Advanced Mode Shape Identification Method for Automotive Application via Modal Kinetic Energy Plots Assisted by Numerous Printed Outputs

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Abstract

Design optimization procedures of full-vehicle simulation models - such a procedure as shown in this paper - require a very fast and reliable mode shape identification. Just because these simulation models necessarily contain a lot of large concentrated masses and mass moments of inertia, e.g., engine, gear, differential, car wheels, steering wheel, mufflers, airbags, and reduced masses from superelement processing, to name just a few, the kinetic energy method is especially destined to accomplish this task. In the present paper, a graphical Modal Kinetic Energy evaluation technique is described in detail. Moreover, the modal kinetic energy plots are a means to investigate the structure's eigenbevior in the low-frequency range, e.g., to see where dynamic vibration absorbers have to be attached and where bushings, and instrumentation for modal testing have to be placed. In summary, the presented graphs make even the most complicated subjects clear and provide the dynamicist with information he can use to achieve a better design quickly. The prints of significant values indicate the degree of coupling between energies in rotational and translational direction per mode and the energy portions of the physical residual chassis structure and the energy portions of appended body and subframe superelements. Representative applications for mode shape identification in automotive engineering, V70, are presented extensively in order to demonstrate the strength of the method. Surely, there are many other applications in the engineering structural analysis field where the advanced mode shape identification method will play key roles.

1. Introduction and Problem Definition

In modal frequency response analysis of a full-vehicle simulation model, such as of the one shown in Figure 1, the following analysis steps

- calculating frequency response predictions,
- determining contributing modes at critical frequencies,
- identifying relevant residual structure mode shapes,
- animating relevant mode shapes on the screen,
- applying the vehicle dynamicist's creativity leading to an innovative redesign

form a complete cycle in the development process. The sequence of pictures in Fig. 3 is the schematic representation of "design optimization procedure" under consideration.

The dynamicist's major goal is to provide the driver and the passengers with a smooth, comfortable ride. Acceleration vs. frequency plots obtained in the first step indicate the critical resonance peaks of the comfort evaluation points.

Since in automotive applications the chassis structure contains numerous viscous dampers - such as shock absorbers, steering damper, dampers used in the hydromount models, etc. - the equations of motion are coupled. Consequently, a single physical displacement in one direction consists of various physical, complex contributions from each of the mode shapes. Therefore, the second step in an analysis using loading with resonance frequency excitation is to calculate the modal contributors, Ref. (5), (8), (9), (10). It is important to note, that using this procedure amazing redesigns may be found by the analyst which automatic structural optimizers are never able to find. By means of the dynamicist's knowledge new elements - such as dynamic vibration absorbers, bushings with high damping, bars, etc. - may be added to the model.

Next, the mode shapes of the modal contributors have to be identified quickly and animated on the screen. Knowing the vibration characteristics of a mechanical system is a requisite for taking suitable measures to reduce obnoxious vibrations.

Finally, having this information the vehicle dynamicist's creativity is required for working on a local redesign.

Many efforts into the identification of mode shapes have been made by several investigators in the past, Ref. (2), (3), (4). Thus, broad and in-depth outlines have been offered. However, these have all relied upon printed outputs. In the search of a quick, accurate, reliable, graphical way to identify mode shapes, step 3, Wamsler, Ref. (1), developed a DMAP alter for plotting modal kinetic energies by grid point for each mode shape in the low-frequency range. The energy plots include the mode shapes' sign (phase) information. Two graphs make it possible to consider the translational and rotational configuration per mode shape on a common basis, viz., on the modal kinetic energies at grid points, achieving a better understanding of the dynamic behavior of the structure.

The plots obtained via an alter offer both a first checking of the structure and identifying mode shapes. In addition, the plots are a means to investigate the structure's eigenbehavior in detail, for instance, to see where dynamic vibration absorbers, bushings, and instrumentation for modal testing have to be placed. The goal of a good design is to avoid energy concentrations. In summary, the graphs make even the most complicated subject clear and provide the dynamic analysis engineer with information he can use to achieve a better design quickly.

In automotive application, responses of high quality are only obtained if both the dynamic mode shapes and the relevant quasi-static mode shapes are present in the analysis, Ref. (7), (11), (12). Both types of mode shapes are obtained via the Component Mode Reduction of the Residual Structure (CMR of RS). Consequently, the CMR of RS has to be applied in all subsequent analyses even in the modal kinetic energy calculation.

Joint efforts of Daimler-Benz AG and MacNeal-Schwendler Corporation have resulted in the melting of two modal kinetic energy alters presented by the authors separately in previous publications leading to an excellent, invaluable visualization tool for ground vehicle dynamicists, Ref. (1), (3), (4).

