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MODULAR APPROACH TO STRUCTURAL SIMULATION FOR VEHICLE CRASHWORTHINESS PREDICTION

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16. Abstract 1 A modular formulation for simulation of the structural 2 deformation and deceleration of a vehicle for crashworthiness 3 and collision compatibility is presented. This formulation 4 includes three dimensional beam elements, various spring 5 elements, rigid body elements, and modal elements.			
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PREFACE

Computer simulation of structural deformation under vehicle crash conditions is an important tool for the prediction and design improvement of vehicle crashworthiness. As an alternative and aid to crash testing it also provides important cost benefits. There are various mathematical models, with varying degrees of sophistication, describing the dynamics behavior of structures. The present study is an attempt to unify the different approaches by using a modular concept. Such an approach can provide the flexibility to model different parts of a vehicle with different degrees of sophistication and computer efficiency, and, hopefully, it can help avoid the need for large, cumbersome software.

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1. INTRODUCTION

Computer simulation of structural deformation under vehicle crash conditions is an important tool for the prediction and design improvement of vehicle crashworthiness. As an alternative and aid to crash testing it also provides important cost benefits. Efficient computer models are needed to provide both economy and flexibility in all analytical simulation efforts. In particular, it is of value to be able to simulate a specific crash situation with a minimum of software. This should lead to simplicity of operation by engineers, and also aid in engineering judgement regarding prediction and design.

The Calspan-Shieh two dimensional frame program, the Battelle FMCCM lumped mass program, together with the 3-dimensional programs under development at Lockheed, the University of Michigan and TSC have been studied as to their feasibility and effectiveness for general usage by engineers.

The Battelle-FMCCM program¹ is essentially a large simplified spring-mass model which is restricted to unidirectional motion. This model handles collinear vehicle-to-vehicle as well as vehicle-to-barrier impact conditions. It can include four masses and up to 35 nonlinear resistances in the form of elasto-plastic springs, hydraulic energy absorber elements, and viscous dampers. The load deformation characteristics are obtained from theoretical and/or experimental data. The crash data must, of course, be interpreted carefully. This model provides the ability to predict overall vehicle response, and is useful in making parameter and design trade-off studies.

In regard to the frame models, the Calspan model² which represents the earliest development in this regard, is applicable to 2-dimensional frame structures. It is essentially a finite element model with straight-beam elements, lumped masses at nodes, and localized plastic hinges at pre-selected nodes. The program is operational, and at present, can be useful for simple front frame and bumper configurations.

The 3-D TSC/DOT model³ presently being developed at the Transportation Systems Center can be regarded as a three-dimensional extension of the Calspan model, in the sense that the assumptions on ideal plastic hinges and lumped masses at nodes are the same as in Reference 2. The analysis and computer implementation of the TSC/DOT model is more heavily based on finite element techniques.⁴⁻⁶ Another three-dimensional frame model, developed by Lockheed and referred to as KRASH⁷ has been installed on TSC equipment. It is a 3-dimensional extension of the FMCCM model, consisting of mass connected straight "beam" elements. Forces and moments are determined from a linear elastic stiffness matrix which is adjusted for plasticity by multiplying by a stiffness reduction factor. The stiffness reduction factor concept is theoretically incorrect in three-dimensional problems, since the stress vector can move on the yield surface having one component increase and another component decrease in the same loading step. However, under some deformation mode, if the proper factor is used, it could produce a reasonable result. CRASH⁸ is a more general three-dimensional frame model which uses plasticity theory. It has been used to model the vehicle to barrier impact of a Mustang. The data indicated that the simulation gave too stiff a result. The work at the University of Michigan falls in the general category of 3-dimensional frame models using the plastic hinge concept, and is discussed in Ref. 9. Reference 9 has also proposed a kind of modular concept to model major components of a vehicle.

Additional features which would produce more complete simulation should, for instance, include the ability to unite a rigid body element (such as the engine block) with a deformable structure, and the ability to allow an optimum mix of several finite elements (i.e., plate, beam and specialized spring elements) to represent vehicle components. Such features can be incorporated conveniently by means of a modular approach. Such an approach can provide both computer efficiency and detailed accuracy in a specific situation; hopefully, to avoid the need for large, cumbersome software.

The main purpose of this report is to describe a unified modular approach for vehicle structural simulation. Such

an approach will provide the flexibility to model different parts of a vehicle with different degrees of sophistication, depending on the type of information one is seeking. For example, in the frontal or oblique impact of a vehicle, detailed modelling of the front end may be necessary; however, the passenger compartment and the rear end can be modelled as a rigid body of finite size. In the case where more detailed information is required for the passenger compartment, such as differences in acceleration at several passenger locations, the elastic motion of the passenger compartment may have to be considered. In this case, a simple modal approximation can be used for the passenger compartment itself.

In the present study, an explicit scheme is used for numerical integration. A beam element which allows large three-dimensional rotations and plastic hinge action, a spring element which can only take tension or compression, a finite size rigid body element and an element using modal approximations are presented in detail in this report.

2. MODULAR FORMULATION AND TIME INTEGRATION OF A DYNAMIC SYSTEM

Consider a vehicle consisting of a finite number of components, such as bumpers, chassis, engine, passenger components, wheels, etc. A schematic sketch is shown in Figure 1. A component itself may consist of one or many individual elements. The behavior of a component (or an element) is expressed in terms of the generalized coordinates at a finite number of points of the component (or the element) in the form of component (or element) mass matrix $\tilde{m}_i$, damping matrix $\tilde{c}_i$ and stiffness matrix $\tilde{k}_i$. Some or all of these points are connected to adjacent components and the discussion of the appropriate matrices will be given in Section 3.

The component matrices can, in general, be functions of the generalized coordinates and/or their rates of change. At the common nodes of all different components one should use the same type of generalized coordinates. Then, all the component (or element)

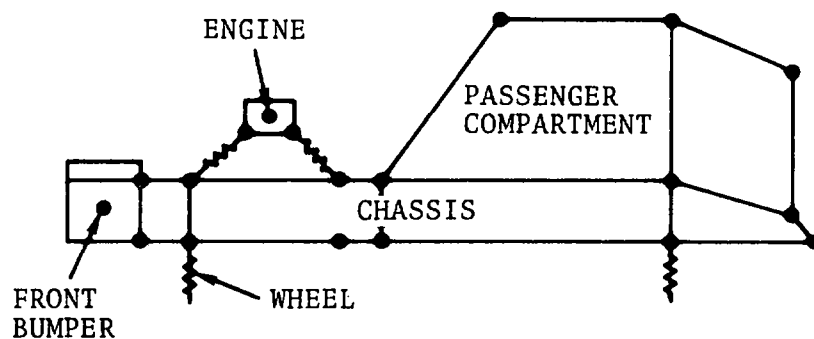


Figure 1. Vehicle Schematic Sketch

matrices can be assembled to yield the following system of equations of motion.

$$\underline{\underline{M}} \ddot{\underline{q}} + \underline{\underline{C}} \dot{\underline{q}} + \underline{\underline{K}} \underline{q} = \underline{F} \quad (2.1)$$

where $\underline{\underline{M}}$, $\underline{\underline{C}}$ and $\underline{\underline{K}}$ are the assembled mass matrix, damping matrix, and stiffness matrix respectively; $\underline{q}$ is the vector and unconstrained coordinates and $\underline{F}$ is the loading vector due to external applied loads or prescribed constraints.

Equations (2.1) comprise a second order system. With given initial values of $\underline{q}$ and $\dot{\underline{q}}$ the time histories of $\ddot{\underline{q}}$ can be obtained by numerical integration. There are many numerical integration schemes; implicit or explicit. To choose a particular scheme to perform numerical integration one will have to be concerned with the possibility of introducing artificial damping and numerical instability (i.e., introducing artificial energy to the system in the integration process). For a linear dynamic system, how these numerical problems are related to the integration stepsize is well understood. In most cases, the implicit schemes introduce artificial damping and are unconditionally stable, while the explicit schemes introduce energy if the stepsize is too large, and are only conditionally stable. However, these assertions have not been established and cannot be considered true for a nonlinear system. An unstable scheme can be made conditionally stable only. In general an implicit scheme will take much longer computer time for each integration step than that of an explicit scheme of comparable truncation error. This is because for a nonlinear system certain matrices must be redefined at each time step. In the implicit scheme, the updated system of equations would have to be resolved, while the explicit scheme only involves updating the right hand side of the system of equations. For large systems this can be very time consuming. Therefore, the explicit integration scheme is used in this study.

The simplest scheme is the central difference method, which allows us to write (2.1) in the form,

$$\begin{aligned} \underline{M}_n (\Delta \underline{q}_{n+1} - \Delta \underline{q}_n) + \frac{\Delta t}{2} \underline{C}_n (\Delta \underline{q}_{n+1} + \Delta \underline{q}_n) \\ + (\underline{K}_n \underline{q}_n - \underline{F}_n) \Delta t^2 = 0 \end{aligned} \quad (2.2)$$

or

$$\begin{aligned} \left(\underline{M}_n + \frac{\Delta t}{2} \underline{C}_n \right) \Delta \underline{q}_{n+1} = \left(\underline{M}_n - \frac{\Delta t}{2} \underline{C}_n \right) \Delta \underline{q}_n \\ - (\underline{K}_n \underline{q}_n - \underline{F}_n) \Delta t^2 \end{aligned} \quad (2.3)$$

where

$$\underline{q}_n = \underline{q}_{n+1} + \Delta \underline{q}_n \quad (2.4)$$

and

$$\dot{\underline{q}}_n = \frac{\underline{q}_{n+1} - \underline{q}_{n-1}}{\Delta t}; \quad \ddot{\underline{q}}_n = \frac{\underline{q}_{n+1} - 2 \underline{q}_n + \underline{q}_{n-1}}{\Delta t^2}$$

The subscript n denotes the value of the quantity at time step n and Δt is the step size. From the given initial values $\underline{q}_0$ and $\dot{\underline{q}}_0$ we have

$$\Delta \underline{q}_1 = \dot{\underline{q}}_0 \Delta t + \frac{1}{2} (\Delta t)^2 \underline{a}_0 \quad (2.5)$$

where $\underline{a}_0$ is the initial acceleration which is evaluated from (2.1)

$$\underline{M} \underline{a}_0 = - \left[\underline{C} \dot{\underline{q}}_0 + \underline{K}_0 \underline{q}_0 - \underline{F}_0 \right] \quad (2.6)$$

The subsequent $\underline{q}$'s and $\Delta \underline{q}$'s are evaluated from (2.3) and (2.4). For a problem where $\underline{M}$ and $\underline{C}$ are independent of time and $\underline{q}$ itself, or are block diagonal matrices, $\Delta \underline{q}$, of each integration step can be determined very rapidly.

It is noted that in (2.3), one will only need the product of $\underline{K}_n$ and $\underline{q}_n$. Thus, in actual computation, there is no need to assemble $\underline{K}_n$. That is, at a given time step, if we let q^i and k_i be the generalized coordinates and the stiffness matrix of the i^{th} component (or element) respectively, then $f^i = k_i q^i$ will be a

generalized force vector of the element due to the deformation q_i^i . In order to obtain $K q$ at this time step, one only has to assemble f_i for all the elements. In this way, there is no multiplication of K and q . The actual multiplication is performed at the element level between k_i and q_i^i . The advantage of assembling f_i instead of k_i is the saving in core storage in the computer. Generally, larger core storage is required to store K (roughly number of degrees of freedom times the average semi-bandwidth). In some instances, f_i can be generated directly without even having to compute k_i . Detailed discussion of the computation of f_i will be given in the next section.

3. ELEMENT MATRICES

Several kinds of elements are discussed which can be used in the dynamic simulation of vehicles. It should be possible to combine them in a variety of ways to simulate vehicle components or groups of components.

3.1 BEAM ELEMENTS

Consider a beam element with local coordinate axes (X_1, X_2, X_3) and global cartesian coordinates (x_1, x_2, x_3) as shown in Figure 2. The two coordinate systems are related by a 3 x 3 matrix, say L which is defined in Appendix A, so that

$$\begin{Bmatrix} X_1 \\ X_2 \\ X_3 \end{Bmatrix}_i = L_{ni} \begin{Bmatrix} x_1 \\ x_2 \\ x_3 \end{Bmatrix}_i \quad (i = 1, 2) \quad (3.1)$$

The subscript i denotes the 1 or node 2. The element stiffness matrix for small elastic strains or small elastic incremental strains with initial stress N (positive for tension) is given by k where

$$k = \begin{bmatrix} \frac{AE}{l} & & & & & & & & & & \\ 0 & a' & & & & & & & & & \\ 0 & 0 & a & & & & & & & & \\ 0 & 0 & 0 & \frac{GJ}{l} & & & & & & & \\ 0 & 0 & -b & 0 & c & & & & & & \\ 0 & b' & 0 & 0 & 0 & c' & & & & & \\ \frac{AE}{l} & 0 & 0 & 0 & 0 & 0 & \frac{AE}{l} & & & & \\ 0 & -a' & 0 & 0 & 0 & -b' & 0 & a' & & & \\ 0 & 0 & -a & 0 & b & 0 & 0 & 0 & a & & \\ 0 & 0 & 0 & \frac{GJ}{l} & 0 & 0 & 0 & 0 & 0 & \frac{GJ}{l} & \\ 0 & 0 & -b & 0 & d & 0 & 0 & 0 & b & 0 & c \\ 0 & b' & 0 & 0 & 0 & d' & 0 & -b' & 0 & 0 & c' \end{bmatrix} \quad (3.2)$$

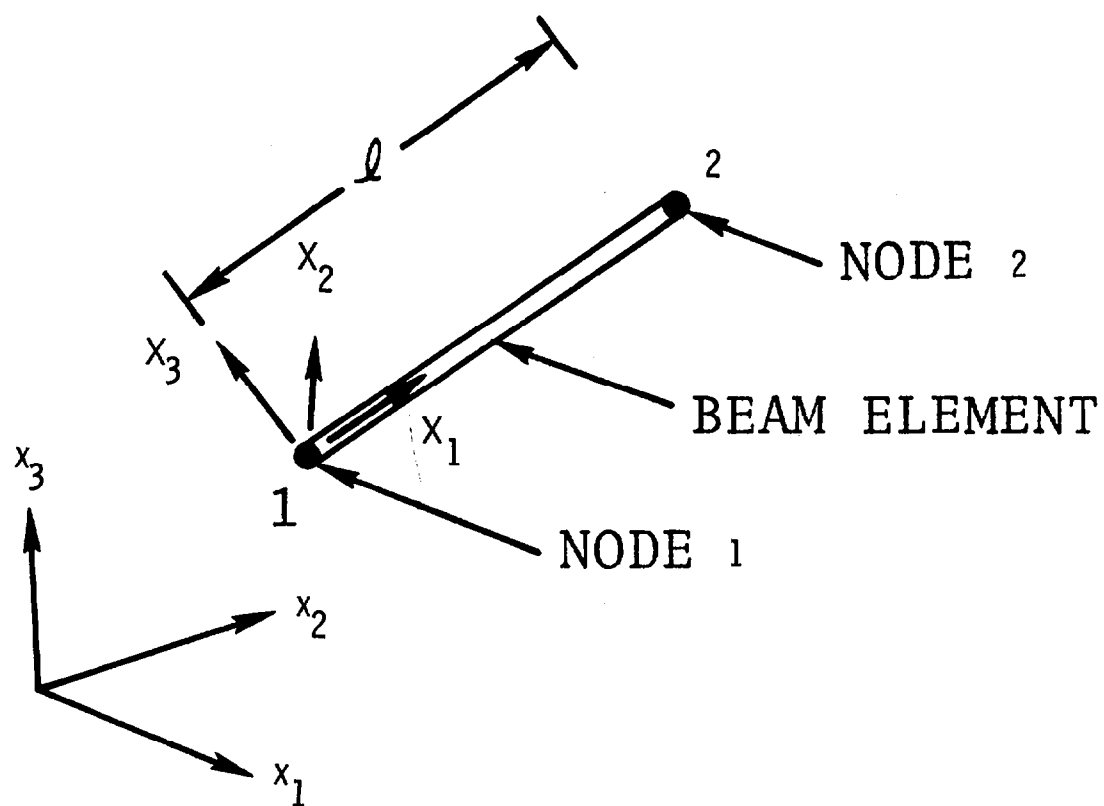


Figure 2. Beam Orientation; Local and Global Axes

in which

$$\begin{aligned}
 a &= \frac{N}{\ell} - \frac{2b}{\ell} \\
 b &= (\cosh \beta - 1)/D \\
 c &= \left(\cosh \beta - \frac{1}{\beta} \sinh \beta \right) / D \\
 d &= \left(\frac{\sinh \beta}{\beta} - 1 \right) / D
 \end{aligned} \tag{3.3}$$

and where

$$\begin{aligned}
 D &= \frac{\ell^2}{EI_2} \frac{\sinh \beta}{\beta} \left(\frac{\sinh \beta}{\beta} - 1 \right) - (\cosh \beta - 1)^2 / N \\
 \beta &= \sqrt{\frac{N\ell^2}{EI_2}}
 \end{aligned}$$

and similarly for a' , b' , c' and d' with I_2 being replaced by I_3 in the values of β and D given in (3.3). The generalized coordinates associated with the element stiffness matrix given by (3.2) are given in incremental form as

$$\Delta \underline{q}^T = \left\{ \Delta \underline{u}_1^T, \Delta \underline{\theta}_1^T, \Delta \underline{u}_2^T, \Delta \underline{\theta}_2^T \right\} = \left\{ \Delta q_1, \Delta q_2, \Delta q_3, \dots, \Delta q_{12} \right\} \tag{3.4}$$

which are the three displacements $\Delta \underline{u}_1$ followed by the three rotations $\Delta \underline{\theta}_1$ of node 1 in the local directions shown in Figure 2; then followed by the corresponding quantities of node 2. The element stiffness matrix given in 3.2 is an exact representation of the beam equation given below. The matrix 3.2 yields values which are equal to those of the closed form solutions at the nodes. It is derived by using the homogeneous solution of the straight beam equation with initial stress as the interpolation function (Ref. 10). In our case the interpolation function is

$$w = C_1 \cosh \sqrt{\frac{N}{EI}} s + C_2 \sinh \sqrt{\frac{N}{EI}} s + C_3 s + C_4$$

which is a solution to

$$EI \frac{d^4 w}{ds^4} + N \frac{d^2 w}{ds^2} = 0$$

and N is the initial stress.

For small increments of elastic deformation, the incremental generalized forces of the element, in correspondence to the generalized coordinates in (3.4), are

$$\Delta \tilde{f} = \left\{ (\Delta \tilde{F})_1^T, (\Delta \tilde{M})_1^T, (\Delta \tilde{F})_2^T, (\Delta \tilde{M})_2^T \right\} = \tilde{k} \Delta \tilde{q} \quad (3.5)$$

where

$$(\Delta \tilde{F})_i = \begin{Bmatrix} \Delta N \\ \Delta Q_2 \\ \Delta Q_3 \end{Bmatrix}_i \quad (\Delta \tilde{M})_i = \begin{Bmatrix} \Delta M_1 \\ \Delta M_2 \\ \Delta M_3 \end{Bmatrix}_i$$

and $(\Delta \tilde{F})_i$ is the force resultant vector at node i while $(\Delta \tilde{M})_i$ is the moment resultant vector at node i , as shown in Figure 3. Note that $\Delta F_1 = -\Delta F_2$ for equilibrium.

For increments involving plastic deformation, $\Delta \tilde{f}$ must be computed differently. The detailed procedure will depend upon the type of plasticity theory used. In the present report, we shall only use the incremental theory of the plastic hinge concept. The yielding surface ϕ , is assumed to be expressible in the form

$$\phi (M_i, N, K) = 0 \quad (3.6)$$

where K is a hardening parameter which depends on the strain history, M_i ($i = 1, 2, 3$) and N are moments and stress resultant respectively. If $\phi < 0$ or $\phi = 0$ and the first variation

$$\Delta \phi = \frac{\partial \phi}{\partial M_i} \Delta M_i + \frac{\partial \phi}{\partial N} \Delta N + \frac{\partial \phi}{\partial K} \Delta K < 0$$

there will be only elastic deformation. When $\phi = 0$, $\Delta \phi \geq 0$ plastic deformation occurs. However, $\Delta \phi > 0$ is not permitted. We must determine the portion of total deformation that is plastic to

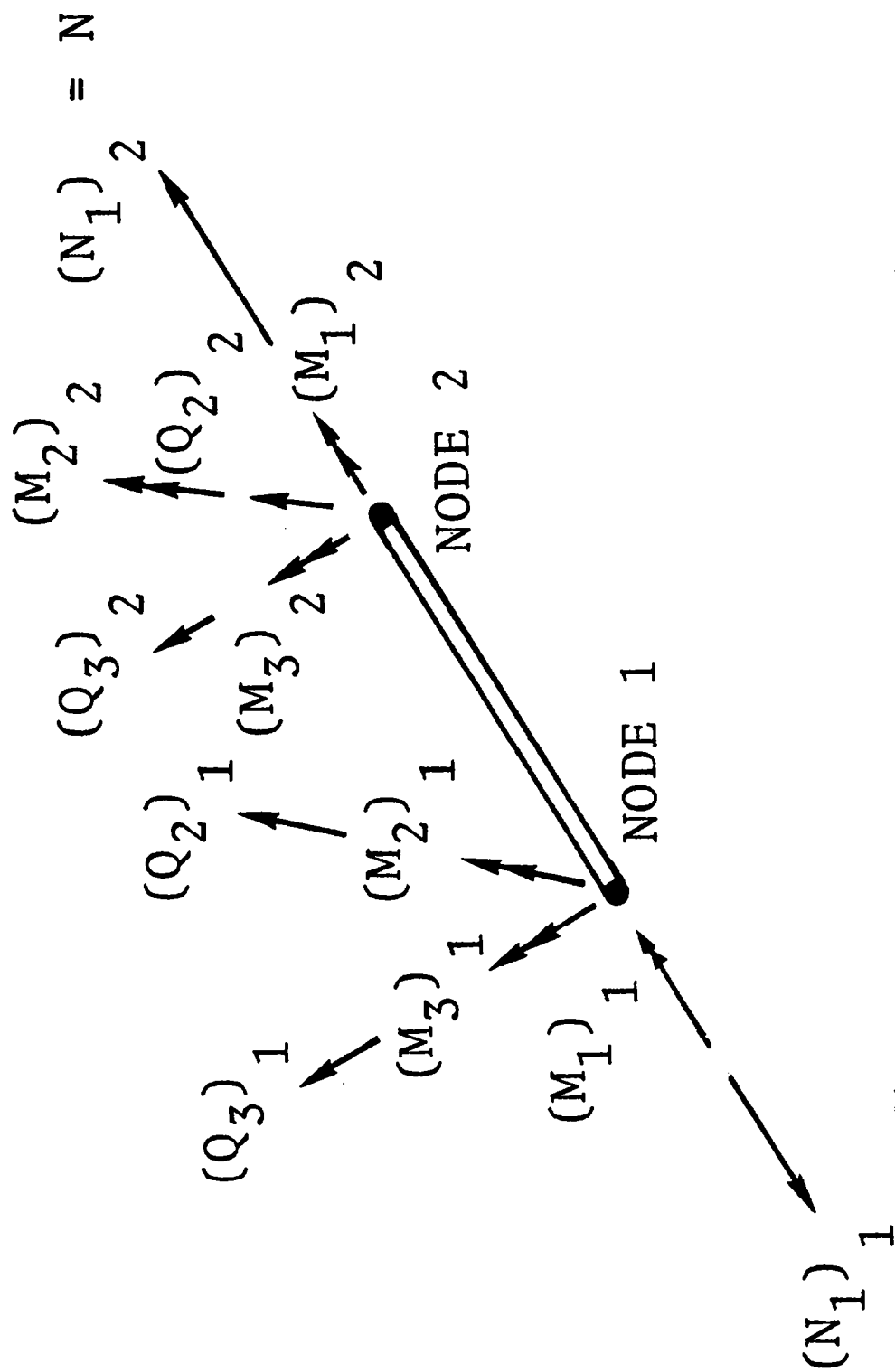


Figure 3. Force Resultants and Couples at Element Nodes

insure $\Delta\phi = 0$. Let $\Delta\gamma$ denote the total incremental strain due to the incremental deformation associated with Δq so that

$$\Delta\gamma = \begin{Bmatrix} \Delta e_1 \\ \Delta K_1 \\ \Delta K_2 \\ \Delta K_3 \end{Bmatrix} \quad (3.7)$$

where Δe_1 , is due to stretching, ΔK_1 , is due to torsion and ΔK_2 and ΔK_3 are due to bending which can be expressed in terms of incremental displacements. For example, ΔK_3 is the bending strain about the x_3 -axis, which is equal to $EI_3 d^2(\Delta u_2)/dx_1^2$ where Δu_2 is the incremental displacement in the x_2 direction. The total incremental strain can be separated into a part due to elastic deformation and a part due to plastic deformation, namely

$$\Delta\gamma = \Delta\gamma^e + \Delta\gamma^p$$

The incremental stresses will be computed from $\Delta\gamma^e$, the elastic incremental strain as follows,

$$\Delta\sigma = \begin{Bmatrix} \Delta N \\ \Delta M_1 \\ \Delta M_2 \\ \Delta M_3 \end{Bmatrix} = \tilde{E} \Delta\gamma^e \quad (3.8)$$

or

$$\Delta\sigma = \tilde{E} \Delta\gamma - \Delta\gamma^p = \Delta\sigma' - \tilde{E} \Delta\gamma^p \quad (3.9)$$

where

$$\Delta\sigma' = \tilde{E} \Delta\gamma = \begin{bmatrix} AE & 0 & 0 & 0 \\ 0 & GJ & 0 & 0 \\ 0 & 0 & EI_3 & 0 \\ 0 & 0 & 0 & EI_2 \end{bmatrix} \Delta\gamma \quad (3.10)$$

in which $\Delta\sigma$ is a function of distance along the beam element. When evaluated at an end point, $\Delta\sigma'$ will yield exact values by virtue of (3.2), and contribute to corresponding components in (3.5). ($\Delta\sigma'$ is equal to the incremental stresses $\Delta\sigma$, if this increment due to Δq is elastic.) We shall evaluate $\Delta\gamma^p$, the incremental strain vector based on incremental plasticity theory. According to the flow rule, $\Delta\gamma^p$ is normal to the yielding surface given by (3.6), i.e., (a schematic diagram is given in Figure 4).

$$\Delta\gamma^p = \lambda \begin{Bmatrix} \partial\Phi/\partial N \\ \partial\Phi/\partial M_1 \\ \partial\Phi/\partial M_2 \\ \partial\Phi/\partial M_3 \end{Bmatrix} \quad (3.11)$$

evaluated on the yield surface

where λ is a proportionality factor, which is to be determined so that the total cumulative stress vector, σ , lies on the yield surface. The total cumulative stress vector can be written as the sum of σ_0 , the stress vector just before the present increment and $\Delta\sigma$, i.e.,

$$\sigma = \begin{Bmatrix} N \\ M_1 \\ M_2 \\ M_3 \end{Bmatrix} = \sigma_0 + \Delta\sigma = \sigma_0 + \Delta\sigma' - \lambda E \begin{Bmatrix} \partial\Phi/\partial N \\ \partial\Phi/\partial M_1 \\ \partial\Phi/\partial M_2 \\ \partial\Phi/\partial M_3 \end{Bmatrix} \quad (3.12)$$

on yielding surface

When the expressions given by (3.12) are substituted into (3.6) an equation is obtained which allows determination of λ .

A simple example of a possible yield criterion is

$$\phi = a_4 N^2 + a_1 M_1^2 + a_2 M_2^2 + a_3 M_3^2 - K_0 = 0 \quad (3.13)$$

where the a 's are constants which depend on the geometry of the cross-section of the beam, and K_0 is another constant related to the square of the yield stress. In the case of elastic perfectly plastic material, K_0 is independent of strain history. To

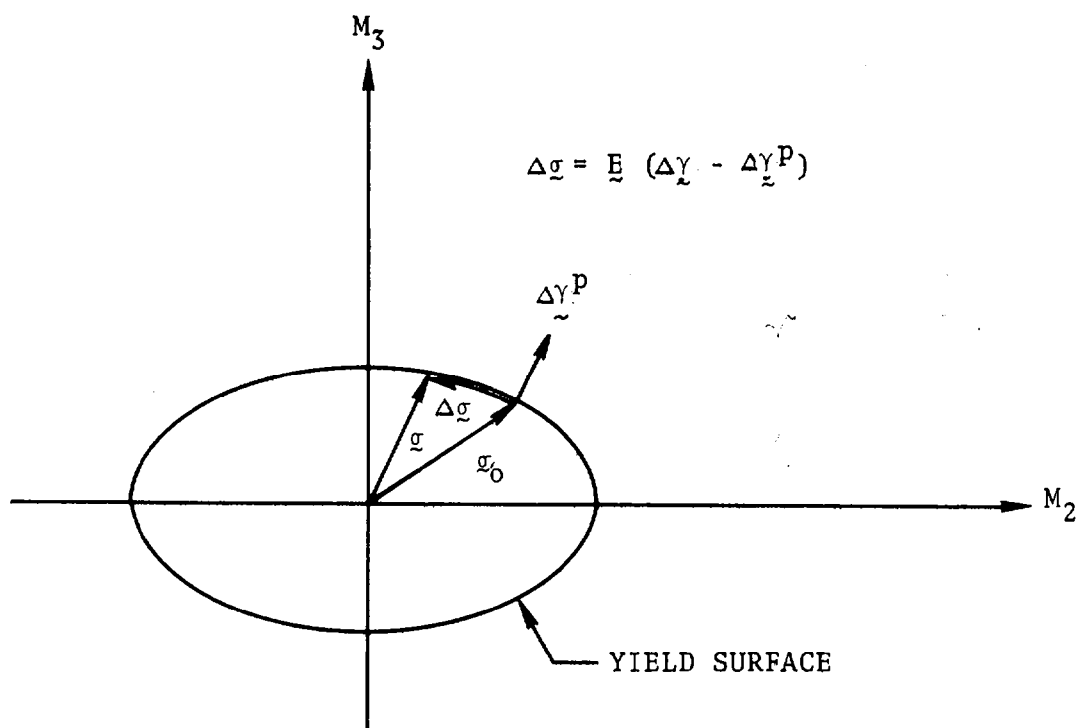


Figure 4. Schematic Diagram of the Stress Vector and the Incremental Plastic Strain Vector

determine λ in this case, we suppose that $\underline{\sigma}_0$ lies within the yield surface given by $\phi = 0$ (Eq. 3.13), while $\underline{\sigma}_0 + \Delta \underline{\sigma}'$ lies outside. The next step is to determine the values of the M's and N on the yield surface. To do this let

$$\underline{\sigma}'' = \underline{\sigma}_0 + \lambda' \Delta \underline{\sigma}' \quad (3.14)$$

and determine the factor λ' so that $\underline{\sigma}''$ just falls on the yield surface (i.e., $\lambda' \Delta \underline{\sigma}'$ is an elastic increment). Substituting (3.14) into (3.13) get

$$A(\lambda')^2 + 2B\lambda' + C = 0 \quad (3.15)$$

where

$$\begin{aligned} A &= a_4 (\Delta N')^2 + \sum_{i=1}^3 a_i (\Delta M'_i)^2 \\ B &= - \left[a_4 N_0 \Delta N' + \sum_{i=1}^3 a_i (M_i)_0 \Delta M'_i \right] \\ C &= a_4 N_0^2 + \sum_{i=1}^3 a_i (M_i)_0^2 - K_0 \end{aligned} \quad (3.16)$$

Then

$$\lambda' = \frac{C}{B \left(1 + \sqrt{1 - AC/B^2} \right)} \quad (3.17)$$

With λ' defined by (3.17), $\underline{\sigma}''$ will be on the yield surface., whose components are N''_1 , M''_1 , M''_2 , and M''_3 . Now in order to compute λ given in (3.11) the following partial derivatives are needed

$$\begin{Bmatrix} \partial\Phi/\partial N \\ \partial\Phi/\partial M_1 \\ \partial\Phi/\partial M_2 \\ \partial\Phi/\partial M_3 \end{Bmatrix} = \begin{Bmatrix} 2a_4 N'' \\ 2a_1 M_1'' \\ 2a_2 M_2'' \\ 2a_3 M_3'' \end{Bmatrix} \quad (3.18)$$

On substituting (3.12) (also using (3.18)) into (3.13), an equation of the same form as that of (3.15) is obtained to determine λ . Once λ is determined, $\Delta\sigma$ within the element can be evaluated from (3.9) and (3.11).

In the present paper we are restricting our consideration to the plastic hinge concept with hinges limited to the end nodes only. In the finite element theory of beams, N and M_1 are constant throughout the element while M_2 and M_3 , in general, vary. In order that internal equilibrium always be maintained (i.e., N and M_1 remain constant throughout the element even if yielding occurs at both ends), the yield surface at node 2 must be modified to the form given by Eq. (3.19) if node 1 has already yielded.

$$\Phi = a_2 M_2^2 + a_3 M_3^2 - K = 0 \quad (3.19)$$

where the constant K is given by

$$K = K_0 - a_4 N^2 - a_1 M_1^2$$

The stress resultant, N , and the twisting moment M_1 are the values at node 1, and fall on the yield surface at node 1. The same remarks hold when node 2 yields first, in which case node 1 and node 2 interchange roles.

Now, when a hinge appears at a node, all plastic deformation is assumed to be concentrated at that node, and the rest of the elements still deforms elastically (i.e., the matrix in (3.2) still applies just outside the plastic zone). For the node which becomes a plastic hinge the moment increments and the stress resultant increments at such a hinge are determined according to Eqs. (3.8) - (3.12); then one can determine the incremental displacements and

rotations at the hinged node that are associated with such incremental moments and forces. For example, if node 1 yields and node 2 does not we can solve for $\Delta\theta_1$ at node one by making use of the three equations implied by the fourth through sixth rows of (3.2). The following expressions are easily derived

$$\begin{aligned}(\Delta\theta_1)_1 &= \ell (\Delta M_1)_1 / GJ + (\Delta\theta_1)_2 \\(\Delta\theta_2)_1 &= \frac{1}{c} \left\{ (\Delta M_2)_1 + b \left[(\Delta u_3)_2 - (\Delta u_3)_1 \right] + d (\Delta\theta_2)_2 \right\} \\(\Delta\theta_3)_1 &= \frac{1}{c'} \left\{ (\Delta M_3)_1 + b' \left[(\Delta u_2)_2 - (\Delta u_2)_1 \right] + d' (\Delta\theta_3)_2 \right\} \quad (3.20)\end{aligned}$$

where the subscript of a parenthesis denotes the node while the subscript of a quantity denotes direction or sense. The ΔM 's given in (3.20) are the incremental moments at node 1, which have been calculated using the flow rule as discussed in relation to Eqs. (3.11) through (3.18), so that plastic deformation has already been accounted for. Subsequently, the ΔM 's and ΔQ 's at node 2 should be recomputed utilizing the equations implied by the eight through twelfth rows of (3.2), where values of the rotations $\Delta\theta$, as computed in (3.20) for node 1 are used. It is clear, then, that the incremental changes in quantities at node 1 due to plastic hinge action will affect the quantities at node 2.

The total cumulative forces and moments are the sum of those existing at the last time step and those of the present increment. In these computations the forces and the moments are expressed in the local beam coordinate system. They must be transformed to global coordinates before they can be assembled properly. First, the matrix L which relates the global coordinates to local coordinates, as defined in Appendix A, must be updated using the results in Appendix A, namely

$$L_i = (L_i)_0 \begin{bmatrix} 1 & \Delta\phi_3 & -\Delta\phi_2 \\ -\Delta\phi_3 & 1 & \Delta\phi_1 \\ \Delta\phi_2 & -\Delta\phi_1 & 1 \end{bmatrix}_i \quad (i = 1, 2) \quad (3.21)$$

where $(L_i)_0$ is the value of L_i at the previous step, and the subscript i denotes node 1 or 2. The $\Delta\phi$'s are rotation increments with respect to global axes. (Note that the vector $\Delta\phi_i$ as defined previously is given by $\Delta\theta_i = (\Delta\theta_1)_i (e_1)_i + (\Delta\theta_2)_i (e_2)_i + (\Delta\theta_3)_i (e_3)_i = (\Delta\phi_1)_i \underline{i} + (\Delta\phi_2)_i \underline{j} + (\Delta\phi_3)_i \underline{k}$ where the local base vector $\underline{e}_1$, $\underline{e}_2$ and $\underline{e}_3$ are defined in Appendix A.)

If we let $\underline{f}$ denote the vector of cumulative values of the forces and moments for an element in the local coordinate system, then the corresponding values in the global coordinate system will be

$$\begin{bmatrix} L_1 & 0 & 0 & 0 \\ 0 & L_1 & 0 & 0 \\ 0 & 0 & L_2 & 0 \\ 0 & 0 & 0 & L_2 \end{bmatrix}^T \underline{f} \quad (3.22)$$

where L_1 and L_2 are the matrices defined by Eq. (3.1).

This section will now be completed with a discussion of the mass matrix. It is clear that a consistent element mass matrix can be derived in a straightforward manner by means of well known methods in finite element theory (Refs. 4-6). However, based on considerations involving questions of numerical instability (Ref. 2), we shall use a lumped element mass matrix herein, given by

$$\underline{m} = \begin{bmatrix} m_1 & 0 & 0 & 0 \\ 0 & m_2 & 0 & 0 \\ 0 & 0 & m_1 & 0 \\ 0 & 0 & 0 & m_2 \end{bmatrix} \quad (3.23)$$

where

$$\tilde{m}_1 = \rho \frac{\ell A}{2} \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad \tilde{m}_2 = \begin{bmatrix} \frac{\rho \ell A}{2} & 0 & 0 \\ 0 & \rho \frac{\ell^3 A}{12} & 0 \\ 0 & 0 & \rho \frac{\ell^3 A}{12} \end{bmatrix} \quad (3.24)$$

where ρ , ℓ and A are the density, length and cross-sectional area, respectively, of the uniform element. The above element mass matrix is associated with the local coordinate system, where the $\tilde{m}_2$ matrix which is concerned with rotations is based on local principal axes. When transformed to global coordinates,

$$m = \begin{bmatrix} \tilde{m}_1 & 0 & 0 & 0 \\ 0 & L_1^T \tilde{m}_2 L_1 & 0 & 0 \\ 0 & 0 & \tilde{m}_1 & 0 \\ 0 & 0 & 0 & L_2^T \tilde{m}_2 L_2 \end{bmatrix} \quad (3.25)$$

the part of the mass matrix due to rotary inertia is a function of time, (since the matrices L_1 and L_2 are functions of time). However, it is felt that the rotary inertia of the beam type structural elements will not affect greatly the dynamics of the system. In the numerical integration we shall treat the element mass matrix as a constant in the form of (3.25), after it is updated for each time step. This approach is correct for the translational inertia portion depicted by $\tilde{m}_1$. However, for the rotary inertia portion, depicted by $L_1^T \tilde{m}_2 L_1$ and $L_2^T \tilde{m}_2 L_2$ the changes from constant values are of the order $\dot{\theta}^2/\ddot{\theta}$ with respect to unity, which is deemed small in crash situations.

3.2 SPRING ELEMENTS

Spring elements can be regarded as special cases of the beam element with $I_2 = I_3 = 0$, and can be handled as such computationally. There are, however, certain special springs which arise naturally in vehicle crashworthiness studies, and so are worthy of

discussion here. In what follows some computational features of such springs will be developed.

(a) Ground spring - This is used to simulate a wheel supporting the vehicle on the ground. The force transmitted through a wheel is always assumed to be in the vertical direction and is either zero or in compression. Let us assume that the point where the wheel touches the ground is node 1 and the point where the wheel supports the vehicle is node 2. In local coordinates, with the X_1 -axis pointing from node 1 to node 2, and considering the order of components in the force vector given in (3.5), all the components in $\tilde{f}$ will be zero in this case except f_1 and f_7 which are

$$f_1 = \max \left[0, (f_1)_0 + k (\Delta q_1 - \Delta q_7) \right] = -f_7 \quad (3.26)$$

where $(f_1)_0$ is the value of f_1 at the previous time step, Δq_1 and Δq_7 are defined in (3.4) and k is a spring constant to simulate the wheel. It is usual to lump the mass of the wheel at node 2.

(b) Tension spring - This is used to simulate a cable restraint system such as a seat belt. Again using local coordinates the only non-zero force components are f_1 and f_7 which in this case are given by

$$f_1 = \min \left[0, (f_1)_0 + k (\Delta q_1 - \Delta q_7) \right] = -f_7 \quad (3.27)$$

with similar definitions for quantities as in (3.26).

3.3 RIGID BODY ELEMENTS

We may simulate the engine block, or a portion of the vehicle which does not involve significant deformation in a crash by a rigid body which is connected at a finite number of points to the deformable part of the vehicle. This rigid body will be referred to as a rigid body element.

The motion of a rigid body can be computed as follows. Consider a rigid body of mass m , and moments of inertia I_1 , I_2 and I_3

with respect to its principal axes. Let $\tilde{u}$ and $\tilde{\phi}$ denote the linear and angular displacement vectors, respectively, of its center of gravity with respect to global axes. Then the equations of motion can be written as

$$m \ddot{\tilde{u}} = \tilde{F} \quad (3.28)$$

$$\frac{d}{dt} \left\{ \tilde{L}^T \begin{bmatrix} I_1 & 0 & 0 \\ 0 & I_2 & 0 \\ 0 & 0 & I_3 \end{bmatrix} \tilde{L} \dot{\tilde{\phi}} \right\} = \tilde{M} \quad (3.29)$$

where, as shown in Appendix A

$$\dot{\tilde{L}} = \tilde{L} \begin{bmatrix} 0 & \dot{\phi}_3 & -\dot{\phi}_2 \\ -\dot{\phi}_3 & 0 & \dot{\phi}_1 \\ \dot{\phi}_2 & -\dot{\phi}_1 & 0 \end{bmatrix} \quad (3.30)$$

and where $\tilde{F}$ and $\tilde{M}$ are the total force and the total moment on the rigid body element; $\tilde{L}$ is the matrix relating the global cartesian coordinates to the principal axes, and is defined similar to that in Eq. (3.1).

If there are N_r nodes in common between the deformable structure and the rigid body, the total force and moment ($\tilde{F}$ and $\tilde{M}$) are obtained by summing over all N_r nodes the forces, $\tilde{f}_i$ and torques $\tilde{T}_i$ which act on the rigid body at the i th node, so that,

$$\begin{aligned} \tilde{F} &= \sum_{i=1}^{N_r} \tilde{f}_i \\ \tilde{M} &= \sum_{i=1}^{N_r} (\tilde{T}_i + \tilde{r}_i \times \tilde{f}_i) \end{aligned} \quad (3.31)$$

where $\tilde{r}_i$ is the position vector measured from the center of gravity to node i .

The central difference explicit scheme used earlier will also be used here to integrate (3.29) and (3.30), namely,

$$\Delta \underline{u}_{n+1} = \Delta \underline{u}_n + \frac{(\Delta t)^2}{m} \underline{F}_n \quad (3.32a)$$

$$\underline{L}_n^T \begin{bmatrix} I_1 & 0 & 0 \\ 0 & I_2 & 0 \\ 0 & 0 & I_3 \end{bmatrix} \underline{L}_n \Delta \phi_{n+1} = \underline{L}_{n-1}^T \begin{bmatrix} I_1 & 0 & 0 \\ 0 & I_2 & 0 \\ 0 & 0 & I_3 \end{bmatrix} \underline{L}_{n-1} \Delta \phi_n + (\Delta t)^2 \underline{M}_n \quad (3.32b)$$

$$\underline{L}_{n+1} = \underline{L}_n \begin{bmatrix} 1 & \Delta \phi_3 & -\Delta \phi_2 \\ -\Delta \phi_3 & 1 & \Delta \phi_1 \\ \Delta \phi_2 & -\Delta \phi_1 & 1 \end{bmatrix}_{n+1} \quad (3.32c)$$

Where the subscript n denotes the value of the quantity at the n th step, and $\Delta \underline{u}_{n+1} = \underline{u}_{n+1} - \underline{u}_n$. Now, the incremental rotations at all the N_r nodes on the rigid body are also $\Delta \phi_{n+1}$ and so the incremental displacements at node i are given by,

$$(\Delta \underline{u}_{n+1})_i = \Delta \underline{u}_{n+1} + \Delta \phi_{n+1} \times \underline{r}_i \quad (3.33)$$

in which $\underline{r}$ is the same as that in (3.31).

3.4 MODAL ELEMENT

During a crash, a component or a part of the vehicle may experience only a small elastic deformation. In such a case one could probably approximate this component as a rigid body. If, however, for one reason or another more detailed response information is desired within this component, it is possible to obtain adequate engineering accuracy by approximating the response using a few natural modes.

In what follows, the procedure of deriving the component matrix using natural modes is briefly described. Let the mass and

stiffness matrix of a given component be denoted by $\underline{m}_c$ and $\underline{k}_c$, respectively. The generalized coordinates associated with these matrices will be denoted by $\underline{q}_c$. Now, these matrices may be determined either by detailed finite element modelling of the component itself or by other well established methods. For complex components the size of such matrices can be very large. This, of course, will depend on how extensive the original modelling is. In such cases, to reduce $\underline{m}_c$ and $\underline{k}_c$ to what will be referred to as a modal form, we may either use assumed modes to approximate the response, or more rationally, derive the modes by solving the natural vibration eigenvalue problem associated with $\underline{m}_c$ and $\underline{k}_c$. The number of modes required, will, of course, depend on the degree of accuracy desired.

Consider the eigenvalue problem

$$\lambda \underline{m}_c \underline{x} = \underline{k}_c \underline{x} \quad (3.34)$$

We can easily determine the first few, say r eigenvalues, λ_j and eigenvectors, $\underline{x}_j$ ($j = 1, 2, \dots, r$) [Refs. 3]. The number r should be sufficiently large or at least equal to the number of rigid body degrees of motion. The generalized coordinate vector, $\underline{q}_c$, is then approximated by a combination of the eigenvectors or modes, namely

$$\underline{q}_c = \underline{X} \underline{a} \quad (3.35)$$

in which

$$\underline{X} = \{ \underline{x}_1, \underline{x}_2, \dots, \underline{x}_r \}$$

The quantity $\underline{a}$ can be called the vector of modal amplitudes [when using $\underline{a}$ as generalized coordinates, the corresponding mass and stiffness matrices can be made diagonal].

To insure compatibility between the component and the rest of the vehicle, it is not convenient to use $\underline{a}$ directly. We shall partition $\underline{q}_c$, $\underline{X}$ and $\underline{a}$ such that

$$\underline{q}_c \begin{Bmatrix} \underline{q}_\alpha \\ \underline{q}_\beta \end{Bmatrix} = \begin{bmatrix} \underline{X}_{\alpha\alpha} & \underline{X}_{\alpha\gamma} \\ \underline{X}_{\beta\alpha} & \underline{X}_{\beta\gamma} \end{bmatrix} \begin{Bmatrix} \underline{a}_\alpha \\ \underline{a}_\gamma \end{Bmatrix} \quad (3.36)$$

where the components of $\underline{q}_\alpha$ and $\underline{q}_\beta$ are the generalized coordinates associated with nodes connected and not connected respectively to the rest of the vehicle. By expressing $\underline{a}_\alpha$ in terms of $\underline{q}_\alpha$ we have

$$\underline{a} = \begin{Bmatrix} \underline{a}_\alpha \\ \underline{a}_\gamma \end{Bmatrix} = \begin{bmatrix} \underline{X}_{\alpha\alpha}^{-1} & -\underline{X}_{\alpha\alpha}^{-1} \underline{X}_{\alpha\gamma} \\ 0 & \underline{I} \end{bmatrix} \begin{Bmatrix} \underline{q}_\alpha \\ \underline{a}_\gamma \end{Bmatrix} = \underline{A} \begin{Bmatrix} \underline{a}_\alpha \\ \underline{a}_\gamma \end{Bmatrix} \quad (3.37)$$

(note that $\underline{X}$ may have to be reordered to assure that $\underline{X}_{\alpha\alpha}$ is nonsingular). The vector $\underline{q}_\alpha$ and $\underline{a}_\gamma$ can be used as the new generalized coordinates. Accordingly, the associated reduced mass and stiffness matrices are then given by

$$\underline{m}_r = \underline{A}^T \underline{A} \quad (3.38)$$

$$\underline{k}_r = \underline{A}^T \underline{D} \underline{A}$$

where $\underline{D}$ is a diagonal matrix of order r with the elements on the diagonal being the first r eigenvalues λ_j ($j = 1, 2, \dots, r$) of (3.34). It can be seen that the order of $\underline{m}_r$ and $\underline{k}_r$ can be much less than that of $\underline{m}_c$ and $\underline{k}_c$; therefore, the total number of degrees of freedom for the final system of equations is reduced.

4. CONCLUSIONS

The basic concept of a modular approach and the procedure of numerical integration for crashworthiness prediction has been developed in the present report. In addition, several elements have been examined which are useful for this purpose. One of the important advantages of using the modular approach in vehicle structure simulation is that it provides great flexibility in modelling specific dynamic behavior economically. Depending on the degree of accuracy desired, one may use simplified spring-mass, or detailed finite element modelling for part or the entire structure.

A computer program based on the modular concept has been developed at TSC. Many examples are being run, and extensive numerical results will be reported in a separate report.

APPENDIX

EVALUATION OF $\underline{L}$ IN EQ. (3.1) AT TIME = 0

The matrix $\underline{L}$ relates the global coordinates to local coordinates. It can be defined either in terms of Euler angles or in terms of the position vectors of three noncollinear points in space. The former approach requires less information, while the latter is easier to visualize. In what follows, $\underline{L}$ shall be defined using the latter approach. Let (x_{i1}, x_{i2}, x_{i3}) be the coordinates of point i , ($i = 1, 2, 3$). The local coordinate frame will have the first base vector $\underline{e}_1$, pointing from point 1 to point 2; the second base vector, $\underline{e}_2$, lies in the plane spanned by the three points and is normal to the first base vector. The third base vector, $\underline{e}_3$, is normal to base vectors $\underline{e}_1$ and $\underline{e}_2$, and satisfies the right hand rule. Therefore, from these definitions, we have

$$\underline{e}_1 = \frac{1}{\ell_1} \begin{Bmatrix} x_{21} - x_{11} \\ x_{22} - x_{12} \\ x_{23} - x_{13} \end{Bmatrix} \quad (\text{A.1})$$

$$\underline{E}_2 = \frac{1}{\ell_2} \begin{Bmatrix} x_{31} - x_{11} \\ x_{32} - x_{12} \\ x_{33} - x_{13} \end{Bmatrix} \quad (\text{A.2})$$

$$\underline{e}_2 = (\underline{e}_1 \times \underline{E}_2) \times \underline{e}_1 = \underline{E}_2 - (\underline{E}_2 \cdot \underline{e}_1) \underline{e}_1 \quad (\text{A.3})$$

$$\underline{e}_3 = \underline{e}_1 \times \underline{e}_2 \quad (\text{A.4})$$

in which

$$\ell_1 = \left[(x_{21} - x_{11})^2 + (x_{22} - x_{12})^2 + (x_{23} - x_{13})^2 \right]^{1/2}$$

and

$$l_2 = \left[(x_{31} - x_{11})^2 + (x_{32} - x_{12})^2 + (x_{33} - x_{13})^2 \right]^{1/2}$$

Then

$$\tilde{L} = \begin{bmatrix} \tilde{e}_1^T \\ \tilde{e}_2^T \\ \tilde{e}_3^T \end{bmatrix} \quad (A.5)$$

i.e., the i th row of $\tilde{L}$ is the i th base vector of the local frame having its components expressed in terms of the global coordinates.

For a beam element, the three noncollinear points are taken as the two end nodes which are chosen as the first two points; a third point is a point in space to indicate the direction of one of the principal axes of bending.

Evaluation of $\Delta\tilde{L}$, and the updating of $\tilde{L}$

The matrix $\tilde{L}$ is

$$\tilde{L} = \begin{bmatrix} \tilde{e}_1^T \\ \tilde{e}_2^T \\ \tilde{e}_3^T \end{bmatrix} = \begin{bmatrix} L_{11} & L_{12} & L_{13} \\ L_{21} & L_{22} & L_{23} \\ L_{31} & L_{32} & L_{33} \end{bmatrix} \quad (A.6)$$

To determine $\dot{\tilde{L}}$, find $\dot{\tilde{e}}_i$ ($i = 1, 2, 3$) where the base vectors $\tilde{e}_i$ are defined in the discussion leading to (A.5). The following relations yield the desired results.

$$\dot{\tilde{e}}_i = \tilde{\omega} \times \tilde{e}_i$$

where

$$\tilde{\omega} = \omega_1 \tilde{i} + \omega_2 \tilde{j} + \omega_3 \tilde{k} \quad (A.7)$$

Then,

$$\dot{\tilde{e}}_i = \begin{bmatrix} i & j & k \\ \omega_1 & \omega_2 & \omega_3 \\ L_{i1} & L_{i2} & L_{i3} \end{bmatrix} = \begin{pmatrix} \omega_2 L_{i3} - \omega_3 L_{i2} \\ \omega_3 L_{i1} - \omega_1 L_{i3} \\ \omega_1 L_{i2} - \omega_2 L_{i1} \end{pmatrix} \begin{matrix} i \\ j \\ k \end{matrix}$$

So that

$$\dot{\tilde{L}} = \begin{bmatrix} \omega_2 L_{13} - \omega_3 L_{12} & \omega_3 L_{11} - \omega_1 L_{13} & \omega_1 L_{12} - \omega_2 L_{11} \\ \omega_2 L_{23} - \omega_3 L_{22} & \omega_1 L_{21} - \omega_1 L_{23} & \omega_1 L_{22} - \omega_2 L_{22} \\ \omega_2 L_{33} - \omega_3 L_{32} & \omega_3 L_{31} - \omega_1 L_{33} & \omega_1 L_{32} - \omega_2 L_{31} \end{bmatrix}$$

or

$$\dot{\tilde{L}} = \begin{bmatrix} L_{11} & L_{12} & L_{13} \\ L_{21} & L_{22} & L_{23} \\ L_{31} & L_{32} & L_{33} \end{bmatrix} \begin{bmatrix} 0 & \omega_3 & -\omega_2 \\ -\omega_3 & 0 & \omega_1 \\ \omega_2 & -\omega_1 & 0 \end{bmatrix} = \tilde{L} \begin{bmatrix} 0 & \dot{\phi}_3 & -\dot{\phi}_2 \\ -\dot{\phi}_3 & 0 & \dot{\phi}_1 \\ \dot{\phi}_2 & -\dot{\phi}_1 & 0 \end{bmatrix} \quad (A.8)$$

where

$$\dot{\phi}_i = \omega_i$$

Equation (A.8) implies

$$\Delta \tilde{L} = \tilde{L} \begin{bmatrix} 0 & \Delta \phi_3 & -\Delta \phi_2 \\ -\Delta \phi_3 & 0 & \Delta \phi_1 \\ \Delta \phi_2 & -\Delta \phi_1 & 0 \end{bmatrix}$$

Then $\tilde{L}$ at the $n+1$ time step is given in terms of its value, $\tilde{L}_n$, at the n th time step as

$$\tilde{L}_{n+1} = \tilde{L}_n + \Delta \tilde{L}_n = \tilde{L}_n + \tilde{L}_n \begin{bmatrix} 0 & \Delta\phi_3 & -\Delta\phi_2 \\ -\Delta\phi_3 & 0 & \Delta\phi_1 \\ \Delta\phi_2 & -\Delta\phi_1 & 0 \end{bmatrix}$$

or

$$\tilde{L}_{n+1} = \tilde{L}_n \begin{bmatrix} 1 & \Delta\phi_3 & -\Delta\phi_2 \\ -\Delta\phi_3 & 1 & \Delta\phi_1 \\ \Delta\phi_2 & -\Delta\phi_1 & 1 \end{bmatrix} \quad (\text{A.9})$$

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TEST
REPORT

DOT #261

CONTRACT NAME: DUMMY RESPONSE IN SIDE IMPACT

CONTRACT NUMBER: DOT-HS-8-01941, TASK ORDER NO. 2

TEST DESCRIPTION: MALIBU INTO VOLKSWAGEN; TEST #3
OF A THREE TEST SERIES

TEST NUMBER: 1960

DATE: FEBRUARY 3, 1981

MUNICARS, INC.

TEST REPORT

CONTRACT NAME: Dummy Response in Side Impact

CONTRACT NUMBER: DOT-HS-8-01941, Task Order No. 2

TEST DESCRIPTION: Malibu into VW; Test #3
of a three test series

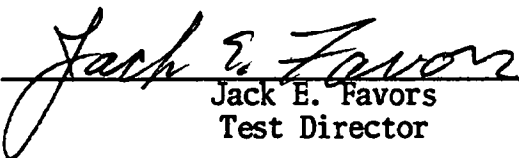
TEST NUMBER: 1960

DATE: February 6, 1981

PREPARED BY:


James R. Artell
Project Engineer

APPROVED:


Jack E. Favors
Test Director

APPROVED:

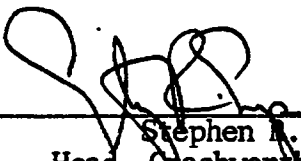

Stephen A. Syson
Head, Crashworthiness Group

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ABBREVIATIONS		
N/A - Not Available		
DNA - Data Not Available		
DNR - Data Not Required		

INTRODUCTION

GENERAL OBJECTIVES

Under Contract DOT-HS-8-01941, Task Order No. 2, Minicars is conducting six side impact crash tests. These tests are broken into two series of three tests each. The objective of each three-test series is to reenact, as closely as possible, a representative real world accident in which one vehicle was impacted on its side. The mechanical response of the new NHTSA side impact dummy, in the impacted vehicle, is recorded and analyzed for comparison to the injuries of the actual accident victim.

Vehicle response is also recorded for both vehicles, in order to more closely duplicate actual crash conditions in subsequent tests.

SECTION 1

TEST OBJECTIVES AND SPECIAL PROCEDURES

OBJECTIVES:

To conduct a side impact car-to-car crash with a newly designed NHTSA side impact dummy seated in the laterally impacted target car. Data from this test will be compared to injury severities of actual accident victims.

SPECIAL PROCEDURES:

A Chevrolet Malibu impacts the driver's side of a Volkswagen sedan. The Malibu's forward velocity is 17.25 ± 0.5 mph. The Volkswagen's forward velocity is 34.5 ± 0.5 mph. Speeds and impact conditions have been determined by a series of SMAC runs and results of previous crash tests.

