

Prediction of Automotive Side Swing Door Closing Effort

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ABSTRACT

The door closing effort is a quality issue concerning both automobile designers and customers. This paper describes an Excel based mathematical model for predicting the side door closing effort in terms of the required minimum energy or velocity, to close the door from a small open position when the check-link ceases to function. A simplified but comprehensive model is developed which includes the cabin pressure (air bind), seal compression, door weight, latch effort, and hinge friction effects. The flexibility of the door and car body is ignored. Because the model simplification introduces errors, we calibrate it using measured data. Calibration is also necessary because some input parameters are difficult to obtain directly. In this work, we provide the option to calibrate the hinge model, the latch model, the seal compression model, and the air bind model. The door weight effect is geometrically exact, and does not need calibration. The capabilities and accuracy of the developed model are demonstrated using the front and rear doors of a production vehicle.

INTRODUCTION

This paper describes a mathematical model to predict the automotive door closing effort. It provides the

minimum required velocity and energy of a vehicle's side swing door between secondary check position and closed position. The model is implemented using visual basic in Microsoft EXCEL. The main effects we consider, are 1) cabin pressure which depends on door area, cabin volume and vehicle vent area, 2) seal compression loading, 3) door weight due to the inclination of the hinge axis, 4) latch effort, and 5) hinge friction.

The door closing effort is one of the quality issues concerning both automobile designers and customers. It gives a first impression on the quality of the vehicle. If a large effort is needed to close the door, the vehicle is usually perceived of poor quality, which is directly linked to a "not pleasing" door closing sound. It is usually necessary to reduce the door closing effort without sacrificing the weather sealing and the acoustic insulation requirements. The door closing effort problem has been studied in the automotive industry. Early works in [1, 2] provide an indication of the current state-of-the-art. The main objective of this work is to predict the energy required to close the door from a small open position when the check-link is assumed to cease functioning (see Figure 1).

Both the door and car body are considered rigid in the analysis. However, the door header deflection is accounted for, indirectly. The header is purposely bent inward to consider the seal pressure effect once the door is closed. The door closing energy mainly consists of five components; the energy losses due to cabin pressure, seal loading, door weight, latch effort, and hinge friction. The first component is caused by the pressure rise in the cabin when the door pushes the air ahead of itself. The second is due to the seal compression, and it represents the work to compress the seal. The third loss takes into account the energy contributed by the door weight when the hinge axis is tilted and therefore, is not vertical. The latching effort and frictional losses at the hinges, are also considered. The non-linear interaction of the door gap, seal load, door weight, latch effort, and hinge friction is calculated using an incremental time stepping numerical scheme.

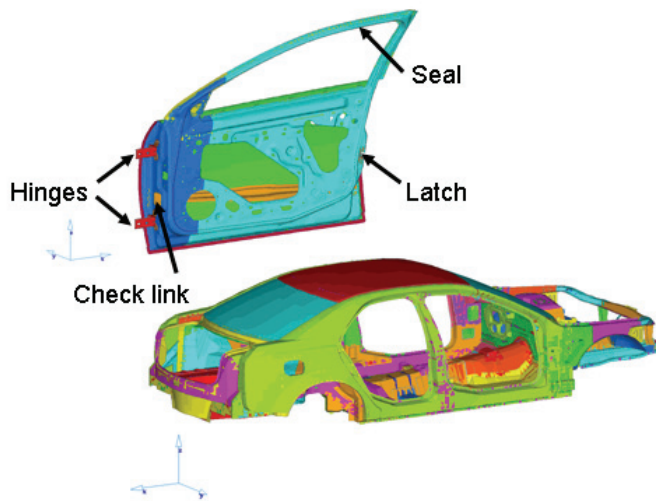


Figure 1. Finite element model of a door and car body [1]

Measurement-based calibration is indispensable because some parameters are difficult to obtain directly and errors are introduced due to the model simplification. For this reason, we provide the option to calibrate the hinge, latch, seal compression, and air bind models. Because the weight effort model is geometrically exact, it does not need calibration.

An example shows that the developed model has the potential to provide a rational prediction with or without full calibration, assuming that the input data is complete and consistent. To our knowledge, the presented model is comprehensive, and close to the current state-of-the-art in predicting the automotive side swing door closing effort. The model can be easily used in parametric studies during the design process of a door closure system.

