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DESIGN AND STANDARDIZATION OF BASE FRAME & ANT VIBRATION MOUNTS FOR BALANCED OPPOSED PISTON AIR COMPRESSOR

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Abstract – Now days, Compressors are widely used in gas gathering, gas processing, and gas storage, chemical and refining applications. Compressors are mounted on the Base frame to carry its weight, to maintain its alignment and to assist in carrying the dynamic loads. Compressors base frame needs an effective design technology to ensure its required performance and functions satisfactorily. This paper represents a case study of the compressor base frame on which high speed reciprocating compressors are mounted. Attempts are basically made to standardize base frame for all type of piston compressor. Also, to design deck systems with effective vibration control, which able to sustain the dynamic forces occur in compressor and motor. Secondarily, anti vibration mounts can be select and arranged as they control the vibration of motor and compressor. By using FEA tool approach is made for the investigations of critical stresses. Also, to plot the harmonic responses of the base frame for acceleration with which it vibrate. Finally compare it with the standard data.

Keywords – *dynamic load, anti-vibration mounts, free vibration response, harmonic response.*

I. INTRODUCTION

Most of the larger high speed compressor models are mounted on Base frames. Compressors are mounted on the Base frame to carry its weight, to maintain its alignment and to assist in carrying the dynamic loads which every compressor generates. Compressors base frame needs an effective design technology to ensure that the base frame as designed performs the required functions, and maintains its integrity. There is also a need to maximize the life of the compressor base frame under the loads to which it is exposed. The base frame (skid) is generally made up of the standard beam or channel sections [1] [2] [3].

A.J. Smalley has given the idea about various loads coming on the compressor base frame and to the foundation block. John P. Harrell has given the idea about the advantages of the skid mounted compressor. The primary advantage of skid mounting is portability and the ability to perform all the necessary integration of engine and compressor. Simon S.Hill and Scott D.Snyder have described the design of vibration absorber using FEA to reduce structural vibration at multiple frequencies with enlarged bandwidth K.Nagaya, A.Kurusu and S.Ikai developed a method of vibration control of structure by using the vibration absorber without damping.

II. DRAWBACKS OF EXISTING MODEL

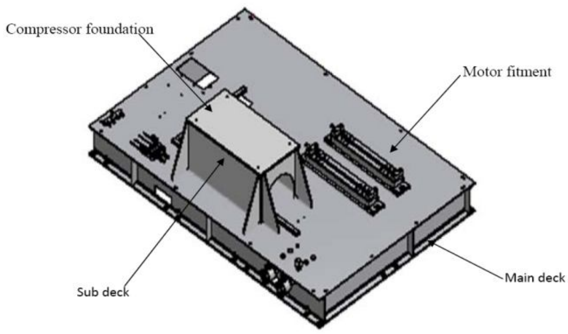


Figure.1 Existing base frame (skid) model.

- Organization is having different models and for each model different base frame design is required.
- Alignment of motor and fitment of motor takes more time while assembling the packaging.
- There is no vibration isolation system available in each existing design.
- In the existing design it is difficult to mount inter cooler and after cooler with the other fitment also.

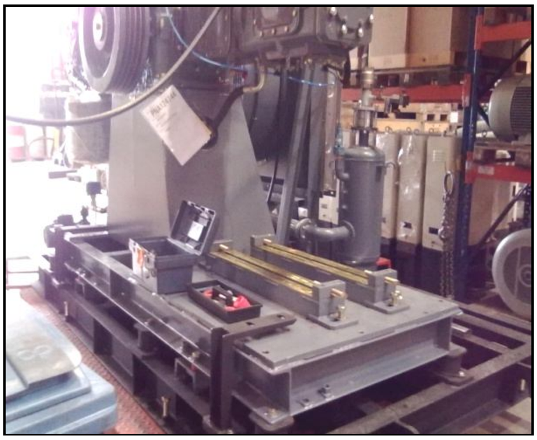


Figure 2. Different type of base frames

III. PROCEDURE FOR BASE FRAME DESIGN

A. Skid Design

TABLE I. Classification of Carbon Steel[16]

Low-Carbon Steel	0.05% to 0.30% carbon
Medium-Carbon Steel	0.30% to 0.45% carbon
High-Carbon Steel	0.45% to 0.75% carbon
Very High-Carbon Steel	0.75% to 1.70% carbon

By observing the material properties for application of skid mild steel is used [16].

