# Design of Rubber Grommets for a Reciprocating Compressor

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***Abstract***

The objective of this research is to design a new geometry of rubber grommet for compressors used in small refrigerators. The rubber grommets are used as vibration isolators that reduce the magnitude of vibration due to the periodic motion of active components in the compressors. The design of rubber grommets must therefore account for the vibration characteristics of each individual compressor and the cost of production. A reciprocating compressor in a commercial refrigerator is selected as the representative system for this study. The characteristics of the selected compressor's vibration response are measured from accelerometers. The geometry and mechanical properties of the rubber grommets used in this system are characterized using standard tests. The vibration response of the system is then numerically simulated in a commercial finite element program to analyze the effectiveness of the existing rubber grommets as vibration isolators. In the simulation, the grommets are constitutively prescribed by a viscohyperelastic model. The viscous part and the hyperelastic part of the viscohypelastic model are related to damping and to stiffness of the rubber grommet, respectively. The magnitudes of vibration response from the simulation are favorably compared to those measured from the compressor. The simulation framework is then used in the design of new rubber grommets that can theoretically absorb the vibration as well as the original design, but the new designs require less mass that leads to a reduction in the cost of production.

***Keyword:*** Rubber Grommet, Viscoelastic, Hyperelastic, Vibration Isolator, Finite Element Simulation

## 1. Introduction

Compressors are the main equipment of refrigeration units such as air conditioners and refrigerators. Reciprocating compressors are one of the common types of compressors because they can achieve a high pressure ratio at a low energy input. However, the cyclic movement of pistons in the reciprocating compressors results in mechanical vibration that can cause undesirable noises. To eliminate the noise, grommets are installed as structural supports for isolating the vibration sources, i.e., the compressors, from the ground. The grommets are commonly made of rubber because they have sufficient stiffness to sustain the weight of the compressors and can reduce the magnitude of vibration by dissipating the vibration energy as heat while being cyclically deformed.

The commercial rubber grommets are available in a wide range of geometry and stiffness. In the reciprocating compressors for household refrigerators, however, some common features in rubber grommets are observed:

1) a grommet has a cap head and a slit for inserting into a hole at the base of a compressor,

2) the part below the slit is the shoulder area for resting the base of a compressor,

3) the part under the shoulder is the base, and

4) at the center of a grommet there is a hole for inserting a bolt that locks the grommet to the ground support of the compressor.

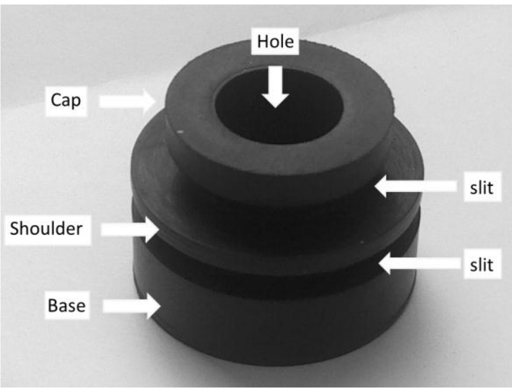


Fig. 1 Common features of a rubber grommet

Among the grommet features, only the base bears the weight of the compressor, and thus it is responsible for isolating the vibration. Thus, the optimization of rubber grommet as a vibration isolator involves the specifications of the base geometry and the selection of rubber properties that match the vibration characteristics of the compressors.

In the industry, the design of rubber grommet is typically provided by the compressor manufacturers, while the grommet manufacturers only serve as made-to-order suppliers. To increase the value of grommet and to reduce the manufacturing cost, a systematic grommet design framework must be developed for the industry.

