Material characteristics of a vehicle door seal and its effect on vehicle vibrations

E. Dikmen *, I. Basdogan *

* Department of Mechanical Engineering, Koc University, Istanbul, Turkey

First Published: November 2008

To cite this Article


To link to this Article: DOI: 10.1080/00423110701689610
URL: http://dx.doi.org/10.1080/00423110701689610

PLEASE SCROLL DOWN FOR ARTICLE
Material characteristics of a vehicle door seal and its effect on vehicle vibrations

E. Dikmen and I. Basdogan*

Department of Mechanical Engineering, Koc University, Istanbul, Turkey

(Received 4 April 2007; final version received 18 September 2007)

The vibration characteristics of the door panels are affected by the weatherstrip seals used in between the doors and vehicle body along the perimeter of the doors. The weatherstrip seals exhibit nonlinear and viscoelastic material properties that vary with frequency, temperature, strain rate and amplitude, and previous load history. The material properties of the seal must be investigated carefully in order to predict the vibration characteristics of the automobiles under different loading conditions.

In this study, we developed hyperelastic and viscoelastic models of the weatherstrip seal to predict dynamic performance of a vehicle door and its effect on the overall vehicle dynamics. For this purpose, first, static compression and stress relaxation experiments were performed on the seal using a robotic indenter equipped with force and displacement sensors and then a finite element model utilising the results of these experiments was developed in ANSYS. Finally, a representative model of the seal was integrated into the finite element model of the vehicle door to investigate its effect on the vehicle vibrations. The model predictions were validated using experimental modal analysis performed on the vehicle door with and without the seal. It was observed that the seal has a significant effect on the vehicle dynamics.

Keywords: weatherstrip seal; hyperelasticity; viscoelasticity; vehicle dynamics

1. Introduction

Automotive weatherstrip seals are used in between the doors and vehicle body along the perimeter of the doors. The seals can be held in place by various methods such as intermittent push pins, continuous carriers, and flange mounts. They are mainly used to prevent water and dust entrance to the passenger compartment in all weather conditions and accommodate for the manufacturing variations.

The weatherstrip seal affects the door vibrations considerably by changing its stiffness and viscoelastic properties. Hence, accurate representation of the door seals is crucial in the simulation models used for predicting the dynamics of the vehicle. For this purpose, the material properties of the seal must be characterised properly.

The weatherstrip seal is generally in the form of dual extrusion bulbs of sponge and dense rubber. The bulbs can have different shapes generally with a height of approximately
15–30 mm. The bulb wall thickness is typically a few millimetres to provide maximum sealing area at a low compression force [1]. The rubber used in the seal can withstand large deformations without permanent deformation and have high damping characteristics. The mechanical properties of the rubber may vary with the amount of deformation, previous load history, temperature, frequency and amplitude of the motion in the presence of mechanical vibrations.

Several groups have conducted experiments with rubber used in the automotive industry to characterise its mechanical properties. Lin et al. [2] presented a simple experimental method to evaluate the frequency dependent stiffness and damping characteristics of rubber mount. Kren and Vriend [3] used dynamic indentation tests in order to determine the viscoelastic properties of rubber. Fenander [4] measured the vertical stiffness and damping of studded rubber railpads, both in a complete track and in a test rig, as a function of frequency under different static preloads. The loss factor of the studded railpads was found to be nearly independent of preload and to increase slightly with frequency. Wagner et al. [1] investigated the deflection response of automotive weatherstrip seal under compressive loading (i.e. compression load deflection – CLD – response). They also investigated the contact pressure distribution and aspiration (loss of contact between the weatherstrip seal and the facing sheet metal surface) due to a pressure differential across the seal. They modelled the seal using Blatz–Ko and Mooney–Rivlin hyperelastic material models and compared the resultant CLD response with the linear response. Stenti et al. [5] developed a simplified model of car door weatherstrip seal and performed nonlinear static and dynamic analysis on the model using commercial finite element code MSC Marc. The CLD response was obtained via static analysis and the effect of the seal on the vibration modes of the door was investigated using dynamic analysis. Lu et al. [6] conducted experiments with a rubber mount and developed a nonlinear finite element model (FEM) to investigate its large deformation behaviour. The results of the experiments agreed well with the results obtained from the FEM. A test structure imitating the vehicle door and the body was constructed [7]. Natural frequencies of the structure were obtained experimentally with and without seals lying between the two frames. It was verified that the seal caused a frequency shift in the test structure. Simulations of the both cases were performed by finite element analysis and an equivalent spring coefficient was determined for the weatherstrip seal. Valenta and Molnar [8] compared Mooney–Rivlin and Neo–Hook hyperelastic material models for silicone rubber using Marc Mentat FEM software. Gur and Morman [9] used nonlinear finite element analysis to determine the conditions favourable to seal system aspiration. They investigated the effects of initial seal height, shape, thickness, constitutive model, friction, and compression due to door closing on aspiration (the loss of contact between the seal and facing metal surface of the door).