TABLE II Material properties used for skid design

Material used	M.S IS:2062 Gr.A
Young's Modulus	210,000 MPa
Shear Modulus	79,000 MPa
Density	7,850 kg/m3
Yield limit	230 MPa

B. Anti-vibration mount selection

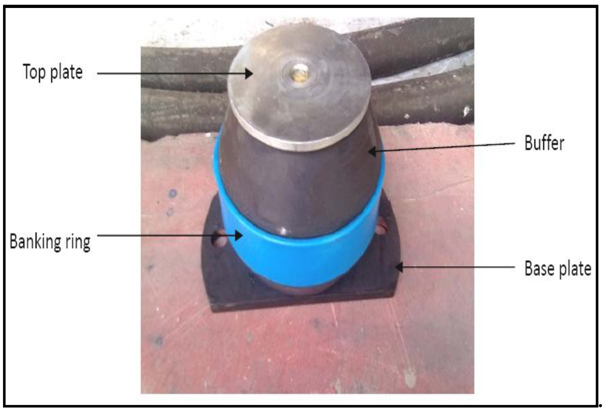


Figure 3. Elastomeric Mount

TABLE III Anti vibrations mount properties [5]

Material used(Buffer)	Oil resistant natural rubber
Static load	710Kg
Dynamic load	1065Kg
Shore Hardness(shore'A')	50-55
Static deflection	20 mm

Axial Stiffness= Static load/Static deflection

=710*9.81/20=348.255 N/mm

The transverse stiffness in other two directions are approximated as 10% of longitudinal stiffness i.e. 34.825 N/mm[13].

C. Load calculations

TABLE IV Weights of different components including the weight of skid

Name of the component	Magnitude	load specification applied as point load
Compressor	2756	on sub deck
Control panel	166	on main deck
Motor	1160	on sub deck
Component side fitted	602	on sub deck
Self weight of sub deck	1250	on main deck
Inter cooler	152	on sub deck
After cooler	180	on main deck
Belt of flywheel	80	on sub deck
Filter	232	on sub deck

TABLE V. Magnitude of dynamic loads.

Sr. no	Name	Magnitude	Load specification
1	Swaying Couple	279 kg-m	Applied at the C. G. of compressor about Z-axis
2	Galloping Couple	324.5 kg-m	Applied at the C. G. of compressor about Y-axis
3	Turning Coupling	265.3kg-m	Applied at the C. G. of compressor about X-axis