MODEL DESCRIPTION – ENERGY SINKS

The energy sinks are due to cabin pressure (air bind), seal compression, door weight, latch effort, and hinge friction.

AIR BIND

The energy loss due to air bind is a substantial contributor to the overall door closing energy. The closing door pushes the air ahead of itself and creates a pressure rise in the vehicle that must be overcome. To predict this effect, we use a simplified one-dimensional ideal fluid dynamic model as shown in Figure 2. The pressure and density at the door are assumed to be the same with those in the cabin. The cabin is assumed to be a tube with the same sectional area with the door. The actual air vent area includes the contribution from the door gap.

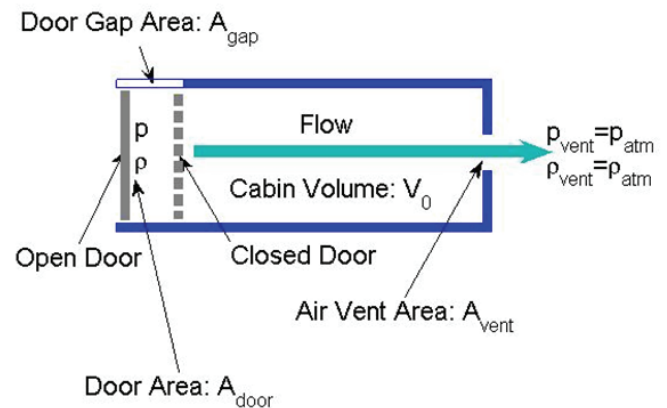


Figure 2. Simplified diagram for air bind

The flow rate q depends on ω , the angular velocity or rotational speed of the closing door, while the door gap area A_{gap} is a function of the door opening angle θ . Based on conservation of mass, the ideal gas law, and Bernoulli's equation relating the flow rate with the corresponding pressure change, one can determine the pressure in the vehicle that will act on the inner surface of the door during closure. The equation of state for the air is

$$p = RT\rho \quad \text{or} \quad pV = RTm \quad (1)$$

where p is the pressure, $\rho = \frac{m}{V}$ is the air density, m and V are the air mass and volume respectively, T is the temperature, and R is the air constant. We assume that the temperature and mass are constant. According to Eq. (1), the change in the cabin pressure is proportional to the change in the cabin air volume; i.e.

$$\frac{\Delta p}{p} = -\frac{\Delta V}{V} \quad \text{or} \quad \frac{\Delta p}{p} = \frac{\Delta \rho}{\rho}. \quad (2)$$

Continuity at the door and vent opening, yields

$$\begin{aligned} \rho \int_{A_{\text{door}}} v dA &= \rho \int_{A_{\text{door}}} \omega r dA = \rho \omega \int_{A_{\text{door}}} r dA \\ &= \rho \omega I_{A_{\text{door}}} = \rho_{\text{vent}} A_{\text{vent}} v_{\text{vent}} \quad (3) \\ q_{\text{vent}} &= A_{\text{vent}} v_{\text{vent}} \end{aligned}$$

where r is the distance of a point on the door to the hinge axis, and A_{door} is the door area. The door area moment around the hinge axis is

$$I_{A_{\text{door}}} = \int_{A_{\text{door}}} r dA, \quad (4)$$

and the actual vent area A'_{vent} is

$$A'_{\text{vent}} = A_{\text{vent}} + c_{\text{vent}} A_{\text{gap}} \quad (5)$$

where A_{vent} is the interior vent area when all doors, windows, and other openings are closed, and c_{vent} is the door seal gap flow coefficient. According to Eq. (3), A_{vent} can be calculated as $A_{\text{vent}} = q_{\text{vent}} / v_{\text{vent}}$ using the leak rate q_{vent} at pressure $\Delta p = p - p_{\text{atm}} = 1$ in of water (249.1 Pa) (see Eq. 9 for the calculation of the average air speed v_{vent} at the vent). The door gap area A_{gap} is obtained by integrating the air vent gap d_{vent} around the door geometric boundary,

$$A_{\text{gap}} = \oint d_{\text{vent}}(\theta, s) ds. \quad (6)$$

Eq. (25) as explained later, provides an expression for d_{vent} .

