Optimization of Reciprocating Compressors using Finite Element Method

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Abstract

The project aims at determination of optimum shell design by the consideration of the natural frequencies of different modified shells. The choice of selecting the compressor shell for low noise is based on the natural frequencies obtained. This is because the noise level of the compressor is basically dependent on its natural frequencies. The finite element reveals that, the natural frequencies are a function of material and stiffness of the body. Hence with the material remaining the same, the geometric of the shell is changed to have different modifications. The natural frequencies of each modification are determined by the modal analysis. This project discusses the process in which the reduction of compressor noise can be achieved by restructuring the compressor shell and decreasing the amplitude of vibration.

Keywords: Optimization, Reciprocating

I. Introduction

Generally, manufacturing household refrigerators, air conditioners and heat pumps etc. use compressors. Hermetically sealed reciprocating compressors are used in household refrigerators. Hermetically sealed reciprocating compressors are Piston cylinder type of "pump".

The compressor consists of cylinder, crankshaft, piston, connecting rod, valves and cylinder head.

The compressor is used to remove the refrigerant. The refrigerator is removed from the evaporator and send to condenser. The condenser is used to cool the refrigerator at a temperature and pressure. During this process, the refrigerator loses its latent heat and converts to liquid.

Refrigerators and air conditioning units produce more noise and vibrations. These vibrations and noise are caused due to compressors. This is because of compressors undergoes severe compression process. The noise of hermitically sealed refrigeration compressor is transferred to surroundings due to direct radiation through the hermitic shell and vibrations through supporting structures.

There are three main groups (classifications) of compressors:

– Reciprocating compressors
– Rotary compressors
– Centrifugal compressors.

II. Finite element modelling of shell

A. *Pre-processor*

The geometric model of the shell, which is modeled in the Unigraphics, was exported using parasolid translator. The following figure shown below is the imported model of the compressor shell.

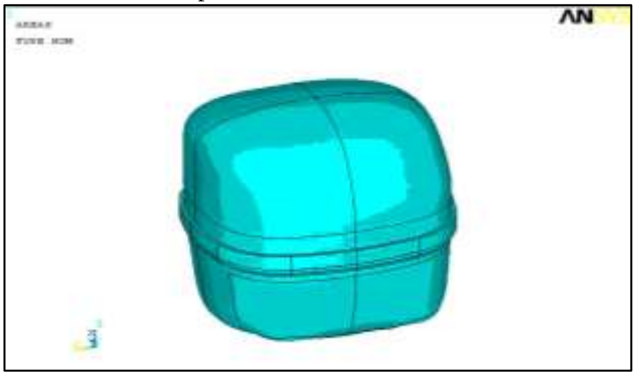

Fig. 1: Shows Compressor Shell

B. Material Properties

- Young's Modulus = 2.07E11Pa
- Poisson' ratio = 0.29
- Density = 7850 Kg/m^3
- No of nodes for element: 4

C. Mesh Generation

Finite element mesh is generated using shell mesh with element size 3mm. The meshed view of the shell is shown in the fig below. While meshing care has been taken to keep minimum 6 elements per wave length which to be calculated as Wave length (λ) = velocity of sound (C) / frequency (f). Where 'f' is the maximum frequency of interest

- In this case f = 5000 Hz
- Velocity of sound for steel/iron metal is 5.1 KM/sec
- Wave length (λ) = 5100/5000 = 1.02 m
- Maximum element length = 1.02/6 = 170 mm
- Element length maintained in analysis = 3 mm

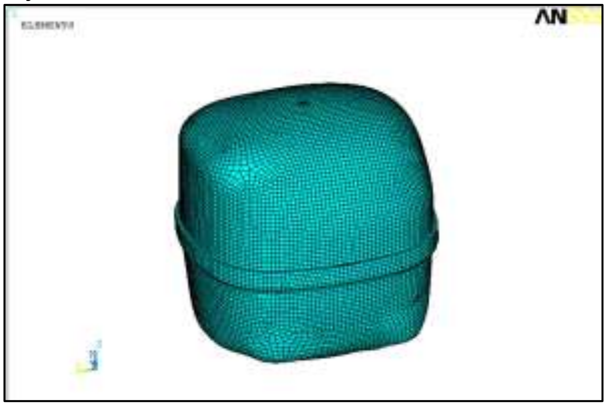

Fig. 2: Shows Finite Element Model of the Shell

D. Boundary Conditions

The structural displacement load is applied on the nodes of the welded areas of the bottom side (base) shell. That is the welded areas are fixed by assigning all degrees of freedom constrained at those locations. The constrained shell is shown in fig below.

