Side Impact Crashworthiness Design: Evaluation of Padding Characteristics Through Mathematical Simulations

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(ESV) Conferences [1,2,3,4]. Additional reports have also been published by the agency regarding these efforts.

Many investigations of real world accidents, crash test results and simulation studies have established that in side crashes of passenger cars, thoracic injuries of occupants are, in large part, caused by occupants' impact against the interior side of the vehicle, primarily the door and other intruding side structures. In order to minimize the thoracic injury potential, the designer must limit the interior impact velocity and maintain the compartment integrity. Sudden collapse of the surrounding compartment structures must be avoided, at least until the incoming vehicle has been slowed down.

A number of vehicle design characteristics (e.g. door thickness at occupant contact areas, pillar design, pillar/floor/roof, attachments, the seat and dash board design etc.) have been identified as having marked influence on dummy measured injury indicating parameters [4]. Vehicle structural characteristics such as pillar strength, door strength and energy dissipation characteristics of various vehicle components also affect the crash severity experienced by the occupant of the struck vehicle. One of the countermeasures that greatly improves the energy dissipation that occurs in the impacted vehicle is the addition of energy absorbing padding material in the door area. NHTSA, among others, has conducted tests of vehicles with and without padding in doors and found that the addition of padding reduces thoracic injury potential (as indicated by the Thoracic Injury Index measured on the dummy [TTI(d)]) by about 30 percent in many production cars tested using the FMVSS 214 dynamic test procedure.

Force-deflection characteristics of energy absorbing padding have been modeled analytically by NHTSA in many previous simulation studies [1,2,3]. The characteristics of a number of different padding materials suitable for applications in car door manufacture have been analytically represented by the function:

\[ F = C_f \left[ S/(S_{\text{max}} - S) \right]^p \]
where \( p \geq 0.0 \)

\[ C_f = \text{a force coefficient related to padding stiffness (lbs)} \]

\[ S = \text{padding deflection (in)} \]

\[ S_{\text{max}} = \text{padding thickness (in)} \]

\[ p = \text{a shape coefficient indicating the shape of the force-deflection curve (unitless)} \]

The objective of this paper is to investigate the effect of a number of these parameters which describe various paddings on the thoracic injury measure TTI(d) by simulating side crashes in a production vehicle with different types of door paddings. Further, a method is also demonstrated to select the optimum padding and its related parameters so that the designer can choose a feasible padding to obtain the lowest possible injury measure in a side crash of the vehicle. This approach could also be used to explore pelvic responses, however, the discussion in this paper is limited to thoracic responses only.

The simulation studies described in this paper are based on test conditions, test devices, and test procedures described in the final rule FMVSS No. 214 (amended) recently announced by NHTSA [5]. However, the model simulates an unrestrained occupant whereas the final rule calls for all occupants to be belted. The production vehicle selected for the simulation study was a 1987 model Nissan Sentra. It was tested by NHTSA using a 3000 lb aluminum honeycomb barrier in the cradled mode simulating a car-to-car collision where the striking vehicle is moving at 30 mph and the struck vehicle at 15 mph, perpendicular to each other. An unrestrained Side Impact Dummy (SID) seated in the driver seating position represented the occupant.

A brief description of the model is provided. The occupant model which was modified from the earlier simulation studies reported in [3] is described in more detail. A detailed discussion of the simulation methodology, analyses and discussion of results and a description of the optimization procedure are presented in this paper.

SIDE IMPACT MODEL

The parametric data used for analysis was generated by the lumped mass side impact model described in [3] and [4] where the one-dimensional occupant model was replaced by the two-dimensional occupant model described in [6]. The speeds referred to in the following discussion, given in the form 30/15 mph, etc., represent intersection collisions of two cars. The first number refers to the striking vehicle speed while the second number relates to that of the struck vehicle. The two-dimensional model accounts for the angular motion and the lateral motion of the occupant. This representation permits the model to more accurately simulate the rotational response of the occupant's thorax about the longitudinal axis in a side collision. Figures 1 through 3 illustrate the mathematical model of the Moving Deformable Barrier (MDB), an unrestrained occupant, represented by the SID, seated in the front seat on the impacted side of the struck vehicle, and the struck car itself.

