STATUS OF ENHANCED FRONT-TO-FRONT VEHICLE COMPATIBILITY TECHNICAL WORKING GROUP RESEARCH AND COMMITMENTS

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ABSTRACT

This paper describes a part of ongoing progress and research conducted by the Front-to-Front Compatibility Technical Working Group (TWG) to enhance vehicle compatibility in vehicle-to-vehicle frontal crashes. As a short-term goal, the TWG developed and implemented Phase I performance criteria, based on static measurements of the Primary Energy Absorbing Structure (PEAS) height, to improve geometrical compatibility. This will enhance structural interaction, through better matching of frontal component geometries, between cars and light trucks, in frontal crashes. Options include better matching of bumper heights, longitudinal frame rail heights, and more evenly distributing impact forces across the fronts of vehicles. All participating manufacturers' new light trucks up to 10,000 pounds Gross Vehicle Weight Rating (GVWR), with limited exceptions, must meet Phase I requirements by September 1, 2009.

The focus of Phase II research for the TWG is the investigation and evaluation of Front-end performance. This will include research to investigate test procedures and performance metrics to assess potential dynamic front-end geometric, stiffness, and any other relevant performance characteristics that would enhance partner protection without any significant degradation in self-protection.

Test and simulation results obtained from frontal impacts with various Load Cell Walls (LCW) and from vehicle-to-vehicle impacts in various frontal impact configurations to support phase II research were analyzed and presented to help assess and improve vehicle compatibility. Average Height of Force (AHOF) obtained from frontal impact with LCW was investigated as a compatibility metric. Initial finding was the AHOF alone is insufficient metric and did not correlate with Aggressivity Metric (AM) defined by NHTSA. Alternative metrics and test procedures are under investigation by the TWG. Phase III research will focus on front stiffness matching between cars and trucks and also on passenger car compartment strength and integrity. The investigation will lead to the development of a test to determine appropriate front-end stiffness characteristics and criteria that would strike an appropriate balance between small vehicle passenger compartment strength and large vehicle energy absorption characteristics.

1. INTRODUCTION

The issue of crash compatibility of passenger vehicles has been around since at least the early 1970s when the widespread introduction of lightweight subcompact cars into a fleet of predominantly large and heavy cars caused some concerns. In recent years the trend of growing sales of sport utility vehicles and pickup trucks have led to renewed public attention to this issue. The National Highway Traffic Safety Administration (NHTSA) [1,2,3], The Insurance Institute of Highway Safety (IIHS), and The International Harmonization Research Activity (IHRA) have identified this trend and have increased the extent of their research in vehicle-to-vehicle compatibility.

The basic concern is the extent to which some vehicle design characteristics adversely influence the outcomes of two-vehicle crashes. Thus, in head-on crashes between two cars the risks for the occupants of the lighter cars increase as the weights of the heavier cars increase. Today, the crash compatibility focus has shifted from concerns about vehicle weight differences to the effects of differences in vehicle heights and front-end stiffnesses in crashes between cars and light trucks.

There are two approaches to improving crash compatibility among passenger vehicles. First and more important is to improve the protection a vehicle provides for its own occupants, which is sometimes referred to as "self protection." This approach is more important because it results in improved protection for vehicle occupants in all crashes, single-vehicle as well as crashes involving other passenger vehicles. Significant improvements in self protection have occurred over the past 20 years or so with the introduction of frontal airbags, better structural designs, increased belt use, etc., and as a result crash compatibility problems are smaller than they otherwise would have been. Self-protection enhancements can reduce occupant risks in all crash modes including frontal, side, and rear. The second approach to improving crash compatibility is to focus on vehicle design characteristics that can reduce the risks for occupants of other passenger vehicles, which is sometimes referred to as "partner protection." Partner protection improvements usually focus on changes to vehicle front-end designs for enhancing the structural interactions between the striking and struck vehicles.

2. BACKGROUND ON THE INDUSTRY AGREEMENT

Over the years individual auto manufacturers have made changes in their vehicles to enhance compatibility. However, in late 2002 the Alliance of Automobile Manufacturers decided to pursue a concerted industry-wide effort to develop performance criteria based on current "best practices," to further enhance vehicle compatibility. To start this process on February 11-12, 2003, the Alliance and the Insurance Institute for Highway Safety (IIHS) cosponsored an international meeting on enhancing vehicle-to-vehicle crash compatibility. Participants were not limited to these two groups; other international experts were included. NHTSA's Administrator Jeffrey W. Runge, M.D., opened the meeting by issuing a challenge to the industry for more progress on compatibility. Other speakers included representatives from Transport Canada, the United Kingdom's Transport Research Laboratory (TRL), the Alliance, Honda, and IIHS.

The technical presentations at the meeting laid the foundation for the industry to work on performance criteria to improve crash compatibility for the North American market. The data presented at the meeting highlighted potential opportunities to further enhance compatibility in both front-to-front crashes as well as front-to-side crashes. At the meeting, the participants agreed to set up two working groups of experts to develop initiatives and actions. One working group was established to address ways to improve compatibility in front-to-side crashes, the other to address front-to-front crashes. Each group included both industry and outside experts. Each group has developed initiatives and performance criteria that participating auto manufacturers are committed to adopt.

One of the key conclusions of the February 11-12, 2003, crash compatibility meeting was that a high priority should be assigned to addressing the issue of reducing injury risks in side impact crashes, especially when the striking vehicles are light trucks. In the short term, the meeting participants concluded that the most effective approach for this problem is to enhance self-protection for passenger vehicle occupants in side struck vehicles. Thus, the industry's specific commitment to enhanced front-to-side crash compatibility by improving self protection in side impacts covers light trucks (vans, pickups, and sport utilities) up to 8,500 pounds GVWR, as well as passenger cars.

In regard to front-to-front compatibility, the conclusions from the February 11-12, 2003, compatibility meeting was that improvements in frontal crash compatibility between cars and light trucks can best be achieved in the near term through improved partner protection. In particular, improved geometric matching between the front structural components of cars and light trucks is the most effective short-term approach, while better matching of frontal stiffness characteristics between cars and light trucks is a longer-term goal. It is important to not compromise the self-protection of occupants of light trucks as the front ends of these vehicles change to further improve partner protection.

Participating manufacturers started their research, investigation and proposed various phases for the development of compatibility performance criteria within two separate working groups for front-to-front and front-to-side compatibility. The working groups will transfer these performance criteria to appropriate internationally recognized voluntary standards to ensure the sustainability of these criteria. From this point in time, the focus of this paper will be on the results and criteria associated with enhancing front-to-front compatibility.

3. FRONT-TO-FRONT COMPATIBILITY WORKING GROUP COMMITMENTS

The TWG held its first meeting on March 10, 2003 and agreed on the following:

- A short-term initial step in addressing further improvements in front-to-front crash compatibility between two colliding vehicles is through better alignment and geometric matching of the vehicle crash structures.
- A barrier face load cell configuration with a 125mm x 125mm load cell array is appropriate to make the determination of the height and distribution of force of an impacting vehicle into the barrier face.
- The TWG agreed to review the use of a deformable face on the barrier for testing with NHTSA in order to ascertain the agency's willingness to include the deformable member as part of future (revised) FMVSS 208 barrier test procedures.

- The TWG agreed that achieving better alignment and engagement between the front-end structures of the impacting vehicles is the necessary first step towards improving compatibility. It was also agreed that manufacturers should begin designing light trucks so their PEAS (Rail or frame) overlap a proportion of the zone established by NHTSA in its bumper standard (49 CFR 581) for passenger cars. This zone of impact resistance for passenger car bumpers is the area between 16 and 20 inches off the ground. By ensuring that light trucks have a significant portion of their front energy-absorbing structures in this zone, these structures are more likely to engage (instead of over- or under-riding) the PEAS of passenger cars in a head-on crash.

4. PHASE I: ENHANCING GEOMETRIC ALIGNMENT OF FRONT ENERGY-ABSORBING STRUCTURES

The TWG developed the following Phase I requirements which were announced on December 3, 2003 as a first step towards improving geometrical compatibility: Participating manufacturers will begin designing light trucks in accordance with one of the following two geometric alignment alternatives, with the light truck at unloaded vehicle weight (as defined in 49 CFR 571.3):

OPTION 1: The light truck's primary frontal energy-absorbing structure shall overlap at least 50 percent of the Part 581 zone AND at least 50 percent of the light truck's primary frontal energy-absorbing structure shall overlap the Part 581 zone (if the primary frontal energy-absorbing structure of the light truck is greater than 8 inches tall, engagement with the entire Part 581 zone is required), OR, **OPTION 2:** If a light truck does not meet the criteria of Option 1, there must be a secondary energy-absorbing structure, connected to the primary structure, whose lower edge shall be no higher than the bottom of the Part 581 bumper zone. This secondary structure shall withstand a load of at least 100 KNewtons exerted by a loading device, as described in the attached Appendix A, before this loading device travels 400 mm as measured from a vertical plane at the forward-most point of the significant structure of the vehicle.