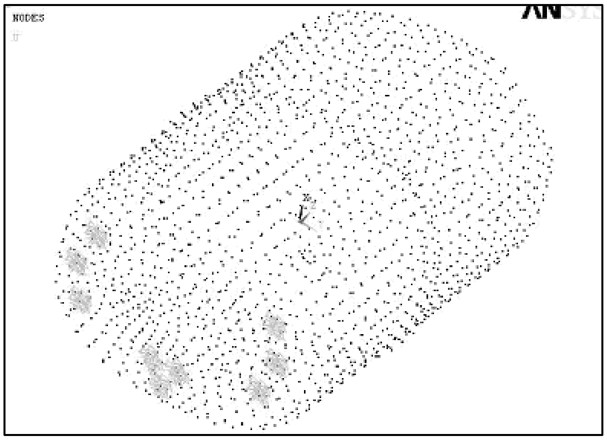

Fig. 3: Boundary Conditions

The Finite Element Model of the shell, will be used for performing modal analysis and power spectrum density analysis in the later coming chapters.

III. METHODOLOGY OF MODAL ANALYSIS

A. Modal Analysis Procedure

The Finite Element model of the compressor Shell generated as mentioned in previous chapter is analyzed for the natural frequencies and the corresponding mode shapes are obtained from modal analysis. The step by step procedure for the modal analysis of the compressor shell is described below.

1) Specify Modal Analysis

Using Analysis options specify Modal Analysis and then click 'OK'.

ANSYS Main Menu> Solution > New Analysis

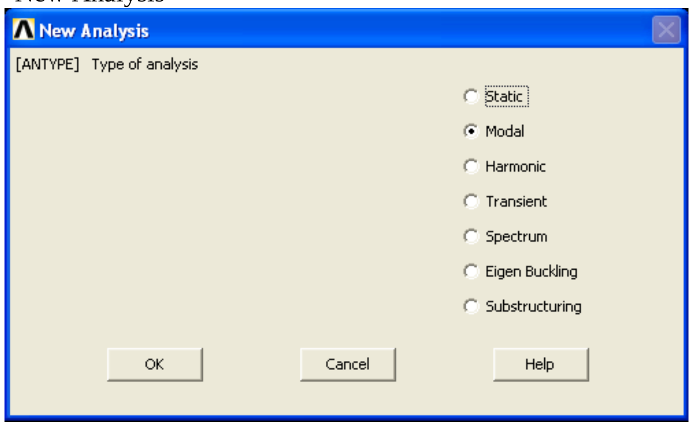

Fig. 4: Modal Analysis Procedure

B. Specify Analysis Options

ANSYS Main Menu> Solution> Analysis Options

Using analysis option under Modal Analysis.

- Specify mode expansion method by picking up "Block Lanczos method". Because it is preferred one and best choice.
- Specify 15 for the "No of modes to expand" option.
- Specify "expand mode shapes".
- Specify "calculate element results".
- And then click 'OK'.

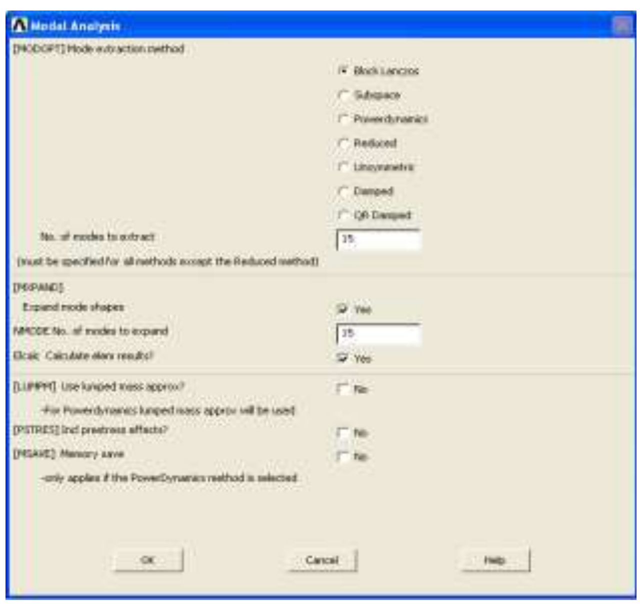

Fig. 5: Specify Analysis Options

On clicking 'OK' in the above window, the following Block Lanczos method options window will be displayed. In this window, the values shown are the default values. All default values are accepted for this Modal Analysis. So accept & click 'OK'.