In this study, we developed a methodology in order to characterise the viscoelastic and hyperelastic material properties of the weatherstrip seal using experimental methods and finite element modelling techniques. The weatherstrip seal investigated in this study is made of ethylene propylene diene monomer (EPDM) sponge rubber. We conducted static compression and stress relaxation experiments using a robotic indenter equipped with force and displacement sensors to characterise the hyperelastic and viscoelastic behaviour of the seal. An inverse solution was developed in ANSYS to obtain the hyperelastic and viscoelastic material coefficients. For this purpose, a finite element model of the seal geometry was constructed and the material coefficients of the model were determined through the optimisation toolbox in ANSYS. The solutions were iterated until the force–displacement and force–time responses of the FEM show a good agreement with the experimental data collected using the robotic indenter.

In order to determine the effect of weatherstrip seal and boundary conditions on door dynamics, an FEM of the vehicle door was constructed using an equivalent linear spring model of the weatherstrip seal. The simplified spring model allows the weatherstrip seal to be readily represented in the full scale vehicle FEM. It also reduces the computational cost.
significantly and makes it possible to perform numerical modal analysis of the door and the vehicle itself. The results obtained from the FEM were compared with ones obtained from the experiments performed on the actual vehicle door in two different settings. In the first experimental setting, the vehicle door was hung from a supporting frame with elastic cords to simulate the free-free boundary conditions. In the second one, the doors were mounted to the vehicle body and the whole structure was suspended through air springs. The effect of the weatherstrip seal on the vibrations of the vehicle door and the vehicle itself was investigated using experimental and numerical modal analysis techniques.

2. Nonlinear material modelling

Rubber exhibits nonlinear force versus displacement response characterised by hyperelastic models. One of the challenges in modelling rubber-like materials is the selection of appropriate hyperelastic material model among many options. Each hyperelastic model is described with a different strain energy function [1, 5].

This section summarises the procedure that we followed to determine the most suitable hyperelastic model of the weatherstrip seal. We first developed a nonlinear FEM of the seal in ANSYS using different strain energy functions. The material coefficients of these functions were calculated via curve fitting to the experimental data obtained from the manufacturer of the seal. This data includes the results of tension, shear, and compression tests performed on the seal ‘material’ under well-defined conditions without considering the effect of seal geometry. After the coefficients were estimated via curve fitting, the compression test was simulated in ANSYS. The CLD responses of the seal for different strain energy functions were obtained by increasing the load incrementally and recording the displacement of the seal at each time step. These results were then compared with the results of the experimental compression test performed on the ‘actual’ seal geometry using our robotic indenter to find the best match. Note that the manufacturer’s data is only available for the seal material and it is necessary to conduct experiments with the actual seal geometry since the geometric factors are known to play an important role in large deformation analysis.

2.1. The FEM model of the weatherstrip seal

A meshed geometric model of the seal was constructed (see Figure 1) and then imported to ANSYS in initial graphics exchange specification (IGES) format. A 2D FEM of the seal was developed with the plane strain assumption since there was no deformation in the depth.
direction. Using a 2D model also reduces the number of FEM computations drastically during the inverse analysis. Plane 182 elements having hyperelastic, viscoelastic, large deflection, and large strain capabilities were used to construct the FEM.