2. Theoretical Background

The kinetic energy for mode j can be written as

$$KE_j e^{i\omega_j t} = \frac{1}{2}\left\{\dot{\Phi}_j\right\}^T \left[M\right] \left\{\dot{\Phi}_j\right\} e^{i\omega_j t} \tag{1}$$

where

$\left\{\Phi_j\right\}$	= j-th eigenvector, modal displacements of n dofs in mode j
$\left[M\right]$	= mass matrix
i	= imaginary unit = $\sqrt{-1}$
ω_j	= natural radian frequency of mode j
t	= time
$\left\{\dot{\Phi}_j\right\} e^{i\omega_j t}$	= $i\omega_j \left\{\Phi_j\right\} e^{i\omega_j t}$, harmonic modal velocity vector
$\left\{\dot{\Phi}_j\right\}$	= $i\omega_j \left\{\Phi_j\right\}$, modal velocities of n dofs in mode j. $\left\{\dot{\Phi}_j\right\}$ is simply the first derivation of displacements, $\left\{\Phi_j\right\}$, with respect to time

Since eigenvectors are known only to constant factor, all scalar factors may be removed. For convenience, the harmonic variation with time may also be dropped. Thus, the kinetic energy for mode j is expressed as

$$KE_j = \{\Phi_j\}^T [M] \{\Phi_j\} = m_j \tag{2}$$

where

m_j = generalized mass for mode j

Considering a set of eigenvectors, kinetic energy can be written in matrix form as

$$[KE] = [\Phi]^T [M][\Phi] = [m_{gen}] \tag{3}$$

where

$[\Phi]$ = eigenvector matrix

$[m_{gen}]$ = generalized mass matrix, diagonal

If the eigenvectors are orthonormalized with respect to $[M]$ then

$$[KE] = [I] \tag{4}$$

where

$[I]$ = identity matrix

Using the special Element by Element multiplication, $\overset{E}{\underset{E}{*}}$, the kinetic energy per mode is split into its contributions.

The kinetic energies by point and mode with contributions of upstream superelements become

$$[ENERGWUP] = [\Phi^T M]^T \overset{E}{\underset{E}{*}} [\Phi] \tag{5}$$

The kinetic energy is the sum of the kinetic energies of the physical masses and mass moments of inertia of the current SE and that of the generalized masses of the upstream superelements.

$$[ENERGWUP] = \frac{KESG}{KESQ} \quad \begin{matrix} \textit{Physical Portion} \\ \textit{Modal Portion} \end{matrix} \tag{6}$$

$[KESG]$ contains the contributions of the physical masses and mass moments of inertia, $[KESQ]$ of the generalized ones. The physical portion, $[KESG]$, also contains the portion from the static part of Component Mode Reduction of Superelements.

The total energy of physical masses and mass moments of inertia, $\left[KESPHYS\right]$, and generalized masses, $\left[KESGEN\right]$, are thus

$$\left[KESPHYS\right] = \left[111111111...\right]\left[KESG\right] \tag{7}$$

$$\left[KESGEN\right] = \left[111111111...\right]\left[KESQ\right] \tag{8}$$

The kinetic energies of physical masses can be summed up as follows

$$\left[KESTR\right] = \left[111000111000...\right]\left[KESG\right] \tag{9}$$

Consequently, the kinetic energies of mass moments of inertia can be determined by

$$\left[KESRO\right] = \left[000111000111...\right]\left[KESG\right] \tag{10}$$

In order to keep the eigenvectors' phase (sign) information for plotting we can write

$$\left[KESS\right] = \left[\Phi^T M\right]^T \overset{E}{\underset{E}{*}} \left[\left[\Phi_{ij}\right]\right] \tag{11}$$

where

$$\left\|\left[\Phi\right]\right\| = \left[\left|\Phi_{ij}\right|\right]$$

The row vector $\left[111000111000...\right]$ used for summation in equation (9) may also be used as a row partitioning vector, $\left[111000111000...\right]^T$, in partition and merge operations in order to interchange energies in translational and rotational directions and, thus, make plotting energies of mass moments of inertia possible.

It follows that

$$\left[KESS\right] \;=>\; \frac{KESSG}{KESSQ} \quad \begin{matrix} \textit{Physical Portion} \\ \textit{Modal Portion} \end{matrix} \tag{12}$$

and

$$\left[KESSG\right] \;=>\; \frac{KESSTR}{KESSRO} \quad \begin{matrix} \textit{Translational Portion} \\ \textit{Rotational Portion} \end{matrix} \tag{13}$$

$\left[KESSM\right]$ is formed from its interchanged partitions

$$\left[KESSM\right] \;<=\; \frac{KESSRO}{KESSTR} \quad \begin{matrix} \textit{Rotational Portion} \\ \textit{Translational Portion} \end{matrix} \tag{14}$$

Finally, $\left[KESSM\right]$ is merged with $\left[KESSQ\right]$ leading to $\left[KESSMM\right]$:

$$[KESSMM] <= \frac{KESSM}{KESSQ} \quad \begin{matrix} Physical\ Portion \\ Modal\ Portion \end{matrix} \tag{15}$$

3. Guidelines for Applying the Method

3.1 Executive Control Section

The DMAP alter 'checka.v70', provided in the sssalter directory, contains the features from this paper.