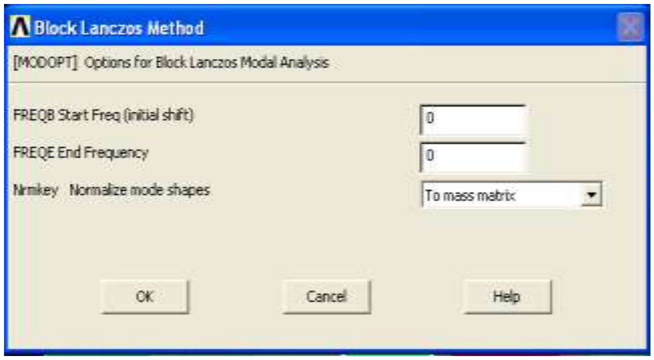

Fig. 6: Block Lanczos method options window

C. Constrain the Shell

Fix the shell at the welded parts on the bottom side of the shell by assigning all degrees of freedom constrained at these locations.
ANSYS Main Menu > Solution > Apply > Displacement > on nodes

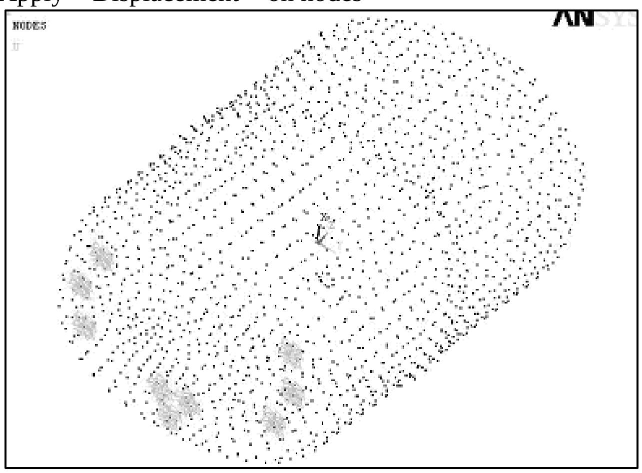

Fig. 7: Constrain the Shell

D. Solve

ANSYS Main Menu > Solution > Solve > Current LS
On pressing the solve option, the following two windows will open as shown below. Review the information in the status window and then close it. Then select 'OK' in the next window to begin solution of current load step.

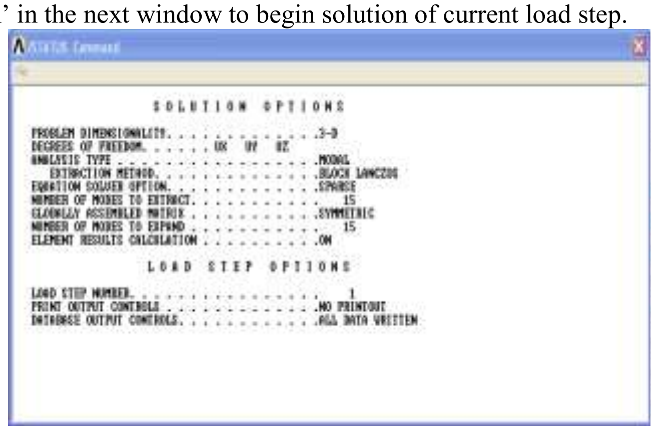

Fig. 8: status window

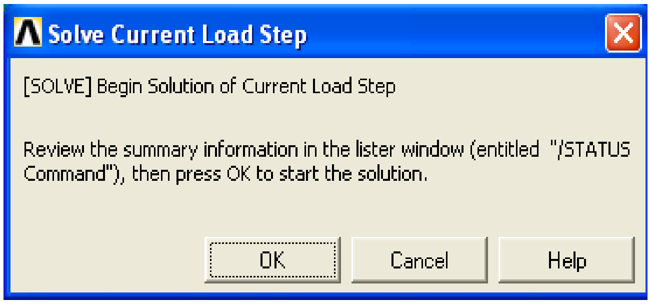

Fig. 9: Solve current load step

Fig. 10: Close the window

E. Review Results

1) List the Natural Frequency

ANSYS Main Menu > Postprocessor > Results Summary

Results summary provides the natural frequencies and mode parameters of the shell. The results obtained in the modal analysis are presented in the next chapter (Results and discussion).