SECTION 2

TEST CONDITIONS

SUMMARY OF CRASH TEST CONDITIONS

Test No.: 1960 Project Title: Dummy Response in Side Impact
 Date: 12-11-80 Time: 1:00 p.m. Temp: 70°F

Subject Vehicle Data

Description: 1972 Chevrolet Malibu Classic

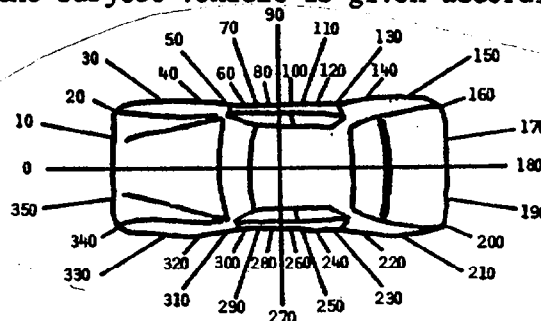
	Actual	Intended
Test weight (lbs)	3870	3688 + 50
Vehicle orientation (degrees)*	0	0
Vehicle velocity (mph)	16.8	17.25
Maximum crush (inches)		
Static	3.25	5
Dynamic	-	6
Maximum intrusion (inches)	0	0

Dummies

	Driver	Middle Passenger	Rt. Front Passenger	Left Rear Passenger	Rt. Rear Passenger
Type	P. 572		P. 572		
	50 M		50 M		
Serial no.	M6-3		M8-3		
Instrumentation					
Head accel.	x		x		
Chest accel.	x		x		
Femur L.C.'s	x		x		
Other					
Restraint system	3 point Belts		3 point Belts		

Remarks:

*The orientation of the subject vehicle is given according to the following diagram:



SUMMARY OF CRASH TEST CONDITIONS (cont'd)

Test No.: 1960 Project Title: Dummy Response in Side Impact

Date: 12-11-80 Time: 1:00 p.m. Temp: 70°F

Impacted Vehicle/Object Data

Description: Volkswagen Sedan

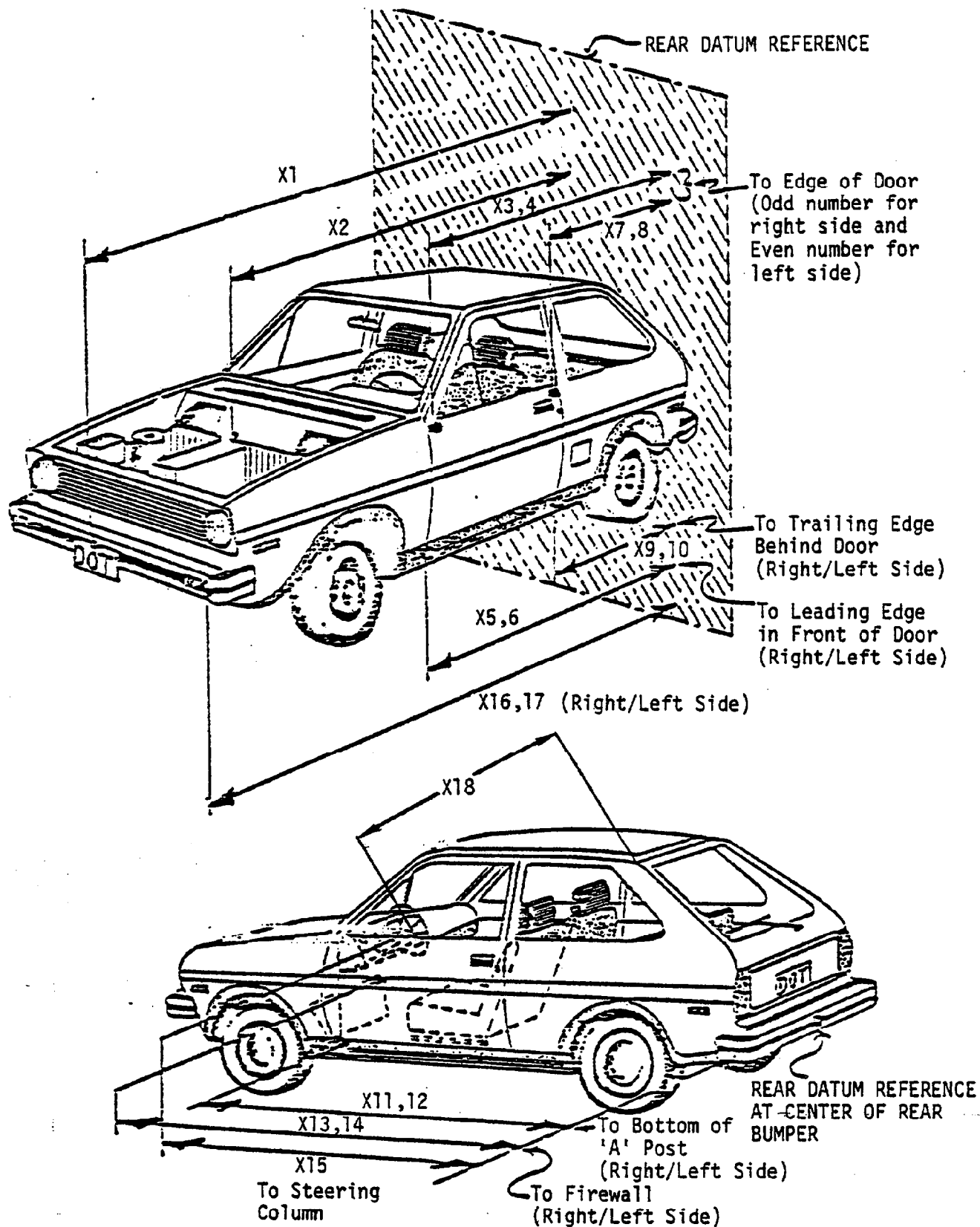
	<u>Actual</u>	<u>Intended</u>
Test weight (lbs)	2100	2085 + 50
Vehicle orientation (degrees)	270	270
Vehicle velocity (mph)	33.0	34.5
Maximum crush (inches)		
Static	11.75	10
Dynamic	-	12

Dummies

	<u>Driver</u>	<u>Middle Passenger</u>	<u>Rt. Front Passenger</u>	<u>Left Rear Passenger</u>	<u>Rt. Rear Passenger</u>
Type	NHTSA				
Serial no.	T1-1				
Instrumentation					
Head accel.	x				
Chest accel.	x				
Femur L.C.'s					
Other - Pelvic Gs	x*				
Restraint system	None				

Remarks: *See page 17 for a summary of the dummy instrumentation.

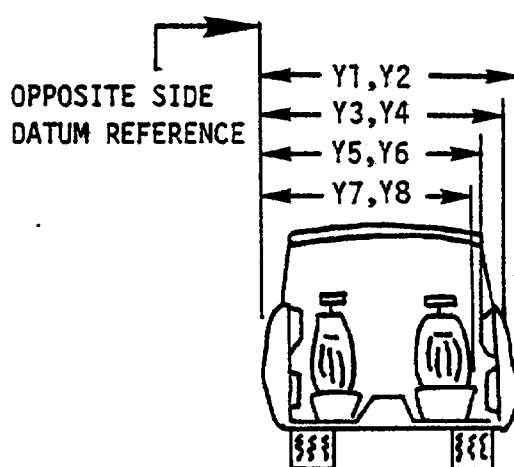
PRE-TEST AND POST-TEST MEASUREMENT POINTS - FRONTALLY IMPACTED VEHICLES



VEHICLE MEASUREMENTS - Chevrolet Malibu

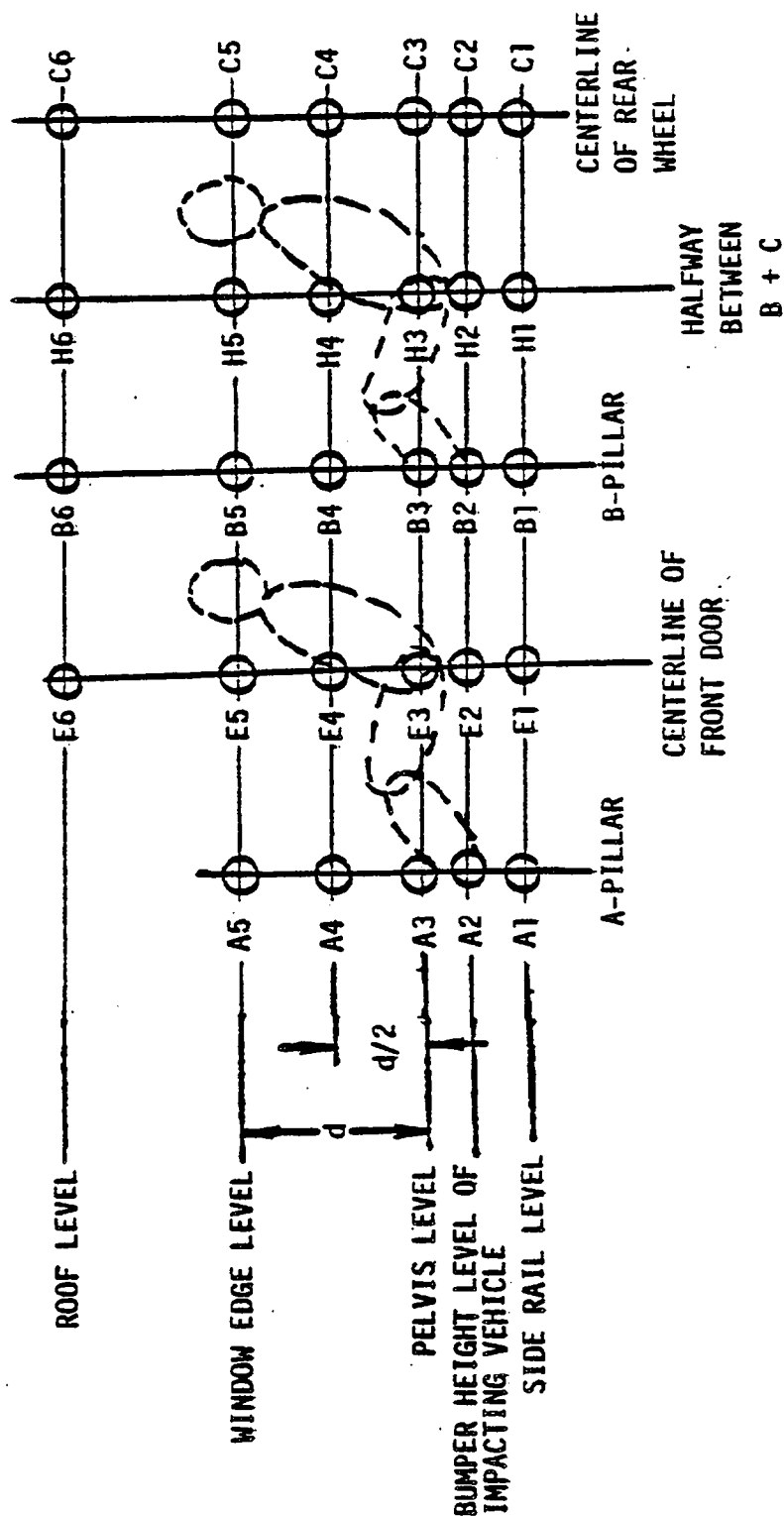
No.	Type of Measurement	Dimensions (inches)	
		Pre-Test	Post-Test
X1	Total Length of Vehicle at Centerline	197.25	194.0
X2	Rear Surface of Vehicle to Firewall	132.5	132.5
X3	Rear Surface of Vehicle to Upper Leading Edge of Right Door	127.1	127.1
X4	Rear Surface of Vehicle to Upper Leading Edge of Left Door	127.25	127.4
X5	Rear Surface of Vehicle to Lower Leading Edge of Right Door	129.5	129.25
X6	Rear Surface of Vehicle to Lower Leading Edge of Left Door	129.9	129.75
X7	Rear Surface of Vehicle to Upper Trailing Edge of Right Door	81.25	81.75
X8	Rear Surface of Vehicle to Upper Trailing Edge of Left Door	81.5	81.5
X9	Rear Surface of Vehicle to Lower Trailing Edge of Right Door	79.6	79.5
X10	Rear Surface of Vehicle to Lower Trailing Edge of Left Door	80.0	79.75
X11	Rear Surface of Vehicle to Bottom of A-Post of Right Side	127.0	126.75
X12	Rear Surface of Vehicle to Bottom of A-Post of Left Side	127.4	127.4
X13	Rear Surface of Vehicle to Firewall - Right Side	141.5	141.0
X14	Rear Surface of Vehicle to Firewall - Left Side	142.3	142.2
X15	Rear Surface of Vehicle to Steering Wheel Center	117.4	117.5
X16	Rear Surface of Vehicle to Right Side of Front Bumper	190.5	191.5
X17	Rear Surface of Vehicle to Left Side of Front Bumper	190.75	188.25
X18	Rear Window Header to Steering Wheel Center	54.0	53.75

PRE-TEST AND POST-TEST MEASUREMENT POINTS -
LATERAL IMPACT VEHICLE - Volkswagen Sedan



No.	Measurement	Value (inches)	
		Pre-	Post-
Y1	Exterior Width at Front Dummy's H-Pt Location	54.8	45.15
Y2	Exterior Width at Rear Dummy's H-Pt Location	55.5	55.3
Y3	Exterior Width at Front Dummy's Shoulder-Pt Location	47.5	45.3
Y4	Exterior Width at Rear Dummy's Shoulder-Pt Location	49.5	48.25
Y5	Interior Width at Front Dummy's H-Pt Location	48.1	41.75
Y6	Interior Width at Rear Dummy's H-Pt Location	N/A	N/A
Y7	Interior Width at Front Dummy's Shoulder-Pt Location	46.9	41.1
Y8	Interior Width at Rear Dummy's Shoulder-Pt Location	N/A	N/A

MAXIMUM OCCUPANT COMPARTMENT INTRUSION WILL BE MEASURED RELATIVE TO AN UNDEFERRED VEHICLE REFERENCE LINE. EXTERIOR INTRUSION SHALL BE MEASURED, AND CONTOUR MAPS OF THE IMPACTED AND IMPACTING VEHICLES WILL BE PREPARED.

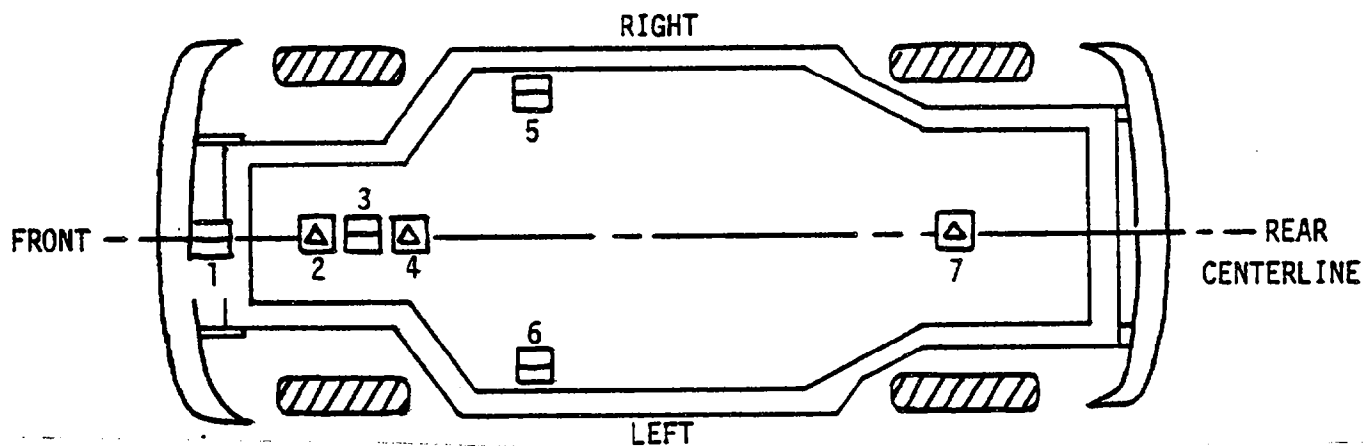


○ LOCATIONS AT WHICH INTRUSION MEASUREMENTS ARE MADE

INTRUSION MEASUREMENT
Volkswagen Sedan

Location (See Figure 1)	Pre-Test (inches)	Post-Test (inches)	Crush (inches)
A1	3.5	8.5	5.0
A2	6.6	11.0	4.4
A3	6.5	12.5	6.0
A4	6.6	10.75	4.15
A5	8.0	9.5	1.5
B1	3.0	6.25	3.25
B2	4.0	15.75	11.75
B3	3.75	15.25	11.50
B4	4.0	14.5	10.5
B5	5.75	12.0	6.25
B6	9.4	11.25	1.85
C1	1.5	2.25	0.75
C2	3.0	4.0	1.0
C3	N/A	N/A	N/A
C4	7.0	3.50	3.5
C5	8.75	6.75	2.0
C6	N/A	N/A	N/A
E1	3.25	7.75	4.5
E2	4.9	12.25	7.35
E3	4.6	14.25	9.65
E4	4.9	13.75	8.85
E5	8.25	10.5	2.25
E6	10.1	10.5	0.4
H1	2.0	4.0	2.0
H2	4.4	5.6	1.2
H3	4.25	4.5	0.25
H4	4.5	5.5	1.0
H5	7.25	8.5	1.25
H6	9.6	9.8	1.2

VEHICLE: BULLET VEHICLE - Chevrolet Malibu



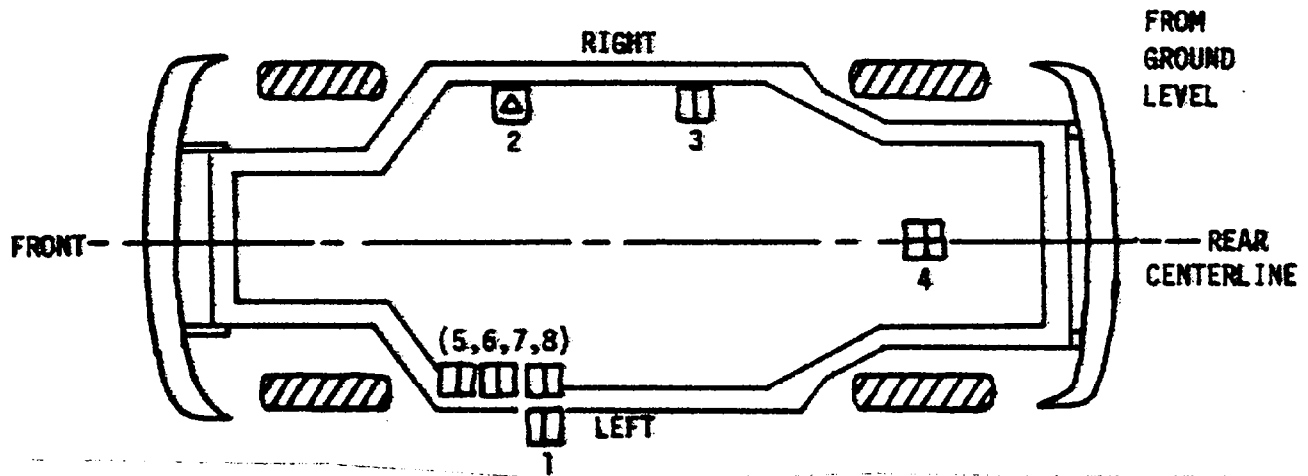
Linear Accelerometers

-  Longitudinal
-  Lateral
-  Vertical
-  Triaxial

Vehicle Transducer Locations

No.	Description of Location	X	Y	Z
1	Behind Front Bumper	x	-	-
2	Engine	x	x	x
3	Front Suspension Right Side (Unsprung Mass)	x	-	-
4	Firewall on Vehicle Centerline	x	x	x
5	Rocker Panel Inside Car	x	-	-
6	Rocker Panel Inside Car	x	-	-
7	Vehicle Centerline, Trunk Floor Right Behind Rear Seat	x	x	x

VEHICLE: TARGET VEHICLE - Volkswagen Sedan



Linear
Accelerometers

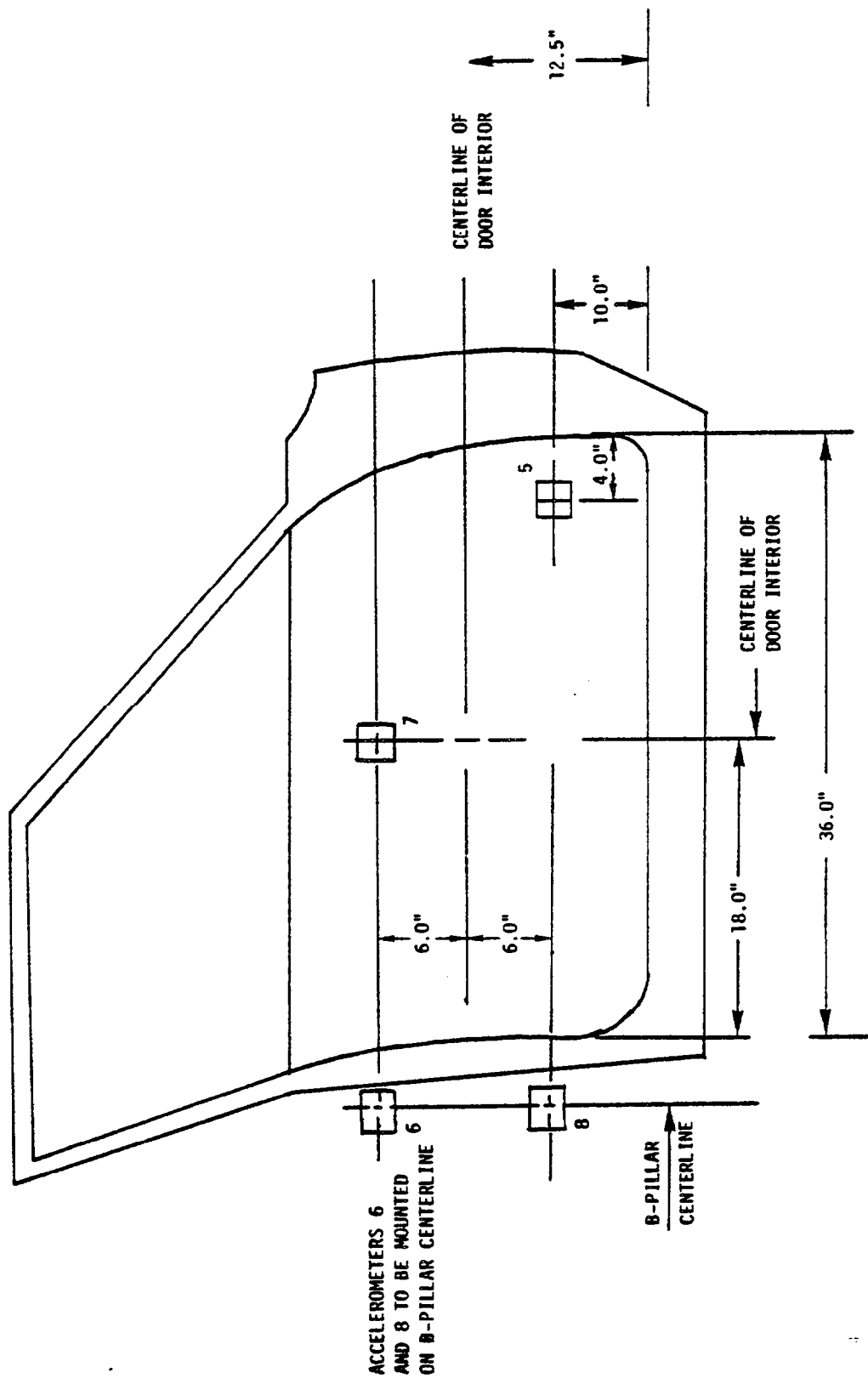
-  Longitudinal
-  Lateral
-  Triaxial

Vehicle Transducer Locations

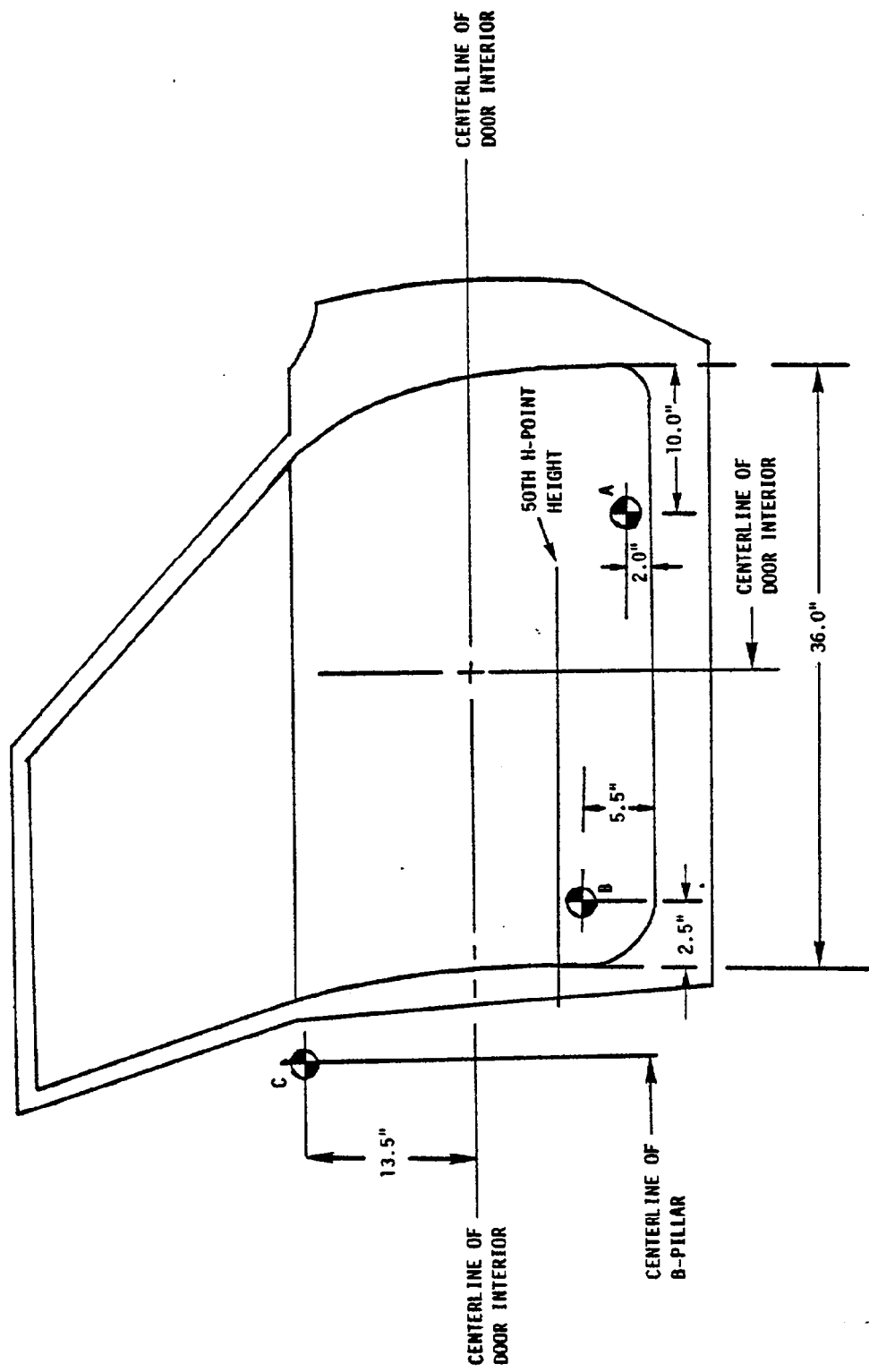
No.	Description of Location	X	Y	Z
Accelerometers				
1	14 Inches Forward of B-pillar Mounted on Rocker Panel (inside)		x	
2	Inside on Rocker Panel Front Door Area	x	x	x
3	Inside on Rocker Panel Front Door Area		x	
4	Trunk Floor on Vehicle Centerline Directly Behind Rear Seat	x	x	
5,6 7,8	See Door Accelerometer Locations		x	

Displacement (String Potentiometer) - See String Potentiometer Locations

A	Lower Forward Location on Left Inner Door
B	Rear Left Inner Door
C	Across B-pillars



DOOR ACCELEROMETER LOCATIONS: Volkswagen Sedan



STRING POTENTIOMETER LOCATIONS: Volkswagen Sedan

INSTRUMENTATION SUMMARY

VEHICLE: TARGET VEHICLE - Volkswagen Sedan

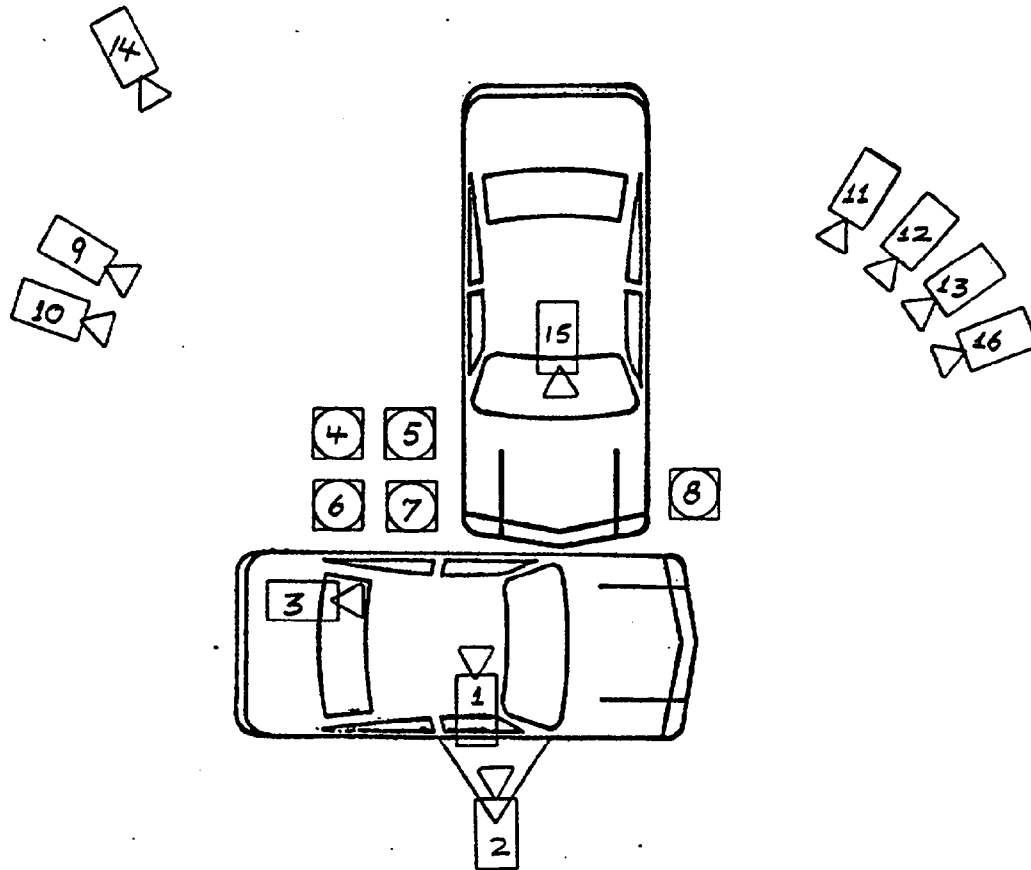
Instrument	Number	Total Number of Channels
Vehicle Accelerometers	8 Locations	11
Dummy*	1 Location	20
Load Cells	None	0
Rate Sensors	1 Location	3
String Potentiometers	3 Locations	3
		Total = 37

VEHICLE: BULLET VEHICLE - Chevrolet Malibu

Instrument	Number	Total Number of Channels
Vehicle Accelerometers	7 Locations	13
Dummies	2 Locations	16
Load Cells	None	0
Rate Sensors	1 Location	3
Strain Gauges	None	0
String Potentiometers	None	0
		Total = 32

*Head Acceleration: 3 Channels (Triaxial)
 Upper Spine Acceleration: 3 Channels (Triaxial)
 Lower Spine Acceleration: 3 Channels (Triaxial)
 Left Ribs Acceleration: 2 Channels
 Right Ribs Acceleration: 2 Channels
 Pelvis Acceleration: 3 Channels (Triaxial)
 Upper Sternal Acceleration: 1 Channel
 Lower Sternal Acceleration: 1 Channel
 Lateral Chest Deflection: 2 Channels

CRASH CONFIGURATION AND CAMERA LOCATIONS



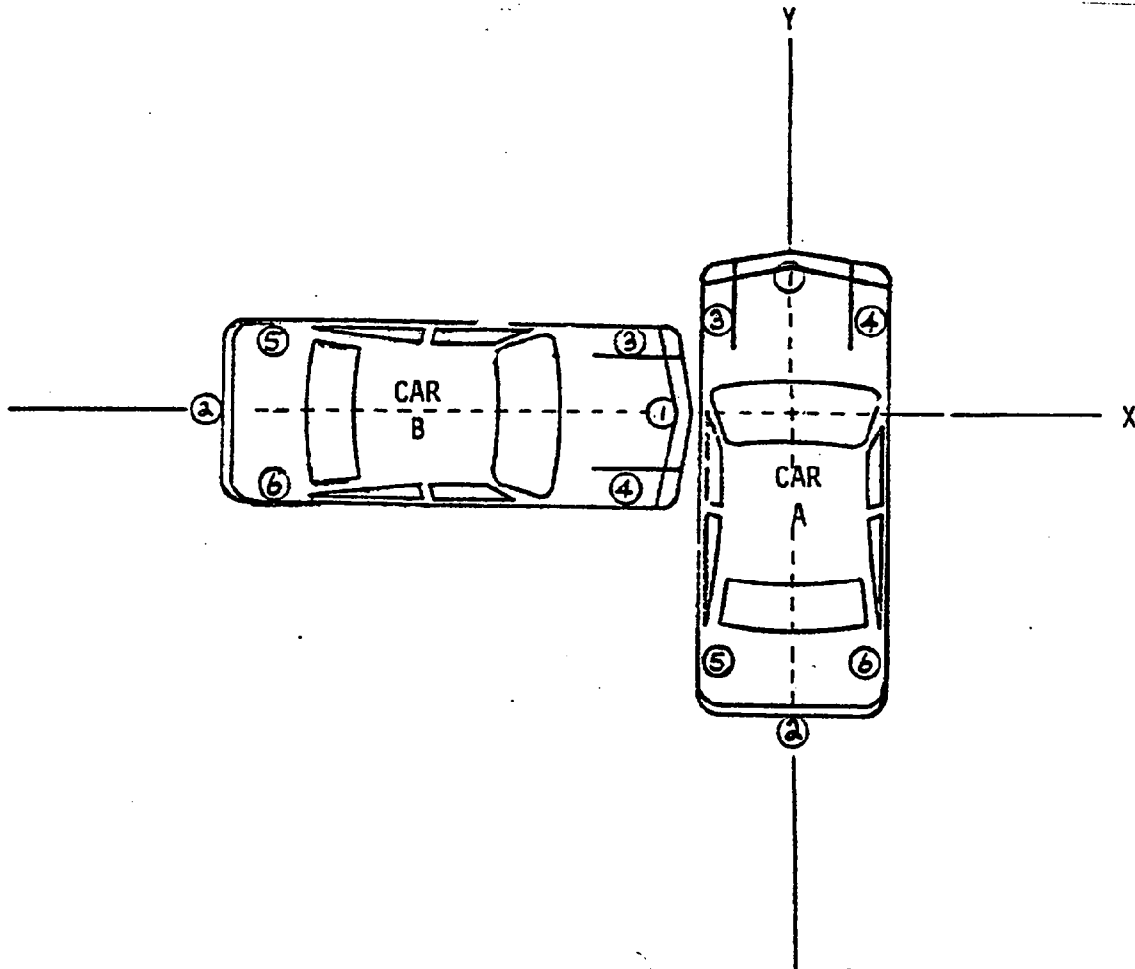
Camera numbers are keyed to Camera Information table.

CAMERA INFORMATION

Camera Number	Location	Type	Lens (mm)	Speed (fps)	Purpose of Camera Data
1	Volkswagen - On-board	Photosonic 1B	8mm	1000	Views dummy through hole cut in roof
2	Volkswagen - On-board	Photosonic 1B	8mm	1000	Dummy kinematics through passenger window
3	Volkswagen - On-board	Photosonic 1B	8mm	1000	View along interior sidewall
4	Tower mounted overhead	Photosonic 1B	13mm	1000	Vehicle speed and crush at impact zone
5	Tower mounted overhead	Photosonic 1B	8mm	1000	Wide angle view
6	Tower mounted overhead	Photosonic 1B	5.3mm	1000	Overall view of both cars
7	Tower mounted overhead	Sequence Polaroid	N/A	N/A	Determine impact point
8	Tower mounted oblique	Photosonic 1B	13mm	1000	Oblique view of impact
9	Offboard	Photosonic 1B	13mm	1000	Close-up of impact zone
10	Offboard	Nikon FM	50mm	Motor-drive	35mm sequence of event
11	Offboard	Photosonic 1B	13mm	1000	Close-up of impact zone
12	Offboard	Nikon FM	35-105 zoom	Motor-drive	35mm sequence of event
13	Offboard	Bronica ETR	75mm	N/A	Peak event still
14	Offboard	Arriflex 16mm	10-120 zoom	24	Documentary
15	Malibu - On-board	GSAP-16mm	17mm	32	Documentary of impact
16	Offboard	Sequence Polaroid	N/A	N/A	Determine impact point

EVENT CONFIGURATION

Test Number: 1960
Car A: Volkswagen Sedan
Car B: Chevrolet Malibu



- 1 Front Bumper Centerline
- 2 Rear Bumper Centerline
- 3 Left Front Wheel Centerline
- 4 Right Front Wheel Centerline
- 5 Left Rear Wheel Centerline
- 6 Right Rear Wheel Centerline

Car A Velocity	<u>33.0 mph</u>
Car B Velocity	<u>16.8 mph</u>
Angle of Impact	<u>270 degrees</u>
Intended Offset	<u>2.75 inches</u>
Actual Offset	<u>-4.25 inches</u>
Offset measured from Point 3 on Car A to Point 1 on Car B.	

SUMMARY OF TEST CONDITIONS

ST VEHICLE INFORMATION:

Vehicle Manufacturer: General Motors
 Make/Model: Chevrolet Malibu Classic
 Body Style: 2-door Coupe Model Year: 1972
 VIN: 37H2L532958 Build Date: 12/71
 NHTSA No.: None Color: Gold
 Engine Data: 8 cylinders; 350 in³ / cc displacement
 Transmission Data: 3 speed () Manual / (X) Automatic
 Date Vehicle Received by Laboratory: DNA
 Dealer's Name & Address: GFE

Data from Certification Label on Left Door Rear Face or 'B' Post:

Vehicle Manufactured by: General Motors
 Date of Manufacture: 12/71 VIN: 37H2L532958
 GVWR: 4780 lbs.; GAWR: Front = 2370 lbs.; Rear = 2410 lbs.

Data from "Recommended Tire Pressure" Label on Door, Post, Glove Box, etc.:

Vehicle Load: Front Rear
 Up to Capacity 26 psi 30 psi

Recommended Tire Size:

E78 x 14

Load Range:

B C D

Vehicle Capacity:

Type of Seats - ☐ Bench
☐ Bucket
☒ Split Bench

Number of Occupants = 3 Front
 (Designated Seating Capacity) 3 Rear
6 Total

Cargo Load = 1100 lbs.Total = 1100 lbs.

Weight of Test Vehicle as Received from Dealer (with maximum fluids):

Right Side = 1880 lbs.Left Side = 1860 lbs.Total Front Weight = 2150 lbs. (57 % of Total Vehicle Weight)Total Rear Weight = 1590 lbs. (43 % of Total Vehicle Weight)Total Delv. Weight = 3740 lbs.Weight of Test Vehicle with Required Dummies and -0- lbs. Cargo:Right Side = 1920 lbs.Left Side = 1950 lbs.Total Front Weight = 2240 lbs. (58 % of Total Vehicle Weight)Total Rear Weight = 1630 lbs. (42 % of Total Vehicle Weight)Total Test Weight = 3870 lbs.Weight of Ballast Secured in Vehicle Trunk Area = 000 lbs.

ST CONDITIONS:

Date of Test: 12-11-80 Time of Test: 1:00 am / pm
 Ambient Temperature: 70 °F at Impact Area

SUMMARY OF TEST CONDITIONS

TEST VEHICLE INFORMATION:

Vehicle Manufacturer: Volkswagen
 Make/Model: Volkswagen
 Body Style: 2-door sedan Model Year: 1971
 VIN: 1112916245 Build Date: 5/71
 NHTSA No.: None Color: Yellow
 Engine Data: 4 cylinders; in³ / cc displacement
 Transmission Data: 4 speed (X) Manual / () Automatic
 Date Vehicle Received by Laboratory: DNA
 Dealer's Name & Address: GFE

Data from Certification Label on Left Door Rear Face or 'B' Post:

Vehicle Manufactured by: Volkswagen
 Date of Manufacture: 5/71 VIN: 1112916245
 GVWR: DNA lbs.; GAWR: Front = DNA lbs.; Rear = DNA lbs.

Data from "Recommended Tire Pressure" Label on Door, Post, Glove Box, etc.:

Vehicle Load: Front 17 psi Rear 26 psi
 Up to Capacity

Recommended Tire Size:

5.60-15

Load Range:

B C D

Vehicle Capacity:

Type of Seats - ☐ Bench
☒ Bucket
☐ Split Bench

Number of Occupants = 2 Front
 (Designated Seating Capacity) 2 Rear
4 Total

Cargo Load = 830 lbs.
 Total = 830 lbs.

Weight of Test Vehicle as Received from Dealer (with maximum fluids):

Right Side = 1050 lbs. Right Rear = lbs.
 Left Side = 1050 lbs. Left Rear = lbs.
 Total Front Weight = 840 lbs. (40 % of Total Vehicle Weight)
 Total Rear Weight = 1260 lbs. (60 % of Total Vehicle Weight)
 Total Delv. Weight = 2100 lbs.

Weight of Test Vehicle with Required Dummies and -0- lbs. Cargo:

Right Side = 1010 lbs. Right Rear = lbs.
 Left Side = 1090 lbs. Left Rear = lbs.
 Total Front Weight = 790 lbs. (38 % of Total Vehicle Weight)
 Total Rear Weight = 1310 lbs. (62 % of Total Vehicle Weight)
 Total Test Weight = 2100 lbs.

Weight of Ballast Secured in Vehicle Trunk Area = 000 lbs.

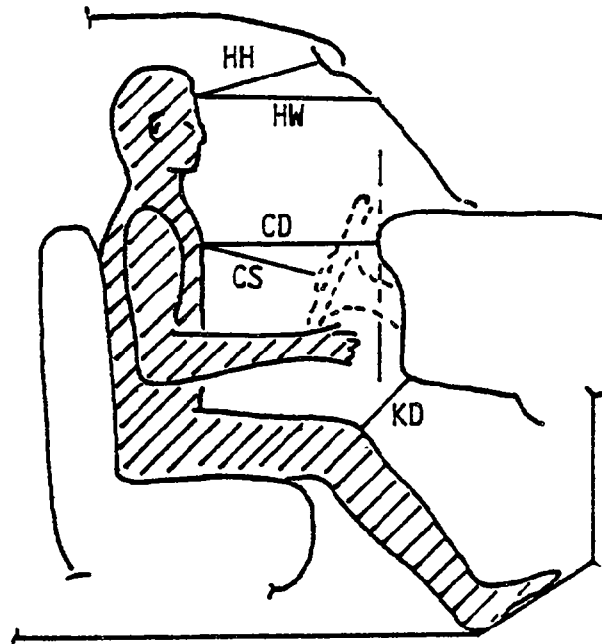
TEST CONDITIONS:

Date of Test: 12-11-80 Time of Test: 1:00 am / pm
 Ambient Temperature: 70 °F at Impact Area

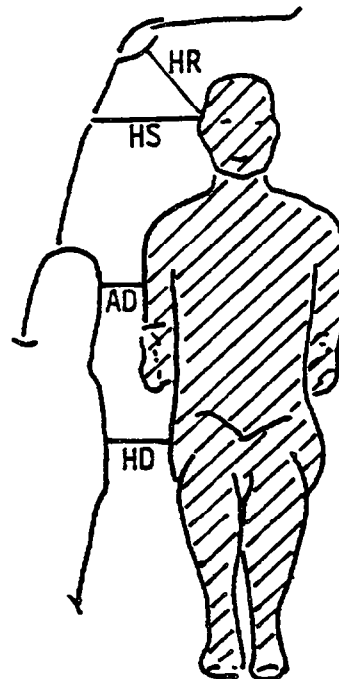
OCCUPANT CLEARANCE DIMENSIONS

Chevrolet Malibu

	Driver	Passenger
HH	17.0	17.0
HW	20.0	18.25
CD	23.75	20.5
CS	15.25	N/A
KDL	6.5	8.5
KDR	6.5	8.75



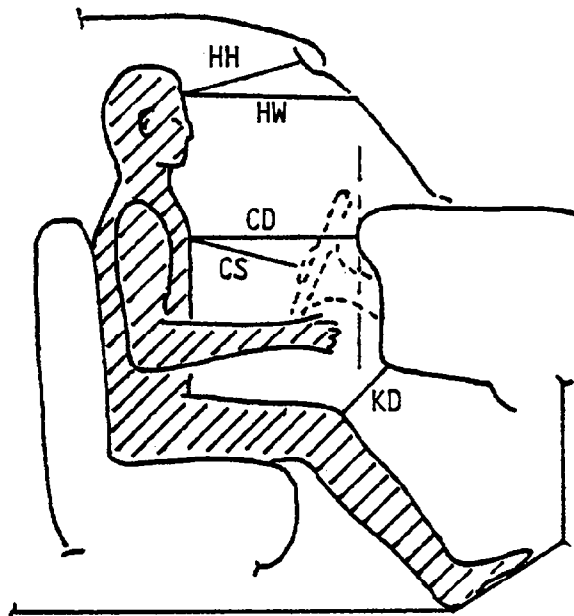
	Driver	Passenger
HR	5.5	5.25
HS	8.0	7.5
AD	4.5	4.25
HD	7.25	7.25



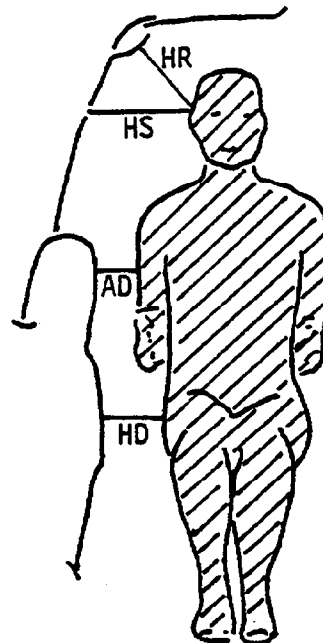
NOTE: ALL DIMENSIONS IN INCHES

OCCUPANT CLEARANCE DIMENSIONS
Volkswagen Sedan

	Driver	Passenger
HH	15.0	
HW	17.0	
CD	19.75	
CS	16.0	
KDL	7.25	
KDR	7.25	



	Driver	Passenger
HR	5.25	
HS	6.5	
AD	2.0	
HD	5.0	



NOTE: ALL DIMENSIONS IN INCHES

SECTION 3

TEST RESULTS

SUMMARY OF RESULTS

Test No.: 1960

Project Title: Dummy Response in Side Impact

Date: 12-11-80

Time: 1:00 p.m.

Temp: 70°F

Subject Vehicle	Impacted Vehicle/Object
-----------------	-------------------------

General

Vehicle/Obj

Description

Chevrolet Malibu

Volkswagen Sedan

Test weight (lbs)

3870

2100

Test velocity (mph)

16.8

33.0

Vehicle Response Data

Max. crush (in.)

Static

3.25

11.75

Dynamic

-

-

Max. intrusion (in.)

0

11.75

Subject Vehicle					Impacted Vehicle/Object				
DR.	MFP	RFP	LRP	RRP	DR.	MFP	RFP	LRP	RRP

Occupant Response

HIC

6

2

64

Head SI

7

3

129

Maximum resultant

head Gs

6

4

47

Maximum A-P head Gs

3

2

30

Maximum S-I head Gs

3

2

25

Maximum R-L head Gs

3

2

27

Chest SI

7

2

-

Chest A-P deflec-

tion (in.) Upper

-

-

0.4

Chest R-L deflec-

tion (in.) Lower

-

-

0.65

Maximum resultant

chest Gs

7

6

-

Maximum A-P chest Gs

4

3

-

Maximum S-I chest Gs

3

3

-

Maximum R-L chest Gs

3

3

-

Maximum resultant

pelvic Gs

-

-

87

Maximum A-P pelvic Gs

-

-

41

Maximum S-I pelvic Gs

-

-

10

Maximum R-L pelvic Gs

-

-

77

Maximum femur load -

right (lbs)

100

200

-

Maximum femur load -

left (lbs)

80

125

-

SUMMARY OF RESULTS (cont'd)

Subject Vehicle					Impacted Vehicle/Object				
DR.	MFP	RFP	LRP	RRP	DR.	MFP	RFP	LRP	RRP

Restraint Response

Left Upper Rib Data

BLUR	7.0
Peak Acceleration (Gs)	59.9
Peak Acceleration (Gs) (3 msec)	46.5

Upper Spine R-L

Peak Acceleration (Gs)	39.1
Peak Acceleration (Gs) (3 msec)	33.3
Peak Power (ft ² /sec ³)	15907

Upper Spine Resultant

Peak Acceleration (Gs)	43.2
Peak Acceleration (Gs) (3 msec)	35.3
Peak Power (ft ² /sec ³)	23302

Lower Spine R-L

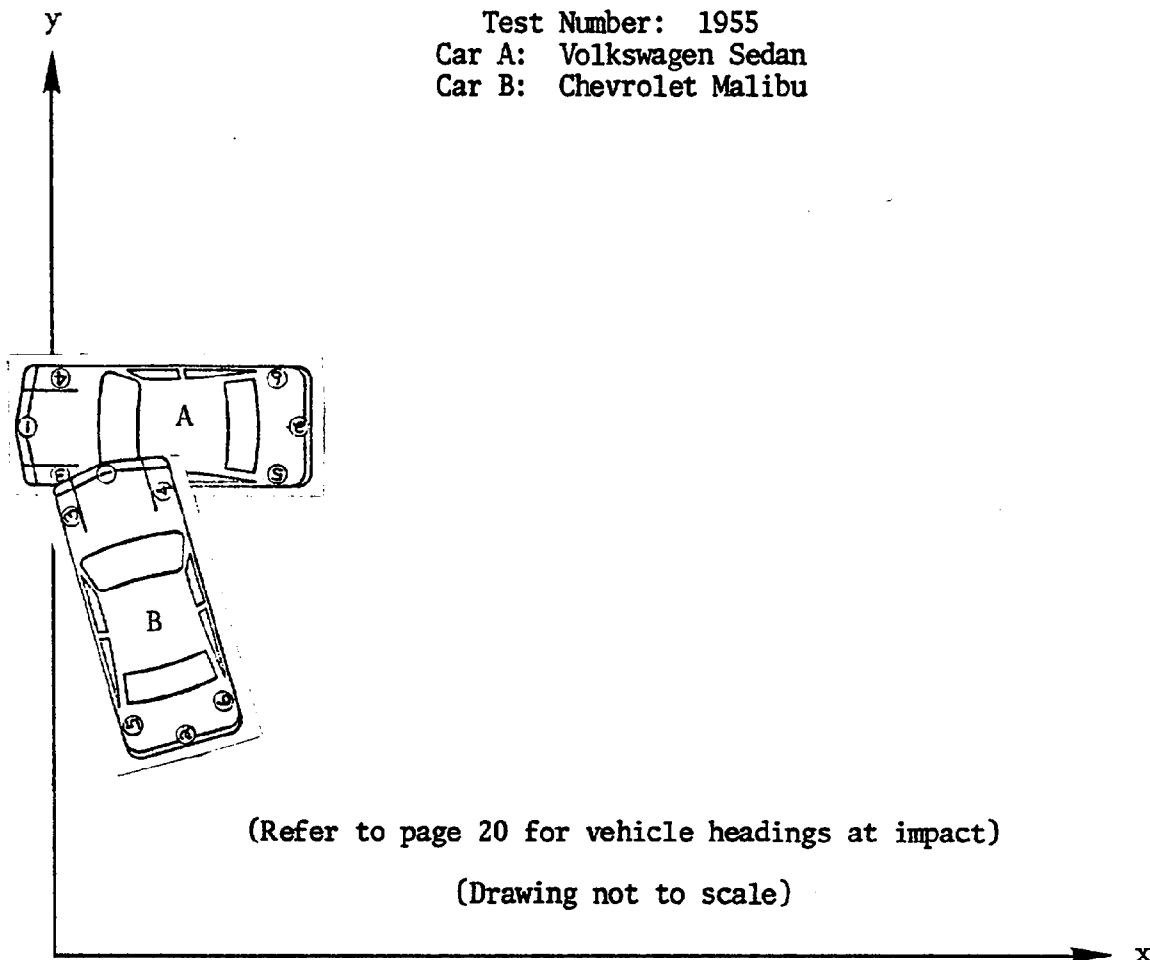
Peak Acceleration (Gs)	46.4
Peak Acceleration (Gs) (3 msec)	44.8
Peak Power (ft ² /sec ³)	22326

Lower Spine Resultant

Peak Acceleration (Gs)	54.5
Peak Acceleration (Gs) (3 msec)	53.6
Peak Power (ft ² /sec ³)	37551

Remarks:

POST-TEST VEHICLE POSITIONS



Position		Car A		Car B	
Number	Description	X	Y	X	Y
1	Front Bumper Centerline	-3'6"	33'4"	2'1"	31'3"
2	Rear Bumper Centerline	9'10"	33'4"	5'5"	16'0"
3	Left Front Wheel Centerline	-1'5"	31'0"	0	28'6"
4	Right Front Wheel Centerline	-11"	36'0"	5'8"	29'11"
5	Left Rear Wheel Centerline	6'10"	30'10"	1'7"	19'5"
6	Right Rear Wheel Centerline	6'10"	35'8"	7'2"	20'5"