Component Mode Reduction of Residual Structure (Mixed Static and Dynamic Reduction of Residual Structure) is a standard feature in V70 now.

3.2 Case Control Section

Following is an example for requesting kinetic energies of masses and mass moments of inertia at grid points in a distinct form.

```
$----------------------------------------------
$
OUTPUT(PLOT)
PLOTTER NASTRAN
PAPER SIZE 30 X 21
$
SEPLOT 0
$
SET 1 = ALL EXCEPT QUAD4 TRIA3
$
SCALE   3.0-03
$
MAXIMUM DEFORMATION 7000.
AXES  MY,X,Z
VIEW  34.27,23.17,0.0
FIND  ORIGIN 1 SET 1
PLOT  MODAL DEFORMATION  0, 10001 THRU 10050,
      MAXIMUM DEFORMATION 1.0,
      SET 1 ORIGIN 1 PEN 2 VECTOR RXYZ
$
$----------------------------------------------
```

Since the natural frequencies of the chassis structure are much lower than those of body and subframes, residual structure plots (PLOT SE 0) are, in general, sufficient. On the other hand, kinetic energy plots of pure shell structures or pure solid structures (SEs) make less sense as, in these cases, swarms of bees would be shown.

In order not to partition the non-linear energies into their components, the VECTOR RXYZ option is recommended in the PLOT command. However, if VECTOR XYZ is chosen then the plotted components are able to show possible coupling between the motions in the different directions. However, the plotted energies in the three directions represent approximations.

3.2 Bulk Data Section

In applying the kinetic energy evaluation method to practical analysis in automotive industry a considerable amount of care is required. Therefore, special attention has to be directed towards the modeling.

Introducing PLOTELs

Many component models, such as the ones of car wheels, engine, discs, etc., are based on a rigid body representation, each incorporating six degrees of freedom which describe general motion. The flexibility is neglected in these cases. Usually the centers of gravity are linked by r-type elements (RBEi) to the mounts or other parts of the structure. Since independent grid points connected by RBEi or MPC do not appear on the structure plots, the kinetic energies of concentrated masses and mass moments of inertia do not appear. To circumvent this lack, the one-dimensional dummy elements PLOTEL have to be introduced for plotting the energies. For convenience, a stand-alone program is used for generating PLOTEL entries related to RBE descriptions.

Similarly, in order to plot kinetic energies of sprung masses connected to CELASi elements, plot elements have to be described in parallel to the corresponding spring elements.

Equally important, for plotting energies of the statically reduced masses from Component Mode Reduction of Superelements at evaluation points, such as the

- driver seat points, front-left, front-right, rear-left, rear-right,
- passenger seat points, front-left, front-right, rear-left, rear-right,

interconnecting PLOTELs have to be introduced between exterior grid points.

Principle Inertia Axes, Displacement Coordinate System, CONM2 Entries

Since car wheels, steering wheel, flywheel, flexible discs, mufflers, etc. do have characteristic principal inertia axes and also free vibrations along these principal inertia axes, the coordinate system used for the description of the principle inertia axes has to also be used for the displacements and kinetic energies, respectively, in order to prevent transformations.

Using the assumption that the centred coordinate systems of the masses, $x, y, z,$ are principle systems, all the products of inertia are equal to zero:

$$I_{xy} = I_{yx} = I_{xz} = I_{zx} = I_{yz} = I_{zy} = 0$$

Consequently, CONM2 elements are used in general rather than CONM1. Then, ***M***, I_{xx}, I_{yy} and I_{zz} are the relevant entries to the CONM2 elements.

SPOINTs

The sum-up and interchange operations in the DMAP alter require that physical SPOINTs do not exist in the description of the model. If they are required nevertheless for any purposes then they have to be represented by grid points with 5 degrees of freedom suppressed. However, SPOINTs for Component Mode Reduction of Superelements and Residual Structure are permitted.

Parameters

```
$-------------------------------------------------------
$
$      PLOT KINETIC ENERGIES IN TRANSLATIONAL DIRECTIONS
PARAM  NOTRKEPL +1
$
$      PLOT KINETIC ENERGIES IN ROTATIONAL    DIRECTIONS
PARAM  NOROKEPL +1
$
$-------------------------------------------------------
```

Other parameters which control the DMAP alter are documented in the checkr.rdm file in the sssalter directory for V70.

4. Identifying Mode Shapes

The purpose of the kinetic energy plots is threefold:

(I) to check the structure,
(II) to identify the mode shapes,
(III) to investigate the structure's eigenbehavior.

What should the dynamicist look for when interpreting modal kinetic energy plots ?