If a light truck has crash compatibility devices that deploy in high-severity frontal crashes with another vehicle, all measurements shall be made with these devises in their deployed state. Not later than September 1, 2009, 100 percent of each participating manufacturer's new light truck production intended for sale in the United States and Canada will be designed in accordance with either geometric alignment Option 1 or Option 2.

Applicability All light truck vehicles with a GVWR up to 10,000 pounds, except, low production volume vehicles, vehicles over 8,500 pounds GVWR with functional criteria which preclude them from meeting the performance criteria, (e.g., postal vehicles, military vehicles, service vehicles used by public and private utilities, vehicles specifically designed primarily for off-road use, and incomplete vehicles), and other vehicles that a manufacturer determines cannot meet the performance criteria without severely compromising their practicality or functionality.

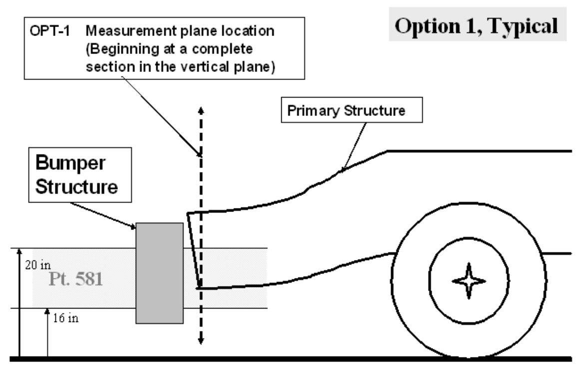


Figure 1. Typical front rail geometry and definition of Part 581 zone for voluntary standard.

Product Information Beginning November 3, 2003, and on each September 1st thereafter, through September 1, 2009 (i.e., November 3, 2003; September 1, 2004; September 1, 2005; September 5, 2006; September 3, 2007; September 1, 2008; and September 1, 2009), participating manufacturers will publicly disclose at least annually, the vehicle nameplates [models] for the upcoming model year that have been engineered according to the front-to-front and front-to-side performance criteria, and provide a 'good faith' estimate of the percentages of the manufacturer's total production for the upcoming model year that are engineered in accordance with the front-to-front performance criteria, respectively.

Confirmatory Data: Beginning November 3, 2003, and on each September 1st thereafter, through September 1, 2009 (i.e., November 3, 2003; September 1, 2004; September 1, 2005; September 5, 2006; September 3, 2007; September 1, 2008; and September 1, 2009), participating manufacturers shall voluntarily provide to NHTSA confirmatory data, engineering judgment, or other analyses demonstrating that vehicles identified above have been designed in accordance with the front-to-front and front-to-side performance criteria, respectively.

5. PHASE II RESEARCH

The TWG developed a matrix of physical tests and simulations in relation to rigid wall barrier impact, full width deformable barrier impact, and vehicle-to-vehicle impact to generate data to support Phase II research (see matrix shown in Appendix B). The purpose of this research phase was three fold. Firstly to identify a dynamic test procedure that evaluates geometrical changes made in vehicles. Secondly, existing metric proposals such as AHOF should be evaluated. Lastly, to evaluate test methods to measure front-end stiffness and develop potential actions to further enhance compatibility between vehicles.

The initial focus of the TWG was on the AHOF to be used as a metric for compatibility to enhance structural interaction between vehicles during frontal impact. The test methods evaluated were full-frontal impacts against a barrier fitted with load cells, load cells covered with a honeycomb (TRL Barrier Face) or without a honeycomb (similar to MIRA Barrier). Other TRL-type LCW with deformable elements such as the one shown in Figure 2 was also used. However, this barrier is 50mm from the ground and has 125mm x 125mm load cell in the interaction zone only, compared to that of TRL barrier which has 125mm x 125 mm load cell array, sixteen cells wide and ten cells high.

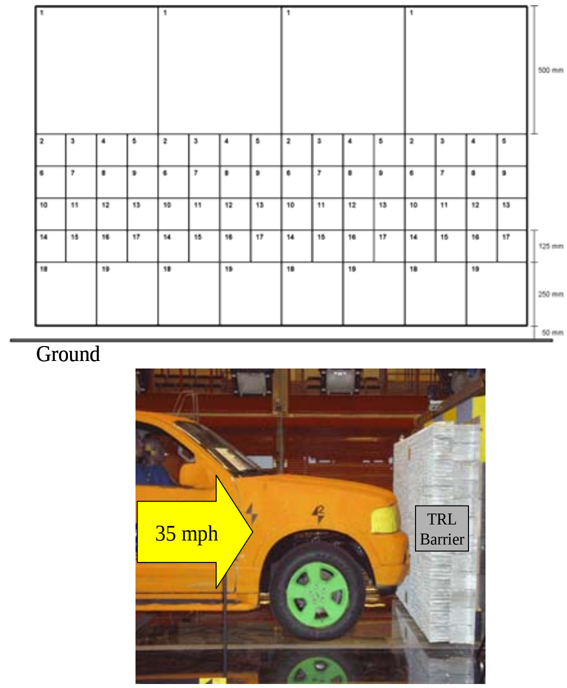


Figure 2. TRL-Type 125x125mm load cell deformable barrier.

Most of the tests and/or simulation planned on Appendix B were executed and completed by the Alliance participants. Although several geometric parameters or metrics such as Height of Force (HOF), AHOF, Homogeneity factors (CV), load distribution, row's force limit, cell force limit, row force percentage, deformation based, and other metrics in the interaction zone can be obtained and investigated, the initial focus of the research was on the AHOF calculation obtained from 56 kph frontal impact with TRL/TRL-type deformable barrier or rigid barrier tests. In addition, vehicle-to-vehicle frontal impact tests and real-world accident data analyses were conducted to validate the relation between the AHOF metric, as a compatibility metric, and the outcome of the occupant injury from crash tests and traffic accidents. Figure 3 shows an example of the load-time history of each cell obtained from a 56 kph impact of a mid-sized SUV. On the same figure it also shows the part 581 zone and locations of the PEAS of typical SUV and passenger cars. Figures 4-6 show typical results such as the HOF, load distribution or load percentage at each row and the deformation or the footprint on the deformable face as potential metrics.

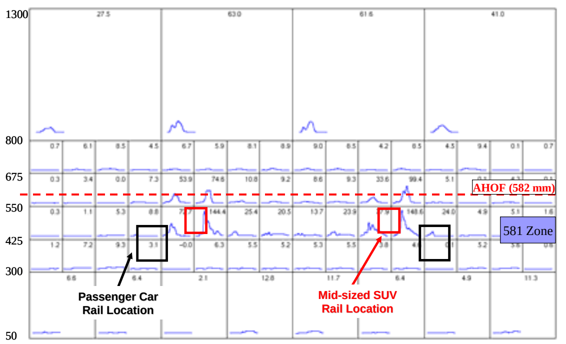


Figure 3. Example of Force-Time history distribution for TRL-type barrier with 125 x 125mm load cells.

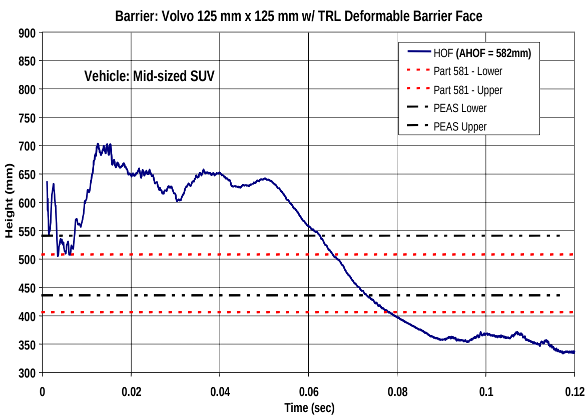


Figure 4. Example of HOF-time history for TRL-type barrier with 125 x 125mm load cells.

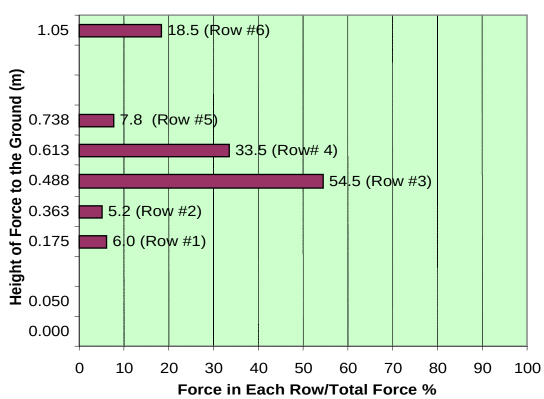
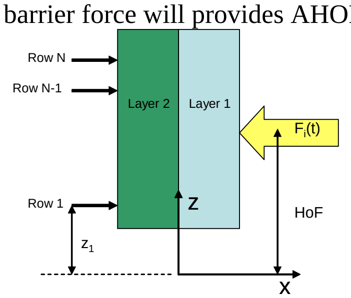


Figure 5. Example of normalized row force distribution for TRL-type barrier with 125 x 125mm load cells.