2.2. Hyperelastic material models

Hyperelasticity refers to the materials which can experience large elastic strain that is recoverable [10]. Elastomers such as rubber and many other polymer materials fall in this category. The constitutive behaviour of hyperelastic materials are usually derived from the strain energy potentials. The hyperelastic material models assume that materials’ response is isothermal. This assumption allows that the strain energy potentials are expressed in terms of strain invariants or principal stretch ratios. Stretch ratio is basically defined for uniaxial tension as

\[ \lambda = \frac{L}{L_0} = 1 + \varepsilon_E, \]

where \( L \) and \( L_0 \) are the final and initial lengths and \( \varepsilon_E \) is the engineering strain. There are three principal stretch ratios \( \lambda_1, \lambda_2, \lambda_3 \) which can be used to define the strain energy potential. The principal stretch ratios \( \lambda_1 \) and \( \lambda_2 \) characterise in-plane deformation while \( \lambda_3 \) defines the out-of-plane deformation. Strain invariants are measures of strain which are independent of the coordinate system used to measure the strains. Typically, three strain invariants are used in strain energy functions which are defined in terms of principal stretch ratios as

\[ I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2, \quad I_2 = \lambda_1^2\lambda_2^2 + \lambda_2^2\lambda_3^2 + \lambda_3^2\lambda_1^2, \quad \text{and} \quad I_3 = \lambda_1^2\lambda_2^2\lambda_3^2. \]

Constitutive behaviour of hyperelastic materials are generally derived from the strain energy potentials. The strain energy potentials, which are generally denoted as \( W \), are functions of principal stretch ratios or strain invariants. Volumetric (with subscript \( b \)) and deviatoric (with subscript \( d \) and bar) terms of the strain energy function for incompressible materials can be separated as shown below:

\[ W = W_d(I_1, I_2) + W_b(J) \]  
\[ W = W_d(\bar{\lambda}_1, \bar{\lambda}_2, \bar{\lambda}_3) + W_b(J), \]

where \( J \) is the ratio of the final volume to the initial volume and \( \lambda_p = J^{-1/3}\lambda_P \), and \( \bar{I}_p = J^{-1/3}I_P \) for \( p = 1, 2, \) and \( 3 \).

There are many different hyperelastic material models. The first step in our approach was to determine the most suitable hyperelastic model of the seal for the subsequent analyses. For this purpose, we first determined the material coefficients of the hyperelastic models via curve fitting to the experimental data on seal material (obtained from the manufacturer) without paying attention to the seal geometry (note that we also performed experiments with the actual seal geometry, which is discussed later). The experimental data obtained from the manufacturer includes the results of simple tension, compression and shear tests performed on a sample of EPDM sponge rubber. The material coefficients obtained for different strain energy potential functions are summarised in Table 1 with the estimated coefficients. Note that the volumetric components of the strain energy functions (\( W_b \)) were neglected due to the unavailability of the data on volume change. A brief description of the hyperelastic material models used in our analysis is as follows:

1. Mooney–Rivlin model: the Mooney–Rivlin model is simple in formulation, therefore it is widely used in many hyperelastic applications. Two-, three-, five-, and nine-term Mooney–Rivlin models are available in ANSYS. The two-term Mooney–Rivlin model is valid up
to 90–100% tensile strains, but is not very accurate to simulate the stiffening behaviour at large strains. The behaviour of the material under compressive loading may also not be characterised well with only the two-term Mooney–Rivlin model [10]. The strain energy function for the two-term Mooney–Rivlin model is

\[ W = c_{10}(I_1 - 3) + c_{01}(I_2 - 3), \]  

where \( c_{10} \) and \( c_{01} \) are the material coefficients. The material constants \( c_{10}, c_{01} \) were calculated via curve fitting to the experimental data in ANSYS.

(2) **Arruda–Boyce model**: the Arruda–Boyce form is a statistical mechanics–based model and generally limited to 300% strain [10]. The strain energy function for the Arruda–Boyce model is

\[ W = \mu \sum_{i=1}^{5} \frac{C_i}{\lambda_i^{2i-2}}(I_i^i - 3^i), \]  

where the constants \( C_i \) are defined as \( C_1 = 1/2, C_2 = 1/20, C_3 = 11/1050, C_4 = 19/7050, C_5 = 519/673750. \) The material constants \( \mu \) and \( \lambda_L \) were determined via curve fitting to the experimental data in ANSYS.