IV. SOLID MODELING

Below figure 4 shows the 3-D assembled model

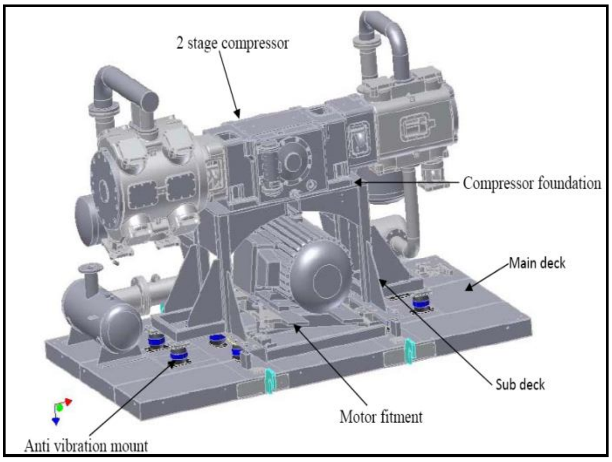


Figure 4. Modified solid model of base frame

Below figure 5 indicate the 3-D model of sub deck

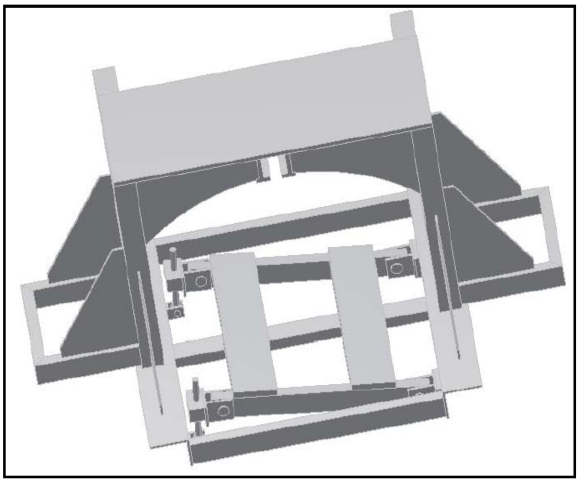


Figure 5 . Developed model of sub deck

Case-I

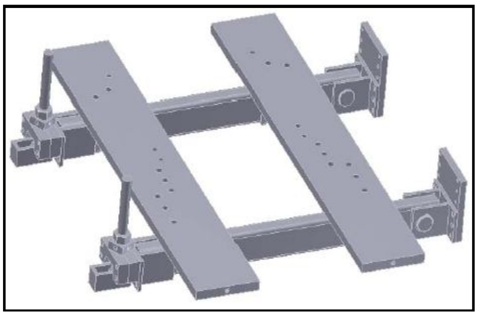


Figure 6 . Developed old model lifting arrangement

Case-II

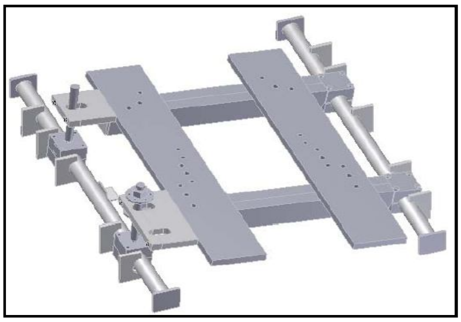


Figure 7 : Developed modified model lifting arrangement

V. FEM approach

A. Meshing-

Basic theme of FEA is to make calculations at only limited number of points and then interpolate the results for entire domain i.e. surface or volume. Any continuous object has infinite degrees of freedom and it's just not possible to solve the problem in this format. Finite element method reduces degrees of freedom from infinite to finite with the help of discretization i.e. meshing. Here for meshing main deck of compressor solid 186 element used. SOLID186 is a higher order 3-D 20-node solid element that exhibits quadratic displacement behavior [14] [18].

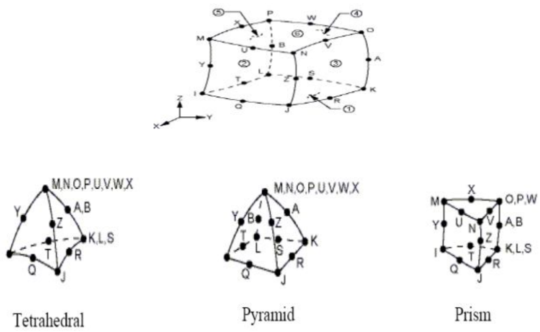


Figure 8.Solid 186 structural solid geometry [14]

B. Free vibration Analysis-

If we disturb any elastic structure in an appropriate manner initially at time $t = 0$ (i.e.by imposing properly selected initial displacements and then releasing these constraints), the structure can be made to oscillate harmonically [18] [9]. Consider example of two degree of freedom system.

Equation of motion is shown as follows.

$$M\ddot{x} + kx = \begin{bmatrix} m1 & 0 \\ 0 & m2 \end{bmatrix}\begin{bmatrix} \ddot{x_1} \\ \ddot{x_2} \end{bmatrix} + \begin{bmatrix} (k_1+k_2) & -k_2 \\ -k_2 & (k_2+k_3) \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (1)$$

$$x = \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} e^{i\omega t} = Xe^{i\omega t} \quad (2)$$

X from the eq.no.2 is known as the systems Eigen vector or Eigen mode which gives mode shapes.

$$[K - \omega^2 M]X = \begin{bmatrix} (k_1+k_2) - \omega^2 m_1 & -k_2 \\ -k_2 & (k_2+k_3) - \omega^2 m_2 \end{bmatrix}\begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (3)$$

$$det[K - \omega^2 M] = (k_1 + k_2 - \omega^2 m_1)(k_2 + k_3 - \omega^2 m_2) - k_2^2 = 0 \quad (4)$$

The roots of this quadratic (ω) as per eq.no.4 are the natural frequency of the system, known as the Eigen values.