PHOTOGRAPHS



FIGURE 1. PRE-TEST VOLKSWAGEN RIGHT SIDE

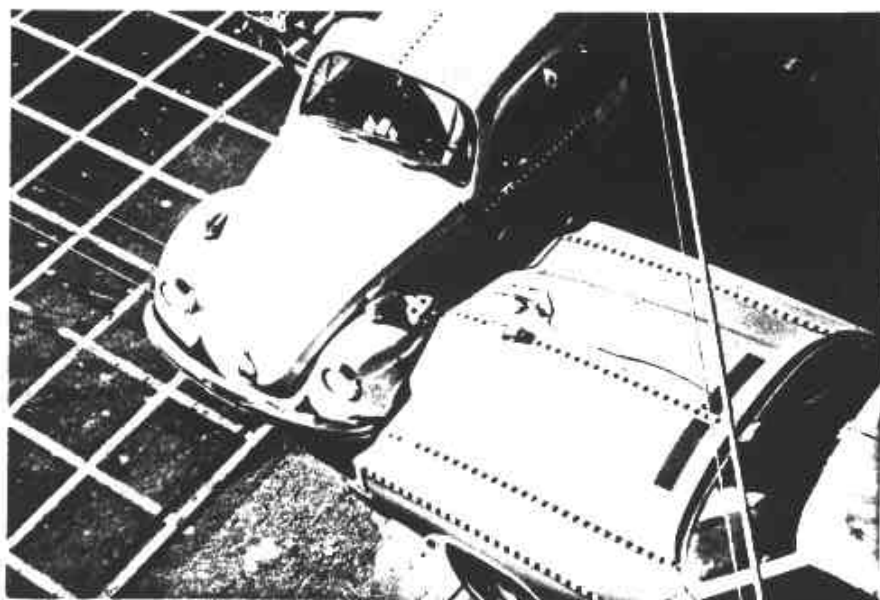


FIGURE 2. PRE-TEST VEHICLE IMPACT POSITIONS



FIGURE 3. PRE-TEST VEHICLE IMPACT POSITIONS



FIGURE 4. PRE-TEST VEHICLE IMPACT POSITIONS

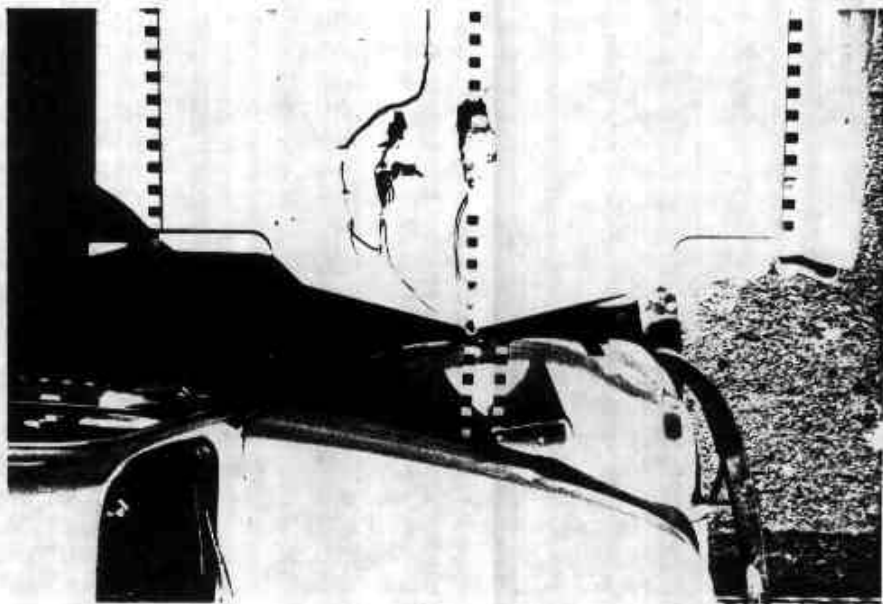


FIGURE 5. PRE-TEST VEHICLE IMPACT POSITIONS



FIGURE 6. PRE-TEST MALIBU FRONT



FIGURE 7. PRE-TEST VOLKSWAGEN OCCUPANT



FIGURE 8. PRE-TEST VOLKSWAGEN OCCUPANT

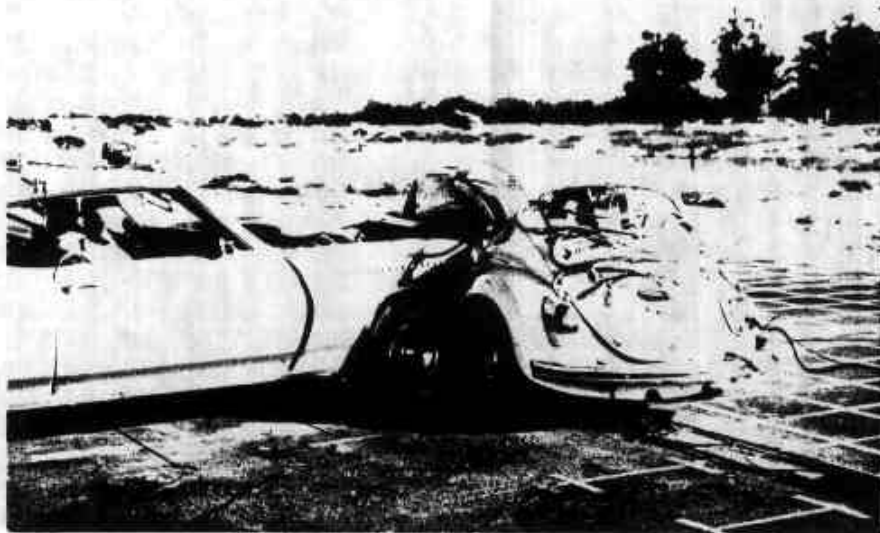


FIGURE 9. TEST 1960 IMPACT



FIGURE 10. TEST 1960 IMPACT

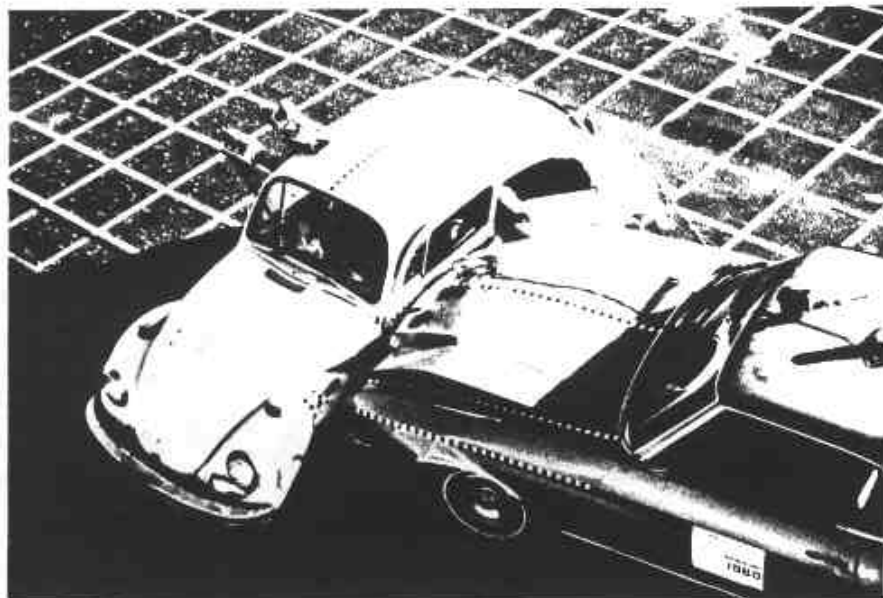


FIGURE 11. POST-TEST VEHICLE POSITIONS



FIGURE 12. POST-TEST VEHICLE POSITIONS

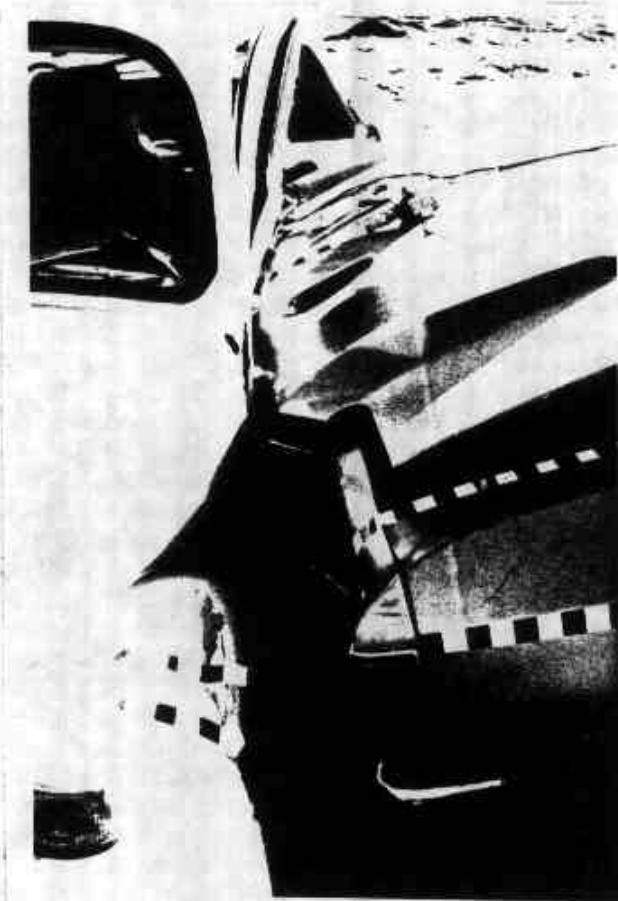


FIGURE 13. POST-TEST VEHICLE POSITIONS

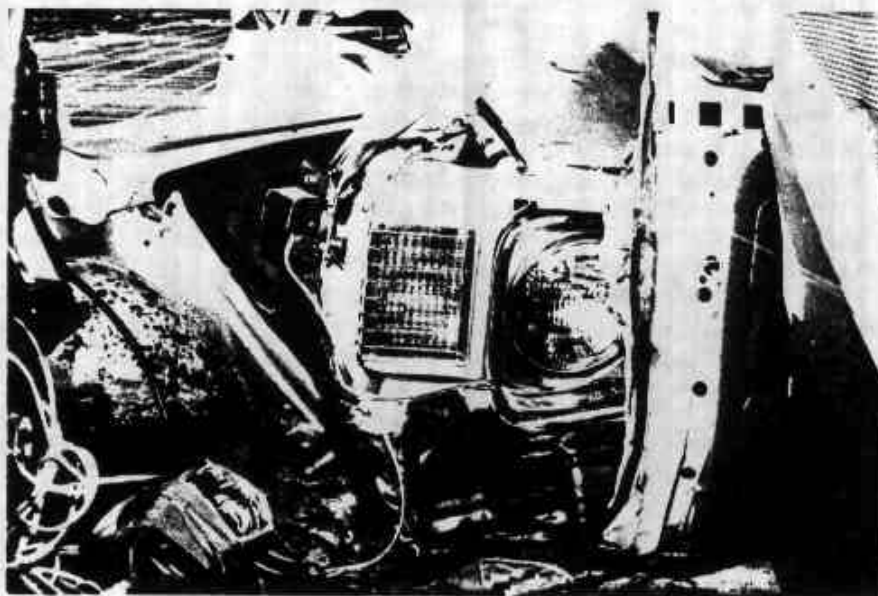


FIGURE 14. POST-TEST MALIBU INTRUSION INTO VOLKSWAGEN



FIGURE 15. POST-TEST VOLKSWAGEN FRONT

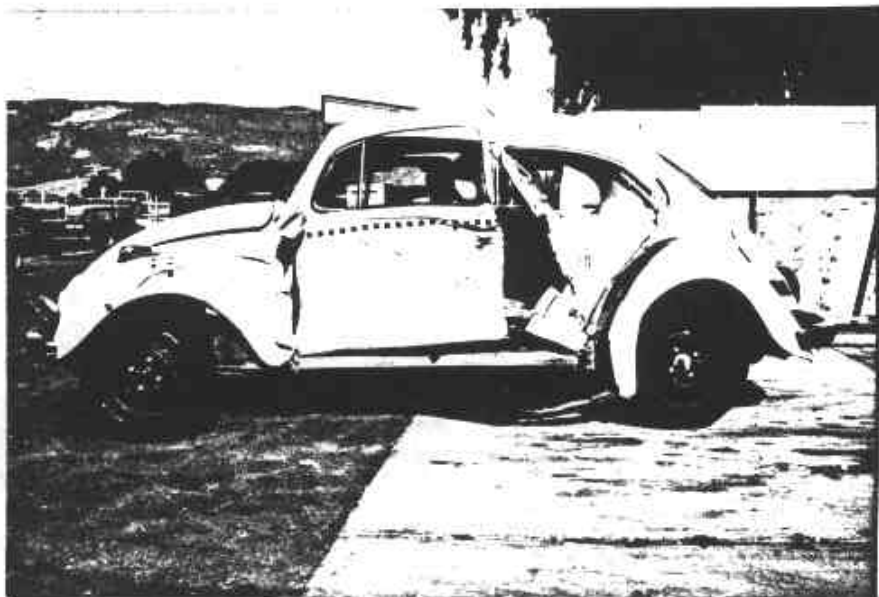


FIGURE 16. POST-TEST VOLKSWAGEN LEFT SIDE



FIGURE 17. POST-TEST VOLKSWAGEN LEFT SIDE

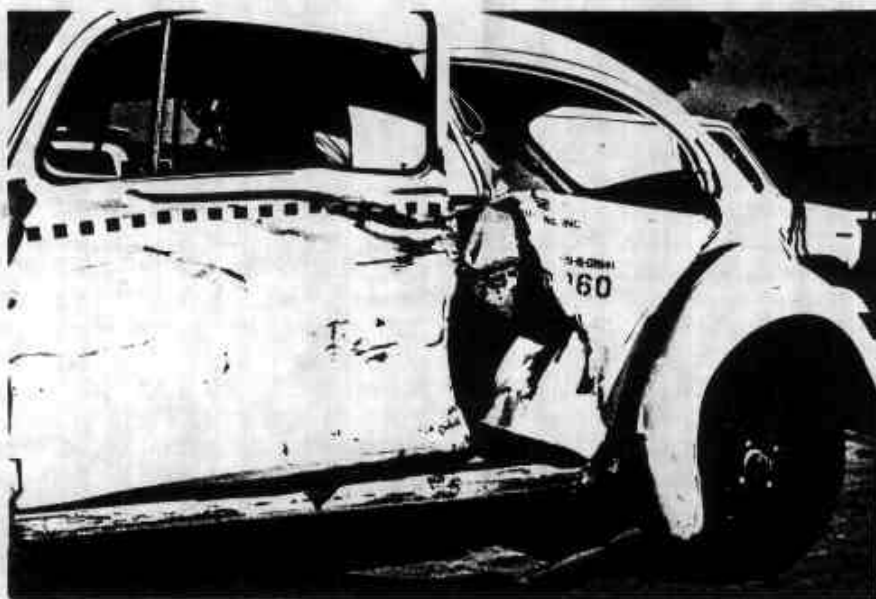


FIGURE 18. POST-TEST VOLKSWAGEN LEFT SIDE



FIGURE 19. POST-TEST VOLKSWAGEN LEFT SIDE



FIGURE 20. POST-TEST VOLKSWAGEN DUMMY POSITION

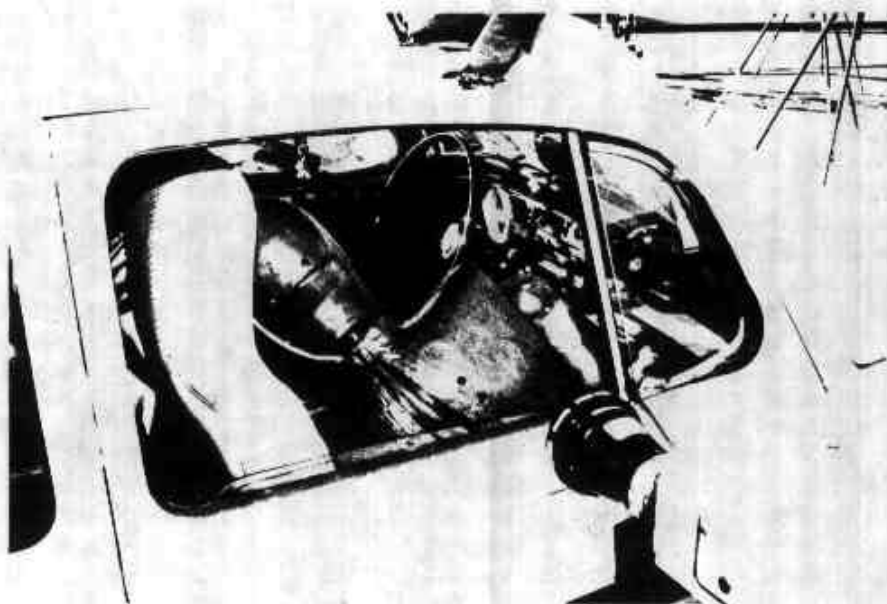


FIGURE 21. POST-TEST VOLKSWAGEN DUMMY POSITION

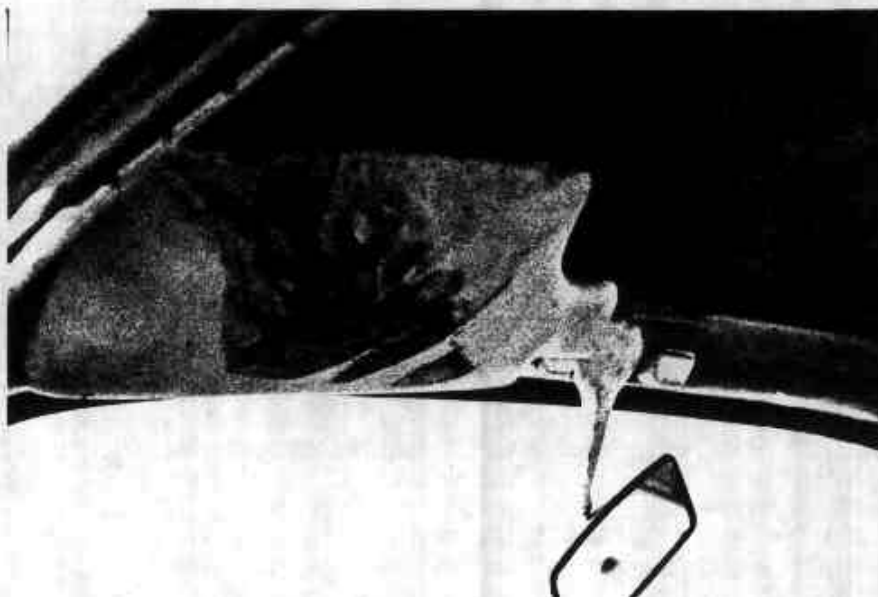


FIGURE 22. POST-TEST VOLKSWAGEN HEAD STRIKE

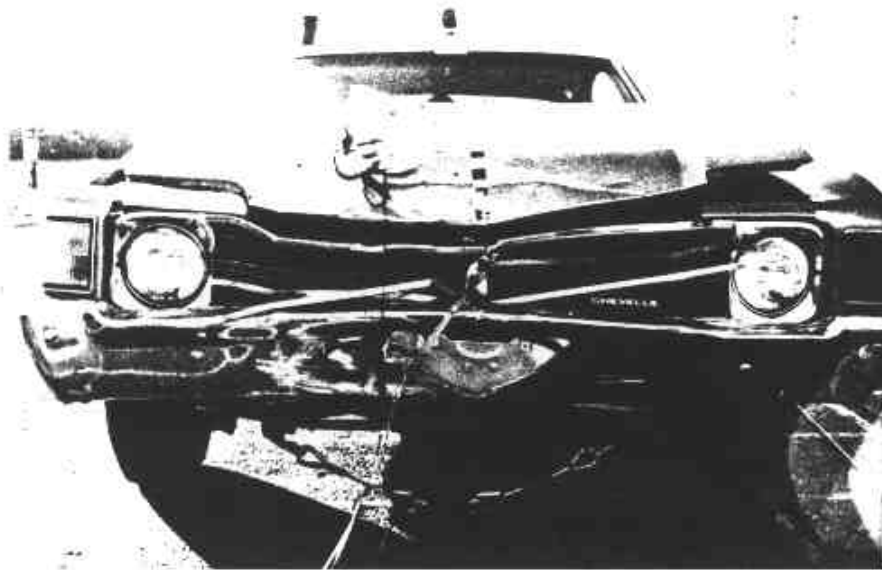


FIGURE 23. POST-TEST MALIBU FRONT



FIGURE 24. POST-TEST MALIBU RIGHT HAND SIDE

TEST DATA TRACES

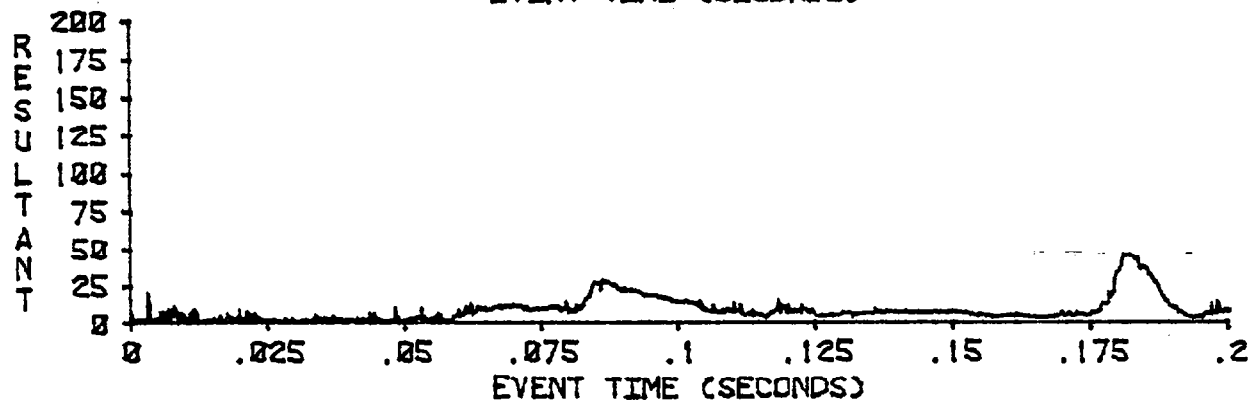
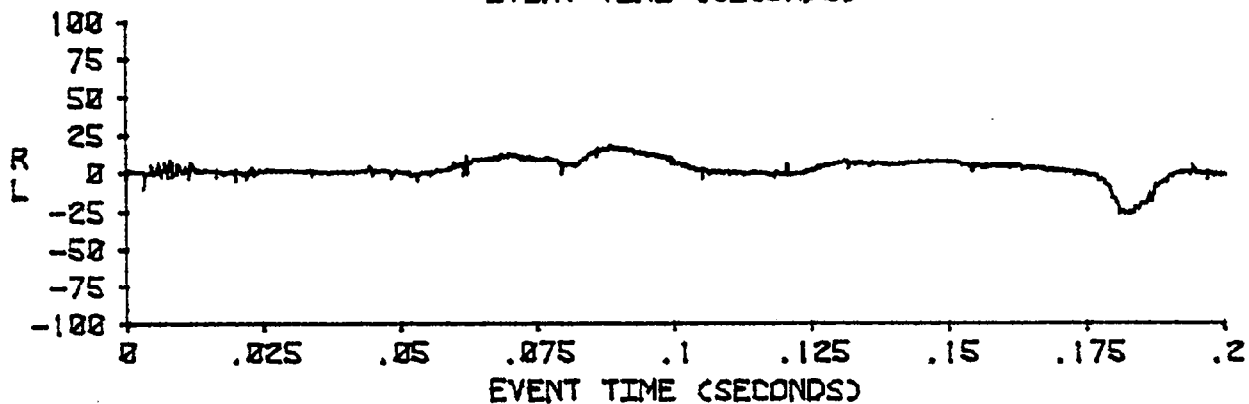
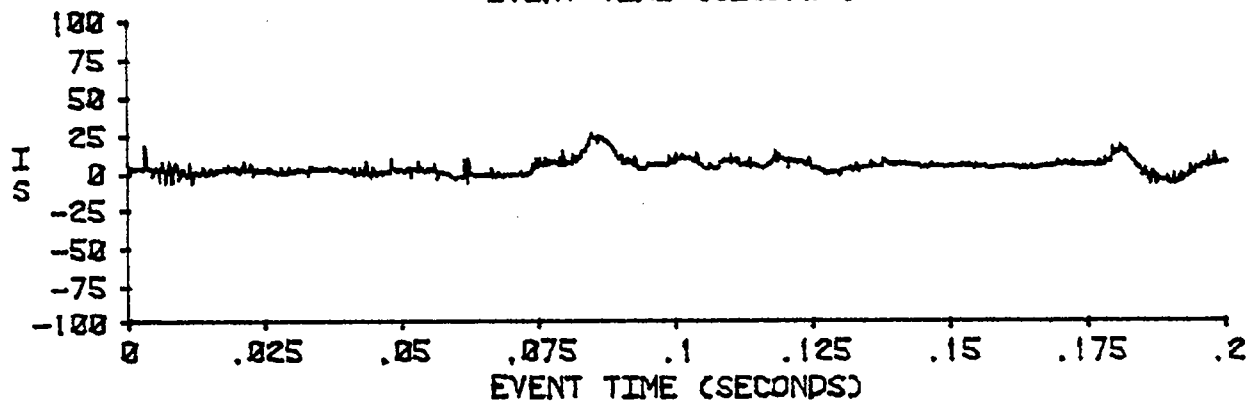
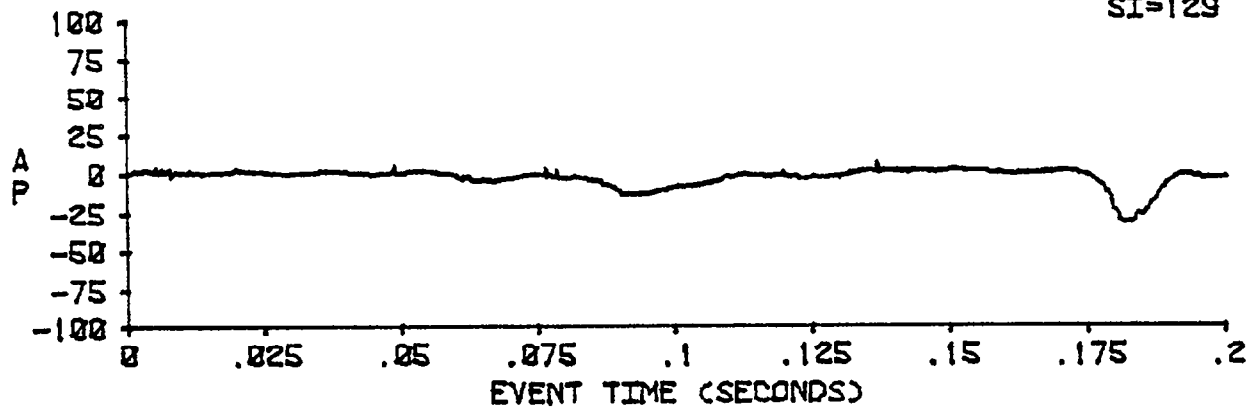
VEHICLE 2 - VW BUG - 33 MPH
DRIVER HEAD G'S
FILTER CLASS 1000

TEST 1960

11 DEC 80

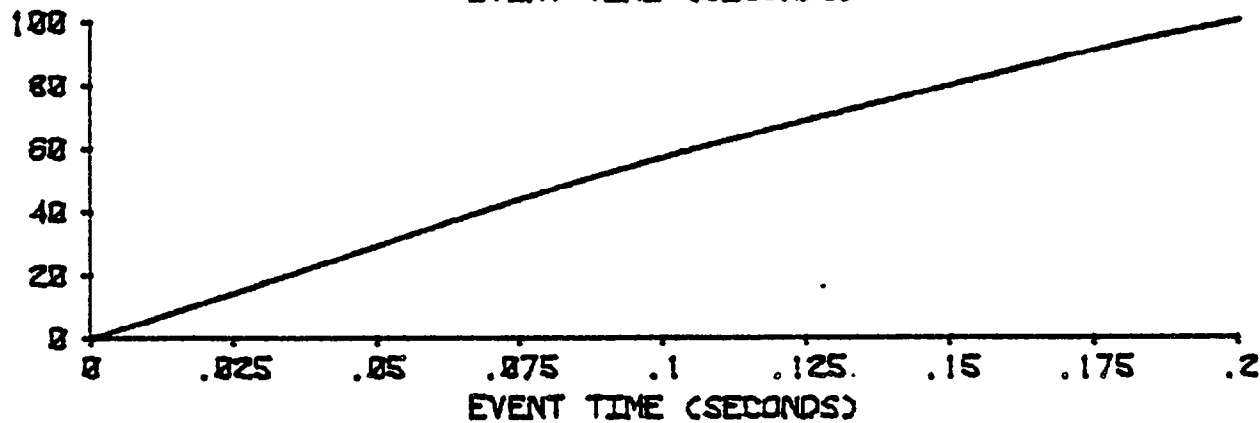
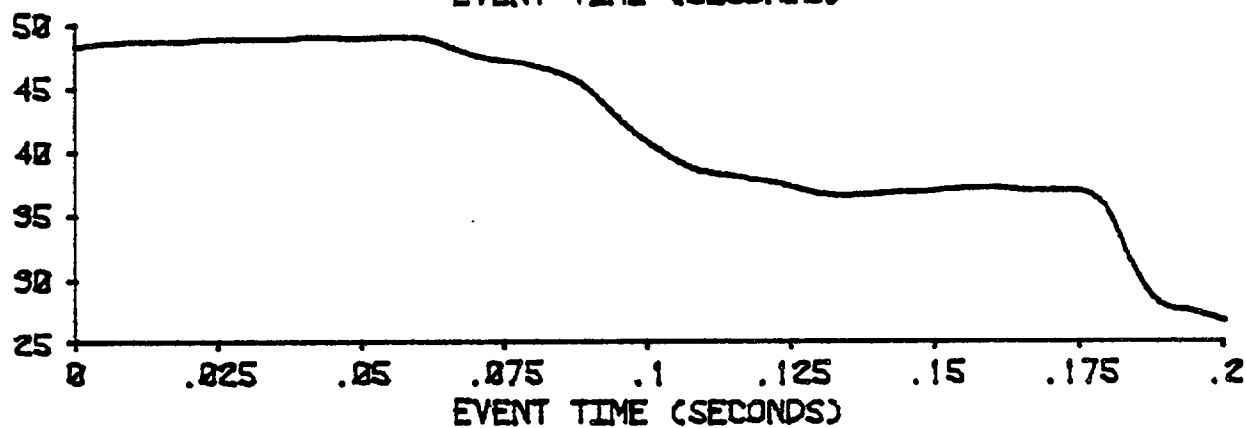
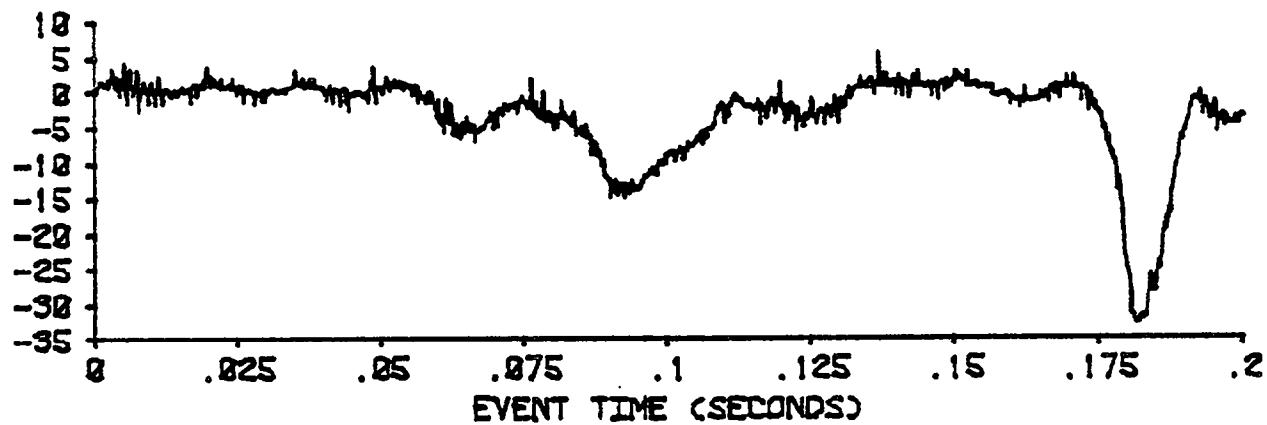
HIC=64 FROM .179 TO .1868 SECONDS

SI=129



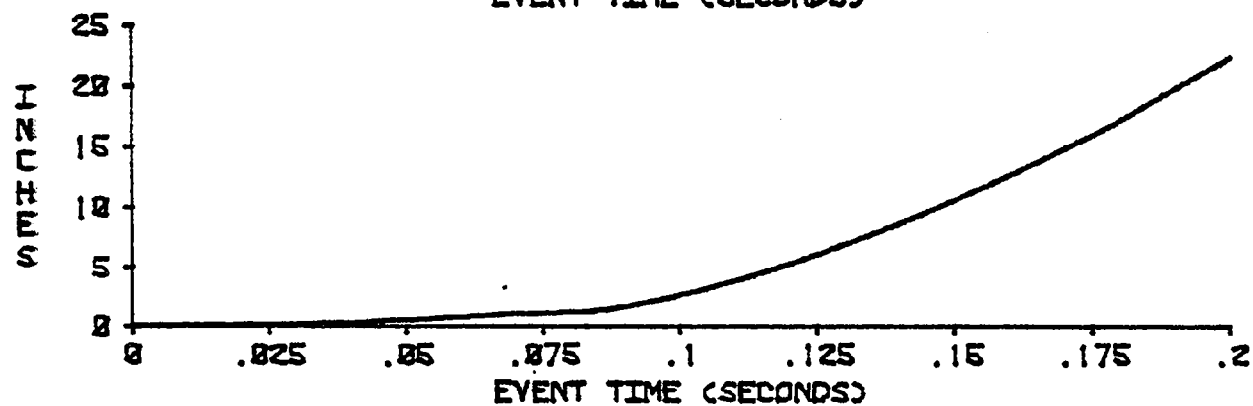
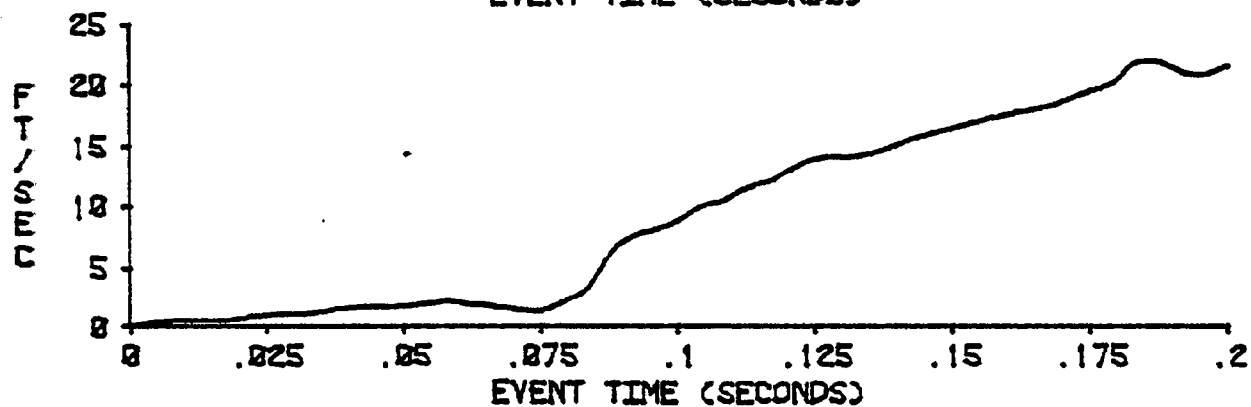
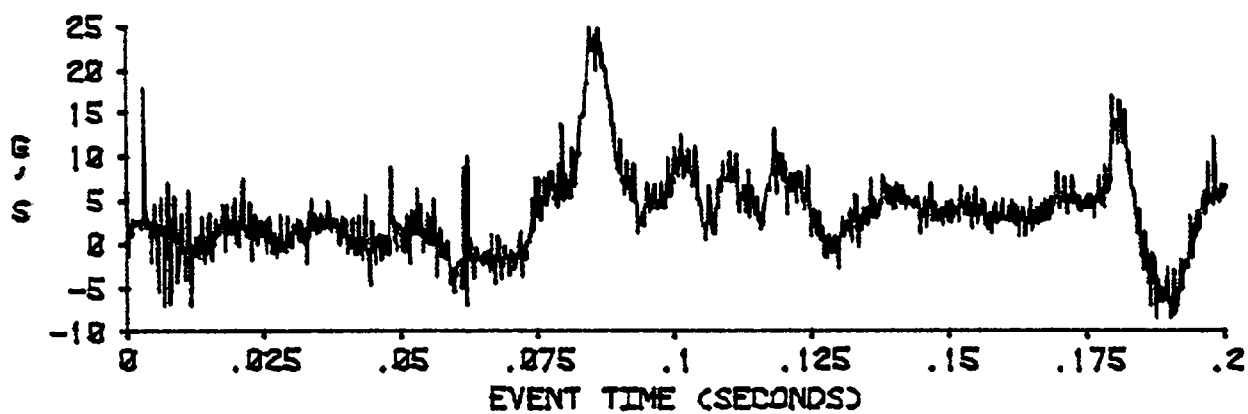
VEHICLE 2 - VW BUG - 33 MPH
DRIVER HEAD AP
FILTER CLASS 1000

TEST 1960
11 DEC 68



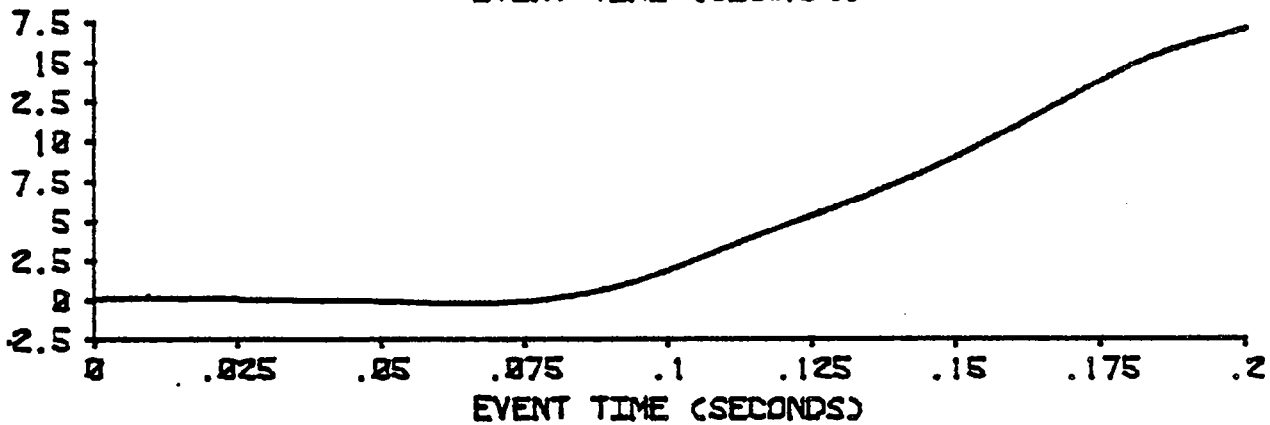
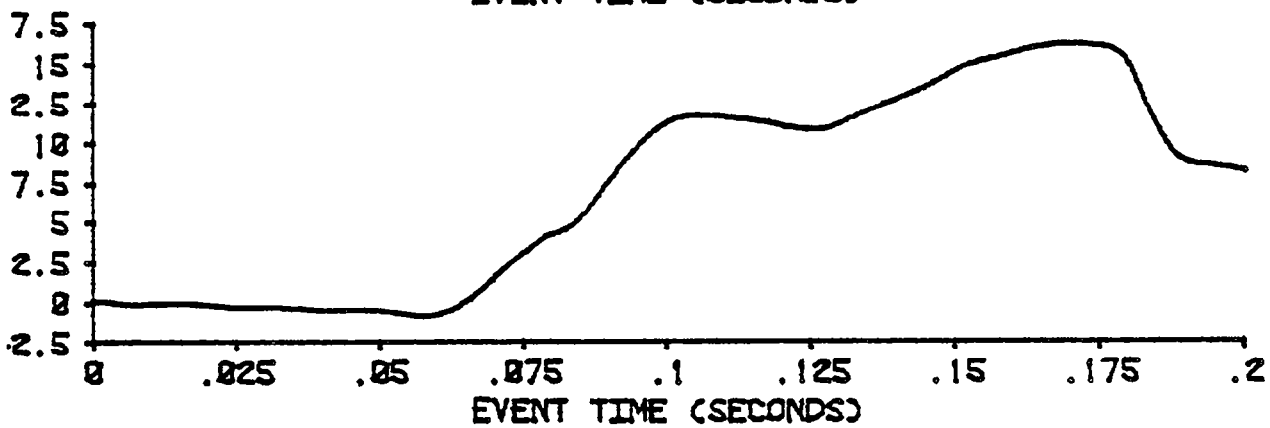
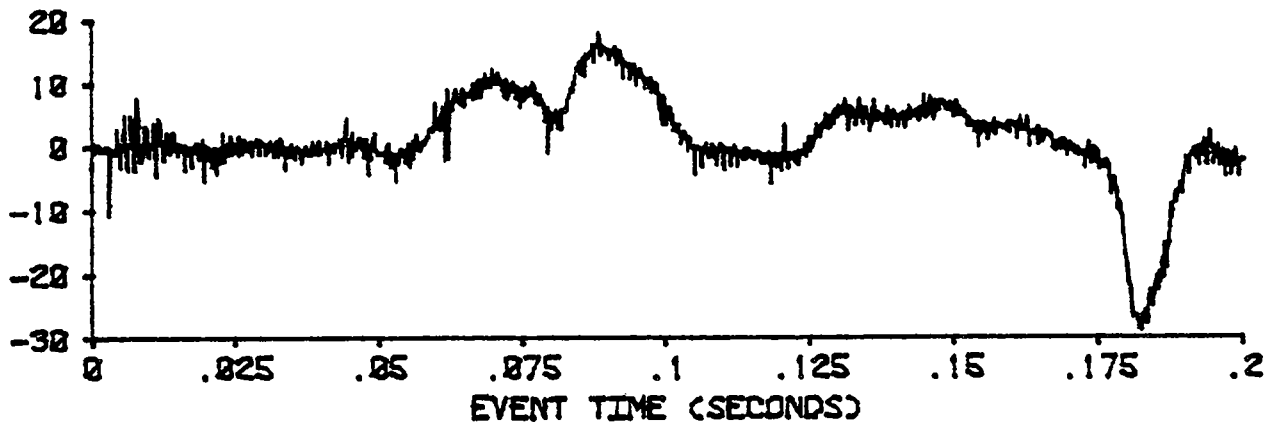
VEHICLE 2 - VW BUG - 33 MPH
DRIVER HEAD IS
FILTER CLASS 1000

TEST 1960
11 DEC 60



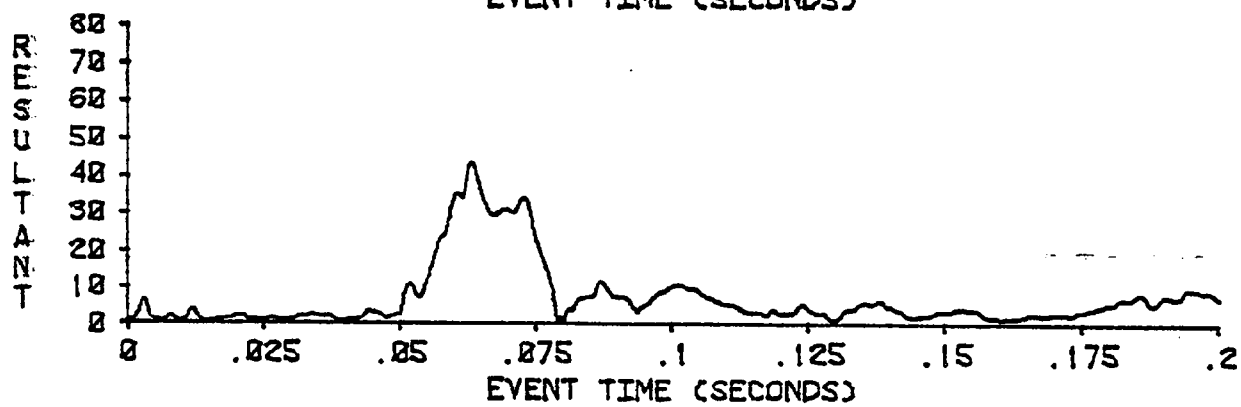
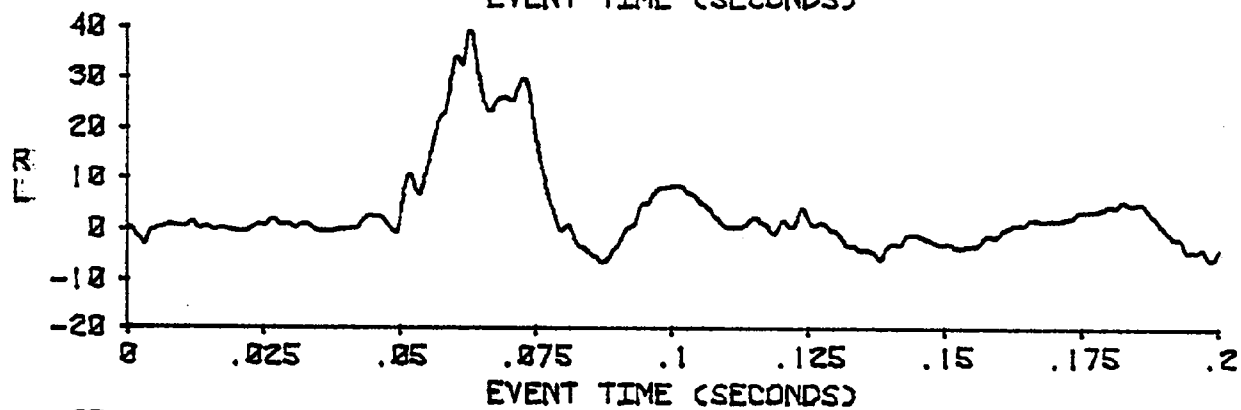
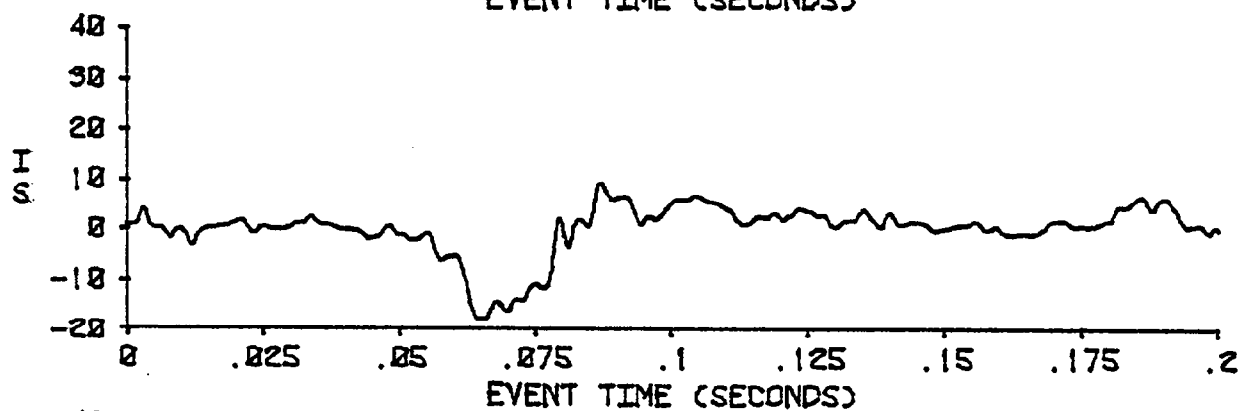
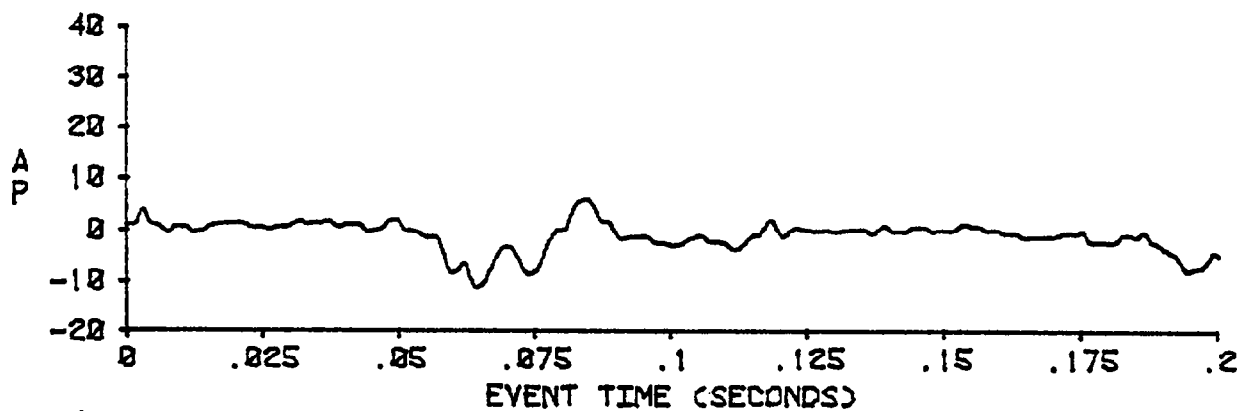
VEHICLE 2 - VW BUG - 33 MPH
DRIVER HEAD RL
FILTER CLASS 1000

TEST 1960
11 DEC 80



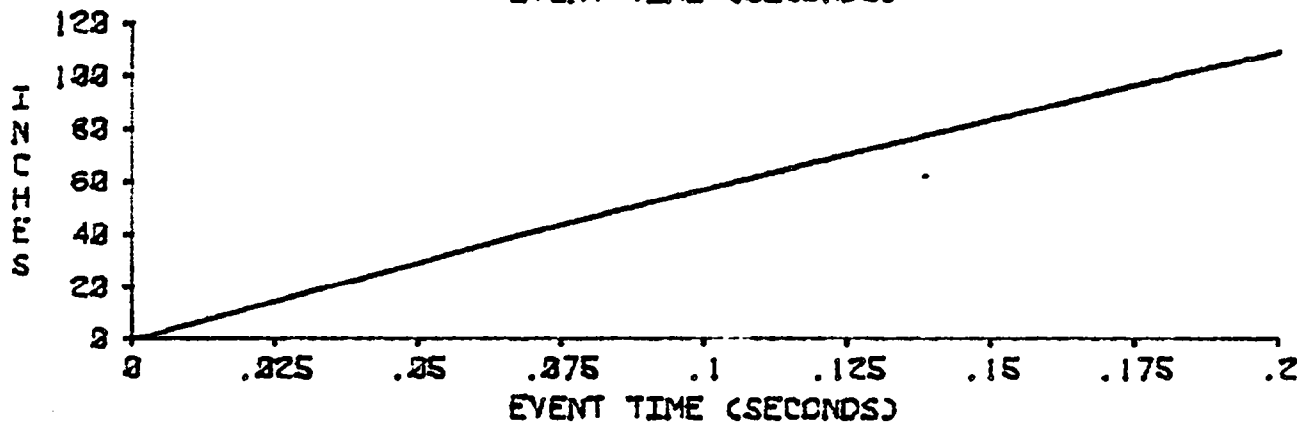
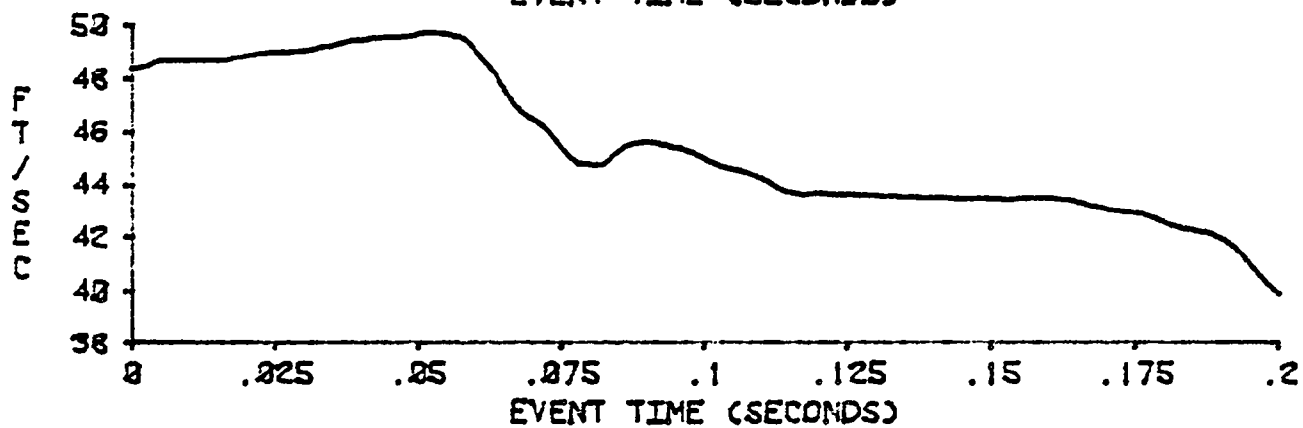
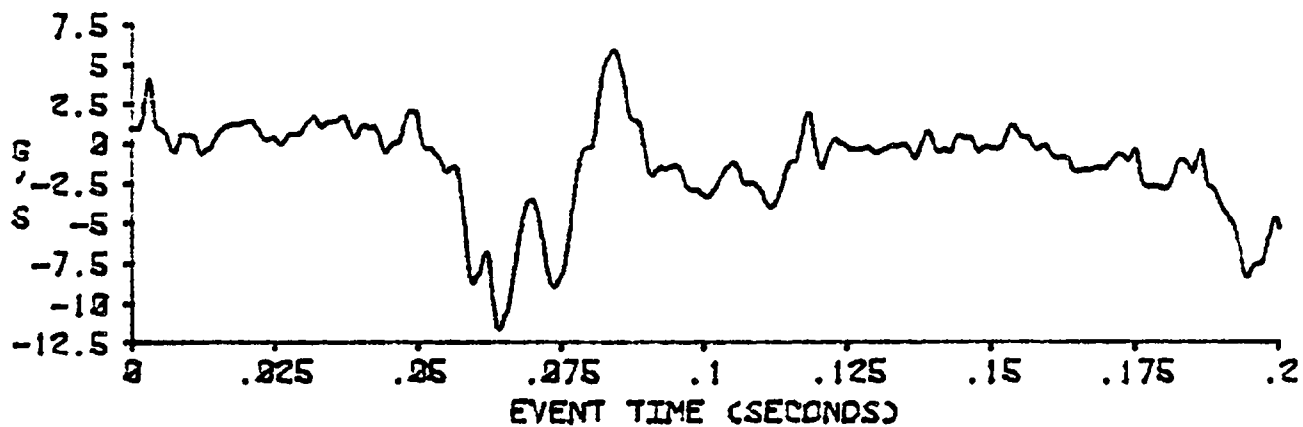
VEHICLE 2 - VW BUG - 33 MPH
DRIVER UPPER SPINE G'S
FILTER CLASS 180

TEST 1960
11 DEC 80
SI=124



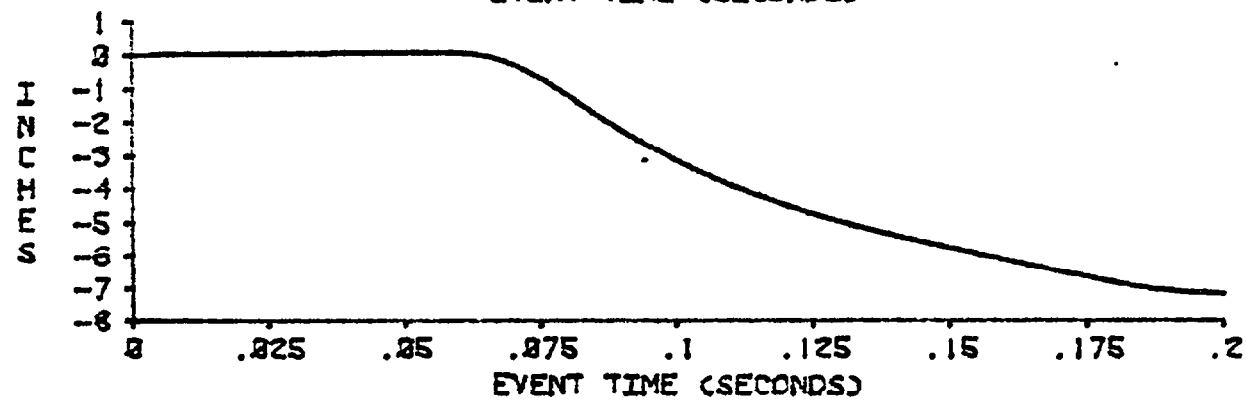
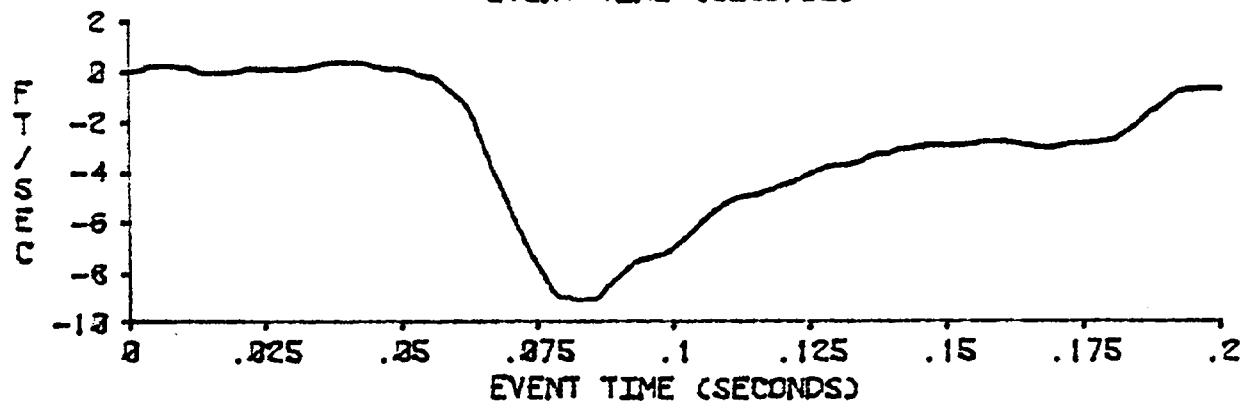
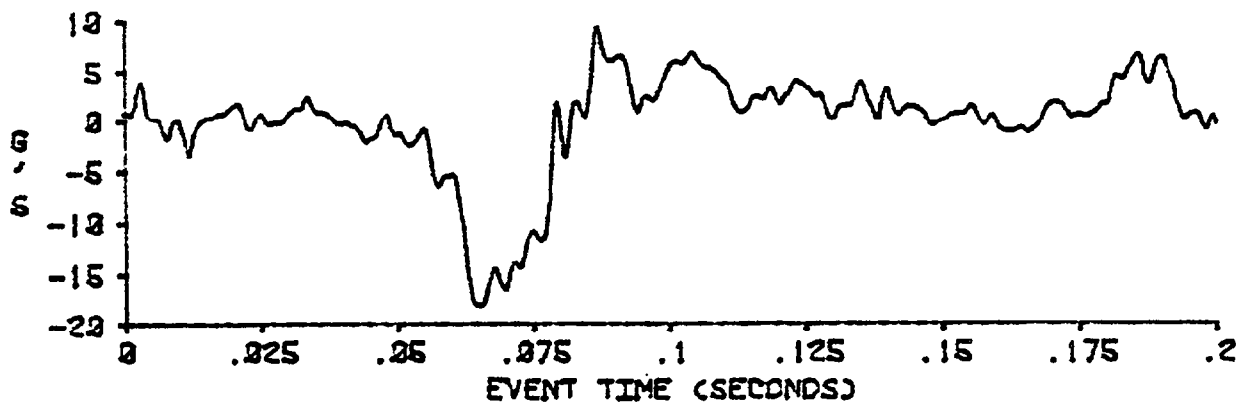
VEHICLE 2 - VW BUG - 33 MPH
DRIVER UPPER SPINE AP
FILTER CLASS 160

TEST 1960
11 DEC 80



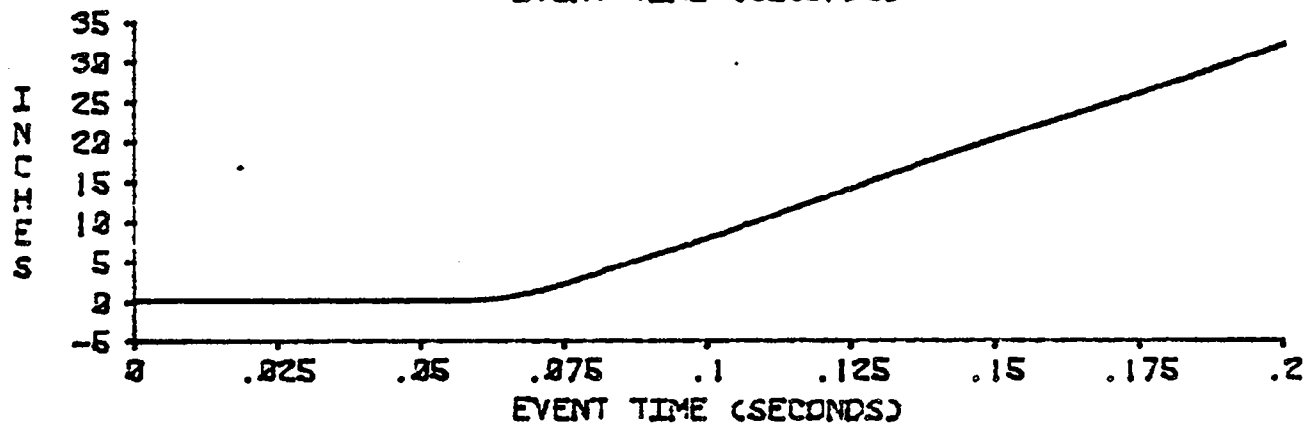
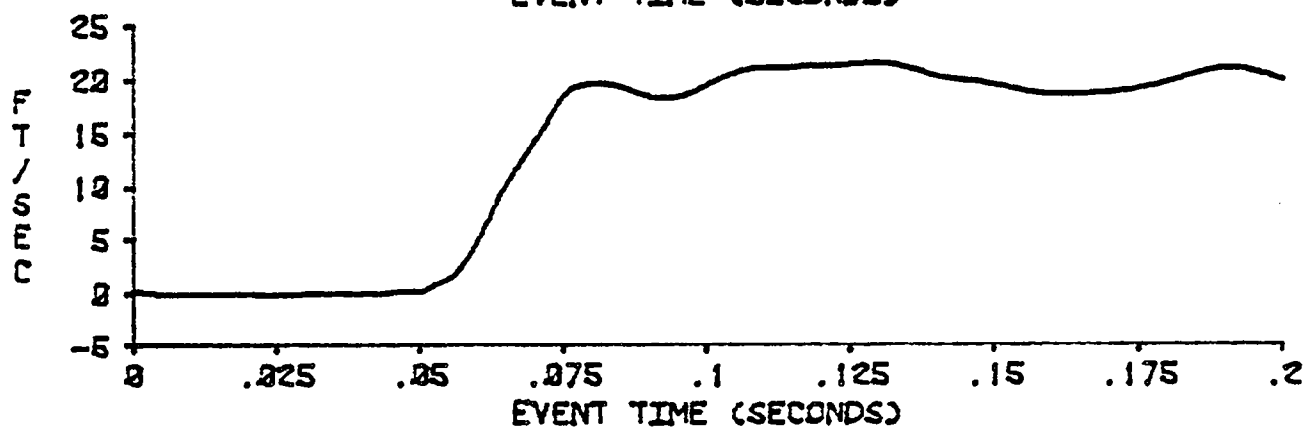
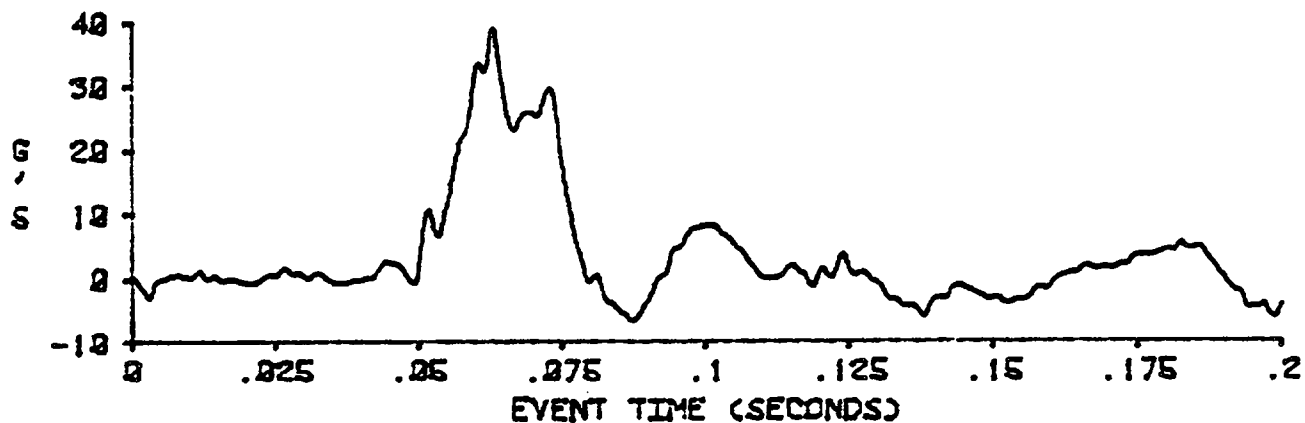
VEHICLE 2 - VW BUG - 33 MPH
DRIVER UPPER SPINE IS
FILTER CLASS 100

