# ON THE COMBINED EFFECT OF THE VARIATION OF DRIVER RESTRAINT CONFIGURATION AND CRASH PULSE FOR HIGH AND VERY HIGH IMPACT VELOCITIES

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**ABSTRACT**

The prerequisites for achieving effective driver protection and the influence of crash pulse shape in 56 and 80 km/h fully distributed perpendicular frontal impacts were investigated. The pulses used were based on the basic external dimensions and properties of a midsize car. The analysis was carried out by combining mathematical analysis with mechanical sled tests.

An analytical tool to generate synthetic realistic crash pulses was developed. Using the tool crash pulses for 56 and 80 km/h frontal impacts in which the initial deceleration level, due to buckling and the total stopping distance were varied. Buckling deceleration levels were varied between 200 and 400 m/s$^2$ and total stopping distances were varied between 0,5 and 0,9 m.

The implications of the shape of the frontal crash pulse and the design of the interior driver protection system for impact velocities of 56 and 80 km/h were analytically studied using computer occupant simulation software. The most promising mathematical modelling results were subsequently evaluated by means of mechanical sled tests. Based on the model predictions injury risk assessments were made for the two different impact velocities

At 56 km/h impact velocity the analysis show that in relation to the FMVSS 208 injury criteria levels a very low injury risk for the driver is achievable with a conventional restraint system and a stopping distance of 0,5 meter and an initial buckling zone of 0,3 m with a deceleration level of 200 m/s$^2$.

At 80 km/h impact velocity the computer analysis and the sled tests show that the dummy responses can be kept below what is stipulated in the FMVSS 208 with an advanced restraint system and a stopping distance of 0,7 m and a square wave type of pulse. That is also true if the stopping distance is increased to 0,9 m.

**Keywords**: Dummies, Compatibility, Models, Airbag, Seat belt, Sled test, Crash Pulse

**IT HAS BEEN** estimated that automotive transportation has killed approximately 25 million people during the last hundred years (Trinca et al 1988). Today about a million people die annually world wide as a consequence of road trauma and perhaps as much as 15 million are being injured every year (Trinca et al 1988). A major part of car occupant fatalities still occur in frontal collisions. In a study from 1997 it was concluded that “high speed testing is necessary to reduce fatalities which occur with a median Equivalent Energy Speed (EES) of 70 km/h in frontal collisions” (Thomas 1997). In the proposed rule making for FMVSS 208 document it can be observed that many occupants are killed in frontal impacts with a delta-v of 56 km/h (35 mph) and above (Department of Transportation 2003) (Figure 1).

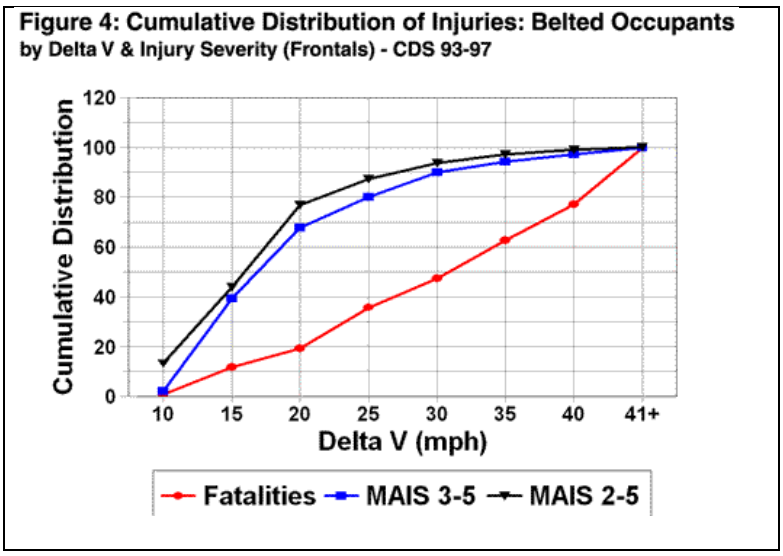


Fig. 1 - Distribution of Injured Belted Occupants by Delta V & Injury Severity (Department of Transportation, 2003)

In the perfect world of road transportation casualties are minimal. The Tomorrows automotive transportation system and speed limits will, to a large extent, be built on what is possible to achieve in terms of passive and active safety. Today's cars have reached a certain level of passive crash protection in frontal collisions but people still get injured or die in frontal collisions due to high speed collisions, incompatibility issues, collisions with narrow objects etc. Frontal collisions with very high delta v (70-90 km/h) are infrequent In addition making meaningful delta-v assessment of such crashes is difficult. Therefore available crash injury statistics is uncertain. It is, however, imperative to research into the possibilities and limitations of frontal passive crash protection in order to create the building blocks for future automotive transportation systems with a very limited number of casualties. This work is an attempt to understand what level of frontal crash protection could be design into a hypothetical mid sized passenger car using advanced knowledge on future crash pulse and restraint system adaptivity.

One of the most effective countermeasures has been the improvement of the crash safety of cars, the so-called “passive safety”. Consequently significant advances in frontal crash protection of passenger cars have been made during the last fifty years. With the introduction of the “New Car Assessment Programs” the crash performance of modern cars have been pushed beyond what is stipulated in mandatory safety standards.

The frontal crash performance of a car is depending of several sub systems such as the deformation characteristics of the front structure, the structural integrity of the car body and the interior restraint system etc. In the early seventies the ESV research project set up a goal to reach frontal crash protection at 80 km/h (50 mph) and indeed some crash tested research vehicles performed well at high impacts velocities (Carter 1972). For a number of reasons, such as cost, weight, size and limitations in available technology it was deemed that such a target was not realistic to apply on the car fleets of the time period. Focus was set at 50 km/h (30 mph) frontal impact crash performance and then later on most cars were to be designed for a crash performance tuned to protect at up to 56 km/h (35 mph) in full frontal impacts and up to 64 km/h (40 mph) in off set impacts (Bendjellal et al 1997).

One important aspect of the conventional crash protection theory has been to avoid “sub-optimisation” by designing for the most frequent crash speed in which occupants are injured or killed (Norin et al 1991, Horsch 1987). The reasoning being that designing for crash protection at very high impact velocities may lead to too stiff car fronts and to too stiff restraint systems resulting in less protection at the most common crash speeds. It is not until recently that new technology has made it possible to avoid this conflict by introducing adaptivity in the performance of the interior restraint systems (Yang et a 2001, Miller and Maripudi 1996, Mackay et al 1998). In the future also the crash pulse i.e. deformation characteristics of the front structure can be made adaptive (Pipkorn 2005 I).