For a time increment Δt , the change in air volume is equal to the air volume displaced by the door, minus the air volume which escapes through the vent and door gap area; i.e.

$$\begin{aligned} -\frac{V}{p} \Delta p &= \Delta V \\ &= \left[\left(1 + \frac{\Delta p}{p} \right) \omega I_{A_{\text{door}}} - A'_{\text{vent}} v_{\text{vent}} \right] \Delta t. \quad (7) \\ &= \left[\left(1 + \frac{\Delta p}{p} \right) \omega I_{A_{\text{door}}} - A'_{\text{vent}} v_{\text{vent}} \right] \Delta t \end{aligned}$$

In Eq. (7), we have used Eq. (2), and the continuity relation of Eq. (3).

Using the following Bernoulli's equation for incompressible flow

$$\begin{aligned} p &= p_{\text{vent}} + 0.5 \rho_{\text{vent}} (v_{\text{vent}})^2 \\ &= p_{\text{atm}} + 0.5 \rho_{\text{atm}} (v_{\text{vent}})^2, \quad (8) \end{aligned}$$

the air velocity at the vent is

$$v_{\text{vent}} = \sqrt{\frac{2(p - p_{\text{atm}})}{\rho_{\text{atm}}}}. \quad (9)$$

Substitution of Eq. (9) in Eq. (7) yields,

$$\begin{aligned} \Delta \bar{p} &= -\frac{p}{V} \left[\left(1 + \frac{\Delta \bar{p}}{p} \right) \omega I_{A_{\text{door}}} - A'_{\text{vent}} \sqrt{\frac{2\bar{p}}{\rho_{\text{atm}}}} \right] \Delta t \\ &= -\frac{\omega I_{A_{\text{door}}}}{V} \Delta \bar{p} \Delta t - \frac{p}{V} \left[\omega I_{A_{\text{door}}} - A'_{\text{vent}} \sqrt{\frac{2\bar{p}}{\rho_{\text{atm}}}} \right] \Delta t, \\ &= -\frac{\bar{p} + p_{\text{atm}}}{V + \omega I_{A_{\text{door}}} \Delta t} \left[\omega I_{A_{\text{door}}} - A'_{\text{vent}} \sqrt{\frac{2\bar{p}}{\rho_{\text{atm}}}} \right] \Delta t \quad (10) \end{aligned}$$

where

$$\bar{p} = p - p_{\text{atm}}. \quad (11)$$

The torque on the door about the hinge axis is therefore, given by

$$T_{\text{air}} = \int_{A_{\text{door}}} r \bar{p} dA = I_{A_{\text{door}}} \bar{p}, \quad (12)$$

and the energy sink increment during Δt is

$$\Delta E_{\text{air}} = T_{\text{air}} \Delta \theta, \quad (13)$$

where the door open angle change $\Delta \theta$ is provided by

$$\Delta \theta = \omega \Delta t. \quad (14)$$

The updated pressure difference at the door is then

$$\bar{p} \leftarrow \bar{p} + \Delta \bar{p}, \quad (15)$$

and the updated energy sink due to the air bind is

$$E_{\text{air}} \leftarrow E_{\text{air}} + \Delta E_{\text{air}}. \quad (16)$$

SEAL COMPRESSION

When the closing door swings further around the hinge axis, at position $\theta = \theta_c$, the seal between the door and body starts compressing along the door seal line and the body seal line, as shown in Figure 3. Here, θ_c is coined the *seal contact angle* of a seal point. It represents the door open angle at which the seal point begins contacting the body (or door). It is noted that all seal points along a seal line do not have the same contact angles. If $\theta > \theta_c$, there exists a door gap that allows air venting. The gap vanishes if $\theta < \theta_c$, and no air flows in or out the cabin at any seal point. In this case, the seal segment is under compression.

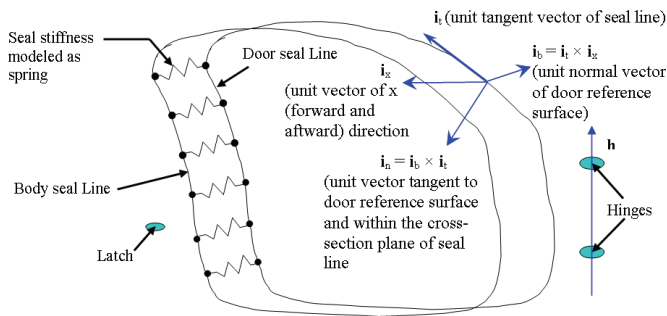


Figure 3. Schematic of the door-seal-body system

The seal is discretized into small segments along the seal line. The stiffness of each segment is represented by a spring connecting a point on the door seal line and a corresponding point on the body seal line. This represents a simple model of the seal flexibility. As explained later, its deviation from reality is compensated using calibration. Figure 4 shows a deformed seal “spring” under seal compression and no slip. A point **A** on the seal bulb, called seal contact point, will potentially contact the body at point **C**. If the body were not there,

