2002

Experimental Study Of Vibration Transmissibility Using Characterization Of Compressor Mounting Grommets, Dynamic Stiffnesses Part-II, Experimental Analysis and Measurements

A. T. Herfat
Copeland Corporation

Follow this and additional works at: https://docs.lib.purdue.edu/icec


This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information. Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
ABSTRACT

The main sources contributing to scroll compressor vibration at running frequency are rotational unbalance in the crank-rotor assembly and the reciprocat ing force due to the Oldham Ring. A rigid body model of the compressor vibrating on resilient grommets has been developed in this paper. The model is used to study the effect of different grommets for vibration suppression of the compressor as well as reducing the vibration transmissibility to the base panel of a system (such as heat pump, or condenser units). This study enables us to select appropriate grommets that provide the desired vibration isolation for the system.

INTRODUCTION

The dynamic components of machines provide work by using forces, moments, and torques. Unfortunately, these components can also produce noise and vibration effects. The adverse effects of noise and vibration disturbance can reduce machine life through the fatigue failure of machine components as well as deteriorate perceived performance.

Appropriate noise and vibration control will improve the machine quality and performance. A damping treatment is used to reduce transient and steady state vibration as well as load transmissibility. It can be accomplished by adding a damping material layer or component (e.g. grommet), such as rubber to the existing structure. The combined system has a higher damping level and thus reduces the unwanted vibration. This procedure is described by using the complex stiffness notation, K*. The concept of complex stiffness results from considering the harmonic response of a damped system. The performance of typical elastometric isolators varies with changes in dynamic input or with the level of vibration that the system is being subjected. The strain sensitivity of the elastomers causes the dynamic characteristics to change.

EXPERIMENTAL PROCEDURE

The Sound and Vibration Laboratory used a Heat Pump unit with a scroll compressor to obtain experimental load transmissibility measurements as well as mounting grommets dynamic stiffness measurements. The load measurements were acquired using grommets with 40, 50, and 60 durometer. The measurements were acquired using the following steps:

1- The driving or excitation loads produced by the compressor were measured using four tri-axial load cells that were preloaded on a steel table, with four steel mounting grommets between the compressor feet and the table.

2- The compressor was re-installed inside the heat pump unit and was running at 45/130/65 F. The tri-axial load cells were used to measure the load transmitted to the base pan of the heat pump unit.
3- Transmissibility was measured for three different mounting rubber grommets: 40 durometer, 50 durometer, and 60 durometer. Transmissibility was measured for the first nine driving load frequencies and in three dimensions, due to rocking, torsion and axial force excitations.

4- For mounting grommet deformation due to axial load, shear loads, and torque that are produced by the compressor, six accelerometers were used both at the top of each compressor foot and at the bottom of the mounting foot (on the base pan).

5- A sixteen-channel data acquisition, along with a signal processing and modal analysis interface, were used for this experiment.

**Definitions and Assumptions:** Analytical method for calculation of the complex compression stiffness and complex shear stiffness of grommets using complex modulus of elasticity and complex shear modulus have been presented in Part-1, first paper.

Assume that the cross sectional area of the grommet in any plane parallel to the compressor feet remained parallel and in a circular shape. In this case the rubber grommets are compressed and restricted completely between the compressor feet and the base-plate in this analysis (fully bonded), see Fig.-1.

![Fig-1](image-url)

Where: $F_s =$Shear Force, and $F_c =$Compressive Force

The equation of motion for a damped-forced system in steady state condition is defined as 

$$[-M \omega_d^2 + K(1 + \frac{j \omega_d C}{K})]X = F_M$$  \hspace{1cm} (1)$$

Complex Stiffness, $K^*$ is defined as

$$K^* = K(1 + \frac{j \omega_d C}{K}) = K(1 + j \eta)$$  \hspace{1cm} (2)$$

Where: 
- $M =$ Compressor mass, Kg
- $K =$ Grommets Stiffness, N/mm
- $X =$ Steady state displacement, maximum amplitude, mm
- $F_M =$ Compressor displacement, maximum amplitude, mm
- $\omega_d =$Compressor driving or excitation force, maximum amplitude, N
- $C =$ Grommet damping, N-s/mm
- $\eta =$ Loss Factor

This is called the Kelvin-Voigt model of a material, the imaginary part of the complex stiffness corresponds to the energy dissipation in the system. Therefore it is called “loss factor” and depends on the drive or excitation frequency. The steady state amplitude at resonance becomes

\[ |X| = \frac{F_M}{K \eta} = \frac{F_M}{2K \xi} \quad (3) \]

Where: \( \xi \) = Damping ratio.

