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ANALYTICAL AND NUMERICAL ANALYSIS OF VIBRATION ISOLATOR FOR HERMETICALLY SEALED COMPRESSOR

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***Abstract*— Customers of domestic goods, such as air conditioners, vacuum cleaners, refrigerators and washing machines, have recently become more exigent in respect to the acoustic comfort. Nowadays, in the refrigeration market, manufacturers are investing in noise control for their products either to mitigate the quality problems or to respect the standards que the Governments create. For that reason, the finite element model is created to analyze the vibratory behavior of isolators used in the compressor mounting. The methodology of the research work is to analyze various materials for dampener for better mechanical properties and least transmissibility of vibration. The main aim is to select better dampener material to get effective results of vibration isolation system. The target of the research work is to optimize the geometrical parameters of isolators for better productivity of the system. The analysis is carried out for stresses, deflection and transmissibility of isolators.**

Keywords—Stiffness, Damping, Transmissibility, Vibration isolation

I. INTRODUCTION

This work aims to study the generation of vibrational energy by sealed positive displacement compressors and the interaction of this with the base of a household refrigerator, through the rubber isolators, also known as shock absorbers. These holes are mounted in the compressor base plate, these holes can vary from that position according to the compressor model. Seeking an optimization of the cooling system, the vibroacoustic point of view, in view of the increasing demands of consumers, the objective of this work was to study and create a review process for the vibration isolators.

Fig. 1. Rubber insulator with compressor

These components are usually made of rubber and have meaningful participation in noise cooling systems, especially in frequencies up to 400Hz. Figure 2 shows how the low-frequency vibrations of a compressor influence the noise of a cooling system in frequency bands up to 400Hz, mostly. In this figure, the noise irradiated by a refrigeration system wherein the compressor operation frequency is not tuned to any mount frequency in the system (dependent on the insulating characteristics) is compared to the noise obtained in a situation where there is alignment between the and operating frequency of the compressor assembly of the system.

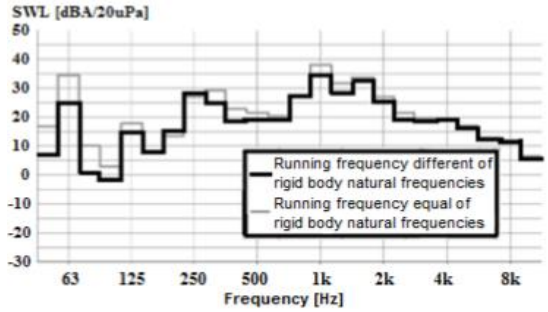

Fig. 2. Low frequency noise optimization with a good choice of grommets.

II. MATERIAL CHARACTERIZATION

A. Rubber

The first step necessary for the evaluation of a vibration isolator is the characterization of the material it is made. Therefore, a brief introduction to the viscoelastic material, particularly rubber, is made

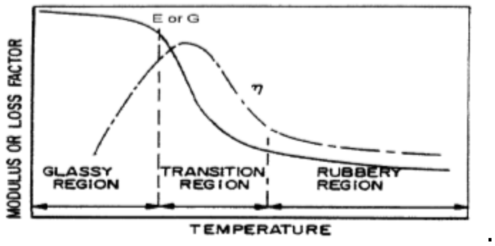

Fig. 3. The Young Modulus and the Loss Factor vs. the temperature.

B. Relationship between hardness and modulus of elasticity

Equation (1) is a semi empirical equation for obtaining the elasticity modulus in Mpa from a shore (A) hardness value proposed by Gent (1958)

$$E = \frac{0.0981(56 + 7.62336S)}{0.137505\,(254 - 2.54S)} \quad ..... (1)$$

III. Analytical Model Analysis

A. Analysis by MATLAB programming

Analytical models are mathematical models that have a closed form solution, i.e. the solution to the equations used to describe changes in a vibration isolation system can be expressed as a mathematical analytic function. For the selection of material, their properties and design parameters to any application it is important to do analytical analysis for achieving effective design. So for analytical analysis of vibration isolation system for compressor, by using MATLAB software the mathematical modeling is generated to easily and quickly calculate the results.