Operating frequency of compressor=24.66 Hz

Case-I

TABLE VI Frequency and mode shape number

Sr. No	Mode	Frequency(Hz)
1	1	45.127
2	2	65.834
3	3	93.621
4	4	99.382
5	5	102.32
6	6	109.69

Case-II

TABLE VII Frequency and mode shape number

Sr. No	Mode	Frequency [Hz]
1	1	65.678
2	2	106.61
3	3	115.06
4	4	116.84
5	5	152.85
6	6	154.65

Here we observe by comparing Case-II to case –I,in case-II natural frequency is away from operating frequency.Hence Case-II is more benifitial.

C. Stress analysis

Load conditions are used as shown in the table IV and V[6][9]

Case-I

TABLE VIII Magnitude of maximum stress and maximum deformation.

Sr. No	Load case	Deformation (mm)	Maximum accountable stress (N/mm^2)	Yield limit (N/mm^2)	Factor of safety
1	Static_1g +dynamic couple	20.769	149.86	230	1.53(>1)

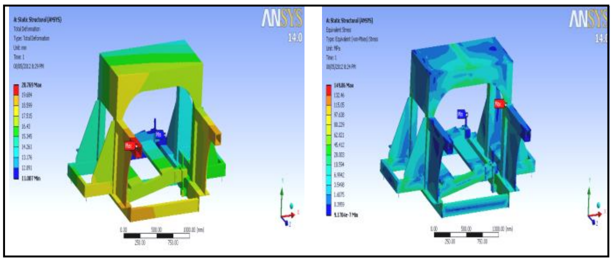


Figure 9.Total deformation and stress plot

Case-II

TABLE IX Magnitude of maximum stress and maximum deformation

Sr. No	Load case	Deformation (mm)	Maximum accountable stress (N/mm^2)	Yield limit (N/mm^2)	Factor of safety
1	Static_1g +dynamic couple	23.311	131.80	230	1.74(>1)

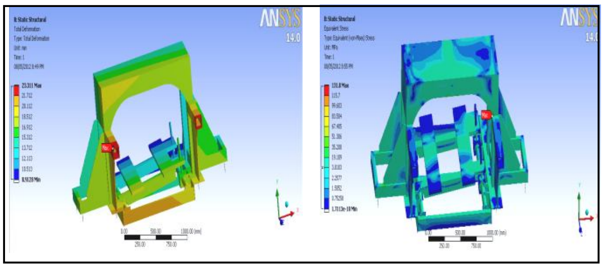


Figure 10.Total deformation and stress plot

D. Harmonic response analysis

When the excitation does not change with time, the solution is a steady state response at the operating frequency. This is known as frequency response analysis [18].

The equation of motion for damped single degree of freedom system follows the eq.no.5

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin \omega t \tag{5}$$

$$\ddot{x} + 2\varepsilon\omega_n\dot{x} + \omega_n^2 x = \frac{F_0}{m}\sin \omega t \tag{6}$$

Complete solution to the under damped system is found as follows shown in eq.no.7

$$x = Ce^{-\varepsilon\omega_n t}\sin(\omega_d t + \varphi) + \frac{F_0}{k}A\sin(\omega t - \emptyset) \tag{7}$$

Magnification factor and phase angle of the steady state solution given by following equation.

$$A = \left\{\left[1 - (\frac{\omega}{\omega_n})^2\right]^2 + \left[2\varepsilon\frac{\omega}{\omega_n}\right]^2\right\}^{-1/2} \tag{8}$$

$$\emptyset = \tan^{-1}\left[\frac{2\varepsilon\frac{\omega}{\omega_n}}{1 - \left[\frac{\omega}{\omega_n}\right]^2}\right] \tag{9}$$

