PARAMETER OPTIMIZATION FOR VEHICLE TO VEHICLE CRASH COMPATIBILITY USING FINITE ELEMENT METHODS

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ABSTRACT

Today, many auto makers and national researchers are interested in the compatibility of standard vehicle crash testing to that of real accident conditions. Current standard tests assume like vehicle to vehicle crash events to address the injury or fatality risk to vehicle occupants. Researchers that investigate vehicle to vehicle crash tests struggle to understand the relationship between aggressivity and injury measurements compared to those of accident investigation data like FARS and NASS GES. Such tests, however, can be simulated with finite element methods to re-enact real accident conditions to predict detailed vehicle measurements in order to make study structural improvements. In order to understand the nature of compatibility and to facilitate structural improvements, robust parametric studies and structural optimization methodologies can be employed to manage the complex, coupled design parameters and geometric changes.

In the past, assembling, executing, managing and interpreting the results has prevented this level of parametric study. StudyWizard, developed by Altair Engineering, is a software technology specifically designed to automate and extract meaningful design information from parametric analytical studies. Using StudyWizard, this paper will demonstrate a strong relationship between aggressivity and injury revealed by FARS data set for an oblique offset vehicle to vehicle crash. For this paper, a full-sized sedan compatibility study is performed for impacts with a light truck and van. The aggressivity characteristics; vehicle mass, stiffness and stackup will be discussed. To conclude, compatibility improvements will be examined.

INTRODUCTION

Even though incompatibility has been known since the 1960s, research activities have not been systematic until recently. However, an understanding and improvement of crash performance of vehicle in frontal and side impact was required before addressing compatibility, because they believed this could reduce casualties immediately.

This incompatibility has often been identified in real traffic accidents. For example, the crash between a passenger car and truck may result in unbalanced deformation and one of the vehicles may suffer great injury.

In studying compatibility, there are many aspects, such as crash between cars, car to pedestrian, car to light truck or van. The aim of this compatibility study is to identify how vehicle safety may be improved by structural changes that are designed to interact better during the crash and by restraint systems that are also designed to act properly in the second collision for each. For ideas on pedestrian protection (considering compatibility between car and pedestrian) evaluation methods are undergoing. EEVC and IHRA activities are making progress in this compatibility.

During the past decade, most automakers have done really well for crashworthiness design of the vehicle structure and restraint system to reduce the occupant injury in specific collision situations, in comparison with other cars or with standard car in FMVSS and other safety standards.

Such tests, however, only back up indications that are commonly calculated for broad categories of collisions. By the way, incompatibility in multi-vehicle crashes have been a concern due to promotion activities that have automakers designing cars with better crashworthiness in rigid barrier or stationary deformable barrier crash tests. The reason why better crashworthiness may result in incompatibility is the possibility that the stronger structure makes the vehicle more aggressive in collisions with other vehicles.

At the same time, the mass of vehicle is also observed as an important factor, driven by the laws of conservation of energy and momentum of two objects. The heavier object has small velocity change and the lighter object high with proportional mass ratio.

With respect to the affect on aggressivity, the stiffness of the vehicle is likely to have small influence compared with the mass of the vehicle.

To study and investigate the aggressivity, generic vehicle data were originally developed in 1981 for use in CRASH3 computer program by CALSPAN under contract to NHTSA. This generic data derived from the statistical analysis of a number of vehicles, not specifying any vehicle, could be useful for statistical studies, where vehicle parameter need to be considered as we simulate and analyze traffic accidents and may be a good preview for a comprehensive understanding of crashworthiness and incompatibility in real accidents.

Reference [1] lists the updated vehicle generic data including mass, stiffness and even suspension information. Some of this data is repeated in Table 1 through 3 below for the reader to glance over real incompatibility information.

Table 1. Generic Data for Passenger Cars

			Class1	Class2	Class3	Class4	Class5
Mass	Total Mass		5.398	6.391	7.607	9.528	10.026
	Roll		2200	2300	3085	3822	3834
	Pitch		12383	15845	20001	24960	23985
	Yaw		13489	17286	23989	29294	29279
Stiffness	Front	A	180.25	184.69	206.64	215.40	288.73
		B	72.11	65.38	69.97	66.70	113.45
		K	95.27	84.51	93.40	87.73	147.56
	Rear	A	172.50	162.33	189.62	186.00	292.40
		B	54.40	49.44	51.77	47.00	138.00
		K	80.82	72.57	80.35	70.16	207.47
	Side	A	88.25	100.00	95.75	137.00	137.00
		B	59.75	66.20	77.75	95.00	95.00
		K	88.92	84.06	97.45	119.12	119.12
Susp	Front	R	96.14	102.39	111.15	95.25	96.00
		D	9.90	9.23	8.69	7.02	6.86
	Rear	R	100.59	97.35	114.39	132.50	173.33
		D	7.99	7.01	6.93	6.96	7.64

Table 2. Generic Data for Pickups

			Class1	Class2
Mass	Total Mass		7.324	11.465
	Roll		2427	5430
	Pitch		17835	40008
	Yaw		19979	43081
Stiffness	Front	A	266.08	219.60
		B	108.92	68.40
		K	140.59	89.10
	Rear	A	258.33	290.67
		B	108.83	123.00
		K	151.77	190.30
	Side	A	103.00	78.00
		B	92.00	40.00
		K	110.55	48.65
Susp	Front	R	149.85	184.17
		D	9.6	8.85
	Rear	R	128.00	157.33
		D	7.86	6.84

Table 3. Generic Data for Multi-purpose and Vans

			Multi-Purpose		Van	
			Class1	Class2	Class1	Class2
Mass	Total Mass		9.069	12.536	8.691	13.050
	Roll		3134	6395	5656	9134
	Pitch		19588	38674	24493	47204
	Yaw		21998	40949	26474	51035
Stiffness	Front	A	266.08	219.60	309.00	358.75
		B	108.92	68.40	135.00	154.75
		K	140.59	89.10	170.36	189.74
	Rear	A	258.33	290.67	281.00	312.00
		B	108.83	123.00	118.02	141.73
		K	161.77	190.30	182.02	221.37
	Side	A	103.00	78.00	96.00	137.00
		B	92.00	40.00	78.00	95.00
		K	110.55	48.65	97.00	119.12
Susp	Front	R	156.33	180.08	164.50	180.73
		D	8.03	7.51	9.41	7.36
	Rear	R	134.22	145.48	146.12	158.00
		D	7.33	6.92	7.62	6.57

$\text{mass} = \text{lb} - \text{sec}^2/\text{in}, \quad \text{inertia} = \text{lb} - \text{sec}^2 - \text{in}$

$\text{Stiffness } A = \text{lb}/\text{in}, B = \text{lb}/\text{in}^2, K = \text{lb}/\text{in}^2$

$R = \text{ride rate} = \text{ln}/\text{in}, \quad D = \text{damp rate} = \text{lb} - \text{sec}/\text{in}$

From Tables 1 through 3, we can directly imagine incompatibility in mass and stiffness along the vehicle category. Specifically, an incompatibility in crash type can that can be inferred is that there is more fatality risk in side impacted car when the stiffness of front and side are discussed.