IV. RESULTS & DISCUSSIONS

A. General Experimental Setup

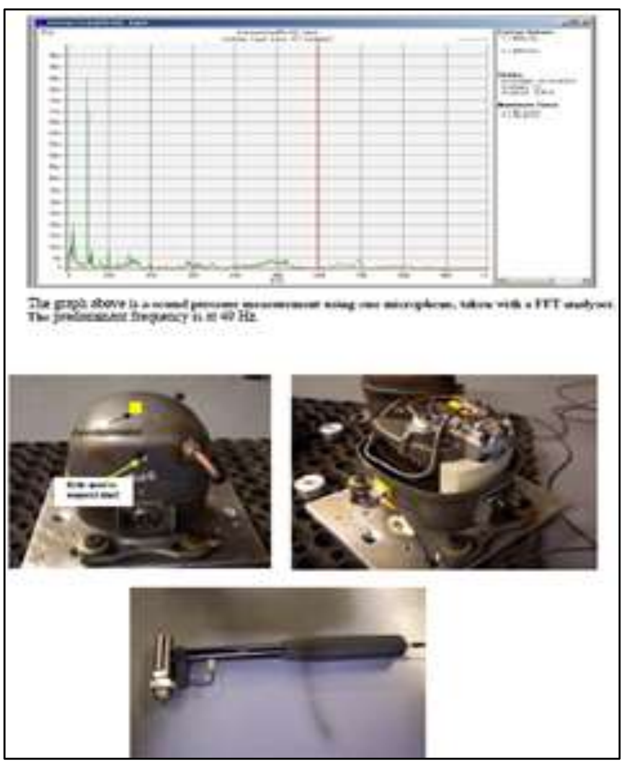

Fig. 11: General Experimental Setup

Experimental values indicating that there is more noise level at a frequency level 3.2k (Hz)

From this chart, it was identified that the noise level of the compressor shell is suddenly increasing at a frequency 3.2k (3200Hz). Hence it was found that there is a problem at a frequency of 3200Hz.

Table – 1
THK134 YCK Test Report

THK134 YCK TEST REPORT													
	Sound Power Test At Brazil Sound Room @ 230V, 50Hz												
Comp No	4030	4036	4037	4038	4039	4040	C-003	C-004	C-005	D001	D002	D-001-R	Min
Freq (Hz)													
100	26.8	28.1	28.8	26	28.5	26.9	28.3	24.5	26.2	25.2	24.6	25.1	24.5
125	22.9	22.6	23.2	23.5	23.3	23.3	26.3	22.9	22.9	22.2	22.8	23.1	22.2
160	17.7	19	19.1	19.1	17.8	18.4	21.8	21.3	22.3	20.9	19.8	18.5	17.7
200	18.9	16.8	16.4	17.5	16.3	16.7	22.2	15.9	15.2	14.9	13.8	15.2	13.8
250	16	11.6	16.4	20.3	14.2	15.9	20.2	13.6	14.1	14.8	14.7	18.5	11.6
315	18	17.4	19.5	31.7	18.8	19	18.5	19.5	16.6	16.7	17.7	16.1	16.1
400	19	25.5	31.3	37.7	31.5	27	28.5	28.2	26.4	25.2	26.8	28.3	25.2
500	19.6	30.6	32.1	41.7	32.2	34.2	29.5	32.7	30.4	29.1	32.4	33.5	29.1
630	18	32.6	25.2	36.8	26.2	39.1	32.6	30.4	29	29.5	25.6	30.7	25.2
800	18.5	26.7	25.8	29.8	24.4	25.2	26.2	28.3	23.3	22.4	20.5	23.8	20.5
1K	19	22.1	26.4	18.3	24.5	25.6	13.2	18.8	13.8	13.6	32.6	9.1	9.1
1.3K	21	25.7	27	21.3	28.7	26.3	18.5	17.5	17.1	14.3	10	16.1	10
1.6K	22	21.7	19.7	20.6	22.5	26.2	20.5	21.1	19.2	14.6	13	21.1	13
2K	24.6	16.3	21.8	21.5	26.5	24.5	20.1	24.5	19.8	16.3	17.2	17.7	16.3
2.5K	28	26.9	29.1	28.6	28.9	30	31.5	33.2	28	23.2	17.1	21.2	17.1
3.2K	38.6	31.9	31.5	32.5	35.2	34.8	38	34.9	30	39.1	37.1	33	30
4K	31.9	23.6	26.3	30.1	28.7	33.8	40.5	32.8	33.6	37	36.9	36.9	23.8
5K	36.8	30.7	32.4	32.1	35.8	34.4	39	37.2	34.9	37.4	34.7	37.5	30.7
6.2K	33.5	31.4	32.8	32.5	32	31.1	37.2	33.6	36.2	33.7	33.7	31.1	31.1

