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ACTIVE ENGINE MOUNTS FOR IMPROVING VEHICLE COMFORT

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ABSTRACT

This paper relates to improving the comfort in vehicle cabins. As the mass of the body and structure of cars is reduced in order to improve fuel economy, the combined mass of the engine and gearbox becomes a proportionately larger part of the total mass of the car. The dynamic behaviour of the car body is thus altered substantially by the degree to which these two masses are coupled. Increasing the effective mount stiffness increases the coupling of body and engine, and generally this is advantageous. The space required for the engine and gearbox in the engine compartment can be reduced if the engine is not expected to move relative to the car body, and this can be achieved by stiff mountings.

The engine produces vibrations which are transmitted to the car structure through the mountings, and these vibrations disturb the comfort of the passengers. It is therefore desirable to reduce this disturbance by isolating the engine from the car body. This isolation is normally achieved by mounting the engine on rubber blocks which are resilient and allow the engine to move in such a way that its inertia balances the forces on the engine. The transmission of vibrations to the car structure is thus reduced.

These two objectives are in conflict with each other: on the one hand, the mountings should be stiff (ideally rigid) in order to tie the engine to the car body and thereby reduce the relative movement of the engine; on the other hand, the mountings should be compliant and allow the engine to move freely in order to provide good vibration isolation.

INTRODUCTION

Various papers have reported work on systems which attempt to overcome this conflict of objectives. One of the most promising ideas is that of active stiffness control where an active control system would modify the stiffness properties of the mounting. This technique has been described in the literature for use in car suspension systems [1, 2]. This type of mounting would be configured to be stiff during the times when large relative displacements of the car engine were likely, and to be compliant otherwise. The result would then be a system which would isolate the vibrations reasonably well except for short periods of time. The poor performance during these short periods of time is an obvious disadvantage of this technique. In addition, the relative movement of the engine and seating due to the mean torque of the engine would not be stopped.

An alternative actively controlled mounting would change its effective length under the direction of a control system. This length would be adjusted to maintain the position of the engine due to the changes in mean torque. However, such a system would be unable to cope with the rapid transient movements of the car body. A third mounting arrangement has been reported [3] in which a detector senses the vibration of the engine and uses this signal to control the length of an active mounting. This mounting responds at the frequency of the engine-induced vibration and is thus much faster in response than the other approaches. However, in an automobile, vibration may be transmitted in all three axes, and thus the engine mount would need to change its length in all of these directions. Additionally, as the engine moves it produces unsteady rotations which are also transmitted to the body through the mounting, and these must also be suppressed. Consequently the application of active mountings has been limited to system where the motion is predominantly in one direction only, and where the mounting incorporates a means of shear decoupling in the other orthogonal directions to reduce the effect of this shortcoming. However, this technique still does not remove the effect of unsteady engine rotations.

THE GENERAL PRINCIPLES OF THE SOLUTION

The mount interface between the vibrating source and the seating as a whole must be designed to be stiff in response to certain vibrations whilst compliant in response to others. If we take into account the fact that, relative to the seating, the effectively rigid body comprising the vibration source may have a total of six degrees of freedom then six or fewer individual mounts, each with control in a single direction, are sufficient to make the interface stiff. However, in order for the interface to be compliant at some frequencies, each individual mount must have the potential to be made compliant in its own control direction by the control system and must also be completely compliant to displacements in the two orthogonal directions and to rotations. This compliance may be achieved by passive means of shear and rotation decoupling. The active mounts of this type are then positioned in optimum locations which are chosen to minimise the complexity of the mounting system while at the same time applying the necessary control forces in the six, or fewer, requisite directions. Fewer than six mounts would be required if the expected distribution of mount gap displacements could be shown to have less than six degrees of freedom. This would only be known from detailed measurements of the dynamics of the combined system.

A basic system for controlling the actuators of a mounting system with more than one mount is shown in Figure 1 and comprises: the sensors (1); a processor (2) receiving input signals from the sensors and providing two output signals, s_1 characterising the vibrations v_1 to be isolated, and signals s_2 characterising the vibrations and static forces v_2 to which the interface is required to be stiff; a controller (3) which has inputs s_1 and which determines the signals w_1 to be fed to the actuators in order to make the interface compliant to the vibrations v_1; a second controller (4) which has inputs s_2 and which determines the signals w_2 to be fed to the actuators in order to make the interface stiff to the vibrations and static forces v_2; a system (5) to combine the signals w_1 and w_2 output by the controllers to produce the drive signals for the actuators (6). In practice, the first and second control systems are likely to be combined in one system. They are presented separately here for clarity.