Unlike the crashworthiness, the concept of aggressivity is about fatality or injury risk to occupants of other vehicle, applied with measurement as either injuries to occupants of vehicles involved in collisions, of specific vehicle as standards, or the average over all, or representative selection. The terms of compatibility appears in the practical problems, as a vehicle shows a different injury response to occupants of any other vehicles, even that have the same crashworthiness.

The researcher who is interested in accident reconstruction would investigate the reason why injuries are different is due to different stiffness, mass and mismatch of main parts for crash performance. At that time, the generic data would be often used for it. Specifically, this mismatch of main parts require us to review the vehicle structure along the impact types, impact position and angle, resulting in different interactions during the collisions.

Compatibility

Reiterating the concept of compatibility, it is more ideal to use a definition that considers the fatality or injury risk to occupants of the other vehicle with considering the subject vehicle. This is more complicated because both the crashworthiness of the other vehicle, and the aggressivity of the other vehicle influence the injury and fatality risks.

In order to understand the nature of compatibility, many automakers and research organization have conducted vehicle-to-vehicle crash tests and tried to set up the relationship between the injury measures and the aggressivity metric (AM) developed by NHTSA aggressivity research program.

$$\text{Aggressivity} = \frac{\text{Driver Fatalities in collision Partner}}{\text{Number of Crashes of subject Vehicle}}$$

This aggressivity metric is well used to indicate how much any vehicle categories are aggressive in real traffic accident, by their relative aggressivity using FAR and GES, as the overall fleet aggressivity ranking. This is a very typical and easier way to understand vehicle aggressivity by vehicle category from reference [2] repeated in Table 4.

Table 4. Vehicle Aggressivity by vehicle category (FARS/GES 1991-94)

Vehicle Category	Driver fatalities in the struck vehicle per 1000 police-reported crashes
Full size van	2.47
Full size pickup	2.31
Sports-Utility vehicle	1.91
Small Pickups	1.53
Minivans	1.45
Large Cars	1.15
Midsize Cars	0.70
Compact Cars	0.58
Subcompact Cars	0.45

In the same reference, crash compatibility also examined in vehicle-to-vehicle crashes, frontal and side impact types, shows a significant difference in the number of fatalities with respect to impact direction, A side impacted car has more than 3 times the fatalities compared to frontal impacted, and 7 times more at it worst. The results of this reference is also repeated in Table 5.

Table 5. Ratio of Fatally-Injured Drivers in LTV-to-Car Frontal and Side Impacts

Vehicle Category	Frontal impacts	Side impacts
Full size van	1 : 6.0	1 : 23
Full size pickup	1 : 5.3	1 : 17
Utility vehicle	1 : 4.1	1 : 20
Minivan	1 : 3.3	1 : 16
Small pickup	1 : 1.6	1 : 11

These crash statistics also reveal incompatibility between car and LTVs (Light Truck and Van including Sport Utility Vehicles) and impact modes.

A different countermeasure for each crash types is apparent. The engineering is a challenge in that the increase of ride-down efficiency has to be balanced with the restraint systems for frontal impact. The reduction of the effective velocity change is accounted as the exchange of momentum between a car structure accelerated by struck vehicle and occupants from impact to separation for side impact.

For instance of the side impact, the possibility of changing initial impact velocity condition or interactions between the car structure and occupants led by slight change of impact angle or tolerance structural stiffness is so high that different occupant responses could often be seen and complicate the understanding of structural effects, when compared with frontal impacts.

While crashworthiness is focused on structural effects and restraints, the accident reconstructionists are used to saying the basic principals to reconstruct the crash accidents are as follows:

- Impact is to the exchange of momentum.

During the collision a linear impulse is produces between the vehicles is

$$\mathrm{Impulse} = \sqrt{2(E_1 + E_2)(M_1 M_2 / (M_1 + M_2)}$$

where E_1, E_2 are the energies absorbed by vehicle 1,2
M_1, M_2 are the mass of vehicle 1,2

- The vehicles are objects strongly influenced by plasticity, but some elasticity works in crash. The restitution is often considered when we calculate the velocity change at relatively low speed impacts showing the small and elastic deformation before the plastic deformation.

- Impact is phenomenon about converting a part of kinetic energy in to plastic deformation energy, and dissipating it. The kinetic energy just before the crash is equal to the plastic deformation energy plus the kinetic energy after crash.
  So, plastic deformation energy is calculated with the following equation involving the restitution coefficient.

$$EA = \frac{M_1 M_2}{2(M_1 + M_2)}(1-\eta^2)(V_{i1} - V_{i2})^2$$

where EA is the energy absorbed by plastic deformation
V_{i1}, V_{i2} are the initial velocity of vehicle 1,2
η is the coefficient of restitution

- The forces engaged in crash are friction force plus impact force. During the crash the total resultant force vector is a function of the reaction force and friction force between vehicles and be calculated with equation as follows:

$$\mathrm{Resultant\ Force} = Fv\sqrt{1+\mu^2}$$

where Fv = Perpendicular Reaction Force
μ = Friction Coefficient

- Non-central impact is not only exchange of momentum but also converting linear momentum in to angular momentum.
  For instance, the non-central and perpendicular impact case is illustrated graphically in figure 1, and the velocity of each vehicle is calculated as follows:

$$V_{1f,Ydir} = \frac{M_1(L_2^{\,2}+\rho_2^{\,2})-\eta I_2}{I_2+M_1(L_2^{\,2}+\rho_2^{\,2})}V_{1i}$$

$$V_{2f,Ydir} = \frac{M_1\rho_2^{\,2}(1+\eta)}{I_2+M_1(L_2^{\,2}+\rho_2^{\,2})}V_{1i}$$

$$\alpha_{2f,center} = \frac{M_1(1+\eta)}{I_2+M_1(L_2^{\,2}+\rho_2^{\,2})}L_2V_{1i}$$

$$\text{Energy Ratio} = \text{Angular Kinetic Energy} / \text{Total Kinetic Energy}$$

$$= 1 \Big/ 1+(\rho_2/L_2)^2$$

where $I_{1,2}$ = Mass of Inertia

$L_{1,2}$ = Distance between CG to impact force position

$\rho_{1,2}$ = radius of gyration

$\alpha_{1,2}$ = Angular Velocity

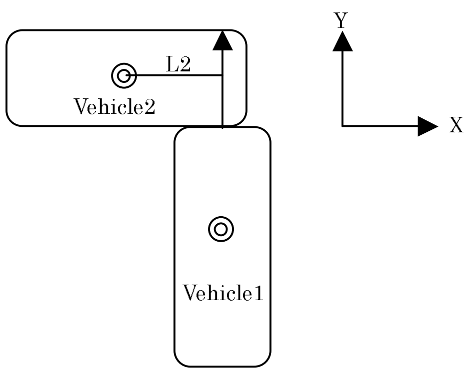


Fig 1. Non-central and perpendicular impact

Consequently, the topic for crash accidents or compatibility study do not involve just one parameter, rather the permutated effects with several parameters. The priority of these parameters may be changed case by case, and have to be studied for the basic incompatibility.

