which is brittle . maximum principle stresses are 10 MPa and which is less than ultimate stress for FG 260 so there is scope for optimization. From harmonic analysis Vibration value of 1.8 mm/s is obtained at location shown in fig 13 which is close agreement with experimental results of vibration test. For optimized crankcase vibration at foot of crankcase is 2.9 mm/s which is within allowable limit given in EFRC. For optimization of crankcase wall thickness of crankcase is reduced form 14 mm to 10 mm which causes weight reduction of 35 Kg. original weight of crankcase is 341.24 Kgs which is reduce to 305.64 Kg. Weight reduction of 10.55 % is achieved in crankcase optimization

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