Figure 6. Example of foot print obtained on a TRL-type barrier deformable face.

5.1 HOF and AHOF Analyses and Conclusions

Figure 7 shows schematic and mathematical definitions for calculating the HOF and AHOF when a force is applied to a barrier, either deformable or rigid, in a normal frontal impact. If all load cells along a certain height are grouped together, the so-called row force may be determined and the height of force (HOF) can then be computed. Normalizing this time dependent height measurement by the total barrier force will provides AHOF.



$$HoF(t) \equiv \frac{\sum_{i=1}^{N} RF_i(t) \cdot z_i}{\sum_{i=1}^{N} RF_i(t)}$$

$$\text{AHoF} \equiv \frac{\sum_t HoF(t) \cdot F_i(t)}{\sum_t F_i(t)}$$

where N is the number of rows and RF_i is the force in row i

Figure 7. Diagram and definition of HOF and AHOF.

TWG members have reported results from crash tests and/or CAE simulations obtained from 56 kph impacts with 125mm x 125 mm rigid LCW, 50mm x 50mm rigid LCW, 125mm x 125 mm deformable LCW using HOF or AHOF metrics. In addition, some of the member companies have conducted vehicle-to-vehicle crashes (test and/or simulation) as a validation. An analysis of NHTSA’s data on Aggressivity Metric (AM) was also conducted to obtain its correlation with AHOF.

When AHOF is calculated for several vehicles it may be compared to the geometrical location of their primary energy absorbing structures. This is done for a small sample of vehicles as shown in Figure 8. Vehicle-to-vehicle crash tests of mid-sized SUV1 (4x4) without SEAS and full-sized SUV (4x2) with blocker beam against a mid-sized passenger car in full frontal impact were also conducted. The AHOF for both SUVs are shown in Figure 8, where HD PU (heavy duty pick-up) AHOF corresponds to that of the full-sized SUV. Both the target and bullet vehicles used a Hybrid III 50th percentile male dummy for the driver and a Hybrid III 5th percentile female dummy for the passenger. The driver seat was at the mid-position while the passenger seat was full-forward. The target vehicle was stationary and the bullet vehicles were moving at a speed adjusted according to impacted vehicles mass ratio to impart a 56 kph velocity change in the target vehicle, which is the passenger car in this case. Figures 9 and 10 show the geometrical alignment, superimposed on 581 zone, for both SUVs front-end structures against that of the passenger car.

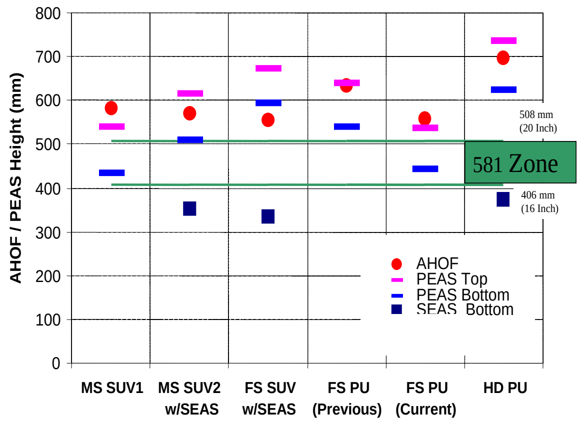


Figure 8. Comparison of AHOF for several vehicles.

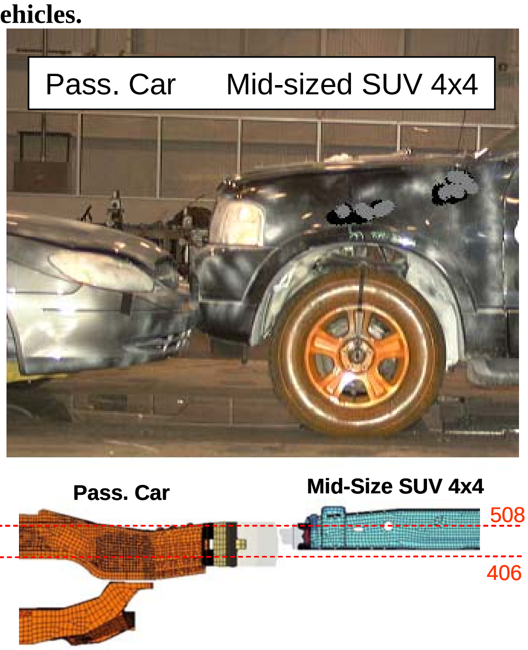


Figure 9. Mid-sized SUV1-to-Passenger car impact.

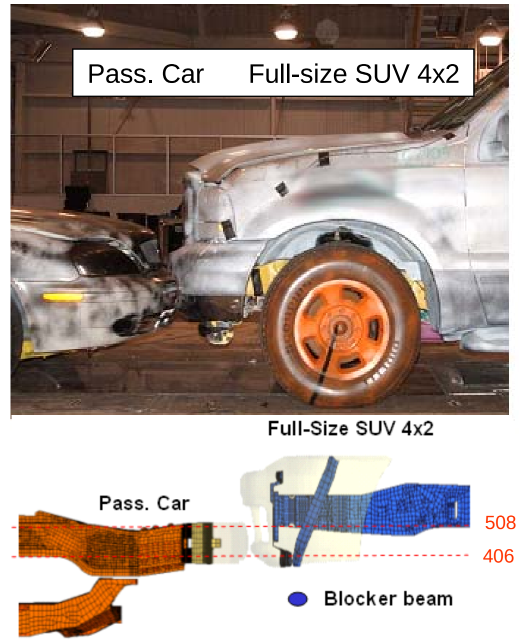


Figure 10. Full-sized SUV-to-Passenger car impact.

Figures 11 shows the driver injury responses of the target vehicle resulted from impacts with mid-sized SUV1 and full-sized SUV that differs in AHOF magnitudes. The injury numbers were normalized to the NCAP values. Occupant responses of the driver hit by full-sized SUV with blocker beam are less severe compared to those resulted from impact with mid-sized SUV1. Comparing results from Figures 8 and 11 it is very clear that the AHOF does not show the beneficial effect of the blocker beam on compatibility demonstrated in vehicle-to-vehicle crash tests. The Full-sized SUV with blocker beam has the highest AHOF compared to that of the mid-sized SUV1 as shown in Figure 8.

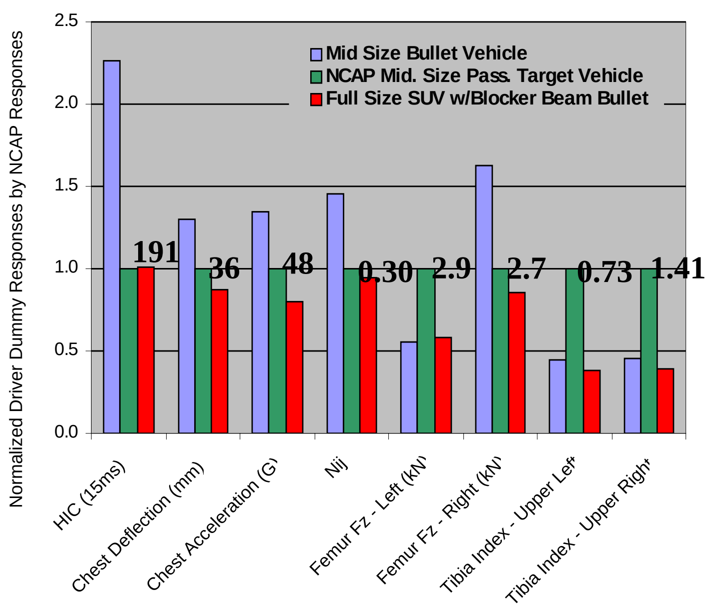


Figure 11. 50th % HIII driver dummy responses of target vehicle.

A separate study considered lowering the overall vehicle height 38 mm and studying the corresponding effect on AHOF [4]. This study concluded that there was little difference in overall AHOF values when vehicle with significantly different PEAS heights were tested against a load cell barrier. In other words, as seen from the data presented in Figures 12 and 13, the HOF and AHOF have significant errors in indicating the change in the actual height of a vehicle and of its structure.