Model Checking

Since a vehicle structure is nearly symmetric, many mode shapes have to be nearly symmetric or cyclic symmetric, too, and, in some cases, of the same frequency.

However, if it is found that considerable differences in kinetic energies per grid do exist between left and right hand side, then, as experience shows, the errors may arise from input errors in mass and/or stiffness and/or geometry (numerical anomalies).

In the analysis of a body shell structure the kinetic energy method may be applied advantageously in order to check wether nonstructural masses such as seats, battery, fenders, spare wheel, etc., are connected correctly onto the sheet metals. It is easily to be seen that concentrated masses do create local mode shapes if the masses are connected, by mistake, to a few grid points only onto the sheet metals. A local mode shape is indicated either graphically by plotted kinetic energy bars per grid or by large kinetic energy terms in a filtered matrix print.

Identifying Mode Shapes

In the low-frequency range, $f < 30$ Hz, i.e., below the first bending mode of the body, the mode shapes are identified easily by interpreting the modal kinetic energy plots, in the high-frequency range, $f > 30$ Hz, by interpreting the printed outputs. The reason for the distinction is that modal kinetic energies are only plotable at physical points. The kinetic energies from SEs at modal coordinates are to be found in the printed output. It is known that the eigenfrequencies of the chassis structure are lower than the one of body and subframe SEs.

It is of interest to note that kinetic energies of the Guyan-reduced masses at SE evaluation master points (e.g., at steering wheel positions 3, 6, 9, and 12 o'clock) are also plotted.

Two graphs make it possible to consider the translational and rotational energy configurations per mode and, consequently, identify the mode shapes.

Investigating the Structure's Eigenbehavior

The plots are a means to investigate the structure's eigenbehavior in detail, for instance, to see where dynamic vibration absorbers, bushings, and instrumentation for modal testing have to be placed. Dynamic vibration absorbers take up obnoxious vibrations. Usually, a full-vehicle simulation model contains vibration absorbers which may not be necessarily recognized.

It is clear from the kinetic energy graphs that a dynamic vibration absorber should be placed at a point of the structure where the kinetic energy per grid point is large and that the device becomes intirely useless when placed at a node of the vibration. The results of a two-mass vibration absorber system can be applied with decent accuracy to a multimass system by replacing the multimass system by an equivalent K-M system:

(1) The mass M of the one-mass system is so chosen that for equal amplitudes at M and the point of multimass system where the damper is attached, the kinetic energy of M equals the kinetic energy of the multimass system in the mode of motion considered.
(2) The spring K of the one-mass system is then so chosen that K/M is equal to the ω^2 of the multimass system in the mode of motion under consideration.

In summary, the graphs make the most complicated subject clear and provide the dynamic analysis engineer with information he can use to achieve a better design quickly. The goal of a good design is to avoid energy concentrations.

5. Representative Application in Automotive Engineering

To verify the novel approach, a superelement normal modes analysis, V70, SOL103, was performed on a full-vehicle simulation model using the jointly developed Modal Kinetic Energy DMAP alter. The plots of Figures 2 and 4 through 6 show the energy distribution of some fundamental vibration modes in the low-frequency range. The plots of kinetic energies of masses and mass moments of inertia are out of frequency order. By interpreting the modal kinetic energy bars the mode shapes can be identified quickly.

6. Concluding Remarks

Various "Modal Kinetic Energy Calculations for Identifying Mode Shapes" have been carried out in the past. However, these have all relied upon printed output. In contrast, the present paper has demonstrated the use of a graphical "Modal Kinetic Energy Evaluation Method" using the standard MSC/NASTRAN plotter. Since kinetic energies are non-linear and a plot module for plotting linear displacements is used, guidelines have been given to the reader in order to prevent transformations. The method is predominantly applicable in analyses of those structures which contain concentrated masses and mass moments of inertia such as the chassis residual structure in an automotive application.

It has been shown that two graphs make it possible to consider the translational and rotational configurations of a mode shape on a common basis, vis., on the modal kinetic energies of masses and mass moments of inertia at grid points per mode shape. Both energies of masses and mass moments of inertia are directly comparable. The plotter displays modal kinetic energie bars placed on the detailed undeformed structure plot.

The main advantage of this method is that it allows the dynamicist to
- check the model,
- get geographical view of the energies per mode,
- recognize the major modal kinetic energies,
- identify the mode shapes quickly,

- achieve a better understanding of the dynamic behavior of the structure,
- obtain information on where to locate dynamic vibration absorbers, bushings, and instrumentation for modal testing.

In summary, the graphs make even the most complicated subject clear and provide the dynamicist with information he can use to achieve an improved design very quickly.

A representative application for mode identification in automotive engineering using V70 was presented extensively in order to demonstrate the strength of the technical enhancement. Surely, there are other applications in the engineering structural analysis field where the advanced mode shape identification method will play key roles.

