Collision Energy Mitigation through Active Control of Future Lightweight Vehicle Architectures

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Abstract: The paper challenges the current state-of-the-art which is accepted by the automotive industry. Present day vehicles are unsophisticatedly over-engineered and, as a consequence, are uneconomic, hence unsustainable. Vehicles currently under development, however, offer tremendous opportunities for shifting from this position to include onboard active safety systems, e.g. collision avoidance. It is argued that future vehicles should be significantly lighter and exploit the developing safety features to the full. Indeed, such a development would reduce the existing need for crashworthiness. The above arguments coupled with parallel developments in smart materials, paves the way towards a new generation of actively controlled vehicle architecture design. Whilst the move to lighter vehicles, with onboard active safety systems and actively controlled structures, may be seen as controversial, there is a convincing case for a paradigm shift towards a truly sustainable transport future.

1 INTRODUCTION

A mechanical structure is an assembly that serves an engineering function, examples being bridges, vehicles and ships. For an automotive vehicle, the major component of the structure is the architecture. When designing the architecture, optimisation of key components of the structure is performed, such as the profile, configuration, size, cross section and material in order to achieve a desired performance (Hunkeler et al., 2013). With the ever-increasing need to reduce CO_2 emissions, hence fuel and energy consumption, the mass of the vehicle structure, which accounts for approximately one quarter of the total vehicle mass, needs to be reduced. Evidence from ongoing research and development programmes has shown that reducing the mass of the architecture is by far the most effective approach for achieving reduced energy consumption (Lotus, 2010). On the contrary, there is a current need to satisfy crashworthiness criteria, with the structure being designed to passively soften on impact in a predetermined manner. As a result, vehicle architectures have in fact increased in mass, with an average increase of 8kg for passenger vehicles per year from 1980 to 2006 (Ellis, 2011). To challenge this, the UK government has set a target to achieve 60% reduction in CO_2 emissions by 2030 (Hickman and Bannister, 2006). For this target to be met, current trends in the vehicle design need to be reversed by introducing radically new innovative ideas for future vehicles.

Various strategies are currently being deployed to achieve reduced emissions, namely developments with materials and the introduction of optimised hybrid and electric drive trains.

Traditionally, steel has been extensively used for vehicle architectures, however, recent years has witnessed a change, with companies, such as Jaguar Land Rover, now manufacturing certain vehicle models from aluminium. The future of lightweight vehicle architecture design is anticipated to be either carbon fibre reinforced plastics or a mixture of materials. However, due to the high production costs, concerns over recyclability and time consuming processes, there is currently some uncertainty over the future use of composites (Ghassemieh, 2011). This leads naturally to the alternative possibility of employing a mixed material vehicle architecture. This involves selecting the most suitable material for a given purpose, with materials such as, aluminium, steel and magnesium being

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employed (Berger et al., 2009).

The development of optimised drive trains for electric and hybrid vehicles is also taking place. However, there are a number of outstanding issues to be overcome, such as, limited range, lack of charging ports, safety of fuel cells, high cost involved with batteries and, with the additional drive train components in these vehicles, an increase in mass. These potential obstacles would appear to be slowing the uptake of electric and hybrid electric vehicles for the present-day average road user.

As mentioned earlier, one of the compelling arguments against reducing mass is the current need to comply with requirements of crashworthiness, since this has evolved to become probably the most important design aspect of a vehicle. Early attempts to absorb energy in a controlled manner during a collision have included hydraulic rams for the longitudinal members of the vehicle architecture (Jawad, 2003) and bumper dampers. Recently, pyrotechnics have been used to actively control the softening in the event of pedestrian impacts (Thatcham, 2012).

The use of advanced driver assisted systems (ADAS) on vehicles, such as autonomous emergency braking, collision avoidance, collision mitigating braking and electronic stability control, which are now being fitted as standard on vehicles, will ultimately reduce the dependence on, or even override, the driver. Effectively, the onboard ADAS and active safety systems in the future are expected to reduce the number of collisions as well as the velocities of such collisions. Significant developments in the deployment of ADAS is currently taking place, with car manufacturers, such as Volvo, investing in active safety systems, with their aim being to achieve zero fatalities or seriously injured passengers in a Volvo by 2020 (Eugensson, 2009).

2 PROBLEM FORMULATION

In this section, current and future trends concerning vehicle mass are presented, along with formulating the problem for actively controlled structures.