**CONCLUSION**

The experimental dynamic stiffness measurements show: 1) All three grommets (40, 50, and 60 durometer) have higher compression stiffness than shear stiffness or dynamic torsional stiffnesses. 2) The shear stiffness is larger than the dynamic torsional stiffness for 50 and 60 durometer grommets. 3) The dynamic torsional stiffness of the 40 durometer grommet is larger than the dynamic torsional stiffness of 50 and 60 durometer grommet. The 50 and 60 durometer grommets have almost similar Dynamic stiffness in the 10-500 Hz frequency range and smaller for these standard rubber grommets (40, 50, and 60 durometer). 4) The highest resonance peaks are found for the axial (vertical) vibration transmissibility at 409 Hz, and for all three grommets. The transmissibility of 50 and 60 durometer grommets are close to each other and larger than that of 40 durometer grommet. The resonance peaks are also found for the axial (vertical) vibration transmissibility around 233 Hz, and for all three grommets, but much smaller than those at 410 Hz. The lateral vibration transmissibility is observed more than axial transmissibility at around 116 Hz, but much smaller than those at 410 Hz. Therefore, in general the axial transmissibility to the base pan of the unit dominates over the lateral ones. However, vibration transmissibility of the grommets changed by frequency variation as well as hardness variation. 5) The analytical axial vibration transmissibility (x) was done only for 60 durometer grommet. The result came up close enough to the experimental results. 6) The nonlinear mechanical properties of these rubber grommets can be viewed through these experimental results as well as analytical.

**ACKNOWLEDGMENTS**

The following are gratefully acknowledged for supporting this project:
Dr. Robert V. Seel, Technical Services Director, Copeland Corporation
David Steinbarger, Technical Services / Quality Vice President, Copeland Corporation
John Roberts, Bruce McPheron, Jim Nelson, and Ken Heberle Lab Technician, and Brian Hughes Lab Supervisor, Sound and Vibration Laboratory, Copeland Corporation

**REFERENCES**

ILLUSTRATIONS

Fig-2: (a) Three Dimensional Resultant Compressor Driving Forces at Compressor Feet. (b) Three Dimensional Resultant Compressor Driving Moments Measured about XG, YG, And ZG Axis. Steel Grommets between Compressor Feet and the Steel Table, at Room Temperature

Fig-3: (a) Compressive Dynamic Stiffness of 60, 50, and 40 Durometer Grommets. (b) Dynamic Shear Stiffness of 60, 50, and 40 Durometer Grommets, Room Temperature.

Note: The Analytical results for 60 Durometer Grommet presented by (x).
Fig-4: Dynamic Torsional Stiffness of 40, 50, and 60 Durometer Grommets at Room Temperature

Fig-5: Compressor Excitation force transmissibility to the Base Pan for 60 Durometer Grommets at Room Temperature. (a) 0-500 Hz Frequency Range, (b) 0-300 Hz Frequency Range
Fig-6: Compressor Excitation force transmissibility to the Base Pan for 50 Durometer Grommets at Room Temperature. (a) 0-500 Hz Frequency Range, (b) 0-300 Hz Frequency Range

Fig-7: Compressor Excitation force transmissibility to the Base Pan for 40 Durometer Grommets at Room Temperature. (a) 0-500 Hz Frequency Range, (b) 0-300 Hz Frequency Range
Fig-8: Axial Compressor Excitation Force Transmissibility to the Base Pan, TRz, for 40, 50, and 60 Durometer Grommets at Room Temperature. (a) 0-500 Hz Frequency Range, (b) 0-300 Hz Frequency Range

Note: The Analytical results for 60 Durometer Grommet presented by (x).