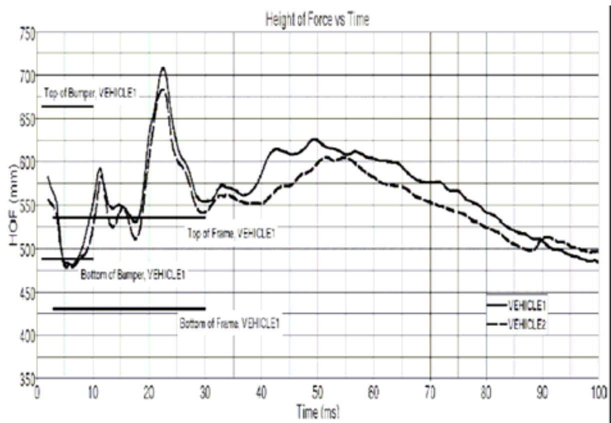


Figure 12. HOF(t) for base vehicle and when lowered 38 mm.

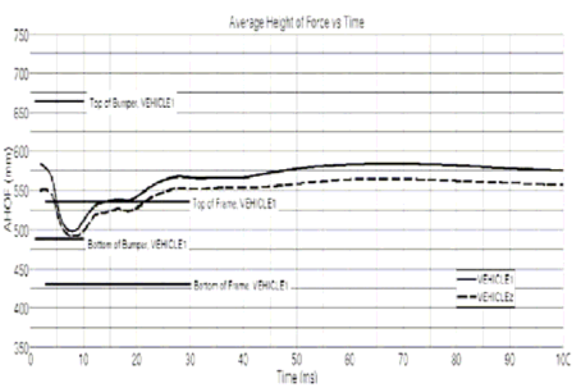


Figure 13. AHOF for base vehicle and when lowered 38 mm.

Conclusions from testing against load cell walls, deformable face barriers, and vehicle-to-vehicle test are that AHOF is not a sufficiently sensitive metric to evaluate the height of the front rail and the effect of SEAS on compatibility. The AHOF does not show the beneficial effect of the Blocker Beam on compatibility demonstrated in the vehicle-to-vehicle tests.

In an effort to determine the correlation between the AHOF measurement and field experience, the fatality related Aggresivity Metric (AM) and AHOF were compared for several vehicles in various categories [5] [1]. To evaluate the characteristics of the vehicle compatibility problem, NHTSA [6] has developed the Aggressivity Metric (AM) which uses the number of driver fatalities in a collision partner normalized by the number of collisions of the subject vehicle (only two vehicle collisions are used) [2]. AM is defined through the relationship:

$$\text{Aggressivity Metric} = \frac{\text{Driver Fatalities in collision partner}}{\text{Number of Crashes of subject vehicle}}$$

The data for this analysis was gathered from two sources. First, AHOF data was obtained from the results of NCAP vehicle tests, measured by a load cell wall, provided by NHTSA. The AM data was provided by NHTSA. The number of vehicle models corresponding to the data appears in Table 1.

1 Toyota Previa Van and T100 PK are included in this analysis even without MY information for the AM value because there was no model change. Therefore, the available AHOF value from any MY of these vehicles can correspond to its AM value.

2 CGNO7424 Ford F150 Pickup Frontal AM (126)

CGNO7628 Chevrolet Tahoe Frontal AM (167)

Table 1.

Number of Vehicle Models Represented in the Datasets

	AM		AHOF
	Front Collision	Side Collision	
# of Vehicles Models	183	201	636

Assumptions

AHOF data was available for a specific subject vehicle from a single model year (MY). However, the AM data did not necessarily correspond to a single MY and therefore, the data was divided into four categories depending on the nature and availability of the dataset:

MY_{xx} ~ MY_{yy}

An AM value for a subject vehicle in the range MY_{xx} ~ MY_{yy} is paired to an AHOF value for a given MY of the subject vehicle in that range. If there are multiple AHOF values that apply to the MY_{xx} ~ MY_{yy} range, then an average is calculated. If the subject vehicle model has many model changes[1] in the MY for which 1 AM value is available and if AHOF values are not available for all model changes within that MY, then the subject vehicle is not included in this analysis.

MY_{xx} ~

In this case, the life of the model is unclear; therefore, the life is assumed to be 4 years. An AHOF value for a particular MY is identified with the AM value for vehicles from MY_{xx} to MY_{xx+4}. If there are multiple AHOF values for vehicles that fit into the range MY_{xx} to MY_{xx+4}, then an average AHOF is calculated.

~ MY_{yy}

In this case, the beginning of the life of the model is unknown. Using the same assumption for model life from #2 above, an AHOF value for a particular MY is identified with the AM value for vehicles from MY_{yy-4} to MY_{yy}. If there are multiple AHOF values for vehicles that fit into the range MY_{yy-4} to MY_{yy}, then an average AHOF is calculated.

MY not listed: For AM values that do not have a MY available, it is not possible to identify a corresponding AHOF value and that particular vehicle cannot be included in the analysis[1]

If there are multiple vehicle models with the same AM value, then an average AHOF value is calculated. The drive train (2WD, 4WD, etc.) is listed in neither the AM nor AHOF datasets. The information on vehicles from the AM dataset includes a designation for the number of doors on a subject vehicle (2-door or 4-door). If the number of doors is not specified for the AHOF value of a subject vehicle, then the vehicle is assumed to be a 4-door vehicle and identified with the AM value for the 4-door subject vehicle. Finally, two outliers with comparable high AM values were excluded.[2]

After applying the above assumptions, the dataset is described in Table 2. Since several AHOF values may be identified with one AM value, AHOF values are used for this analysis to enhance the n-value of the dataset. Further, the number of models with available AHOF data is greater than the number of models with AM data. The database was divided into two categories. Front-to-front corresponds to vehicles colliding in the x-direction from 11 to 1 o'clock. The front-to-side condition corresponds to vehicles struck on either side (7 to 11 o'clock or 1 to 5 o'clock) by the front of the bullet vehicle.

Table 2.

Vehicle Models with Corresponding AM and AHOF Data

	AM Data	AHOF Data
Vehicle models used for front-to-front analysis	77 models (42%)	143 models
Vehicle models used for front-to-side analysis	80 models (40%)	149 models

The data was grouped by vehicle type and by mass. In Figure 14, results by vehicle type show that for passenger cars the correlation was extremely low ($R^2 = 0.002$). For SUVs, AM tends to decrease with increasing AHOF, however this correlation is also extremely low ($R^2 = 0.3$). Conversely for Vans, AM tends to increase with increasing AHOF, also with low correlation ($R^2 = 0.05$). Such low correlation values, whether for increasing or decreasing relationship between AM and AHOF, suggests no relationship between the two variables.

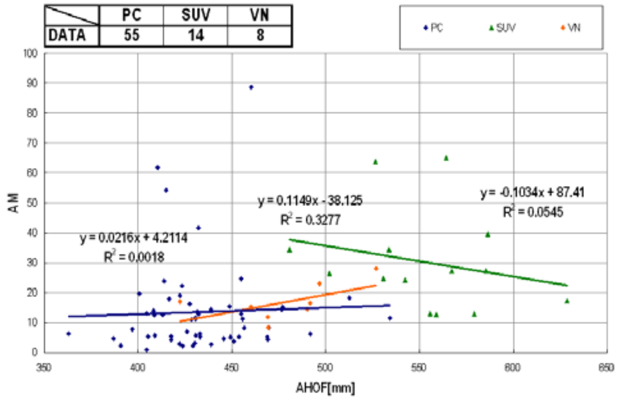


Figure 14. Correlation between the Aggresivity Metric and AHOF for vehicle types.

Finally, the vehicles were divided into weight classes based on their GVWR and the results are presented in Figure 15. The weight classes range from 1000kg to 3000kg in increments of 500 kg giving a total of 4 weight classes. This classification was performed to eliminate any confounding effects due to the different weights. Again it was seen that there was no significant difference in the relationship between AM and AHOF for different weight classes.

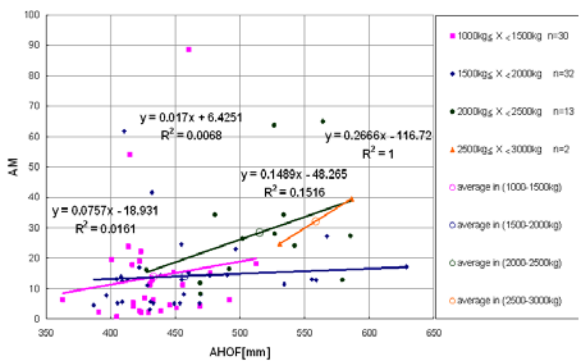


Figure 15. Correlation between the Aggresivity Metric and AHOF for vehicle weight class.