2.1 Vehicles of Dissimilar Mass

The move towards lightweight vehicles will inevitably involve even greater differences in masses between vehicles, such as passenger vehicles, lorries and trucks. A small vehicle can be of mass as little as 800kg whereas a laden/unladen lorry could easily be a factor 20/10 times heavier (or even more) i.e. presenting problems to smaller vehicles. This can be highlighted via an example, illustrating that the future low mass vehicle is vulnerable compared to a larger vehicle. Consider, for example, a collision between a moving and stationary vehicle with dissimilar masses, such that a larger vehicle initially travelling at a velocity, given by $V_a =$ 12m/s, collides with a smaller stationary vehicle with a velocity, $V_b = 0$. Denote the mass of the vehicles as m_a and m_b given by 1000kg and 500kg, respectively. Denote the final velocity, i.e. after the collision, as V_f. It is well known that the conservation of momentum can be expressed as $(m_a V_a + m_b V_b) = m_{a+b} V_f$, where $m_{a+b} = m_a + m_b$. It can be deduced that the final velocity of the combined mass of the two vehicles is 8m/s.

From the principle of conservation of energy, the kinetic energy before and after the collision must be equal, consequently $\frac{1}{2} m_a V_a^2 + \frac{1}{2} m_b V_b^2 = \frac{1}{2} m_a V_f^2 + \frac{1}{2} m_b V_f^2 + \Delta E$, where ΔE is the collision energy dissipated within the vehicle structures. It can be deduced that ΔE for this particular collision is 24kJ.

It is known (Schmidt et al., 1998) that the ratio of absorption of energy from a collision is proportional to the change in the vehicle velocities, denoted ΔV_a and ΔV_b where $\Delta V_a = |V_f - V_a|$ and $\Delta V_b = |V_f - V_b|$. It can also be deduced that the ratio of ΔV_a:ΔV_b is the same as m_b: m_a, so that the smaller of the two vehicles is always the more vulnerable.

The above example serves to highlight the need for active control of automotive structures in order to share the energy absorption where, should a small vehicle collide with a larger vehicle, the larger vehicle structure will soften to absorb the smaller vehicle (with the smaller vehicle being allowed to stiffen on impact). The objective of the approach is to control the structural properties to ensure optimum mitigating energy absorption in an actively controlled manner.

2.2 Brief Review of Smart Materials

Rapid advances in the electronics industry have taken place in recent years with on-board embedded micro-processors and control systems being applied in a wide range of applications, with the automotive sector providing many examples, e.g. ADAS. Developments have also taken place in the area of smart materials, whereby electronic devices, e.g. piezoelectric systems, are bonded to material to produce enhanced structural properties. Whilst the application of actively controlled structures have been reported in the literature for aircraft, bridges, buildings and spacecraft to enhance their structural properties, there have been little or no published reports in the automotive sector (Gabbert, 2002).

Smart materials have the ability of possessing functions such as sensing, actuating and controlling. These functions can be used in a structure where there is a need to react under the influence of the environment, i.e. an induced force (Gupta and Srivastava, 2010).

An example of a smart material is a piezoelectric device (formed by an alloy of lead (Pb), Zinc (Zn) and Titanium (Ti)) which is often referred to as PZT. When a mechanical stress is applied, the piezoelectric effect produces a charge caused by the motion of electric dipoles within the material, known as the direct effect. This can be used for energy harvesting. Piezoelectric materials also exhibit a reciprocal effect, known as the converse effect. When an electric field is applied the result is a mechanical response, in this case a displacement. Other examples include shape memory alloys, where given an electric current input, i.e. heat, the shape of the structure can be changed, thus, varying its rigidity (Leo, 2007).

This brief introduction has demonstrated the potential use of smart materials for changing the properties of vehicle architectures.

2.3 Position Statement

Because vehicle architecture design is currently driven by crashworthiness performance, it follows that during normal every-day driving conditions the vehicle structure is unsophisticatedly over-engineered; with the architecture being significantly different if crashworthiness requirements could be met in a more efficient manner, i.e. actively controlled, as outlined above.