(3) **Ogden model**: the Ogden form is directly based on the principal stretch ratios rather than the strain invariants and defined as

\[ W = \sum_{i=1}^{N} \frac{\mu_i}{\alpha_i} (\lambda_1^{\alpha_i} + \lambda_2^{\alpha_i} + \lambda_3^{\alpha_i} - 3). \]

The Ogden model may be applicable for strains up to 700% [10], but it is computationally more expensive to implement than some of the other models. The material coefficients \( \mu_1 \) and \( \alpha_1 \) were estimated for the first-order Ogden model via curve fitting to the experimental data in ANSYS.

(4) **Blatz–Ko model**: the Blatz–Ko model is more commonly used for modelling compressible foam–type rubbers. The strain energy potential function of the Blatz–Ko model is

\[ W = \frac{\mu}{2} \left( \frac{I_2}{I_3} + 2\sqrt{I_3} - 5 \right), \]  

where \( \mu \) is a material constant and was determined by curve fitting.

(5) **Gent model**: the strain energy function for the Gent model is:

\[ W = \frac{\mu}{2} \frac{J_m}{J_m} \ln \left( 1 - \frac{I_1 - 3}{J_m} \right)^{-1}, \]

where the material constants \( \mu \) and \( J_m \) were determined via curve fitting.
2.3. **Static compression experiments**

To find the best matching hyperelastic model to the actual behaviour of the seal, real compression experiments were performed with the seal itself using a robotic indenter and the results were compared with the numerical compression experiments performed with the seal geometry in ANSYS. A robotic arm equipped with position sensors was used to compress the seal (Figure 2). A flat plate was attached to the tip of the arm as an indenter. A PID controller was implemented to move the indenter from an initial position to a desired position in 3D space in discrete time steps at a slow rate of 0.05 mm/s (to minimise the dynamic effects) while compensating for the positional errors. A force transducer (Nano 17 from ATI Industrial Automation) was attached to the robotic arm for the purpose of measuring the force response of the seal during compression. The Nano 17 has a force range of ±70 N in the normal direction, ±50 N in other principal directions and has a resolution of 1/1280 N along each of the three orthogonal axes when attached to a 16-bit A/D were converter. The force and position data were acquired using a 16-bit analogue input card NI PCI-6034E (National Instruments) with a maximum sampling rate of 200 kS/s [11]. The compression experiments were repeated many times and the results were averaged in order to minimise the experimental errors.

The results of the static compression experiment for the seal sample were compared with the numerical simulations performed in ANSYS. The material coefficients estimated for the different strain energy functions were used in ANSYS to conduct numerical compression experiments with the FEM of the seal. A rigid line was used to compress the seal geometry uniformly in 2D. A total displacement of 14 mm was applied to the line and the force response of the seal was recorded to a file. The seal was compressed with a speed of 0.05 mm/s in ANSYS to generate force–displacement data in similar conditions to the actual compression experiments.

Figure 3 shows the results of the FEM simulations and the compression experiment performed with the robotic indenter. The hyperelastic model that shows the best agreement (based on minimum error criteria) with the experimental data was the Arruda–Boyce model (see Figure 3). This model is used in the subsequent analysis of the viscoelastic behaviour of the seal.

![Figure 2. Experimental setup for compression tests.](image-url)
Figure 3. Static compression experiment compared with the numerical simulations using different hyperelastic models.

3. Viscoelastic material modelling

Rubber-like materials such as the weatherstrip seal show viscoelastic behaviour. Viscoelasticity is a rate-dependent behaviour where the material properties may be both time- and temperature-dependent. In order to characterise the viscoelastic behaviour of the weatherstrip seal, stress relaxation experiments were conducted. The robotic indenter was programmed to compress the seal to a predefined indentation depth in 1 s and then it was held there for 40 s while the force relaxation response of the weatherstrip seal was recorded as a function of time. Stress relaxation experiments for the indentation depths of 4 and 8 mm (corresponding to the compression speeds of 4 and 8 mm/s) are shown in Figure 4. It is observed from Figures 4a and 4b that the seal exhibits viscoelastic behaviour and the force response is influenced by the rate of loading.