This research provides a framework to design geometry of the grommet to fit a specific compressor. Due to the limited availability of raw materials, the optimization process is performed on the geometry of the grommet while assuming the same material properties as that of the original grommet. A commercial reciprocating compressor is selected as the representative system for this study. The characteristics of vibration response of the selected compressor are analyzed and used as loading inputs in numerical simulation of the compressor system. The simulation is created in a finite element framework using a commercial program, ABAQUS. In the simulation, the rubber grommets are constitutively prescribed as a viscohyperelastic material where the hyperelastic part and the viscous part correspond to the stiffness and the damping responses of the material, respectively. The simulation of the existing grommets is validated by comparing the calculated vibration output to the experimentally measured values. Two new designs of the rubber grommets are then analyzed using the validated simulation framework. The simulation results show that the new designs have similar vibration response but require lower masses than the original design. The simulation framework can be applied to analyze the grommets for other industrial applications.

## 2. Literature Review

Theoretical framework of vibration isolators have been rigorously analyzed in a number of standard vibration textbooks, such as [1], [2]. Typically, the performance of a vibration isolator is classified by its transmissibility (T) – a ratio of the force transmitted out of the system versus the force generated within the system.

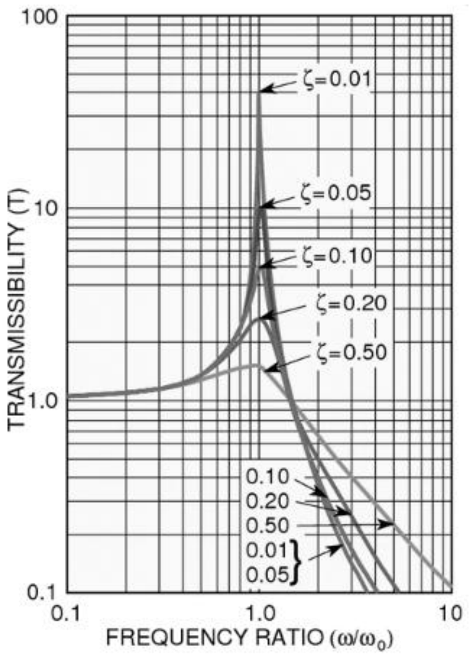


Fig. 2 Transmissibility curves

Fig. 2 shows a transmissibility curve plotted as a function of a ratio between a working frequency ($\omega$) and the natural frequency of the system ($\omega_0$) with variations in the damping ($\zeta$). The lower the transmissibility value is, the better the vibration isolator is regarded. Vibration isolators are employed in a wide range of applications. Luca et al. [3] provided a review of the active and passive vibration isolators particularly for the civil engineering applications. Ashrafiuon [4] carried out a dynamic analysis of an airplane engine mount systems by assuming a linear vibration response up to the tested frequency of 80 Hz. The analysis covers the transmissibility and the amplitude of vibration for three fundamental models. The engine mounts are modeled as a three dimensional spring with a hysteresis damping. Yu et al [5] analyzed the dynamic stiffness and the damping that depend on both frequency and amplitude of engine mounts. They demonstrated that the passive hydraulic mount is better than the elastomeric mount especially at low frequencies. For the study on rubber grommets, Tawan [6] performed a study of the dimensional effects on the mechanical behavior of rubber grommets as engine mounts.

## 3. Experiments

### 3.1 Physical properties of the compressor

The selected compressor unit taken from a mid-size household refrigerator consists of a 457 BTU/Hr reciprocating compressor seated on four identical rubber grommets that are attached on top of an aluminum base. The mass of the compressor unit is 8.41 kg. The center of gravity (CG) of the compressor is located by measuring the weight distribution of the freely suspended compressor. Fig. 3 shows the locations of the grommets and the CG in the compressor.