TEST 1960
11 DEC 60



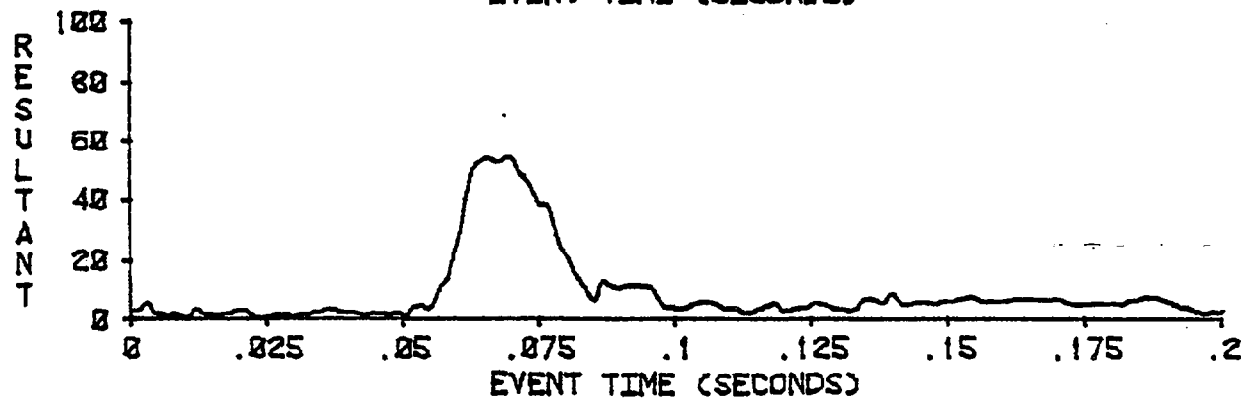
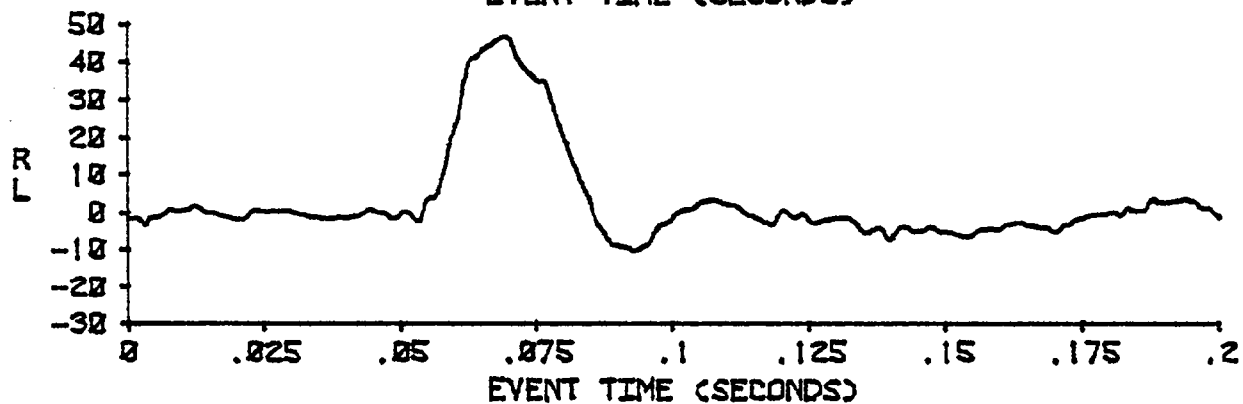
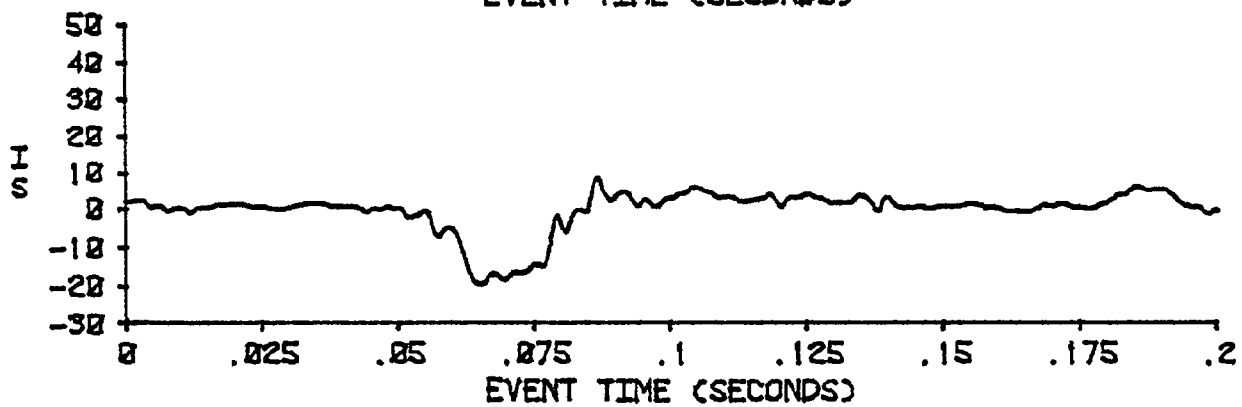
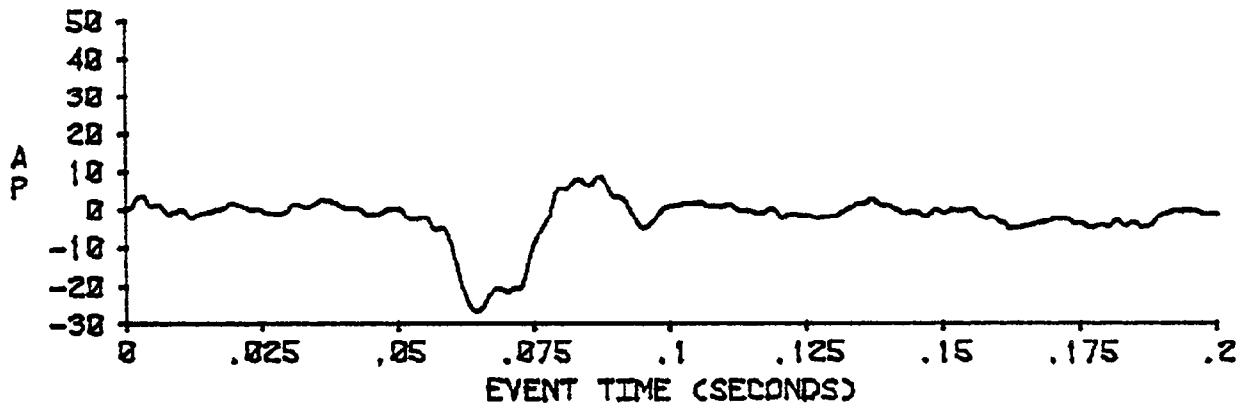
VEHICLE 2 - VW BUG - 33 MPH
DRIVER UPPER SPINE RL
FILTER CLASS 100

TEST 1960
11 DEC 60



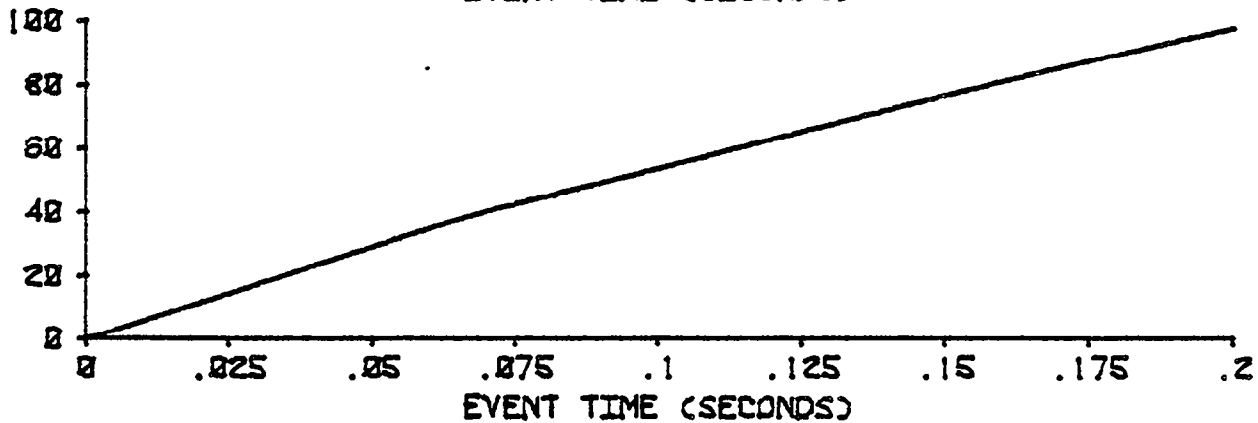
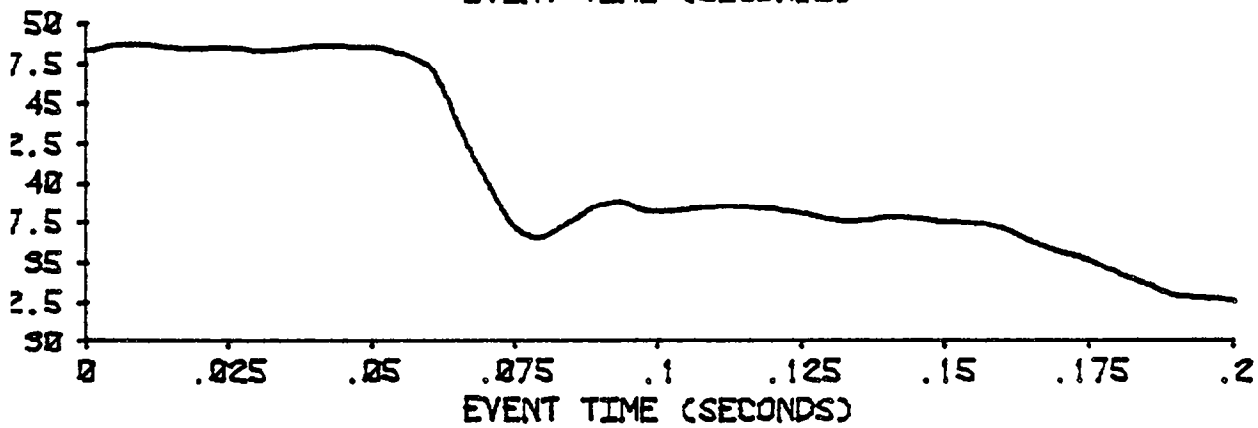
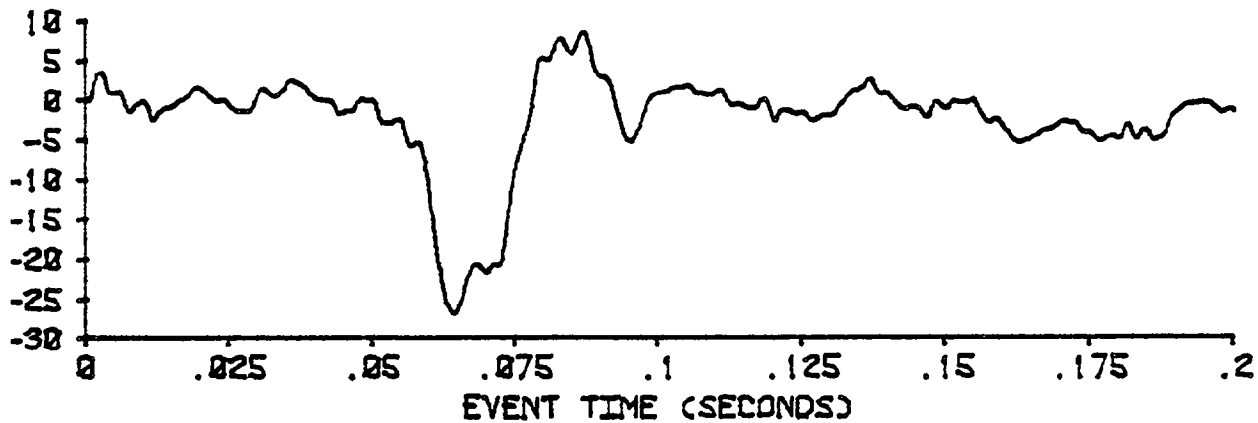
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LOWER SPINE G'S
FILTER CLASS 180

TEST 1960
11 DEC 80
SI=295



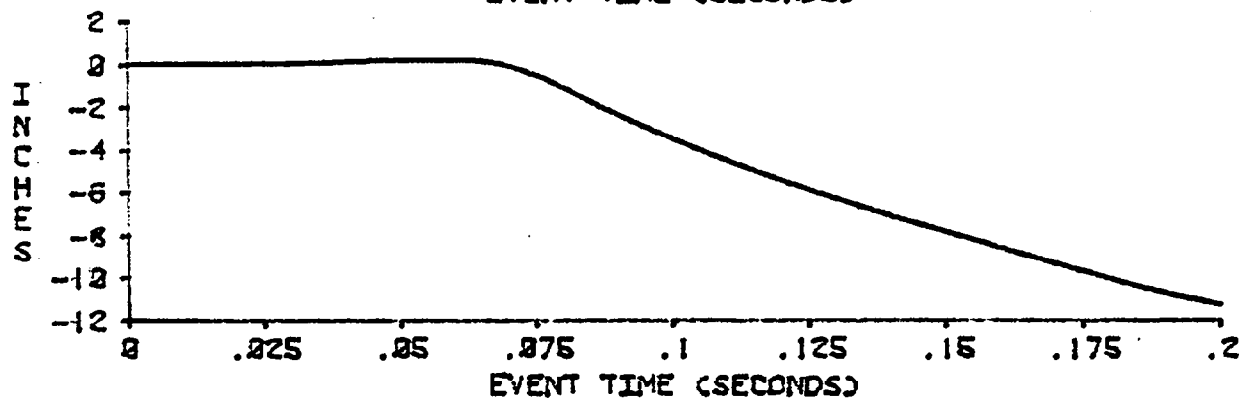
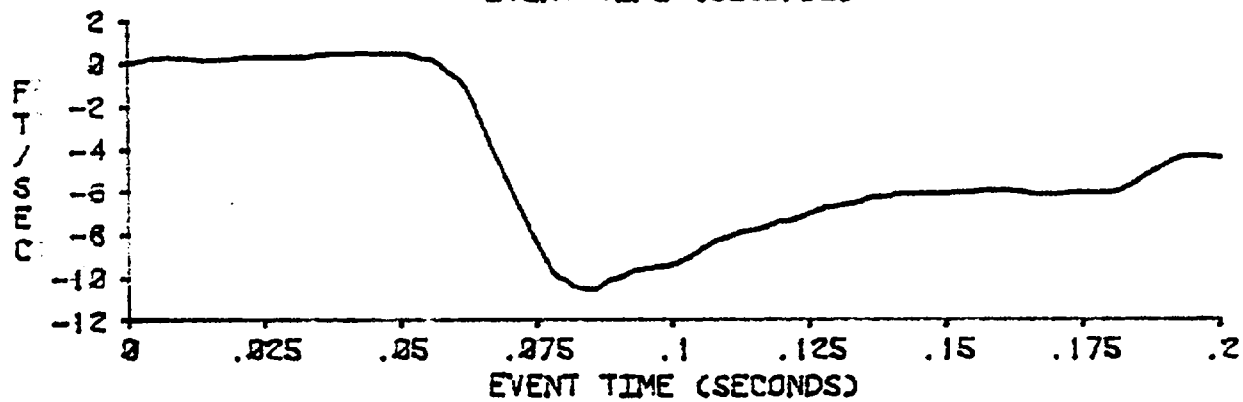
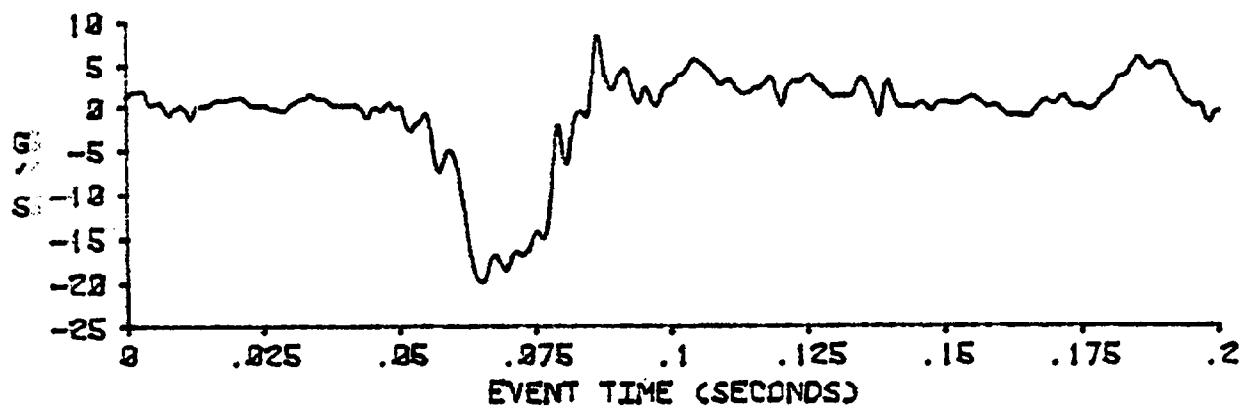
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LOWER SPINE AP
FILTER CLASS 160

TEST 1960
11 DEC 80



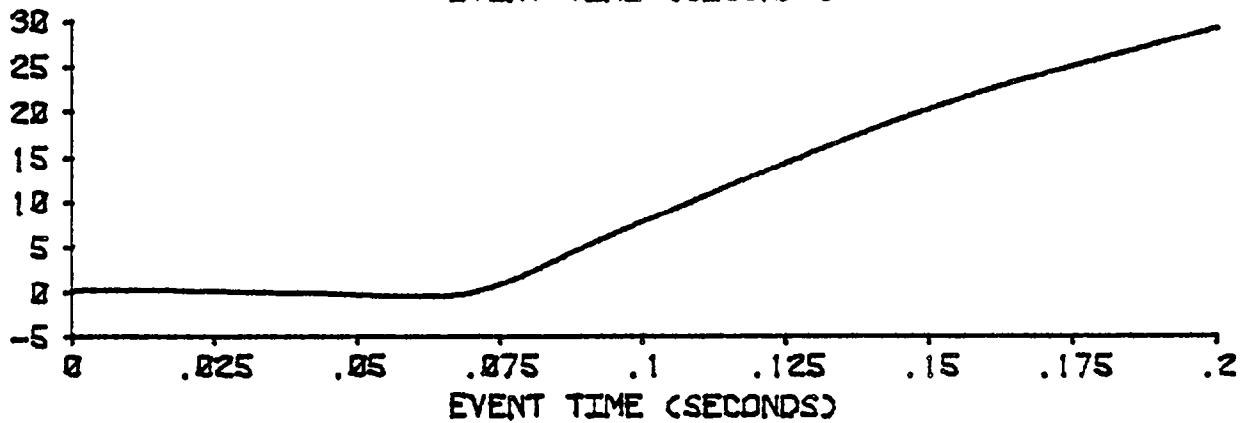
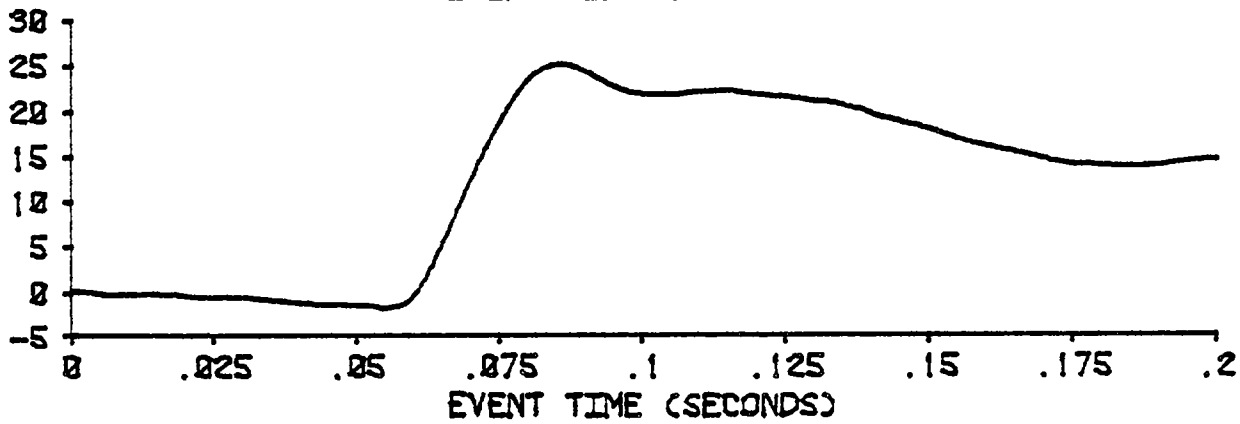
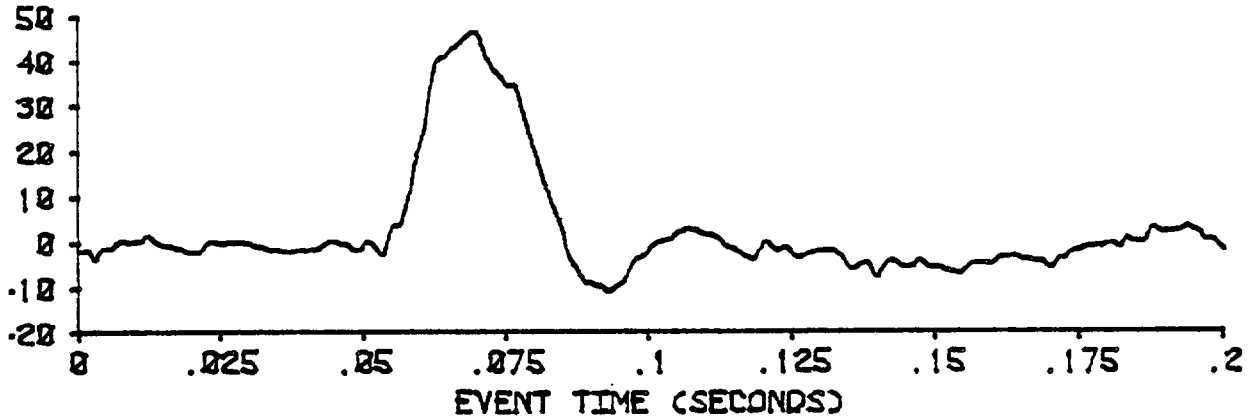
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LOWER SPINE IS
FILTER CLASS 160

TEST 1960
11 DEC 60



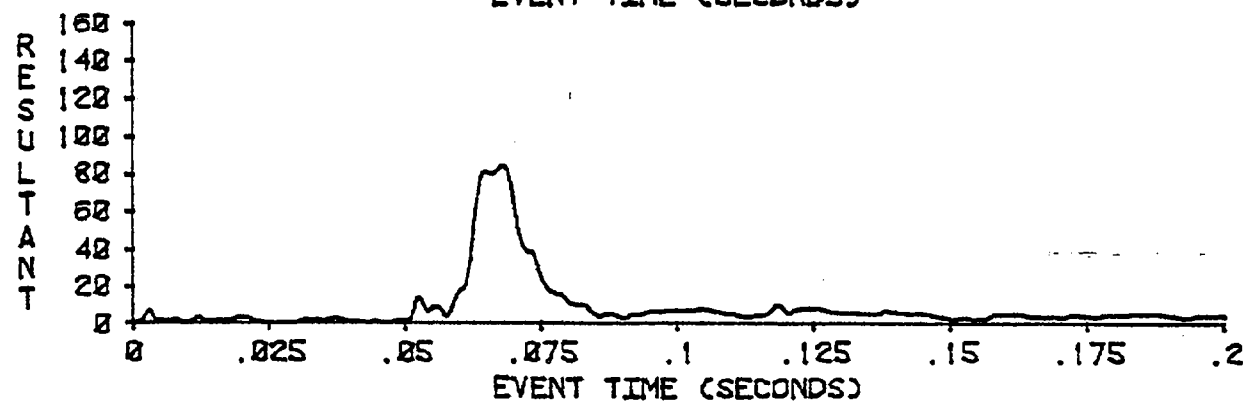
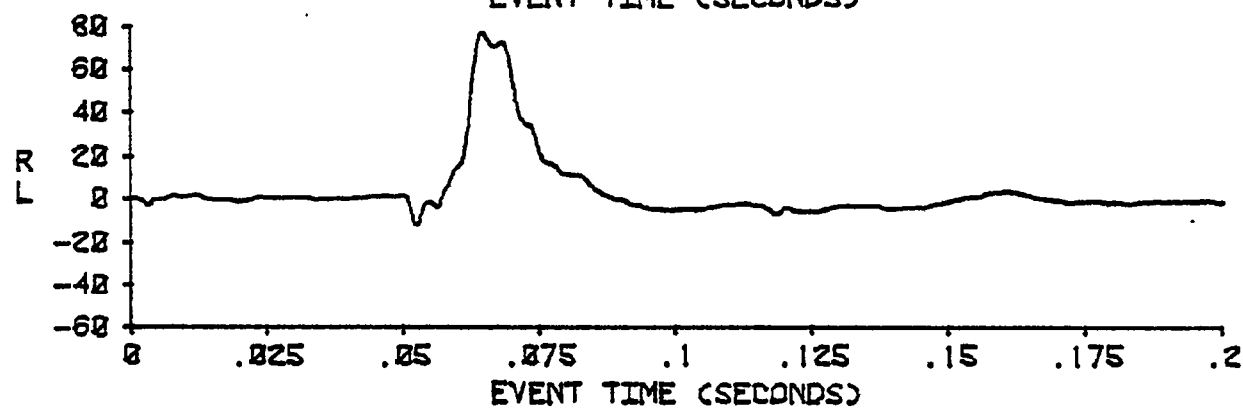
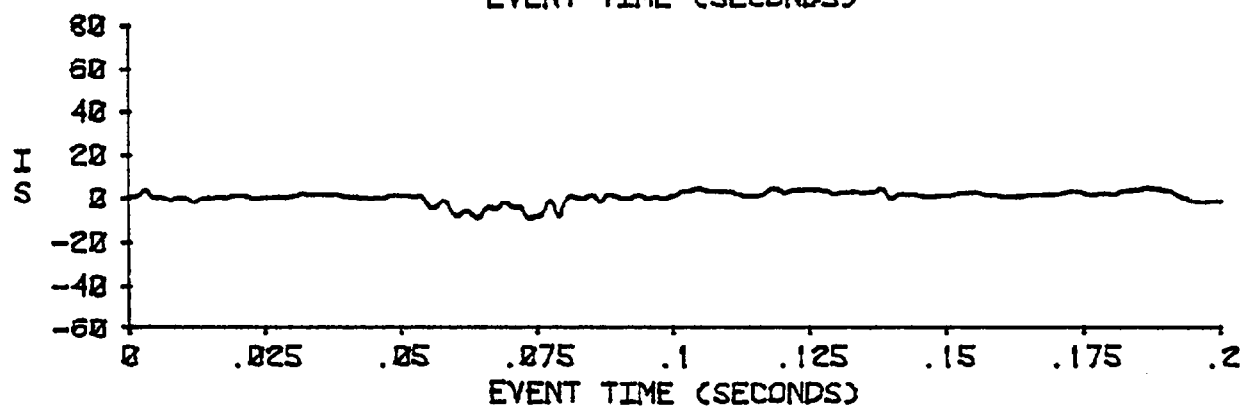
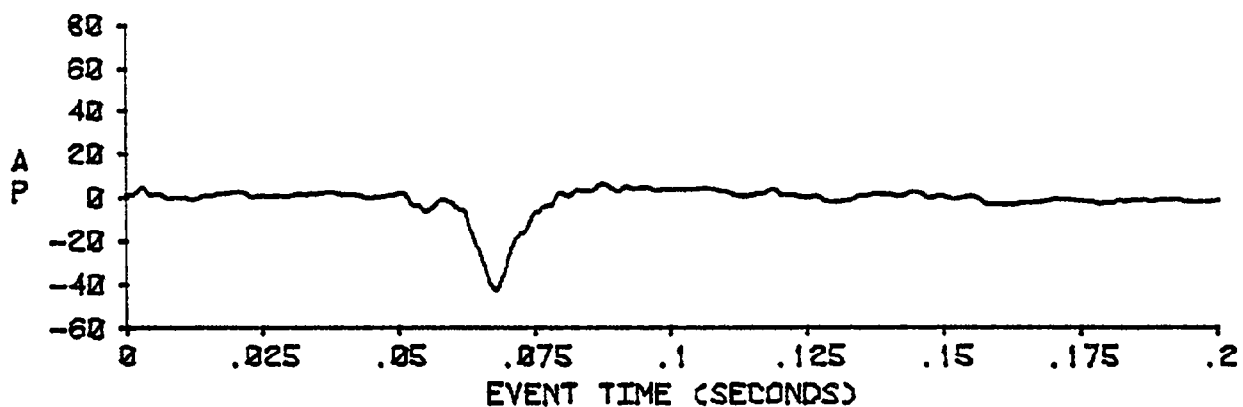
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LOWER SPINE RL
FILTER CLASS 160

TEST 1960
11 DEC 80



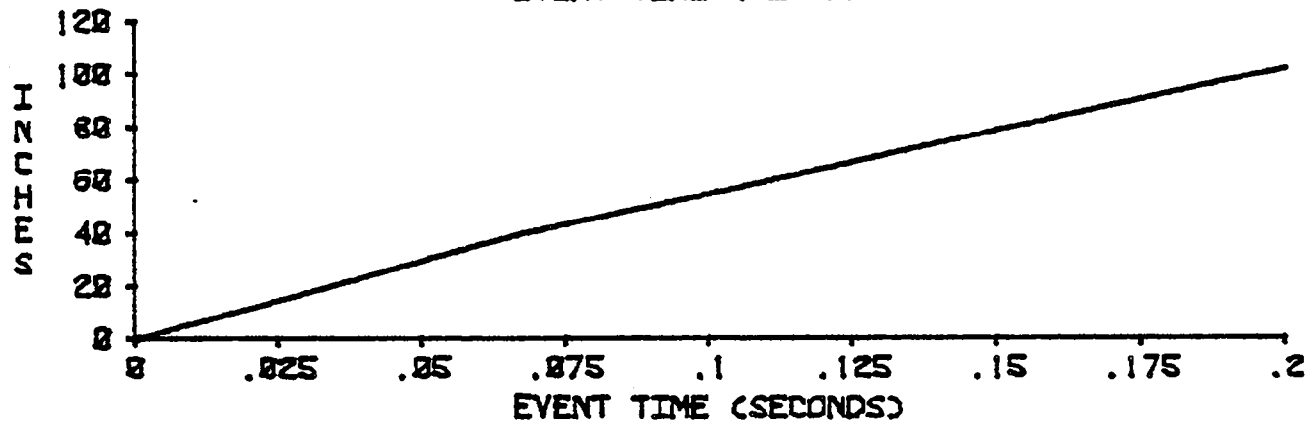
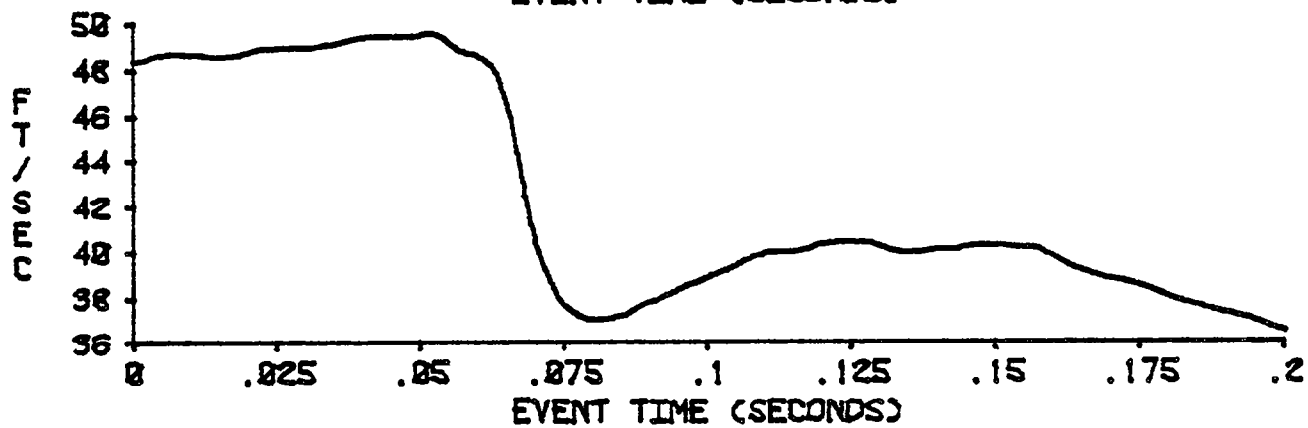
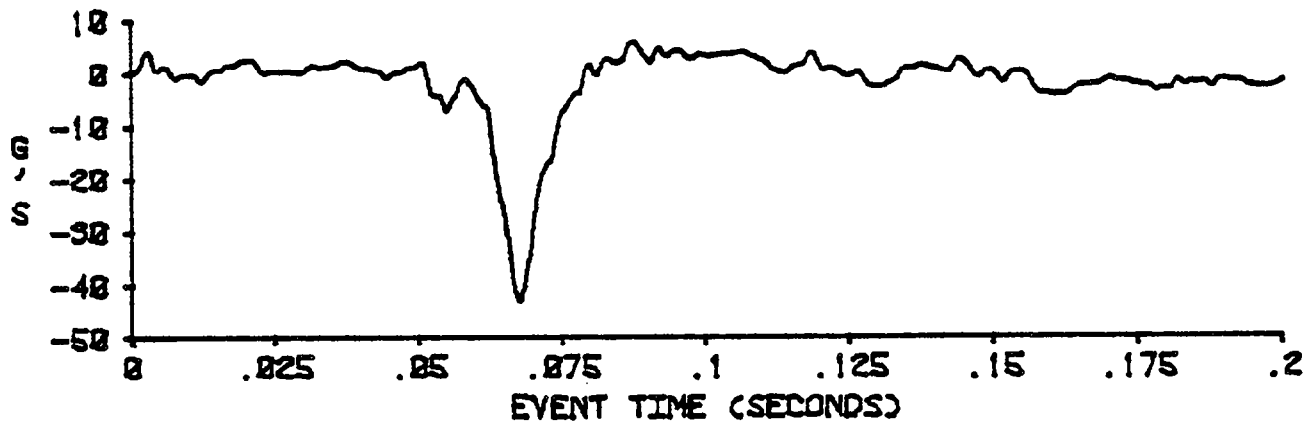
VEHICLE 2 - VW BUG - 33 MPH
DRIVER PELVIS G'S
FILTER CLASS 180

TEST 1960
11 DEC 80



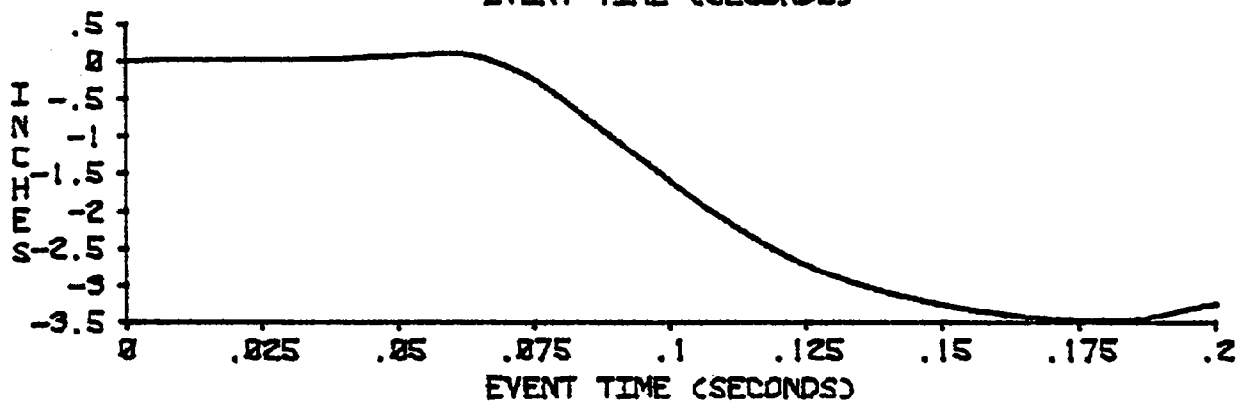
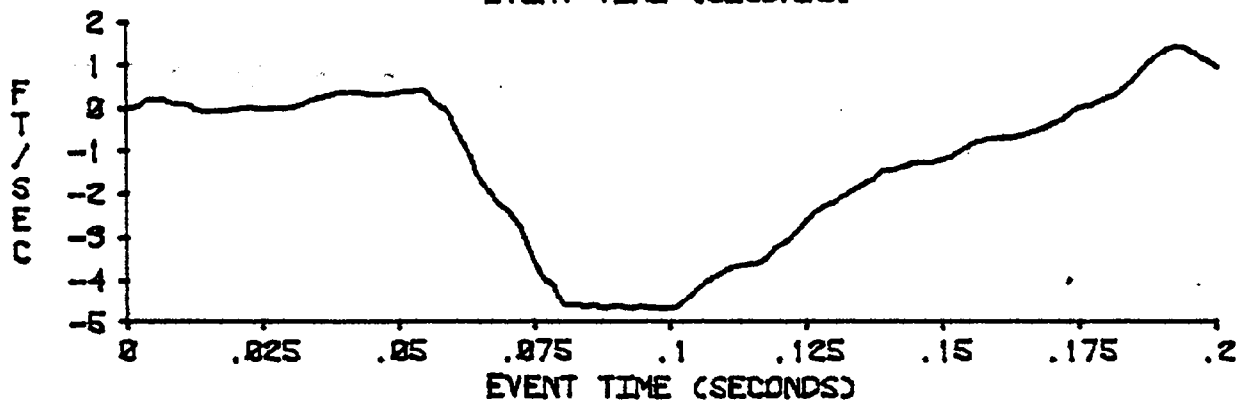
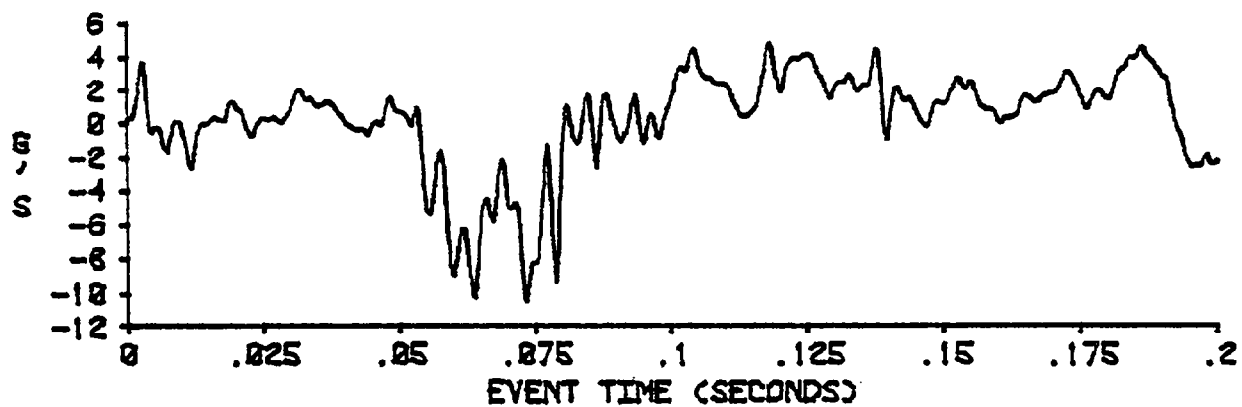
VEHICLE 2 - VW BUG - 33 MPH
DRIVER PELVIS AP
FILTER CLASS 180

TEST 1960
11 DEC 80



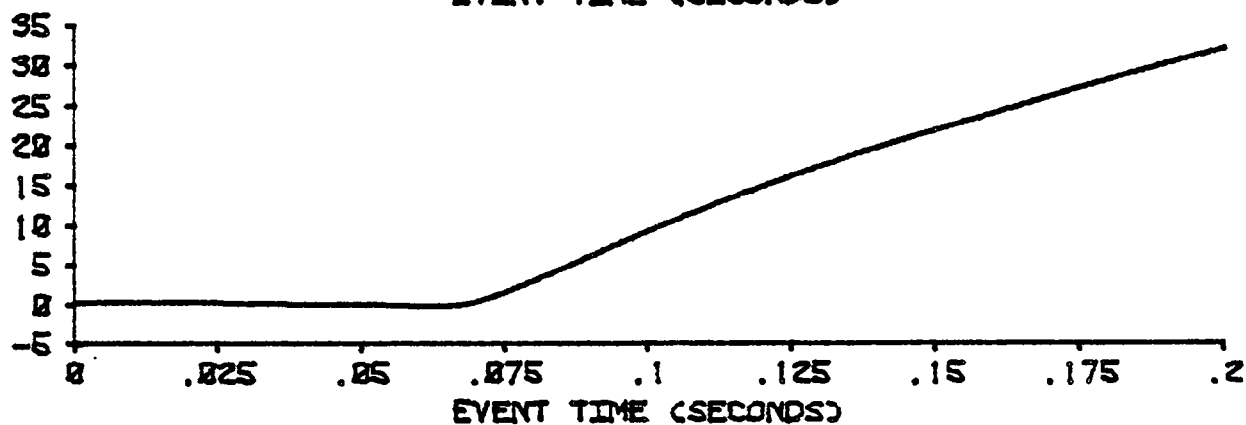
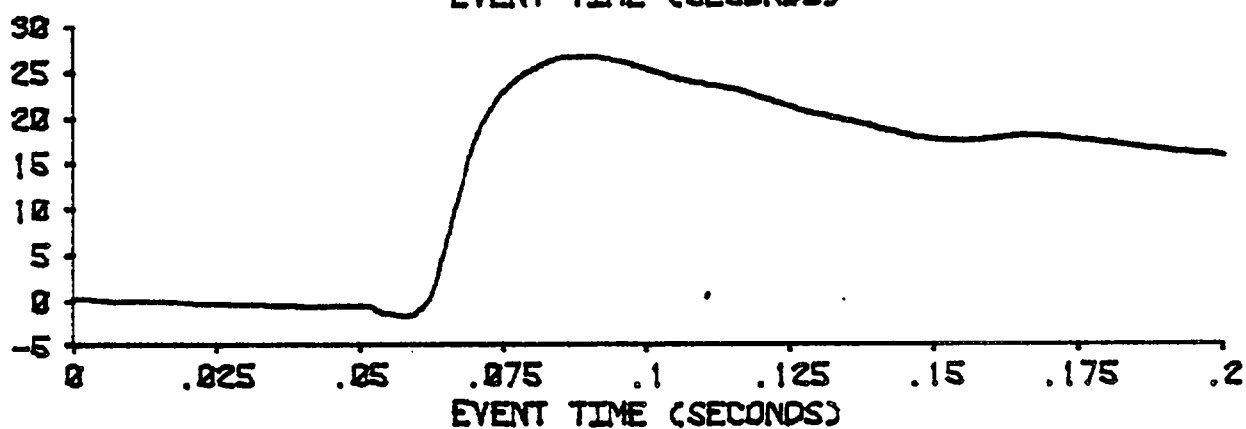
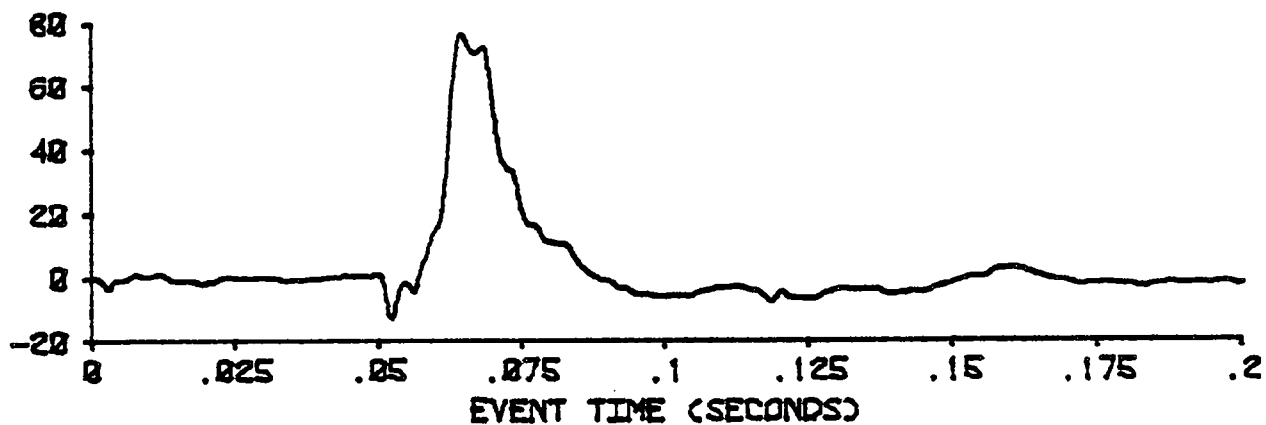
VEHICLE 2 - VW BUG - 33 MPH
DRIVER PELVIS IS
FILTER CLASS 180

TEST 1960
11 DEC 60



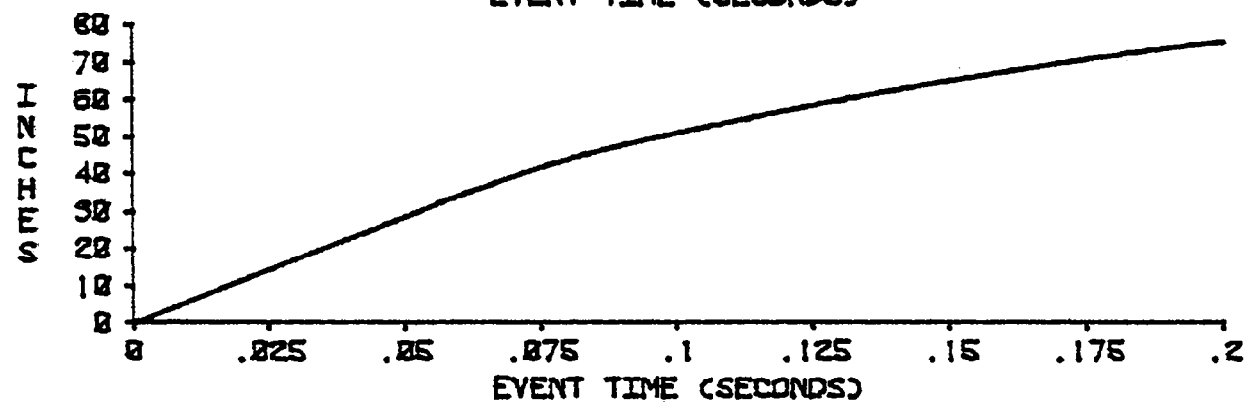
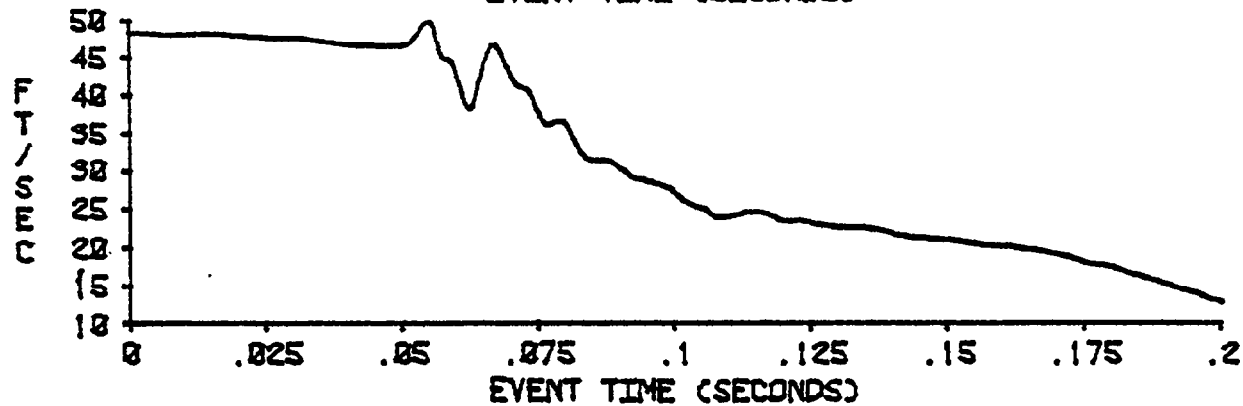
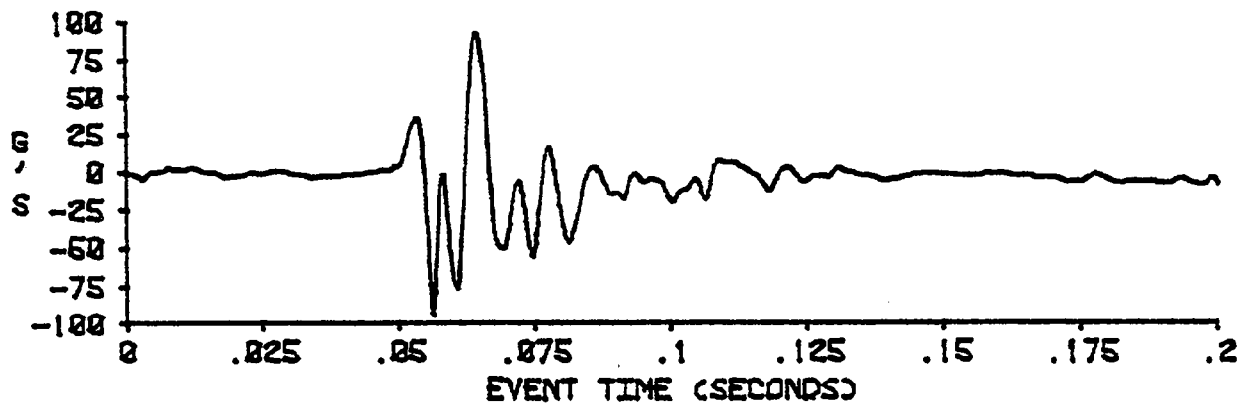
VEHICLE 2 - VW BUG - 33 MPH
DRIVER PELVIS RL
FILTER CLASS 180

TEST 1960
11 DEC 60



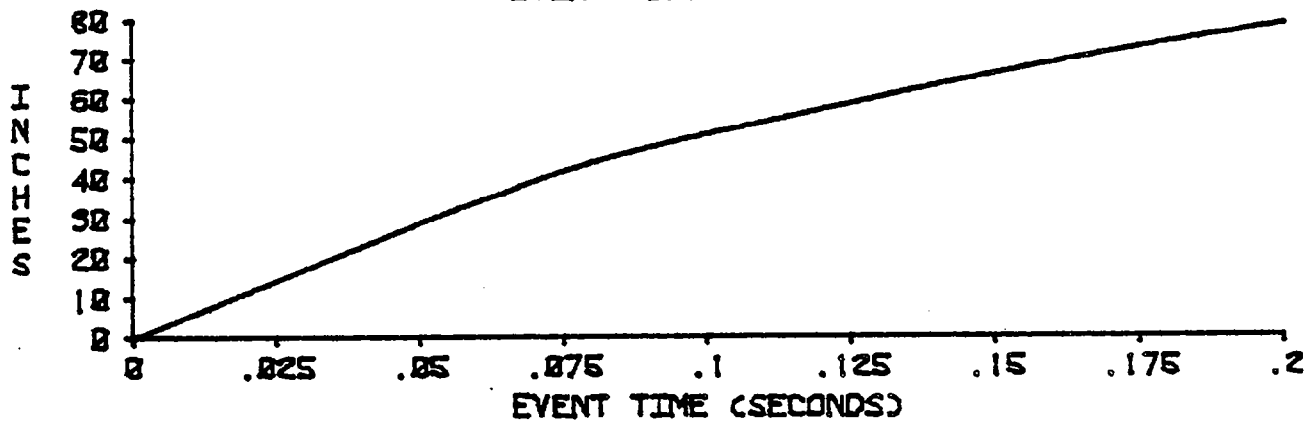
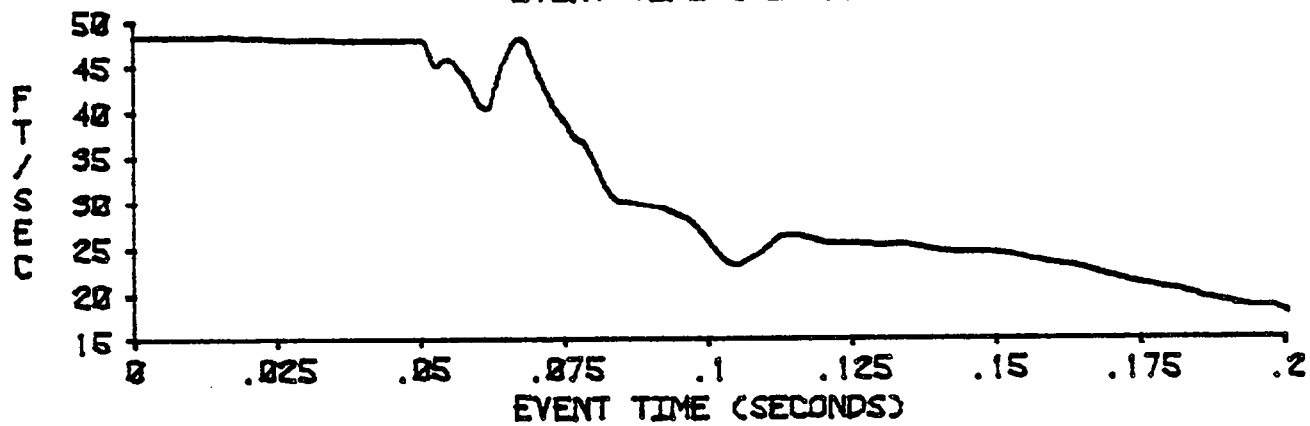
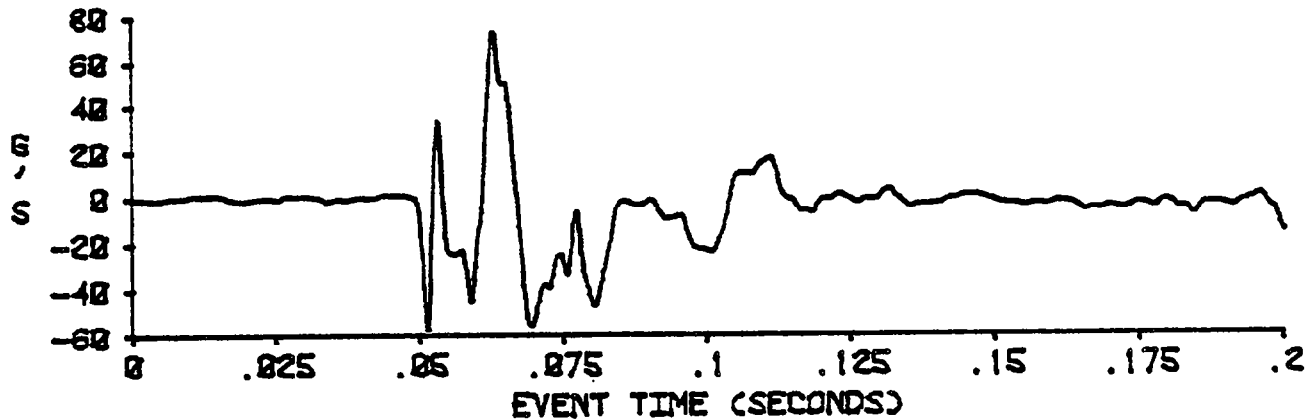
VEHICLE 2 - VW BUG - 33 MPH
DRIVER UPPER STERNAL AP
FILTER CLASS 160