It is at this juncture that the hub of the issue becomes apparent. This issue, coupled with rapid developments in active safety and in the deployment of ADAS, forms a convincing premise for the position statement. Thus, the position statement is as follows: Due to effective onboard safety systems in the future, collisions will be fewer and of lower velocity, thus, markedly reducing the required levels of crashworthiness. If this is the case, then future vehicle structures will be lighter, thereby exacerbating the dissimilar mass problem, outlined in Section 2.1. Therefore, it is argued that vehicles need to stiffen or soften, allowing the structural properties to be actively controlled depending on the collision being encountered. For example, if two vehicles of different mass collide at the same velocity, it would be expected that the lighter vehicle would stiffen and the larger vehicle would soften in order to optimally share the collision energy. Hence, active control of smart structures is deployed to mitigate the effects of these collisions in order to control the energy absorption that is required for each vehicle. It is conjectured in this position paper that advances in smart materials, such as shape memory alloys and piezoelectric materials coupled with predictive and adaptive control, will lead to research to provide a better solution to the collision energy mitigation problem.

Following the argument through, future vehicles equipped with ADAS and active control, will be significantly lighter, hence, improving efficiency, satisfying CO_2 legislation and at the same time, maintaining or improving safety, whilst reducing the current requirements for crashworthiness; rather tending more towards a reduced aggressivity of the vehicle fleet.

3 MODELLING STRUCTURES FOR CONTROL

3.1 Preliminary Considerations

Each member or beam within a vehicle structure is considered to be modelled as a mass, spring and damper system. This is analogous with electrical systems being modelled as combinations of capacitance, inductance and resistance. In a mechanical system, energy is stored in the mass and spring elements and dissipated through damping. There are basically two conceptual modelling approaches when dealing with mechanical structures, namely the nodal approach, where displacements, velocities and accelerations at specific points (or nodes) of a structure are of interest, and the modal approach, where the spectral properties, i.e. the eigenvalues, eigenvectors and corresponding natural frequencies of the entire structure are of interest. Whilst the use of the nodal approach, leads to dependencies of nodes on each other (i.e. coupled), use of the modal approach gives rise to each mode being independent of each other (i.e. uncoupled). As it will become clear, the design of control algorithms for modal control is considerably simpler than the control of a nodal system, particularly when considering only a few modes. As a consequence, attention will be given to a particular form of the state-space modal model, which is developed here for the design and realisation of modal control algorithms.

It is assumed that the stiffness and dissipative damping may be actively controlled in certain members of the structure, hence changing the overall structural properties. The simple interconnected two member structure given in Figure 1 is considered for the purpose of illustrating the modelling and control approach.

The simple structure illustrated in Figure 1 has two degrees of freedom denoted by the displacements q_1 and q_2, hence two structural modes. As a starting point it is convenient to consider a nodal model, which is subsequently transformed to a modal model for both modelling and control. In this regard it is convenient to assume that when a vehicle, modelled as an interconnected structure, is unconstrained, the spring stiffness and damping factors at the extremities of the structure, k_1, k_3, d_1 and d_3 in Figure 1, are set to zero. However, upon collision with an obstacle (or another vehicle) it is assumed that, e.g. k_1 and d_1, become non-zero. In effect an unconstrained vehicle may be considered as being in rigid body mode. In constrained mode, i.e. when k_1 and d_1 become non-zero, the vehicle then becomes a flexible structure.

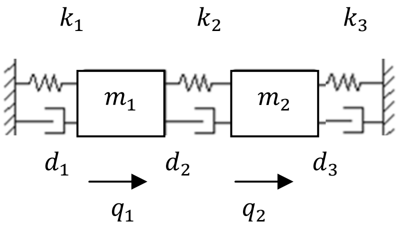

Figure 1: Interconnected two-member system, where m_1, m_2 denote the system masses, k_1, k_2, k_3 denote the spring stiffness coefficients and d_1, d_2, d_3 denote the damping coefficients.

3.2 Nodal Model

It is convenient to begin by considering a flexible structure in the familiar nodal coordinates represented by the following second order matrix differential equation:

$$M_n\ddot{q} + D_n\dot{q} + K_n q = B_n u \tag{1}$$

$$y = C_{nq}q + C_{nv}\dot{q} \tag{2}$$

where the subscript n denotes nodal representation. Let n_d denote the number of degrees of freedom, r denote the number of outputs of interest and s denote the number of inputs. The quantities in (1) and (2) are defined as:

q is the n_d x 1 nodal displacement vector
$\dot{q}$ is the n_d x 1 nodal velocity vector
$\ddot{q}$ is the n_d x 1 nodal acceleration vector
u is the s x 1 input vector
y is the r x 1 output vector
M_n is the n_d x n_d nodal mass matrix
D_n is the n_d x n_d nodal damping matrix
K_n is the n_d x n_d nodal stiffness matrix
B_n is the n_d x s nodal input matrix
C_{nq} is the r x n_d nodal output displacement matrix
C_{nv} is the r x n_d nodal output velocity matrix