the seal contact point should move to position **B**. In fact, the **A-C-B** path is part of a circular trajectory of the seal contact point as it revolves around the door hinge axis **h**.

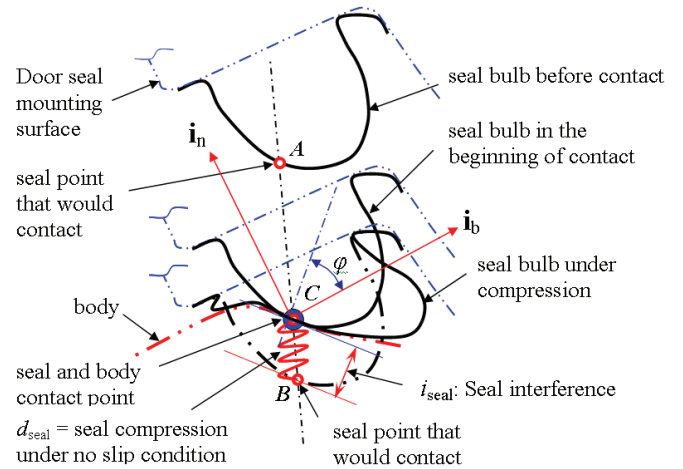


Figure 4. Seal compression with no slip

The **3-D** seal (spring) compression **CB** is given by

$$\mathbf{d}_{\text{seal}} = d_{\text{seal}} \mathbf{i}_d, \quad (17)$$

where

$$d_{\text{seal}} = \begin{cases} 0 & \text{for } \theta \geq \theta_c \\ r(\theta_c - \theta) & \text{for } \theta < \theta_c \end{cases} \quad (18)$$

is the compression deformation of the seal spring model in 3-D, and

$$\mathbf{i}_d = \frac{\mathbf{h} \times \mathbf{r}}{\|\mathbf{h} \times \mathbf{r}\|_2} \quad (19)$$

is the unit tangent vector of the seal point's circular trajectory. The 3-D *seal compression* $\mathbf{d}_{\text{seal}}$ is not in general, in the seal cross-sectional plane spanned by $\mathbf{i}_b$ and $\mathbf{i}_n$. The nominal seal interference i_{seal} represents not only the interference in the seal cross-sectional plane, but also in a direction normal to the tangent line at the contact position **C**. An interference angle φ is then defined with $\mathbf{i}_b$, the unit normal vector of the door reference surface. The vector form of the nominal seal interference i_{seal} can be then written as

$$\mathbf{i}_{\text{seal}} = i_{\text{seal}} (\mathbf{i}_b \cos \varphi + \mathbf{i}_n \sin \varphi). \quad (20)$$

The angle between $\mathbf{d}_{\text{seal}}$ and $\mathbf{i}_{\text{seal}}$ can be easily determined by

$$\varphi_d = \tan^{-1} \frac{\sqrt{1 - [\mathbf{i}_d \cdot (\mathbf{i}_b \cos \varphi + \mathbf{i}_n \sin \varphi)]^2}}{\mathbf{i}_d \cdot (\mathbf{i}_b \cos \varphi + \mathbf{i}_n \sin \varphi)}. \quad (21)$$