Making use of following formulation shown in eq.no.10 is known direct frequency response analysis.

$$[-\omega^2 M + i\omega C + K]u(\omega) = F(\omega) \tag{10}$$

This formulation gives modal frequency response analysis.

$$[-\omega^2 MX + i\omega CX + KX]\varepsilon(\omega) = F(\omega) \tag{11}$$

Following figure 11 shows point locations where frequency response curve plotted

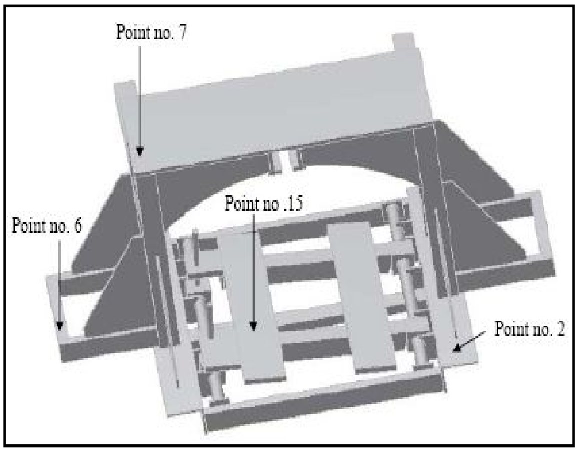


Figure11. Location of points for which graph plot

Case-I

Following figure shows frequency response curve at various points in X, Y, Z directions respectively.

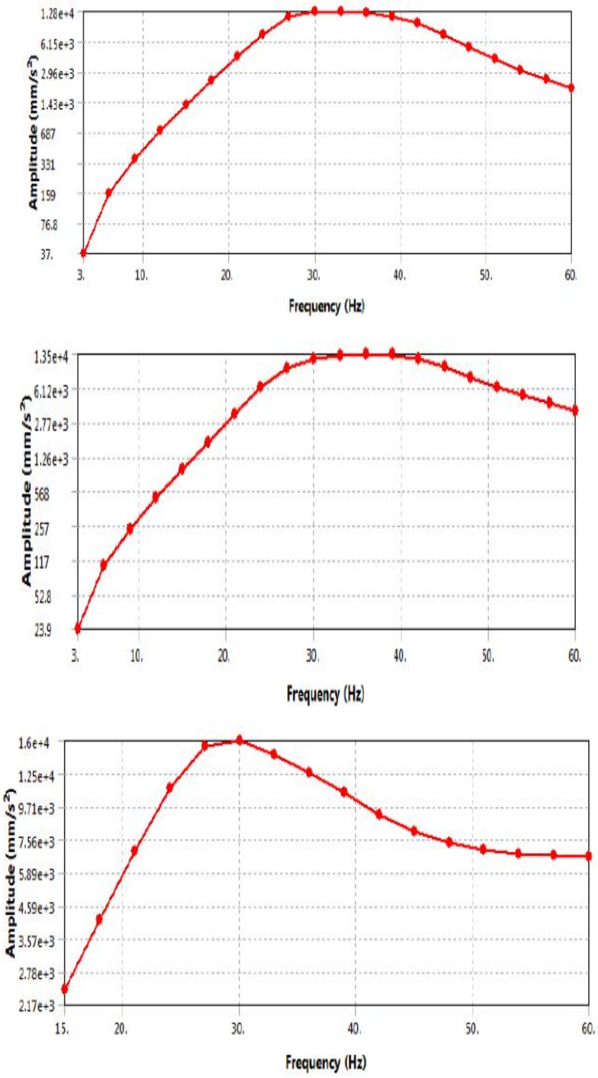
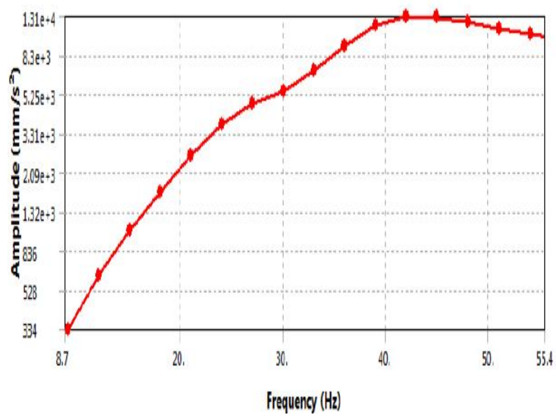
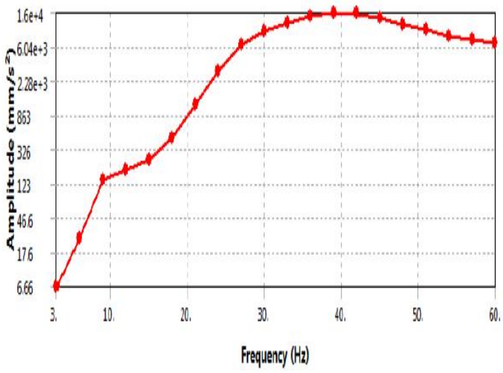