For convenience, let the masses $m_1 = m_2 = 1$, the stiffness values $k_1 = k_2 = 2$ and $k_3 = 0$ and let the damping matrix be proportional to the stiffness matrix, such that $D_n = 0.01K_n$. Consider the case of a force input u at mass 2, with outputs being velocity of mass 2 and displacement and velocity of mass 1. This yields the following matrices: $M_n = diag(m_1,\ m_2)$ so that $M_n = I_2$. The stiffness and damping matrices are:

$$K_n = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 + k_3 \end{bmatrix} = \begin{bmatrix} 4 & -2 \\ -2 & 2 \end{bmatrix} \tag{3}$$

$$D_n = \begin{bmatrix} d_1 + d_2 & -d_2 \\ -d_2 & d_2 + d_3 \end{bmatrix} = \begin{bmatrix} 0.04 & -0.02 \\ -0.02 & 0.02 \end{bmatrix} \tag{4}$$

The input and output matrices are:

$$B_n = \begin{bmatrix} 0 \\ 1 \end{bmatrix}, C_{nq} = \begin{bmatrix} 1 & 0 \\ 0 & 0 \end{bmatrix}, C_{nv} = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix} \tag{5}$$

3.3 Modal Model

The modal coordinate presentation is obtained from the nodal representation via a transformation. By setting the damping matrix D_n in (1) to zero and considering the unforced case (i.e. u = 0) the nodal representation takes the form

$$(M_n\ddot{q} + K_n q) = 0 \tag{6}$$

Assume that the solutions are of the form $q = \phi e^{j\omega t}$ where ϕ is a n_d x 1 vector so that $\dot{q} = j\omega\phi e^{j\omega t}$ and $\ddot{q} = j^2\omega^2\phi e^{j\omega t}$ i.e. $\ddot{q} = -\omega^2\phi e^{j\omega t}$. Substitution into (6) leads to

$$(K_n - \omega^2 M_n)\phi e^{j\omega t} = 0 \tag{7}$$

It is also known that when K_n and M_n are symmetric and positive definite the roots of $det(K_n - \lambda M_n) = 0$ are real, where the $\lambda_i = \omega_i^2$ are necessarily positive and represent the squares of the natural modes or frequencies of the structure. Equation (7) is essentially an eigenvalue problem (Wilkinson, 1965). Because K_n is positive definite there exists an orthogonal matrix Φ such that $\Phi^T K_n \Phi = diag(\omega_1^2, \omega_2^2 ... \omega_n^2) = \Omega^2$, where Φ is known as the modal matrix and comprises the n eigenvectors of dimension n_d x 1 as its columns, i.e. the non-trivial solutions of (7), where the ϕ_i are in fact the eigenvectors.

Define the modal variables as q_m, $\dot{q}_m$ and $\ddot{q}_m$ such that $q = \Phi q_m$, $\dot{q} = \Phi \dot{q}_m$ and $\ddot{q} = \Phi \ddot{q}_m$ substituting for q, $\dot{q}$ and $\ddot{q}$ in (1) and pre-multiplying by Φ^{T} leads to

$$\Phi^T M_n \Phi \ddot{q}_m + \Phi^T D_n \Phi \dot{q}_m + \Phi^T K_n \Phi q_m = \Phi^T B_n u \quad (8)$$

$$y = C_{nq}\Phi\ q_m + C_{nv}\Phi \dot{q}_m \quad (9)$$

Through the similarity transformation the modal matrix has the effect of diagonalising the mass and stiffness matrices.

Denote the new n x n modal mass matrix, modal stiffness matrix and modal damping matrix, respectively, as

$$M = \Phi^T M_n \Phi \quad (10)$$

$$K = \Phi^{\mathrm{T}} K_n \Phi \quad (11)$$

$$D = \Phi^T D_n \Phi \quad (12)$$

Note that D may not always be a diagonal matrix, however, for convenience proportional damping, whereby $D = \propto_1 K + \propto_2 M$, $\propto_1, \propto_2 > 0$, is often employed. It is commonly argued that the damping within a structure is difficult to define accurately and at best is only roughly approximated.