In a previous study frontal crash protection at 80 km/h (50 mph) with a crash pulse from a crash tests at 80 km/h was investigated (Mellander 2005, Pipkorn et al 2005 II) It was found that for a stopping distance of 0.9 m a restraint system can be designed for efficient protection of a vehicle occupant.

Modern vehicles do not have 0.9 m available deformation distance. Therefore crush distance and crash pulse shape was introduced in this study. The deformation characteristics of the car governing the frontal crash protection cannot be isolated from other issues such as size, cost, mass and crash compatibility. The crash compatibility issues have been widely discussed during the last decades. Therefore crash compatibility considerations were included in the study.

In a study in Japan the frontal crash pulse at 56 km/h (35 mph) into a flat rigid barrier for a large number of cars was examined (Ono et al 2003). There is a significant spread in acceleration level (Figure 2). In the same study the force deflection characteristics of the full frontal crush zone were averaged by vehicle type (Figure 2).

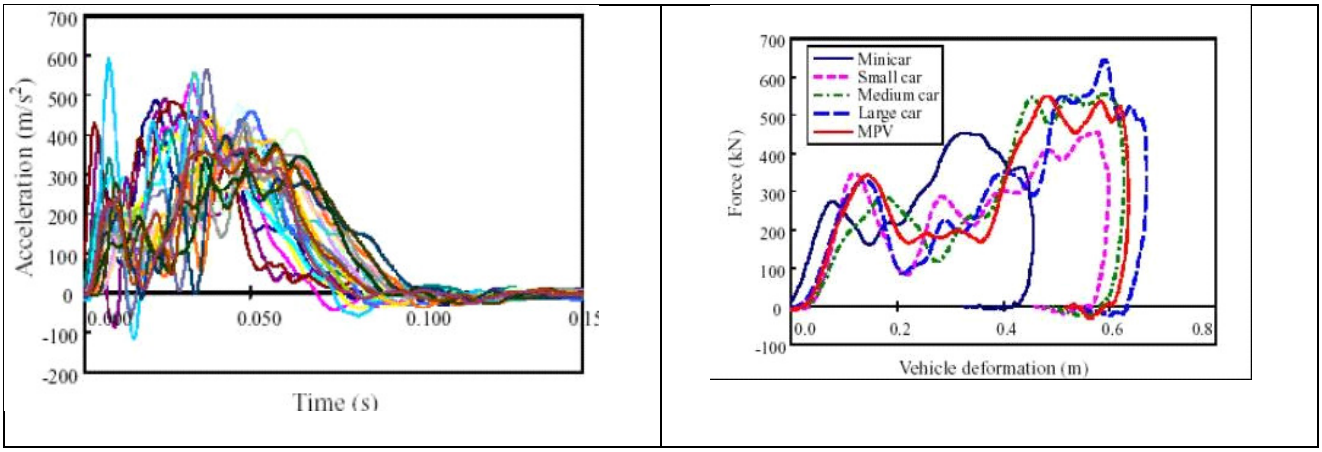


Fig. 2 - Acceleration Time Histories and Averaged Force Deformation Characteristics in Full Frontal Impact Tests at 56 km/h (35 mph) (Ono et al 2003)

From this it can be concluded that a force level of approximately 250-300 kN for the first 300- 400 mm is used by many manufacturers before the force level rises to a second plateau to limit the crush depth and avoid intrusion. That results in a first plateau deceleration between approximately 170-200 m/s$^2$ for a medium size car with a weight of 1500 kg. In order to limit compartment intrusion the force level has then to rise in proportion to the length of the remaining crush zone.

## METHODOLOGY

PULSE GENERATION: The objective of the study was to investigate the influence of crash pulse shape and restraint configuration on occupant protection at 56 and 80 km/h (35 and 50 mph) impact velocity using computer analysis combined with mechanical sled testing. The crash pulse was divided into two separate stages. The first stage constitutes the controlled buckling of the side members or crash box in the front structure and in the second stage the front structure is stiffening up in order to achieve the correct crush depth (Figure 3).

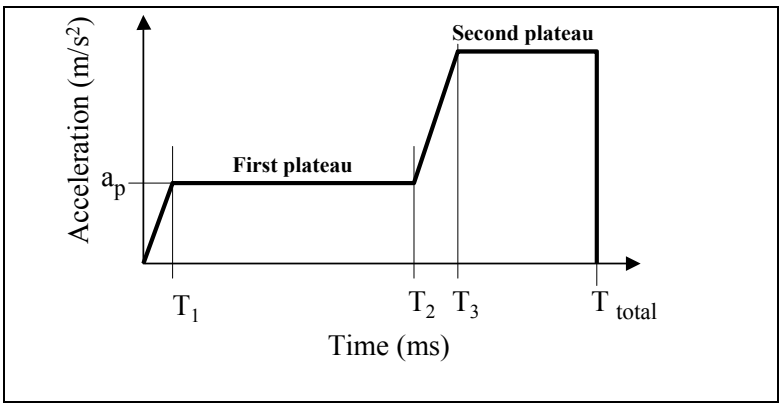


Fig. 3 - Description of the Form of the Hypothetical Crash Pulse

A theoretical crash pulse that contains the defined impact velocity (delta-v if no rebound) and the defined crush distance was derived. In the approach taken the crash pulse was approximated by a set of straight lines in a number of time intervals, $T_1$ to $T_n$ (Figure 3).

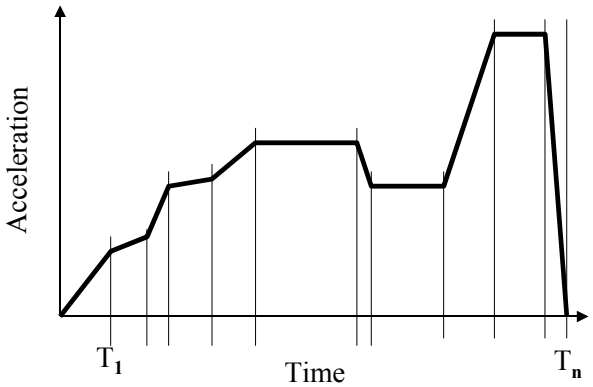


Fig. 4 - A Hypothetical Crash Pulse Approximated with Straight Lines

For each one of these time intervals, the deceleration described as a straight-line, the equation is easily defined. A linear acceleration profile with an increased acceleration b to c in a time interval $T_b$- $T_c$ (for the purpose of the demonstration it is the same as the fifth part of the hypothetical crash pulse in figure 4) is shown in figure 5.