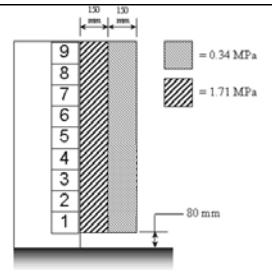
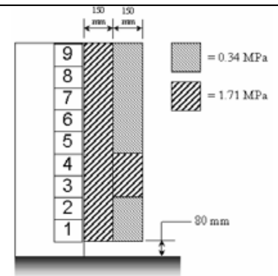
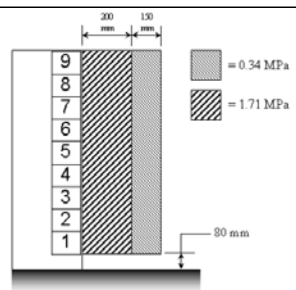
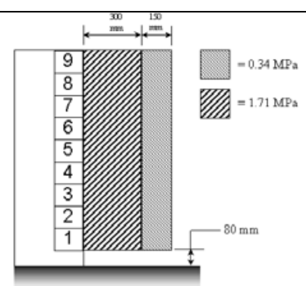
From this study, it can be concluded that there is poor correlation between AM and AHOF. In terms of fatalities, evaluating vehicles using AHOF alone will not necessarily provide reductions in vehicle aggressivity in the field. Continued research of appropriate metrics is recommended to evaluate measures that will improve field compatibility. With AHOF discounted as single compatibility metric, several subgroups were formed to further study vehicle compatibility tests and metrics. At Phase 2, the TWG concluded that geometric compatibility is an important first step to improve compatibility between vehicles. Also, stiffness and geometry must be considered together for the long-term direction of further improvements in fleet-wide compatibility. The current AHOF/HOF definition alone is no sufficiently sensitive to discriminate changes or variations in front-end structures that are beneficial for compatibility. Based on this the TWG formed three sub-groups to support phase II research

6. PHASE II, SUBGROUP 1: FIXED BARRIER TESTS AND METRICS

This subgroup was organized to evaluate potential changes to the TRL deformable barrier to improve SEAS detection and to explore new LCW metrics that could be used with a full overlap test to predict structural interaction. It was determined that a deformable element barrier should be used for investigation in lieu of a rigid wall for several reasons. Foremost is that a deformable barrier would allow for improved detection of secondary energy absorbing structure, which can be set back from the front of the vehicle and otherwise undetectable in an impact with a rigid wall. A deformable barrier can also reduce the high decelerations that can result from stress wave effects at the front of the rails in rigid wall impacts with the effect that the initial phase of the impact is more representative of vehicle-to-vehicle impacts. Additionally, deformable barriers reduce engine dump loading that may otherwise confound the measured force data and can detect strain effects due to cross-car load transfer through crossbeam structures.

Where appropriate, barrier tests designed to assess compatibility should be adaptable to current NCAP / FMVSS 208 test setups, in order to minimize number of tests necessary during vehicle development. The baseline deformable barrier was developed by TRL consisting of two 150 mm thick layers of aluminum honeycomb. The stiffness of the layers is 0.34 MPa and 1.71 MPa for the front and rear layers, respectively. The second layer of the baseline barrier is segmented along each load cell row and column, meaning the deformable layer will not transfer load to adjacent cells. Using this design as a baseline configuration, three modifications were identified for exploration (seen in Table 3).

Table 3.
Barrier configurations considered for evaluation

Baseline (TRL) Barrier	
Barrier 1	
Barrier 2	
Barrier 3	

Barrier 1 adjusts the stiffness of only rows 3 and 4 in the front-most layer to 1.71 MPa. The intended purpose here was to provide a path for the secondary energy absorbing structure (SEAS) to transfer force to the barrier. The second barrier and third barriers increase the thickness of the second layer by various degrees to determine if added depth would allow the barrier to reach further back into the test vehicle to pick up the SEAS. In the case of the third barrier, the rear layer is not segmented as it is in the baseline TRL barrier. This is necessary to avoid crush instability in the honeycomb. Four metrics were proposed by the TWG for barrier evaluation, as defined in Tables 4 and 5.

Table 4

Definitions for proposed barrier metrics

Metric 1: Total force in select rows	$\mathrm{M1} \equiv \max_t\left[RF_3(t) + RF_4(t)\right]$
Metric 2: Total force % in select rows	$\mathrm{M2} \equiv \dfrac{\max_t\left[RF_3(t)\right] + \max_t\left[RF_4(t)\right]}{\sum_{i=1}^{N} \max_t\left[RF_i(t)\right]}$
Metric 3: Homogeneity, distribution of force in selection area	$\mathrm{M3} \equiv \dfrac{\sum_{i=1}^{nc}\left(\dfrac{L - f_i}{L}\right)^2}{nc}$
Metric 4: Distribution of deformation into layer 2 in Evaluation Area	$\mathrm{M4} \equiv \dfrac{A_C}{EA}$ for all $u_x^{Frt}(y,z) > 150\mathrm{mm}$
Other Metrics have also been proposed such as vertical and horizontal [negative] deviation from a target value.	

Table 5.
Notation used in metric definitions.

$$A_C \equiv \begin{cases} \text{Area of the simple, closed curve defined} \\ \text{by nodes/points with} > 150\,\text{mm deformation.} \\ \text{Green's Theorem} \Rightarrow A_C = \frac{1}{2}\int (ydz - zdy) \end{cases}$$

$$f_i \equiv \text{(peak) load } F_i(t) \text{ in load cell i in HSA} = \max_t\left[F_i(t)\right]$$

$$EA \equiv \text{Evaluation Area, } 250 \cdot 1500 = 375{,}000\,\text{mm}^2$$

$$HSA \equiv \text{Homogeneity Selection Area}$$

$$L \equiv \text{(peak) average load per cell in } HSA = \frac{\sum_{i=1}^{nc} f_i}{nc}$$

$$nc \equiv \text{number of load cells in } HSA$$

$$N \equiv \text{number of barrier Rows}$$

$$RF_i(t) \equiv \text{Force-time history of Row i}$$

$$TF(t) \equiv \sum_{i=1}^{N} RF_i(t), \text{ force-time history for barrier}$$

$$u_x^{Frt}(y,z) \equiv \text{axial displacement for nodes on front barrier face lying within EA}$$

Homogeneity Selection Area is the area covered by load cells lying in rows whose peak row force is > 5% of the peak total force AND whose peak column force is > 3% of the peak total force.

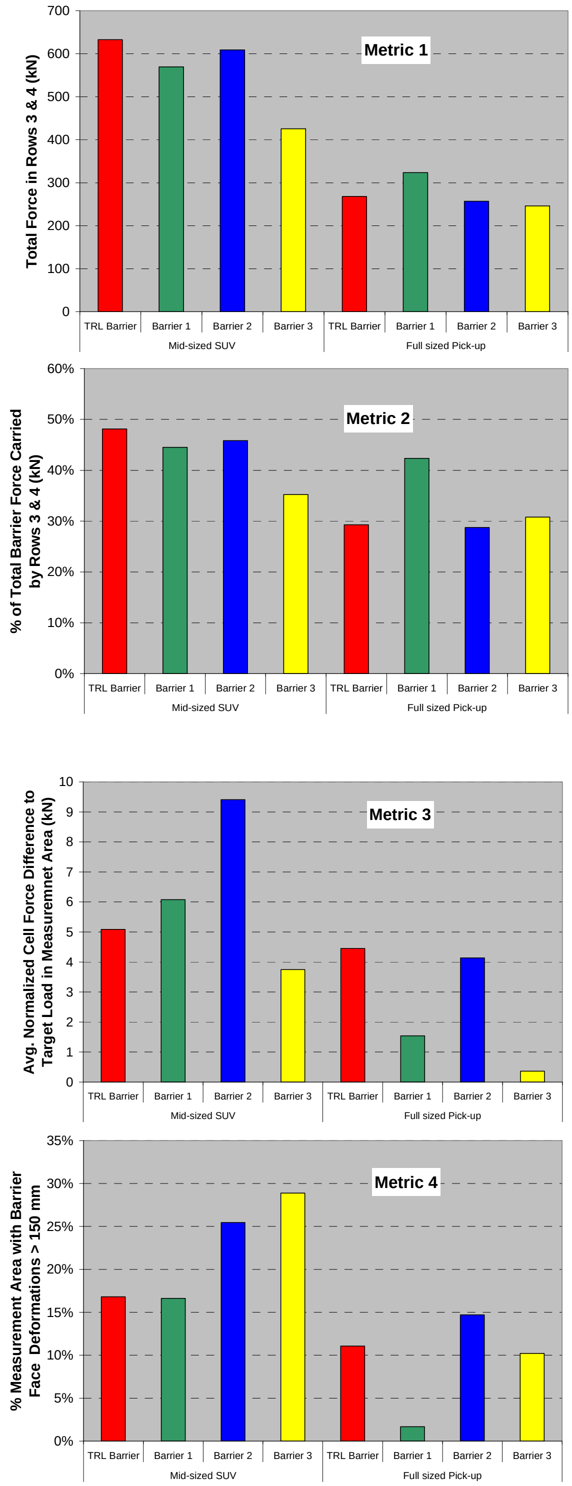
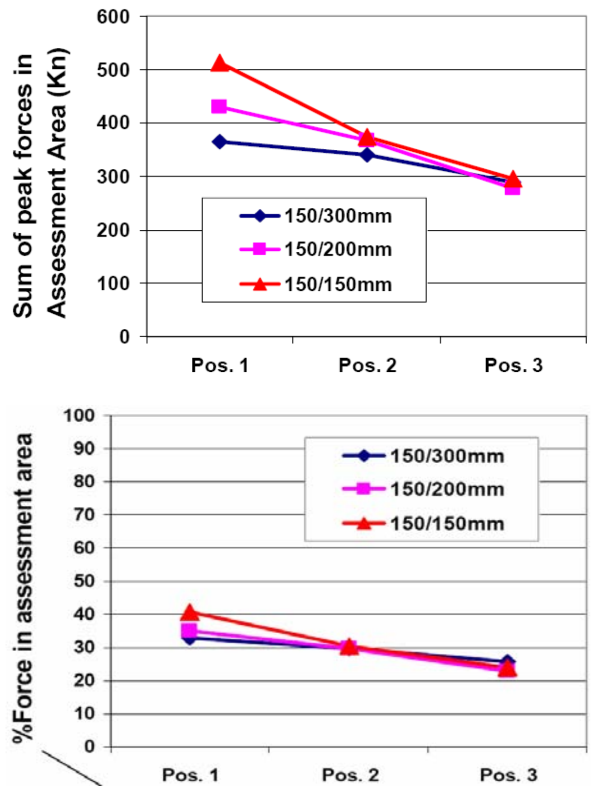