Making use of the new notation and pre-multiplying throughout by M^{-1} leads to

$$\ddot{q}_m + M^{-1} D\ \dot{q}_m + M^{-1} K\ q_m = M^{-1}\Phi^T B_n u \quad (13)$$

$$y = C_{nq}\Phi\ q_m + C_{nv}\Phi \dot{q}_m \quad (14)$$

Introducing new diagonal matrices Ω and Z, where Ω is a diagonal matrix of natural frequencies and Z is a diagonal matrix of damping factors, i.e. $\Omega = diag(\omega_1, \omega_2 ... \omega_n)$ and $Z = diag(\xi_1, \xi_2 ... \xi_n)$, leads to the convenient notation

$$\ddot{q}_m + 2Z\Omega\ \dot{q}_m + \Omega^2\ q_m = B_m u \quad (15)$$

$$y = C_{mq} q_m + C_{mv} \dot{q}_m \quad (16)$$

where,

$$B_m = M^{-1}\Phi^T B_n, C_{mq} = C_{nq}\Phi, C_{mv} = C_{nv}\Phi$$

It is important to note that the above modal representation is a set of uncoupled equations. This greatly simplifies the analysis since each mode may be considered separately. The overall structural response is the sum of the modal responses. Consequently it is possible to express (15) and (16) equivalently as

$$\ddot{q}_{m_i} + 2\xi_i \omega_i \dot{q}_{m_i} + \omega_i^2 q_{m_i} = b_{m_i} u \quad (17)$$

$$y_i = c_{mq_i} q_{m_i} + c_{mv_i} \dot{q}_{m_i} \quad (18)$$
$$i = 1 ... n$$

$$y = \sum_{i=1}^{n} y_i$$

where n is the number of modes.

3.4 State-Space Modal Model

It is convenient to represent the structure in state-space form. The general form of a state-space representation is:

$$\dot{x} = Ax + Bu \quad (19)$$

$$y = Cx \quad (20)$$

where the form of the triple (A, B, C) will depend on the choice of the state variables x.

One particular intermediate choice for the state variables is

$$x = \begin{bmatrix} x_1 \\ \cdots \\ x_2 \end{bmatrix} = \begin{bmatrix} q_m \\ \cdots \\ \dot{q}_m \end{bmatrix} \quad (21)$$

leading to

$$\dot{x} = \begin{bmatrix} 0 & I \\ -\Omega^2 & -2Z\Omega \end{bmatrix} x + \begin{bmatrix} 0 \\ B_m \end{bmatrix} u \quad (22)$$

$$y = [C_{mq} \quad C_{mv}] x \quad (23)$$

which has basically converted the n second order differential equations of (17) and (18) to $2n$ first order differential equations.

It is now worthwhile considering a particularly appealing form of state-space representation whereby the state-space modal model makes use of the triple (A_m, B_m, C_m), whereby the state vector is redefined

$$x = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix} \quad (24)$$

where each component consists of two states

$$x_i = \begin{bmatrix} x_{i1} \\ x_{i2} \end{bmatrix} \text{ or } x_i = \begin{bmatrix} q_{m_i} \\ \dot{q}_{m_i} \end{bmatrix} \quad (25)$$

This leads to

$$A_m = \begin{bmatrix} x & x & & & & \\ x & x & 0 & & 0 & \\ & & x & x & & \\ 0 & & x & x & 0 & \\ & & & & x & x \\ 0 & & 0 & & x & x \end{bmatrix} \tag{26}$$

$$= diag\,(A_{m_i})$$

where the A_{m_i} are 2 x 2 blocks, $i = 1 \dots n$

$$B_m = \begin{bmatrix} B_{m_1} \\ B_{m_2} \\ \vdots \\ B_{m_n} \end{bmatrix} \text{ and} \tag{27}$$

$$C_m = \begin{bmatrix} C_{m_1} & C_{m_2} & \cdots & C_{m_n} \end{bmatrix} \tag{28}$$

the B_{m_i} are 2 x s blocks and C_{m_i} are r x 2 blocks.