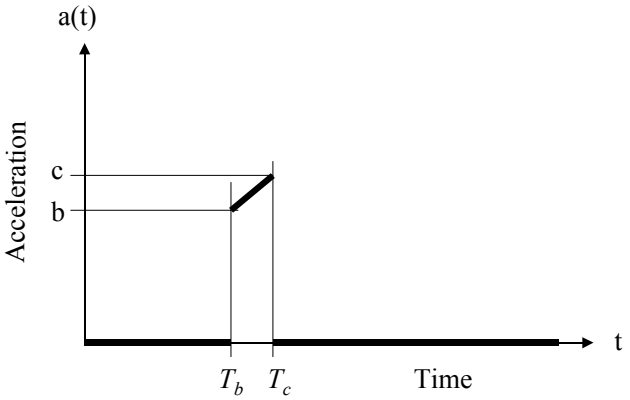


Fig. 5 - A Hypothetical Crash Pulse with a Linear Acceleration Profile and an Increased Acceleration in Time Interval $T_b$-$T_c$

The acceleration a(t), velocity v(t)and stopping distance s(t) (distance travelled) at any point in time for the crash pulse in figure 4 can be found using the following expressions.

$$a(t)=\begin{cases} 0 & t \le T_b \\ b+\dfrac{c-b}{T_c-T_b}(t-T_b) & T_b < t \le T_c \\ 0 & t > T_c \end{cases}$$

$$v(t)=\begin{cases} 0 & t \le T_b \\ b(t-T_b)+\dfrac{c-b}{T_c-T_b}\times\dfrac{(t-T_b)^2}{2} & T_b < t \le T_c \\ b(T_c-T_b)+\dfrac{c-b}{2}\times(T_c-T_b) & t > T_c \end{cases}$$

$$s(t)=\begin{cases} 0 & t \le T_b \\ \dfrac{b(t-T_b)^2}{2}+\dfrac{c-b}{T_c-T_b}\times\dfrac{(t-T_b)^3}{6} & T_b < t \le T_c \\ \dfrac{b(T_c-T_b)^2}{2}+\dfrac{c-b}{6}(T_c-T_b)^2+\left(\dfrac{c+b}{2}(T_c-T_b)\right)\times(t-T_c) & t > T_c \end{cases}$$

To achieve the velocity and stopping distance (distance travelled) for the full theoretical hypothetical crash pulse in figure 4 the expressions for all the time intervals can then be summed. In order to reduce the complexity of the problem some parameters defining the crash pulse in figure 4 had to be manually set. Typical crash pulses were studied and $T_1$ and the interval $T_2$-$T_3$ was set to be 0,005 seconds (Figure 3). If the impact velocity and the crush depth are known, see expression below, the system of equations can be solved using standard mathematical computer software.

$$\begin{cases} v(x,y)=v_0 \\ s(x,y)=s_0 \end{cases}$$

For compatibility reasons the first plateau of the crash pulse at 56 km/h (35 mph) was set to 200 m/s$^2$ with a stopping distance of 0,3 m and the stopping distance was varied in increments 0,4, 0,5 and 0,6 m. A deceleration level of 200 m/s$^2$ corresponds to a buckling force of 300 kN for a 1500 kg car in a full frontal impact. Since the acceleration $a_p$ in the first stage was set to 200 m/s$^2$ (see above) with a length of the buckling zone of 0,3 m the two unknown will be the magnitude of the plateau acceleration in the second stage and the total duration of the crash pulse