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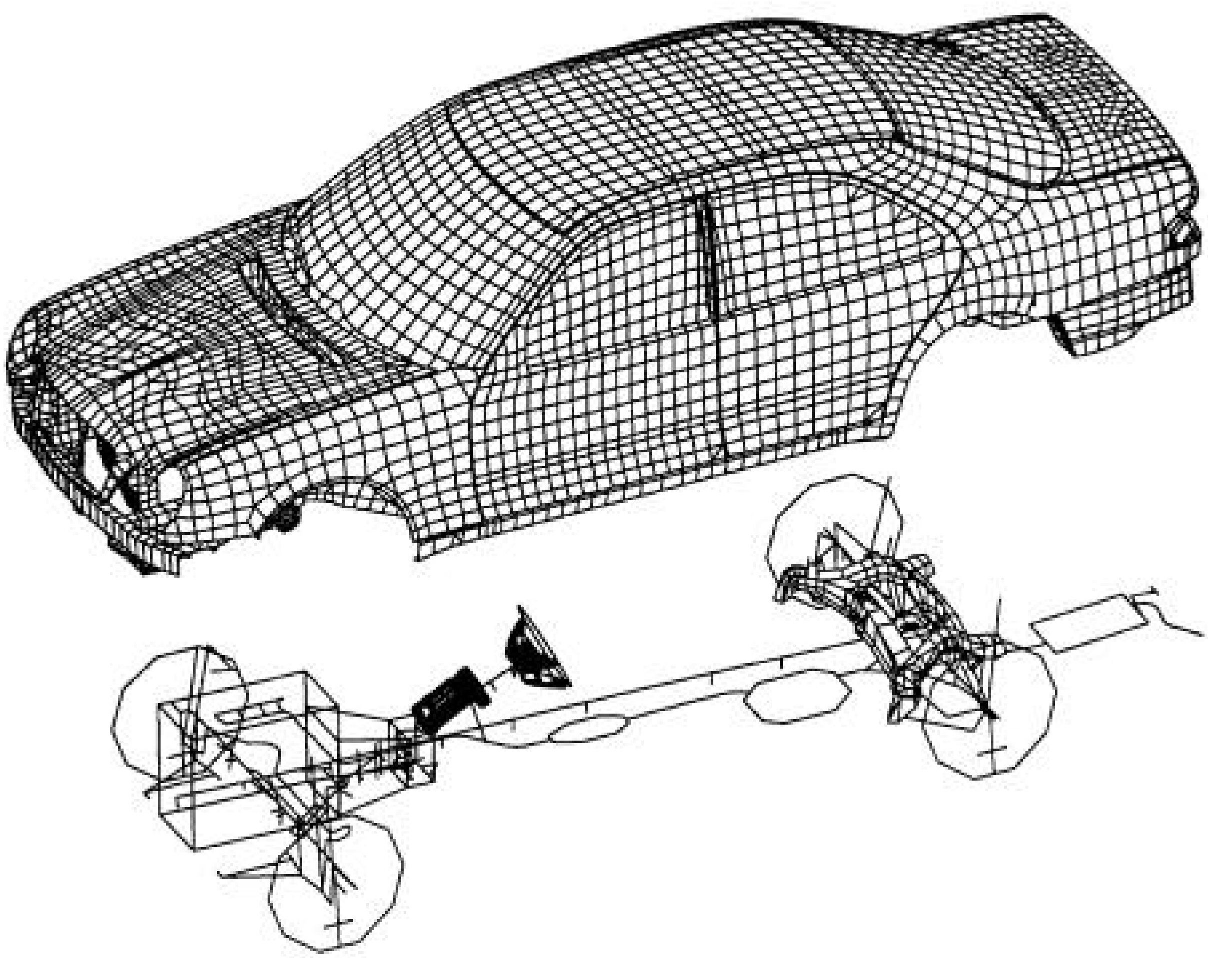