Another consideration regards the injury mechanism to represent the load path to transfer impact forces to the occupant, and what is the major reason to result in fatal injury during the crash.

Comparing side impact and frontal impact, occupants mainly restrained by seat belt and airbag supplemented in frontal impacts, are injured at the second impact time as usual, but in side impacts occupants did not have specific restraint system. Recent side airbag or inflatable tubular system meant to reduce injuries are suffering with the bullet car hitting them directly.

Many aspects of incompatibility are not well understood, but studies focus on engineering spot issues until a comprehensive awareness of compatibility is understood completely, and the methodology for it accepted commonly.

Finite Element vehicle models and optimization

In crash analysis and simulation, the finite element method is regarded as the most powerful tool because of the ability to perform parametric studies requiring iterations in the design of vehicle structure.

For crash simulation, explicit FE method such as LS-DYNA, PAM-CRASH and RADIOSS, ABAQUS-explicit are well used and are common for virtual crash tests.

FHWA/NHTSA National Crash Analysis Center at George Washington have developed various FE vehicle models which are available to download for all researchers of the world who are likely to do crash simulation.

In this study, the 3 FE models extracted from FHWA/NHTSA National Crash Analysis Center-Website are listed on table 6.
(http://www.ncac.gwu.edu/archives/model/index.html)

Table 6. FE models used in LTV-to-Car compatibility study, extracted from FHWA/NHTSA National Crash Analysis Center-Website

Model Name	Model Size
Ford Taurus	Modified Model 28,400 elements
Chevrolet C2500 Pickup	Detailed model 54,800 elements
Dodge Caravan	Detailed model 329,300 elements

Design of experimental and Optimization methods are generally known as the best tools used for design of vehicle structure, because it can drive an optimal conclusion which can be evidenced in a systematic and mathematical sense.

But there are major difficulties as well that have been identified in crash simulation case. The literature mentions that crash simulation models must be accurate to prevent the numerical error affecting the response. Things such as contact routines, rigid linkage, element types/sizes, and sampling frequency may cause degradation of spectral response, more than the effects of parameter changes, and requiring extensive

computational resources and time [4].
Another difficult thing that may be said is that "the nature of crash phenomenon" is chaotic as it involves the interaction of hundreds of parts which come into contact causing force oscillations within a very short time. Bifurcation also happens and is known as the qualitative change appearing in the response with respect to change of parameters [5].

It is impossible to assess the crash structure system in view of this bifurcation because bifurcations mean the change of topology in the character of the solution. Contemporary crash simulation techniques are likely to conclude that the optimization of vehicle crashworthiness is possible, because it is always the study about the representative crash case like FMVSS crash tests and simply selecting the best settings of the varied parameters, what designer wants to change. Those who catalogue crash analysis and simulation as chaos, however, may say "must be careful in optimization study". One must take into consideration the sensitivity of initial conditions or uncertainty, such as change of initial velocity, mass, impact angle and interactions between car structure parts, and qualitative leverage from structural improvements or for different crash types.

Ignoring the uncertainty and excluding the possible effects coming from the numerical errors and change of topology, etc, the optimization technique may attempt to solve the crash problems in a relatively easy way. And, it is believed that the optimization theory using sequential approximation method can solve the sensible crash problems if design parameters could be defined and designer wants to get the optimum result within its variations [6][7].

Response Surface optimization for crash problems

The engineering of structural design is a challenge to determine the best one through an iterative process where a series of structural changes regarded as reasonable trials show off the different response.
Using optimization and FE simulation tools, engineers might be allowed to change the design parameters and to get the efficient search for right combination of design parameters for a certain design.

The FE codes for analyzing the nonlinear dynamic response of structures have a time discretization which is accomplished through the central difference operator, that expresses velocities at time t+ Δ t/2 and displacements at t+Δt explicitly in terms of present accelerations, velocities, displacements as follows [8][9][10];

$$
\begin{aligned}
&{}^{t}\ddot{\mathbf{d}} = \mathbf{M}^{-1}\left({}^{t}\mathbf{f}\right) \\
&{}^{t+\Delta t/2}\dot{\mathbf{d}} = {}^{t-\Delta t/2}\dot{\mathbf{d}} + {}^{t}\ddot{\mathbf{d}}\Delta t \qquad \ldots\ldots\ldots\ldots\ldots(1) \\
&{}^{t+\Delta t}\mathbf{d} = {}^{t}\mathbf{d} + {}^{t+\Delta t/2}\dot{\mathbf{d}}\Delta t
\end{aligned}
$$

The advantage of this method is only the mass matrix has to be simply inversed.
However this explicit method is conditionally stable with Δ t as the disadvantage to be considered carefully.
From the nonlinear crash experience that looked at plastic folding, thin shell elements are not accommodated by higher order due to plastic hinge line. The fold may be located in the any nodal points of finite elements model where the more dense nodal points are doing well enough to represent the crippling.

There are two distinct optimization schemes. One is the local approximation using most important gradient, design sensitivity analysis. The other is global approximation method which uses higher order analytical expressions to approximate the dependence between structural responses and design parameters with few analyses.
The local approximation scheme is based on the premise that small changes of design parameters occur in each optimization step, and in conjunction with nonlinear dynamic FE code, uses an explicit time integration scheme, where differentiation of the state equations (1) with respect to the design variables p_j yields

$$
\begin{aligned}
&\frac{d^{t}\ddot{d}}{dp_j} = M^{-1}\left(\frac{\partial^{t} f}{\partial p_j} + \frac{\partial^{t} f}{\partial^{t} d}\frac{d^{t} d}{dp_j} + \frac{\partial^{t} f}{\partial^{t}\dot{d}}\frac{d^{t}\dot{d}}{dp_j} - \frac{d^{t} M}{dp_j}\ddot{d}\right) \\
&\frac{d^{t+\Delta t/2}\dot{d}}{dp_j} = \frac{d^{t-\Delta t/2}\dot{d}}{dp_j} + \frac{d^{t}\ddot{d}}{dp_j}\Delta t \qquad \ldots\ldots\ldots\ldots(2) \\
&\frac{d^{t+\Delta t} d}{dp_j} = \frac{d^{t} d}{dp_j} + \frac{d^{t+\Delta t/2}\dot{d}}{dp_j}\Delta t
\end{aligned}
$$

This linear gradient relationship for each optimization step is likely not to be maintained under the significant change of time step during the calculation, even change of material stiffness to keep the constant time step.
In this case, there are notable non-linearities and warring convergence, simply when we think about the significant change of time step and several different element size models, Consequently, the global approximation seems to be a better solution and a more general way of getting the optimized solution with respect to more wide design parameters.