Built the MATLAB programing for-

- Arbitrary number of isolators
- Mass matrix
- Stiffness matrix
- Eigen values
- Natural frequencies
- Mode shapes
- 3 dimensional transmissibility
- Steady state sine vibration frequency response

Prerequisite data for analysis-

- Number of isolators or mounting points
- Mass of a compressor
- Location of the Center of gravity of a compressor
- Moment of inertia about x, y, and z axis at center of gravity of compressor
- Stiffness values of individual isolator in x, y, and z axis
- Distance of each isolator mounting point to the C.G. of a compressor
- Damping ratio of isolator material
- Frequency of a compressor
- Amplitude of vibration

The suitable method to get dynamic modulus of that material is to perform compression test on UTM. By finding out the elastic modulus at operating frequency and temperature, it is used to calculate the stiffness values by theoretical formulae. The theoretical formulae should be used according to geometry profile of the isolator. The alternate method to calculate the stiffness is ANSYS software. First of all, create the model of isolator and provide the material properties such as dynamic young's modulus, poisons ratio and density etc. After that mesh the geometry and apply the force on loading platform and obtaining displacement or deformation of geometry in ANSYS software tool.

The damping ratio of material is obtained only experimentally and this is dynamic property.

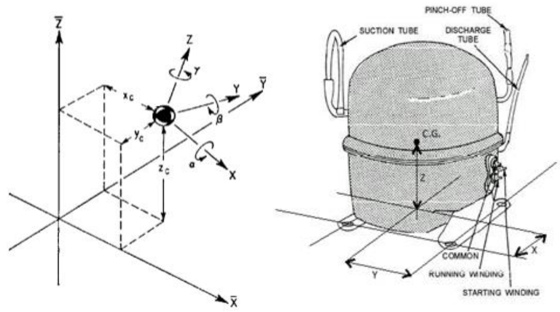

Fig. 4. Coordinate system notation of isolator location

The transmissibility graphs are plotted for acceleration as well as relative displacement transmissibility vs. frequency for x-axis, y-axis and z-axis excitation. For steady state sine vibration base calculations, the program is generated by which it is easy to get the response to the given frequency and amplitude of vibration.

B. Static and modal analysis by ANSYS APDL

The first step is to complete the preprocessor part i.e. to select the element type as MASS21 for mass as a compressor and SOLID185 as a grommet. Then provide real constants of mass properties and inertia values to the selected mass.

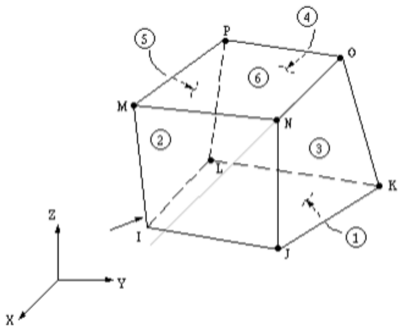

Fig. 5. Element SOLID185

The second step is to give the material properties such as dynamic modulus, poisons ratio and density etc. Then the modeling of isolation system is done systematically as create one node for an element at an origin, and select it as mass by element attributes with auto numbered through nodes. Create the geometry of individual grommet with proper dimensions at the center and copy this geometry at the locations as per the distance from C.G. of a compressor to the isolation mounting point according to a coordinate system. The next step is to create the system mesh: the origin node is considered as a rigid body mass, while the grommets with quadratic hexahedral elements. After finishing the mesh, the fixed-fixed joints are created between the grommets loading surface nodes and the origin node mass by coupling rigid region option. The boundary conditions and joints are defined: the nodes of the lower grommet surfaces are fixed to the platform by applying zero displacement and zero DOF constrains to the lower surface. Define load: apply the global gravity inertia values to the specific axis before solve the case.

The first analysis is a static analysis (static properties for the grommet), used to calculate the grommet deformation due to the gravity acceleration. Considering the deformed mesh obtained from the static analysis, a modal analysis with Block Lanczos algorithm with considering pre stressed effect should be carried out. Then the general post processing results will be achieved and further analysis can be carried out.