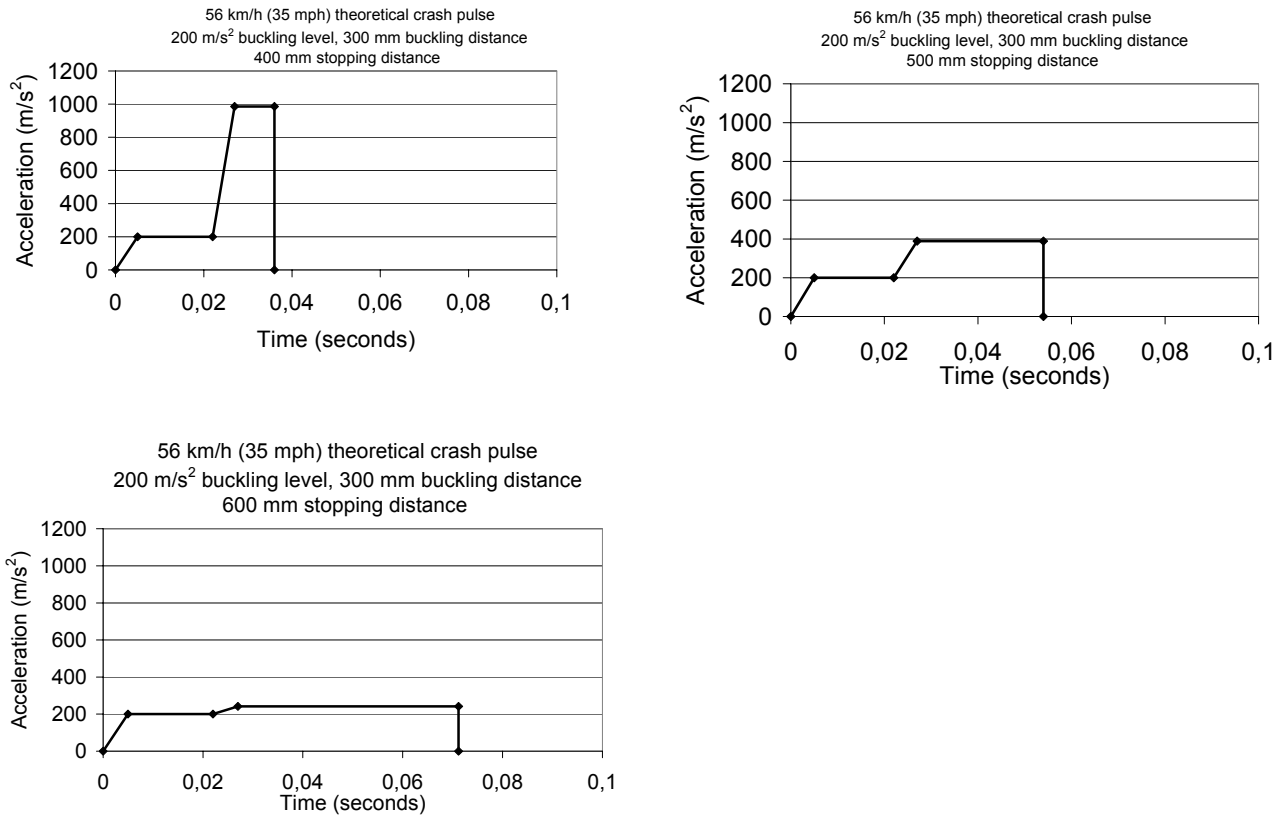


Fig. 6 - Synthetic Realistic Crash Pulses at 56 km/h (35 mph) Crash Velocity

For the 80 km/h (50 mph) study the buckling zone was set to 0,5 m, the stopping distance to 0,7 m and the acceleration plateau in the first buckling phase of the crash pulse was varied between 200 m/s$^2$ and 400 m/s$^2$ (Figure 7). It was, however, found that setting such a low acceleration level as 200 m/s$^2$ resulted in very high plateau acceleration in the second phase (975 m/s$^2$). Therefore the crash pulses used in the computer analysis had a first acceleration plateau of 300 and 400 m/s$^2$ respectively in the first stage.

By increasing the crush depth to 0,9 m the plateau acceleration in the first stage was reduced to 200 m/s$^2$ keeping buckling zone of 0,5 m and a realistic level of acceleration in the plateau acceleration in the second stage. The calculated crash pulse is shown in figure 7.

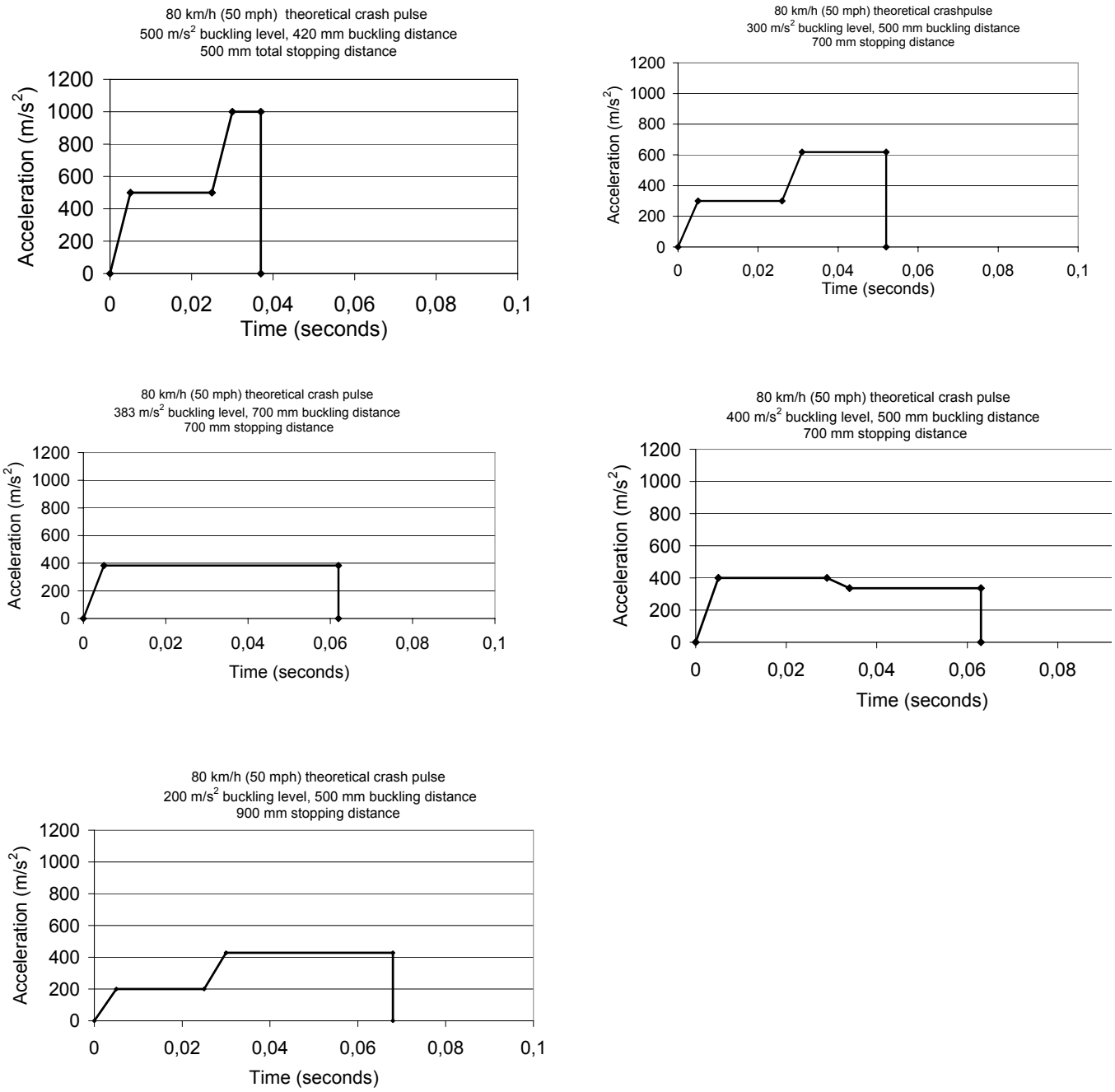


Fig. 7 – Synthetic Realistic Crash Pulses in 80 km/h (50 mph)

Some of the crash pulses derived with the theoretical crash pulse tool predicted extreme conditions such a second stage acceleration at 1000 m/s$^2$ etc and therefore these pulses considered to be unrealistic and ruled out for further analysis.

The synthetic crash pulses with 80 km/h (50 mph) impact velocity that were mimicked in the mechanical sled testing were pulses with approximately 0,7 and 0,9 m stopping distance (Figure 8). For the pulses with 0,7 m stopping distance as a target the initial accelerations due to buckling were 300, 350 m/s$^2$and the actual buckling distances were 0,6 and 0,76 m. For the pulse with 0,9 m stopping distance as a target the initial acceleration due to buckling was 180 m/s$^2$ and the actual buckling distance was 0,45 m.