Figure 1: The full-vehicle simulation model

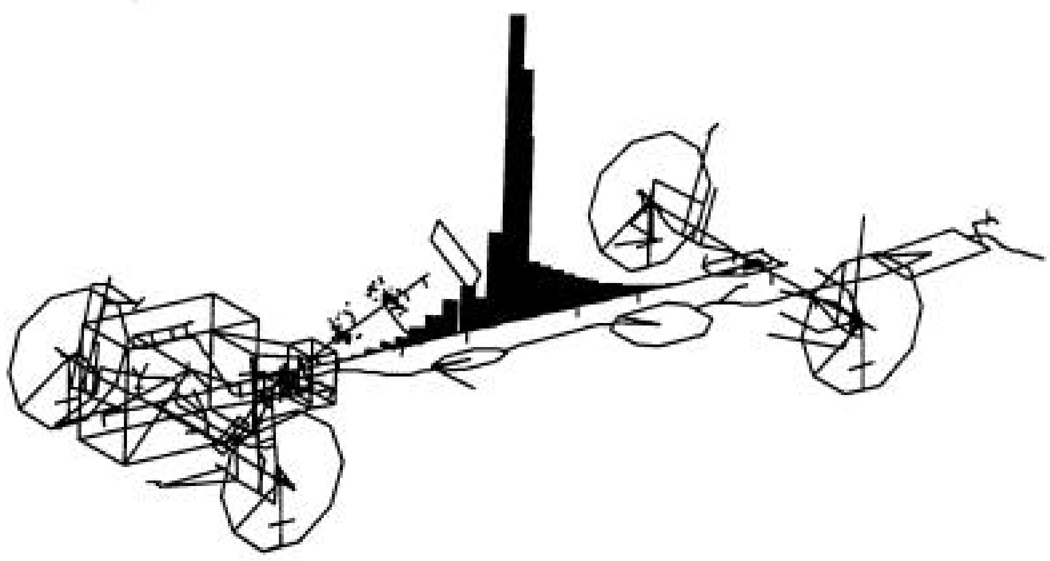
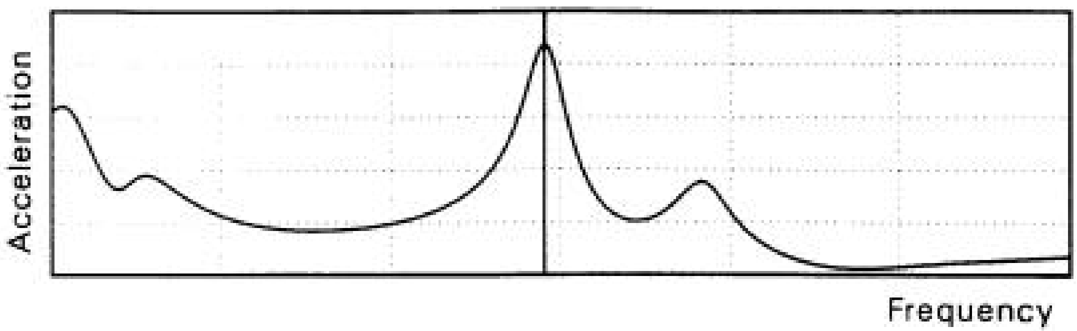


Figure 2: Bending of drive shaft about lateral axis. Translational KE plot

a)



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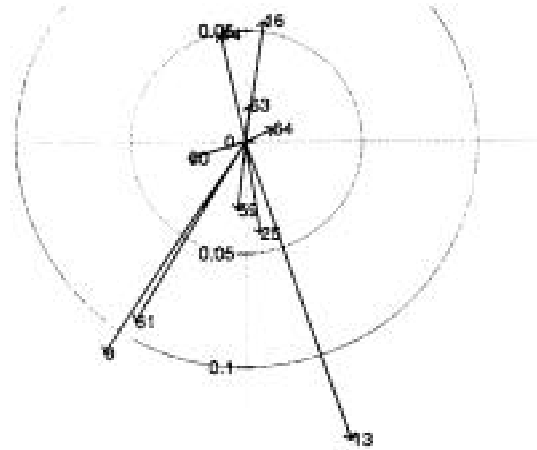
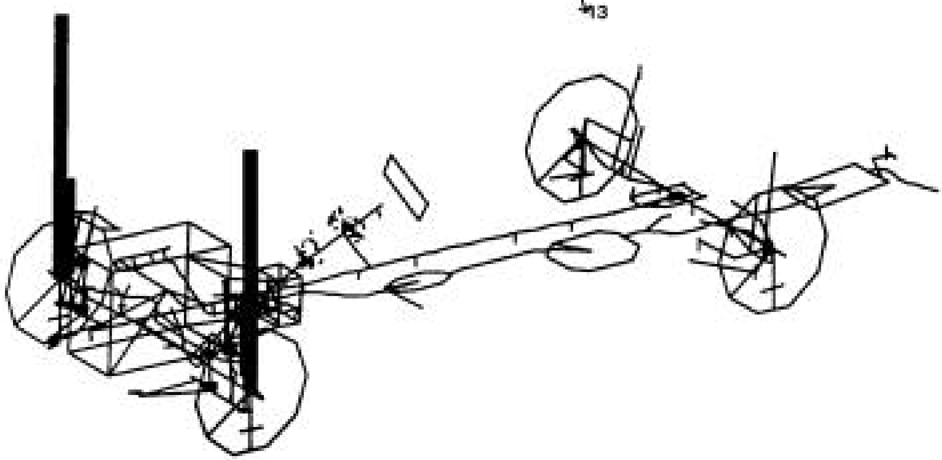


c)