**Figure 1: Lumped Spring Mass Model for MDB**

**Moving Deformable Barrier (MDB)** - The response of the MDB is characterized by a one-dimensional lumped mass model. It consists of a single mass constituting the total mass of the vehicle and two non-linear energy absorbers which represent the softer face and the stiffer sections of the aluminum honeycomb front. The motion of the MDB is restricted so as to move in the lateral direction of the struck car.

**Figure 2: Lumped Sprint Mass Model for Door Padding and Driver In Struck Vehicle**

**Side Impact Dummy (SID)** - The SID response behavior was modeled in the same manner as the two-dimensional model developed by Hasegawa et al [6] where four masses and ten non-linear energy absorbers characterize the major body elements of the dummy participating in the crash. The motion of each mass is described at the center of gravity by three degrees of freedom, two translational and one rotational, where the translational motions are those in the lateral and vertical
direction, respectively, and the rotational motion is in the roll direction. Two of the masses represent the thorax, one representing an equivalent mass of the thorax about the longitudinal axis which comes in contact with the impacted car door surface (the outer thoracic mass) and the other representing the remaining equivalent mass of the thorax (the inner thoracic mass). These two masses are interconnected in parallel by three equally spaced non-linear energy absorbing springs and a viscous damper arranged between their centers of gravity. The pelvis and head masses are each connected to the inner thoracic mass (spine), through a simple energy absorbing shear element. The door interacts with the thoracic and pelvic masses through four energy absorbers, one in the pelvic region and three in the thoracic region. The abdominal insert of the SID and its interaction with the door were not included in this model. In the same manner, the interaction between the SID and the seat was neglected.

The effective mass and force versus deflection traces for all of the mass and energy absorbing elements in this model, except for the force versus deflection traces characterizing the two elements which couple the head and pelvis to the spine, are from [1]. These two elements were described by the linear force-deflection traces provided in [6]. The inner to outer thoracic force versus deflection traces from [1] were equally apportioned among the three thoracic energy absorbers in this model. The moments of inertia of the four body part masses representing the occupant and the spacing between all of the energy absorbers and the centers of gravity of the masses were from [6].

The following responses were simulated by the two-dimensional SID model:

1. pelvic response (measured at c.g.)
2. head response (measured at c.g.)
3. upper rib response (measured at the upper energy absorber attachment point on outer thorax mass element).
4. lower rib response (measured at the lower energy absorber attachment point on outer thorax mass element).
5. upper spine response (measured at the upper energy absorber attachment point on inner thorax mass element).
6. lower spine response (measured at the lower energy absorber attachment point on inner thorax mass element).

Struck Stationary Car - The struck car is characterized by nine masses and twenty one non-linear energy absorbers
and its motions are in the lateral direction only. The energy absorbers interconnect the various masses and represent different components of the struck vehicle. Four masses are used to model the response of the upper and lower, inner and outer sections of the door. The upper and lower sections of the A and B pillars are represented by an additional four masses. The remaining masses of the struck vehicle components are lumped with the rear seat occupant and are represented by a single mass designated as the passenger compartment. The pillar sections and the door mass are interconnected by non-linear energy absorbers. In the same manner, the passenger compartment is also connected to the pillars.

**Padding Model** - In modeling a padded door, two strips of padding were used, one attached to the lower-inner door surface and the other to the upper-inner door surface allowing contacts with the pelvic and thoracic regions respectively. These strips of padding were modeled as one-dimensional, non-linear energy absorbers (see Figure 2). The mass in this model represents an effective mass of the contacted region of the pad and was included for two reasons. First, it allowed the energy absorbed in each strip of padding to be calculated independently, without determining this from force-deflection curves composed of both padding and outer thorax or pelvis without the padding effective mass. Secondly, our parametric characterization modeling studies on the dynamic behavior of the dummy in side impacts determined that outer thoracic accelerations can be influenced by the effective mass of the pad depending on the selection of its mass and energy absorbing element properties.