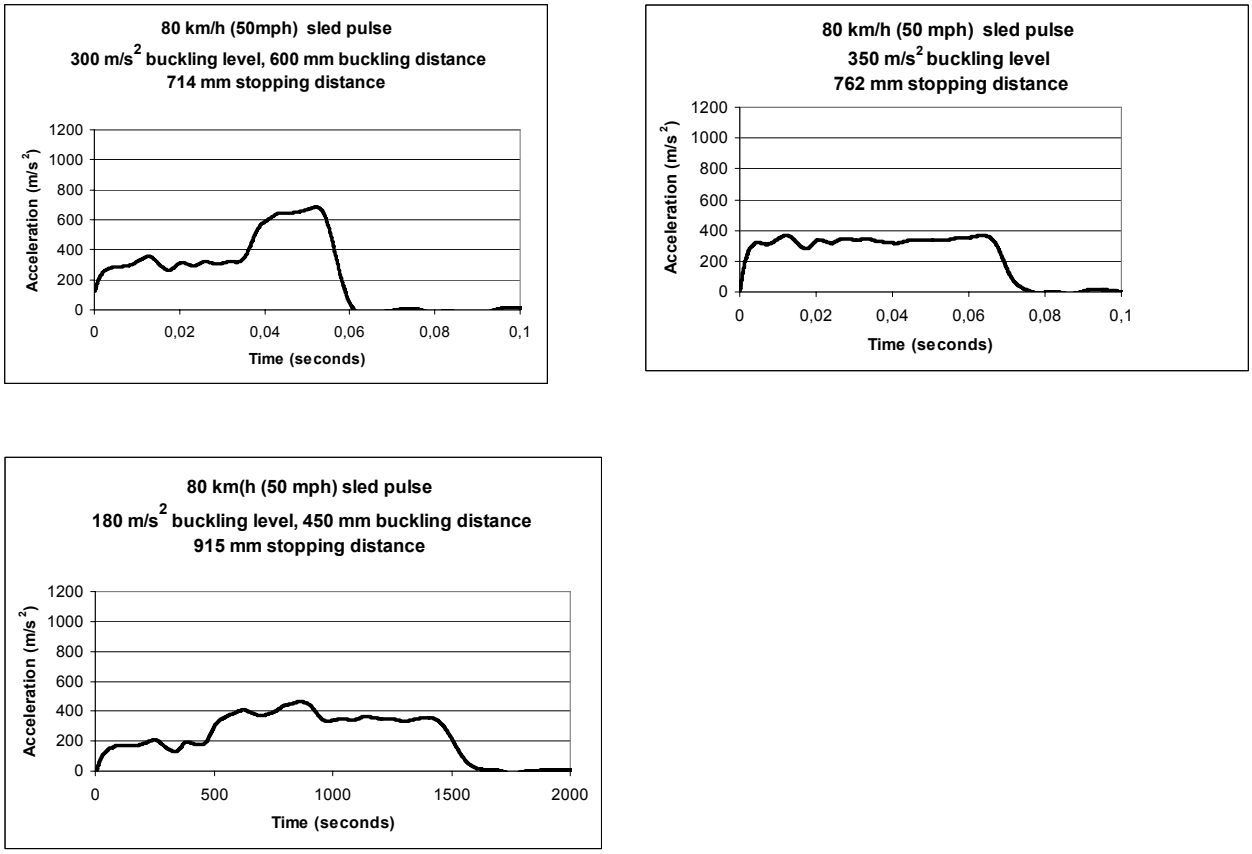


Fig. 8 - Mechanical Sled Test Crash Pulses

MATHEMATICAL OCCUPANT MODEL: To evaluate the occupant protection system for the derived crash pulses at high and very high impact velocity a mathematical model was developed. The geometry of the occupant compartment in the mathematical model was based on the geometry of the occupant compartment of a common mid size vehicle. The mathematical model was a multi-body dynamics model (MADYMO) that incorporated a 50%-ile HIII-dummy, a windscreen, a ceiling, a seat, a knee bolster, a belt system, an airbag, a steering wheel and an energy absorbing collapsible steering column (Figure 9).

The model was validated by means of results from mechanical sled tests. The predictions and results that were used for validation and evaluation were head acceleration, chest acceleration, chest deflection, pelvis acceleration, femur force, belt forces, steering column yield distance and airbag pressure.

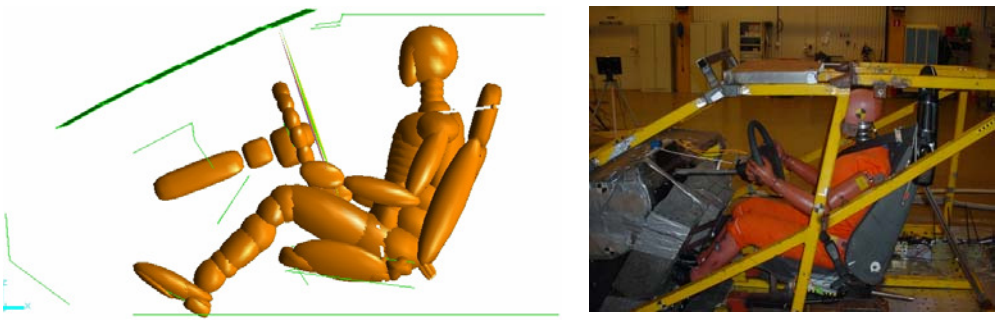

Fig. 9 - Mathematical Model and Mechanical Test Set Up

MECHANICAL SLED TEST SET UP: A mock-up of the driver environment of the same mid sized car as was used in the mathematical analysis was built and mounted on the sled. The mock-up included seat, steering wheel with column, air bag, knee restraints and seat belt (Figure 9). The driver restraint systems consisted of a three-point seat belt with an upper B-pillar mounted retractor and 60-litre air bag mounted in a

state of the art steering wheel. The seat belt had a dual load-limiter with a force level of 5 and 3kN. There was no limitation to the spool out due to the force limiting. In all tests the seat belt had a pre-tensioning device in the diagonal belt and in the outboard part of the lap belt. The load limiters were triggered prior to the air bag. The applied pre loading force was 2 kN. The quasi-static elongation of the seat belt was 10%. The lap belt anchorages were equipped with force limiters that were designed to let the anchorages slide forward at a force of 6 kN. The steering wheel column was oriented horizontally and the attachment between the steering wheel and the column was angled to achieve a realistic steering wheel angle. The column had a special collapse mechanism to allow for a stroke of 250 mm at predetermined force levels. The deformable element consisted of aluminium honeycomb of which the area could be varied to control the collapse force. A reinforced standard seat was used in all tests. A steel plate was built in under the seat cushion in order to avoid excessive seat cushion deformation and seat chassis deformation during testing. The seat was positioned in the mid position with a 26° seat back angle. The knee bolsters consisted of energy absorbing polypropylene. A 50%-ile Hybrid III dummy as defined in US Code of Federal regulation 49 Part 572 was used in the tests. The dummy was conventionally instrumented to measure head, chest and pelvic linear acceleration in three orthogonal directions. Chest deflection was measured at mid sternum. Femur forces were measured with axial load cells in each leg.