B. Original Housing

Fig. 12: Original Housing

1) Modified Housing: 1

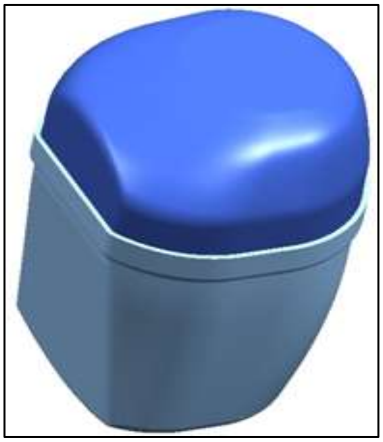

Fig. 13: Modified Housing: 1

2) Modified Housing: 2

Fig. 14: Modified Housing: 2

Natural Frequencies obtained for various housings in Modal analysis in ANSYS
The natural frequencies obtained for the original housing and the three modified housings (compressor shell) are shown in the table below.

Table – 2

Without legs			
Original housing	*Modified housing1*	*Modified housing2*	*Modified housing3*
Frequency value	*Frequency value*	*Frequency value*	*Frequency value*
2939	*1844*	*3158*	*2281.3*
3214	*2614*	*3524*	*3300.9*
3235	*2702*	*3552*	*3362.9*
3361	*2820*	*3790*	*3448.7*
3715	*2945*	*4177*	*3639.9*
4149	*3262*	*4471*	*3803.4*
4365	*3300*	*4690*	*3922.1*
4492	*3567*	*4702*	*4779*
4495	*4001*	*5053*	*4891*
4632	*4072*	*5286*	*5107*
4707	*4191*	*5322*	*5115*
4893	*4395*	*5413*	*5381*
4944	*4462*	*5542*	*5399*
5069	*4484*	*5595*	*5687*

Comparison of natural frequencies for different compressor housings without legs

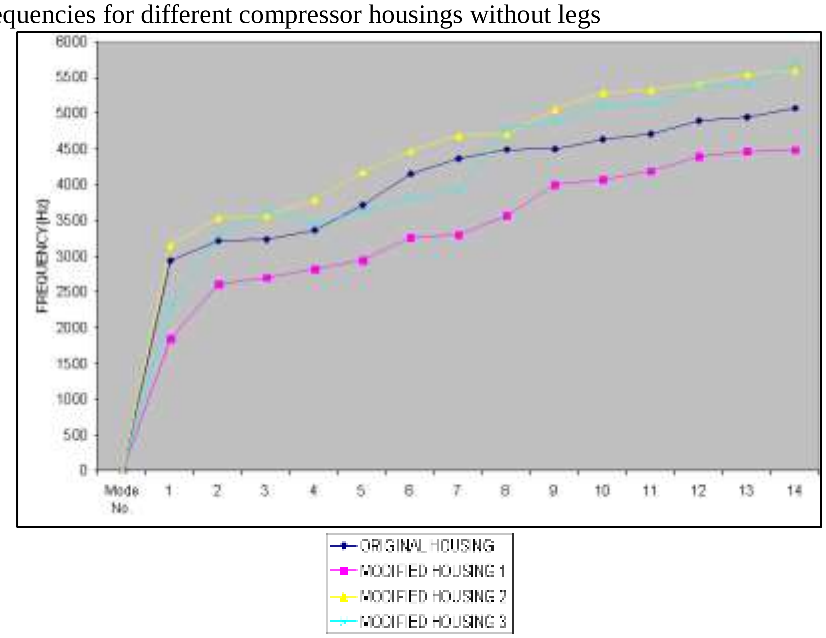

Fig. 15: Comparison of natural frequencies for different compressor housings without legs