IV. Results

Fig. 6. Static stress analysis of neoprene isolators

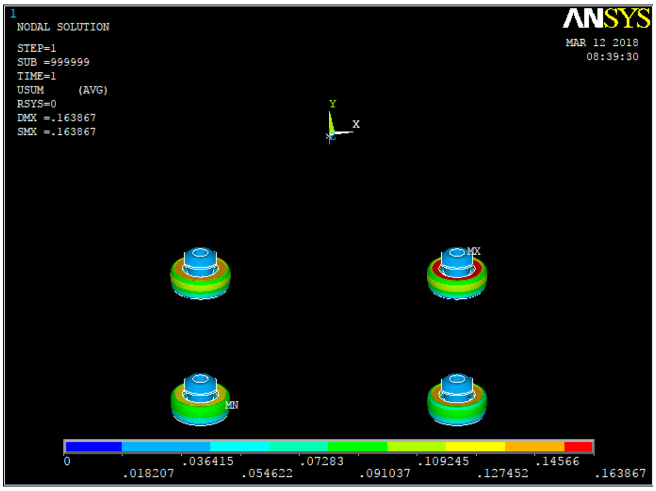

Fig. 7. Static displacement analysis of neoprene isolators

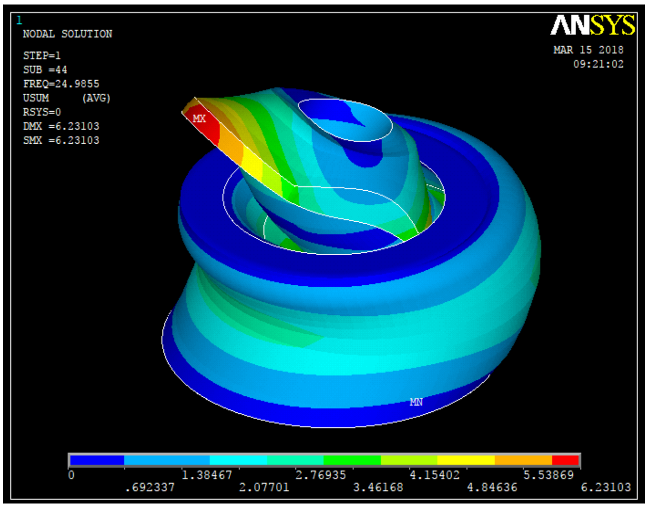

Fig. 8. Enlarged view of mode shape for natural frequency

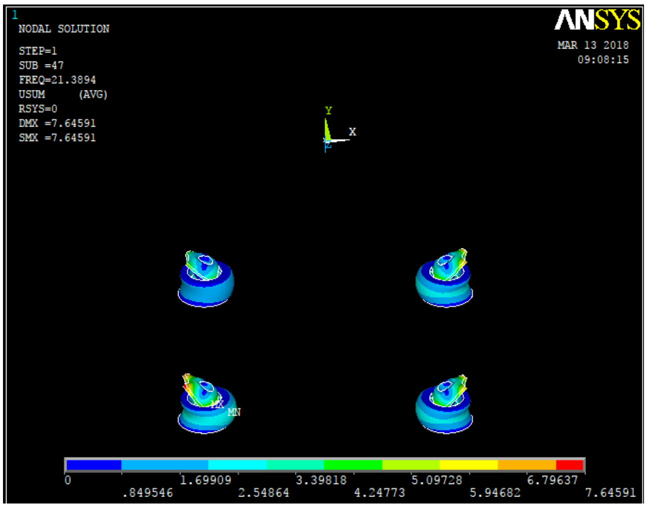

Fig. 9. Mode shape for natural frequency

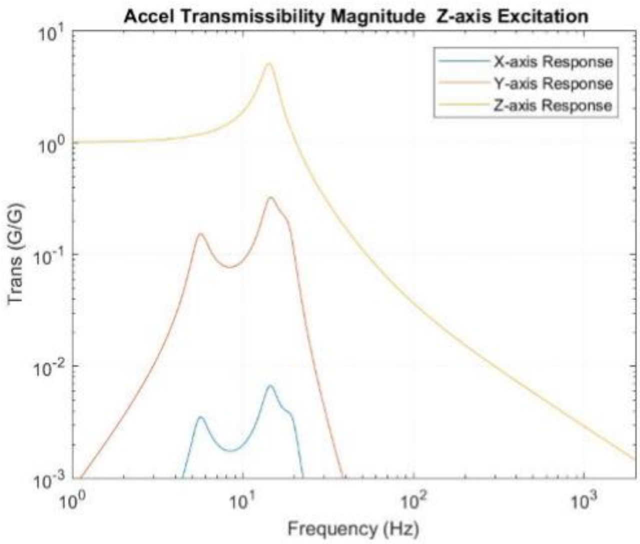

Fig. 10. Transmissibility vs Frequency graph of neoprene rubber

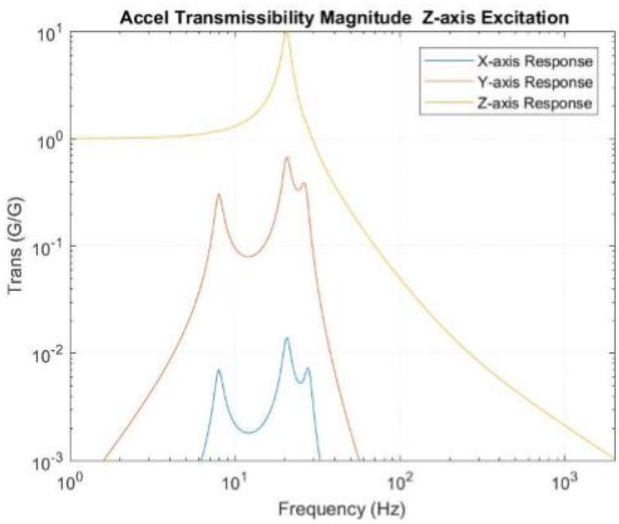

Fig. 11. Transmissibility vs Frequency graph of natural rubber

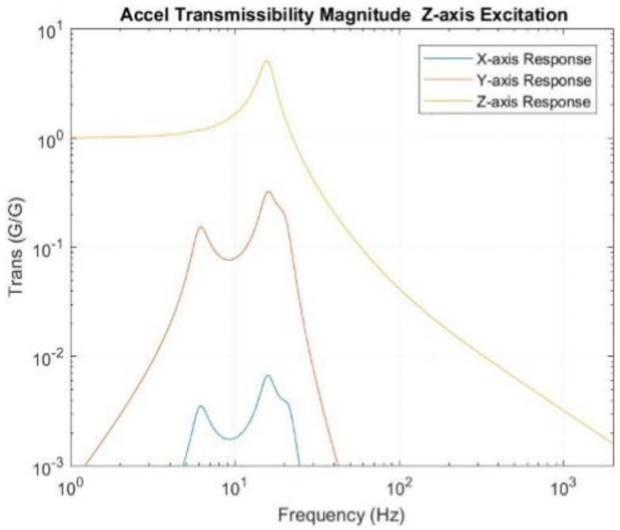

Fig. 12. Transmissibility vs Frequency graph of natural rubber

TABLE I. ANALYTICAL ANALYSIS RESULTS

Material	**Max. tensile stress**	**Max. compressive stress**	**Max. deflection**	**Transmissibility(A)**
Unit	***psi***	***psi***	***in***	
Neoprene	59.54	121.68	0.1638	0.1112
Natural	118.58	242.44	0.1607	0.2125
EPDM	67.91	138.91	0.1579	0.1324

V. CONCLUSION

This paper presented the importance of vibration isolators, also known as grommets, and how a good grommet choice or project may optimize appliance noise in the range from 0 to 400Hz. Obtaining the rubber properties from the techniques and using the static and the dynamic models presented in this paper, it is possible to analyze, design and optimize vibration isolator for refrigeration compressors. Based on the static analysis and using a modal analysis with grommet pre-deformation, a simplified model of a compressor assembled on grommets was analyzed, showing its six rigid body natural frequencies and some of its modes. Hence, these analysis technique tools can be used for designing the better isolation system for any kind of hermetically sealed compressor.

Future work also address the effect of the pressure of the refrigerant fluid and the suction and discharge lines in the dynamics of the compressor when it is mounted on a chassis of a typical refrigeration system. This would show the limitations of the model presented here, providing the design of numerical models closer to reality.