TEST 1960
11 DEC 60



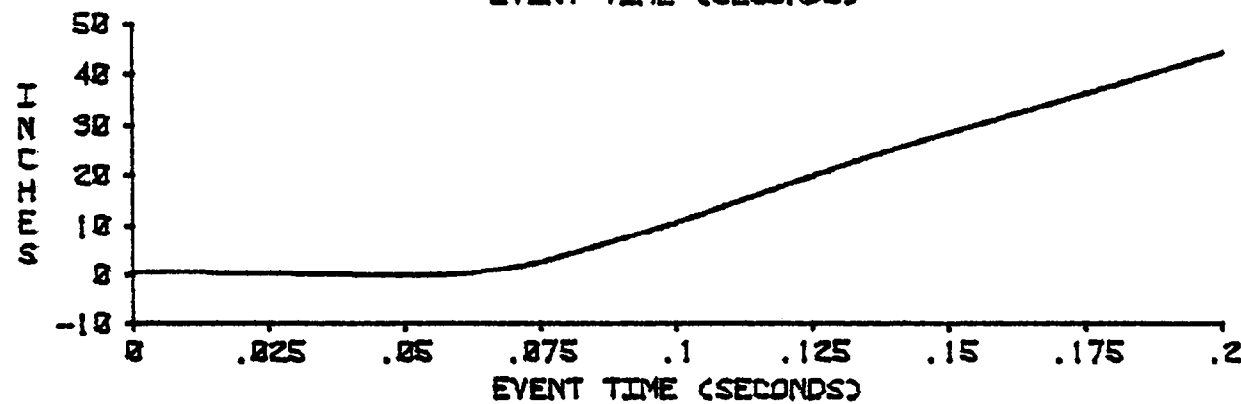
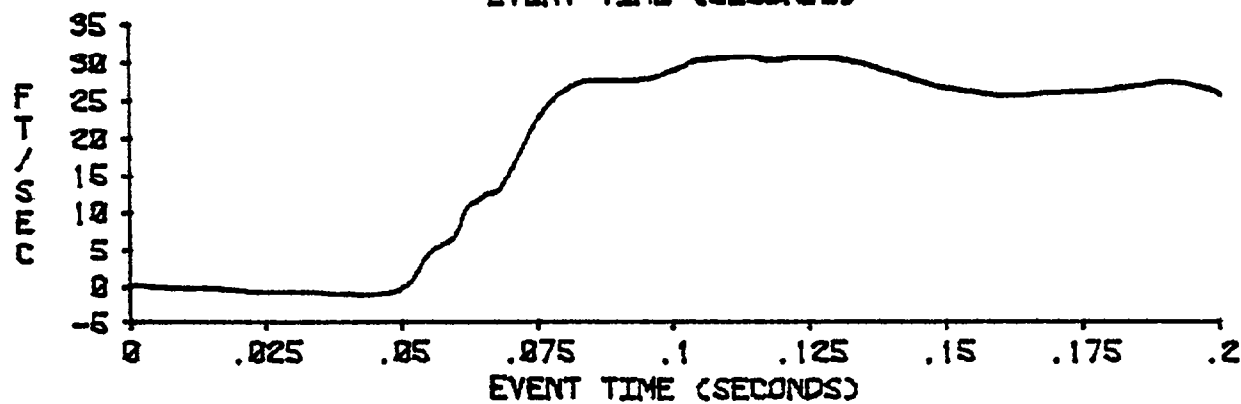
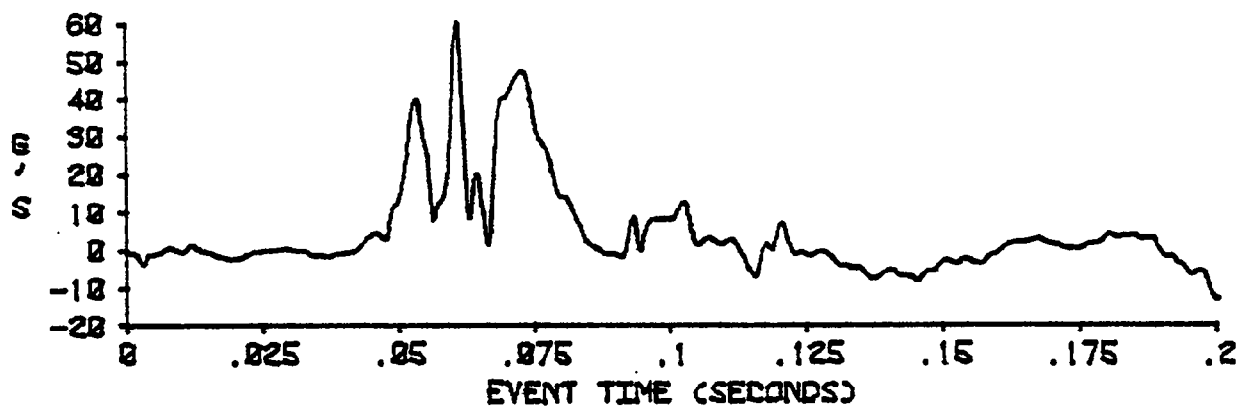
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LOWER STERNAL AP
FILTER CLASS 180

TEST 1960
11 DEC 60



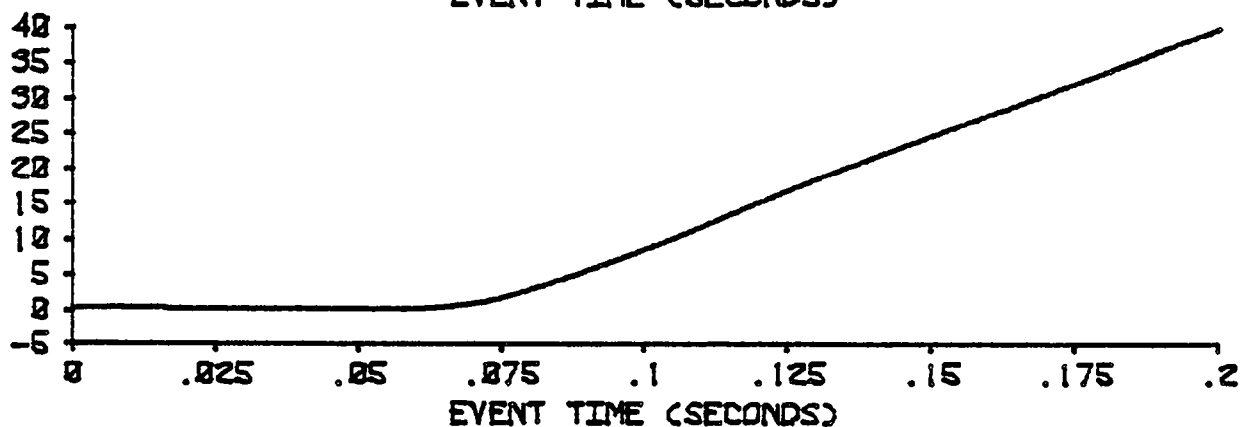
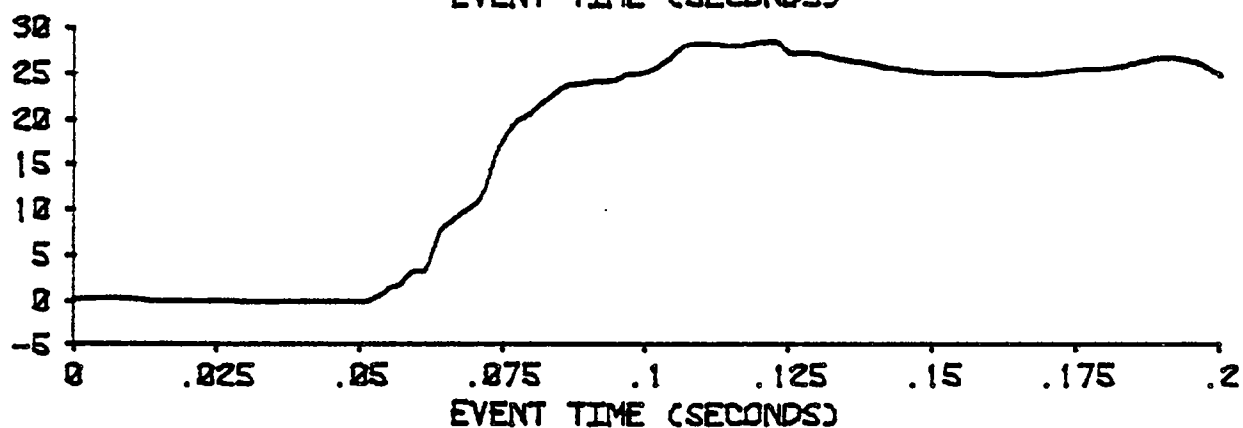
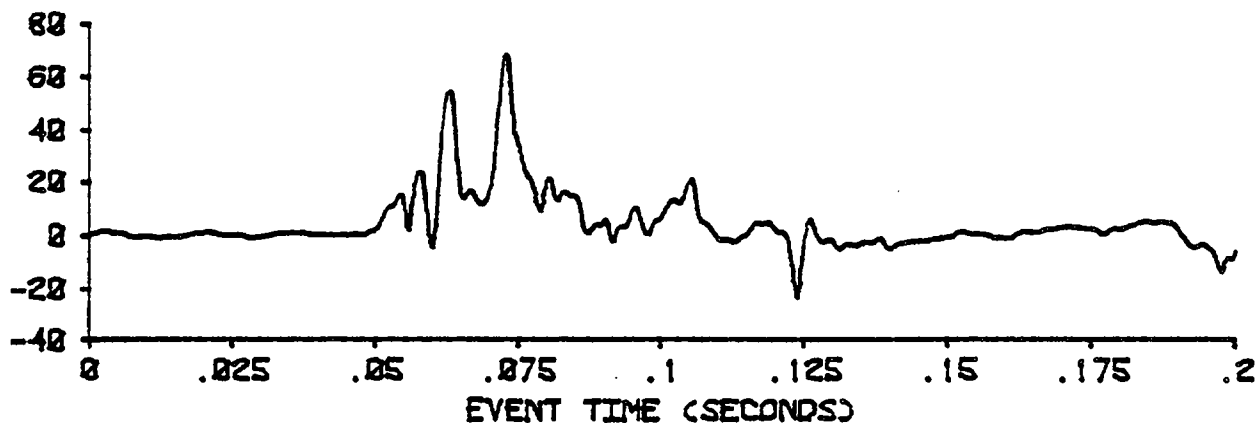
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LEFT UPPER RIB RL
FILTER CLASS 100

TEST 1960
11 DEC 60



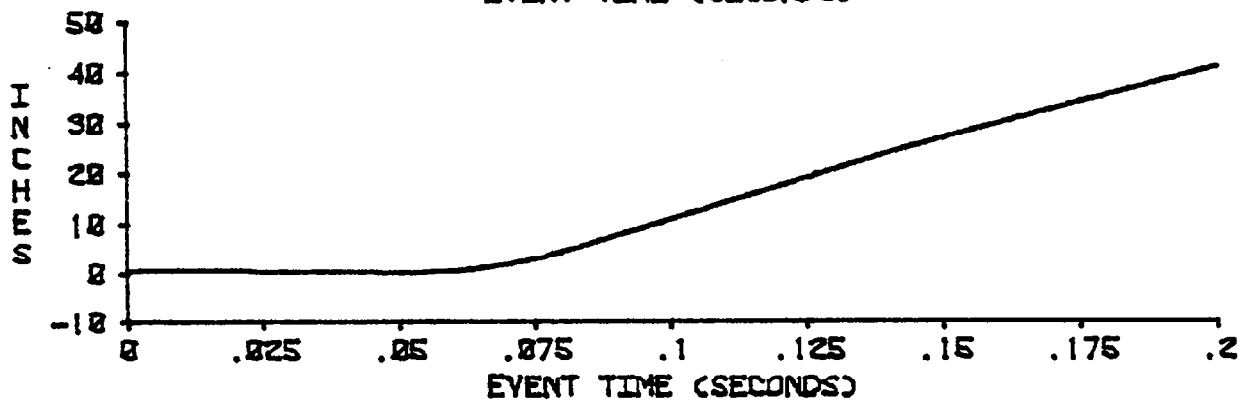
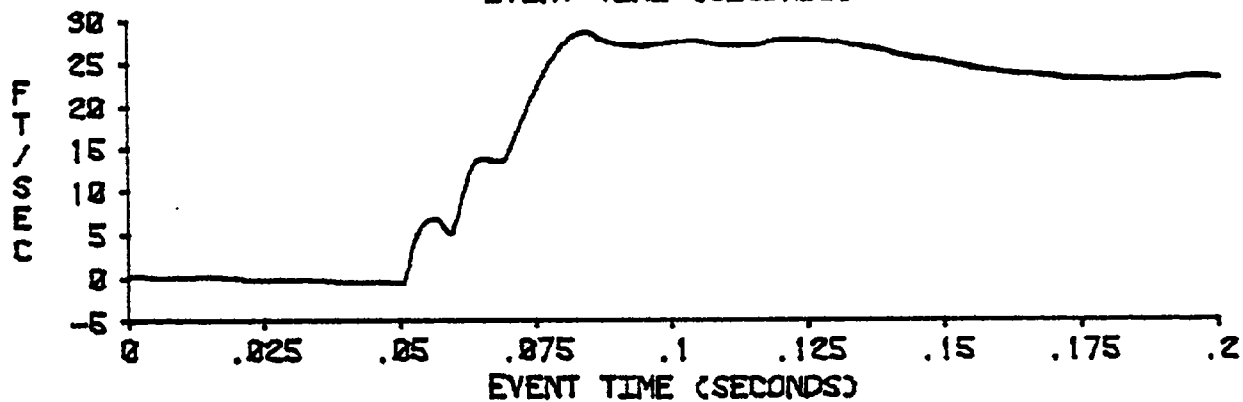
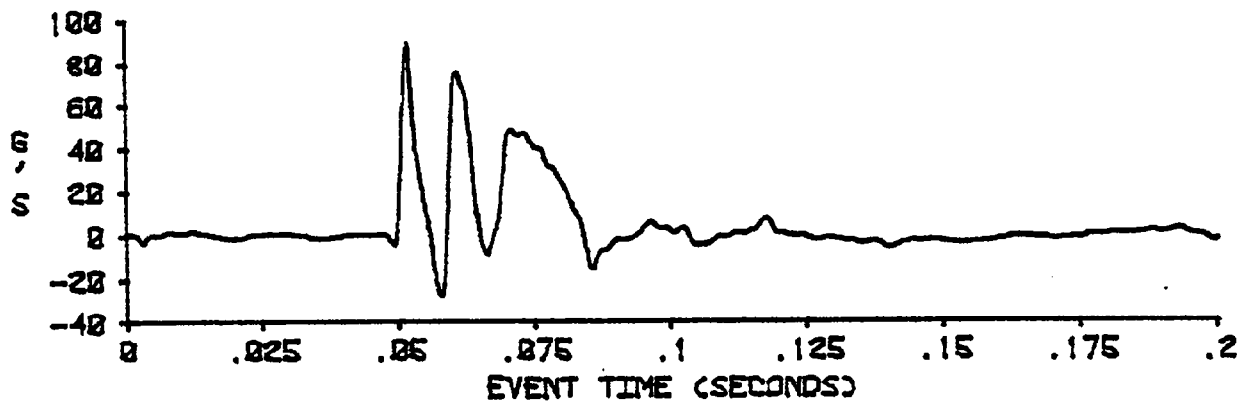
VEHICLE 2 - VW BUG - 33 MPH
DRIVER RIGHT UPPER RIB RL
FILTER CLASS 180

TEST 1860
11 DEC 80



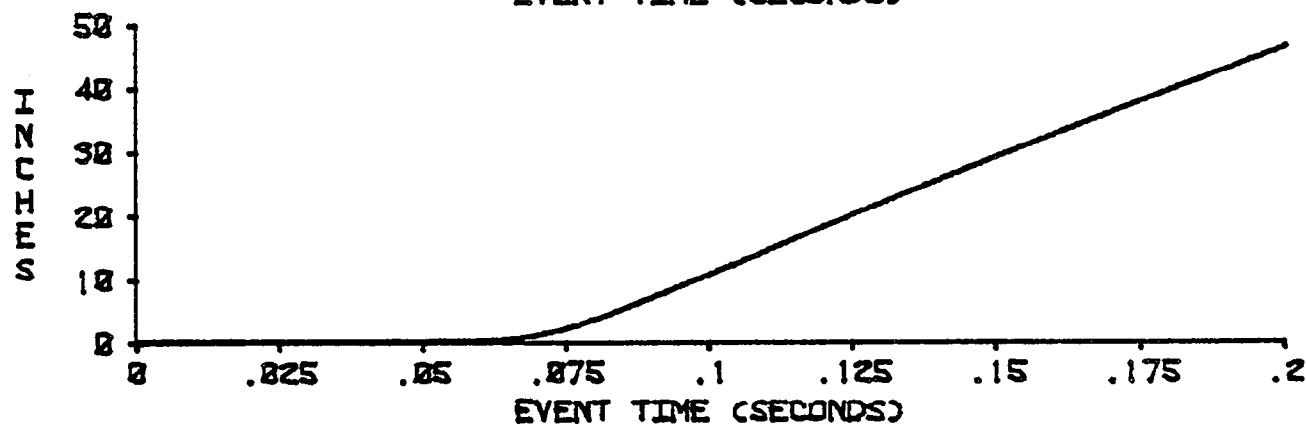
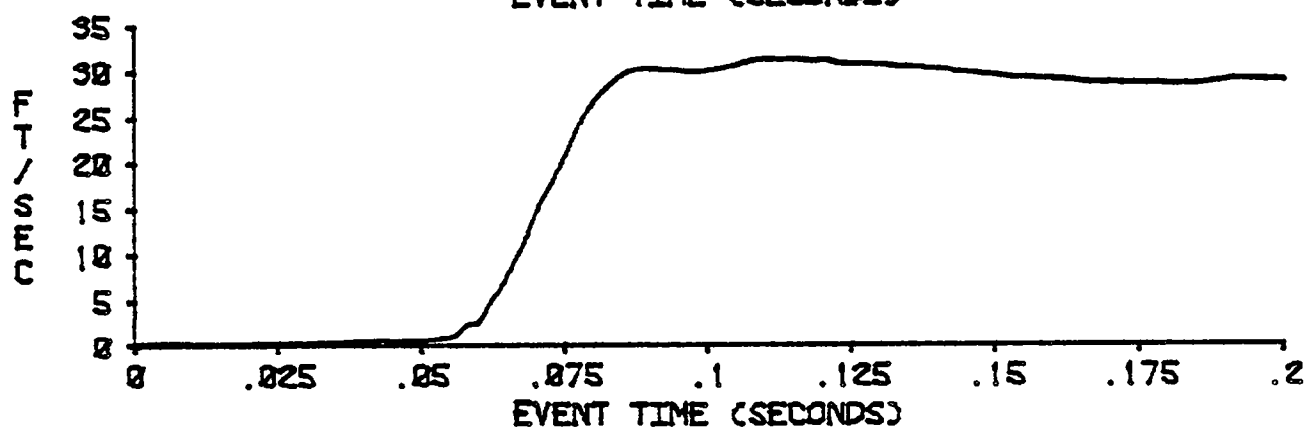
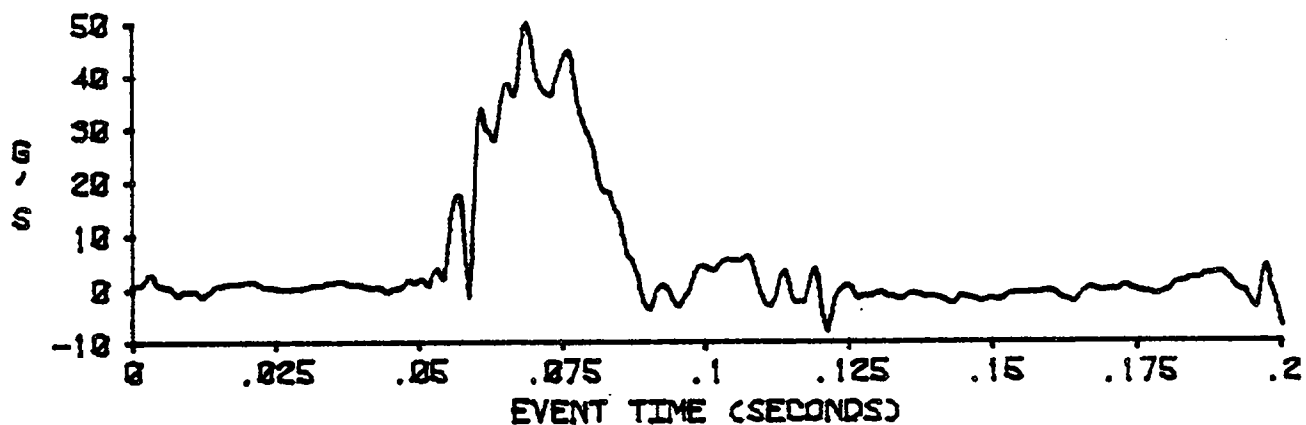
VEHICLE 2 - VW BUG - 33 MPH
DRIVER LEFT LOWER RIB RL
FILTER CLASS 168

TEST 1960
11 DEC 80



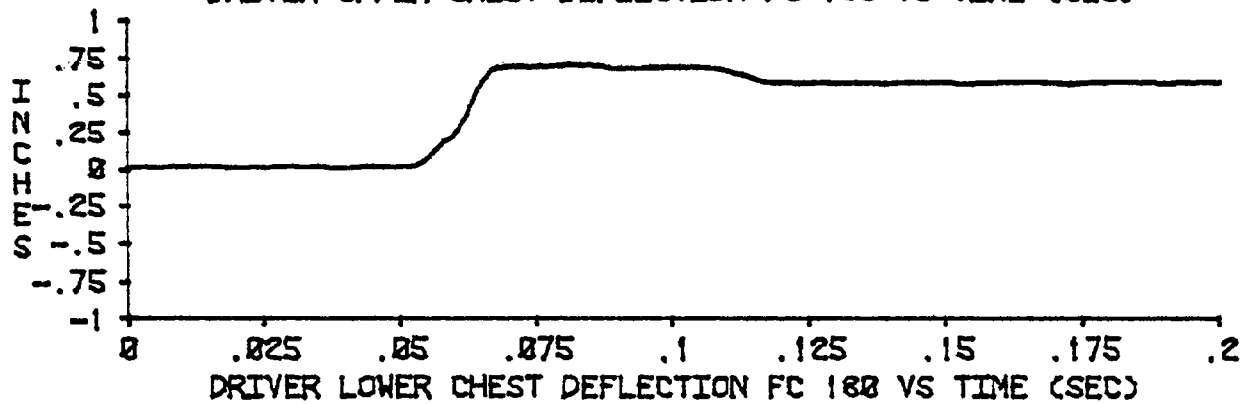
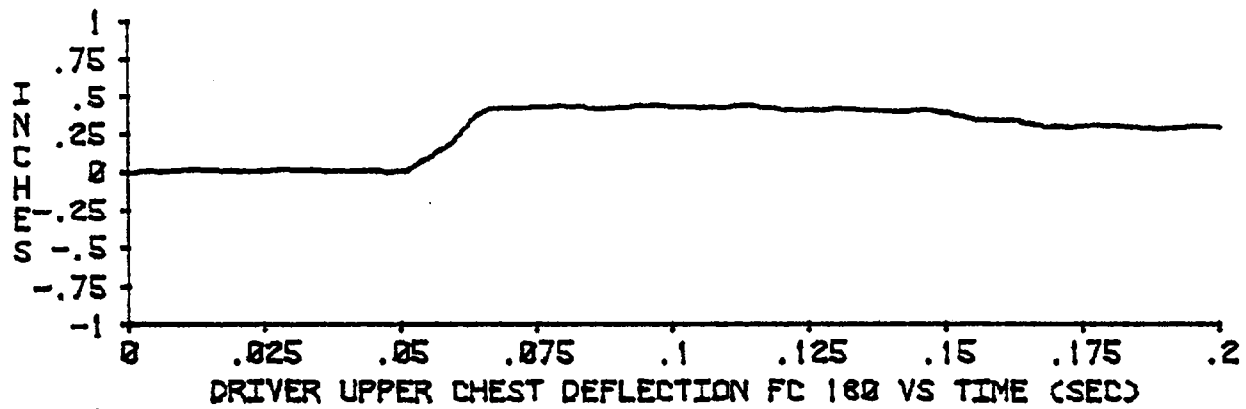
VEHICLE 2 - VW BUG - 33 MPH
DRIVER RIGHT LOWER RIB RL
FILTER CLASS 180

TEST 1960
11 DEC 60



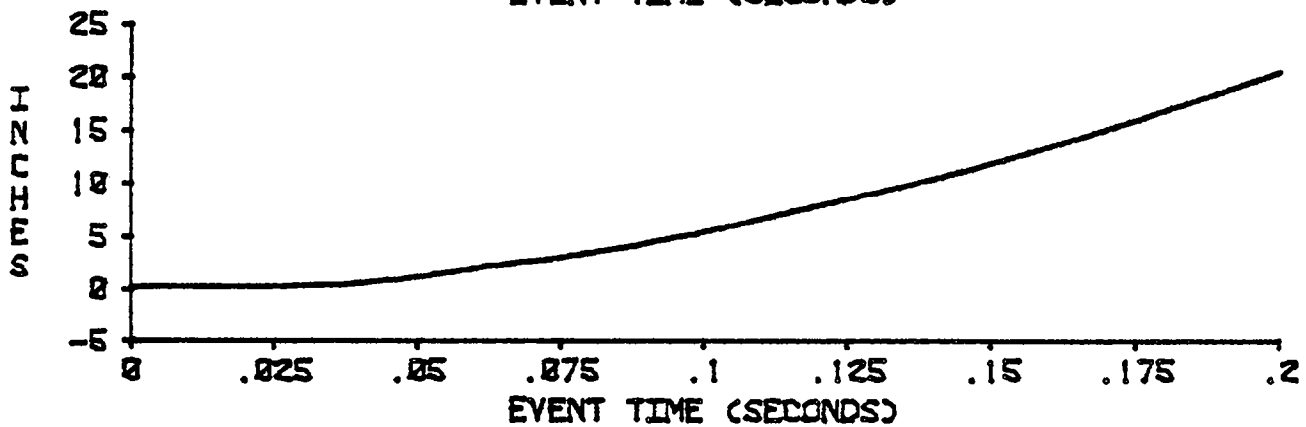
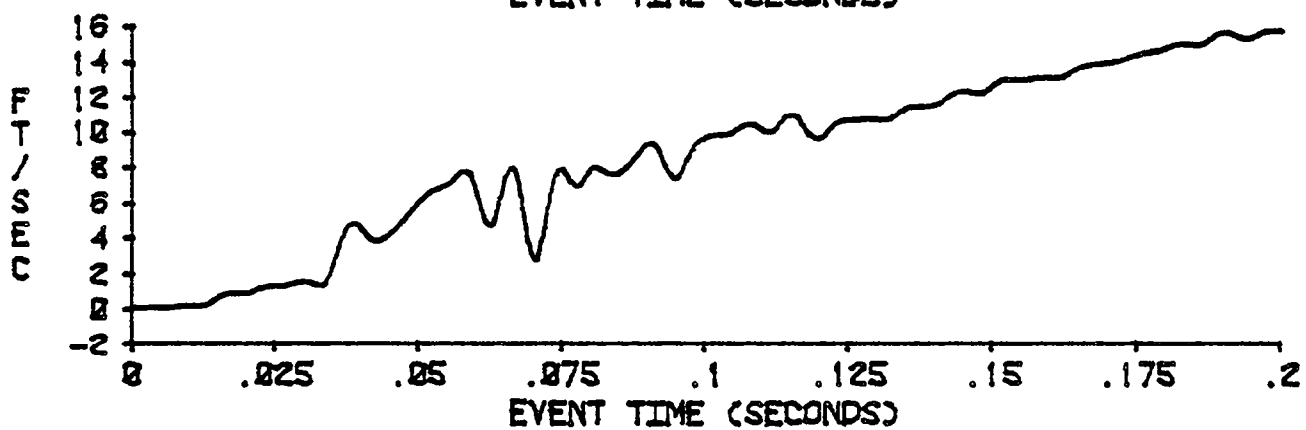
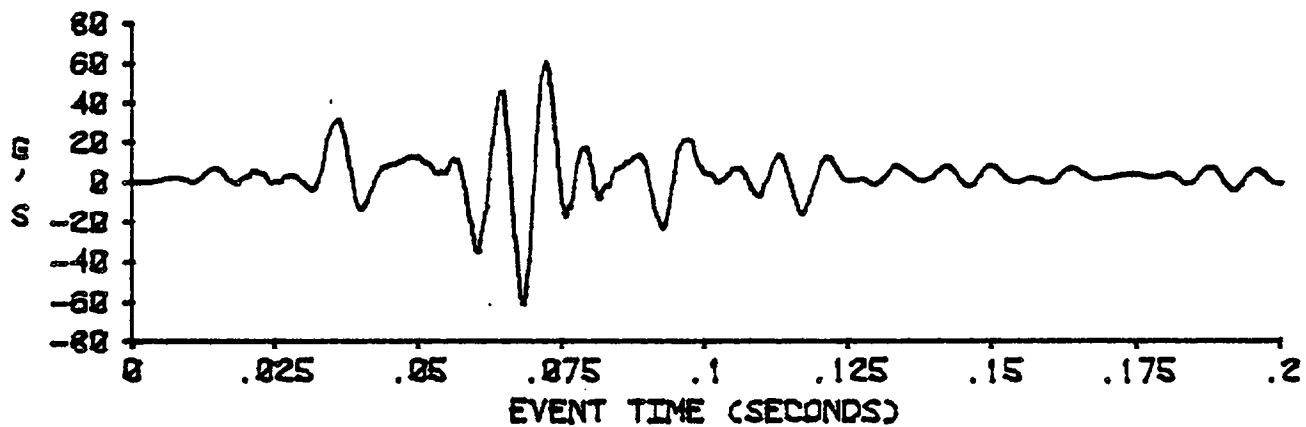
VEHICLE 2 - VW BUG - 33 MPH

TEST 1960
11 DEC 60



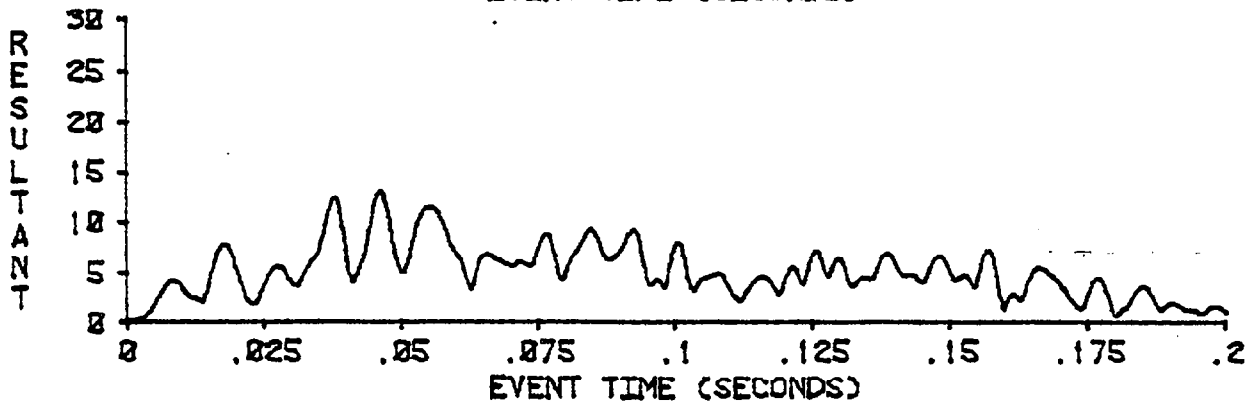
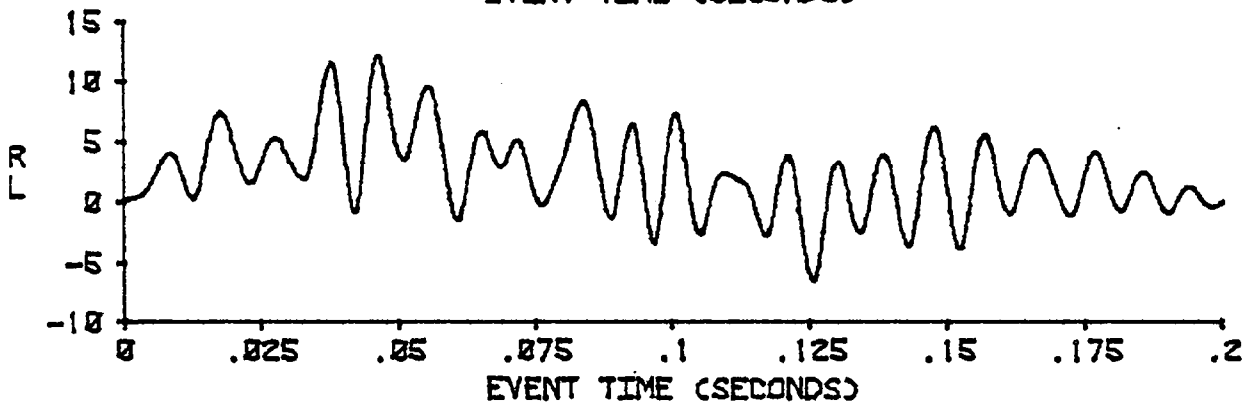
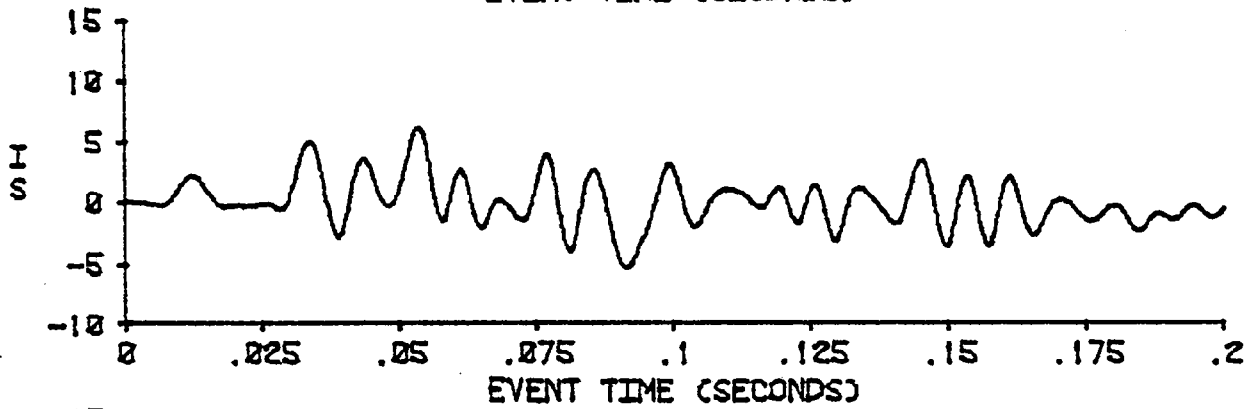
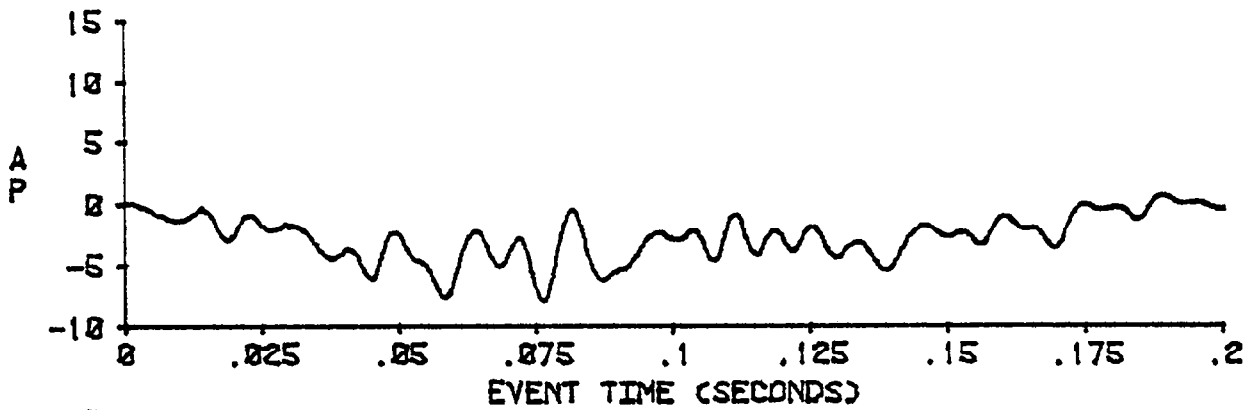
VEHICLE 2 - VW BUG - 33 MPH
LEFT REAR ROCKER PANEL RL
FILTER CLASS 60

TEST 1960
11 DEC 60



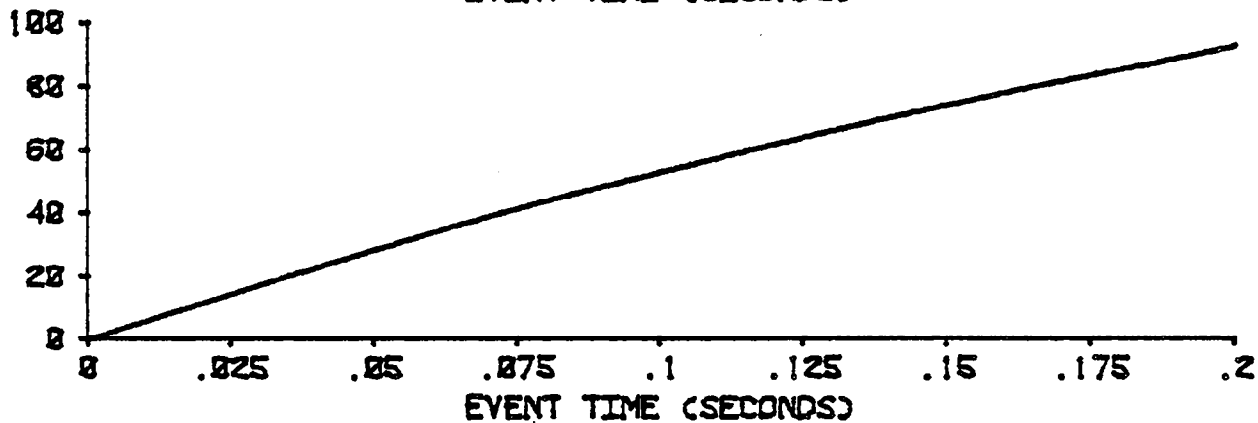
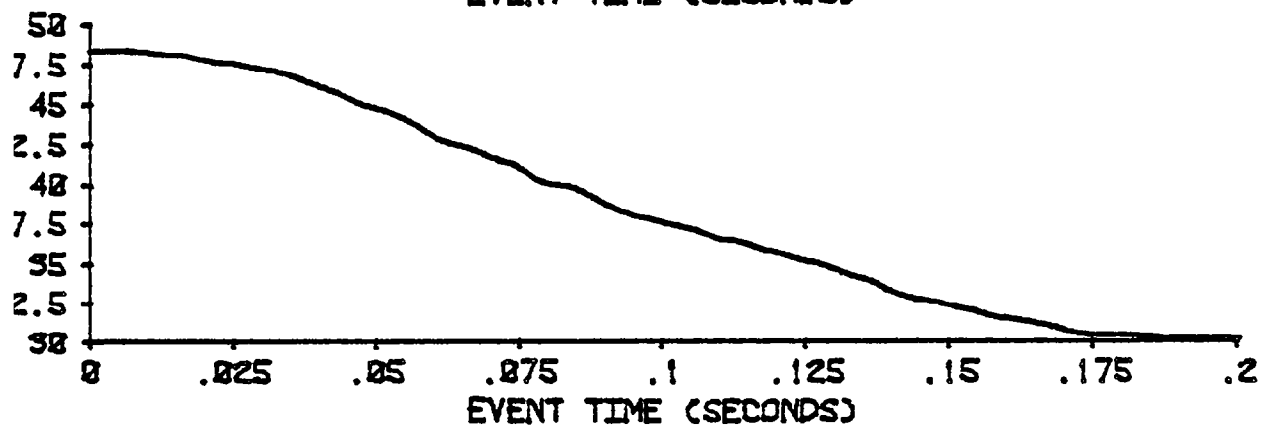
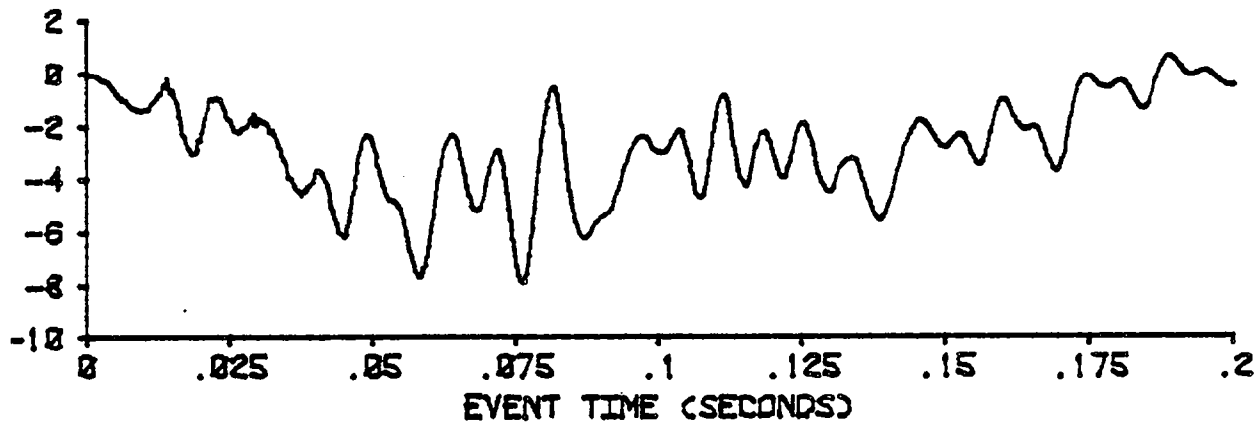
VEHICLE 2 - VW BUG - 33 MPH
RIGHT FRONT ROCKER PANEL G'S
FILTER CLASS 60

TEST 1960
11 DEC 80



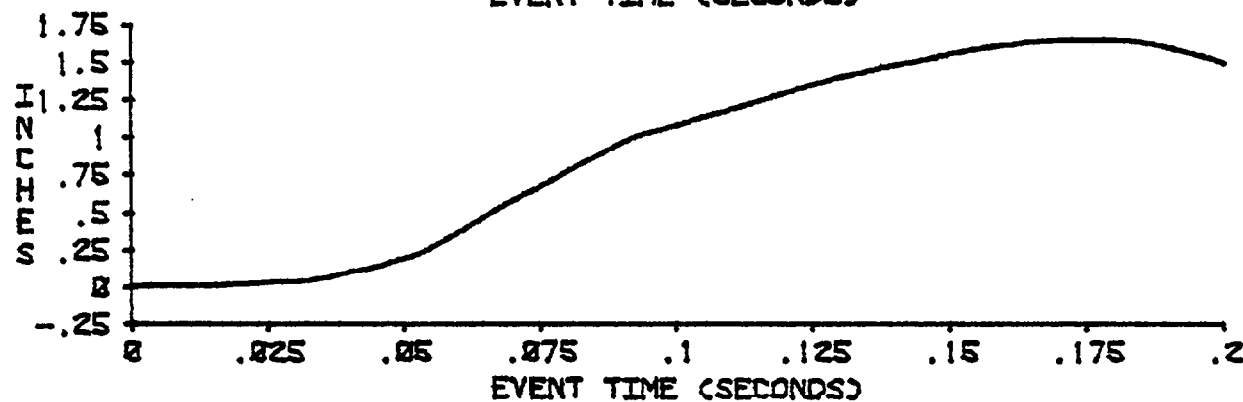
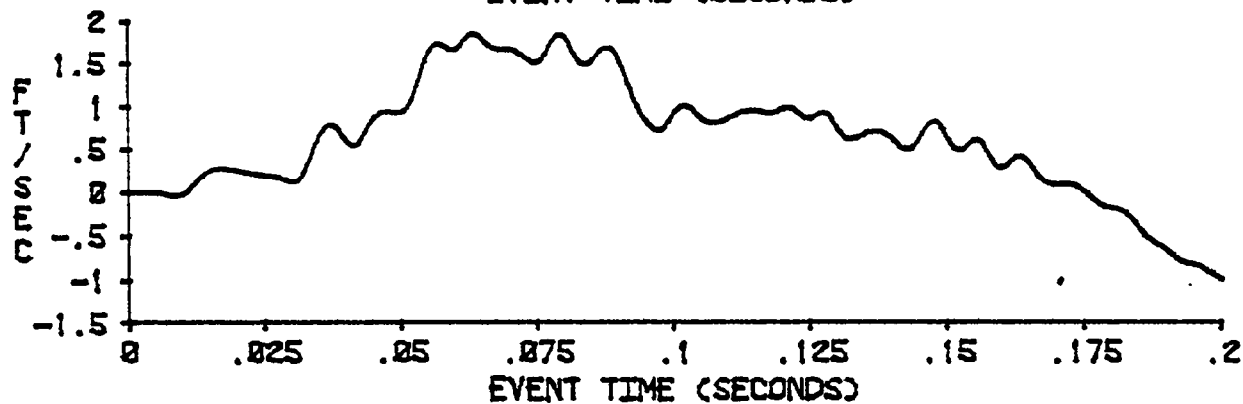
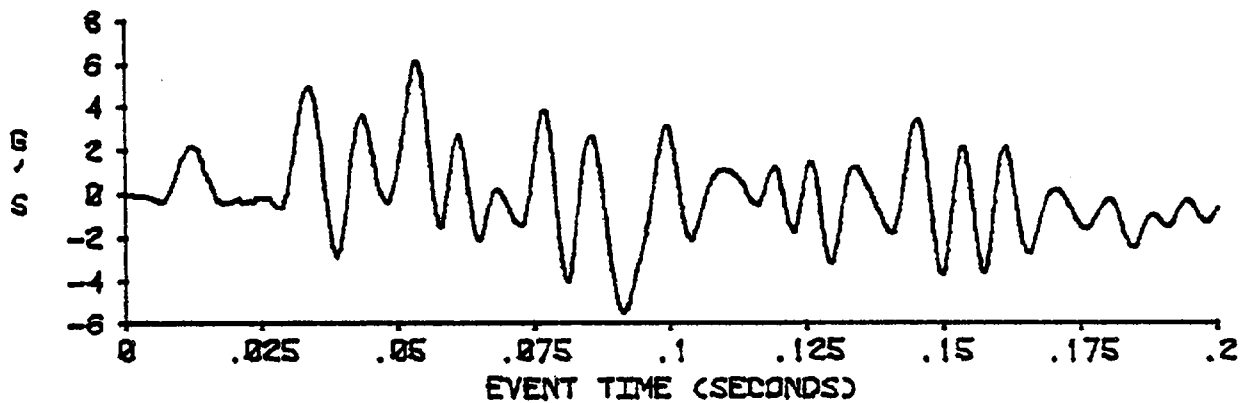
VEHICLE 2 - VW BUG - 33 MPH
RIGHT FRONT ROCKER PANEL AP
FILTER CLASS 60

TEST 1960
11 DEC 60



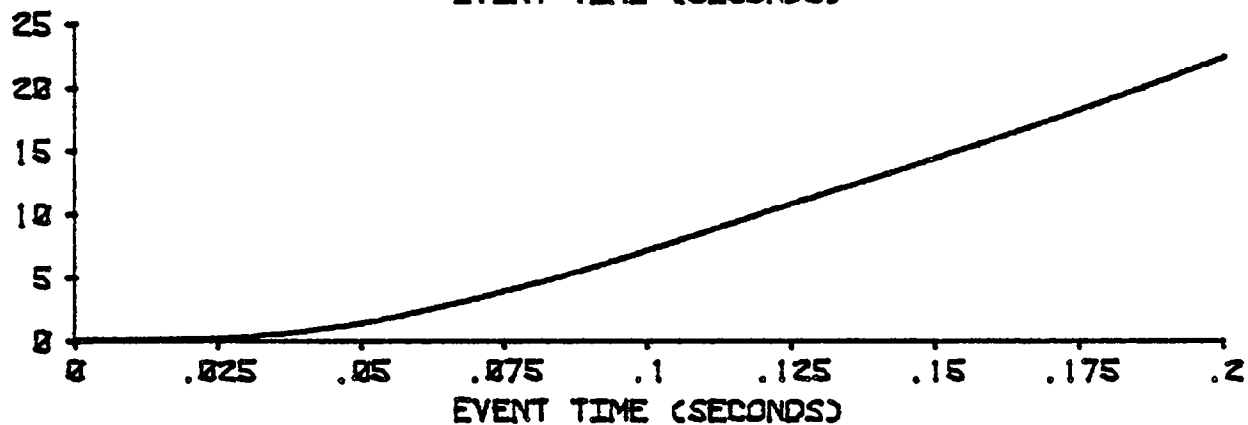
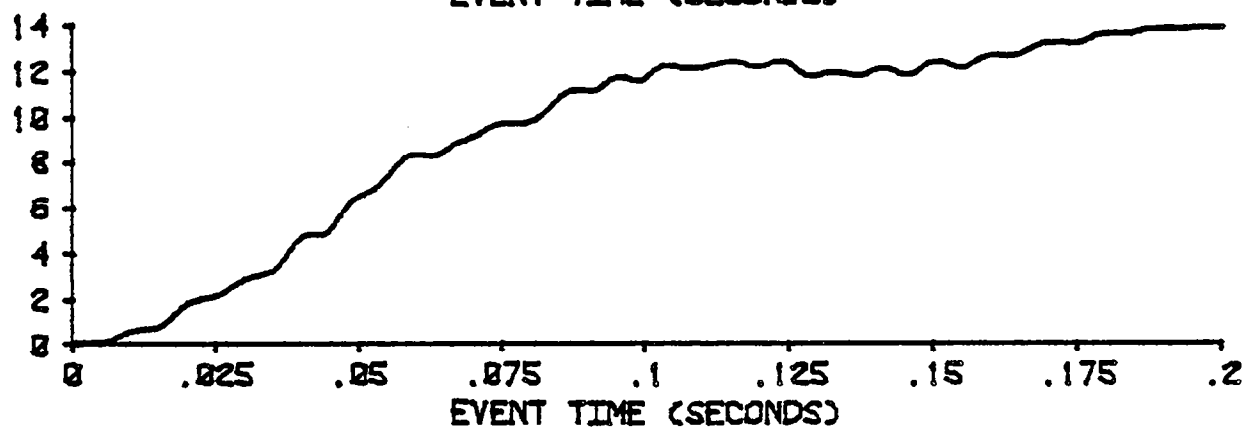
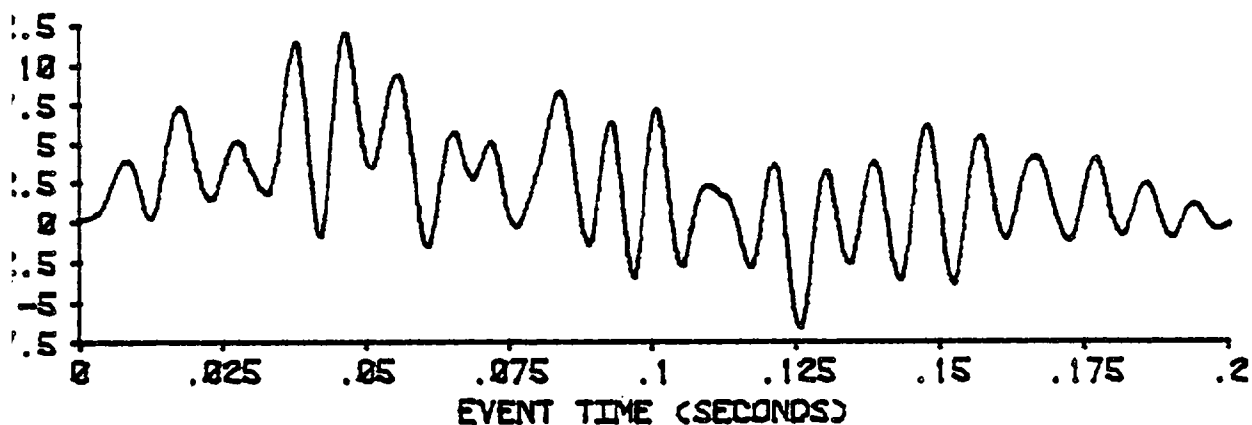
VEHICLE 2 - VW BUG - 33 MPH
RIGHT FRONT ROCKER PANEL IS
FILTER CLASS 60

TEST 1960
11 DEC 60



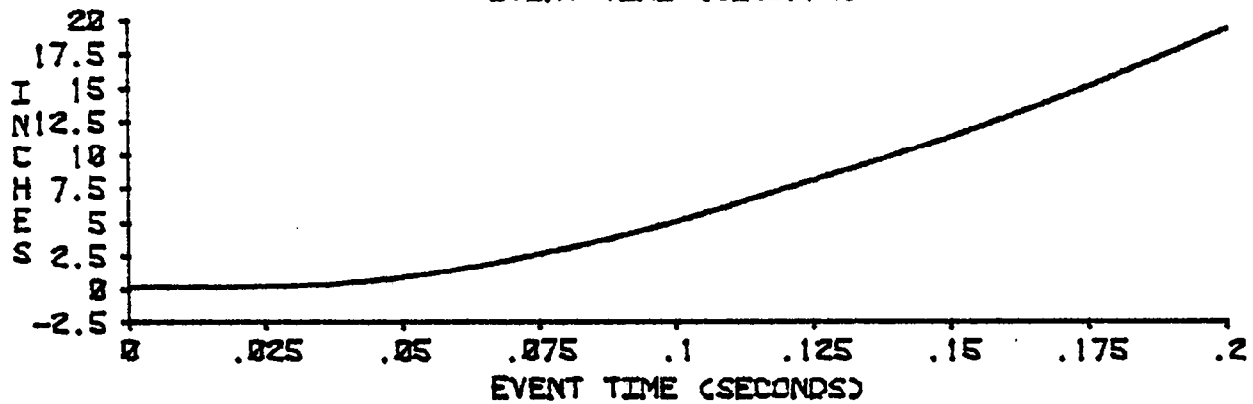
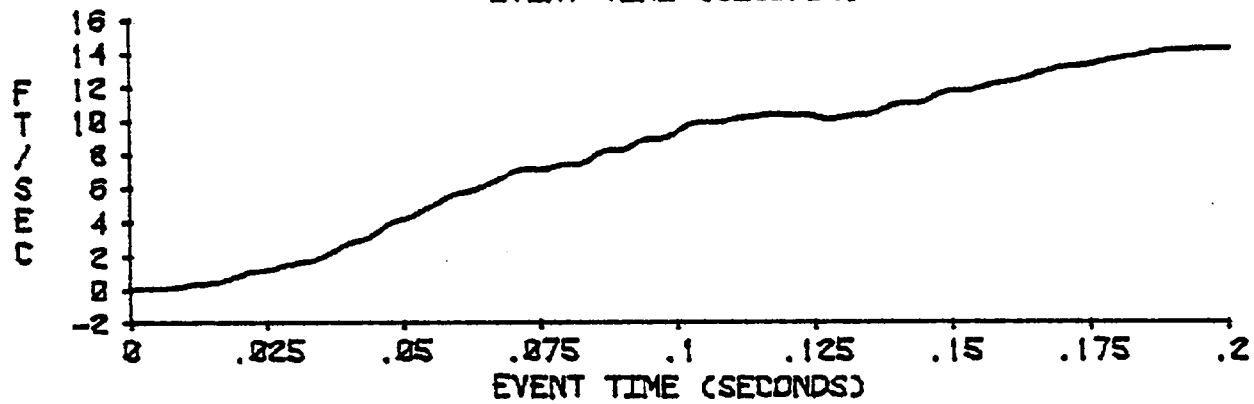
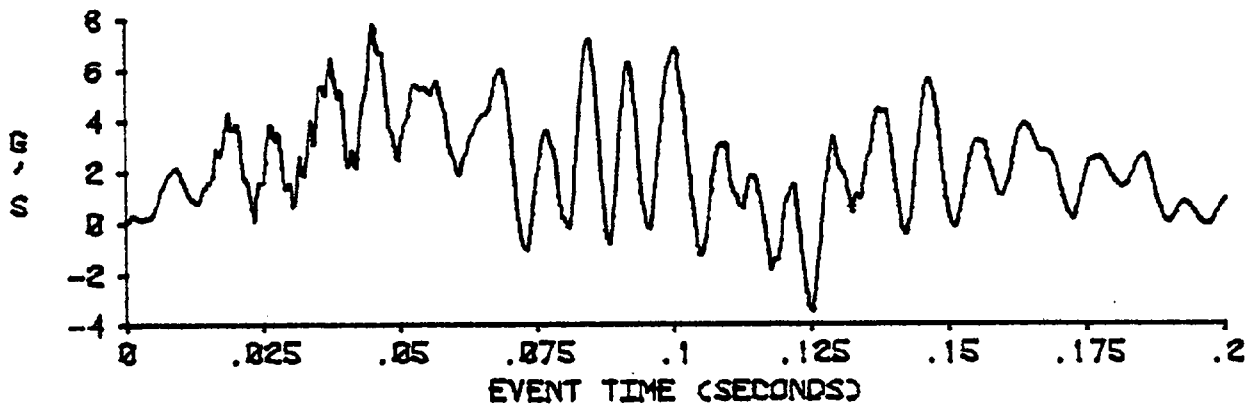
VEHICLE 2 - VW BUG - 33 MPH
RIGHT FRONT ROCKER PANEL RL
FILTER CLASS 60