## RESULTS

MATHEMATICAL OCCUPANT MODELLING RESULTS AT IMPACT VELOCITY 56 KM/H (35 MPH): The computer runs at 56 km/h (35 mph) were carried out with crash pulses as shown in figure 7. The first stage of the crash pulse remains the same but the second plateau acceleration in the second stage is different as is the duration to achieve different crush depth namely 0,4, 0,5 and 0,6 m respectively. The results are shown in relation to the "new" FMVSS 208 injury criteria threshold levels (Appendix A). The vertical axes are the results divided by the FMVSS 208 injury criteria threshold levels.

With a 200 m/s$^2$ buckling level acceleration and a stopping distance of 0,5 m the dummy responses were all below the FMVSS 208 injury criteria levels (Figure 10). However with a stopping distance as short as 0,4 m chest acceleration and chest deformation values were above the FMVSS 208 injury criteria level. Increasing the stopping distance to 0,6 m reduces the dummy responses to well below FMVSS 208 injury criteria levels.

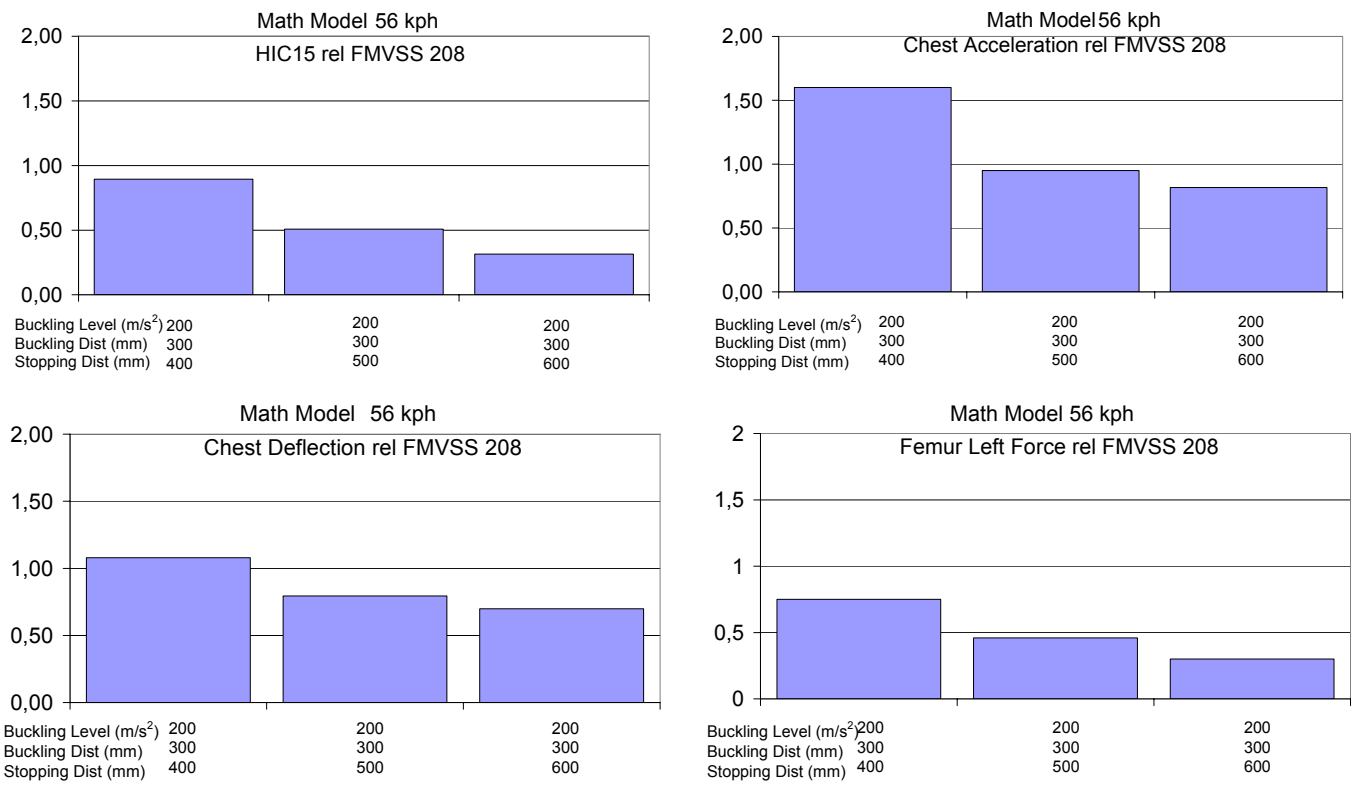


Fig. 10 - $HIC_{15}$, Chest Acceleration, Chest Deformation and Femur Left Force Predicted by the Model

MATHEMATICAL OCCUPANT MODELLING RESULTS AND IMPACT VELOCITY 80 KM/H (50 MPH):The computer runs at 80 km/h (50 mph) were carried out with crash pulses as shown in figure 9. In the results from the simulations at 80 km/h (50 mph) impact velocity $HIC_{15}$ was significantly below the FMVSS 208 injury criteria levels for all buckling levels and stopping distances (Figure 11). For chest acceleration the crash pulse with 0,5 m stopping distance and the pulse with 300 $m/s^2$ buckling level and 0,7 m stopping distance were above the FMVSS 208 injury criteria levels. For chest deflection all evaluated configurations at 80 km/h (50 mph) had chest deflection significantly less than the FMVSS 208 injury criteria levels (Figure 11).