Let $\psi_i(\mathbf{b})$ be the response of interest. Then, a polynomial $\hat{\psi}_i(\mathbf{b})$ of the degree n can be introduced such that

$$\psi_i(b) \approx \hat{\psi}_i(b) = a_{i0} + \sum_{p=1}^{m} a_{ip} b_p + \sum_{p=1}^{m} \sum_{q=1}^{m} a_{ipq} b_p b_q + \sum_{p=1}^{m} \sum_{q=1}^{m} \sum_{r=1}^{m} a_{ipqr} b_p b_q b_r + \ldots \quad \text{..........(3)}$$

where the quantities $a_{i0}, a_{ip}, a_{ipq}, a_{ipqr}$ are constants that are determined from the results of the system analyses by solving a linear system of equations. The number of analyses that are necessary depends on the polynomial degree *n* of Equation (3). The function of Equation (3) is called the *response surface*.

Using the equation above, an approximation to the classical optimization problem can be established:

$$\hat{\psi}_i(\mathbf{b}) \Rightarrow \min, i = 0$$
$$\hat{\psi}_i(\mathbf{b}) \le 0, i = 1, \ldots, n_\psi$$

A modified version of the standard response surface method, as described above, analyzes the designs as the optimization proceeds and will be called the sequential response surface method which is used in *Altair StudyWizard*, (it is to be renamed with *HyperStudy* adding robust and stochastic analysis). This optimization software is interfaced with several FE solvers such as *LS-DYNA* (Hallquist, 1998) or PAM-CRASH. Consider a quadratic response surface as

$$\psi_i(b) \approx \hat{\psi}_i(b) = a_{i0} + \sum_{p=1}^{m} a_{ip} b_p + \sum_{p=1}^{m} \sum_{q=1}^{m} a_{ipq} b_p b_q \quad \text{...........}\mathbf{(4)}$$

The coefficients a_{i0}, a_{ip}, a_{ipq} are determined using a least square fit of the function $\hat{\psi}_i$ on the existing designs. In the first step $s=1+m$ designs are analyzed, and the coefficients a_{i0}, a_{ip} are determined. The next *m* designs are analyzed to determine the coefficients a_{ipp}. Additional designs are used to determine the remaining coefficients as it becomes necessary. After $s=1+m+(1+m)\,m/2$ designs, an algorithm is used to weigh the designs. The optimization procedure is as follows:

1. Analyze the initial design and *m* perturbed designs
2. Use a least square fit to determine the polynomial coefficients for the objective function $\hat{\psi}_o(\mathbf{b})$ and each constraint $\hat{\psi}_i(\mathbf{b})$

   If $s=(1+m)$, then calculate a_{i0}, a_{ip}

   If $1+m < s < 2(1+m)$, then calculate a_{i0}, a_{ip}, a_{ipp}, $i = 1, s-1-m$

   If $1+2m < s < 2+m+(1+m)\,m/2$, then calculate a_{i0}, a_{ip}, a_{ipp} and the successive off diagonal elements a_{ipq}

   If $s > 1+m+(1+m)\,m/2$, then weight the designs and calculated all a_{i0}, a_{ip}, a_{ipq}
3. Solve for the approximate optimum design using mathematical programming
4. Analyze the approximate optimum design
5. If the design has converged, stop
6. Back to 2

The procedure is shown in Figure 2. for a problem with one design variable. The theory of sequential response surface optimization is further explained by Schramm and Thomas in reference [6][11].

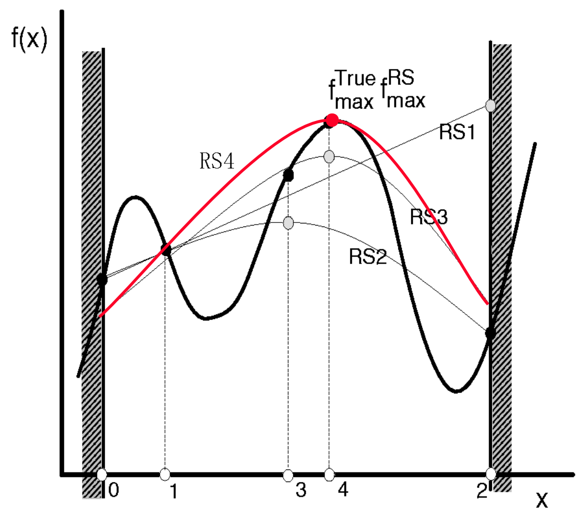


Fig 2. Response Surface Method

In general, crash problems are often defined as follows;

Objective:	$W(\mathbf{p}) \Rightarrow \min$
Subject to	$\mathbf{M}\,{}^t\ddot{\mathbf{d}} = {}^t\mathbf{f}({}^t\mathbf{d}, {}^t\dot{\mathbf{d}}, \mathbf{p}),\ 0 \le t \le t_e$
with	${}^0\mathbf{d} = \mathbf{d}_0,\ {}^0\dot{\mathbf{d}} = \dot{\mathbf{d}}_0$ ....(5)
Constraints:	$\mathbf{g}(\mathbf{d}, \dot{\mathbf{d}}, \ddot{\mathbf{d}}, \mathbf{p}) \le \mathbf{0}$
Side constraints:	$\mathbf{p}^l \le \mathbf{p} \le \mathbf{p}^u$

The objective function is a structural response like acceleration of vehicle, mass of certain body, deformation of vehicle or whatever engineer could be interested in.

The constraints are also structural responses.

In the structural responses, displacements, velocities, accelerations, stresses are grouped as time dependent

responses, often requiring the noise signal to be filtered, in order to make up a feasible engineering result.
The other responses, like energies, injury criteria, and crash performance taken with average in general may be used directly in definition of response function for objective or constraints because they are integral type.
Using the equations (5), objective and constraints can be calculated and determined by combination of computational analysis programs, ordinary FE and optimization.