For modeling, the effective mass of the pad was assumed to be the mass of the padding material in direct contact with the SID, determined from the density, the thickness of the pad and the contact area. The contact area of the SID was assumed to be 45.5 sq. inches.

A general expression relating the force of impact to padding deformation at each instant of time was used to describe the dynamic force versus deflection characteristics of the padding strips:

\[ F_d = C_f \left[ \frac{S}{S_{\text{max}}} \right]^p \]  

**EQ (1)**

where \( p \geq 0.0 \)
- \( F_d \) = the dynamic force
- \( C_f \) = a dynamic force coefficient
- \( S \) = the padding deflection
- \( S_{\text{max}} \) = the padding thickness
- \( p \) = a shape coefficient

This expression was validated with experimental force versus deflection traces acquired from special impacts carried out on a variety of padding materials with different thicknesses. The deflection rate was not included in this expression since our modeling studies of a SID impacted at speeds ranging from 15 mph to 35 mph by doors equipped with a number of polymeric closed cell foam paddings three inches thick, concluded that a padding force expression which includes strain rate can be approximated quite accurately by a force-deflection expression developed from dynamic impact data.

Padding hysteresis was not modeled in our study for side collisions ranging between 25/12.5 mph and 35/17.5 mph, and padding materials 3 inches in thickness. Instead, the pad’s bottomed-out stiffness was used for the unloading stiffness. This assumption was based on studies carried out by Low and Prasad [7] on a 3 inch thick APR pad which concluded that the SID responses were not very sensitive to hysteresis effects at a door impact speed of 20 mph, a speed which is within the range of door impact speeds generally observed for a variety of car designs in 30/15 mph side collisions.

**VALIDATION OF THE SIDE IMPACT MODEL**

**MY-87 Nissan Sentra, 90° Impact** - We elected to use a lightweight 2-door, 2510 lb, 1987 Nissan Sentra which was crash tested by NHTSA in a 90° configuration as the candidate vehicle in this paper. The Nissan Sentra in this crash test remained stationary and the MDB was towed into its side. This was performed at a closing velocity of approximately 33.5 mph at a crab angle of 26°. The impact point on the Nissan Sentra was 37 inches forward of the center line of the wheel base. (This crash test simulated a 90° impact collision where the striking vehicle moved at 30 mph and the struck car moved at 15 mph). The MDB weighed 3000 lbs and a SID was used in the driver seating position.

![Figure 4a: Experimental and Simulated Driver Responses](1987 Nissan Sentra at 30/15 mph, 90° Impact)

Figure 4a compares the simulated and measured peak upper and lower rib, upper and lower spine, and lateral and vertical head accelerations. The upper rib, lower rib, and lower spine acceleration-time histories along with peak G’s
obtained from simulations and crash tests are compared in Figure 4b. The time histories show acceptable match between simulations and test results. TTI(d) here is defined as an average of the maximum peak rib and peak lower spine accelerations. Note that the simulated peak accelerations are based on unfiltered model traces and the measured peak accelerations are based on Finite Impulse Response (FIR) filtered traces. Comparison of the simulated and measured peak pelvis accelerations is not shown since the experimental pelvic data had anomalies in it.

![Graphs of Upper and Lower Rib/Spine Accelerations](image)

**Figure 4b: Experimental and Simulated Driver Acceleration Time Histories**

(1987 Nissan Sentra at 30/15 mph, 90° Impact)

In general, the model peak accelerations agreed well with the test data. One exception was the upper peak spine acceleration which was approximately 20 G's lower than the measured value. The reason for this difference is not clear. However, since we were interested in applying this model to determine TTI(d) as a function of door padding and car side structural interactions, matching the response of non-TTI related channels was not of primary importance.