From the above observation, First three mode shapes obtained in Ansys for different models without legs:

C. *Original Housing*

1) Mode 1

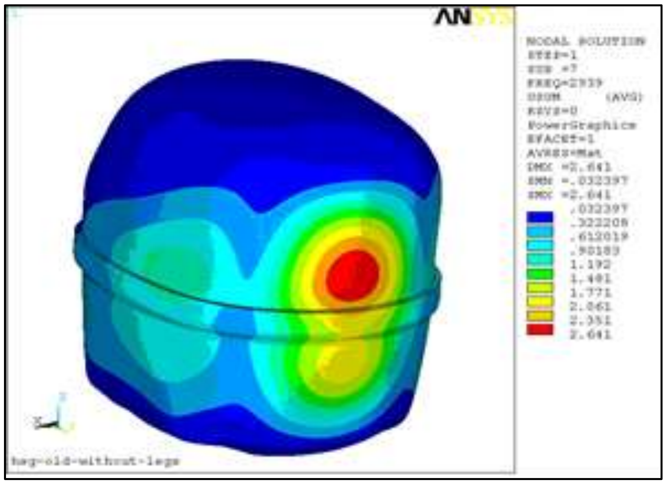

Fig. 16: Original Housing Mode 1

2) Modified Housing: 1

a) Mode 1:

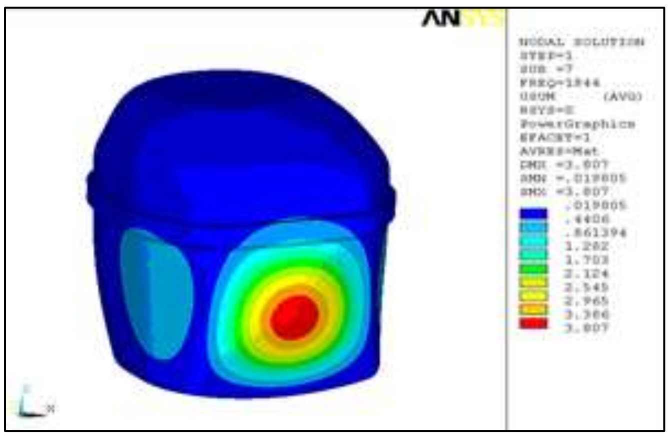

Fig. 17: Modified Housing Mode 1

b) Mode 2:

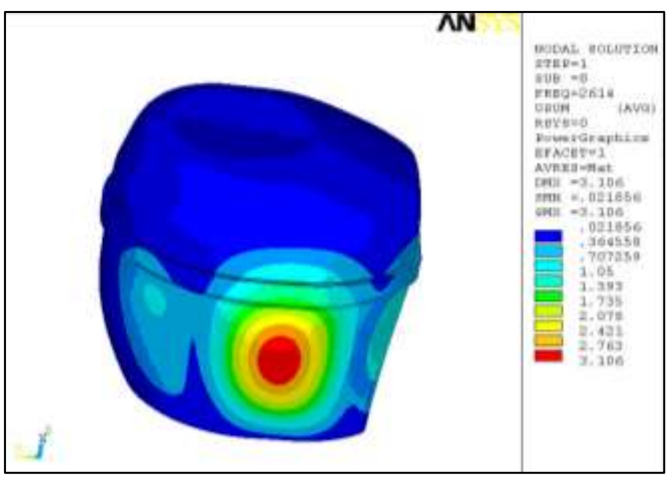

Fig. 18: Modified Housing Mode 2

c) Mode 3

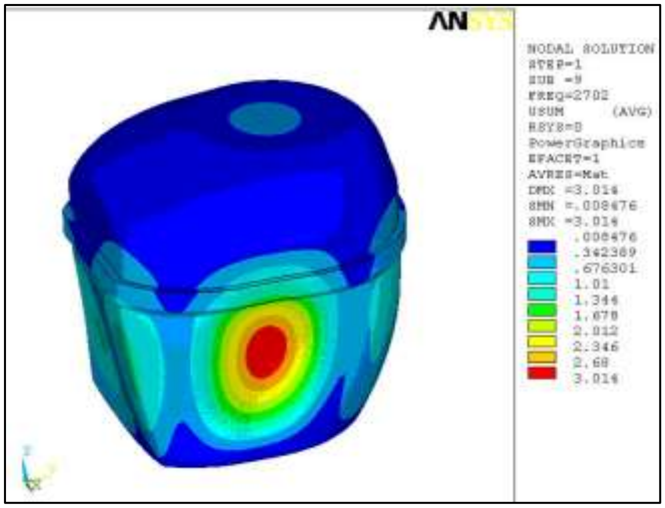

Fig. 19: Modified Housing Mode 3

3) Modified Housing: 2

a) Mode 1:

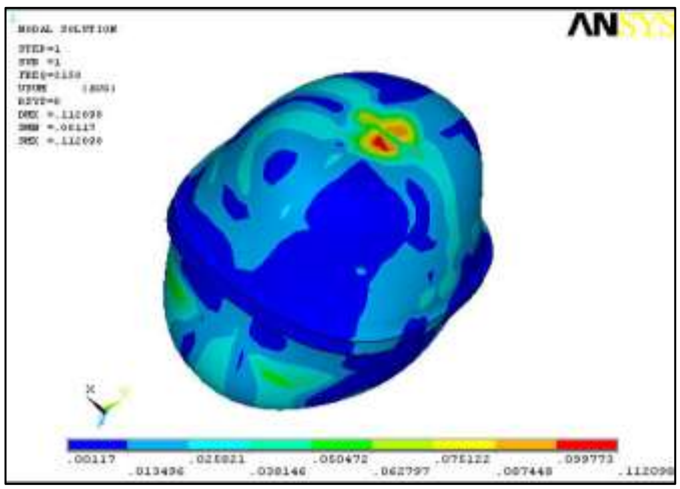

Fig. 20: Modified Housing 2 Mode 1

b) Mode 2

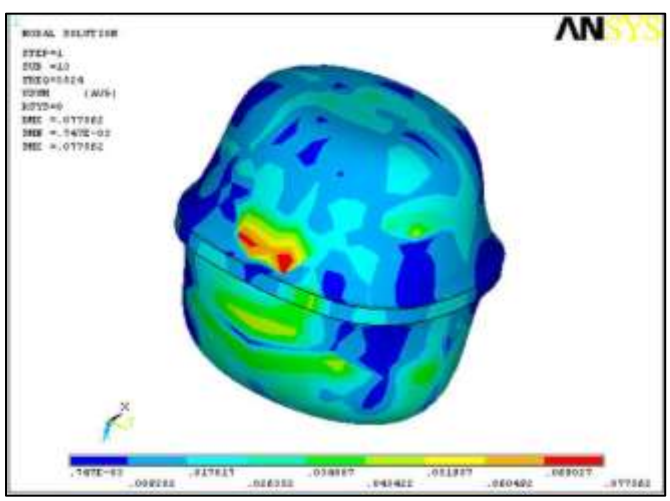

Fig. 21: Modified Housing 2 Mode 2

c) Mode 3

Table – 3

Mode	*Frequency*
1	*483.85*
2	*527.76*
3	*1448.0*
4	*2281.3*
5	*3300.9*
6	*3362.9*
7	*3448.7*
8	*3639.9*
9	*3803.4*
10	*3922.0*