Industrial application of structural optimization is divided into the sizing and shape optimization distinguished by the type of design parameters.
Size optimization is about the change of dimensions of structure for instance, thickness of structure.
Shape optimization is about the change of boundary of the structure and its design parameters are to comply with certain design requirements that require moving the nodes of FE mesh, both on the surface and in the inner of mesh, and keeping good shaped elements.
The shape functions that are commonly accepted to prescribe the surface and the inner parts of FE model can be linear or quadratic.
The nodal positions during optimization are computed as follows;

$$x_n = {}^0x_n + \sum_{i=1}^{NDV} \frac{\partial x_n}{\partial b_i} \Delta b_i$$

For this shape optimization, the creation of shape functions is fully supported in HyperMesh, Altair's finite element preprocessor [12].

StudyWizard

Parameter study and optimization can be usefully combined to investigate and optimize the behavior of a structural model. Response surface methods can handle just a few design variables since otherwise the computational effort is too high. Especially in crash analysis, where a single analysis run requires from several hours to days of computer time, any effort to reduce the wait time needs to be made. The number of design variables for an optimization should be limited to about ten.
If the number of design variables is very high, it is advisable to first run a screening Design of Experiment to determine design variables of large influence.
Altair's StudyWizard combines parameter studies using Design of Experiments approaches and sequential response surface optimization [13]. The same parameterized model can be used in both approaches. The StudyWizard is integrated in the post-processing software HyperView [14]. This way all the interfaces to different solvers can be accessed and multi-disciplinary optimization can be performed.

The creation of parameterized models can be automated with custom wizards. For the use of the StudyWizard with LS-DYNA such a parameterization wizard is available. The user can select the type of design variables such as shell thickness, shape, materials, load curves from a wizard window and the parameterized deck is created automatically. Almost no deck editing is involved.
(http://www.altair.com/, http://www.altairjp.co.jp)

Crash Optimization

From the computational crash simulations, a designer could pick up the weak points and make feasible design changes. The procedure of definitive changes depends upon the engineering sense or engineering standard database made by a previous study.
If they have no idea how to measure crash problems and how to attempt crash optimization due to a new study, the process would be as follows;

- Exclude the possible numerical errors with respect to change of design parameters or its instability and extract design parameters considering crash phenomenon that have no change qualitatively.

- Design of Experimental method is used to understand the factorial effects in design parameters and sample the design type. For the sampling design parameters, the Latin square technique is often used.

- Global approximation methods may facilitate optimization for crash problem. A second order Response Surface type model is made up with wide crash design parameters and is implemented to do optimization of the complex structural crash problems.

Frontal Oblique Offset Crash Type

In order to reconstruct the real aggressivity characteristics such as mass, stiffness, geometric profiles, NHTSA has conducted a series of vehicle-to-vehicle crash tests. A typical example of fatality in LTV-to-Car crash types comes from reference [2]. The results are repeated in figure 3 to 4.
According to the research based on the 1988-1996 NASS-CDS files, this crash type is severe in both crash pulse and intrusion. Therefore it is proper to estimate the restraint systems and structural improvements at the same time because restraint systems are generally designed by crash pulse basis and car structure are designed by deformation basis.

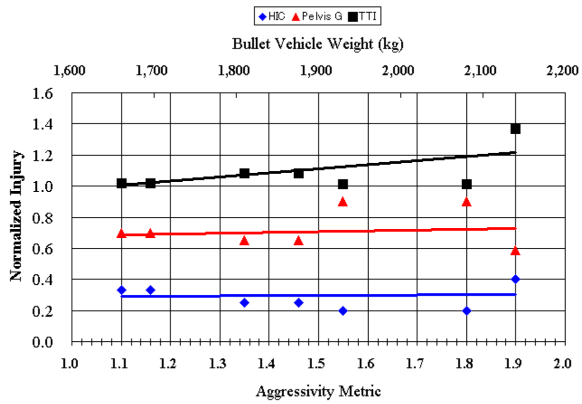


Fig 3. Side Impact Injury Measurements with respect to Aggressivity Metric and Bullet Vehicle Weight.

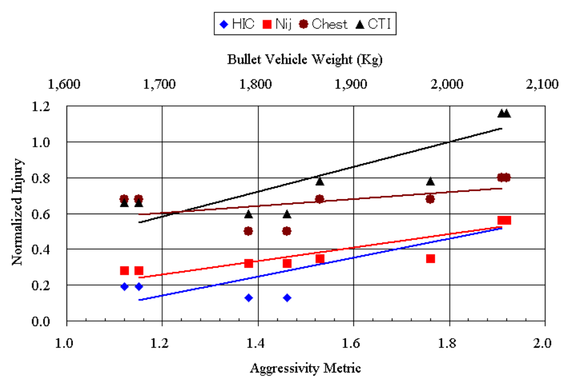


Fig 4. Frontal Impact Injury Measurements with respect to Aggressivity Metric and Bullet Vehicle Weight.

In this study, they concluded the side impact test series did not show a strong relationship between the injury measures of the target vehicle and either the mass of the striking vehicle or the aggessivity metric based on FARS/GES data.

The oblique offset test series, however, shows good correlation of driver injury measure with both the vehicle mass and the aggressivity metric.

And they commented that more research should be done to better understand compatibility. Additional factors may be affecting vehicle aggressivity, even though the frontal offset tests show a good correlation.

In the determination of oblique offset test series, analysis of the NASS crash data files of drivers in frontal collision, one representing with the highest frequency and serious-to-fatal injury risk is addressed.

For these reasons, the optimization for vehicle to vehicle crash compatibility using FE methods should consider the frontal oblique offset case only, due to the fact that more clarity is seen in the frontal impact than side impact.

As in reference [2], the crash type is shown in figure 5.

To see figure 5, this crash type should be catalogued as the central collinear collision is not so much in rotational effects during the crash and the simulation substantiate this by representing very small amount of rotational acceleration till ending of translational deceleration.

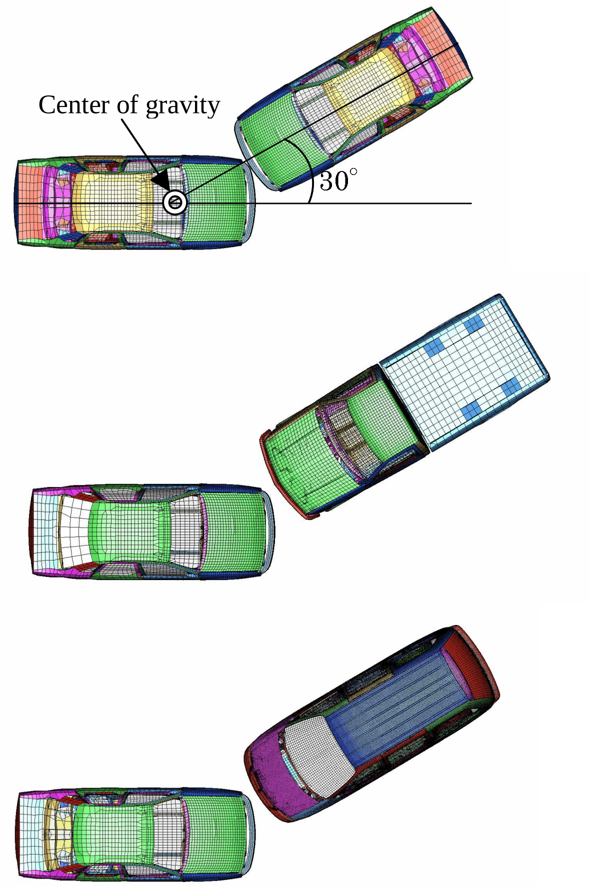


Figure 5. Frontal Oblique Offset Crash configuration

Optimization of car-to-LTV

To represent the LTV-to-Car crash types, the vehicle categories were selected and tuned as the table 7 shows.