The model results also showed that the upper and lower rib and lower spine acceleration-time history profiles matched the experimental data quite satisfactorily, even though the measured and simulated peak upper spine accelerations were not matched. In these results, the peak of the upper spine acceleration was shifted slightly later in time than the measured peak upper spine acceleration. The simulated head accelerations, although matching the measured peak accelerations, were low frequency in character and did not contain the aperiodic higher frequency content observed in the measured accelerations.

In this modeling exercise, a 3.5 inch inner door surface to pelvis spacing was used and the spacing dimension at center of gravity (c.g.) of the outer thorax-to-door was 4.35 inches. Because of the non-uniform crush of the door in the vertical plane at the time the door contacted the SID thorax, the door spacing at the thoracic area, which yielded reasonable model results, was 3.92 inches for the spacing between the door and the lower section of the outer thorax and 4.78 inches for the spacing between the door and the upper section of the outer thorax.

The force-deflection characteristics for the various energy absorbers connecting the masses within the Nissan Sentra door, A and B-pillars, and main car structure were estimated from crash test acceleration pulses, static and special dynamic impact tests performed on door interiors, and special dynamic impacts performed on pillars. Force-deflection traces were assumed for those energy absorbers where no force deflection data was available and these traces were fine-tuned through repeated simulations. Door thickness at the pelvic and thoracic contact levels was obtained from measurement acquired from the vehicle. The MDB mass and force-deflection characteristics were derived from MDB crash test data.

### Parametric Data

Door padding and struck car side stiffness were parametrically investigated to determine the optimum padding characteristics for reducing thoracic injury potential. The struck car considered was the 1987 Nissan Sentra in 90° impacts involving striking MDB/stuck Nissan Sentra at speeds of 25/12.5 mph, 30/15 mph and 35/17.5 mph. Three inch thick strips of padding material were positioned on the interior surface of the door at the thoracic and pelvis contact regions. The SID pelvis was positioned four inches and the outer thorax was positioned 4.85 inches from the padded door surface. The two parameters characterizing the door padding material were the coefficients $C_t$ and $p$ in EQ (1). Struck car side stiffness was characterized by the outer door-to-pillar and pillar-to-passenger compartment stiffnesses (EA-3,4,5,6 and EA-18,19,20,21, respectively) (see Figure 3). The stiffness variations are defined using multiplication factors for the baseline car force levels. A car stiffness of 1.0 pertains to the baseline car.

The SID driver response data, described in the Side Impact Model section, was constructed through model simulations using Program Fiat-B [8], a three dimensional computer simulation program designed for lumped mass vehicle crash studies. The analytical expression, EQ (1), provided the force-deflection data for door padding. Over six hundred simulations were performed, including the following variations in car side stiffness and padding $C_t$ and $p$ for the three striking car speeds of the MDB:

1. five car stiffnesses (0.43 0.714, 1.0, 1.43 and 1.714 nominal stiffness);
2. six $C_t$'s shown in Table 1 for materials dynamically tested at DOT/NHTSA/VRTC (National Highway Traffic Safety Administration, Vehicle Research & Test Center);
3. six padding shapes, p's (0.0,0.1, 0.3, 0.5, 0.7 and 0.9) which fall in the range of p's for the padding materials shown in Table 1.
Table 1: Padding Characteristics from Dynamic Tests

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (pcf)</th>
<th>$C_f$ (lbs)</th>
<th>$p$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mattress Foam</td>
<td>2.0</td>
<td>541.4</td>
<td>0.800</td>
</tr>
<tr>
<td>Ethafoam 220 (Dow)</td>
<td>2.2</td>
<td>987.6</td>
<td>0.475</td>
</tr>
<tr>
<td>Arcel 310 (Arco)</td>
<td>1.5</td>
<td>1729.5</td>
<td>0.373</td>
</tr>
<tr>
<td>Arcel 512 (Arco)</td>
<td>2.5</td>
<td>2409.5</td>
<td>0.218</td>
</tr>
<tr>
<td>Dytherm 3.0 (Arco)</td>
<td>3.0</td>
<td>4034.5</td>
<td>0.172</td>
</tr>
<tr>
<td>Dytherm 4.0 (Arco)</td>
<td>4.0</td>
<td>6566.3</td>
<td>0.217</td>
</tr>
</tbody>
</table>