a) Calculating Modal Frequency Response Predictions

b) Determining Contributing Modes at Resonance Frequency

c) Identifying Relevant Mode Shapes

d) Animating Relevant Mode Shapes on the Screen

e) Applying the Dynamicist's Creativity Leading to a Redesign

Figure 3: Development process loop

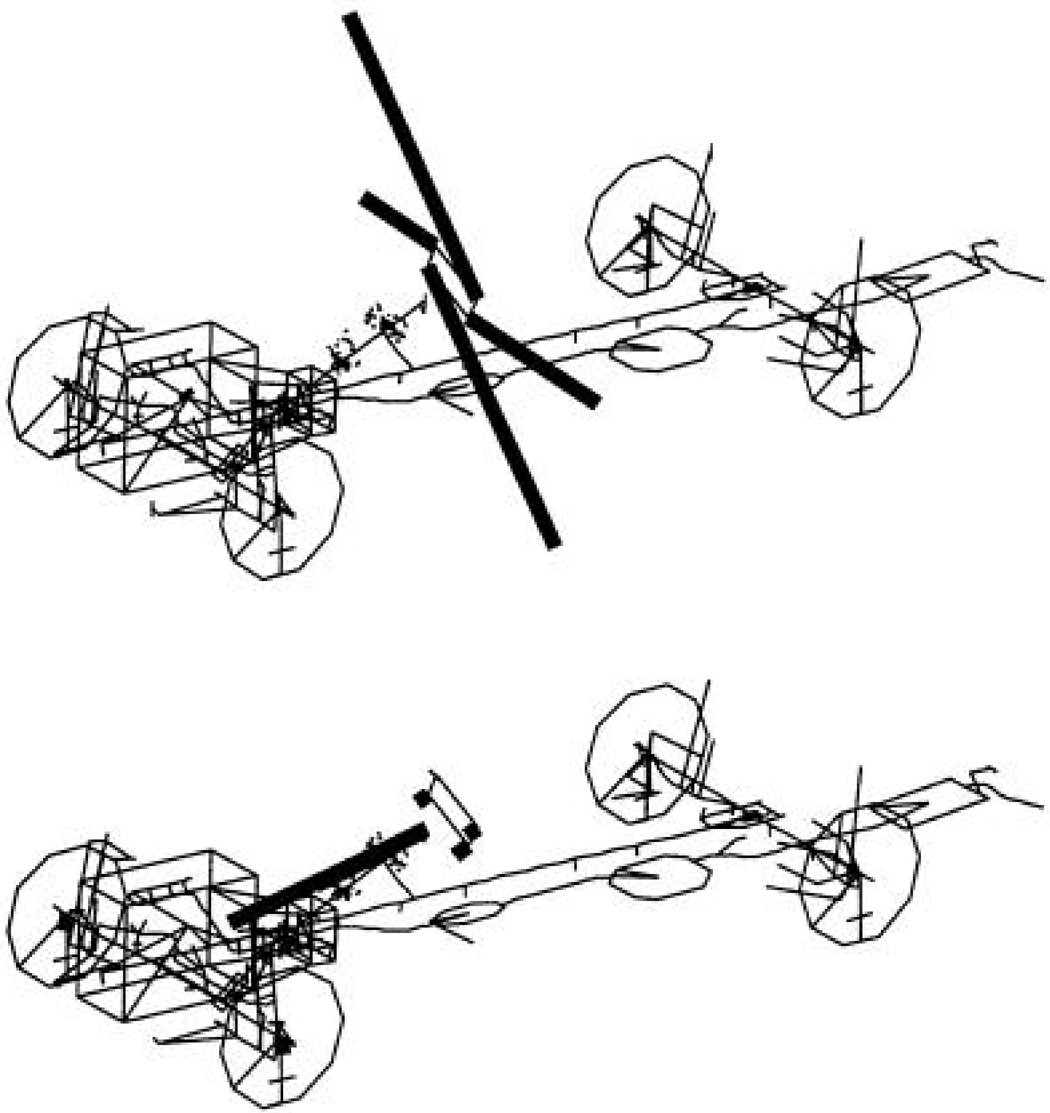
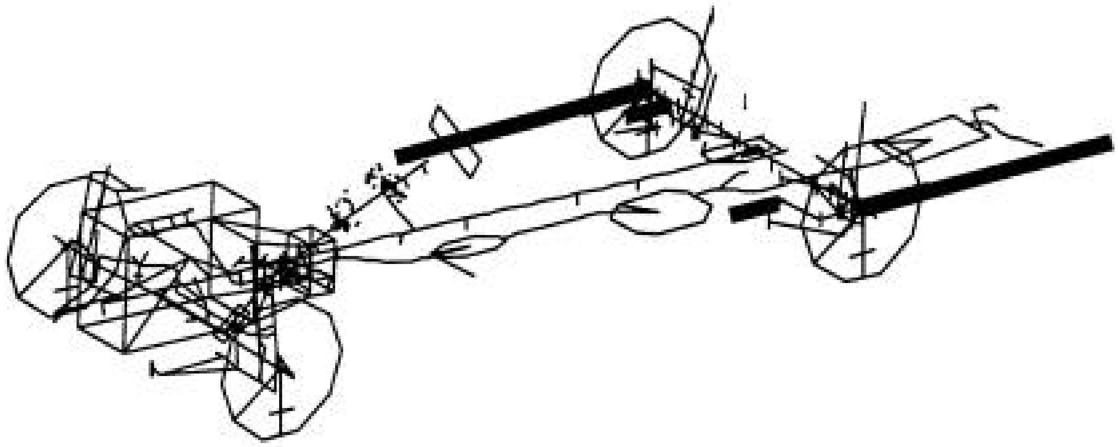