Figure.12 Frequency response at point number 2





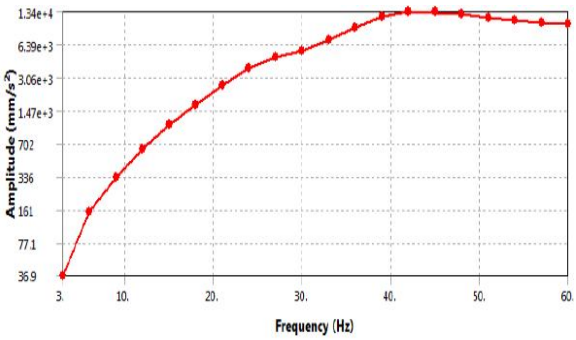
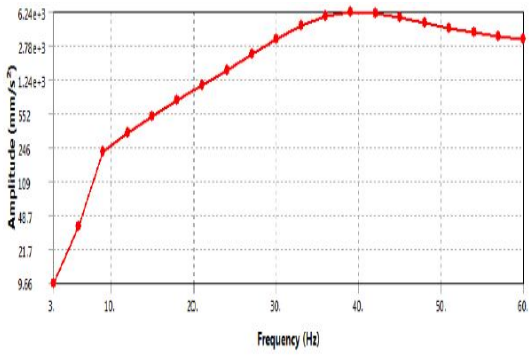
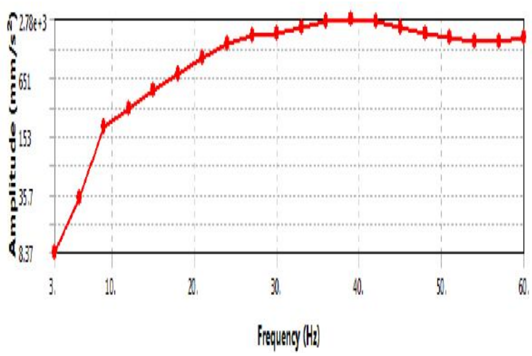