ANSYS utilises the Prony series representation of the stress relaxation function to model viscoelasticity. The stress expression for large strain viscoelasticity based on the two-term Prony series can be written as:

\[ S = \int_{0}^{t} \left[ \alpha_\infty + \sum_{j=1}^{2} \alpha_j \exp \left( -\frac{t - \tau}{\tau_j} \right) \right] \left( 2 \frac{d}{d\tau} \frac{dW}{dC} \right) d\tau, \]  

where \( \alpha_1, \tau_1, \alpha_2, \tau_2 \) are the viscoelastic material coefficients (note that \( \alpha_\infty = -(\alpha_1 + \alpha_2) \)), \( W \) is the strain energy function (note that the Arruda–Boyce model was used based on the results of the static compression experiments) and \( C \) is the right Cauchy–Green deformation tensor.

4. Inverse solution

In order to estimate the hyperelastic (\( \mu \) and \( \lambda_L \)) and viscoelastic (\( \alpha_1, \tau_1, \alpha_2 \) and \( \tau_2 \)) material coefficients of the weatherstrip seal in tandem, an inverse solution must be developed.
The characterisation of material properties based on experimental data is considered as the inverse problem. Figure 5 illustrates the steps of our inverse solution. We used the optimisation toolbox of ANSYS to solve the inverse problem. In order to perform an optimisation analysis in ANSYS, three set of variables must be defined. The first set consists of independent quantities called design variables (DVs) to be determined by the optimisation algorithm. The second set contains state variables (SVs) constraining the model and, finally, a dependent variable (objective function) to be minimised must be determined. To calculate the hyperelastic and viscoelastic parameters of the weatherstrip seal, $\mu$, $\lambda$, $\alpha_1$, $\tau_1$, $\alpha_2$ and $\tau_2$ were assigned as DVs. Force data samples representing the relaxation behaviour were chosen from the experimental
force relaxation data and then compared with the force values (i.e. SVs) obtained from the FEM solution. We defined the objective function as the difference between the experimental stress relaxation response and the one obtained from the ANSYS solution.

The optimisation algorithm minimises the objective function defined as:

$$\text{Error} = \sum_{j=1}^{n} (F_{j}^{\text{EXP}} - F_{j}^{\text{FEM}})^2,$$

where, $F_{j}^{\text{EXP}}$ is the experimental force value of $j$th data point, $F_{j}^{\text{FEM}}$ is the force value obtained from the FEM simulation at the corresponding time, and $n$ is the number of force data samples. To obtain $F_{j}^{\text{FEM}}$, the stress relaxation experiments were simulated in ANSYS using the finite element model of the seal having hyperelastic and viscoelastic characteristics. The hyperelastic material behaviour was modelled using the Arruda–Boyce strain-energy function and the viscoelasticity was modelled using the two-term Prony series. The hyperelastic coefficients estimated from the static compression tests in Section 2 were used as initial guesses for the optimisation iterations. The initial guesses for viscoelastic models were selected arbitrarily. The solutions were iterated until the total error between the experimental and numerical force values was less than a predefined threshold value.

Experimental force relaxation curve for 8 mm indentation (strain rate is 8 mm/s) and the FE simulation with the material properties obtained from the inverse FE solution are shown in Figure 6a. In order to further justify the inverse solution, the seal was compressed up to 14 mm at a rate of 4 mm/s (considering the dynamical effects) using the robotic indenter and the force–displacement response was compared with the one obtained from the numerical solutions (Figure 6b). Figure 6b demonstrates that the coefficients obtained from the finite element
Figure 6. (a) Stress relaxation experiments and finite element simulation at a rate of 8 mm/s. (b) Compression experiment at a rate of 4 mm/s using the coefficients obtained from the 8 mm/s relaxation experiment.

solution at 8 mm/s provides a good match between the experimental and numerical results even for a different rate of loading at 4 mm/s. The hyperelastic and viscoelastic coefficients calculated by the inverse solution method for 4 mm/s and 8 mm/s compression speeds are tabulated in Table 2.

5. Effect of the seal on vehicle vibrations

In order to determine the effect of the weatherstrip seal and boundary conditions on the door dynamics, we performed experimental and numerical modal analysis. As discussed in the earlier sections, the weatherstrip seal shows nonlinear and viscoelastic material behaviour. However, performing modal analysis using hyperelastic models is not possible in FEM
Table 2. Hyperelastic and viscoelastic coefficients calculated by the inverse solution.