Clearly the ith mode, $i = 1 \dots n$, has the state-space representation, i.e. the triple, $(A_{m_i}, B_{m_i}, C_{m_i})$ which are all independent, so that

$$\dot{x}_i = A_{m_i} x_i + B_{m_i} u \tag{29}$$

$$y_i = C_{m_i} x_i \tag{30}$$

$$y = \sum_{i=1}^{n} y_i \tag{31}$$

The advantage of the above state-space modal representation is that there are no couplings between modes, thus each state is independent. Equations (24) to (31) may be expressed in block diagram form as illustrated in Figure 2.

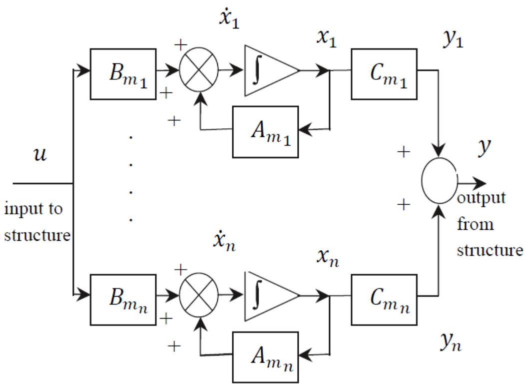

Figure 2: Block diagram of modal state-space representation indicating no coupling between modes.

4 ILLUSTRATIVE EXAMPLE

Consider the simple two-member structure in Figure 1 and described in Section 3.1.

Using matrices (3) and (4) it is clear that M_n and K_n are symmetric and that K_n is positive definite (with successive determinant minors being positive) and D_n represents proportional damping. In the nodal coordinate representation $q, \dot{q}$ and $\ddot{q}$ represent displacement, velocity and acceleration of two nodes. In this case $n_d = n$ i.e. the number of degrees of freedom in terms of nodes is equal to the number of modes, with each mode being a natural frequency of the combined structure.

Consideration is now given to equation (7), for the non-trivial case, i.e. $\phi \neq 0$. So that the eigenvalue problem $det(K_n - \omega_i^2 M_n) = 0$, i = 1, 2, is to be solved, where ω_i^2 denotes the square of the natural modes or frequencies in rad/s. Noting the form of M_n and letting the $\omega_i^2 = \lambda_i$ for i = 1, 2, the equation can be re-stated in the normal eigenvalue form

$$det(\lambda I - K_n) = 0 \tag{32}$$

yielding λ_1= 5.236 and λ_2= 0.764, which are real positive and distinct. These are necessarily positive due to the fact that $\lambda_i = \omega_i^2$. Hence the natural frequencies are: ω_1 = 2.288 rad/s and ω_2 = 0.874 rad/s, so that

$$\Omega = \begin{bmatrix} 2.288 & 0 \\ 0 & 0.874 \end{bmatrix}$$
$$\text{and} \quad \Omega^2 = \begin{bmatrix} 5.236 & 0 \\ 0 & 0.764 \end{bmatrix} \tag{33}$$

The eigenvectors in equation (7) corresponding to the $\omega_i^2 = \lambda_i$ are denoted ϕ_i and these are obtained from

$$(K_n - \lambda_i I)\phi_i = 0 \tag{34}$$

It can be deduced that for λ_1 and λ_2

$$\phi_{11} = -1.618\phi_{12}$$
$$\phi_{21} = 0.618\phi_{22} \tag{35}$$

The modal matrix Φ in standard form may thus be expressed as

$$\Phi = \begin{bmatrix} -1.0 & 0.618 \\ 0.618 & 1.0 \end{bmatrix} \tag{36}$$

(In standard form the largest element in each ϕ_i is normalised to unity.)

It should be noted that these eigenvectors are not unique and can be replaced by any arbitrary non-zero scalar multiples.

It is sometimes convenient to normalise the vectors such that the Euclidian norm is equal to unity, but this is not considered here. However, noting that Φ^T has the same diagonalising properties as Φ^{-1}, i.e. the inverse of the modal matrix, Φ^{-1} is used in the example in place of Φ^T, see Section 3.3.

Consider the modal state-space representation (24) to (28), where it can be shown that:

$$\dot{x} = \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix}$$

$$= \begin{bmatrix} 0 & 1 & & \\ -\omega_1^2 & -2\xi_1\omega_1 & & \\ & & 0 & 1 \\ & & -\omega_2^2 & -2\xi_2\omega_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ b_{m_1} \\ 0 \\ b_{m_2} \end{bmatrix} u \quad (37)$$

$$y = \begin{bmatrix} 1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} \quad (38)$$

q_{m_i}, $\dot{q}_{m_i}$ are the ith model displacement and velocity, respectively.