If vibration isolation is required outside the frequency bandwidth of the control system, it may be obtained by passive decoupling of the mount in its controlled direction at some or all frequencies outside the controller bandwidth. Moreover, if the vibrations v_1 and v_2 occupy non-overlapping bands of the frequency spectrum, then one of the controllers could be omitted and the desired behaviour of the interface achieved by a single controller and passive decoupling. This relies on the fact that passive systems have an isolation performance that can be designed to vary with frequency, generally providing increased isolation with increasing frequency.

THE CAR MOUNTING SYSTEM

We will now describe the specific implementation of a system designed for an automobile.

The specific objectives of the mounting interface in this case are twofold:

1. to isolate the dynamic vibration of the car engine caused by the cyclic fluctuations in torque, out of balance forces and other rotation or reciprocation effects over a set of engine orders whose frequencies vary with the frequency of the machine;

2. to appear stiff in response to other vibrations and static forces such as vibration or displacement of the seating and non-cyclic fluctuations in torque.

These specific objectives are achieved by designing the active mount control system in the following way. By means of accelerometers positioned on the seating in the line of action of the mount and a tachometer on the engine, the processor determines the periodic components of the movement of the seating at integer multiples of the firing cycle frequency of the engine and feeds them to the first controller to produce a vibration isolating signal. Control systems which achieve this have been described in the literature [3]. The processor also subtracts the periodic components from the total vibration signal obtained from the sensors on the seating, and combines the resulting residual signal with that obtained from motion sensors mounted on the machine. This combined signal forms the drive signal for the second control system which takes the form of a feedback controller. The output from this feedback controller is then added to the vibration isolating signal produced by the first controller and fed to the actuator at each mount. The feedback controller is designed to work only for frequencies in a given range; passive elements are therefore incorporated into the mount in series and in parallel with the actuator, thereby making the mount passively compliant above this frequency band and supportive of the static load of the machine. Moreover, the mount is compliant above this frequency band and supportive of the static load of the machine. The mount is also compliant at multiples of the frequency of the firing cycle for which the vibrations are to be isolated, because the feedback controller input has no components at these frequencies.

We now need to look in more detail at the non-cyclic inputs to the system and the feedback controller. The engine of the vehicle is mounted on the body structure and is required to remain fixed relative to the body while the vehicle is subject to a variety of disturbances such as acceleration, braking, cornering and road effects.

The dynamics of the non-periodic displacements from equilibria of the gaps between the engine and the seating structure can be described in a state space formulation as

$$\frac{d\underline{x}}{dt} = A\underline{x} + B\underline{u} + \underline{e}$$

where $\underline{x}$ denotes the state vector of the variations in the gaps, matrix A defines the unforced dynamics of the gaps, B relates the control forces to the gap dynamics, $\underline{u}$ is the vector of feedback components of the control forces and $\underline{e}$ represents the disturbance terms due to effects uncorrelated with the engine frequency, eg mean engine torques, acceleration and road effects. A block diagram of the second control system is shown in Figure 4. In order to make the interface stiff over a frequency band up to a specific cut off frequency, the system dynamics need only be controlled over the frequency band of interest. This may be achieved by low pass filtering the state $\underline{x}$ to produce $\underline{y}$, a process which may be described in the state space formulation by

$$\frac{d\underline{y}}{dt} = C\underline{y} + H\underline{x}$$

The disturbance term $\underline{e}$ may similarly be described as the output of a filter driven by a white noise input w and having state $\underline{\Omega}$ and modelled by the equations

$$\frac{d\underline{\Omega}}{dt} = F\underline{\Omega} + \underline{b}w;$$

$$\underline{e} = E\underline{\Omega}$$

Defining $\underline{v}$ to be the vector formed by concatenating the three state vectors $\underline{x}$, $\underline{y}$ and $\underline{\Omega}$ in order, the entire dynamics of the system may be described by the single state space equation

$$\frac{d\underline{v}}{dt} = M\underline{v} + N\underline{u} + \underline{d}w$$

where

$$M = \begin{matrix} A & 0 & E \\ H & C & 0 \\ 0 & 0 & F \end{matrix} \qquad N = \begin{matrix} B \\ 0 \\ 0 \end{matrix} \qquad \underline{d} = \begin{matrix} 0 \\ 0 \\ \underline{b} \end{matrix}$$

The aim of the control signal $\underline{u}$ is to maintain the vector $\underline{y}$ at zero or to restore it to zero in some optimal manner if it should deviate. A suitable measure of the quality of the control is

$$J(\underline{u}) = \int_0^\infty (\underline{z}'(t)\underline{z}(t) + \underline{u}'(t)\Lambda\underline{u}(t))dt$$

where

$$\underline{z} = G\underline{y} = \underline{y}, \qquad G = [0\ I\ 0],$$

Λ is a diagonal matrix and a prime denotes a vector transpose. This cost (or penalty) functional serves to penalise large controls and deviations of the filtered gap states $\underline{y}$ from zero. This problem is in a standard form for solution by the method described in [4] which leads to a linear feedback control in the form

$$\underline{u} = -K\underline{v}$$

Implementation of the control requires a knowledge of the state vector $\underline{v}$, which may be reconstructed from the available measurements of the motions of the vehicle and engine, possibly by means of a state observer, as also proposed in [4].