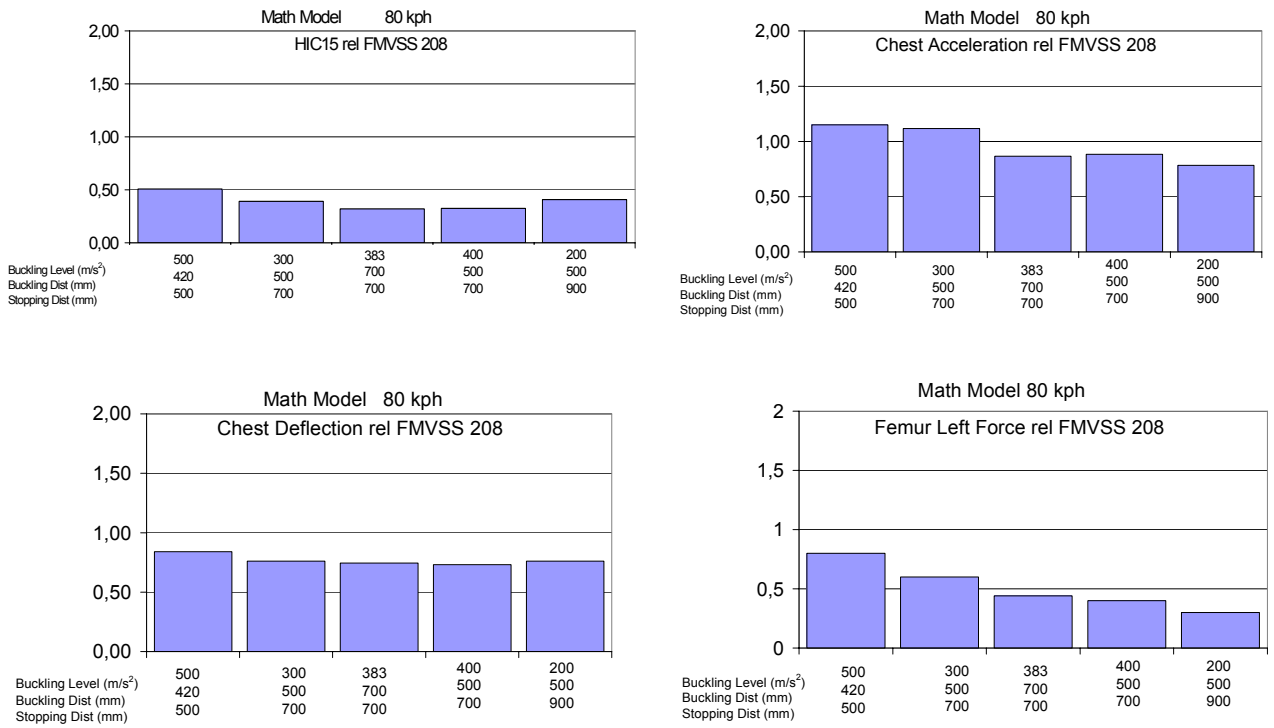


Fig. 11 - HIC15, Chest Acceleration, Chest Deflection and Femur Left Force in 80 km/h Predicted by the Model

MECHANICAL SLED TEST RESULTS: Due to the limitations of the mechanical test equipment some of the simulation pulses defined for 80 km/h (50 mph) was not simulated in the mechanical tests. The pulses that were evaluated mechanically were pulses with 0,714, 0,762 and 0,915 m stopping distance (Figure 8). The lowest dummy values were obtained in the test with a crash pulse with 0,915 m stopping distance. All values were below the FMVSS 208 injury criteria levels. For the tests with approximately 0,7 m stopping distance lowest dummy values were obtained in the test with a rectangular pulse shape. All dummy values but $HIC_{15}$ were below the FMVSS 208 injury criteria levels. In the test with the step shaped crash pulse with a buckling acceleration level of 300 $m/s^2$ $HIC_{15}$ and chest acceleration were above the FMVSS 208 injury criteria levels while chest deflection and femur force were below. All NIJ values were below the FMVSS 208 injury criteria levels.