The poles of each modal property are the complex conjugate pairs, for $i = 1, 2$

$$-\xi_i\omega_i + j\omega_i\sqrt{1 - \xi_i^2} \quad (39)$$

$$-\xi_i\omega_i - j\omega_i\sqrt{1 - \xi_i^2} \quad (40)$$

Applying the similarity transformation $\Phi^{-1}\, D\, \Phi$ it may be deduced that the resulting damping in the modal representation is given by

$$Z = \begin{bmatrix} 0.05236 & 0 \\ 0 & 0.00764 \end{bmatrix} \quad (41)$$

Making use of Ω and Ω^2 from (33) and combining with Z from (41) and substituting into (37), it may be deduced that A_m is given by

$$A_m = \begin{bmatrix} 0 & 1 & & \\ -5.236 & -0.240 & & \\ & & 0 & 1 \\ & & -0.764 & -0.0134 \end{bmatrix} \quad (42)$$

and, using (16), the input vector B_m^T is given by

$$B_m^T = [0 \quad 0.447 \quad 0 \quad 0.724] \quad (43)$$

To illustrate the modal control approach, attention is given to increasing both the damping and the natural frequency of the triple $(A_{m_2}, B_{m_2}, C_{m_2})$ such that ω_2 and ξ_2 are given by 1.748 rad/s and 0.0153, respectively, hence increasing their values by a factor of 2.

Figure 3 shows the uncontrolled response of the flexible structure and the effect of modal control applied to the triple $(A_{m_2}, B_{m_2}, C_{m_2})$, on the overall performance. The controller satisfactorily achieves the required damping and natural frequency for the second mode without changing that of the first.

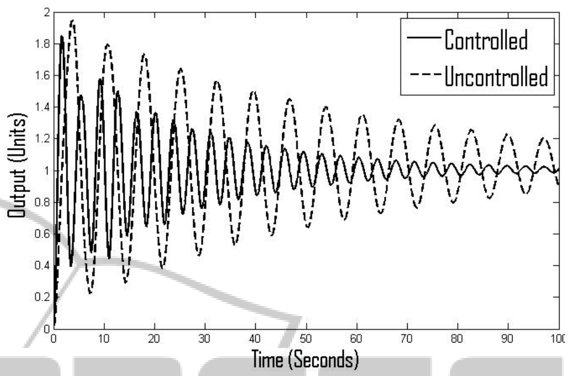

Figure 3: Illustrating the application of modal control to the example of Figure 1.

The above example has served to illustrate that it is possible via modal control to independently change a mode of a given structure without affecting the other modes.

5 CONCLUSIONS

The paper has provided a premise statement and has made assumptions on the factors influencing future design of the passenger vehicle fleet. Based on these assumptions, new generations of vehicles will be lighter, able to achieve CO_2 emission reductions and also, because such vehicles will be equipped with advanced driver assisted systems and active safety devices, will become more efficient, safer and environmentally friendly.

The presence of the lighter vehicles in the fleet prompts the need for optimum energy absorption between vehicles of dissimilar masses. Smart materials provide a means of achieving these desirable mechanical/structural properties given a particular collision scenario. As such larger vehicles will be required to soften upon impact and give way to smaller vehicles which, contrary to current practice, will be allowed to stiffen. Consequently, collision energy mitigation control strategies need to respond rapidly in advance, making use of predictive and adaptive procedures, with full exploitation of vehicle to vehicle communication and onboard safety systems. The need for the current levels of crashworthiness and the accompanying crumple zones in future vehicles is therefore challenged. In fact it is argued that reduced aggressivity among colliding vehicles should replace crashworthiness as a key future safety design criteria.

The paper has provided a convincing case for developing future lighter vehicles with advanced safety features, capable of mitigating the effects of collisions via active control of onboard smart materials to achieve maximum energy absorption. The future vehicles described above will also meet the increased demands regarding CO_2 legislation, which must be achieved to develop an economic, efficient, safe and sustainable transport system for future generations.

The resulting design represents a radical paradigm shift from current automotive industry practice. Whilst this may be met with scepticism from some quarters, it will be embraced and seen as a sustainable approach for achieving CO_2 reduction, lightweight structures, cooperative vehicles and enhanced active safety systems. All of which represent key goals along a road map towards achieving improved/sustainable future vehicle engineering systems.

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