An arrangement of mounts suitable for use in the car application is shown in Figure 2, wherein the arrows indicate the direction of control of each mount. An example of a mount which could be used to implement the mounting system is shown in Figure 3, where an engine (7) is mounted on a seating (8). The mount is attached to the seating by means of a ball and socket joint (9), which acts to decouple rotations, and to the machine by a similar device (10). Blocks of anisotropic rubber (11 and 12) are attached to the opposite sides of the ball and socket joints to provide the required shear decoupling in the directions orthogonal to the controlled direction. The rubber blocks (11 and 12) are separated by a coil which contains the actuator (14), which is controlled by the control line (15) connected to the control system described above. The actuator may be of a hydraulic, electromagnetic, piezoelectric or other type.

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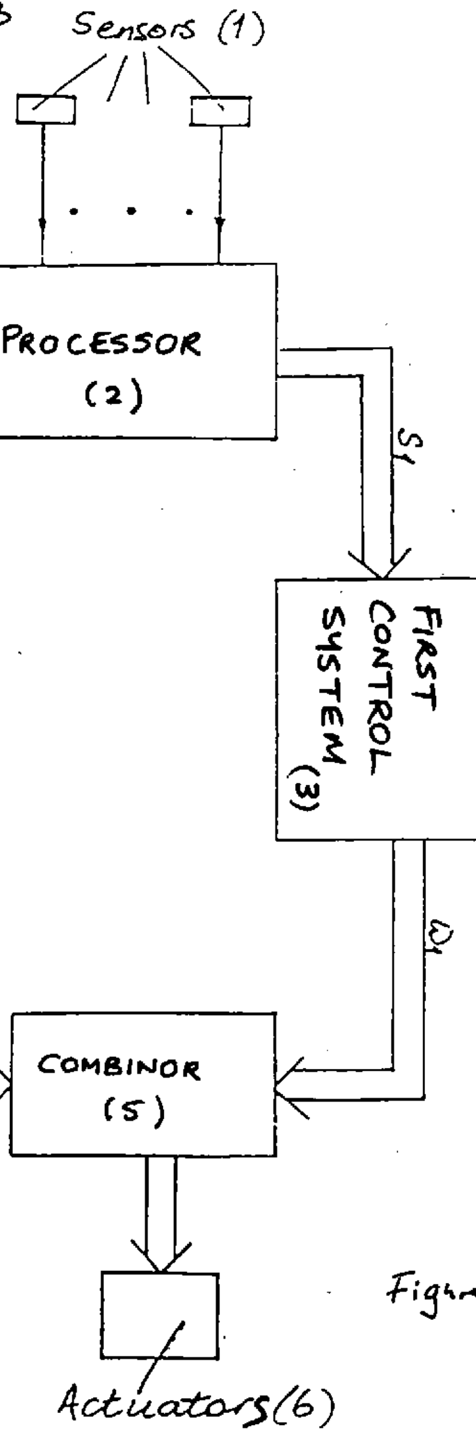

Figure 1.

Figure 1

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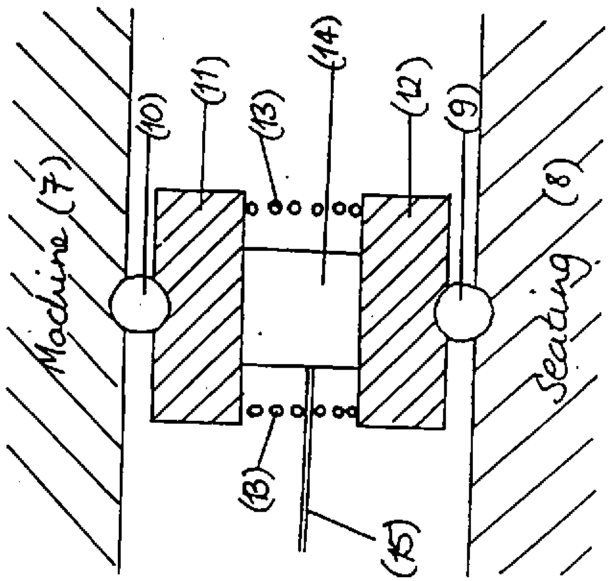