Figure 5 shows examples of force-deflection traces depicting the six shapes considered for the optimizations for $C_f = 1729.5$ lbs. When $p = 0.0$, a constant force pad is obtained. A linear stiffness of 100 klbs/in was used for the bottoming out stiffness since Eq (1) is independent of deflection when $p = 0$. The same stiffness of 100 klbs/in was used for the unloading portion of the force-deflection curve.

Figure 5: Dynamic Force Deflections for Analytical Pads
(Traces of 3" Thick Pads with $C_f = 1729.5$ lbs)

In addition, parametric data was constructed for two other padding conditions, $C_f = 0$, and $\infty$. A nonlinear multiple regression was applied to express the simulated TTI(d) as a function of the two analytical padding variables $C_f$ and $p$ for the 30/15 mph side impact condition.

Some selective trends of the injury measures with regard to the parameters described in the optimization study are discussed below.

PARAMETRIC RESULTS FROM SIMULATIONS

Effect of Padding Variables $C_f$ and $p$ - An example of the response of TTI(d) to different padding force-deflection prescriptions, as described by $C_f$ and $p$ in the analytical padding expression Eq (1), is shown in Figure 6 where curves of constant $p$ are shown for $p = 0.0$ (a constant force pad) to 0.9. The effect of both material coefficients $C_f$ and $p$ on TTI(d) are clearly noted, particularly on the minimum TTI(d) where the optimal amount of energy is absorbed by the pad during occupant contact (see Figure 7). These curves show that padding materials characterized by shape coefficients lying between 0.0 and 0.3 are most effective in reducing TTI(d). It was determined that at the optimal padding conditions, $C_f = 1200$ lbs and $p = 0.1$, the TTI(d) was effectively reduced by approximately 40%. Note that the TTI(d) curves for the lower $p$'s dip sharply at the lower values of $C_f$'s. As $p$ assumes higher values, a lessor change in TTI(d) occurs. The minimum TTI(d) occurs in these cases at the higher values of $C_f$'s. All curves eventually approach asymptotically the TTI(d) value for an un padded door condition as indicated by $C_f = \infty$ in Figure 6. $C_f = 0$ denotes a pad with a voided material space of 3 inches. In this case, the occupant is seated 7 inches from an un padded door prior to door contact.

In Figure 6, the trace of the minimum TTI(d)'s as a function of $C_f$ is defined as the optimal TTI(d) curve. For a padding effectiveness of approximately 20-30%, this optimal curve shows $C_f$ ranging from 800 to 2000 lbs and $p$ ranging between 0.0 and 0.3.

Note from Figure 7 that the maximum energy was absorbed by the constant force pad ($p = 0.0$) while the pad for minimum TTI(d), denoted by $C_f = 1200$ lbs and $p = 0.1$, absorbed slightly less energy. Simulations were not conducted at intermediate values between $p = 0.0$ and $p = 0.1$ and may yield slightly lower TTI(d) values. Also, it is interesting to note that 2 inch crush of the pad which has a slightly higher $C_f = 1750$ lbs absorbed roughly the same amount of energy as the optimal pad of $C_f = 1200$ lbs in this example, which gave a maximum crush of 2.75 inches. The TTI(d) for these two pads are approximately the same.
The effectiveness of the manufacturers padding materials presented in Table 1 on minimizing the TTI(d) in the 30/15 mph collision are compared to the minimum TTI(d) trace from Figure 6 as a function of C_f in Figure 8. In addition to these materials, another material manufactured by ARCO, Arsan 601, was found to exhibit characteristics similar to the pad simulated with C_f = 1200 lbs and p = 0.1.