TEST 1960
11 DEC 60



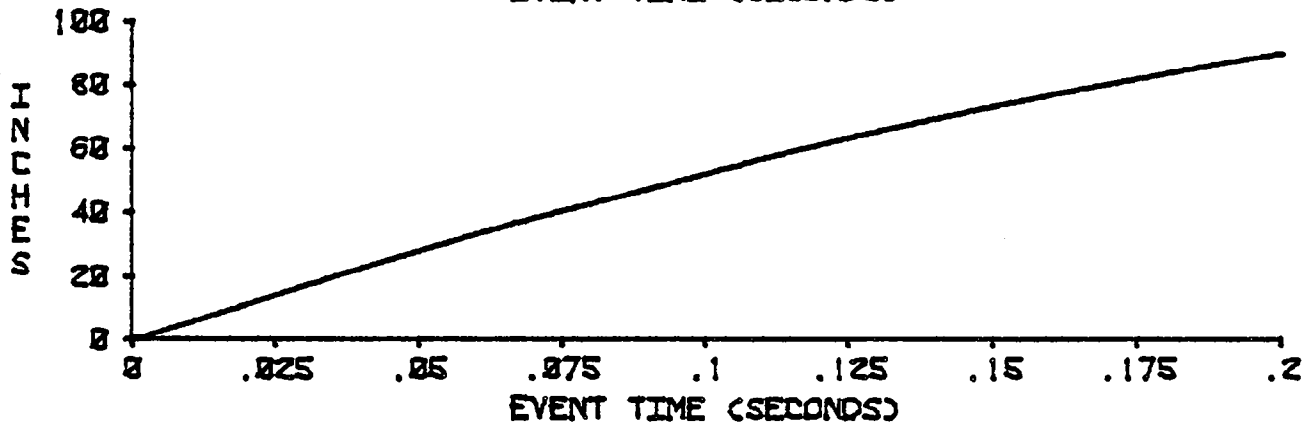
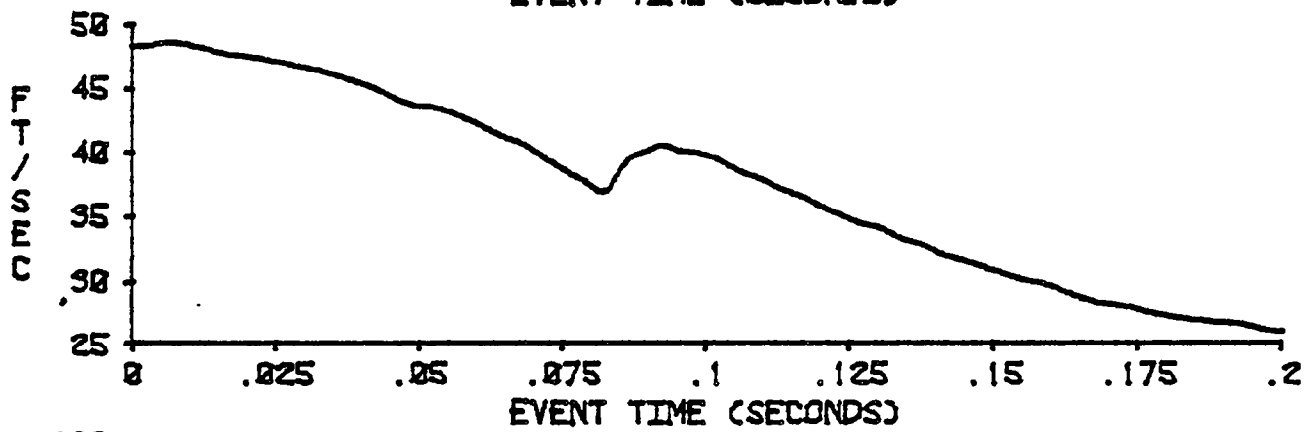
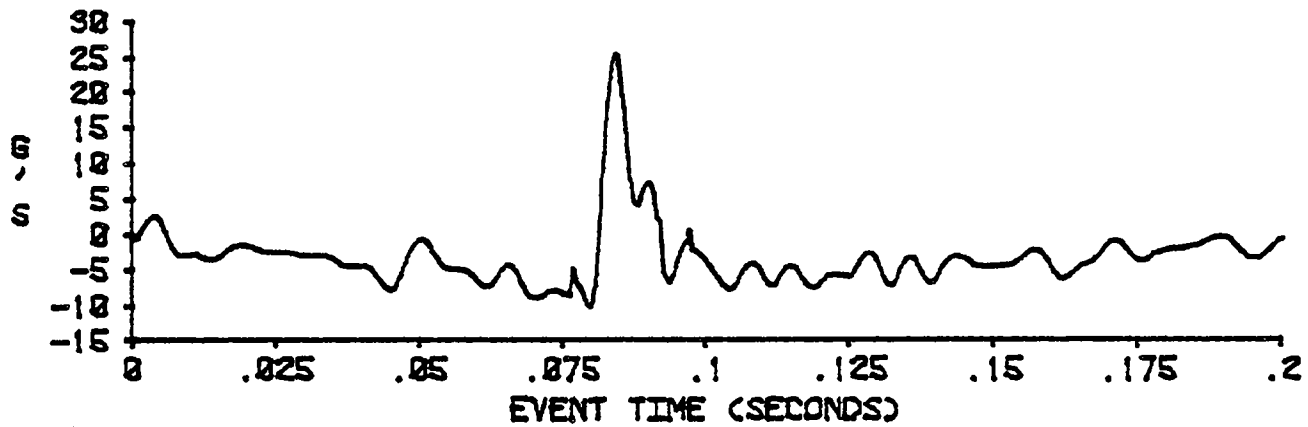
VEHICLE 2 - VW BUG - 33 MPH
RIGHT REAR ROCKER PANEL RL
FILTER CLASS 68

TEST 1968
11 DEC 68



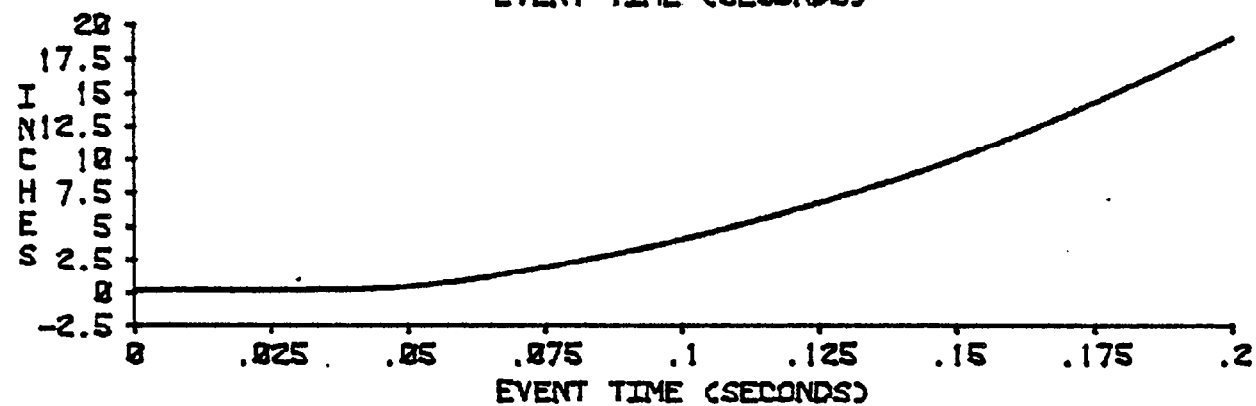
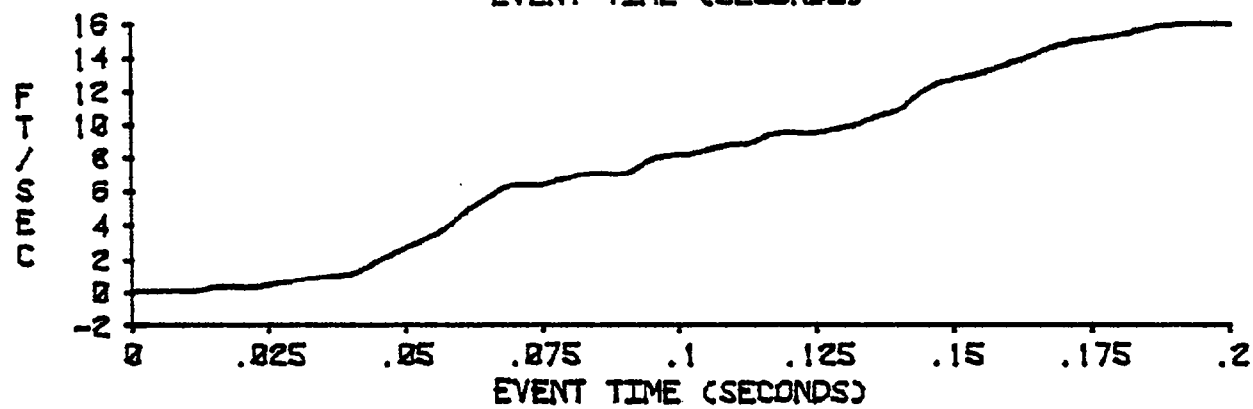
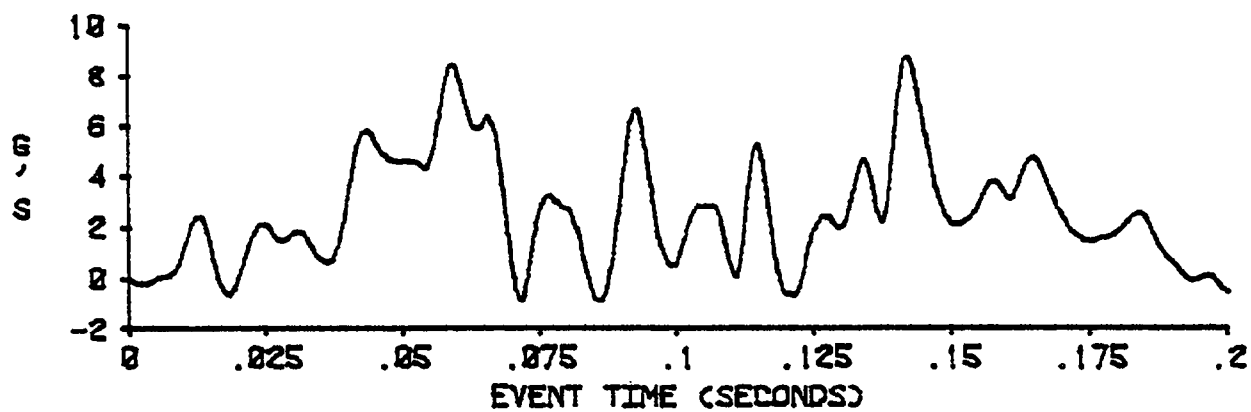
VEHICLE 2 - VW BUG - 33 MPH
CAR MID TRUNK AP
FILTER CLASS 60

TEST 1960
11 DEC 60



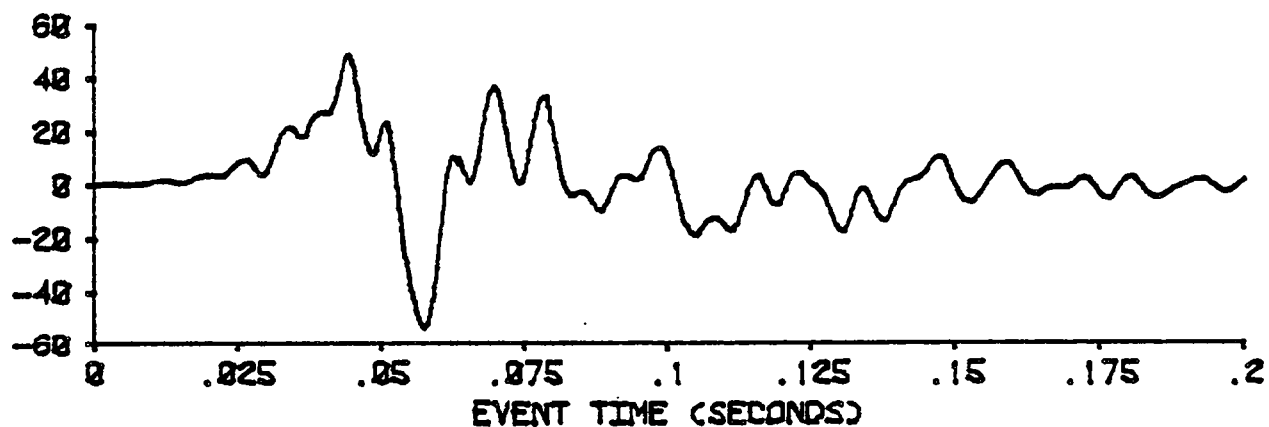
VEHICLE 2 - VW BUG - 33 MPH
CAR MID TRUNK RL
FILTER CLASS 68

TEST 1968
11 DEC 68

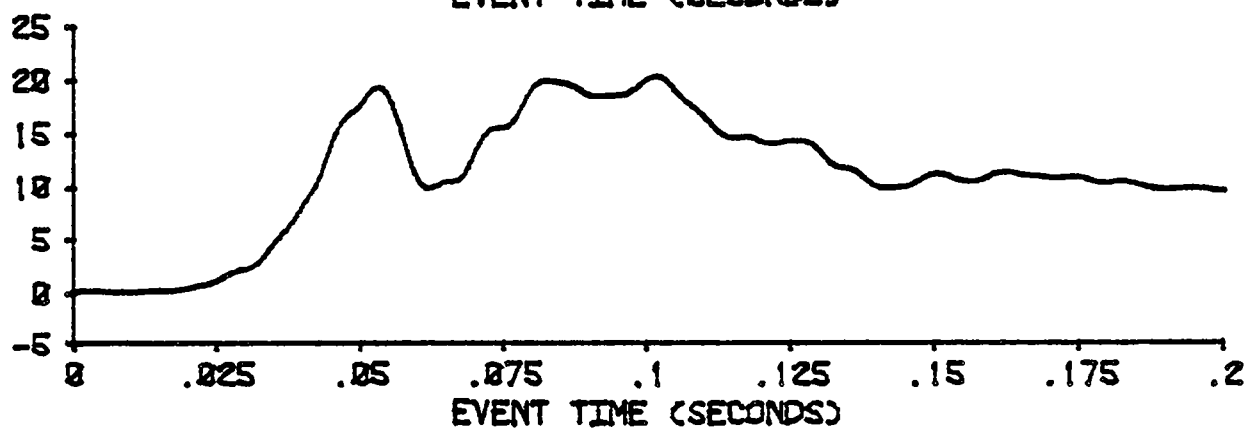


VEHICLE 2 - VW BUG - 33 MPH
LEFT DOOR UPPER RL
FILTER CLASS 60

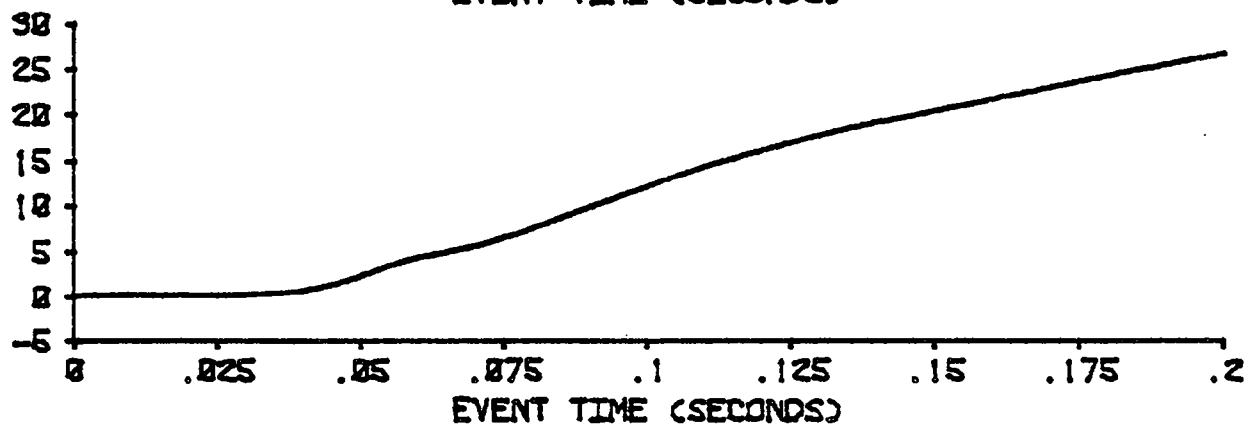
TEST 1960
11 DEC 68



EVENT TIME (SECONDS)



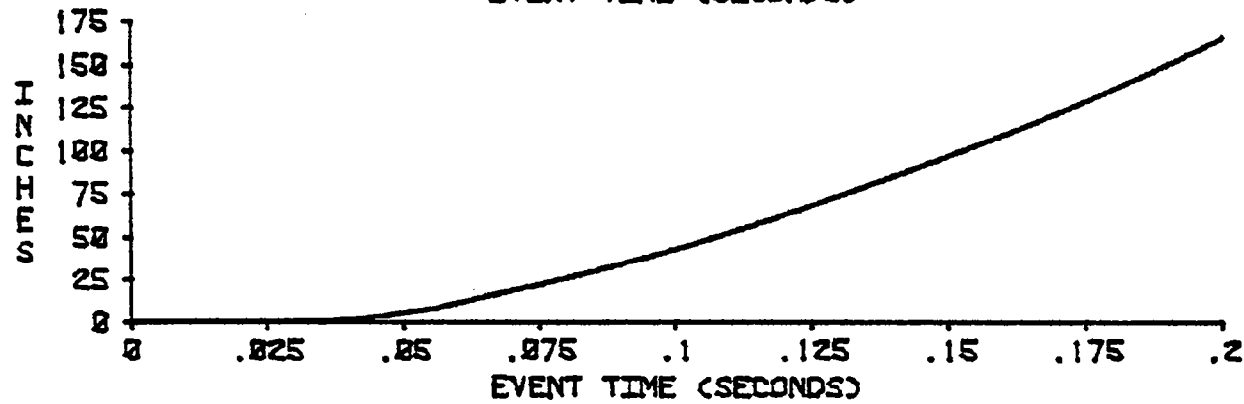
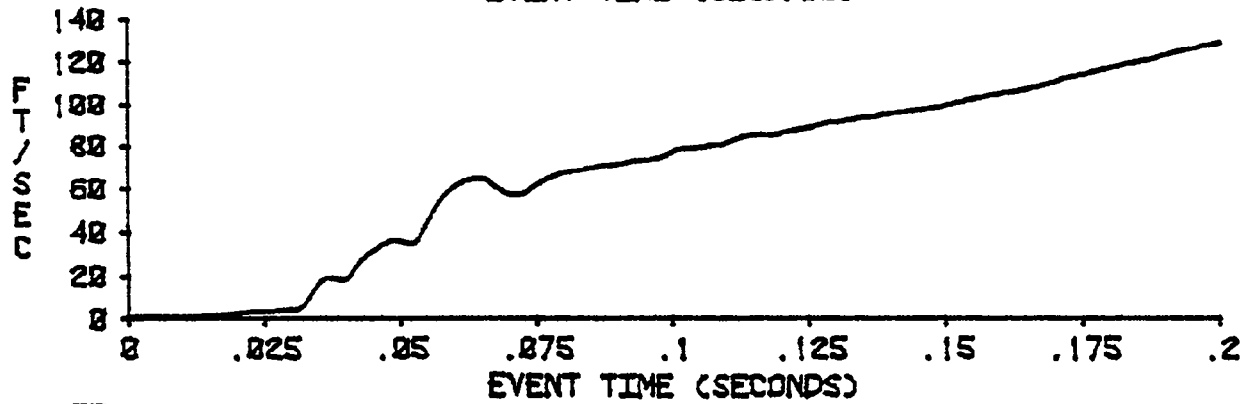
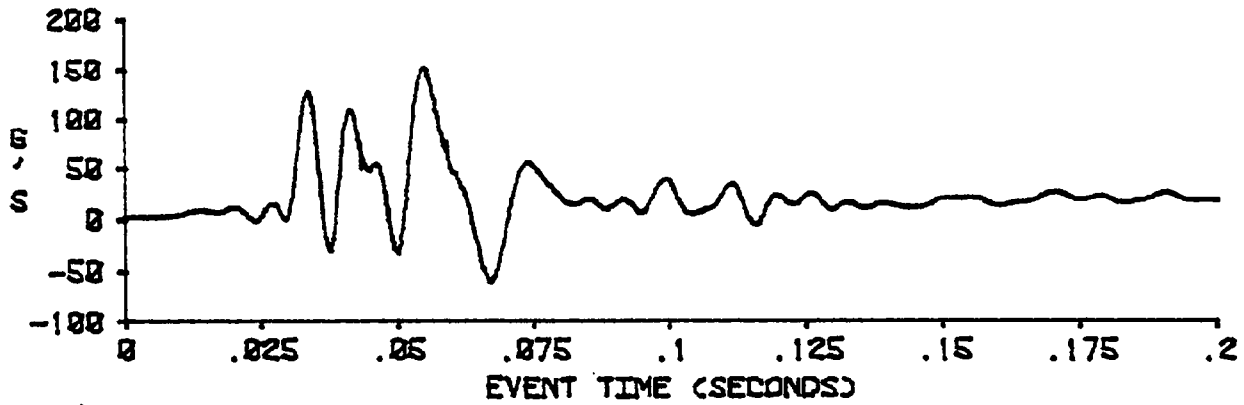
EVENT TIME (SECONDS)



EVENT TIME (SECONDS)

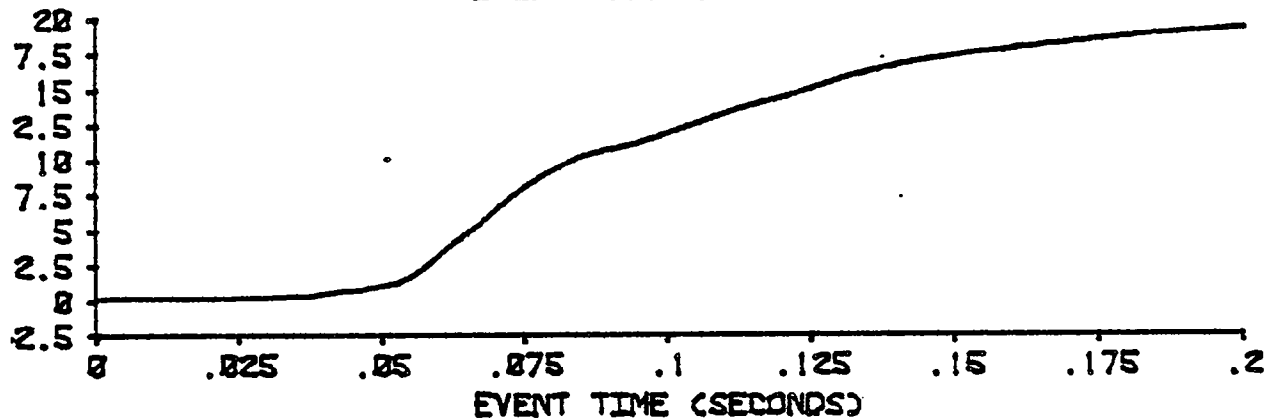
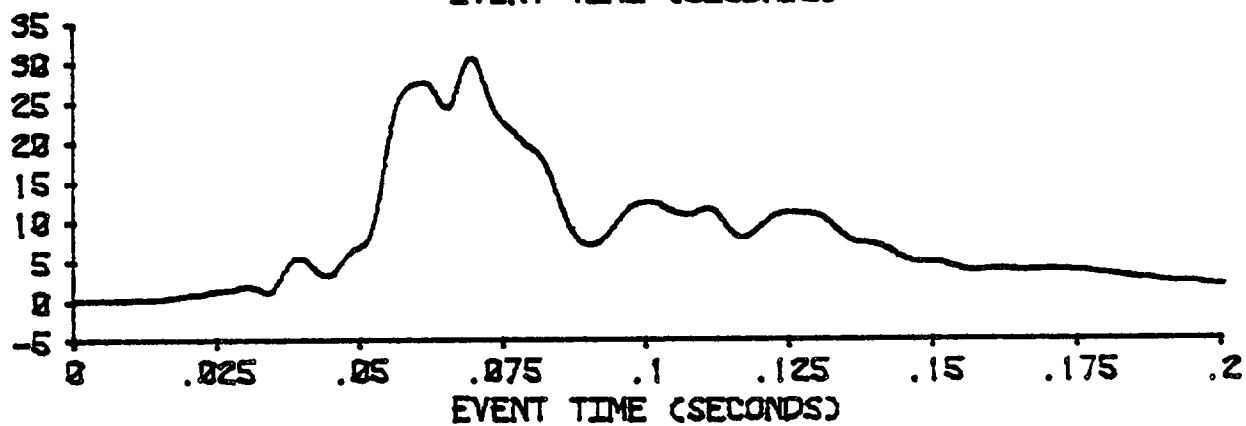
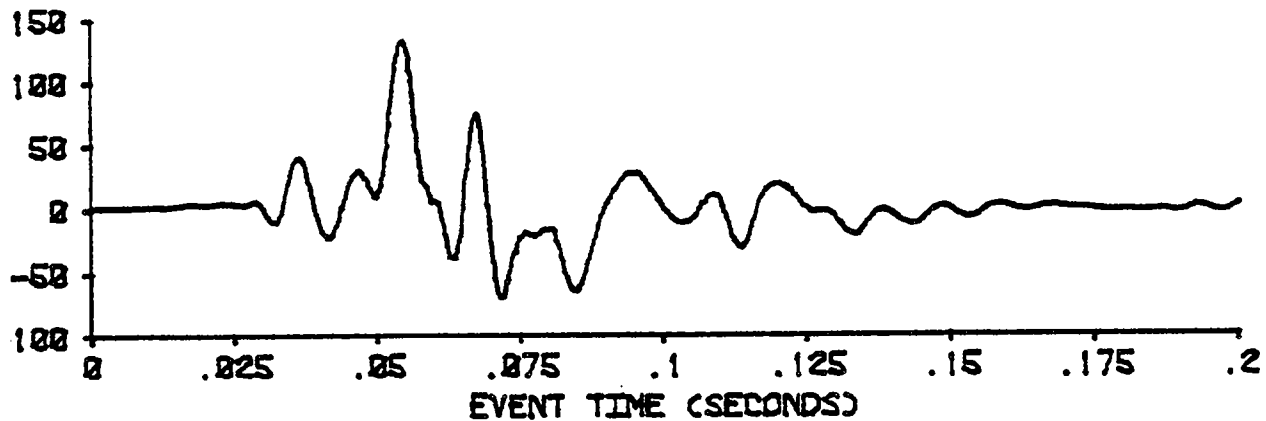
VEHICLE 2 - VW BUG - 33 MPH
LEFT DOOR LOWER RL
FILTER CLASS 60

TEST 1960
11 DEC 60



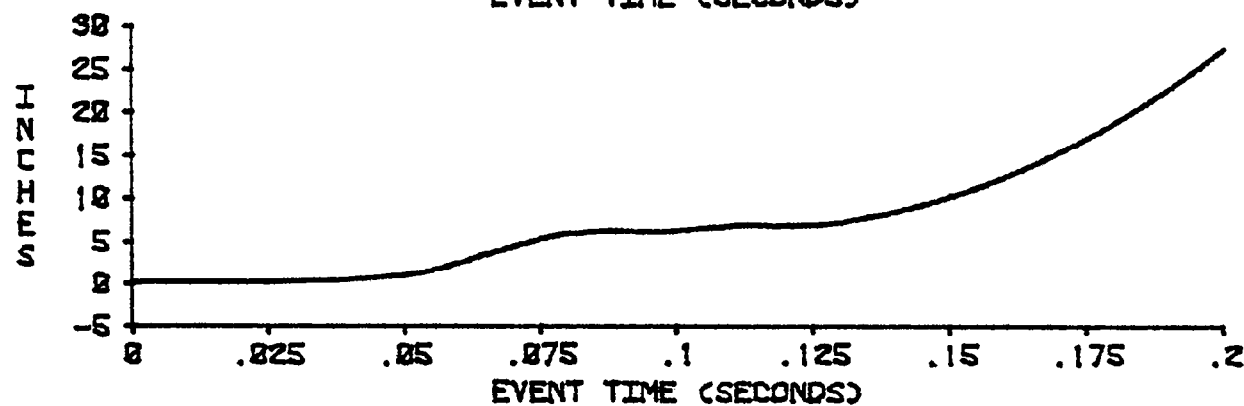
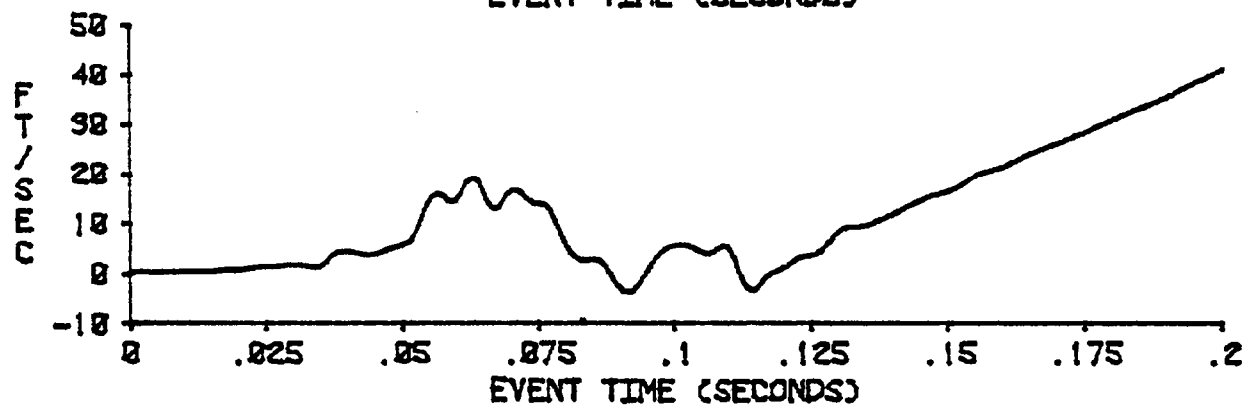
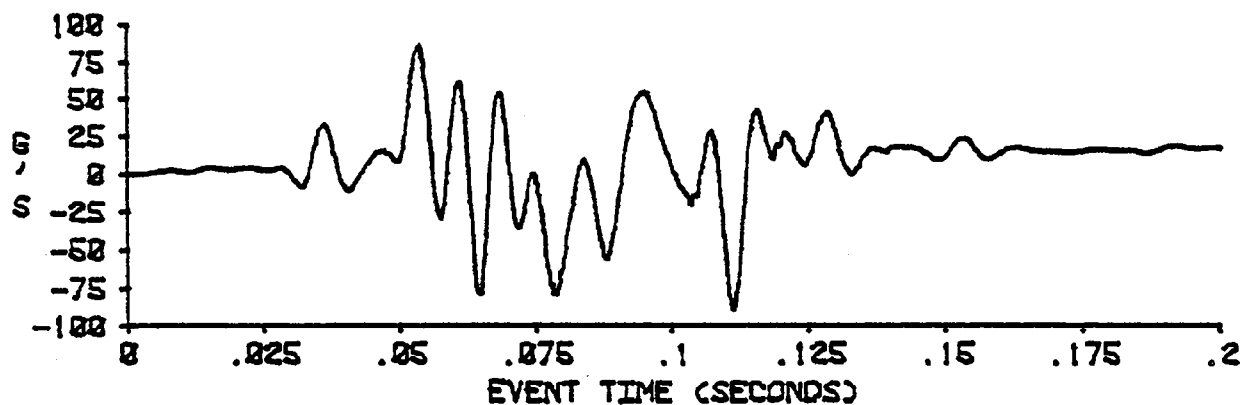
VEHICLE 2 - VW BUG - 33 MPH
B PILLAR UPPER RL
FILTER CLASS 60

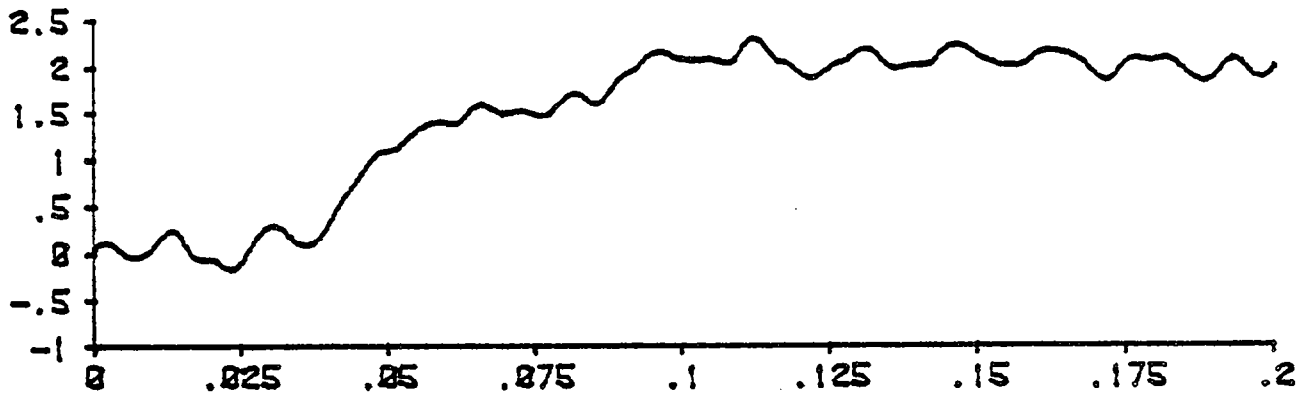
TEST 1960
11 DEC 60



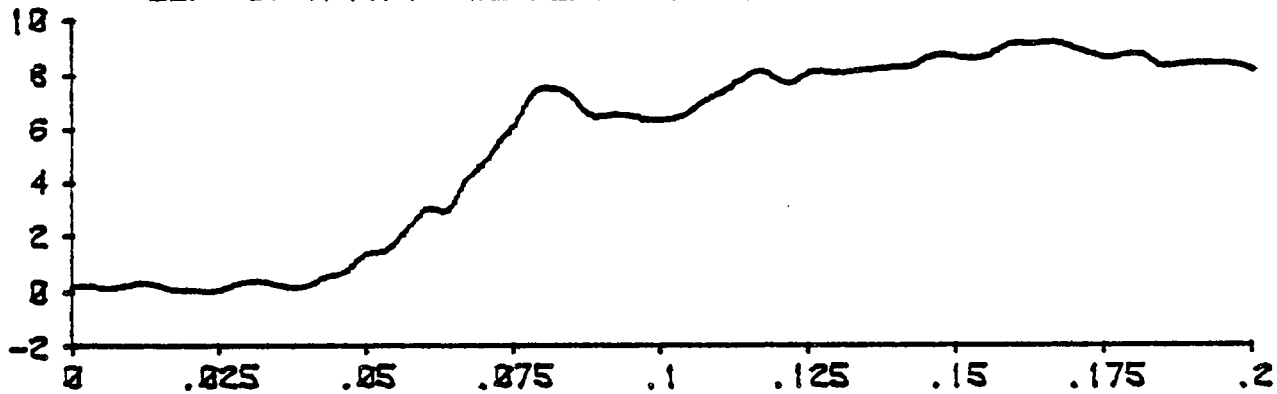
VEHICLE 2 - VW BUG - 33 MPH
B PILLAR LOWER RL
FILTER CLASS 60

TEST 1960
11 DEC 60

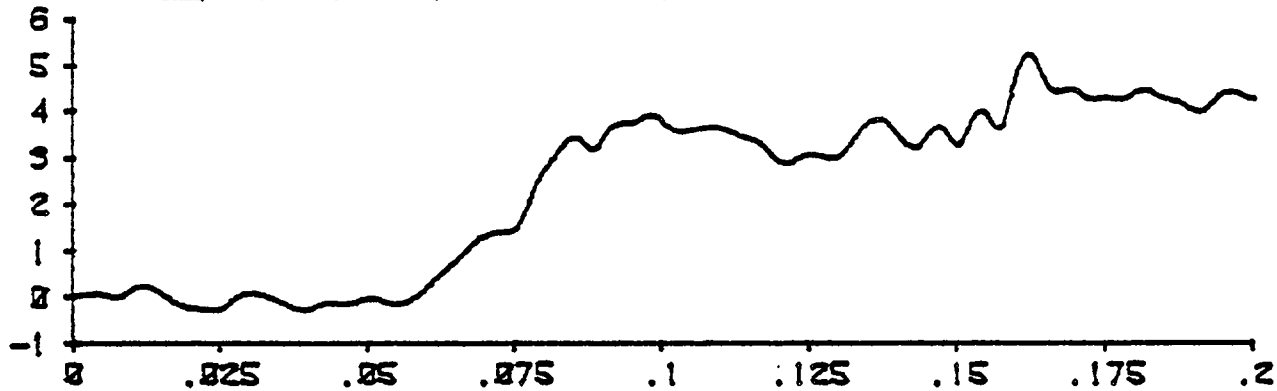




LEFT DOOR FRONT DISPLACEMENT RL FC 68 VS TIME (SEC)



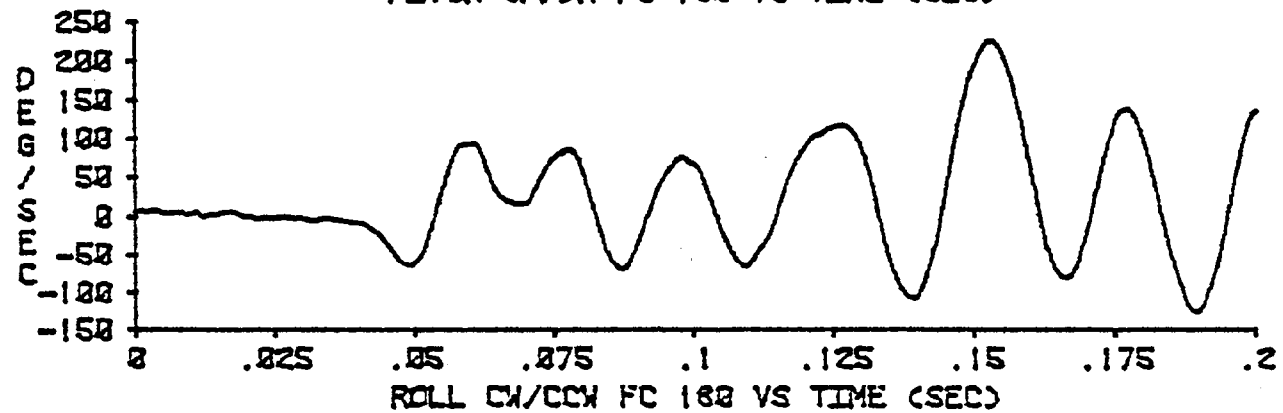
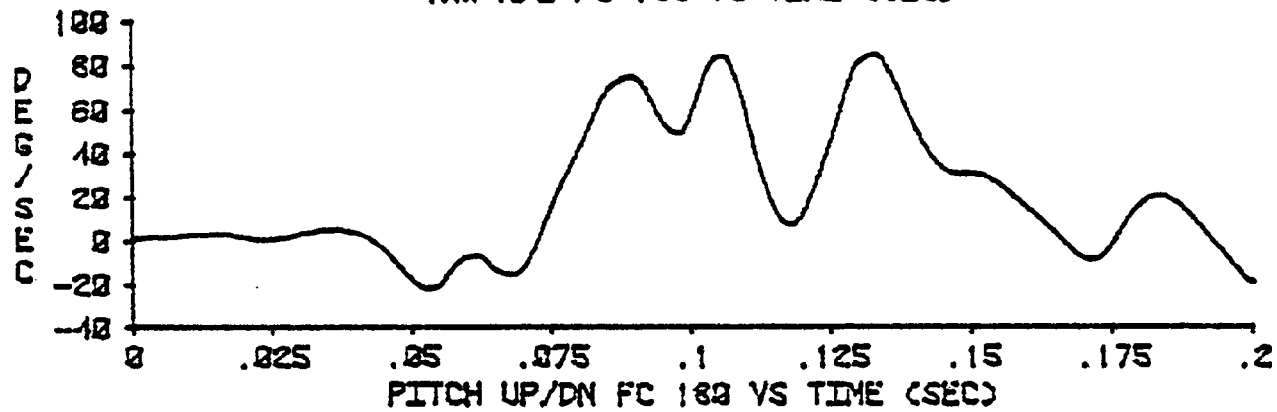
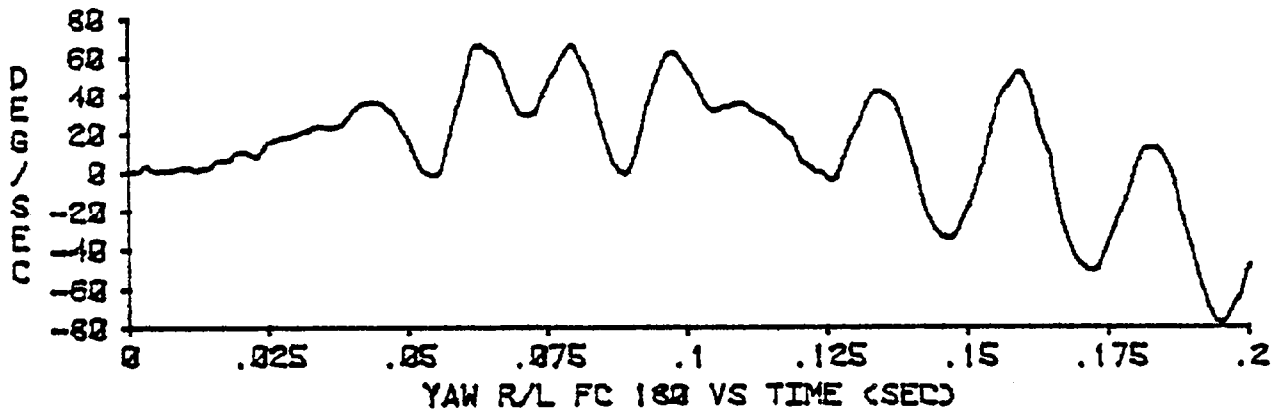
LEFT DOOR REAR DISPLACEMENT RL FC 68 VS TIME (SEC)



LEFT B PILLAR DISPLACEMENT RL FC 68 VS TIME (SEC)

VEHICLE 2 - VW BUG - 33 MPH

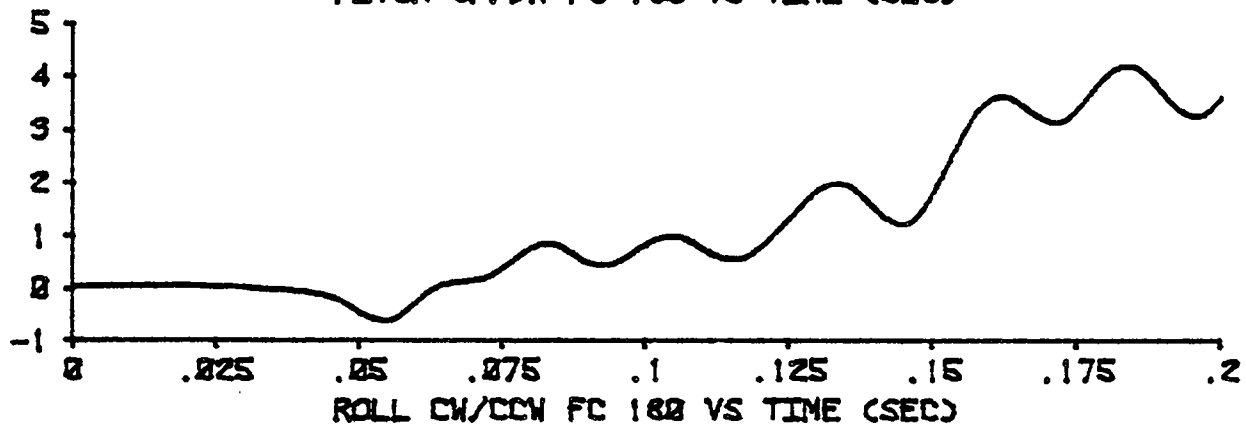
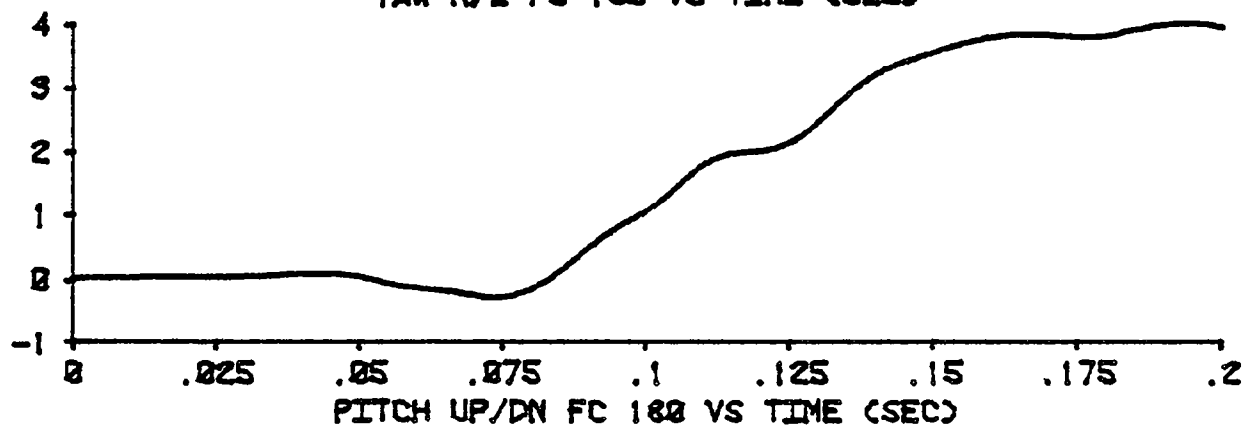
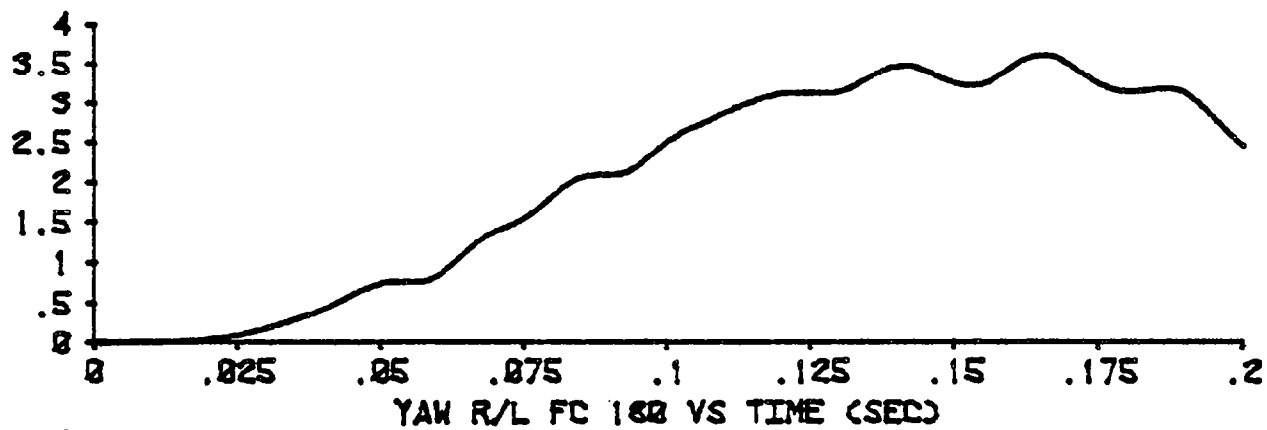
TEST 1962
11 DEC 62



VEHICLE 2 - VW BUG - 33 MPH

TEST 1960

11 DEC 60



VEHICLE 1 - CHEV MALIBU - 16.8 MPH

TEST 1960

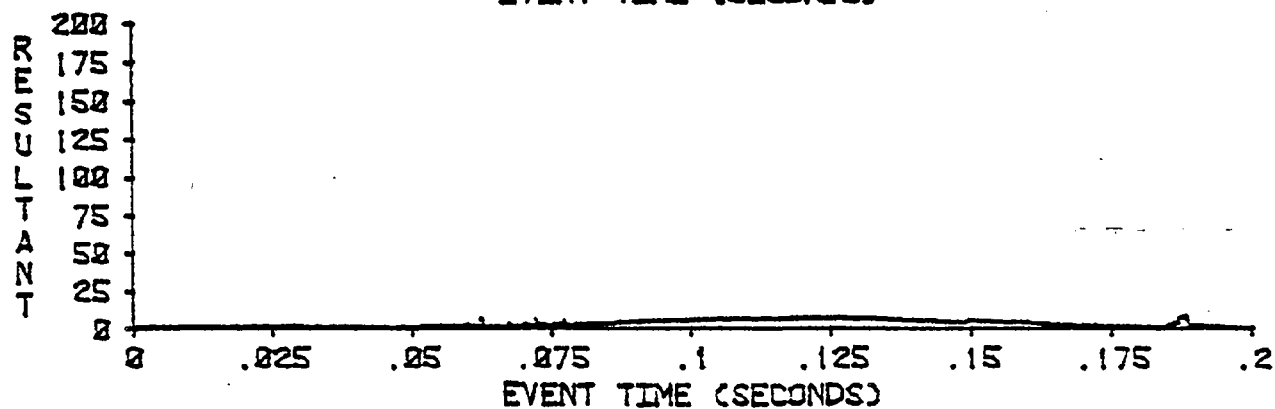
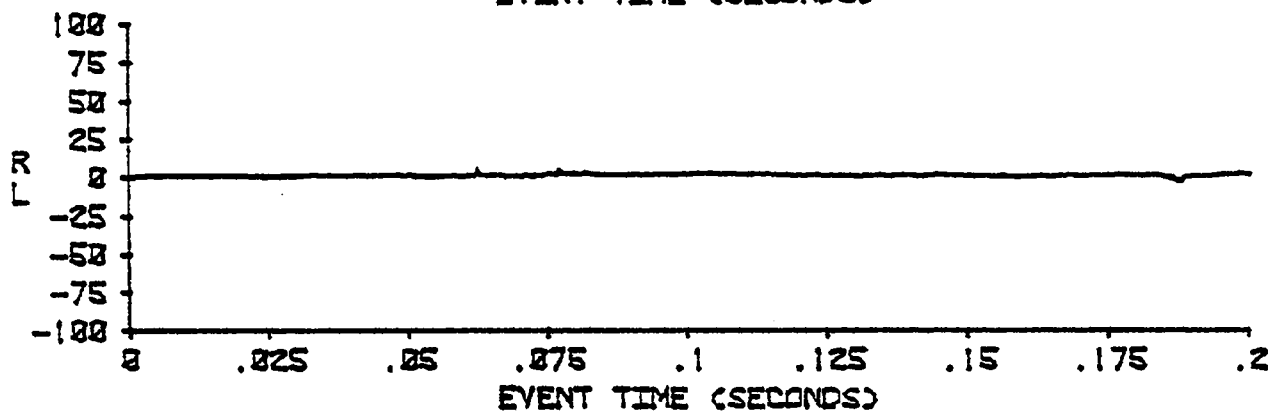
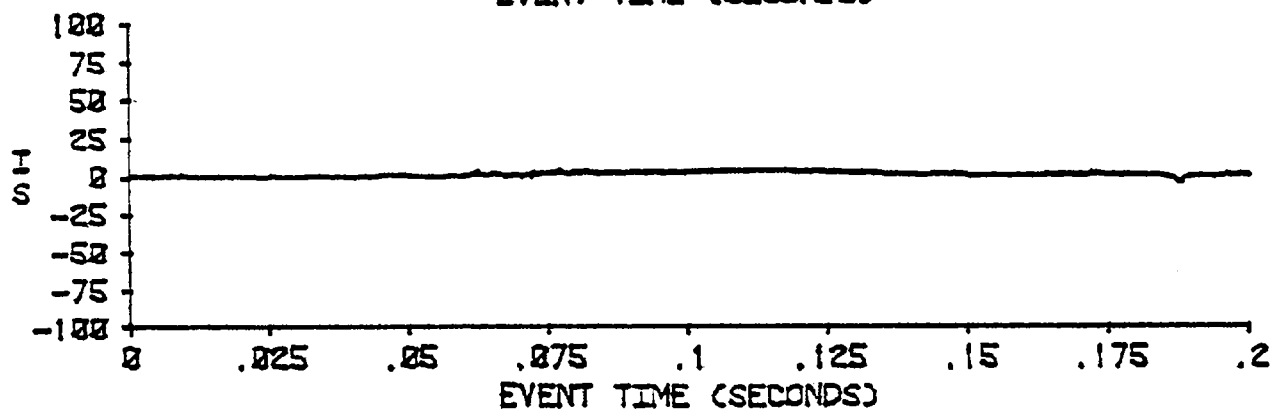
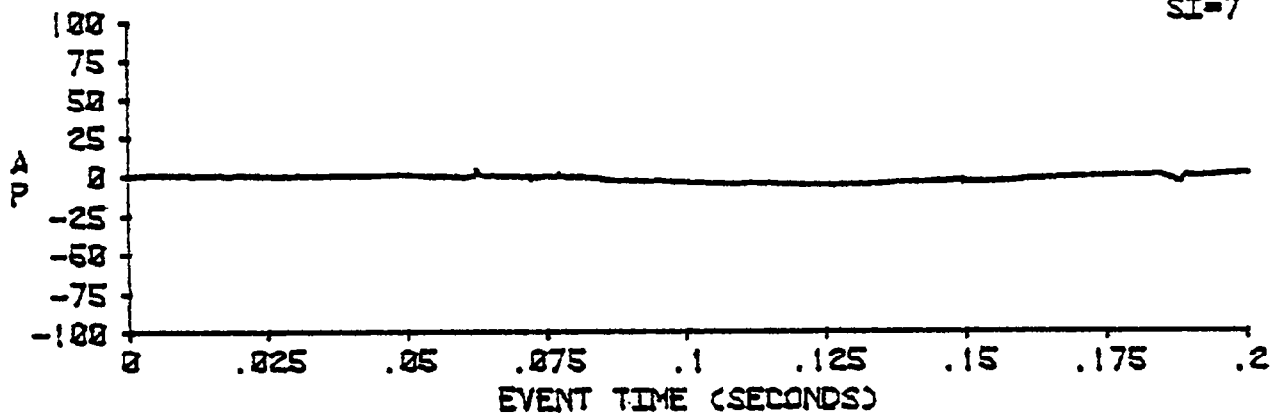
DRIVER HEAD G'S

11 DEC 80

FILTER CLASS 1000

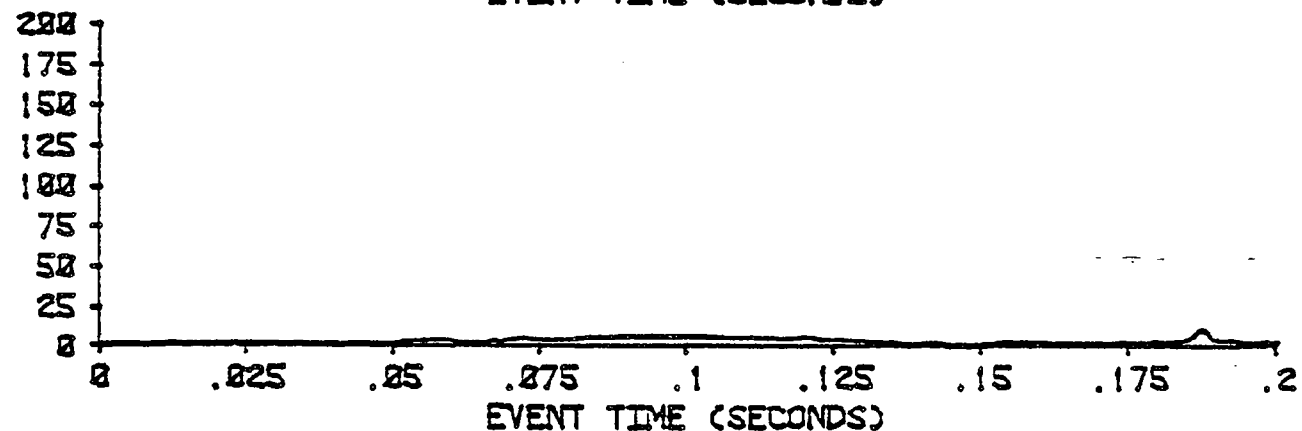
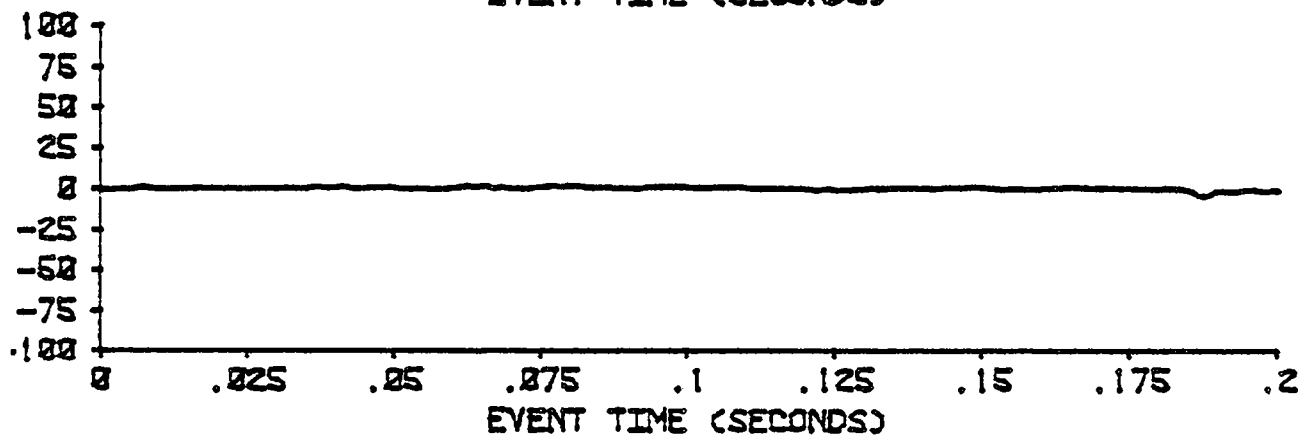
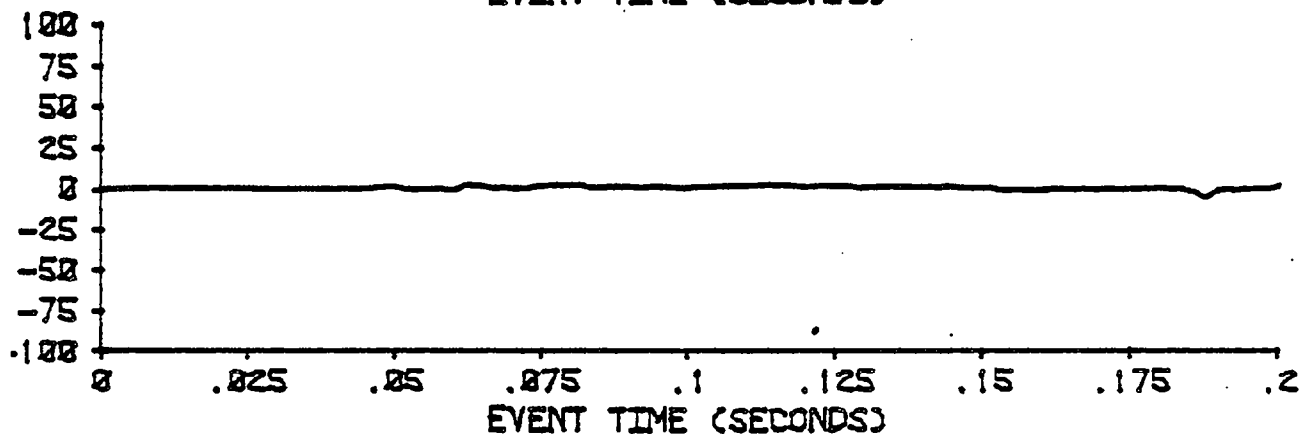
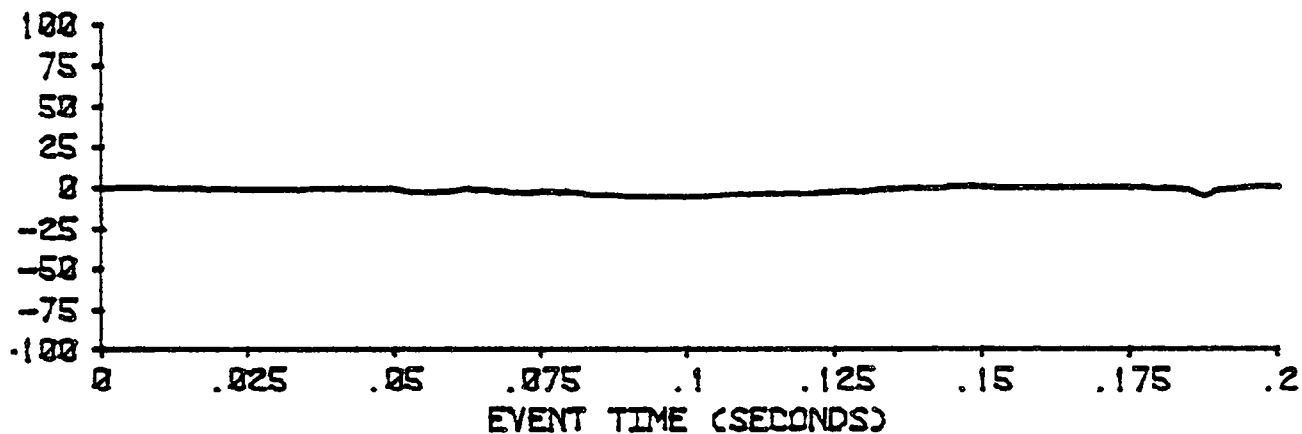
HIC=6 FROM .079 TO .1596 SECONDS