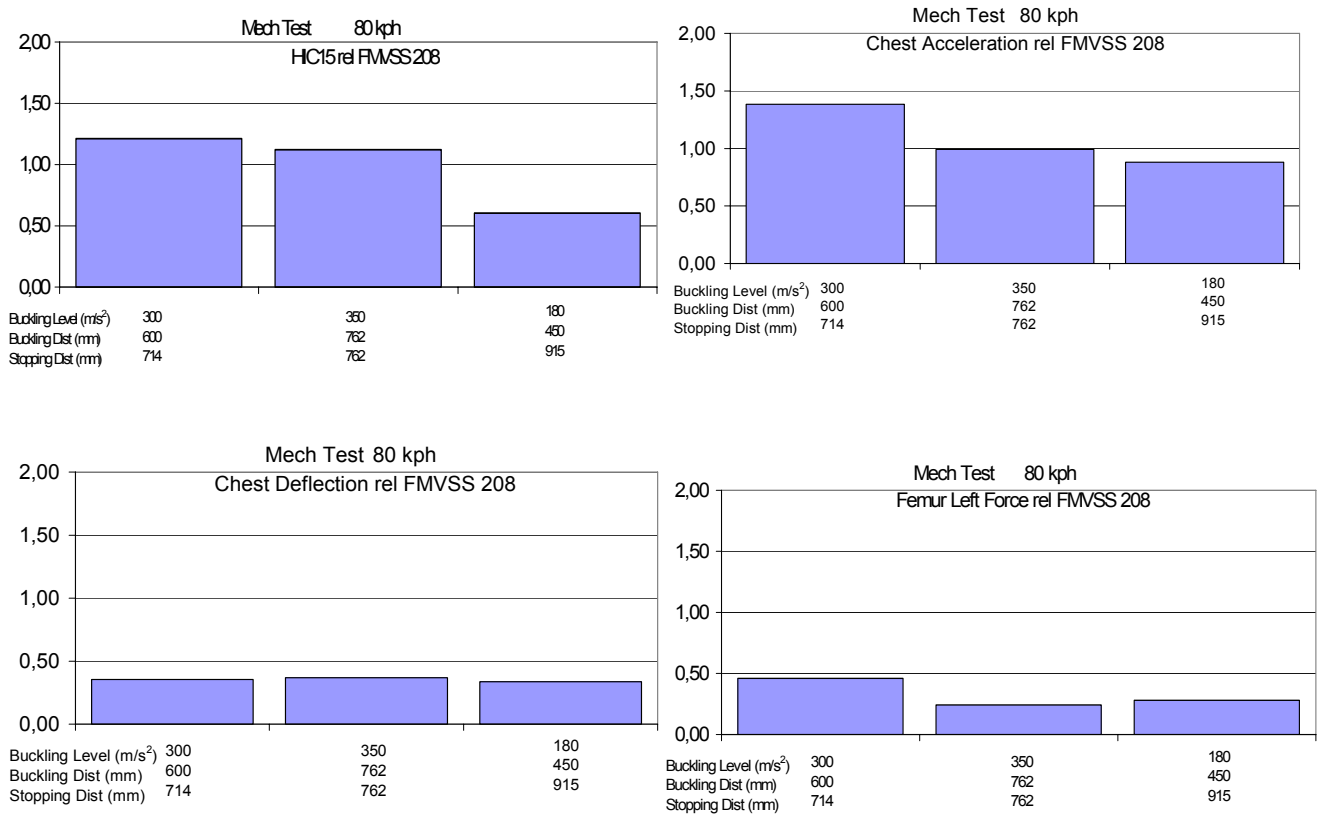


Fig. 12 - $HIC_{15}$ , Chest Acceleration, Chest Deflection and Femur Left Force in Mechanical Sled Tests

**DISCUSSION**

Due to the fact that the objective of the study was to emphasize occupant protection in 80 km/h (50 mph) frontal impacts therefore the results in 56 km/h (35 mph) were considered to be sufficient even though the restraint system can be tuned for additional reductions in dummy response at 56 km/h (35 mph). In the study a conventional driver side air bag with a conventional in crash triggering was used. A larger airbag in combination with pre-crash triggering can add additional improvements to the occupant protection system. However, in the proposed protection system at 80 km/h the ride down distance of the occupant was greater than what is available in most vehicles on the market today.

The concept for high speed protection of the occupant is based on that the occupant early on in the crash is connected to the car resulting in greatest possible ride down distance of the occupant. In addition the ride down distance of the occupant is enlarged through advanced force limiting in the seat belt as well as in the energy absorbing steering column. The air bag is restraining the head and the neck and is distributing and limiting the forces applied on the chest. The drawback with such a system is that once the occupant move forward, not being connected to the car, the performance of the protection system has to rely on the endurance of the load limiting devices. If there is a bottoming out taking place the occupant's velocity relative the car will result in high forces on the occupant. Another aspect is the appropriateness of the tuning the restraint system to a specific occupant mass. In this study the collapse force of the steering column has been a set to absorb as much as possible of the occupant's energy without applying too high restraining forces on the occupant. For example with an occupant of less weight the collapse mechanism may not have triggered and consequently the systems potentials for crash protection would not have been utilized. A restraint system designed to protect the occupant at very high impact velocities can be too stiff for the occupant at low impact velocities. In addition it can be too stiff for the elderly population with lower tolerance limits. However, with proper tuning of an adaptive restraint system (belt and bag) good protection can be achieved in both high and low impact velocities.

In modern cars with sloping A-pillars the forward motion of the occupant is sometimes restricted and the head can impact the windshield header during a ride down in a frontal crash. This problem was not addressed in the study and needs further attention.

In the mechanical sled tests the chest deflection was always low and it can be partly due to the relatively high position of the diagonal band that compressed the chest in the upper portion and partly due to the distributed loading of the air bag. Since the potentiometer measuring chest deflection is located at mid sternum it may not correctly pick up the deformation in the upper part of the dummy thorax.

The mathematical model was validated by means of sled tests in both 56 km/h and 80 km/h prior to carrying out the pulse analysis. Generally good agreement between predictions from the model and results from the mechanical sled tests was obtained. However in the mathematical model the neck forces and moments were not evaluated in the pulse analysis due to the poor agreement between the forces and moments in the validation of the mathematical model.

The most promising results from the mathematical occupant analysis were reproduced in the mechanical sled tests. However, due to poor reproduction of the theoretical crash pulses in the mechanical sled tests the results from the sled tests cannot be compared to the results from the computer analysis but they should rather be regarded as indications that the trends in the computer analysis was the same in the sled testing.

The results prove that a compatible crash pulse can yield low dummy response even with a rather short stopping distance and that occupant protection systems for occupant in full frontal impacts at 80 km/h (50 mph) can be designed even for a car in the midsize segment. In the study the basic external dimensions and properties of a midsize car were used. It would, however, require considerable redesigning of the structural members in the compartment in order to achieve the necessary stiffness characteristics and avoid intrusion or occupant compartment collapse. In the coming next step of work finite element analysis of a car body will be made in order to assess the implication on the structural elements and the weight and the cost of the car body.

The hypothetical crash pulses chosen for the analysis in this project were a result of a specific reasoning as outlined above. More research into the compatibility aspects and the aspects of self-protection and the influence on crash pulse must be performed. Can intrusion be avoided at impact velocities of 80 km/h (50 mph) with realistic penalties in cost and weight? The concept of pre-crash sensing should be introduced and its influence should also be investigated.

**CONCLUSIONS**

- A theoretical tool has been developed to produce crash pulses with a specific crush depth, delta-v and pulse shape
- Computer analysis shows that effective full frontal impact protection at 56 km/h (35 mph) can be achieved with a crush depth of 0,7 m and a crash pulse that has a first low acceleration stage for compatibility reasons
- Computer occupant analysis and mechanical sled testing shows that effective full frontal impact protection at 80 km/h (50 mph) can be achieved with an advanced restraint system.

    1. with a crush depth of 0,7 m with a square wave type of crash pulse with an advanced restraint system
    2. with a crush depth of 0,76 m with a square wave type of crash pulse with an advanced restraint system
    3. with a crush depth of 0,915 m with a crash pulse that has a first low acceleration stage for compatibility reasons
    4. with a crush depth of 0,76 m with a crash pulse that has a first low acceleration stage for compatibility reasons the

- The need for a more efficient crash pulse at higher crash speeds point towards the necessity for adaptivity in the frontal structures.

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## APPENDIX A: FMVSS 208 INJURY CRITERIA LEVELS

| | |
|---|---|
| $HIC_{15}$ | 700 |
| NIJ | 1 |
| Chest Acceleration | 60 g |
| Chest Deflection | 63 mm |
| Femur Force | 10000 N |