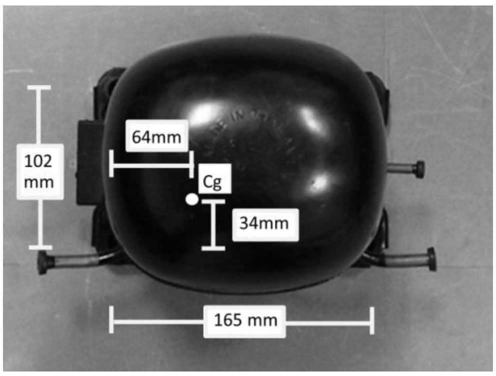


Fig. 3 Locations of CG and grommets

The rubber grommet has a cross-section shown in Fig 4. Each grommet weights 12.8 grams.

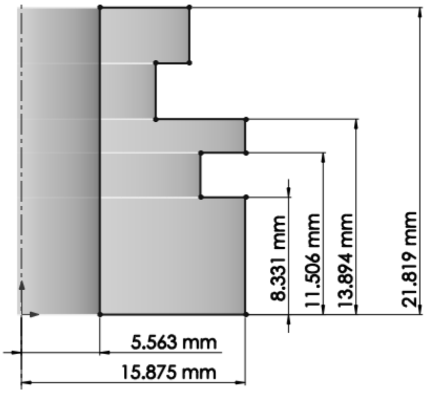


Fig. 4 Cross section of the original grommet

### 3.2 Vibration response

The vibration response of the compressor is measured using a set of accelerometers. The accelerometers are attached at the base of the compressor to measure the acceleration originated at the source of vibration, and at the aluminum support underneath the grommets to measure the acceleration transmitted through the grommets. Fig. 5 shows the locations of the accelerometers. The signals at the source are used as inputs in the simulation, whereas those at the base are for comparing with the simulation response.

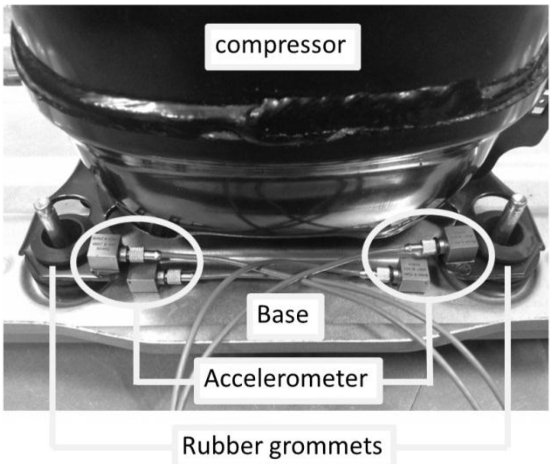


Fig. 5 Measurement of vibration response using accelerometers

### 3.3 Measuring Material Properties

Because the shape of rubber grommets does not allow standard test methods for measuring material properties, a request is made to the grommet manufacturer to provide the rubber in a form of a sheet. The rubber sheet is tested under two conditions: the elastic properties are measured from uniaxial tensile tests and the viscous properties are measured from stress relaxation tests. Both tests use rubber specimens prepared according to ASTM D412. The tensile test specimens are loaded at a crosshead speed of 500 mm/min. The relaxation tests are conducted by initially stretching the specimen to a 25% nominal strain at a speed of 500 mm/min and then holding for 600 seconds. The test results shown in Fig. 6 and 7 for the tensile test and the relaxation test show a typical nonlinear time-dependent behavior of an elastomer.