Figure 4: Steering. The energies of the reduced masses of the steering wheel super-element can be seen (top). The energies of mass moments of inertia relate to the steering wheel and the steering column (bottom).

Top: translational KE plot. Bottom: rotational KE plot



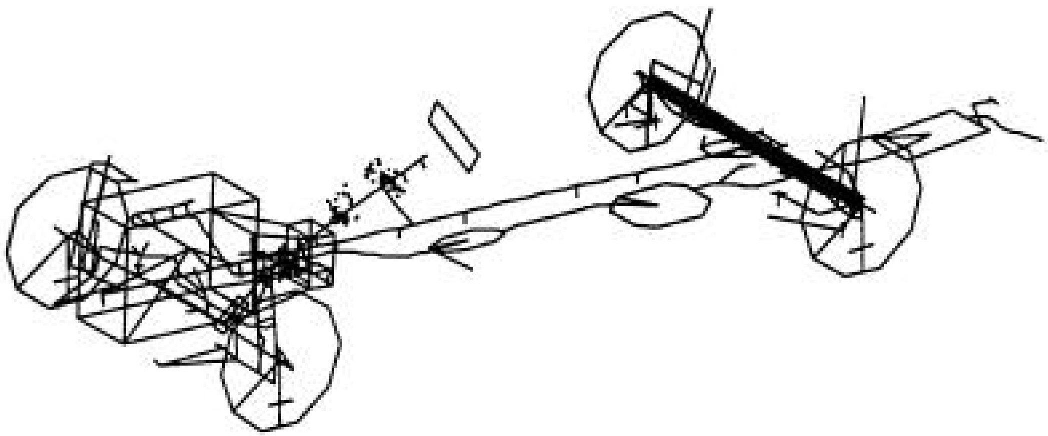


Figure 5: Longitudinal motion of rear wheels, out of phase, coupled with rotation of rear wheels, out of phase.
Top: translational KE plot. Bottom: rotational KE plot

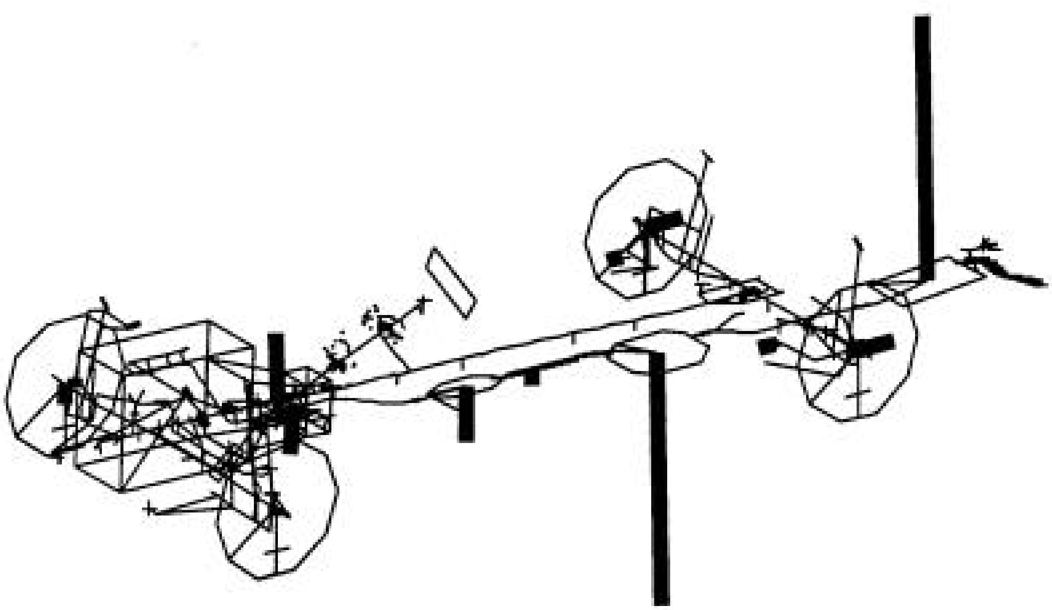


Figure 6: Bending of exhaust system about lateral axis. Translational KE plot