Table 7. Frontal Oblique Offset Impact Vehicle Weights

Vehicle Category	Target Vehicle	Striking Vehicle
Passenger Car Class 3		Ford Taurus (W=1,511kg)
Pickup Class 2	Ford Taurus (W=1,511kg)	Chevrolet C2500 (W=2,254Kg)
Van Class 1		Dodge Caravan (W=1,910Kg)

In order to get more accurate body deceleration signal, the original Taurus FE model mesh density (of which the dash and floor was too rough to give us a feasible response) was modified as moderate element size between front and rear mesh density.

The deformable to rigid option of LS-DYNA, allowing deformable parts to be switched to rigid, was used in the Dodge Caravan model that reach 329,300 elements because this is too big to fulfill optimization study.

A parameter study or a DOE (design of experiments) study has been often used for determining which parameters are most influential on the response.

In this compatibility study, the optimization study follows a DOE study that could give us main parameter effects to be considered at first.

DOE and Optimization study is implemented using StudyWizard, and the crash problem is solved using LS-DYNA. Numerical solutions are performed on DELL Intel Xeon 2.4 GHz (4 CPUs, 4GB Memory100GB free disk Windows 2000). A single analysis of Taurus-to-Taurus impact takes about 7 hours, Taurus-to-C2500 pick up truck about 10 hours and Taurus-to-Caravan about 21 hours.

At the first step, it is necessary to define the parameter that is to be considered or likely to affect the incompatibility problem.

Among the many parameters we could discuss, the following design parameters (as mentioned in table 8) are selected first, because many times crash engineers experience those in crashworthiness.

For a DOE of 4 design parameters for each impact, $L_{16}(2^{15})$ Latin square is used because 4 design parameters of each come to 12 design parameters in 3 impact models.

Table 8. Design Parameters for DOE study

Design Parameters	Description
Engine Position	Movement forward and backward (+/- 30mm)
FRT Member Inner Thickness	Upper Value 3mm Lower Value 1mm
Sub Frame Member Thickness	Upper Value 3.6mm Lower Value 1.5mm
Dash Lower PNL Thickness	Upper Value 3.6mm Lower Value 1.0mm

To extend the commonly accepted idea of head injury criterion (HIC) calculation, the average deceleration means imposing comparative impulse as to different crash conditions. . This average deceleration, however, is not blocked with specific maximum duration time to pick up the burst signal and weighted as like mathematical HIC calculation.

A plot of the relationship between the vehicle weights and average deceleration calculated by integration of deceleration curve with respect to time and then divided by its time duration is shown in fig 6. This substantiates the understanding of aggressivity metric in frontal oblique offset impact case as was done and mentioned in reference 2.

One must carefully consider the fact that structural modifications are included here. The heavier bullet vehicle weight imposes more impulse in general. Thus, the possibility of reducing the impulse with structural modification is found, when we see the bottom case on the heavier bullet vehicle weight. But it should be aggressivity to the other passenger car.

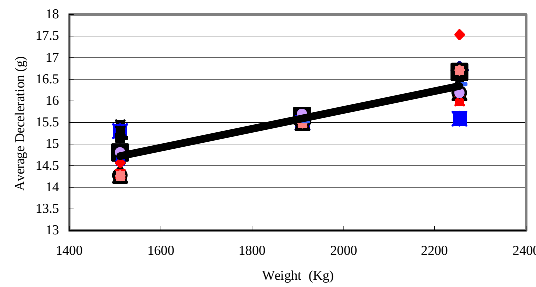


Fig 6. Changing average deceleration with respect to bullet vehicle weight and structural modification.

Finding the main parameter, which most influences the responses of average deceleration and displacement is the goal here. The average deceleration monitored at driver side is influenced more by the sub frame member thickness and Engine position as shown in fig 7.

Meanwhile, the thickness of the front side member and dash does not result in a change in the average deceleration.

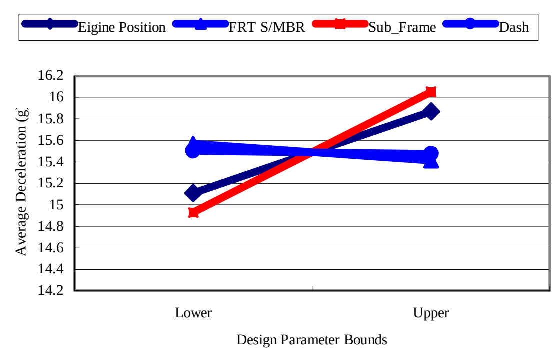


Fig 7. Design Parameters influence in Average Deceleration.

Fig 8 shows that the dash thickness and engine position influenced the other displacement responses monitored at several points of dashboard. Sub frame thickness also shows more influence with an inverse proportional relationship.

A review of fig 9 (which illustrates the deformed shape) shows the reason why this front side member does not affect both deceleration and displacement. The front side member have designed to get axial high deformation energy in the full-lap frontal impact is easy to bend in this frontal oblique offset, because of loading angle changed from perpendicular to 30 degree angled impact condition. But sub frame shows more stable axial deformation due to originally angled layout against oblique impact angle and its support by front crossed member.

Therefore we may say the side member is necessary to change in initial shape instead of its thickness and the shape optimization for this matter of changing initial geometry is efficient to change deformation mode close to axial.

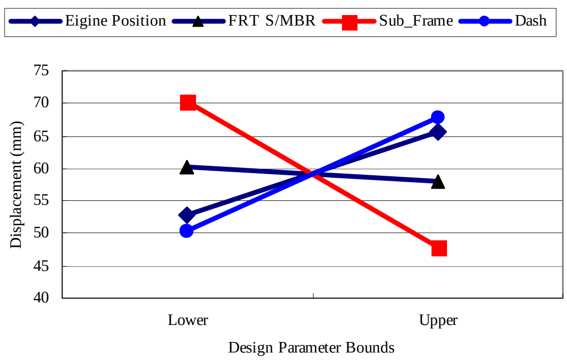


Fig 8. Design Parameters influence in Displacement.