Fig.5.33.Frequency response at point 7 in Z-direction

Figure.13Frequency response at point number 7

Case-II





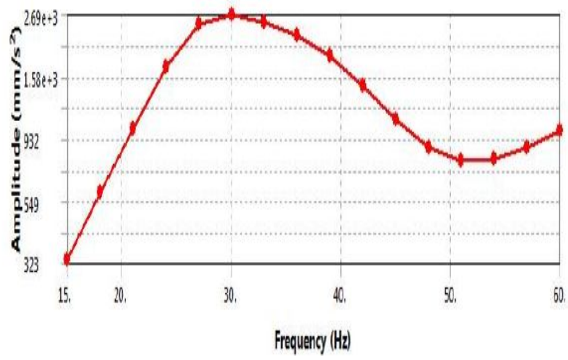
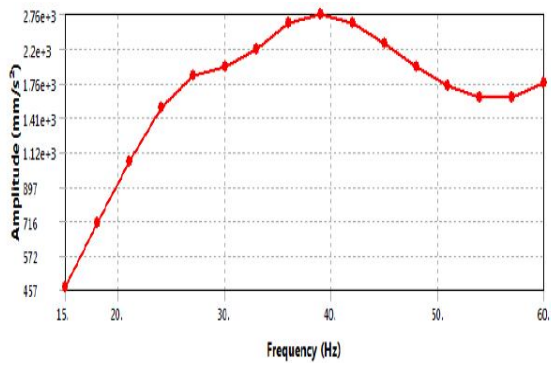
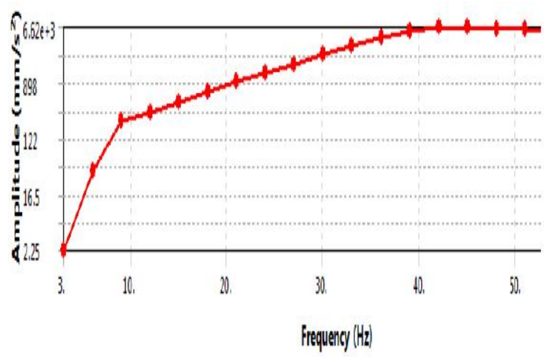


Figure.14 Frequency response at point number 2





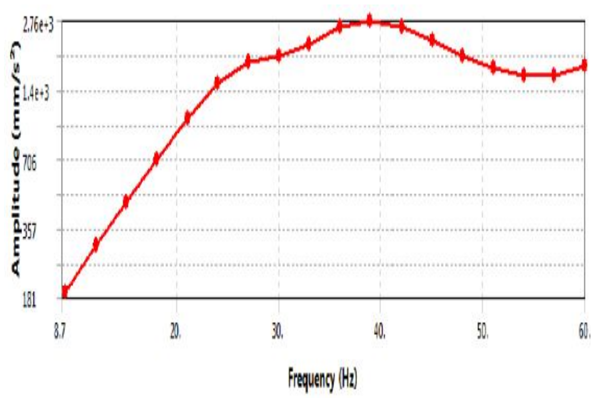


Figure.15Frequency response at point number 7

TABLE X. Acceleration results for case-I and case-II at various points using FEA.

Point no	Axis direction	Case-I mm/s^2	Case-II mm/s^2
2	x	6180	1569
	y	6980	1502
	z	7414	1715
6	x	7850	1792
	y	8688	1124
	z	9294	1806
7	x	3706	1498
	y	3084	1303
	z	3771	1498
15	x	11196	2288
	y	12690	2504
	z	14611	2553

VI. EXPERIMENTATION

International standards ISO 2372, BS 4675 and others recommend measurement of vibration severity (velocity in mm/s RMS) as most suitable for industrial purposes.VIB-10 vibrameter is a sturdy, battery powered instrument, based on ISO standards used for testing purpose here.

TABLE XI Acceleration results for case-I and case-II at various points using testing equipments.

Point no	Axis direction	Case-I mm/s^2	Case-II mm/s^2
2	x	6662	1549
	y	6817	1436
	z	7901	1800
6	x	8057	1918
	y	9141	1239
	z	8676	1874
7	x	3563	1444
	y	3408	1227
	z	3873	1637
15	x	12390	2376
	y	11310	2575
	z	15490	2634

VII. SUMMARY AND CONLUSION

- Observing result and discussion of free vibration analysis for case-I and case-II it seems that case-II is more comfortable than case-I and existing base frame at pre design criteria.
- Both models are suitable at design criteria for minimum deformation and maximum stress. The stress observed in case-I 149.86 N/mm^2 which is far away from the yield limit and factor of safety obtain 1.53 greater than 1 so design is safe. Also in case-II stress observed 131.80 N/mm^2 which is far away from the yield limit 230 N/mm^2 and factor of safety obtain 1.74 greater than 1 so design is safe. Hence case-II comparatively better than old model.
- As per ISO 10186 part-6 vibration reading at compressor bed not exceeds the limit of 17.88mm/s. The results obtained in experimental and analysis work both compared with ISO standard. While observing table 'X' and table 'XI' case-II full fill the requirements of vibration criteria.
- Anti-vibration mounts are used for each assembly that's why it help to reduce the vibration going up to main deck and protect the other systems mounted near the compressor from damage by reason of vibration.