<table>
<thead>
<tr>
<th></th>
<th>Compression speed @ 4 mm/s</th>
<th>Compression speed @ 8 mm/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>α₁</td>
<td>0.11198</td>
<td>0.1499</td>
</tr>
<tr>
<td>α₂</td>
<td>0.80049E-01</td>
<td>0.08</td>
</tr>
<tr>
<td>τ₁</td>
<td>1.8583</td>
<td>1.8583</td>
</tr>
<tr>
<td>τ₂</td>
<td>9.98</td>
<td>9.98</td>
</tr>
<tr>
<td>μ</td>
<td>0.20629</td>
<td>0.1525</td>
</tr>
<tr>
<td>λₐL</td>
<td>1.9813</td>
<td>1.9851</td>
</tr>
</tbody>
</table>

Performing these simulations is also computationally very expensive, especially when dynamical analysis of the whole vehicle is required. For these reasons, we used a linear spring model of the seal in our numerical modal analysis. The stiffness coefficient of the seal used in the FEM of the vehicle door was obtained from the CLD data performed on the real seal geometry. A fifth order polynomial was fitted to the experimental CLD curve given in Figure 3. The polynomial function was differentiated to calculate the stiffness values at various compression depths. Figure 7 shows the tangent stiffness values for 6–14 mm displacements of the weatherstrip seal, which is known to be the compression range of the seal when the seal is in place between the door and the vehicle body.

5.1. Experimental modal analysis of the door

In order to determine the effect of the weatherstrip seal and boundary conditions on the door dynamics, modal analysis experiments were performed in two different configurations of the vehicle door. In the first configuration, the door was hung from a supporting frame with elastic cords to simulate the free-free boundary conditions. In the second one, doors were mounted to the vehicle body and the whole structure was suspended through air springs. Air springs in the second configuration were adjusted such that they had low natural frequencies to satisfy...

Figure 7. Stiffness values obtained by differentiating the experimental CLD curve for 6–14 mm compression range.
the free-free condition. Therefore, the only difference between these two configurations is the boundary conditions such as the hinge connections and the locks. The two configurations are shown in Figure 8.

Frequency shifts and damping contribution of weatherstrip seals on the door were investigated by performing experiments with and without the seal in the second configuration. The effect of boundary conditions was investigated by comparing the results of the experiments in the first and the second configurations without the seal. The experimental setup included a laser Doppler vibrometer (LDV) that was used to measure vibration velocities of the door at different locations on the structure (see Figure 8). The structure was excited at a wide range of frequencies via an electrodynamic shaker and the velocity data measured by the LDV was collected through the use of a data acquisition system. The transfer functions that relate the excitation input to the velocity output were measured at various locations of the structure and then transferred to a modal analysis software, ME’ScopeVES [12], to extract the modal parameters.

5.1.1. Results of experimental modal analysis

The first set of modal analysis experiments was performed in order to determine the effect of the boundary conditions on the door dynamics. For this purpose, first and second configurations were used and the seals were removed from all of the contacting surfaces in order to eliminate the seal effect on the door dynamics. Figure 9 illustrates the first mode shape of the door in the first and second configurations. Boundary conditions changed the first natural frequency from 37.89 Hz to 32.93 Hz. Due to the effect of hinge connections around the points 63 and 58 shown in Figure 9b, the motion of the left and right corners of upper part of the door disappeared compared to the mode shape in Figure 9a.

The second set of modal analysis experiments was performed in order to determine the effect of the seal on the door dynamics. For that purpose, the second configuration was used with and without the seal. The effect of the weatherstrip seal in door dynamics can be seen in Figure 10. The frequency of the first mode dropped considerably from 42.06 Hz to 32.93 Hz when the seal was removed. However, the effect on the mode shape was not significant. The seal behaves like a spring, increases the overall stiffness of the door and hence affects its dynamics considerably. The frequency shift of the second mode was less than the first mode,

Figure 8. Two configurations used for the experimental modal analysis of the door. (a) Door hung freely. (b) Door mounted to the vehicle body.
which changed from 64.88 Hz to 61.44 Hz when the seal was removed (see Figure 11). The earlier studies on the same subject also report that the effect of seal on door dynamics decreases in higher frequencies [1,7]. Moreover, it is observed that the small amplitude vibrations at the upper and lower right corners of the door damped out when the seal was in place. Table 3 illustrates the results of the experimental modal analysis with different boundary conditions and configurations.