We have therefore, the following projection relationship between the 3-D compression deflection and the nominal seal interference

$$\mathbf{d}_{\text{seal}} \cdot (\mathbf{i}_b \cos \varphi + \mathbf{i}_n \sin \varphi) = d_{\text{seal}} \cos \phi_d = i_{\text{seal}}, \quad (22)$$

or equivalently,

$$d_{\text{seal}} = \frac{i_{\text{seal}}}{\cos \phi_d}. \quad (23)$$

It is clear from Eq. (23) that the 3-D compression deflection d_{seal} and the nominal seal interference i_{seal} are different. Once the nominal seal interference i_{seal} or $(i_{\text{seal}}, \varphi)$ is known from CAD design data for example, at the fully closed door position ($\theta = 0$), Eq. (23) and Eq. (18) yield the following seal contact angle

$$\theta_c = \frac{d_{\text{seal}0}}{r} = \frac{i_{\text{seal}0}}{r \cos \phi_d}, \quad (24)$$

which provides an important threshold for the calculation of the 3-D seal compression deflection according to Eq. (18) when $\theta \leq \theta_c$, and for the calculation of the air vent gap. The latter is given by

$$d_{\text{vent}} = \begin{cases} r(\theta - \theta_c) \cos \phi_d & \text{for } \theta > \theta_c \\ 0 & \text{for } \theta \leq \theta_c \end{cases}. \quad (25)$$

It represents the gap in the normal direction to the contact line, and is used to calculate the door gap area as shown in Eq. (6).

In practice, the nominal seal interference i_{seal} can be measured at the closed door state using a CAD tool such as Catia, where the seal cross-sectional geometry may not be representing the installed state (see Figure 5). In this case, the seal lip interference l_{seal} , will be of

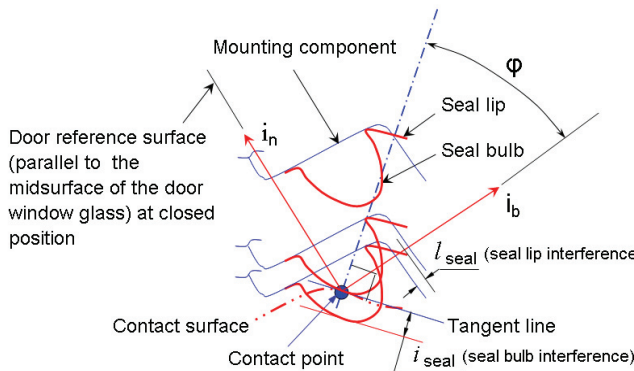


Figure 5. Schematic of a door mounted seal cross-section

interest, representing the amount of interference of the seal lip with the seal mounting surface. l_{seal} compensates for the seal bulb interference using $i_{\text{seal}} := i_{\text{seal}} + l_{\text{seal}}$. Note that the seal interference, i_{seal} , is the maximum distance from the seal bulb to the tangent line at the contact, and must be measured in the normal direction to the tangent line at the contact point.

The seal stiffness is usually given by the Compression Load Deflection (CLD) curve. Figure 6 shows a set of CLD curves used in the example of this paper.

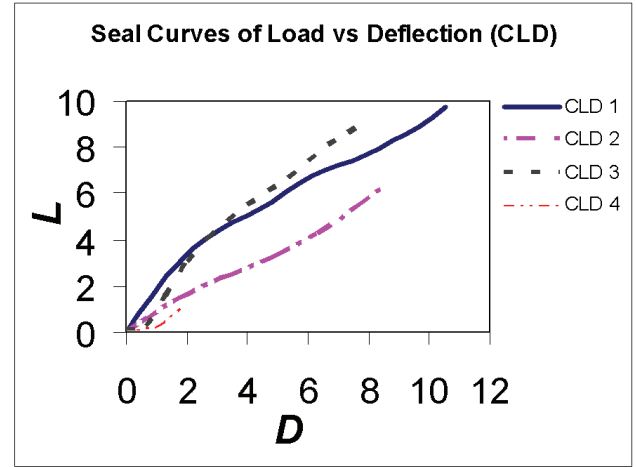


Figure 6. Seal force L (N/100mm) vs compression D (mm)

A normalized seal compression load deflection (CLD) curve for a seal segment of length l is used based on the compression distance d_{seal} (see Figure 5), and the compression force f_{seal} . The relation between the normalized compression force L and the normalized compression distance D is

$$L = L(D); D = d_{\text{seal}}; L = \frac{f_{\text{seal}}}{l}, \quad (26)$$

resulting in the following force for a seal segment

$$f_{\text{seal}} = l L(d_{\text{seal}}). \quad (27)$$

The resulting torque around the hinge axis due to the seal compression forces is given by the following summation over all seal segments

$$T_{\text{seal}} = \sum_i f_{\text{seal},i} r_i \quad \text{or} \quad T'_{\text{seal}} = c_{\text{seal}} T_{\text{seal}}, \quad (28)$$