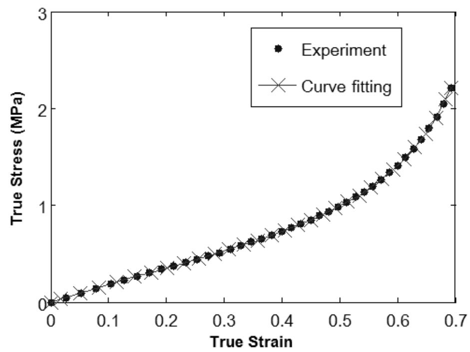


Fig. 6 Uniaxial tensile response

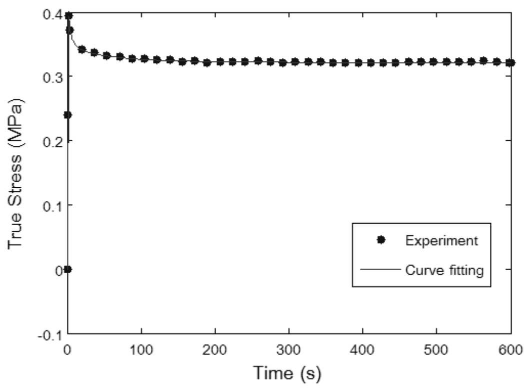


Fig.7 Relaxation test results

## 4. Simulation

A numerical simulation replicating the compressor-grommet system is created in the finite element software ABAQUS. The compressor is modeled by a set of rigid mass points acting on top of the grommet. The aluminum base is represented by a discrete rigid body with an infinite mass. The rubber grommet is modeled by a three-dimensional deformable body with the measured dimensions. The grommets are classified based on their positions in the compressor system in Fig 8. The base is set to remain stationary. Frictionless contact is assumed at the grommet-based interface. The grommets are meshed by 8-node linear brick hybrid elements with reduced integration (C3D8RH). The grommets are constitutively prescribed as a hyperviscoelastic material. The hyperelastic parameters are fitted from the uniaxial tensile test using the Marlow model. The viscoelastic part of the model is fitted to the relaxation data using a Prony series of relaxation modulus with the following equation:

$$g(t) = 1 - \sum_{i=1}^{N} g_i \left(1 - exp\left(-\frac{t}{\tau_i}\right)\right), \quad (1)$$

where $g(t)$ is the relaxation modulus, $g_i$ is a nondimensional multiplier, and $\tau_i$ is the relaxation time. The relaxation spectrum coefficients fitted from the experimental data are listed in Table (1)

The analysis is performed in three steps. First, a static dead load representing the weight of the compressor is applied on each grommet. Because the center of mass of the compressor is not aligned with its geometric center, the assigned weight on each grommet varies according to the load distribution. Second, the first mode natural frequencies of the system are calculated using * FREQUENCY option. Third, the vibration response of the system is analyzed in *MODAL DYNAMIC using the measured vibration response as input at the point masses at the top of each grommet. The simulation output is measured by averaging nodal acceleration at the grommet-aluminum base interface.

Table 1 Coefficients of relaxation modulus

| $g_i$ | $\tau_i$ |
|---|---|
| 0.008 | 10000 |
| 0.009 | 8000 |
| 0.01 | 5000 |
| 0.01 | 600 |
| 0.06 | 50 |
| 0.1 | 5 |
| 0.015 | 1 |