**Figure 7: Padding Absorbed Energy vs Maximum Crush**
(1987 Nissan Sentra, 30/15 mph, 90° Impact)

**Figure 8: TTI(d) Performance of Manufacturer Pads**
(1987 Nissan Sentra, 30/15 mph, 90° Impact)

**Effect of Impact Speed** - Figure 9 shows the optimal TTI(d) curves plotted against C_f at lateral impact speeds of 25, 30 and 35 mph. The minimum TTI(d) increases with impact speed. Most importantly, the TTI(d) appeared to be effectively reduced in a narrower range of C_f’s at the higher 35 mph impact speed than at the lower 25 mph impact speed. It should also be observed that the optimal C_f, that is the C_f corresponding to minimum TTI(d), increases with impact speed; for example, C_f = 1000 lbs for the 25 mph impact, 1200 lbs for the 30 mph impact and 1600 lbs for the 35 mph impact. The shape coefficient, p, for minimum TTI(d) also increases in value with impact speed; 0.1 for 25/12.5 mph impact, 0.1 for 30/15 mph impact and 0.3 for 35/17.5 mph impact. The shape of the padding force-deflection curve thus begins to depart from the p = 0.0 shape at the lower impact speed to shapes described by p’s between 0.1 and 0.3 to effectively reduce the TTI(d) at the higher impact speeds. Interestingly, these curves show that if the padding characteristics, C_f and p, were selected on the basis of the lower 25/12.5 mph impact, this would result in a pad which would be approximately 12% less effective in minimizing the occupant TTI(d) at the higher 35/17.5 mph impact. Padding characteristics selected on the basis of the higher 35/17.5 mph impact appear to minimally affect (less than 4%) the pad’s effectiveness at the lower impact speed of 25/12.5 mph. Finally, the optimal performance of padding is effective in a slightly smaller range of C_f’s as the speed is increased.
Figure 9: Effect of Impact Speed and Padding $C_f$ on Optimal TTI(d)
(1987 Nissan Sentra, 90° Impact)

Figure 10: Effect of Car Normalized Side Stiffness and Padding $C_f$ on Optimal TTI(d)
(1987 Nissan Sentra, 30/15 mph, 90° Impact)
Effect of Car Stiffness - Figure 10 shows the optimal TTI(d) curve for three car side stiffnesses: 0.714 nominal, 1.0 nominal (baseline) and 1.714 nominal stiffness for the 30/15 mph impact. The optimal TTI(d) curves are similar, with the exception that the TTI(d) is more sensitive to \( C_f \) in the case of the 0.714 nominal stiffness car than in the case of the 1.714 nominal stiffness car. This is because, generally, padding is more effective in reducing the TTI(d) of cars of lower side stiffness than of cars of higher side stiffness. The optimal \( C_f \) for the minimum TTI(d) tends to shift slightly to lower values with increases in the side structural stiffness. For example, at the 30/15 mph impact the minimum TTI(d) occurred for the 0.714 nominal stiffness car at \( C_f = 1400 \) lbs, while the minimum TTI(d) occurred for the 1.714 nominal stiffness car at \( C_f = 1100 \) lbs. P’s showed a trend towards higher values with decreases in the side structural stiffness due to the inability of the softer structure to absorb the impact energy from the MDB effectively prior to occupant contact with the door.