### 5.2. Results of finite element simulation of vehicle

An FEM of the half vehicle body was constructed (see Figure 12) in order to verify the linear spring assumption used for the weatherstrip seal. A linear spring model based on the results of
Table 3. The results of experimental modal analysis performed on the vehicle door.

<table>
<thead>
<tr>
<th></th>
<th>Free-free condition</th>
<th>With the seal</th>
<th>Without the seal</th>
</tr>
</thead>
<tbody>
<tr>
<td>First mode</td>
<td>37.89 Hz</td>
<td>42.06 Hz</td>
<td>32.93 Hz</td>
</tr>
<tr>
<td>Second mode</td>
<td>75 Hz</td>
<td>64.88 Hz</td>
<td>61.44 Hz</td>
</tr>
</tbody>
</table>

Figure 12. Finite element model of the half vehicle body.

The compression experiment (Figure 7) was used to represent the weatherstrip seal in the FEM of the vehicle. For the static compression range shown in Figure 7, the equivalent stiffness coefficient of the seal goes up to $k = 2000 \text{ N/m}$ for a 100 mm sample. Stiffness values of 500, 1000, and 2000 N/m were used in simulations and the highest spring coefficient at 2000 N/m provided the best results when the frequencies and the mode shapes from the experiments and numerical simulations were compared. Figure 13 compares the experimental modal analysis results with the ones obtained from the finite element analysis at 40 Hz. Figure 13a shows

Figure 13. Mode shapes of the door attached to the vehicle body. (a) FEM simulations. (b) Experimental modal analysis.
the finite element simulation of the half vehicle body obtained using the spring coefficient of 2000 N/m for the seal. When it is compared to Figure 13b, the right door experimental modal analysis results at 40 Hz, it can be observed that the mode shape of the right door can be predicted accurately when the highest stiffness value is used in the simulations.

6. Discussion and conclusion

In this study, an FEM was used to determine the CLD behaviour of a vehicle weatherstrip seal for different hyperelastic models and the model predictions were compared with the experimental results obtained using a robotic indenter. The Arruda–Boyce model was found to be the most suitable strain energy function for modelling the hyperelastic material properties of the weatherstrip seal. An inverse FE solution was developed to determine the viscoelastic material properties of the weatherstrip seal in tandem with the hyperelastic coefficients through optimisation iterations. This approach was verified by comparing the results of the experiments with forward FEM solutions for different loading rates. Characterising the viscoelastic and hyperelastic material properties of the weatherstrip seal is crucial, especially when detailed modelling of the seal is required to investigate the dynamics of the seal itself. This experimentally verified FEM displaying hyperelastic and viscoelastic behaviour of the seal can be used to investigate the door closing effects and the aspiration conditions in the future.

However, modelling the weatherstrip seal using the hyperelastic and viscoelastic models is not practical when the whole vehicle dynamics is investigated. The detailed modelling of the seal increases the computational cost significantly. Besides, performing the numerical modal analysis with nonlinear elastic models is not feasible with the FEM packages. For that purpose, we calculated the equivalent spring coefficient of the seal for different compression depths using the experimental CLD curve. The stiffness values for 6–14 mm compression range were calculated and used in the FEM simulations. The results of the numerical computations were compared with the experimental modal analysis results in order to determine the best value of the spring coefficient for the seal.

Experimental modal analysis results were also used to demonstrate the effect of the seal and boundary conditions on the vehicle dynamics. Boundary conditions both altered the natural frequency and the mode shape of the door. In addition, it was observed that the weatherstrip seal contributes to the door dynamics in the first mode more than the second mode.

Acknowledgements

The authors kindly acknowledge the support of the Scientific and Technological Research Council of Turkey (TUBITAK) and Ford Otosan AS. We also would like to acknowledge the technical support given by Mert Doganli from Ford Otosan AS and the technical discussions made with Tuncal Yuksel from Standard Profil AS.

References