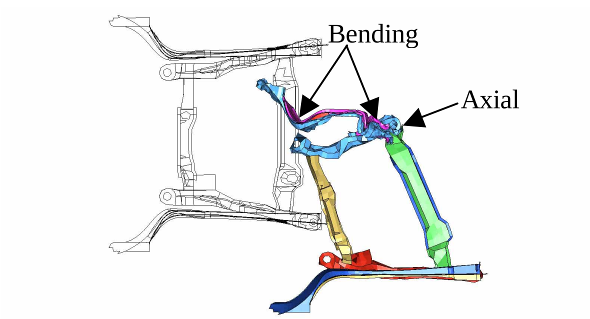


Fig 9. The main members deformed shape during the frontal oblique offset impact

In the deformation of each crash as shown in fig 10, not surprisingly the target Taurus has more deformation than the same bullet Taurus, and that deformation is in the proportion to bullet car weight in general. The deformed shapes of the target car show the more possible injury through the more deformation of compartment as it pinpoint the collapse of center pillar especially in Taurus-to-C2500 due to the high impact position overriding the passenger car.

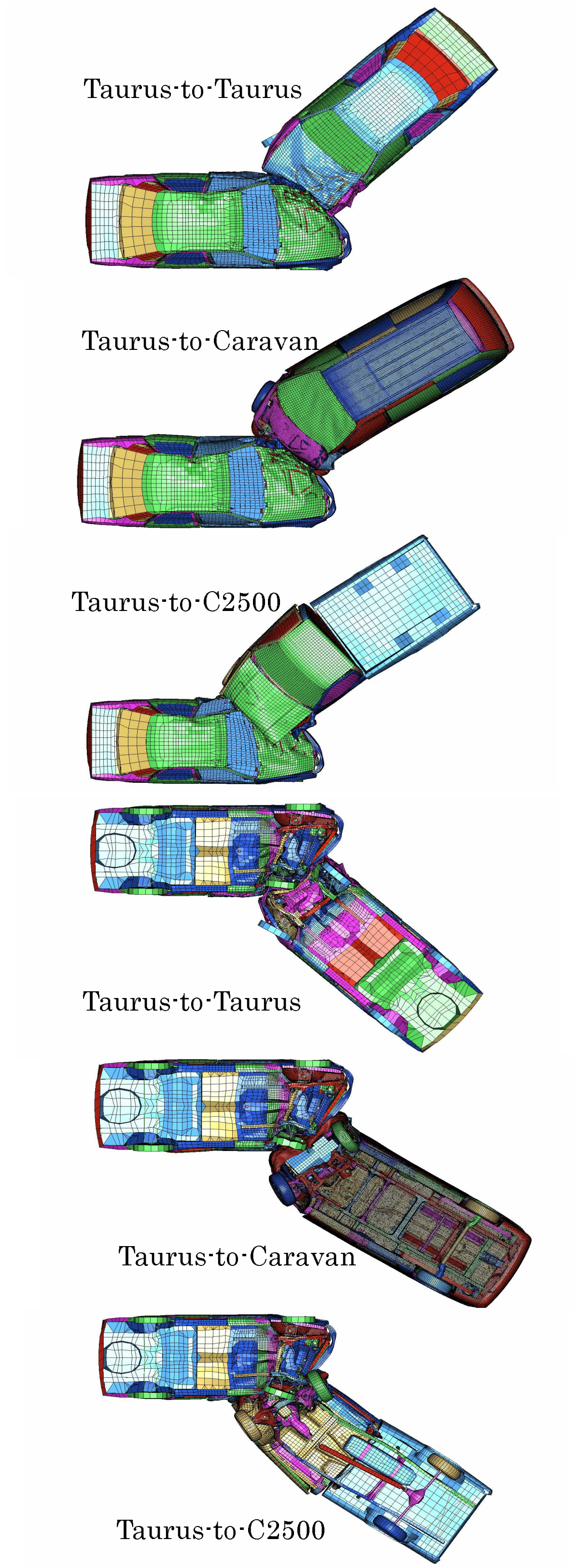


Fig 10. Simulated deformation in the Car-to-LTV impact

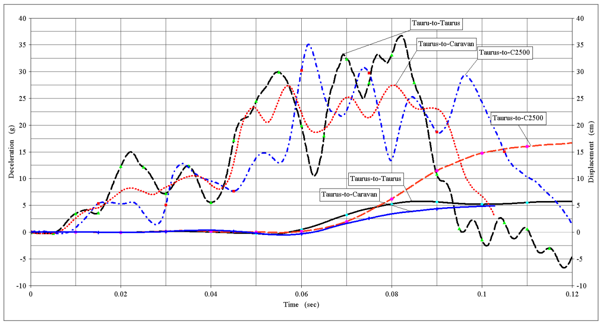


Fig 11. Body Deceleration Curve and Dashboard Intrusion.

Having the DOE results, an optimization study of compatibility follows it, utilizing parameters as listed in table 9 that is distinguished with shape and size variables. Shape optimization refers to the geometric shape changes while size optimization considers the gauge effects of any parts.

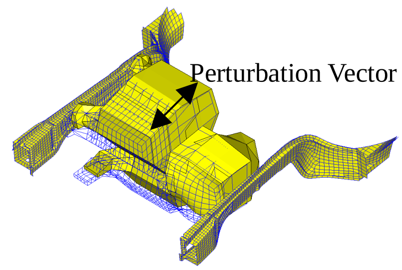
In this study, both size and shape optimization have been utilized simultaneously to consider the efficient ways of changing geometry.

Table 9. Design Parameters and its bounds for optimization (Shape and size variables)

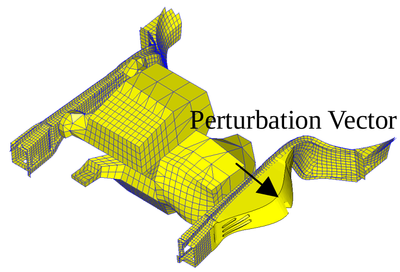
Design Parameters	Initial Value	Lower Value	Upper Value
Engine Position	0.0	-30	30
FRT Member Wider Shape	0.0	0	20
FRT Member Inner Thickness	1.5	1	3
FRT Member Reinf Thickness	1.5	1	3
Sub Frame Longitudinal Thickness	1.930	1.5	3.6
Dash LWR Thickness	1.120	1.0	3.6

The shape optimization is used in the engine positioning and shape of front longitudinal side member. To define the shape variable perturbations, nodal domains are defined having vectors specified at the vertices, and internal nodes rounded by that domains are interpolated linearly. This is very similar to finite element shape function algorithm in basic idea. Specifying vectors at the center between vertices, 2nd order shape variables could be represented.

Specifically, the shape optimization of the front side member needs to be considered because the original deformed shape of it was very easy to bend at change of section combined with impact loading direction of frontal oblique offset impact, shown in previous DOE study.