SI-7



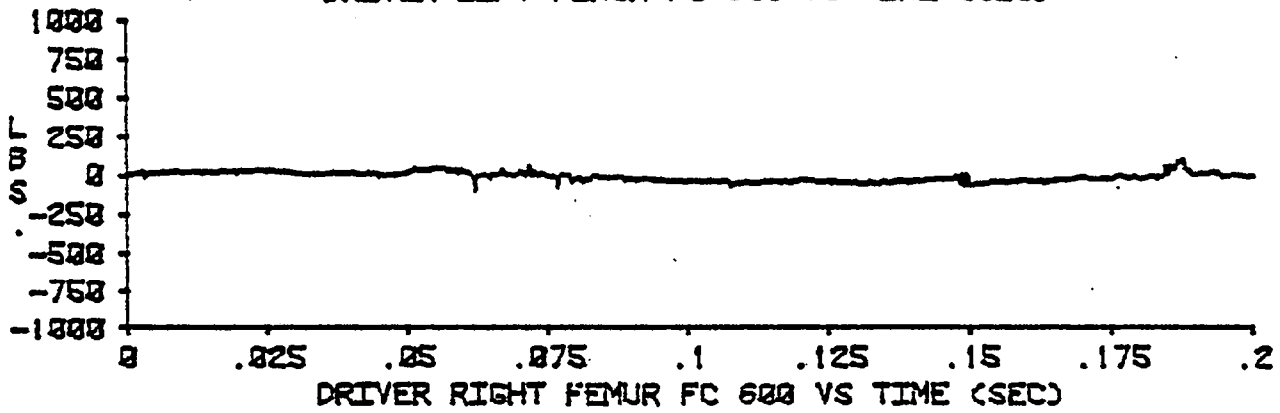
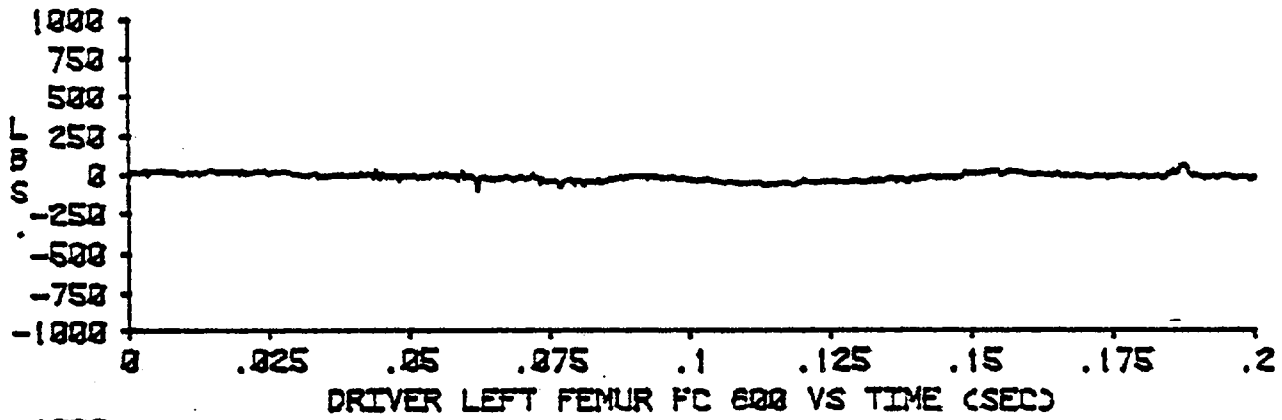
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
DRIVER CHEST G'S
FILTER CLASS 180

TEST 1960
11 DEC 80
SI=7



VEHICLE 1 - CHEV MALIBU - 16.8 MPH

TEST 1860
11 DEC 80



VEHICLE 1 - CHEV MALIBU - 16.8 MPH

TEST 1960

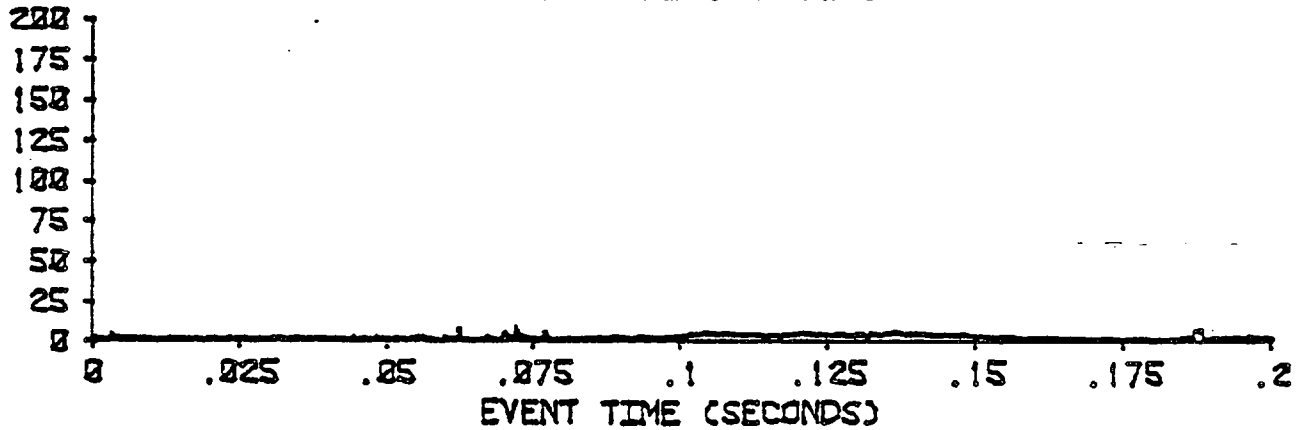
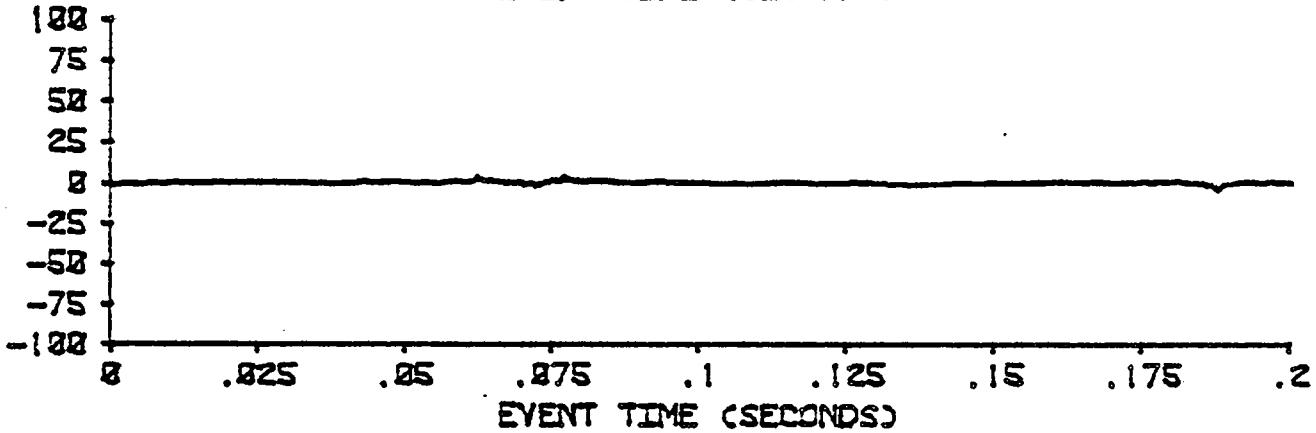
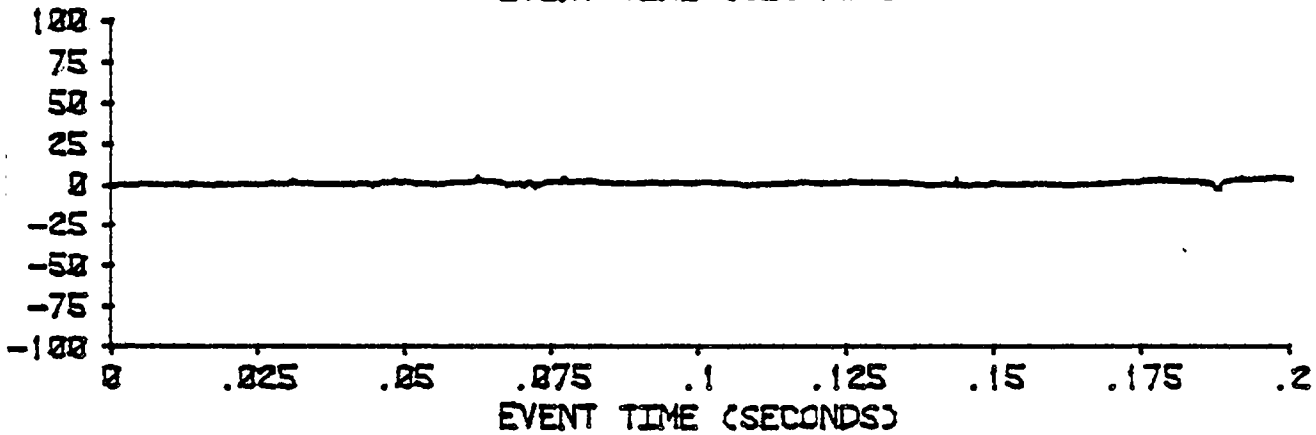
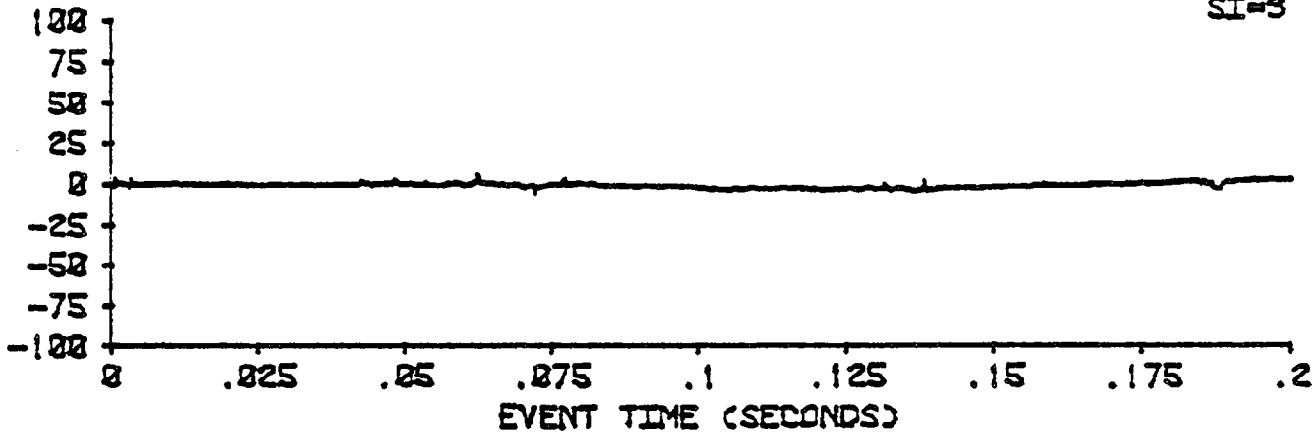
PASS. HEAD G'S

11 DEC 80

FILTER CLASS 1000

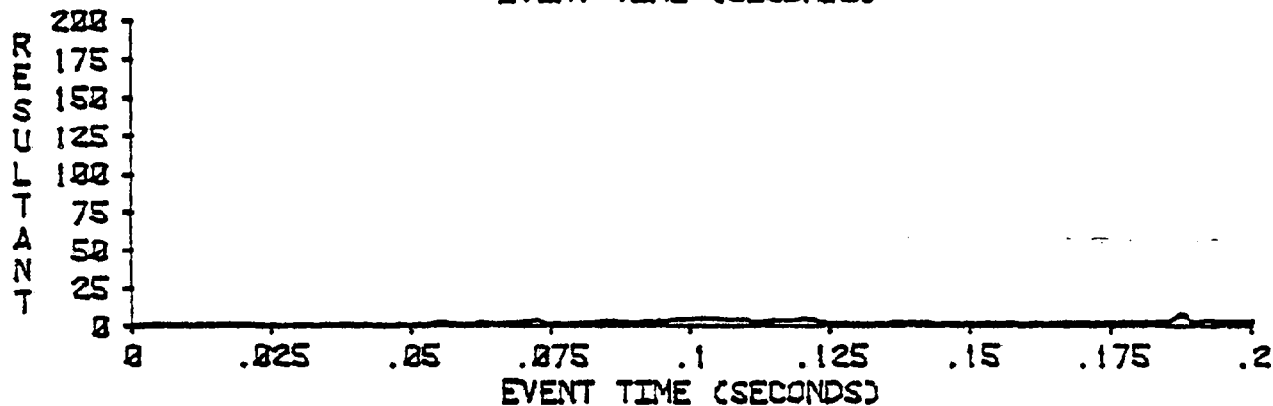
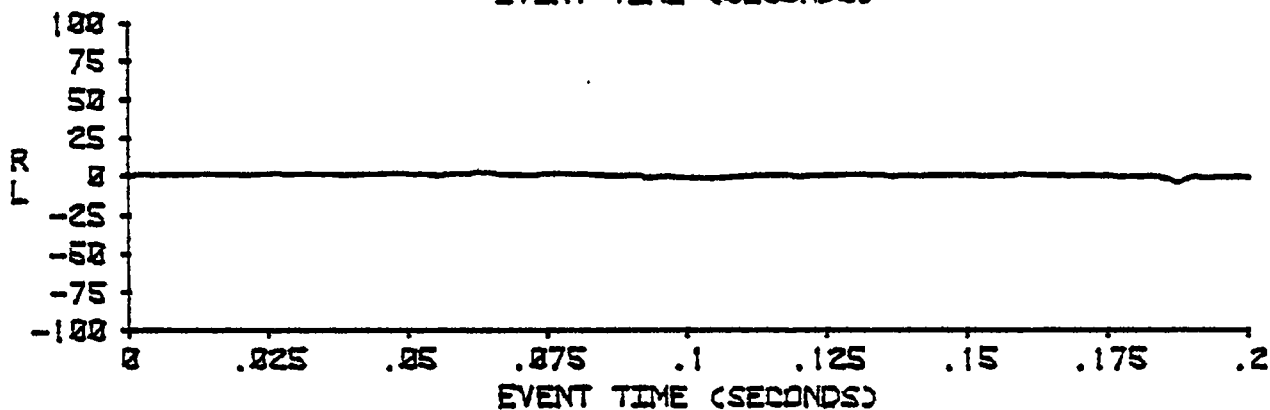
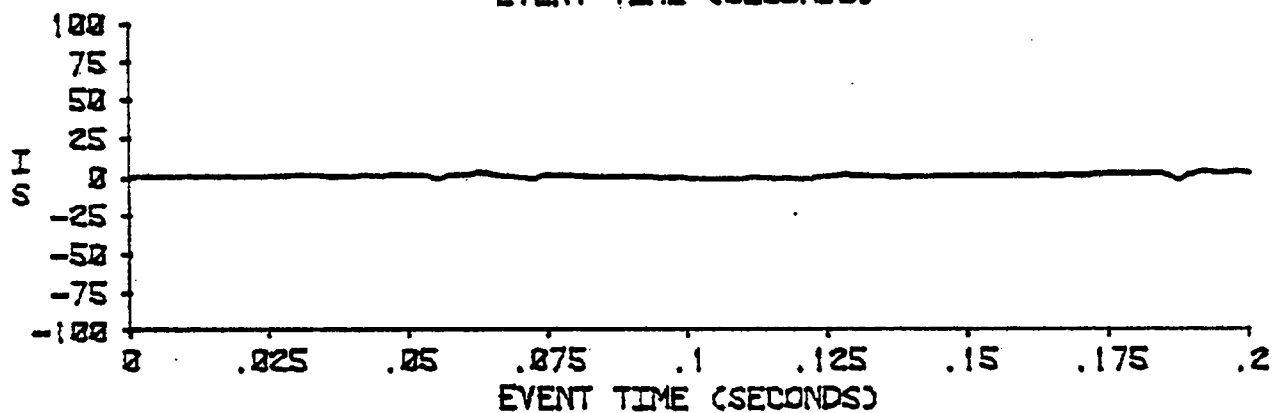
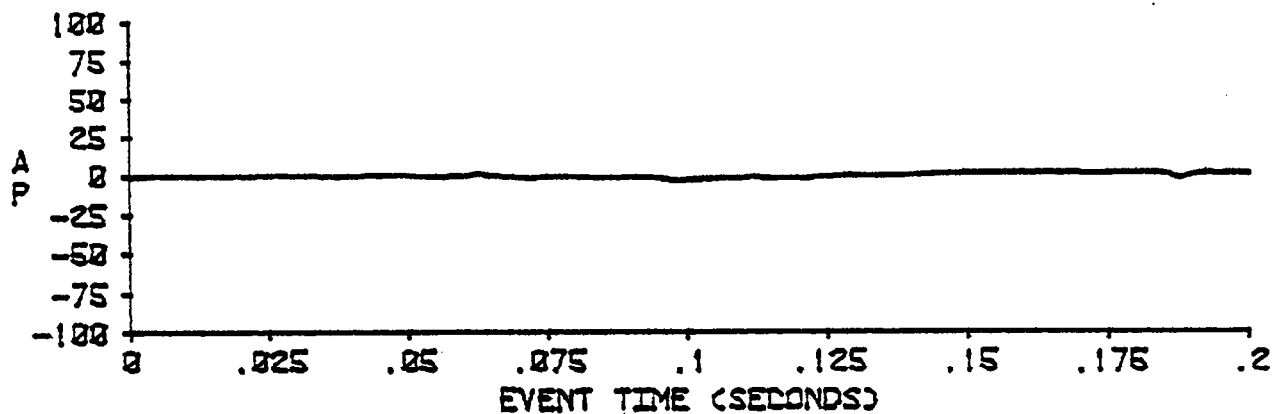
HIC=2 FROM .069 TO .1986 SECONDS

SI=3



VEHICLE 1 - CHEV MALIBU - 16.8 MPH
PASS. CHEST G'S
FILTER CLASS 180

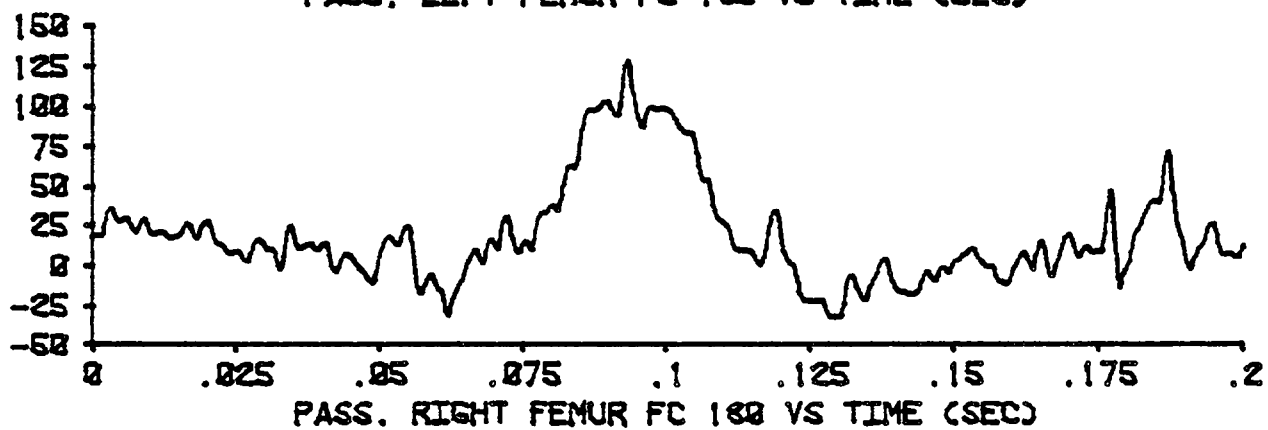
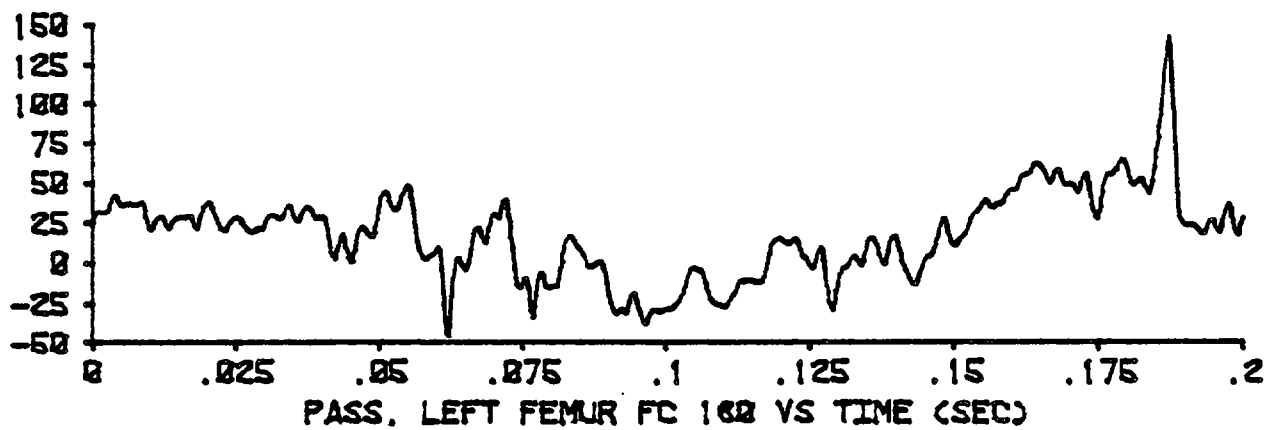
TEST 1960
11 DEC 80
SI=2



VEHICLE 1 - CHEV MALIBU - 16.8 MPH

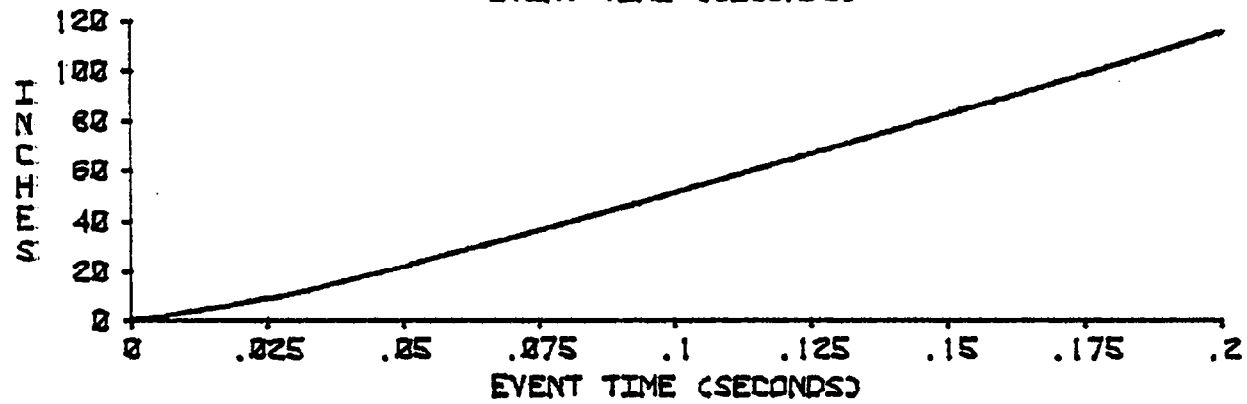
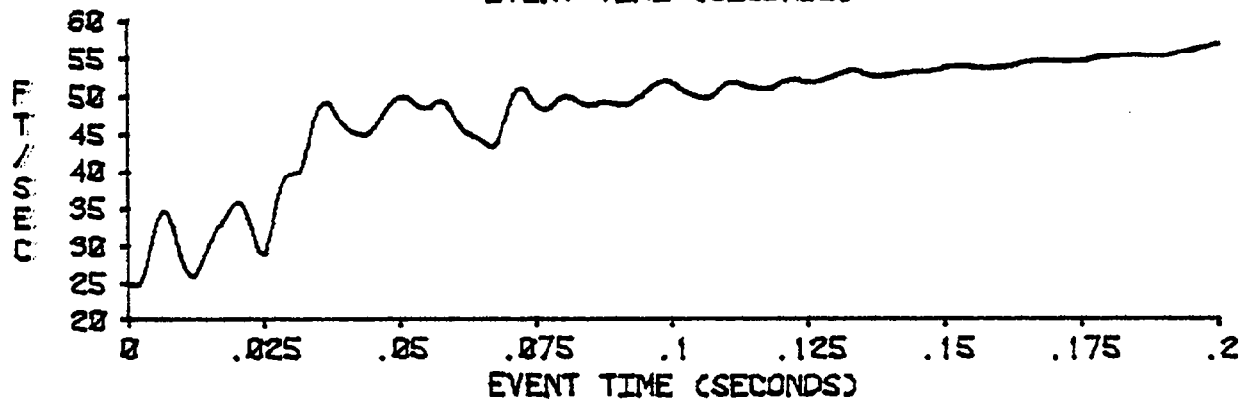
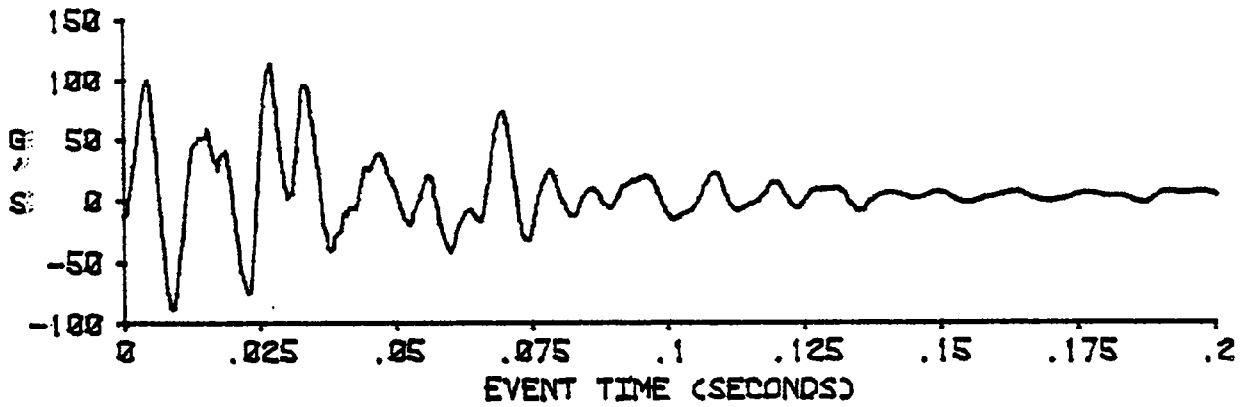
TEST 1860

11 DEC 80



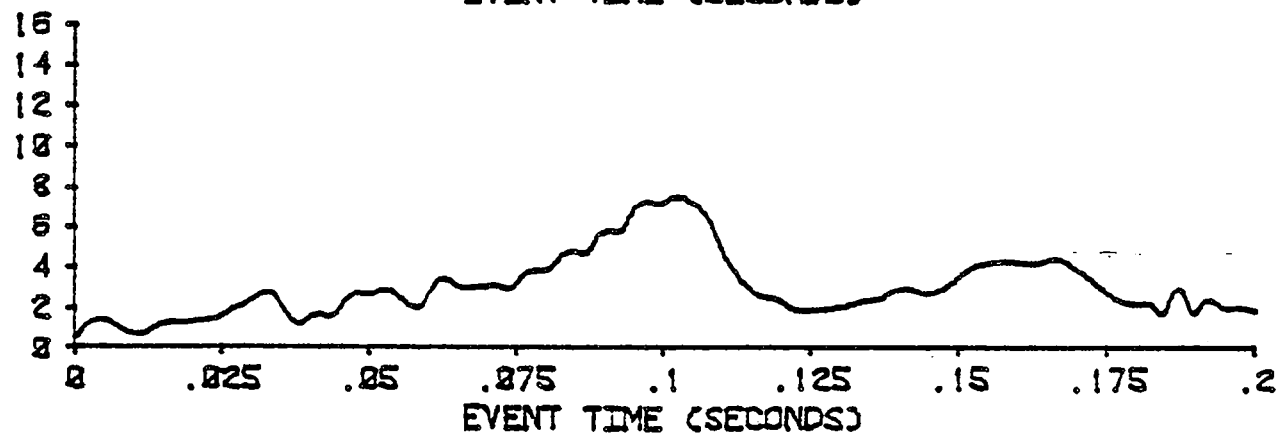
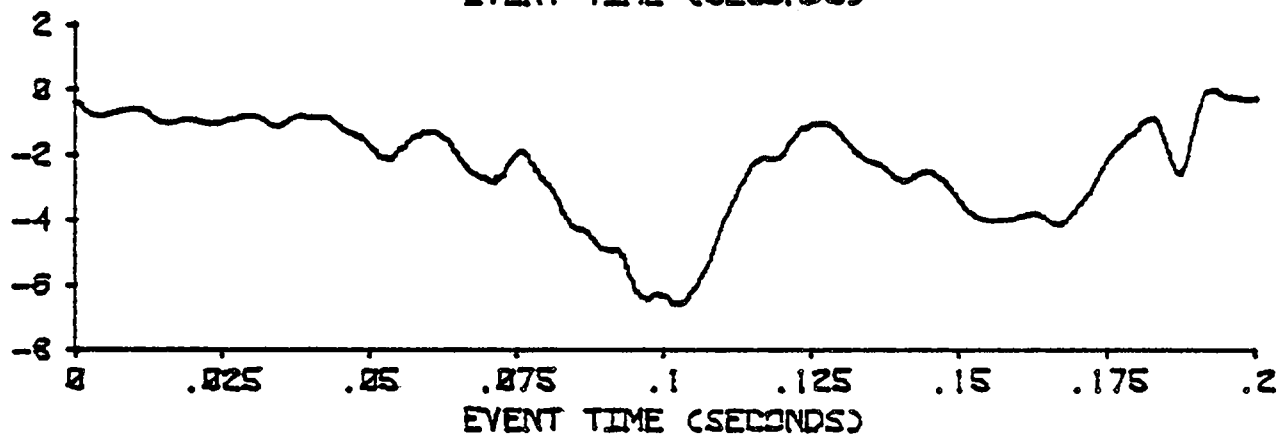
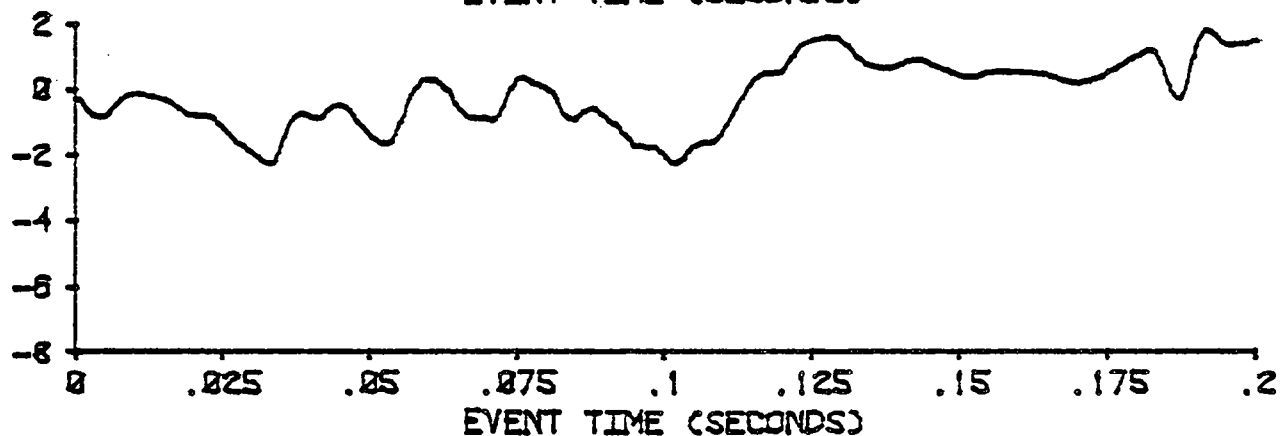
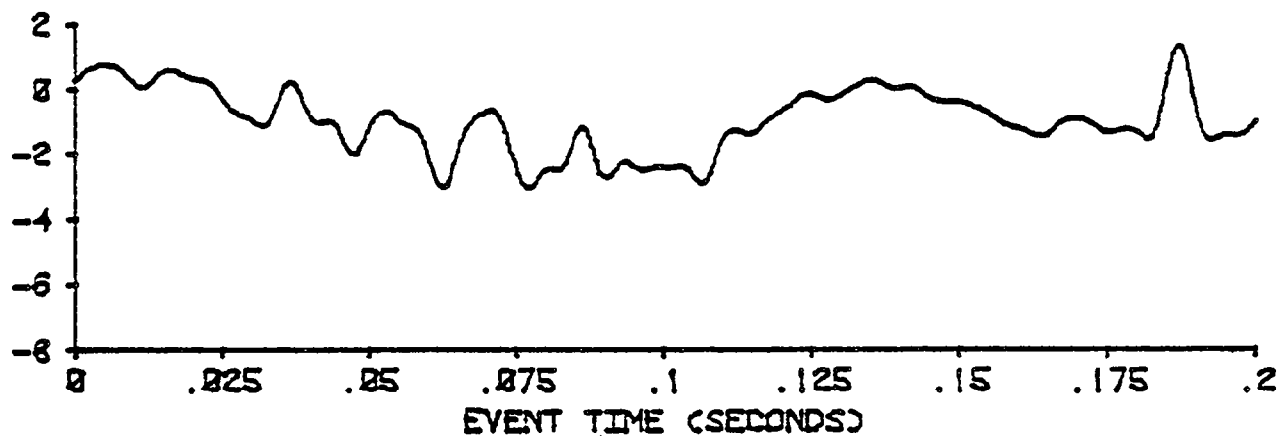
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR BUMPER AP
FILTER CLASS 60

TEST 1960
11 DEC 60



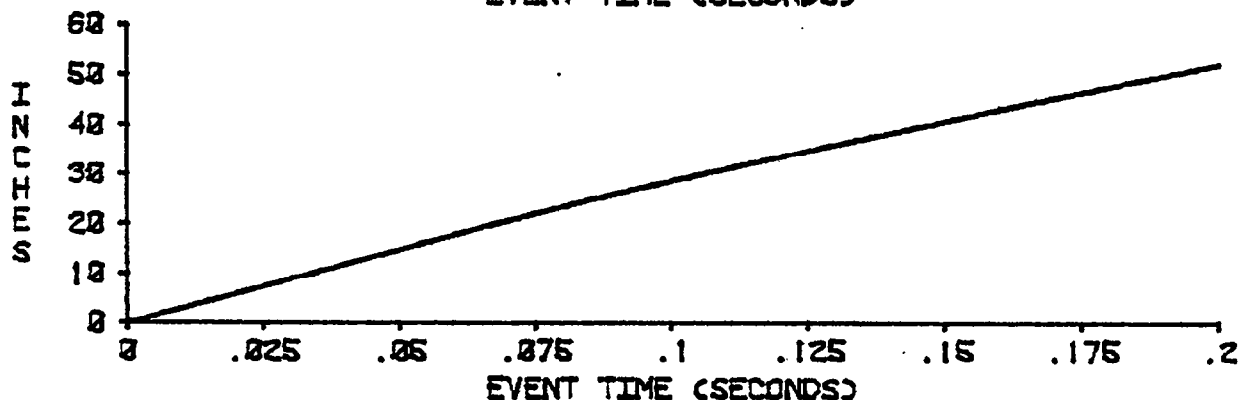
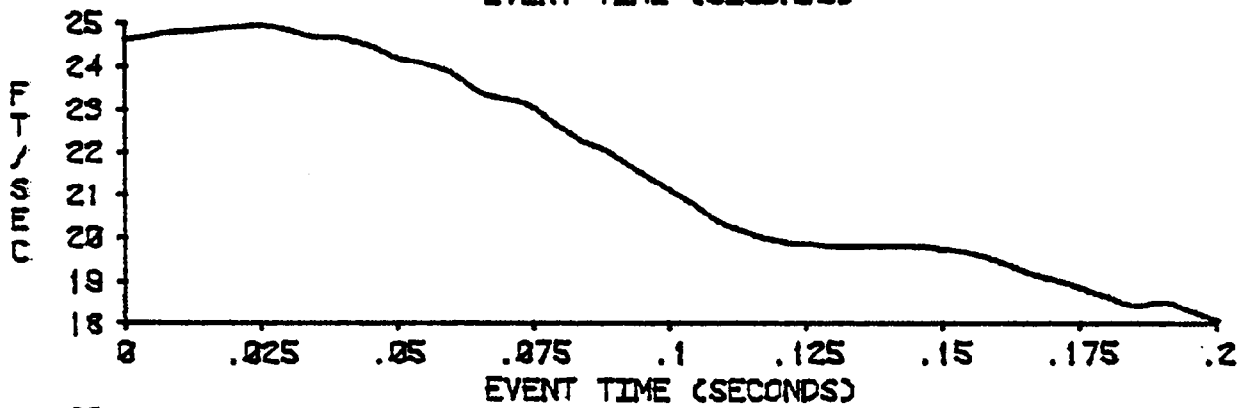
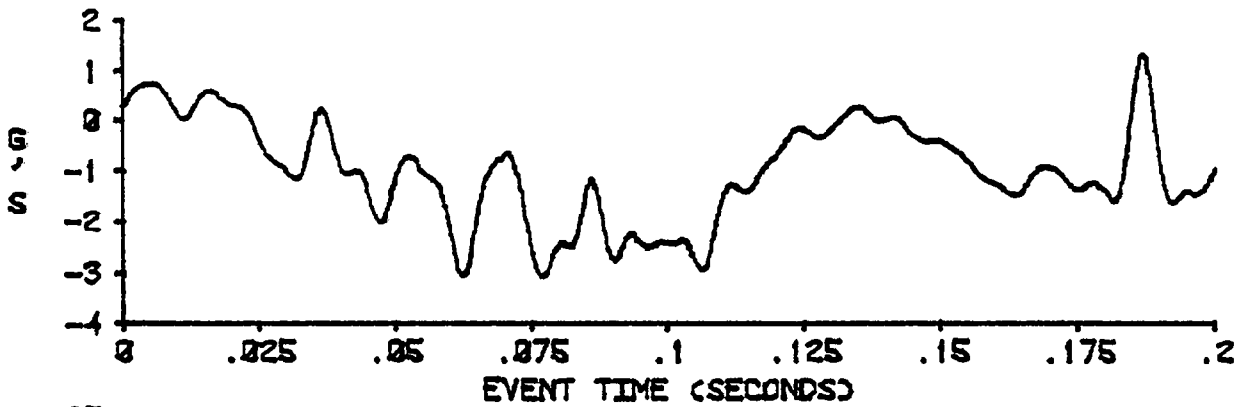
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR ENGINE G'S G'S
FILTER CLASS 60

TEST 1960
11 DEC 80



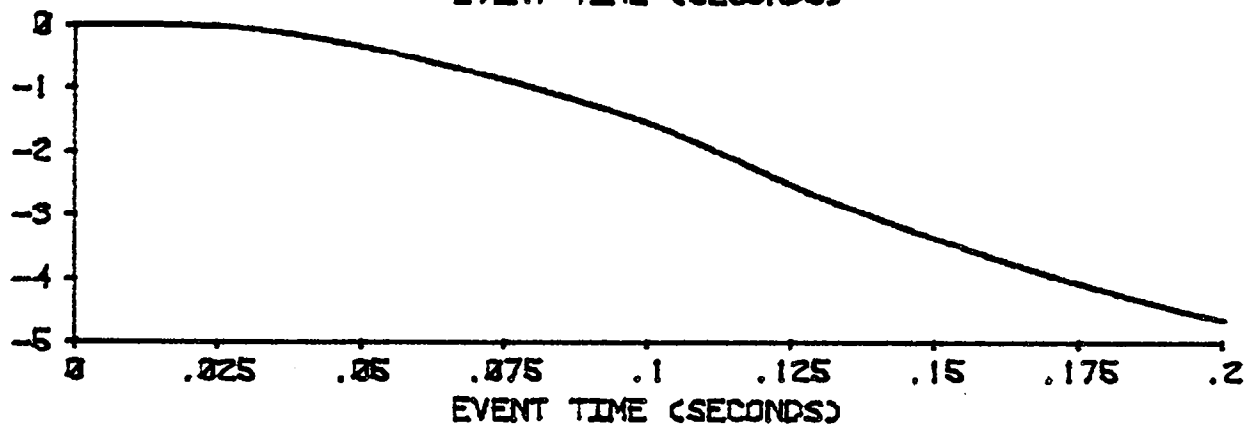
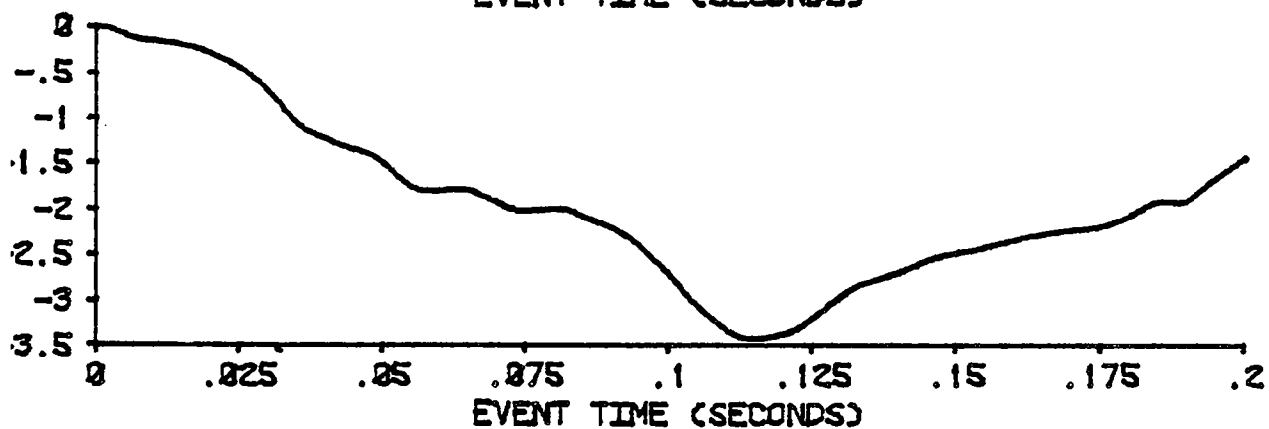
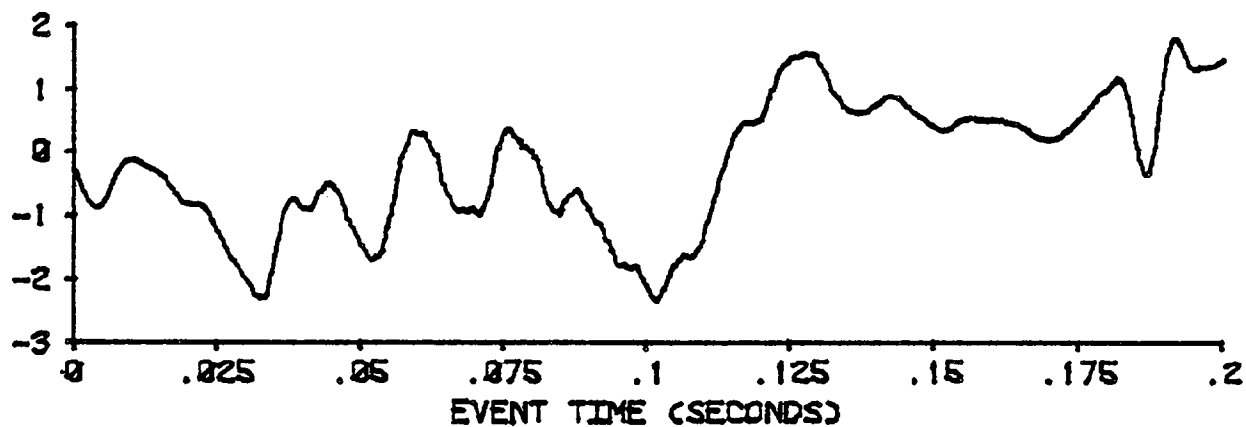
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR ENGINE AP
FILTER CLASS 60

TEST 1960
11 DEC 60



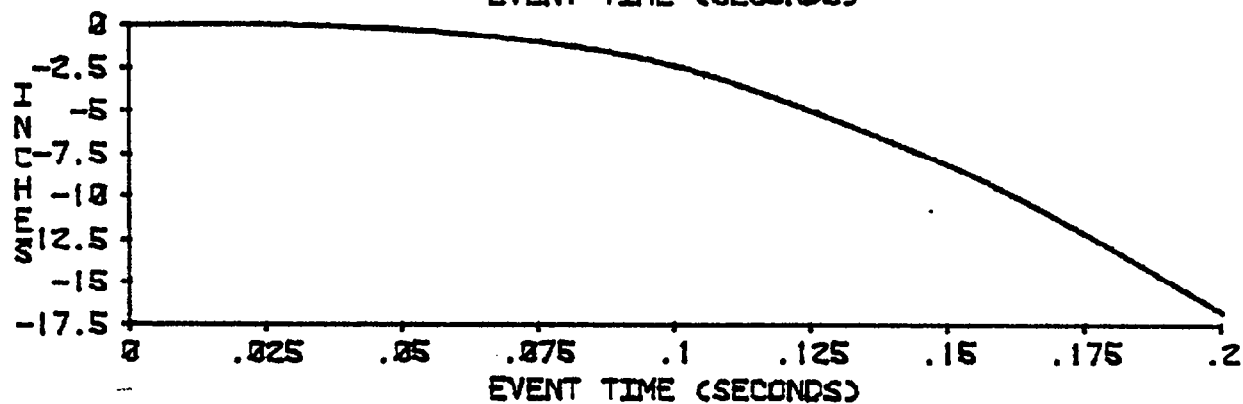
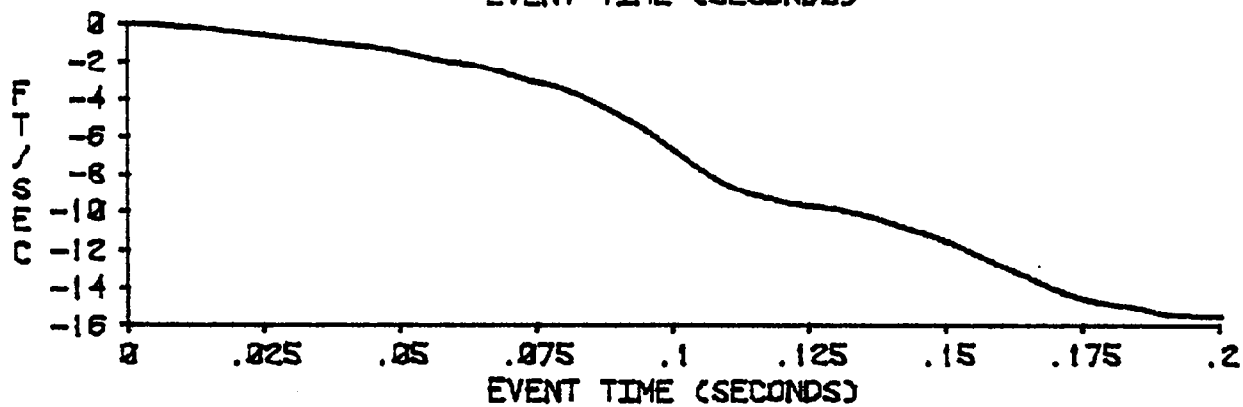
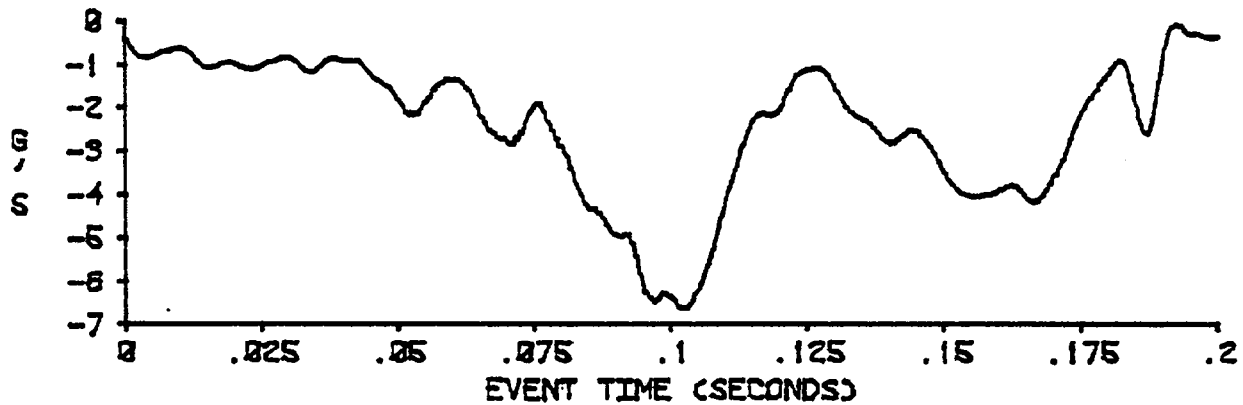
VEHICLE 1 - CHEV MALIBU - 16.6 MPH
CAR ENGINE IS
FILTER CLASS 60

TEST 1960
11 DEC 60



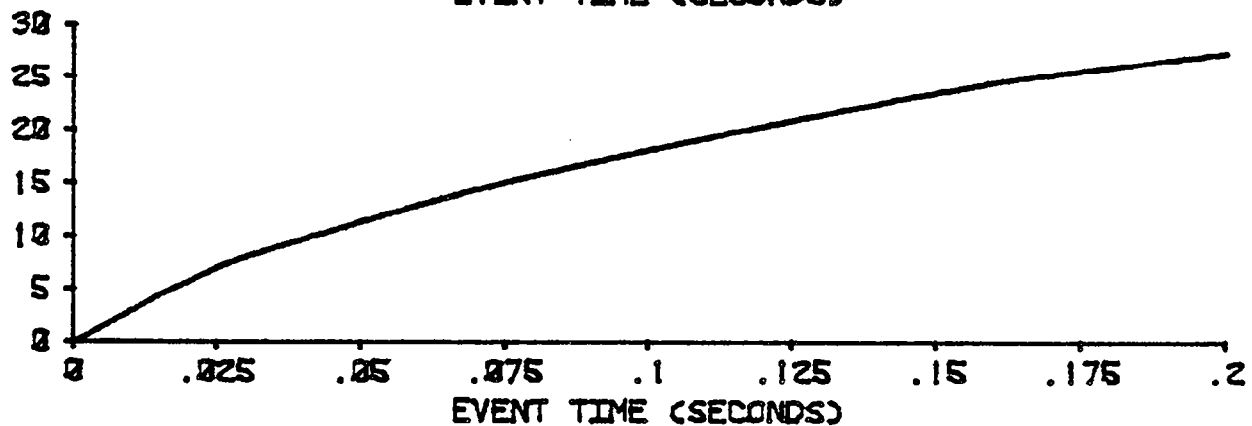
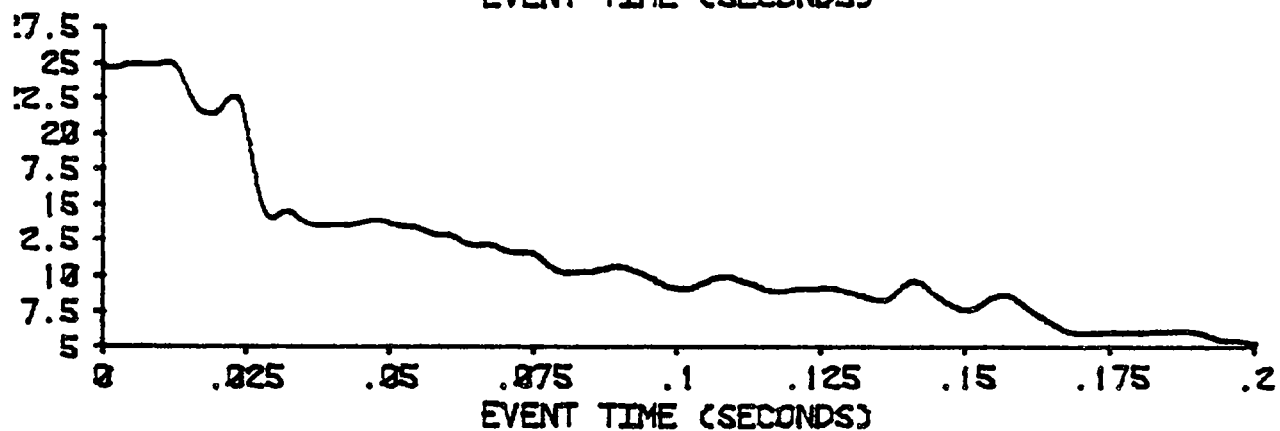
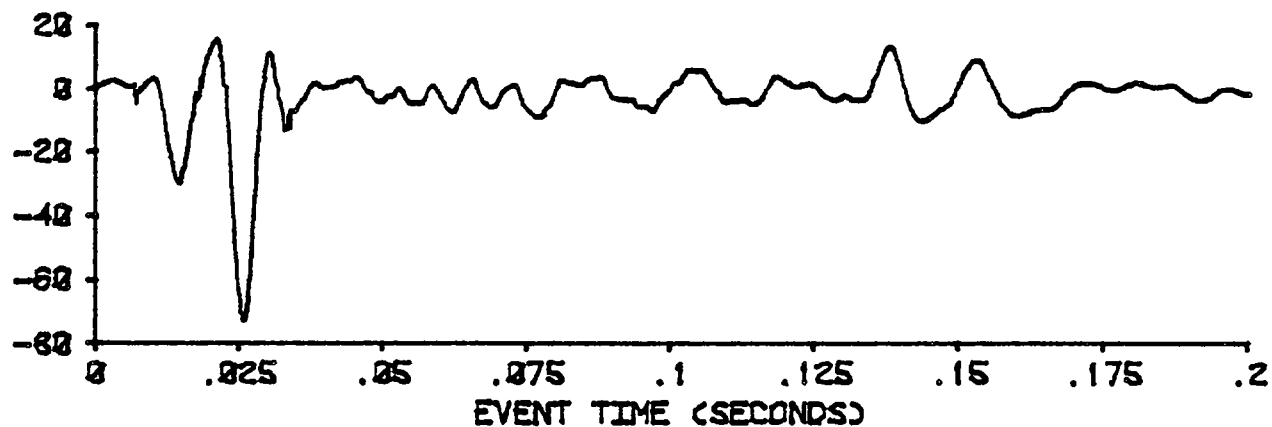
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR ENGINE RL
FILTER CLASS 60