Figure 16. Comparison of for metrics for various barrier designs using simulation.

Barrier 1 adjusts the stiffness of only rows 3 and 4 in the front-most layer to 1.71 MPa. The intended purpose here was to allow the secondary energy absorbing structure (SEAS) to transfer force to the barrier. The second barrier increases the thickness of the layer 2 to 200 mm. The third barrier configuration continues to thicken the second layer to 300 mm as well as eliminating the segmentation of it. This non-segmented characteristic was intended to investigate the possibility of capturing the lateral load transfer actual vehicles experience

Metrics one and two use a force-based criterion to measure barrier differences. A third study investigated changing the location of a SEAS structure on detection by layered barriers. A change in the depth of the second layer did not appear to affect the detection of SEAS as seen in Figure 17 below.



$$\frac{\sum \text{force maxima within assessment area}}{\sum \text{force maxima total}}$$

Figure 17: Effect of changing the forward position of SEAS on force in an assessment area.

It can be seen from Figure 17 that changing the depth of the second barrier layer leads to less effective SEAS detection. Examining current test results with rigid barriers (LCW), deformable barriers (LCW), and further work has been initiated to develop a simple test procedure and metrics using a LCW as a compatibility measurement tool.

The subgroup has also investigated the International Harmonized Research Activity (IHRA) compatibility working group phase I test proposal [7]. The test configuration is a full width test carried out at 56 km/h into a wall equipped with an array of 125 mm x 125 mm load cells and the TRL deformable barrier face. The aim of the proposal is to ensure that all vehicles have sufficiently strong structure in a common interaction zone. This zone is defined vertically as 330-580 mm high, essentially the third and fourth load cell rows when the LCW is positioned with a ground clearance of 80 mm. A new metric based on peak cell loads has been proposed. It consists of vertical and horizontal components. These are complementary but could be applied separately.

The aim of the vertical component is to ensure that there is sufficient vehicle structure in alignment with the common interaction area. It sets a target row load and calculates the load below the target row for each row in the common interaction zone, i.e. rows 3 and 4.

$$D_{VNT} = \sum_{Row(i)=3}^{4} IF\left[\{R_i \leq TR_i\} THEN, ABS(R_i - TR_i),\ ELSE = 0\right]$$

where:

$$\text{Row Load } R_i = \sum_{j=1}^{allcolumns} f_{ij}$$

TR_i = Target row load
F_{ij} = peak cell load

If a performance limit of zero is required for this metric, then it is effectively a minimum row load requirement. The subgroup examined test data from 8 FWDB tests with various LTVs and showed that a row load greater than 100 kN was a good indicator if the LTV had either PEAS and / or SEAS in alignment with that row provided that the SEAS had a crossbeam structure.

The aim of horizontal component is to assess if crossbeam(s) or comparable structure on SEAS have sufficient strength. The metric would encourage a crossbeam strength that tended to match the stiffness of the front of the longitudinals. It sets a target cell load for the row based on the total row load and calculates the load below target cell load for each cell between the rails for each row in the common interaction zone. The subgroup intends to evaluate this metric further in future work.

In summary, the subgroup intends to perform additional work to evaluate the IHRA proposal.

Major issues that this work will address include the test robustness, in particular its sensitivity to the vertical alignment of the vehicle with the LCW, and validation, including the degree to which the metric affects the fleet and the benefits of changing to meet it. The TWG will continue their research to evaluate the proposed and new compatibility metrics.

7. PHASE II, SUBGROUP 2: VEHICLE-TO-VEHICLE OR MDB-TO-VEHICLE IMPACT TEST

The purpose of this subgroup's activities was to study vehicle-to-vehicle impacts with the focus of developing a performance protocol for classification of the under-ride/over-ride condition and also to provide an alternative performance procedure to simplify the geometry matching of PEAS/SEAS. Additional research will be towards development of a uniform test protocol for Phase-III research.

Figure 18. Full-frontal, vehicle-to-vehicle testing between mid-sized SUV and passenger car.

Figure 19. Full-frontal, MDB-to-vehicle testing for passenger car.

Objectives for this task are the development of requirements for vehicle-to-vehicle simulation and crash tests to demonstrate the minimization of Under-Ride/Over-Ride in a vehicle-to-vehicle frontal impact conditions. It is desired to establish a single standard partner (the struck) vehicle to be used for all tests. This vehicle will be a mid-size passenger car and will be representative of a model with a four-star rating and a weight around 1600 kg. The moving deformable barrier (MDB) will represent this average. And each OEM will be able to test a mid-sized vehicle (~1750 Kg, 4*, Acceptable) with it. The MDB would provide an equivalent target for all OEM compatibility testing.

The goal of establishing a uniform test vehicle is to enable manufacturers to start engineering for improved compatibility as soon as possible. Therefore, it is recommended that a manufacturer be allowed to select a vehicle as a 'reference passenger car' according to the following three criteria: First, select a high volume mid-size or smaller passenger car that is sold in the USA as a regular production vehicle and the vehicle mass should not exceed 4000lbs (NHTSA NCAP test configuration). Secondly, the vehicle must be rated at least as 4 stars in US Frontal NCAP test. Lastly, the vehicle must be rated as at least 'Acceptable' in IIHS ODB tests. These target characteristics will ensure a good balance between equitable requirement and a quick start. The TWG will undertake the development of one uniform test vehicle surrogate as mentioned earlier. The first priority test mode will be Full overlap using 50th & 5th percentile dummies, and the second priority will be partial overlap, again using 50th & 5th percentile dummies. Determination of the particular measurement criterion is an ongoing issue. Potential candidates are the use of NASS data including partner vehicle acceleration and cabin intrusion, occupant performance measures, and IIHS intrusion values.

The first option for the measurement of injury is occupant compartment accelerations using both vehicles front and rear sills, similar to the NCAP test. The use of the dummy's injury values (as in full-frontal and offset testing) are options. In this case, the performance criterion for this test would be pulse severity not to exceed the US-NCAP performance. Secondly, occupant compartment intrusion values are being considered. Here, a partial overlap test could be utilized and the body side aperture deformation measured. Open issues are the selection of a target vehicle and the selection of static post-test points to be measured (e.g. I/P, Dash, Cowl, Toe-Board, or the steering column).

It should be noted that an additional over-ride/under-ride performance criterion is being considered. If in the NCAP full frontal test, a plane of maximum crush is established, the "NCAP plane", is defined, then a "Safety Zone" extending from the NCAP plane to a reference point (for instance the 'A' Pillar, Windshield, etc.) would be defined (see Figure 20). Future structural designs under consideration should ensure that the rails of the LTV do not intrude into the safety zone beyond a [TBD] value.

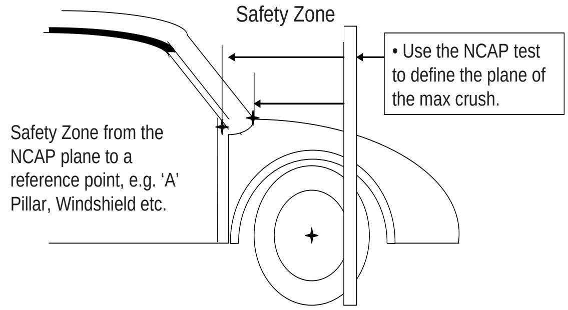


Figure 20. Definition of "NCAP safety plane".