(a) Engine forward and rearward movement



(b) Front Side Member Wider perturbation

Fig 12. perturb FE mesh for shape optimization

The problem is formulated as a typical crashworthiness concept ;

Minimize : intrusion of compartment

Within design range :

$$25g \le \max(\text{Deceleration}) \le 45g$$

$$10g \le \text{mean}(\text{Eng Room Decel}) \le 15g$$

$$13g \le \text{mean}(\text{Compart Decel}) \le 15g$$

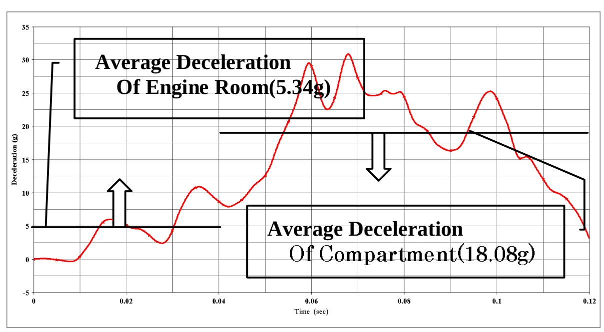


Fig 13. Concept of Design Range

Even though the role of restraint system has not been investigated clearly at this time, the seat belt could be effective to moderate injury level at even high deceleration. The range of deceleration, therefore, may be set to the upper 45g. At that value the seat belt can manage to reduce the injury level without so much venturing. The lower 25g is not thought to be an issue in designing the car structure.

Other design ranges for the engine room average deceleration and the compartment average deceleration, as shown in figure 13, are associated with improving crashworthiness by the increase of front stiffness leading to decrease force level at the second impact stage.

In order to set the threshold, the most moderated impact condition of car-to-car impact was used, and then it were taken out as the increased 10g of lower bound at front stiffness and the moderated 15g of upper bound at second impact stage. In the mean time, the design variables were selected as only rearward of front side member, it may have not related with pedestrian impact.

Minimizing the intrusion into the compartment would make sense if the car structure is going to be strong simultaneously. And it is doable design method to prevent occupant injury suffering from large deformation to the extent that restraint system could not support to moderate injury.

Unfortunately, the number of design parameters are limited because of the computation time, as this full vehicle impact models take about 3 days for one iteration. To think this 6 parameters, sum up to 18 parameters for 3 impact models, it is required to execute 18 runs at least for initialize stage of optimization progress.

The intermediate results are shown in fig 14 and fig 15, the change of objective dashboard intrusion and the change of body deceleration for each.

We hope to show the optimized results at coming 18th ESV conference.

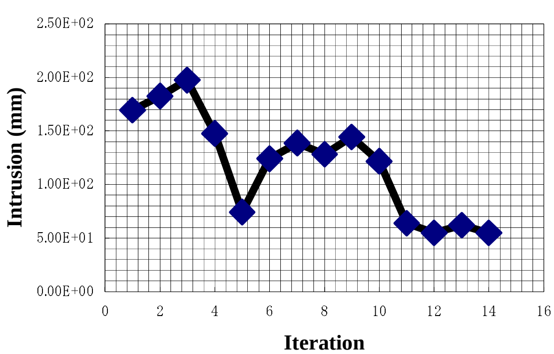


Fig 14. Optimization History for objective function as intrusion minimize

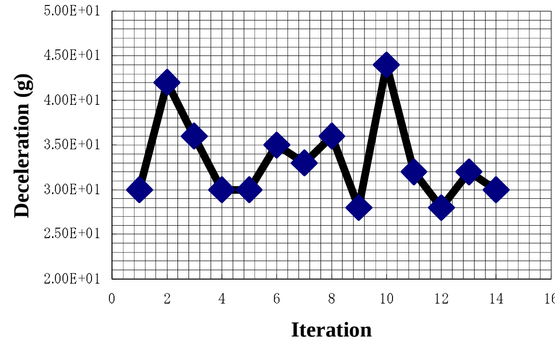


Fig 15. Optimization History for constraint function with body deceleration

Figure 16 compares curves between the initial run and intermediate run at iteration 14, showing some of the changed responses.

In the ideal optimized results to the compatibility problem, the balanced increase of body deceleration and the minimization of intrusion are desired. The intermediate results, therefore, are not sufficient to conclude because deceleration is also decreased along with the decreased intrusion.

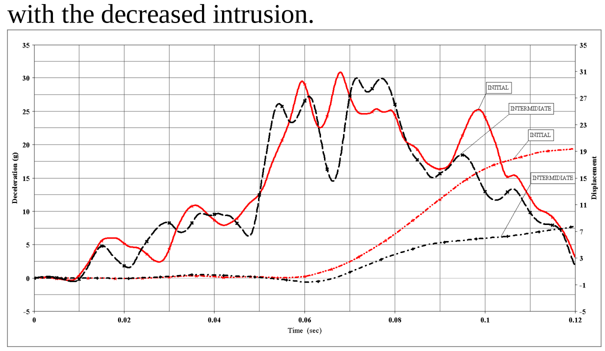


Fig 16. Body deceleration curve and intrusion between initial and intermediate at 14 run.

CONCLUSIONS

The main objective of this study is to show that bullet vehicle weight deepens aggressivity by the DOE study coupled with a optimization method and a nonlinear dynamic FE code. Then, these same methods are demonstrated to propose an optimized passenger car structure against the incompatibility of LTV to car.

As a starting view for compatibility problems, the parameters considered are in the line of crashworthiness, because of the similarity in structural improvement and response analysis.

Combining structural improvement and bullet vehicle weight, the DOE results substantiate that the tendency is close to aggressivity metric, and that structural improvements also can leverage the incompatibility.

The results of the DOE indicate that engine position and sub frame are more influential in both deceleration and displacement responses. In contrast with full-lap or offset case, the front side member is believed to act as the main resistance and crash energy absorbing member having not influenced both responses.

This means front side member need to change its shape to be more effective in resistance force and crash energy absorbing, ideally it should exhibit an axial deformation mode. In this case, shape optimization technique has facilitated the consideration of geometric design space more efficiently.

Even though optimization followed DOE study, the final results are unfortunately unavailable because it is still running and just in initial step.

The optimization for vehicle-to-vehicle crash is a very time consuming job. Of course it depends on the FE model size and the number of design parameters. In this case it took about 2 months just for the initial step with 18 design parameters.

Another problem for FE model size is that enough detail should be included, allowing representation of the large deformation suffered with aggressive bullet vehicle having heavy weight and stiffer structure. The frontal oblique offset crash especially shows more severe deformation of compartment.

The challenge is to reduce the number of elements not affecting the response to be discussed or to seek the way blocked consideration at specific time zone sequentially.

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