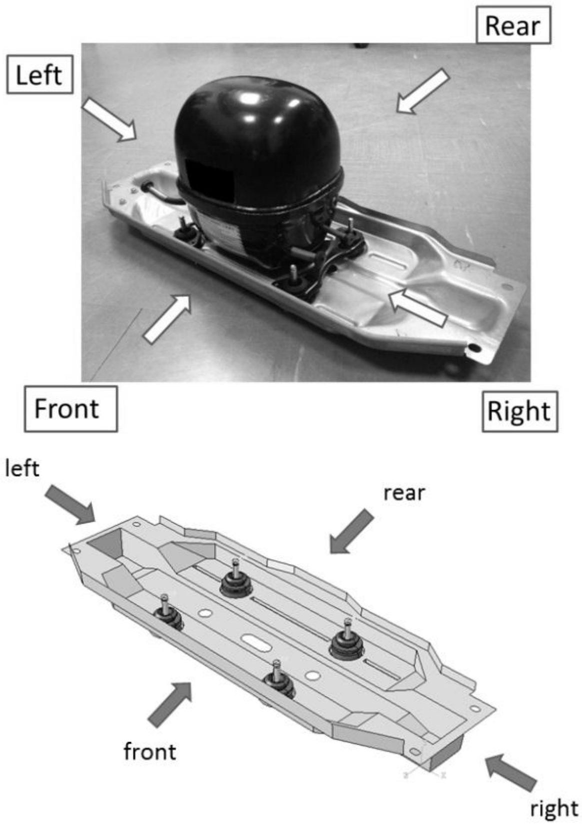


Fig. 8 Simulations of the grommets

## 5. Results and Discussion

The averaged transmissibility curve of the compressor-grommet system is shown in Fig 9. A representative input time-domain accelerations representing the steady state response of the compressor is plotted in Fig 10, and the corresponding experimental measured output and simulation result is shown in Fig 11.

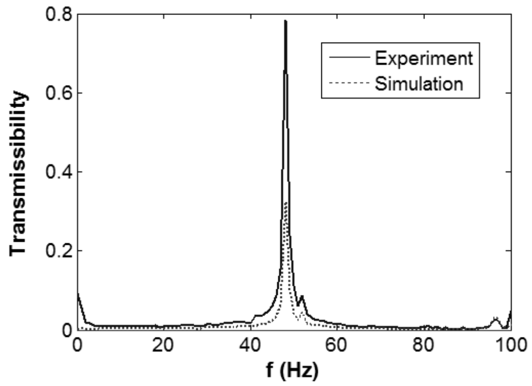


Fig. 9 Transmission curves between the experiment and the simulation

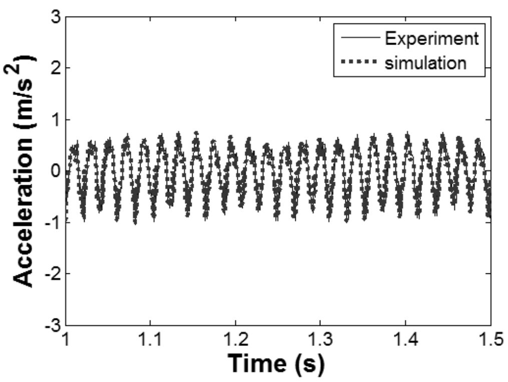


Fig. 10 Input signal at front-left grommet

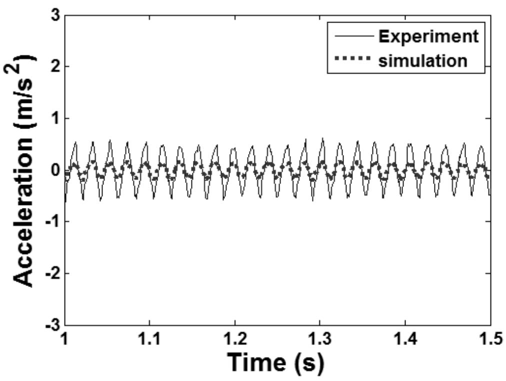


Fig. 11 Output signal at front-left grommet

The output plots show that the measured outputs and the simulation responses are all in-phase, suggesting that the simulation can accurately capture the motion of the compressor system. Yet the simulations always under-predicts the magnitudes of the outputs because the stiffness calculated in the simulation is obtained from a quasi-static tensile test but the vibration is a dynamic behavior. The differences in magnitude are further illustrated in terms of the root mean square (RMS) of the accelerations in Table 2.

Table 2 RMS amplitudes of the input and output accelerations of the measured signals and the simulation.

| Grommet Position | RMS Input signals ($m/s^2$) | | |
|---|---|---|---|
| | Test | simulation | % error |
| Front-left | 0.471 | 0.440 | -7.10 |
| Front-right | 0.813 | 0.742 | -9.59 |
| Rear-left | 0.619 | 0.576 | -7.43 |
| Rear-right | 0.725 | 0.663 | -9.34 |
| | RMS Output signals ($m/s^2$) | | |
| | Test | simulation | % error |
| Front-left | 0.344 | 0.144 | -58.11 |
| Front-right | 0.420 | 0.153 | -63.64 |
| Rear-left | 0.583 | 0.128 | -78.12 |
| Rear-right | 0.532 | 0.154 | -70.96 |