8. PHASE II, SUBGROUP 3: SUPPLEMENTARY TEST FOR SEAS EVALUATION

Efforts for this subgroup's activities centered on development of a performance metric and evaluation procedure for Secondary Energy Absorbing Structure (SEAS) that meet the criteria proposed in TWG Phase I recommendations (Option 2). SEAS Evaluation Candidates investigated were the Dynamic Override Barrier test and Quasi-static test based on ECE R93 (Front Under-run Protection). Both tests and simulations were conducted for development of this evaluation.

The Subgroup examined SEAS structures with respect to over-ride potential through the test setup seen in Figure 21. For a test vehicle weight of 1967 kg with a speed of 19.8 kph and barrier upper surface height of 508mm, the strength (force-displacement performance) of the primary front structure components was determined. The overlap of the barrier with the bumper beam is 56mm, however, the frame (PEAS) did not overlap the barrier since the barrier width was 400mm.

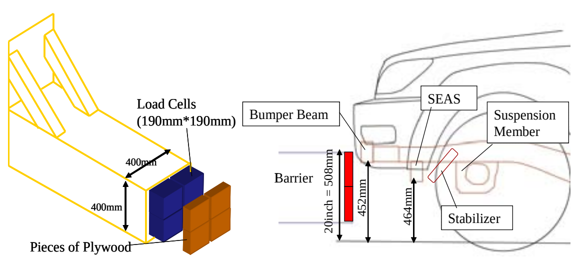
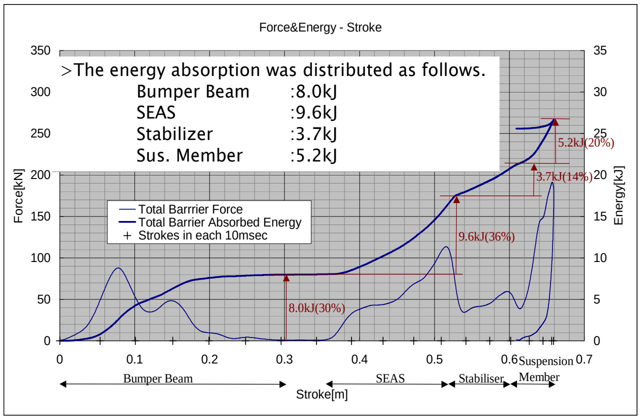


Figure 21. A proposed hardware and test setup for over-ride and SEAS strength testing.



Force-Stroke and Energy-Stroke of total barrier force

Figure 22. Force-displacement performance for various front structure components.

This test showed an energy distribution between the bumper beam, SEAS, stabilizer bar, and suspension members of 8.0, 9.6, 3.7, and 5.2 kJ, respectively. It should be noted from Figure 22 that there is 150 mm stroke span between the bumper beam and initial force accumulation of the SEAS. The majority of the energy is dissipated by the bumper beam and SEAS.

SUV-to-Barrier Simulation included SUV-to-override barrier (full width barrier = 2000 mm). The barrier was lowered to clear other front structures and impact the SEAS first and the vehicle speed was 56 kph.

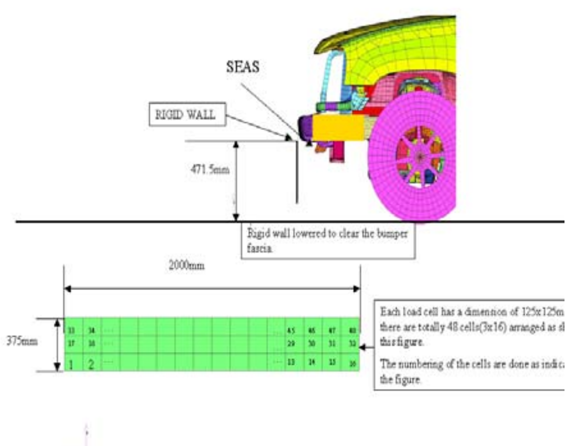


Figure 23. Partial rigid wall simulation for evaluation of SEAS.

The SEAS evaluated were shown to be effective though direct loading simulations by a partial rigid wall (see Figures 23 and 24).

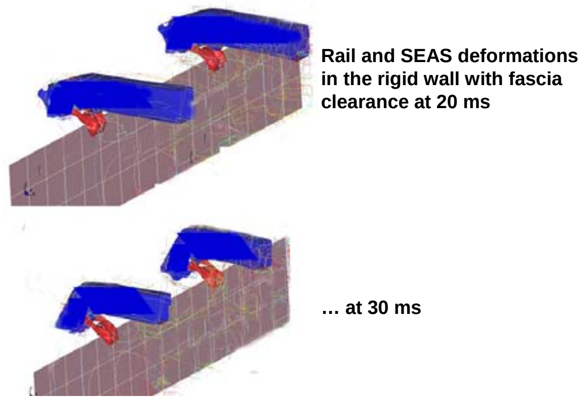


Figure 24. Rail deformations when impacted by a partial rigid wall.

This additional proposal was a low, continuous load cell barrier. Simulations were done where only the SEAS were contacted to evaluate their effectiveness. The effectiveness of SEAS was first examined in vehicle-to-vehicle tests and in these simulations.

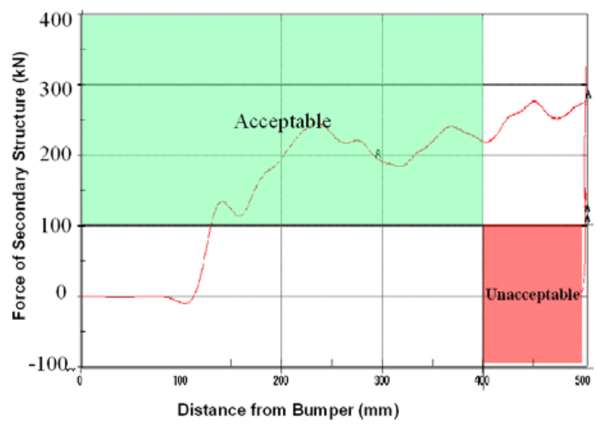


Figure 25. Proposed force – displacement ranges of acceptability for SEAS.

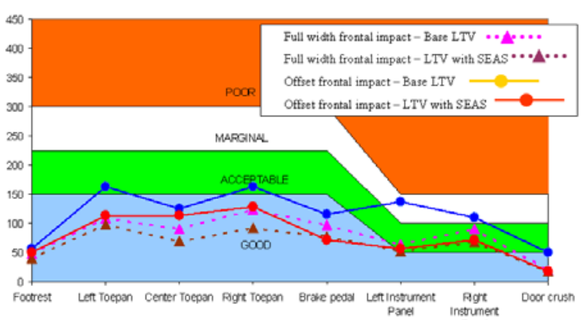


Figure 26. Ranges of acceptable performance for LTV with and without SEAS.

SUV-to-car simulations including non-continuous and continuous SEAS were investigated by the subgroup 3. For all SUVs simulated (2100kg – 2900kg), the SEAS was shown to be effective in reducing intrusions to the struck car (Figure 26) if a minimum force of 60 kN is seen by each rail with less than 400 mm of displacement. This amounts to a total force of 120 kN on the SEAS.

The TWG has agreed on the following test procedures and performance criterion for SEAS. The SEAS shall withstand a load of at least 100 KNewtons exerted by a loading device, as described in the attached Appendix A, before this loading device travels 400 mm as measured from a vertical plane at the forward-most point of the significant structure of the vehicle.

9. PHASE III: STUDIES FOR FRONTAL COMPATIBILITY IMPROVEMENT

In this phase of the research, focus will be on stiffness matching and passenger car structural integrity. This will pertain to the study of front-end stiffness performance by investigating tests over the next two years to determine appropriate front-end stiffness characteristics and criteria to evaluate small vehicle passenger compartment strength and integrity. The criterion will be to develop a test procedure to enhance partner protection without any significant decrease in self-protection. Test procedures under consideration are load cell barrier tests and vehicle-to-vehicle tests.

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10. REFERENCES

[1] Hollowell, W.T. and Gabler, H.C. “NHTSA’s Vehicle Aggressivity and Compatibility Research Program”, 15th International Conference on the Enhanced Safety of Vehicles, Paper No. 96-S4-O-01, Melbourne, Australia, 1990.

[2] Gabler, H.C. and Hollowell, W.T., “ The Aggressivity of Light Trucks and Vans in Traffic Crashes”, SAE Paper No. 980908, Detroit, 1998.

[3] Summers, S., Prasad, A. and Hollowell, W.T., “ NHTSA’s Vehicle Compatibility Research Program”, SAE Paper No. 1999-01-0071, Detroit, 1999.

[4] Verma, M. K. , Nagappala, R. , Tung, Y. J. , "Significant Factors in Height of Force Measurements for Vehicle Collision Compatibility", SAE Technical Paper 2004-01-1165, Warrendale, PA, 2004.

[5] "Analysis of the Correlation Between a Vehicle Compatibility Performance Metric and Vehicle Aggressivity", Toyota Draft paper, Feb 2, 2005.