Figure 3

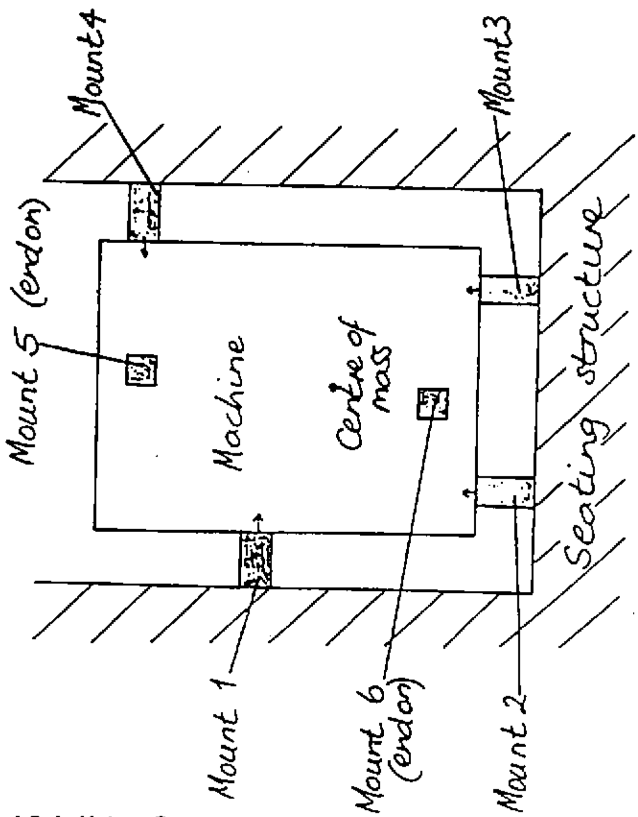

Figure 2.

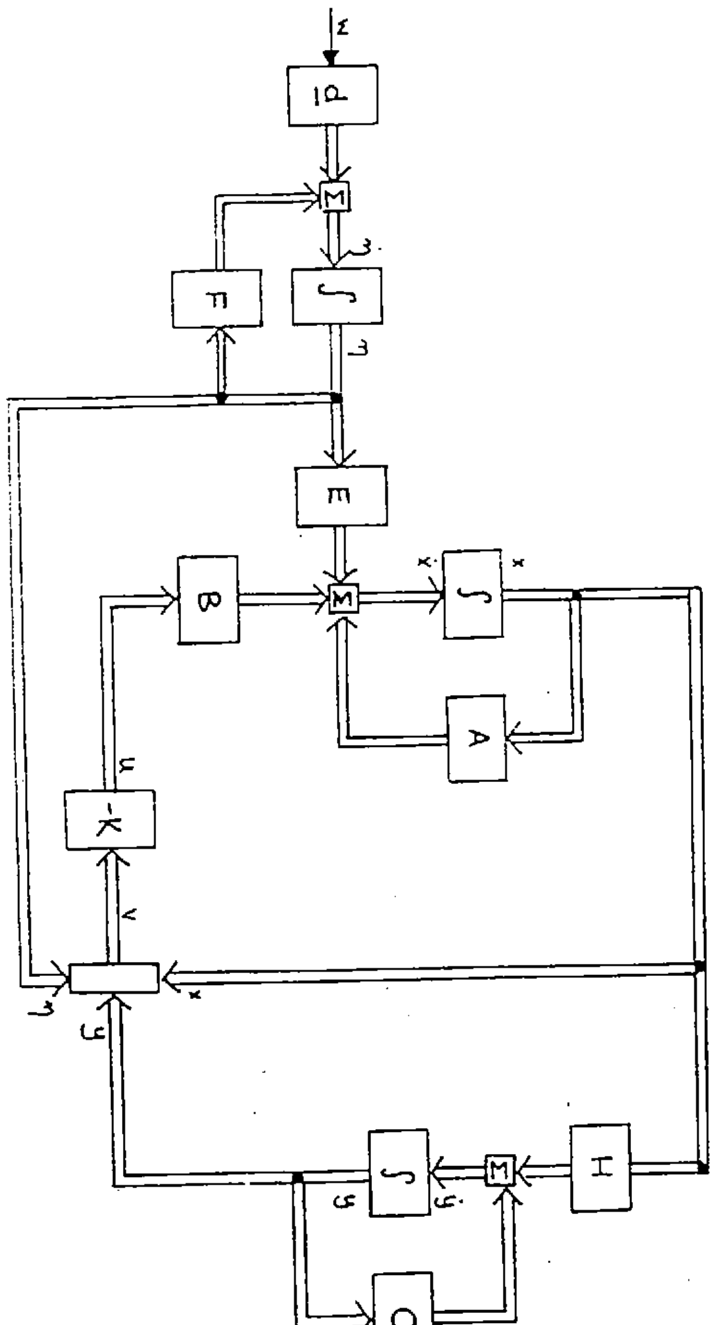

Figure 4

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