In the design of vibration isolator, the isolator should have a natural frequency ($\omega_0$) that is much lower than the working frequency ($\omega$) and a high damping so that the transmissibility is as low as possible. The natural frequency is related to the stiffness $K$ and the mass $m$ of the isolator by $\omega_0 = \sqrt{\frac{K}{m}}$. On the other hand, the damping property of a grommet is independent of the geometry, and is solely dependent on the hysteresis property of the material. Therefore, a grommet with an improved transmissibility can be designed by decreasing $K$ or by increasing $m$. However, the objective of the industry is to reduce the mass of the grommet for cost saving. The grommet design therefore must balance between a decrease in $K$ and a decrease in $m$.

Two new grommet designs are created and analyzed using the same simulation framework. All modifications are made to the base of the grommet. The 1st design attempts to reduce the mass as well as the stiffness simultaneously. In comparison, the 2nd design attempts to reduce the mass but maintains the stiffness by keeping the same stiffness. Fig. 12 and 13 show the 1st and 2nd grommet designs, respectively.

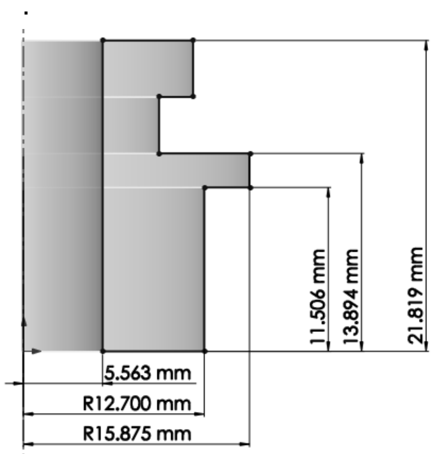


Fig. 12 1st new grommet design

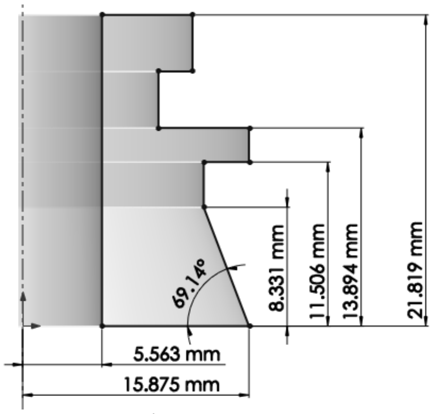


Fig. 13 2nd new grommet design

The mass and the vibration responses of the original design and the two new designs are summarized in Table (3). Both new designs require less mass and can absorb the vibration better than the original model. However, the grommets with low stiffness are not always desirable because they might be able to sustain the weight of the compressor. Further, a large deformation of rubber leads to a shorter fatigue life. The product life cycle of the original grommet design and the two new designs must be further evaluated.

Table 3 Responses of new grommet designs

| Grommet Position | output acceleration(m/s2) | | | | |
|---|---|---|---|---|---|
| | original | 1st design | % change | 2nd design | % change |
| Front-left | 0.144 | 0.118 | -2.84 | 0.140 | -17.96 |
| Front-right | 0.153 | 0.132 | -1.83 | 0.150 | -14.00 |
| Rear-left | 0.128 | 0.113 | -1.33 | 0.126 | -11.61 |
| Rear-right | 0.154 | 0.128 | -2.66 | 0.150 | -17.23 |
| Mass (g) | 12.8 | 9.99 | -21.95 | 11.3 | -11.72 |

## 6. Conclusion

A finite element simulation has been created to analyze the vibration response of a compressor-grommet system. The simulation framework is applied in the analyses of two new grommet designs. The results show that the new design can have a similar characteristic of vibration while requiring less mass than the original grommet.

## 7. Acknowledgement

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