GENERAL OBSERVATIONS

The shape of the force-deflection curve as defined by \( p \) had a strong influence on the inner and outer peak thoracic G’s. For example, in the case of the 35/17.5 mph impact, see Figure 11, it was observed that the outer thoracic peak G appears to be effectively reduced when \( p \) equaled about 0.25. The inner thoracic peak G, on the other hand, appears to be effectively reduced when \( p \) equaled about 0.15. Besides this, both thoracic peak G’s appear to be minimized at approximately the same value of \( C_f \) the minimum outer thoracic peak G occurring at a slightly higher value of \( C_f \) than the minimum inner thoracic peak G. Since TTI(d) is composed of the two peak G-levels, this example suggests that a pad should be chosen which combines the effect of both \( C_f \) and shape factor. This indicates a pad of more rounded than square shape force-deflection characteristics at low value of crush to effectively absorb the impact energy of the outer thoracic mass element when it comes in initial contact with the pad. Also, at the same time as the pad further deforms, it must provide continuous efficient absorption to reduce the later inner thoracic accelerations. Note that the energy absorbed by padding in this case is usually less than the optimal energy absorbed by a constant force non-bottoming out pad under minimum force of impact. Slow bottoming out features are required to preclude higher second peak G levels in the outer thoracic mass. The same trend in shape factors as just discussed were also observed at the other two collision speeds of 25/12.5 mph and 30/15 mph. See Figures 12 and 13.

It was also observed that:

1. TTI(d) appeared to be more sensitive to the occupant’s rotation during door encounters with the softer padding material than the stiffer one. For example, in the case of the 30/15 mph impact there was a 9 to 15 G difference between the upper and lower inner thoracic peak G’s with the softer \( C_f = 541.5 \) lb pad whereas there was a 3 to 6 G difference between the upper and lower inner thoracic peak G with the stiffer \( C_f = 4034.5 \) lb pad.

2. Padding inertial effects did not appear to be a factor on the occupant responses since the SID’s outer thoracic effective mass was much larger than the padding material’s effective mass.

3. All padded door TTI(d)’s were less than the unpadded door TTI(d)’s.

OPTIMIZED PADDING CHARACTERISTICS

Table 2 illustrates the size and range of energies absorbed by the optimal pads for the different side stiffness and impact speeds parametrically studied. Values of \( C_f \) and \( p \) are also displayed along with the optimized and unpadded door TTI(d)’s.

<table>
<thead>
<tr>
<th>Lateral Impact Speed (mph)</th>
<th>Side Stiffness (normalized)</th>
<th>Absorbed Energy (in-lbs)</th>
<th>Min. TTI(d) (G)</th>
<th>Optimal ( C_f ) (lbs)</th>
<th>Optimal No Pad TTI(d) (G)</th>
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<tr>
<td>25</td>
<td>0.43</td>
<td>2920</td>
<td>53</td>
<td>970</td>
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<td>1400</td>
<td>0.3 120</td>
</tr>
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</table>

Simulated Data            * - Base

It is seen from Table 2 that a 30/15 mph crash simulation of a relatively soft car (normalized side stiffness = 0.43) yields a TTI(d) of 81 G’s with a 3 inch pad while a baseline car with a relatively stiffer structure (normalized side stiffness = 1.43) gives a TTI(d) of 84 G’s. It must be noted that even though less energy is transferred to the occupant simulated in the padded case, the ribs are
undergoing approximately the same velocity change under the two impact conditions. The velocity change in the “soft car” case is about 1 mph higher than the velocity change in the “stiff car” case. This indicates that the side impact severity in the padded and baseline cases appear to be about the same.

Interestingly, the absorbed energies and optimized TTI(d)’s appear to vary linearly with $C_t$ according to the different side stiffness and impact speeds listed in Table 1 for the 1987 Nissan Sentra with the 3 inch door padding. We found that these minimum TTI(d)’s can be approximated as a function of $C_t$ by the following expression which was derived based on a non-bottoming out chest impacting a constant force pad backed by a rigid wall.

$$TTI(d)_{optimal} = \frac{C_t}{M_{eff}} \quad \text{EQ (2)}$$

where $M_{eff} = 2M_o(M_{ot}+M_{it})/M_{ot}+3M_{ot}$ is an effective mass and $M_{ot}$ and $M_{it}$ are the outer and inner thoracic masses, respectively. This relationship is only applicable to the SID thoracic mass distribution.