REFERENCES

[1] A. J. Smalley, J. S. Mandke, P.J.Pantermuehl, and R.D.Drummond"Reciprcating Compressor Foundation: Loading, Design, Analysis, and Mounting & Repair". GMRC, Report, Mechanical and fluid engineering division, South west Research Institute, Dec-1999, TA 93-1.

[2] John.Pete.Harrell,Danny.M.Deffenbaugh,Anthony.J.Smalley,Robert.A.Hart"Advanced reciprocating compression technology",U.S.department of energy, National energy technology,Dec-2005,Projet no.-18.11052,Doe. no.-DE-FC26-04NT.

[3] Simon S.Hill, Scott D.Snyder, "Design of an Adaptive Vibration Absorber to Reduce Electrical Transformer Structural Vibration", Journal of Vibration and acoustic, Vol.124, Oct.2002, pp-606-611.

[4] K.Nagaya, A.Kurusu, S.Ikai, "Vibration Control of a Structure by using A tunable Absorber And An optimal Absorber under Auto Tuning Control", Journal of Sound and Vibration, 1999, 228(4), pp-773-792.

[5] Yunhe Yu , Nagi G. Naganathan , Rao V. Dukkipati, " A literature review of automotive vehicle engine mounting systems", Mechanical, Industrial and Manufacturing, Engineering Department, The University of Toledo,OH

43606, USA, Mechanism and Machine Theory 36 (2001) 123-142.

[6] Naiuan Ye and Torgeir Moam, Static and fatigue analysis of three types of aluminum box stiffener/ web frame connection, International Journal of fatigue, 29(2007), 1426-1433.

[7] H.Moradi,F.Bakhtiari,M.R.Movahhedy, "Tunable Vibration Absorber Design to Suppress Vibration "An application in boring manufacturing process, Journal of sound and Vibration,Vol-318,April2008,pp-93-108.

[8] E.C.Lee, C.Y.Nian, Y.S.Tang, "Design of Dynamic vibration absorber against vibration in operations", Journal of material Processing Technology, Vol-108, Issue3, January2001, pp-278-285.

[9] Grenland group, "Vibration analysis of gas lift compressor foundation",Jeovan publication third edition-2006.

[10] Dennis H. Shreve, "Introduction to vibration technology", Director of Marketing IRD Mechanalysis, Inc. Columbus, Ohio 43229 November 1994.

[11] L. Broerman,Benjamin A. White, P.E. j ,Jason T. Gatewood, "Pulsation and vibration control for reciprocating compressor",Gas Machinery Conference 2011.

[12] William D. Brant, Jack Moll ,Anthony J. Smalley,Shu-jin Fang, James P. Lee, "Foundations for Dynamic Equipment", Reported by ACI Committee 351, ACI 351.3R-04.

[13] Eberle, Kelly and Harper, "Dynamic Analysis of Reciprocating Compressors on FPSO Topside Modules", Chris Engineering Design Beta Machinery Analysis , 5th Conference of the EFRC March 21st / 23rd, 2007, Prague.

[14] J.N. Reddy, "An Introduction to the Finite Element Method", Tata Mcgraw Hill, 2nd edition-2003.

[15] N.S.Gokhale, S.S.Deshpande, S.S.Bedekar, A.N.Thite, "Practical finite element analysis",1st edition-2008.

[16] Paul C. Hanlon, " Compressor handbook", Library of Congress Cataloging-in-Publication Data, McGRAW-HILL-2001.

[17] Khurmi, R, " Theory of Machines", 14th ed.,S. Chand & Co. Ltd., New Dehli 2005; ISBN 9788121925242.

[18] Vince Adams,Abraham Askenazi, "Building better products with finite element analysis",On word press,2530 camino entrada santa fe, NM 87505-4835 USA.

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