TEST 1960
11 DEC 80



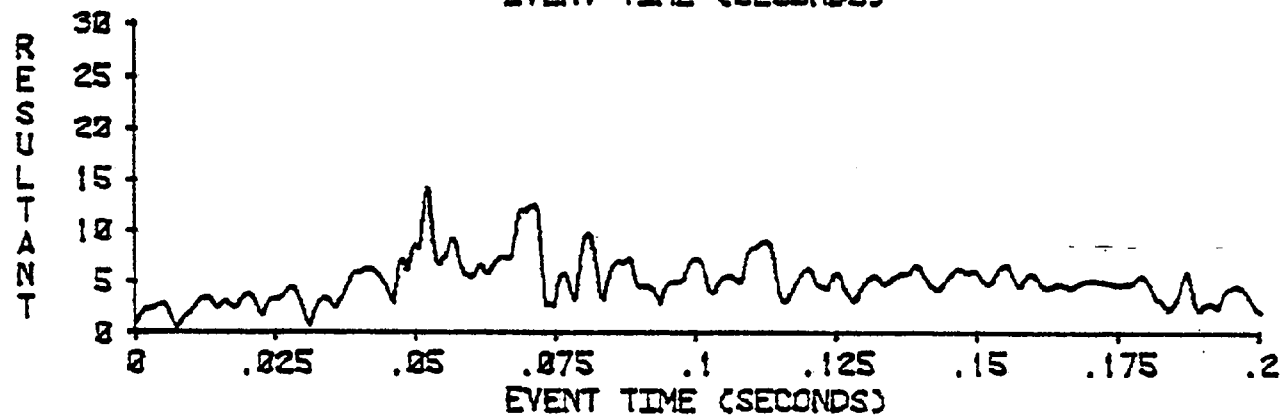
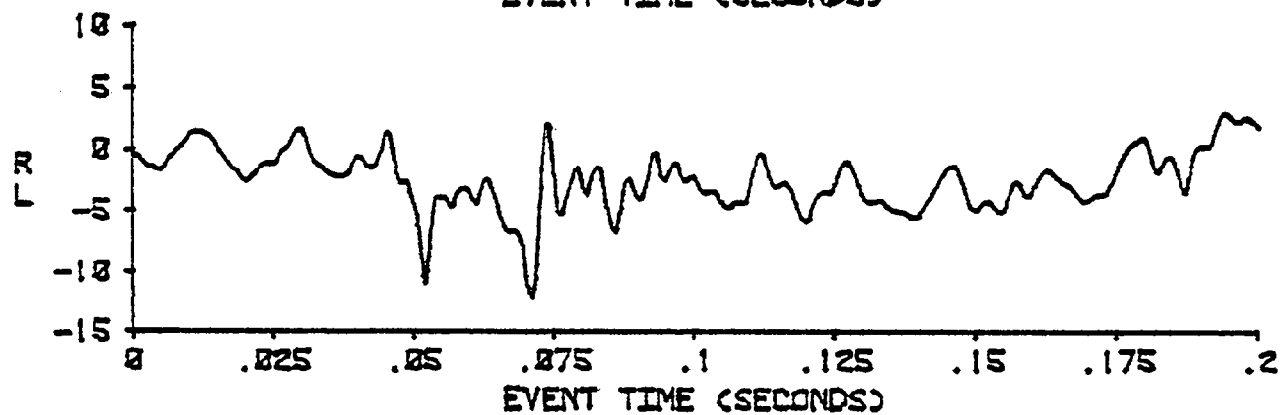
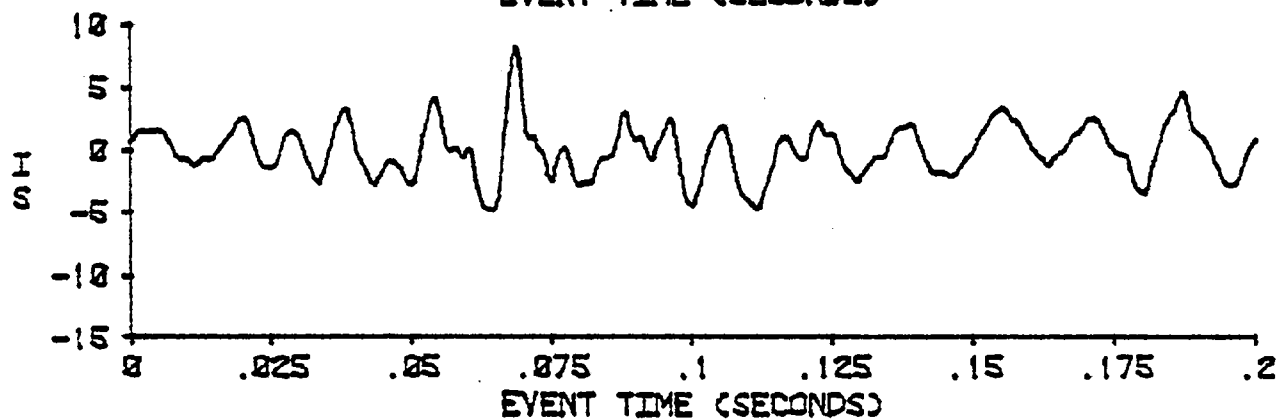
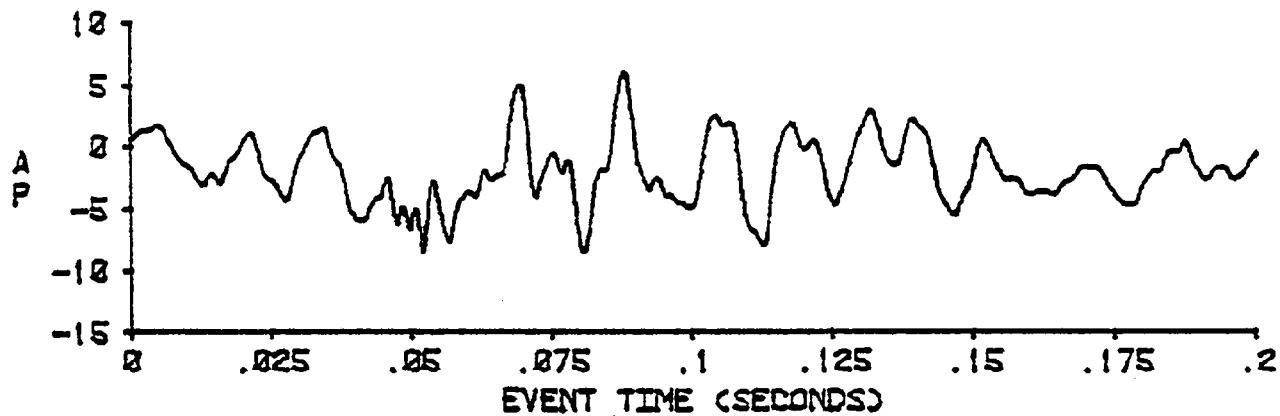
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
RT. FRONT SUSPENSION AP
FILTER CLASS 68

TEST 1968
11 DEC 88



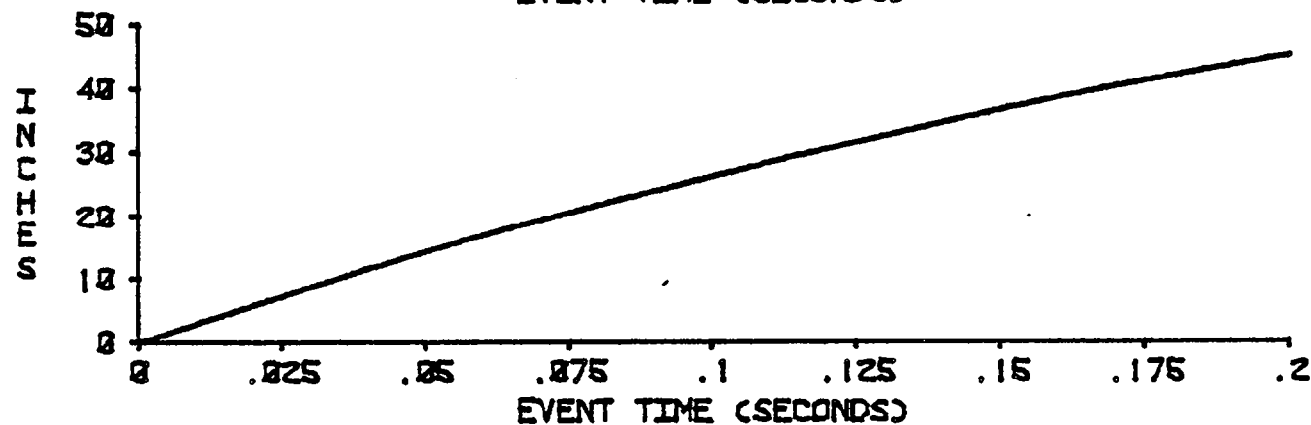
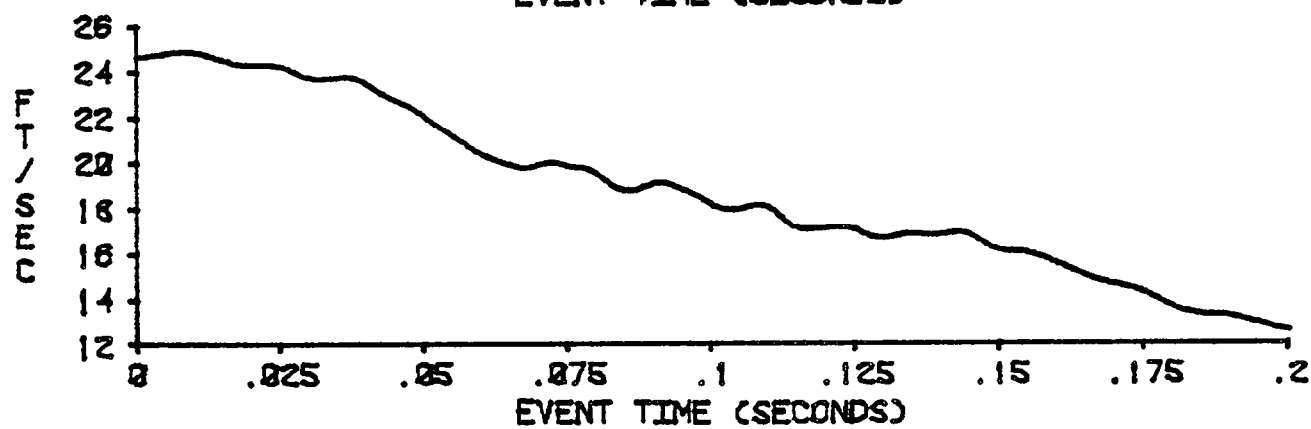
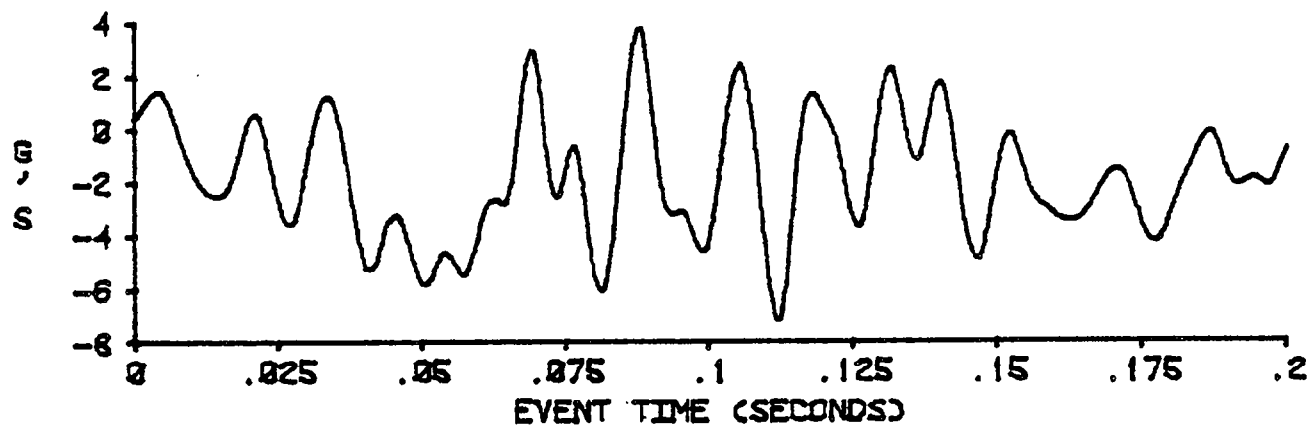
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR FIREWALL G'S
FILTER CLASS 180

TEST 1960
11 DEC 68



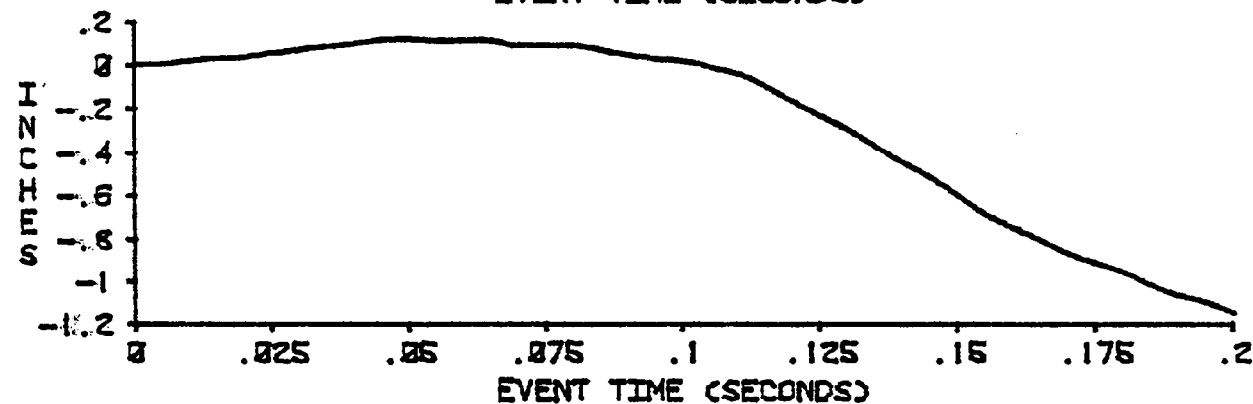
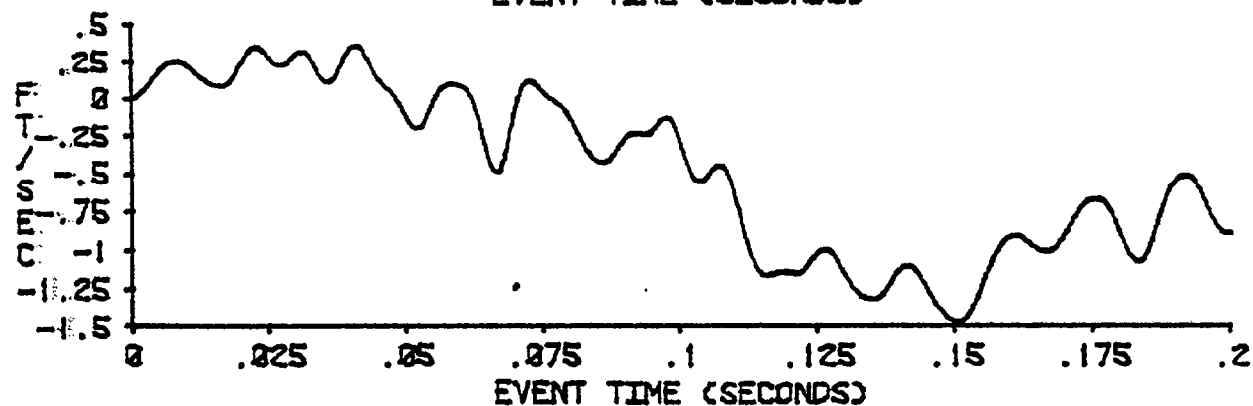
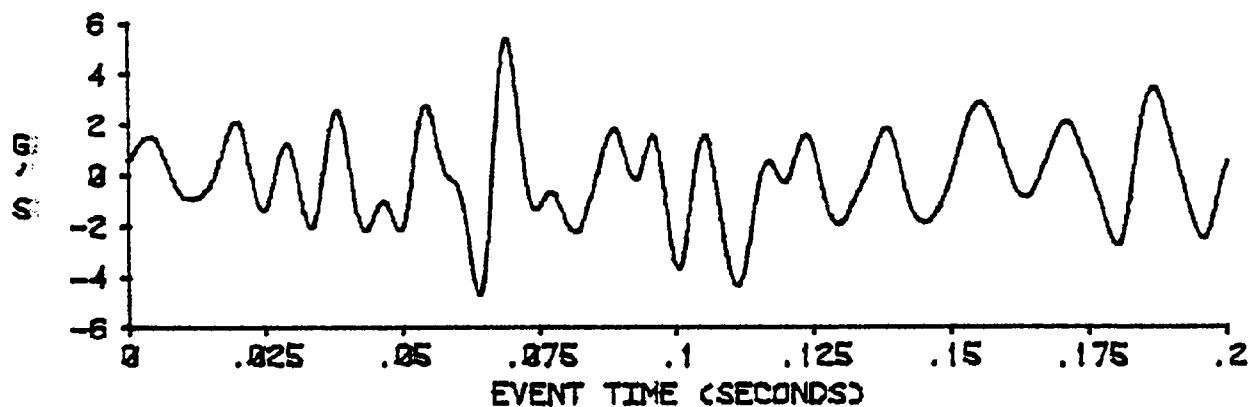
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR FIREWALL AP
FILTER CLASS 50

TEST 1960
11 DEC 60



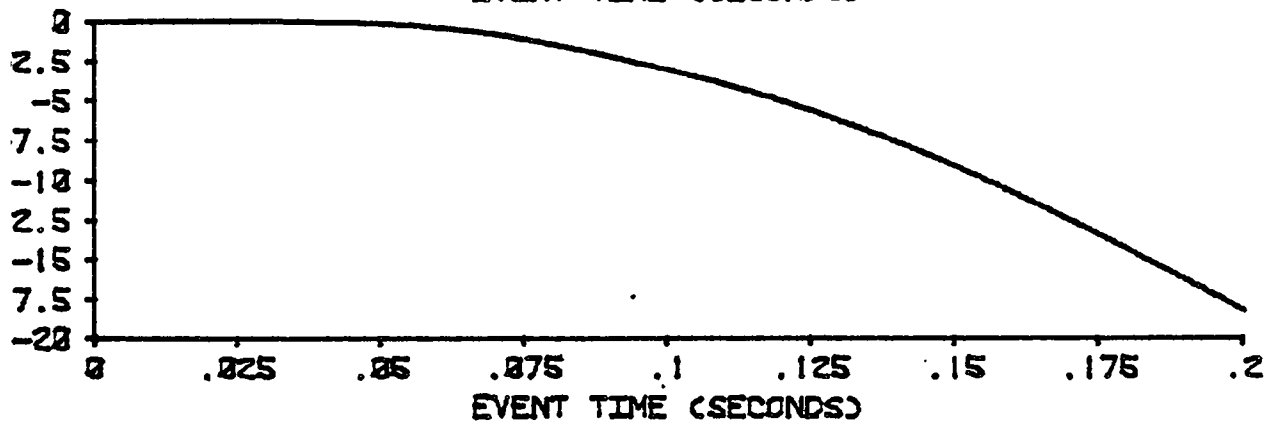
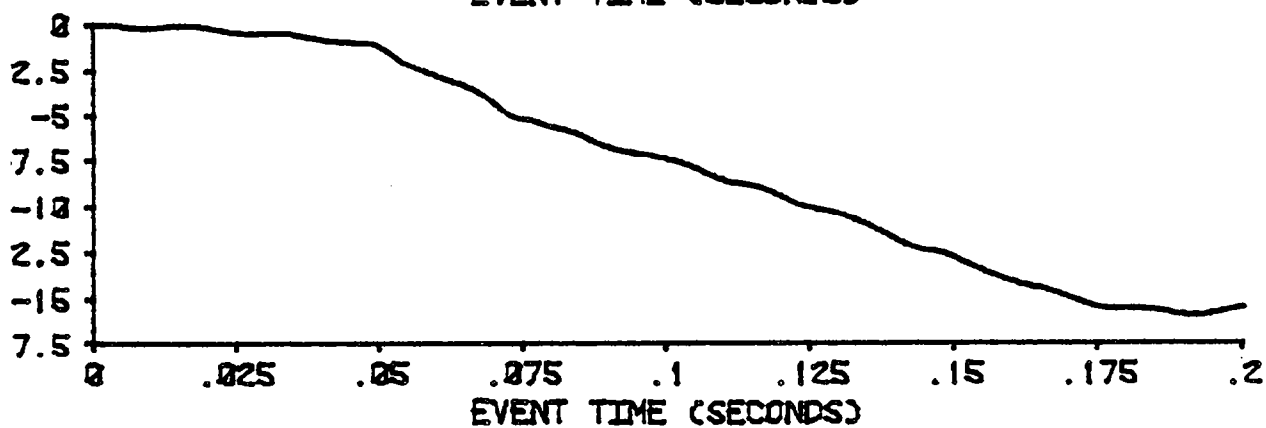
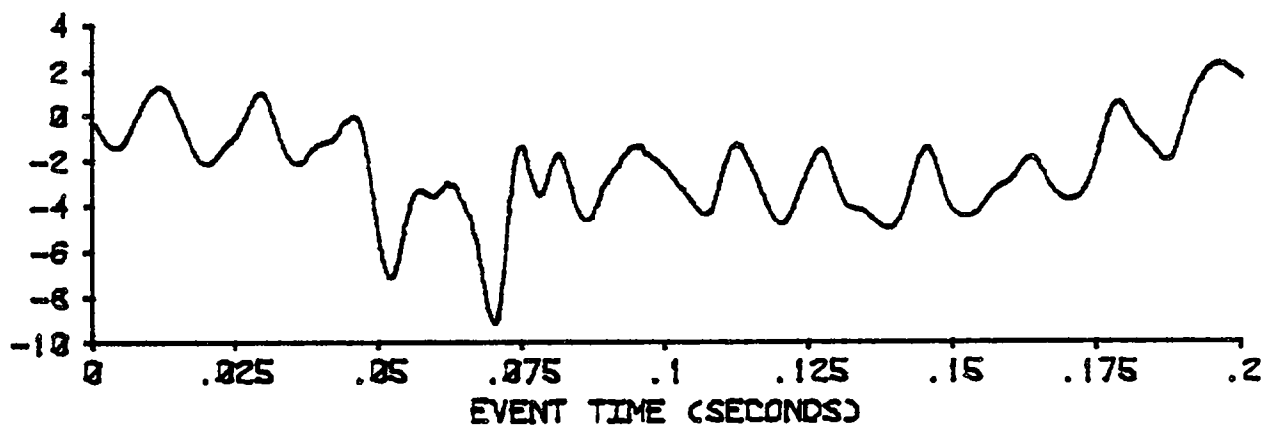
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR FIREWALL IS
FILTER CLASS 60

TEST 1960
11 DEC 60



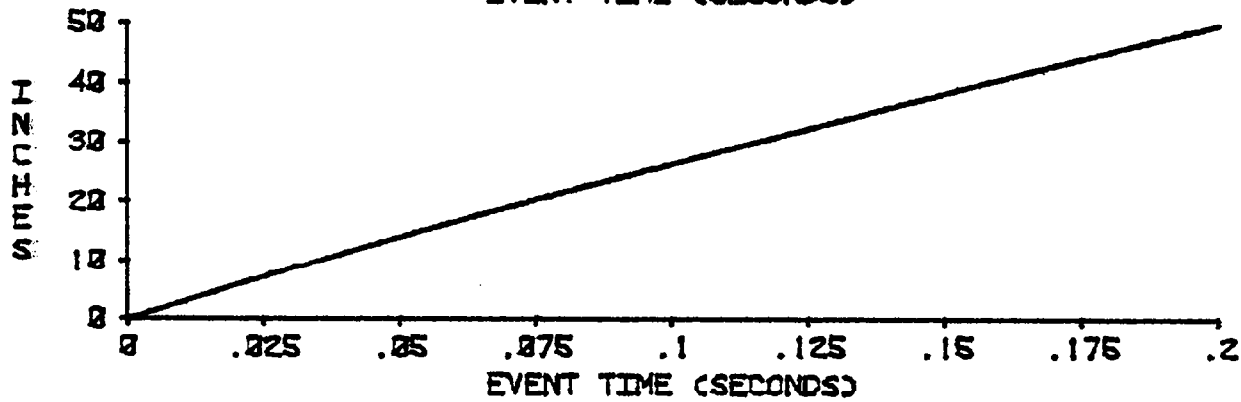
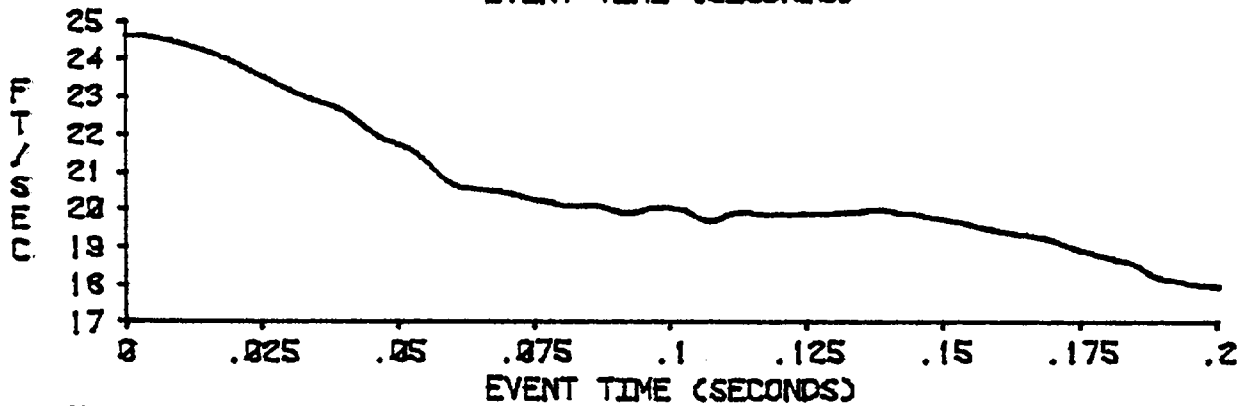
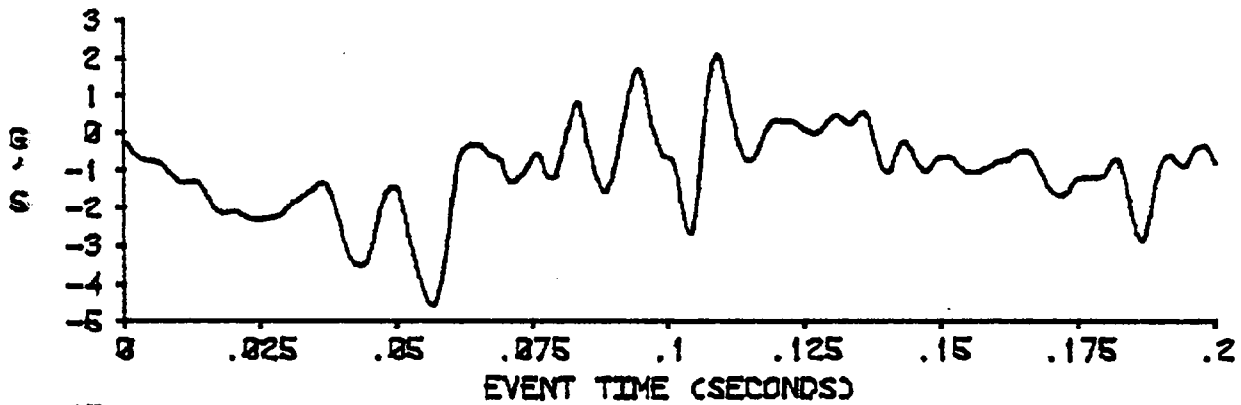
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR FIREWALL RL
FILTER CLASS 60

TEST 1960
11 DEC 60



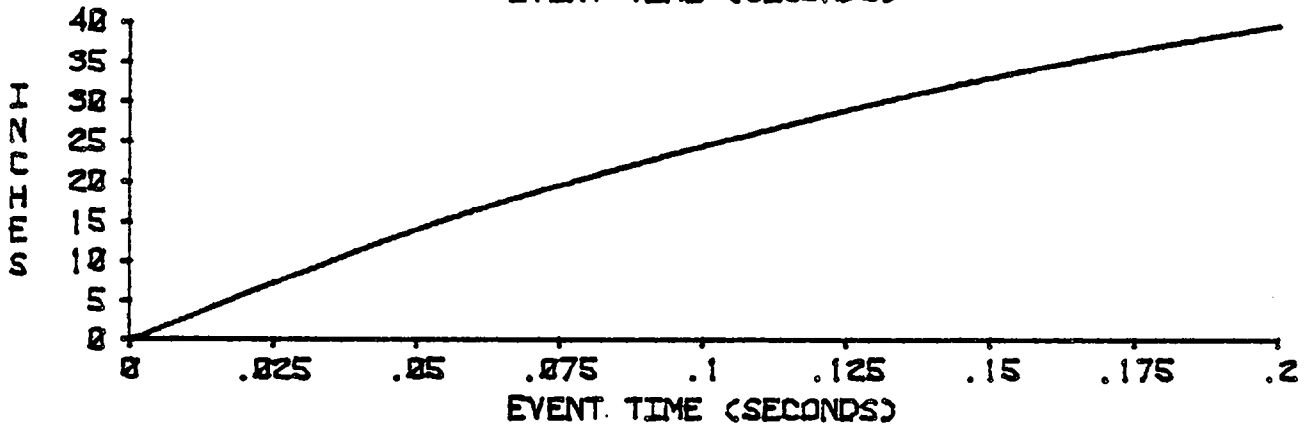
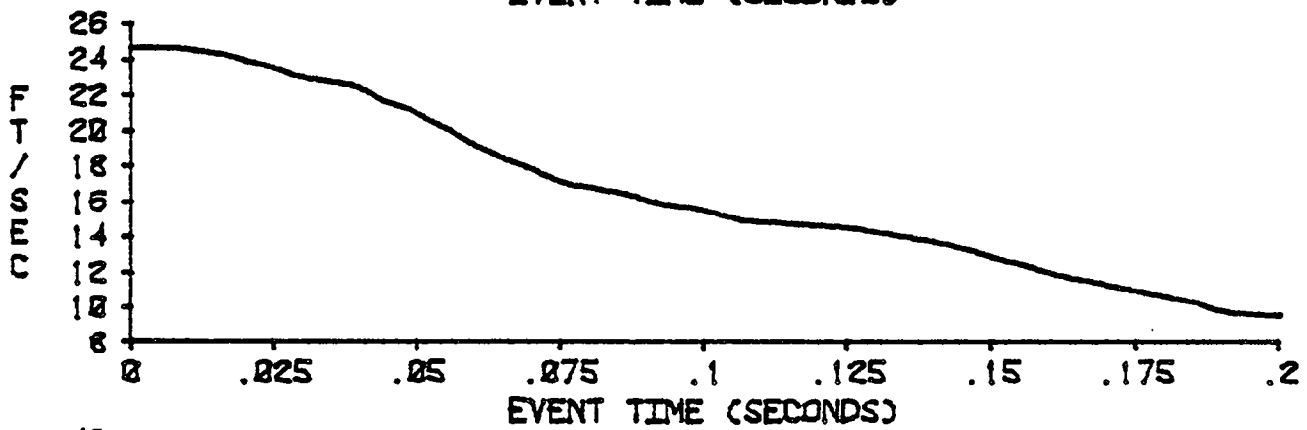
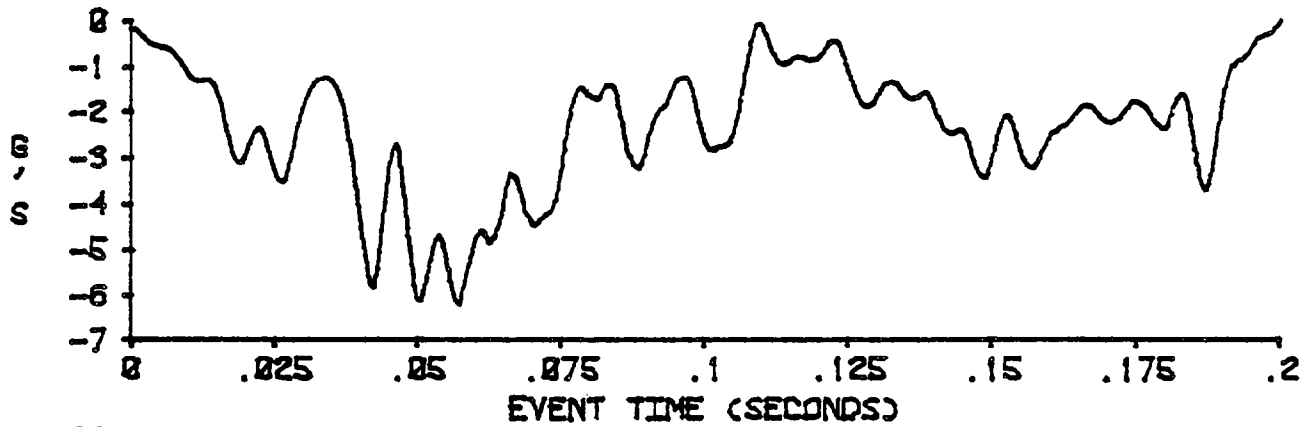
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
RIGHT FRONT ROCKER PANEL AP
FILTER CLASS 68

TEST 1960
11 DEC 60



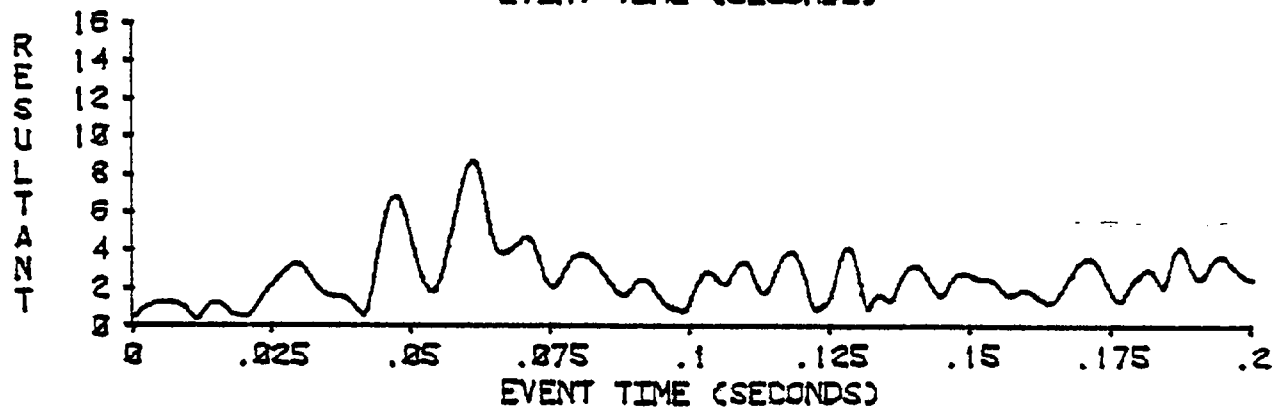
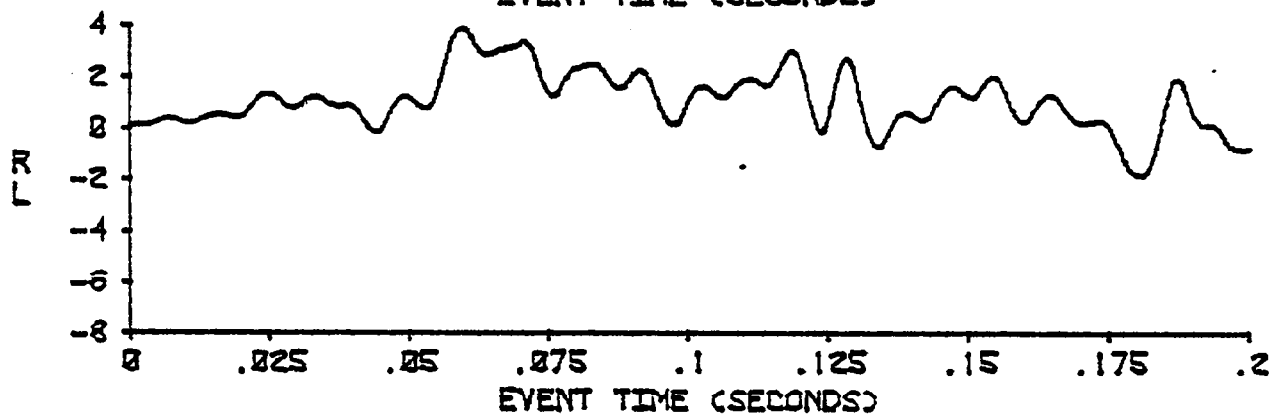
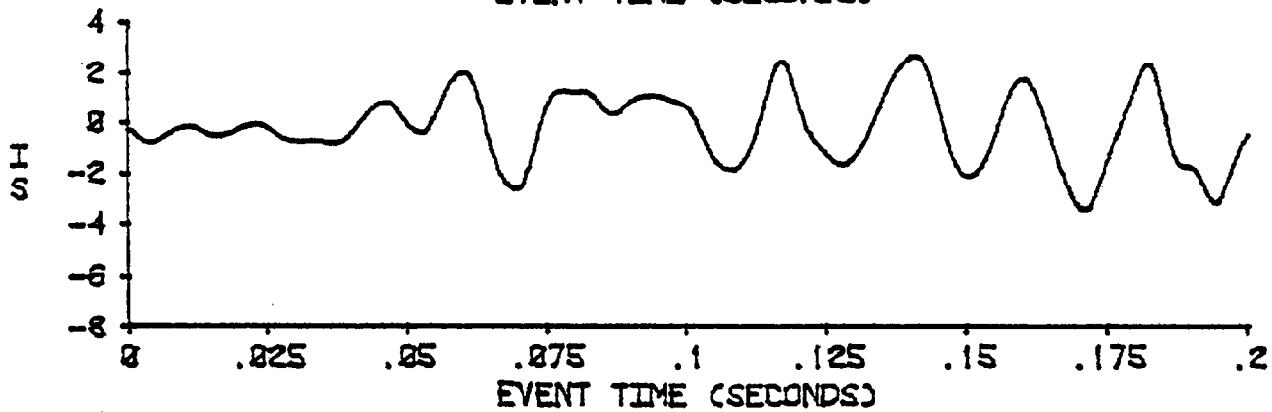
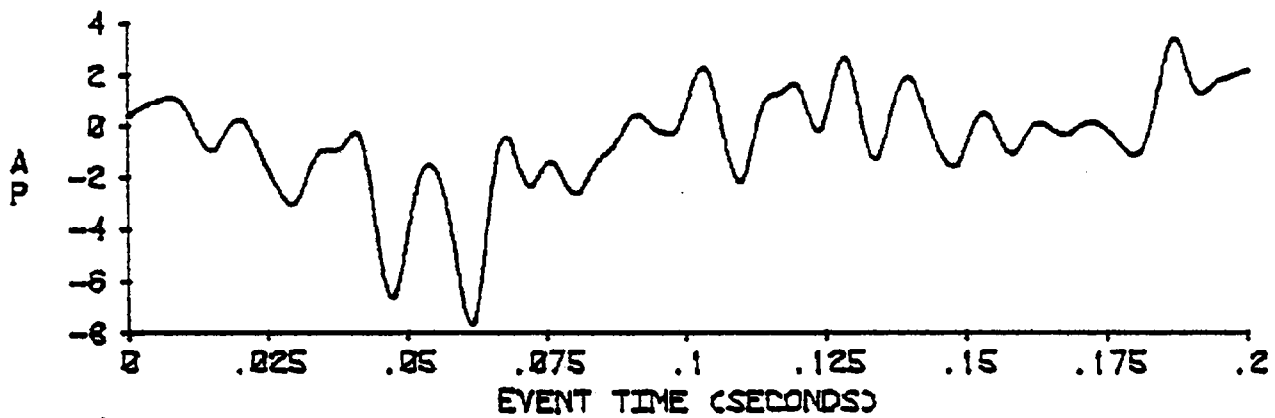
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
LEFT FRONT ROCKER PANEL AP
FILTER CLASS 60

TEST 1960
11 DEC 68



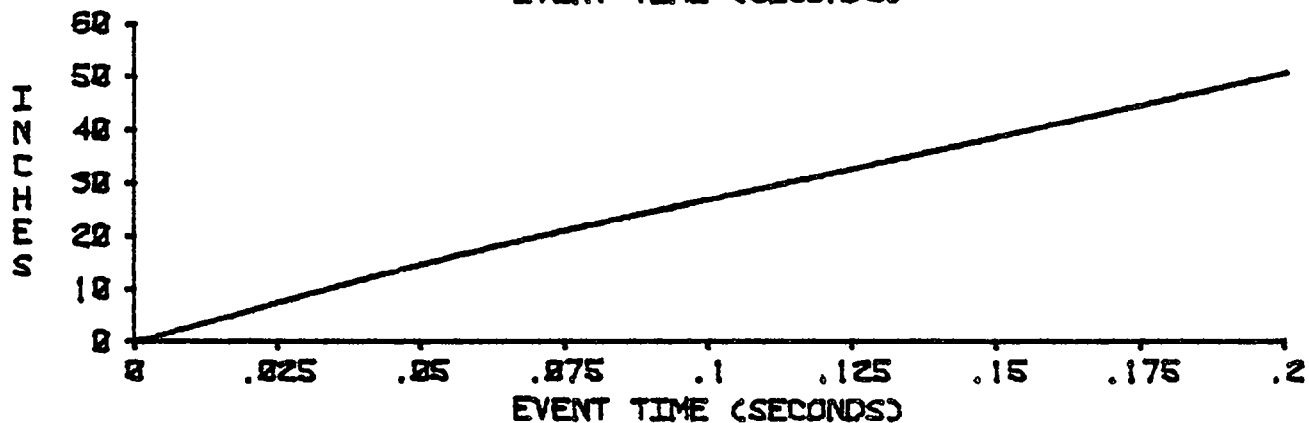
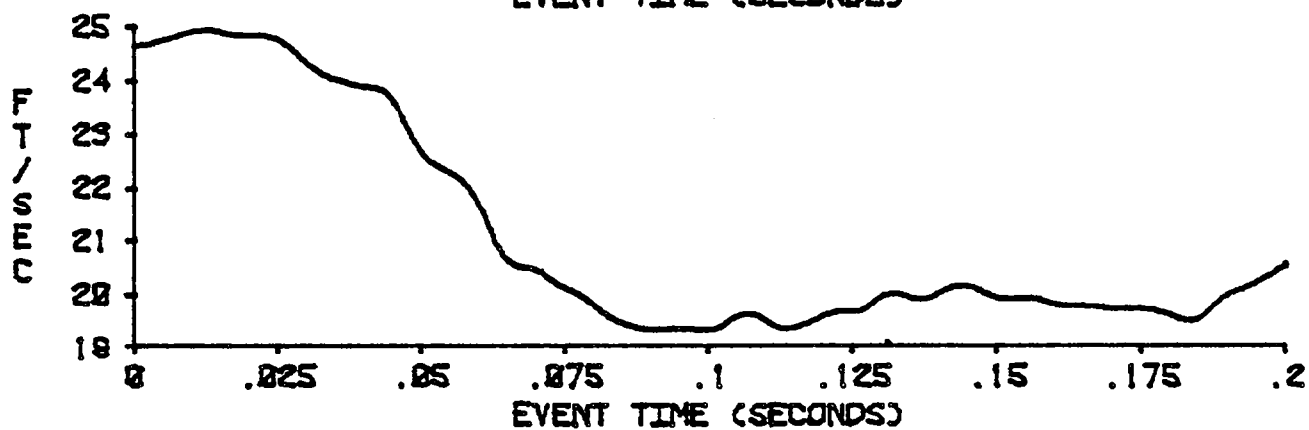
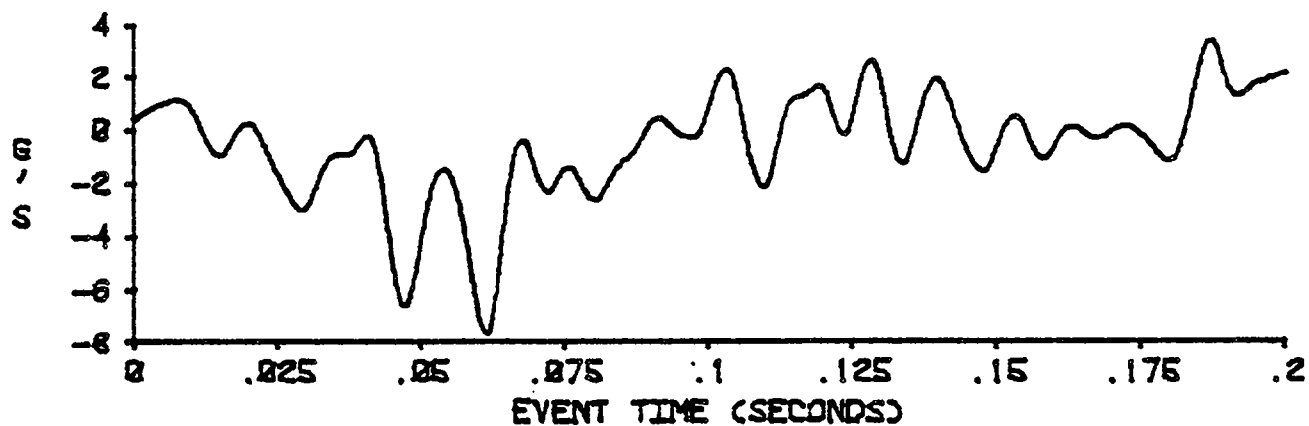
VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR MID TRUNK G'S
FILTER CLASS 60

TEST 1960
11 DEC 80



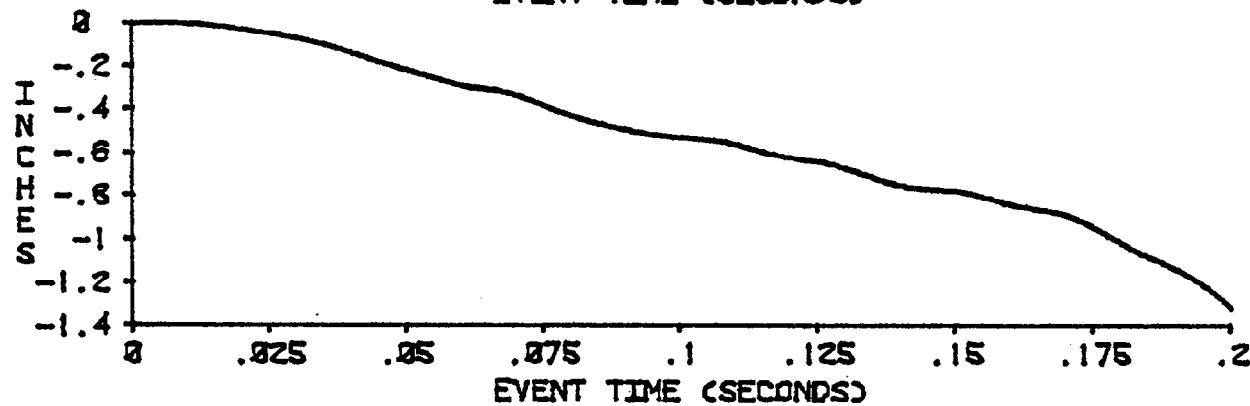
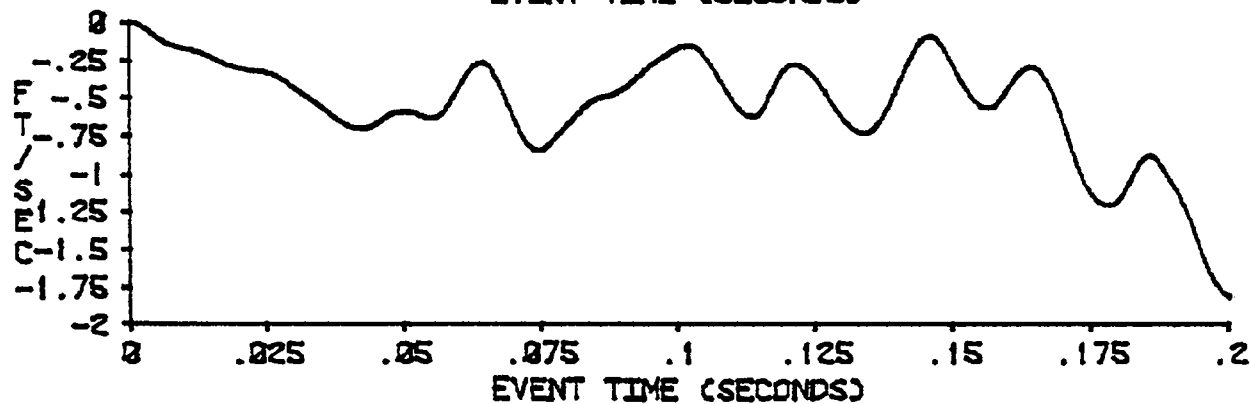
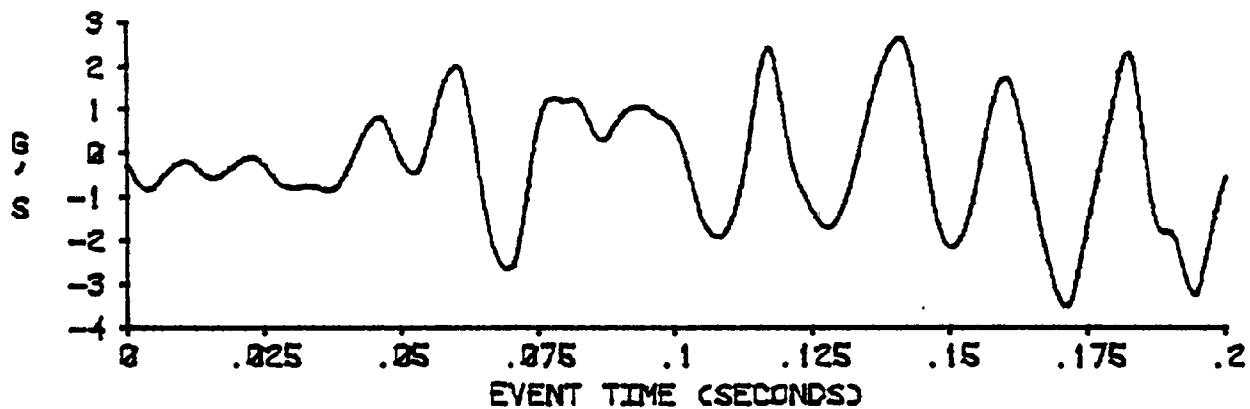
VEHICLE 1 - CHEV MALIBU - 16.6 MPH
CAR MID TRUNK AP
FILTER CLASS 68

TEST 1968
11 DEC 68



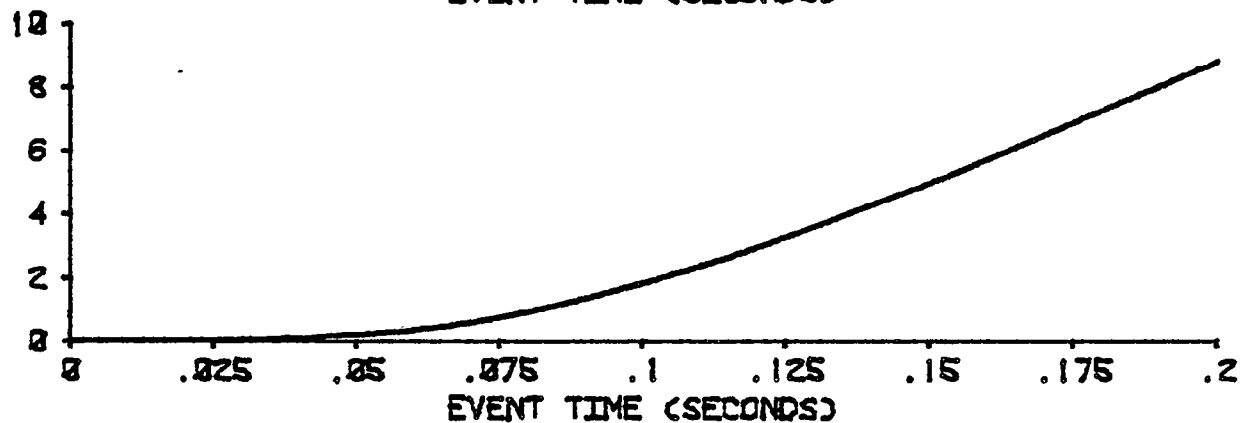
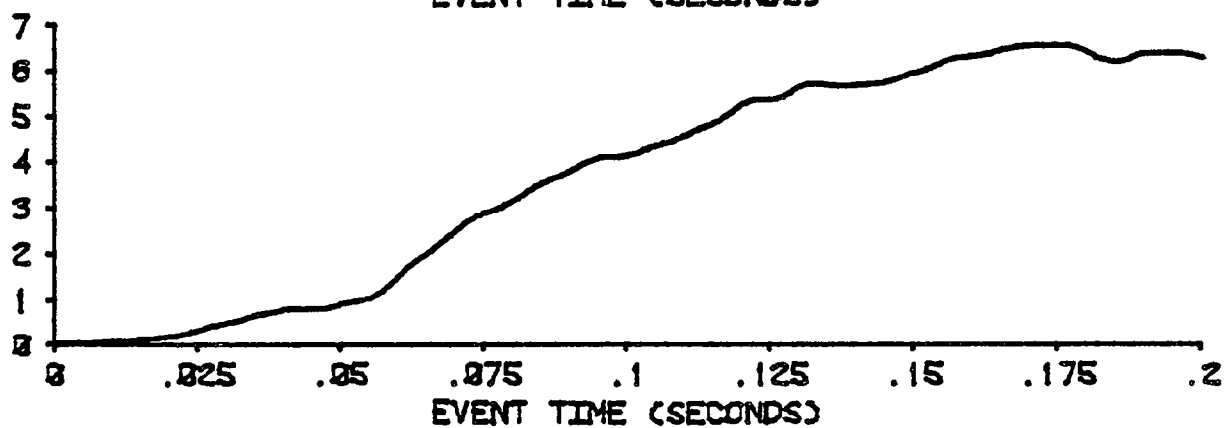
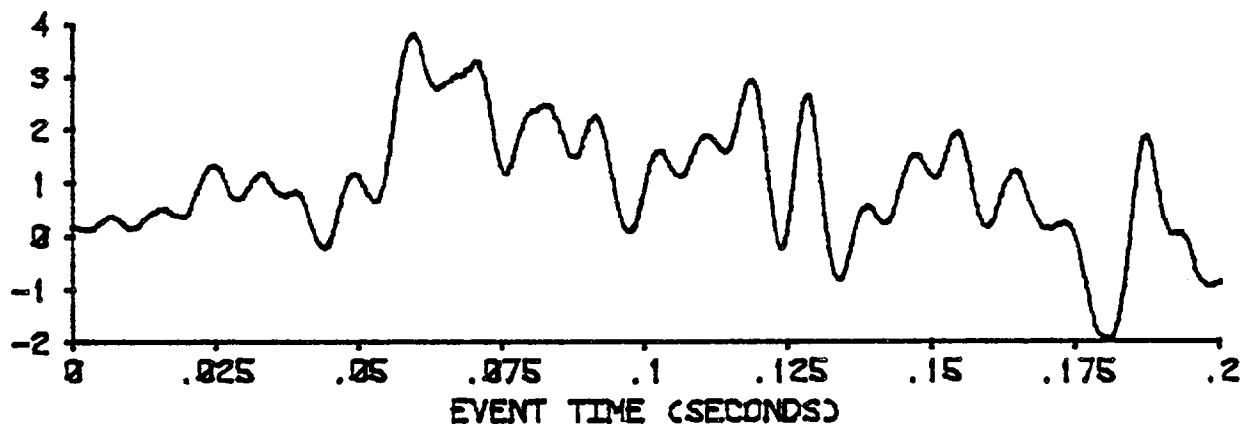
VEHICLE 1 - CHEV MALIBU - 18.8 MPH
CAR MID TRUNK IS
FILTER CLASS 60

TEST 1960
11 DEC 60



VEHICLE 1 - CHEV MALIBU - 16.8 MPH
CAR MID TRUNK RL
FILTER CLASS 60

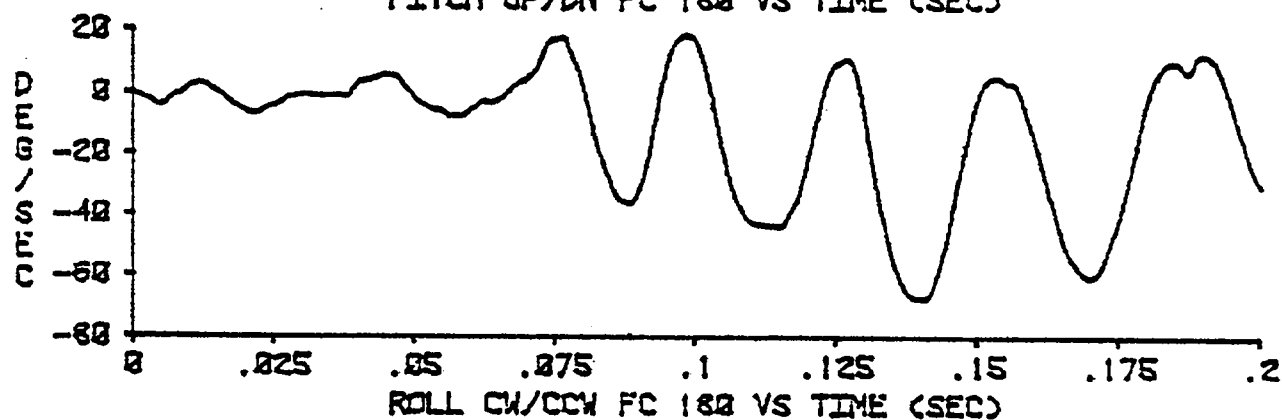
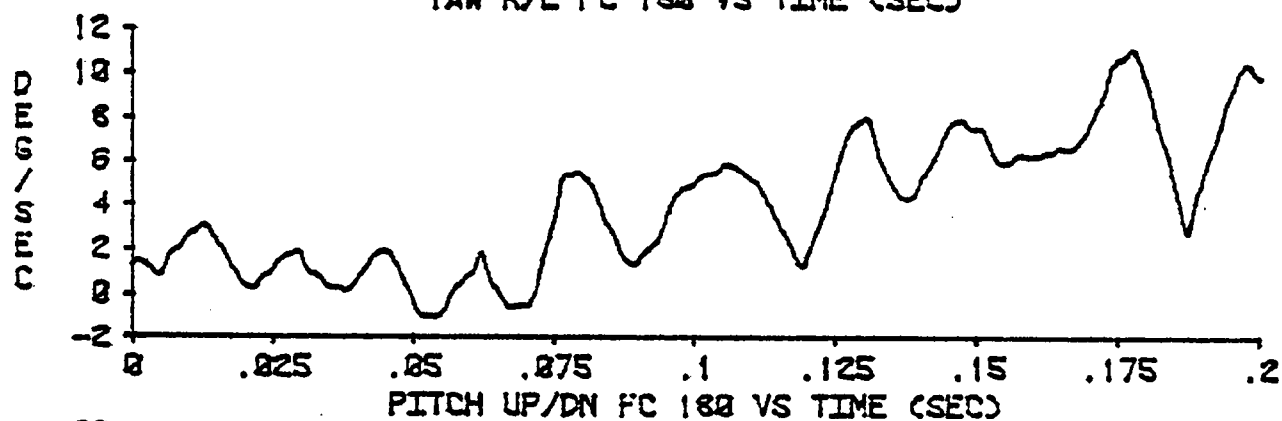
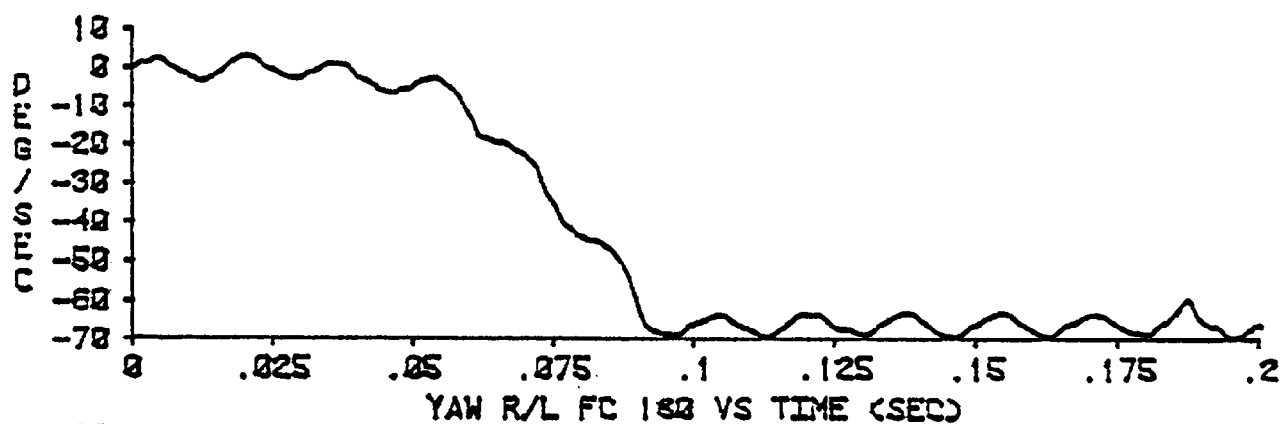
TEST 1960
11 DEC 60



VEHICLE 1 - CHEV MALIBU - 16.8 MPH

TEST 1960

11 DEC 80



VEHICLE 1 - CHEV MALIBU - 16.6 MPH

TEST 1960

11 DEC 68