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Illusrations

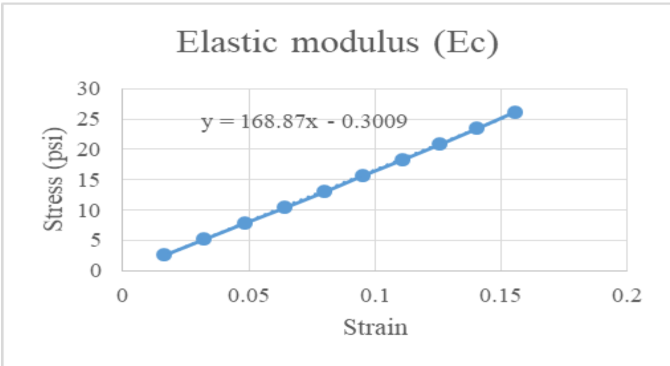

Fig. 13. Stress-strain graph of neoprene rubber

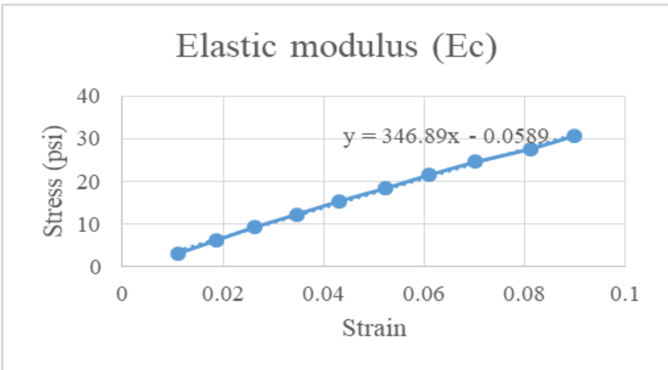

Fig. 14. Stress-strain graph of natural rubber

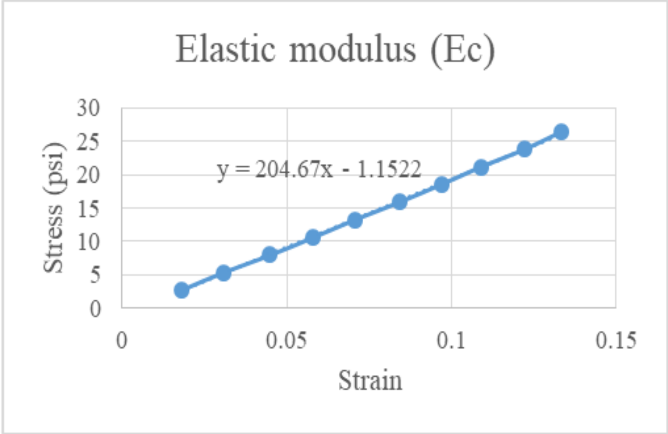

Fig. 15. Stress-strain graph of EPDM rubber

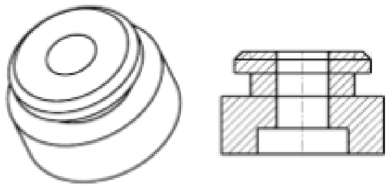

Fig. 16. Isolator geometries

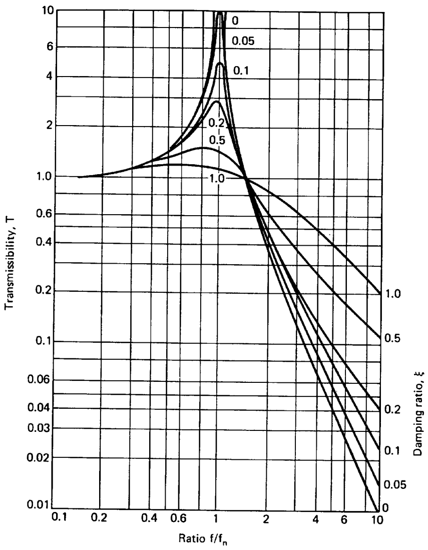

Fig. 17. Transmissibility vs. Frequency ratio graph