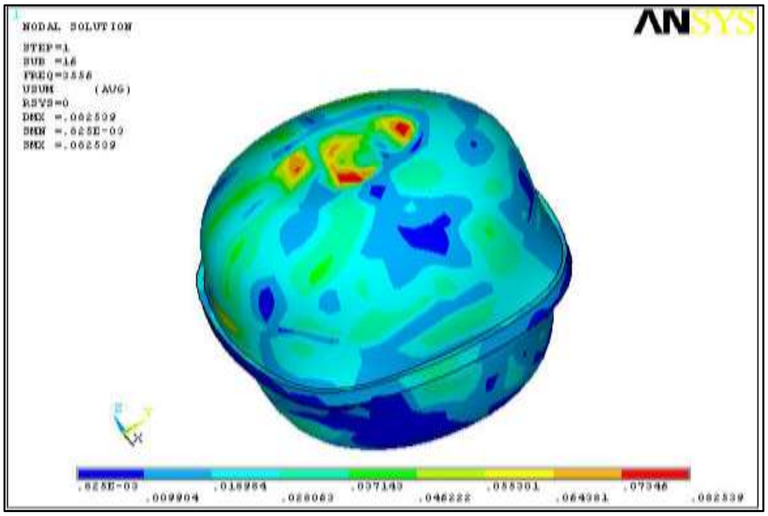

Fig. 22: Modified Housing 2 Mode 3

V. POWER SPECTRUM DENSITY ANALYSIS

The modes extracted from the finite element model of the compressor in modal analysis are used to evaluate the random response. All the vibration modes falling in the frequency range 2000-4000 are used for response spectrum analysis.

A constant damping ratio corresponding to 2% damping is used for analysis.

Broadband PSD as mentioned in the graph below is fed as input acceleration PSD in g^2/Hz. unit displacement excitation is applied at all the DOF restrained in the FEM.

Response displacement, velocity and acceleration solution relative to base excitation are obtained from analysis.

The 1σ displacements for the compressor are plotted. The maximum 1σ displacement values are 1.1, 1.1 and 3.2mm in x, y and z direction respectively. Displacement is maximum at the top surface. Top node 6917 was taken to find the response of the system.

The 1σ acceleration values for the compressor are 104.782, 88.121 and 248.312 mm/sec^2 in x, y and z directions respectively. The maximum stresses for the compressor are 353Mpa, 332Mpa and 58.7Mpa in the three directions respectively and are plotted below.

VI. RESULTS FROM PSD ANALYSIS

A. *Conclusion*

1) Margin of Safety

The following definitions applied for evaluating the safe working of the compressor.
Concept of margin of safety:

- UTS: Ultimate tensile strength
- YS : Yield strength
- WS: Working stress
- AS : Allowable stress

Allowable stress = $\min\left(\frac{UTS}{1.5}, \frac{YS}{1.25}\right)$

Margin of safety = $\left(\frac{AS}{WS} - 1\right)$

2) Material Properties

The material properties for the given body are mentioned below.

Material: 15-5 PH steel; H 1025 heat treatment condition; ref: AMS 5959K.

1) UTS= 1069Mpa.
2) YS= 1000MPa.

Allowable stress in tension = min $\left(\frac{1069}{1.5}, \frac{1000}{1.25}\right)$ =800Mpa.

Allowable stress in shear = 400Mpa.

The margin for safety for the 1 σ stress in the three perpendicular directions are tabulated below.

Table – 4

Direction	*Maximum working stress*	*Margin of safety*
X	*353*	*2.26*
Y	*332*	*2.40*
Z	*58.7*	*13.6*

From the above discussion we can conclude that the excitation frequencies do not coincide with the natural frequencies of a system. So the system is working under safe conditions.

REFERENCES

[1] Kinsler, L. E., Frey, A. R., Coppens, A. B, and Sanders, J. V., “Fundamentals of Acoustics,” John Wiley & Sons,Inc., 1982.

[2] Bassi, F., Pelagalli, L., Rebay, S., Betto, A., Orefice, M., Pinto, A., “Numerical Simulation of a Reciprocating Compressor for Household Refrigerators”, Proc. Compressor Engineering Conference at Purdue, West Lafayette, Indiana, USA, p. 97-104, 2000.

[3] Imaichi, Noriak Ishil, Shiro Saito, Kenichiro Imasu, Shigeru Muramatsu Kensaku, Proceedings of the July 1980 International Compressor Engineering Conference at Purdue Pp.90-96, High Frequency Vibratory Components Caused by Elastic Vibrations of the Crankshaft in Refrigerating compressors.

[4] Katsuya Minnami, Hitoshi Hattari, Makoto Hayano, Proceedings of the July 1998 International Compressor Engineering Conference at Purdue Pp.408-413, Lubrication Analysis of Reciprocating Compressor for HFC refregerents.

[5] Hyuk Lee, Byung-ha Kwon, Sung-oun Park " The Reduction of the Low-Frequency Band Noise in Hermetic Refrigerator Compressor", Proceeding of the 1998 Purdue Compressor Technology Conference