SUMMARY AND CONCLUSIONS

The effectiveness of door padding and side structural stiffness to reduce the potential for occupant thoracic injury in side impacts was evaluated through simulations on one make/model of a car in a 90° impact. The lumped mass computer model reported in Experimental Safety Vehicle (ESV) Conference was used after being modified to include the thoracic responses in a two-dimensional dummy. Parametric investigations were carried out on a three inch thick padding material in a 1987 Nissan Sentra car for different levels of side stiffness. The SID dummy, seated four inches from the interior surface of the door, was used. The padding force versus deflection curves were obtained from an analytical expression used in earlier modeling [2,3]. This expression was verified with experimental data from special dynamic tests carried out on a variety of materials of different strengths, shapes and thicknesses. The two material constants, $C_t$ and $p$ in this expression, formed the basis for the parametric changes, where $C_t$ represents the 50% crush and $p$ represents a factor which modifies the shape of the force-deflection curve. $p = 0$ designates a pad which gives a constant force which is only a theoretical pad. The following was observed from simulations carried out at the MDB lateral impact speeds of 25, 30 and 35 mph.

In general, the car-to-car parametric simulations, based on the non-linear expression showed that occupant responses are strongly dependent on the two coefficients $C_t$ and $p$. To achieve the lowest TTI(d), it was necessary to select $C_t$ and $p$ such that the pad absorbs, through crush of the material before bottoming out, most of the occupant's impact energy during its contact with the door. It was also determined that $C_t$ and $p$ for minimum TTI(d) vary depending on the speed of the collision and the structural side stiffness of the impacted vehicle. As expected, the $C_t$ for minimum TTI(d) increases with increase in side impact closing speed and decreases with increase in the side structural stiffness.

Our study indicated that a pad which absorbs the initial impact of the outer thoracic element efficiently and also provides continued absorption to effectively reduce the response of the inner thoracic element minimizes injury in the thoracic area. Under this condition, TTI(d) reductions were found to be of the order of 30% at 25/12.5 mph, 40% at 30/15 mph and 54% at 35/17.5 mph. The lower impact speed favored a force-deflection shape of $p$ approximately equal to 0.1 while the higher impact speed favored a force-deflection shape of $p$ close to 0.3 to effectively reduce the TTI(d), (see Figures 11 through 13).

Thoracic accelerations can most strikingly be influenced by the shape of the padding force-deflection curve. The shapes characterized by $p$'s in the range 0.1 to 0.3 were found to be most effective in reducing TTI(d) over a narrow range of $C_t$'s, while the shapes characterized by $p$'s in the range 0.5 to 0.9 were found to be much less effective in reducing the TTI(d). Their effectiveness, however, extended over a broader range of $C_t$'s. (Since, no paddings were identified from the experimental tests which were characterized by shape coefficients greater than 1.0, these shapes were assumed to be non-representative and therefore were not considered in the study.)

A padding material which minimized the TTI(d) at the 35/17.5 mph impact appears to be also effective in reducing the TTI(d) at the 25/12.5 mph impact, since the pad does not differ significantly in performance from a material which minimized the TTI(d) at the 25/12.5 mph. However, it is not clear whether a pad which minimizes the TTI(d) at the 35/17.5 mph impact condition will offer TTI(d) reductions at the much lower impact of 10/5 mph. Thus, further studies will be necessary.

It was determined that a pad selected to effectively reduce the TTI(d) of a low stiffness car can also be used to effectively reduce the TTI(d) of a higher stiffness car.

Since TTI(d) was shown in the simulations to be sensitive to the choice of the two padding coefficients $C_t$ and $p$, it is possible to develop a mathematical expression which functionally relates TTI(d) to the padding material coefficients and a car's Thoracic Trauma Indexes at $C_t=0$ and as $C_t \to \infty$. In the future, the authors intend to develop such a generalized expression which, knowing the expected baseline performance of a passenger car, would be useful in selecting the padding parameters to achieve the desired design goals of TTI(d).
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