[6] Gabler, H. C. and Hollowell W. T., “NHTSA’s Vehicle Aggressivity and Compatibility Research Program”, Proceedings of the Sixteenth International Enhanced Safety of Vehicle Conference, Paper No. 98-S3-O-01, Windsor, Canada (1998).

[7] Edwards, M., "Additional Results for TWG Compatibility", Presentation to Alliance TWG, Feb, 2004

APPENDIX A

Test procedure for Phase II Recommendation for SEAS Conformance to these requirements may be assessed by either of the two procedures below.

Procedure A: Quasi-static Force Application for Evaluating Secondary Structure

1 Definitions

1.1 Secondary Energy Absorbing Structure – SEAS

2 Requirements

2.1 SEAS Location. The SEAS must be connected to the primary energy absorbing structure of the vehicle.

2.2 SEAS Strength. The SEAS must resist the force level specified in S2.2.1 without exceeding the total force application device travel distance specified in S2.2.2.

2.2.1 A minimum force of 100 kN

A maximum horizontal travel of the force application device of 400 mm as measured from the forward-most point of the significant structure of the vehicle. The forward-most point of the significant structure of the vehicle as defined at 3.3.6.

2.3 Secondary Energy Absorbing Structure – SEAS

3 SEAS Test Procedures. The procedures for evaluating the SEAS to the criteria of S2 are specified in S3.1 through S4.0

3.1 Force Application Device. The force application device employed in S3.4 of this section consists of a rectangular solid made of rigid steel. The steel solid is 125 mm in height, 25 mm in thickness. For the measurements, the top edge of the solid shall be placed so that its first contact is only with the SEAS. The width of the solid must be at least the horizontal (y-direction) dimension of the SEAS. The face of the block is used at the contact surface for application of the forces specified in S2.2.1. Each edge of the contact surface has a radius of curvature of 5 mm plus or minus 1 mm.

3.1.1 The solid rectangle of S3.1 shall be rigidly attached to a device capable of applying quasi-static load as specified in S3.4.

3.1.2 Instrumentation with a minimum accuracy of 5 percent plus or minus 5 percent shall be used for measuring the load and will be placed in the force application device so that it measures the actual load being transmitted into the vehicle SEAS.

3.1.3 Travel of the force application shall be measured in a horizontal direction from the point of foremost significant structure on the vehicle, this 'foremost point of significant structure' as defined at 3.3.6.

3.2 Vehicle Preparation. The vehicle should be prepared such that it is secured in the stationary position.

3.2.1 The vehicle must be secured on a rigid, horizontal fixture (± 0.250º) so that it is adequately restrained at the vehicle underbody and also at the sides to prevent rearward movement of the whole vehicle during the test. Good engineering judgment will be required to provide maximum support, for the maximum area possible.

3.2.2 Secure the vehicle in the tie-down fixture as described in S3.2.1. A sufficient number of horizontal and vertical tie-downs shall be used to prevent movement under load. The vehicle may be secured to the loading fixture using wire rope, turnbuckles, strap plates, etc.

3.2.3 An unyielding vertical face shall support the vehicle rear bumper to prevent rearward movement.

3.3 Positioning the Force Application Device. Before applying any force to the guard, locate the force application device such that:

3.3.1 The center point of the contact surface of the force application device is aligned with the SEAS at the vehicle horizontal centerline.

3.3.2 The force application device top edge shall be no higher than 455 mm

3.3.3 The force application device must be vertically positioned so as to insure that the first point of contact during the test is with the SEAS.

3.3.4 If necessary to achieve the condition achieved in S3.3.3, any structure in front of the SEAS should be removed before force application.

3.3.5 The longitudinal axis of the force application device passes through the horizontal centerline of the vehicle and is perpendicular to the vertical axis of the vehicle.

3.3.6 Forward-most Point of Significant Structure: The forward-most point of significant structure on the vehicle is defined as the first point on the vehicle structure that participates in the management of the forces generated in high severity frontal crashes.

3.3.7 Alignment: The front face of the force application device is aligned with the horizontal plane passing through the foremost point of significant structure on the vehicle.

3.4 Force Application: After the force application device has been positioned according to S3.3 of this section, apply the load per the force application procedures described in S3.4.1 through S3.4.2.3.4.1

3.4.1 Rate of Travel: Apply force continuously such that the force application device travel rate does not exceed 12.5 mm per second until the minimum force in S2.1.1 has been exceeded or until the force application device has traveled the total distance in S2.1.2 from the position in S3.3, whichever occurs first.

3.4.2 Direction of Travel: During each force application, the force application device is guided so as to travel only horizontally in a direction perpendicular to the surface of the device during the entire test. At all times during the application of force, the location of the longitudinal axis of the force application device remains constant.

Procedure B: Dynamic Force Application for Evaluating Secondary Structure

4.1 As an alternative, this measurement may be made with a ‘loading attachment’ to a fixed barrier. The vehicle will move into this attachment at the minimum velocity that will result in at least 400mm of horizontal travel by the forward-most point of the significant structure of the vehicle. The movement of the vehicle shall be horizontally in a direction perpendicular to the plane of the loading attachment.

4.2 This attachment shall be designed to perform as the force application device described in S3 for the quasi-static test procedure and will have the same dimensions and instrumentations.

4.3 The test shall be performed by removing as necessary any structure in front of the SEAS (e.g. bumpers, fascias etc) so as to insure that the first point of contact of the loading attachment is with the designated SEAS on the vehicle.

APPENDIX B

Testing to Support Development of Dynamic Test Procedures and Performance Criteria to Promote Geometrical Compatibility

F-to-F Compatibility Proposed Tests	**BARRIER TESTS**		**VEHICLE-TO-VEHICLE TESTS**		
	NCAP With 125mmx125mm Load Cell	**TRL Barrier**	**Small Size Pass. Car**	**Mid Size Pass. Car**	**Full Size Pass. Car** As Indicated by Other Test Results
Mid SUV WITHOUT SEAS (Secondary energy absorbing structure)	Physical Tests 50x50 Load cells GM - 2 tests -Jan 15, 04 Simulations DCX GM - complete BMW	Physical Tests GM –2 tests March 1, 04 Explorer (pre-2002) [4900 lbs] Explorer (Current) [4900 lbs] Simulations DCX GM - March 1, 04 BMW	Physical Tests MMC(Japan Spec Veh)	Physical Tests Explorer (Current) - 50% Offset/Collinear [4900 lbs] -Full engagement/ Collinear [4900 lbs]	
Mid SUV WITH SEAS	Physical Tests Honda Toyota (4 Runner) Simulations DCX MMC GM	Physical Tests GM – March 30, 04 Ford Honda Toyota (4 Runner) Simulations DCX MMC GM – March 15, 04	Physical Tests Toyota (4 Runner) Toyota (4 Runner-60mm)		
FULL SUV WITHOUT SEAS	Physical Tests Nissan *Expedition (pre-2003) * [5650 lbs] *Expedition (Current) * [5900 lbs] *50mmX50mm Simulations DCX	Physical Tests Nissan Simulations DCX			
FULL SUV WITH SEAS	Physical Tests Simulations	Physical Tests Simulations		Physical Tests Excursion (Current) - Full engagement/ Collinear [7500 lbs]	
SMALL PICKUP WITHOUT SEAS					
SMALL PICKUP WITH SEAS					

Testing to Support Development of Dynamic Test Procedures and Performance Criteria to Promote Geometrical Compatibility, Cont'd.					
F-to-F Compatibility Proposed Tests	**BARRIER TESTS**		**VEHICLE-TO-VEHICLE TESTS**		
	NCAP With 125mmx125mm Load Cell	**TRL Barrier**	**Small Size Pass. Car**	**Mid Size Pass. Car**	**Full Size Pass. Car** As Indicated by Other Test Results
MEDIUM PICKUP WITHOUT SEAS	Simulations DCX (1500 Series)	Physical Tests F150 (Current) [5200 lbs] F150 (2004) [5800 lbs] Simulations DCX (1500 Series			
MEDIUM PICKUP WITH SEAS	Simulations DCX (1500 Series)	Simulations DCX (1500 Series)			
LARGE PICKUP WITHOUT SEAS	Physical Tests GM – 2 Tests completed Simulations DCX (2500 Series) GM – completed	Physical Tests Ford Simulations DCX (2500 Series) GM – April 30, 04			
LARGE PICKUP WITH SEAS	Simulations DCX (2500 Series)	Physical Tests F250 (Current) [7400 lbs] GM – April 30, 04 Simulations DCX (2500 Series) GM – April 30, 04			
LARGE SEDAN	Physical Tests	Physical Tests			
SMALL SIZE CAR	Physical Tests Honda	Physical Tests Honda VW	NA	NA	NA
MID SIZE CAR	Physical Tests Honda	Physical Tests VW Honda	NA	NA	NA