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ACTIVE VIBRATION ISOLATION OF DIESEL
ENGINES IN SHIPS

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Abstract

This study was concerned with an investigation of the feasibility of using active vibration isolation to reduce low order harmonic vibration transmission through engine mounts to the hulls of ships. The control objective was to minimise the motion of the intermediate mass of a two-stage passive isolation system for all six degrees of freedom. Two control strategies were investigated; kinetic energy control and modal control. For kinetic energy control, the cost function used was the sum of the squared error signals at the error sensors, which is proportional to the kinetic energy of the mass. For modal control, the error sensor signals were combined to provide modal information to each controller channel so that the controller could specifically target each of the six degrees of freedom of the intermediate mass. Both numerical and experimental results demonstrated that the total vibratory energy of the intermediate mass could be reduced for low order harmonics of the engine operating speed.

INTRODUCTION

The traditional approach for isolating the transmission of vibratory energy from a vibration source, such as a reciprocating engine, to a flexible support structure is to place passive isolators between the vibration source and the receiving structure. The selected vibration isolator must support the static load of the machine and must also
have a sufficiently low stiffness so that the translational and rotational resonance frequencies of the machine, mounted on the isolators, are considerably less than the frequencies of the dominant disturbances generated by the machine. A passive isolator capable of adequate isolation at low frequencies can sometimes result in insufficient support stability for the equipment. One promising way around these problems is to use active vibration isolation with passive isolators to reduce the vibratory energy transmission. Ahn et. al. [1] proposed a hybrid-type active vibration isolation system using an electromagnetic actuator and air-springs. In their work, the electromagnetic actuator was used as an active element and the air spring acted as the passive element. It was demonstrated that this hybrid control system could provide a better isolation performance than the passive system alone. Gardonio et. al. [2-5] studied the theoretical effectiveness of various control strategies for active vibration isolation. In their work [3], the minimisation of the total power transmission through the mounts to the receiver was compared with several control strategies: the minimisation of the axial velocities or forces; the minimisation of the axial power transmission and the minimisation of the sum of the squared axial velocities and weighted squared forces. It was concluded that the cancellation of the total power transmitted from the source to the receiver through the mounts was the optimal control strategy for the cases they studied.

In this paper, the feasibility of using active vibration isolation to minimise the motion of an intermediate mass of a two stage passive ship engine mount is investigated in the laboratory. The work involved the use of a feedforward controller to cancel the total vibratory energy of the intermediate mass of a test rig simulating one of diesel engine mounts in a ship. The motion of the intermediate mass (a total of six degrees of freedom) was sensed using accelerometers to provide the error signals and several inertial shakers were mounted to the mass to act as control actuators. Both numerical simulation results and real time control results are provided.

**TEST RIG MODEL**

Figure 1 shows the setup of the test rig, which represents a two-stage hybrid vibration isolation system. In the figure, for the passive system, the top plate is connected to the intermediate mass through 4 rubber isolators and another 6 isolators are mounted between the intermediate mass and the bottom plate. The intermediate mass was from

![Figure 1: Two stage hybrid vibration isolation system](image-url)
an actual ship isolator. Sufficient rubber blocks were used on each side of the intermediate mass to provide a similar static deflection to that achieved by installed isolator. The number of blocks used in the tests was considerably less than the number used on the ship, as the mass of the top plate was much less than the mass of engine supported in the actual installation. The specifications of the system are detailed in previous work [6].

For the active control system, seven error sensors (E₁–E₇) and seven pairs of control actuators (A₁–A₇ and AA₁–AA₇), which allowed control at a total of six degrees of freedom of the rigid mass were located on the intermediate mass as mapped in Figure 2. Two actuators of each pair (Aᵢ and AAᵢ) were wired in series and connected to the controller through one channel. In Figure 2, M₁–M₇ indicate the monitor sensor locations which were used to evaluate the control performance for numerical simulation and R, P and T represent pitch, rotation and torsional motion of the mass about the X, Y and Z axes respectively.

![Figure 2: Error sensor, monitor sensor and control actuator locations](image)

To increase the mechanical output of the control shakers within the frequency range of interest, two sets of control shakers Aᵢ and AAᵢ were tuned to resonate around the fundamental and 2.5th order of the engine operating speed respectively [7]. A ‘simulated’ primary disturbance was generated by feeding vibration signals that were recorded on a ship into the primary inertial actuators attached to the top mass.

**CONTROL STRATEGIES**

The quadratic optimisation technique, which has been documented thoroughly by Nelson and Elliott [8] and Hansen and Snyder [9], has been widely used in active noise and vibration control simulations and is summarised in Table 1 for the case study. In the table, for kinetic energy control, $Z_e$ is the matrix of transfer functions from the control sources to the error sensors; $Q_e$ is the vector of the control forces and $V_p$ is the vector of the primary disturbance signals at the error sensors. For modal control, $Z_m$ is the matrix of modal transfer functions from the modal control actuators to the modal error sensors; and $V_m$ is the vector of the primary disturbance for each mode.
Table 1 Control strategies

<table>
<thead>
<tr>
<th>Control Strategies</th>
<th>Kinetic Energy Control</th>
<th>Modal Control</th>
</tr>
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<tbody>
<tr>
<td>Error Criterion</td>
<td>$J = Q_e^H A Q_e + Q_e^H b + b^H Q_e + c$</td>
<td></td>
</tr>
<tr>
<td>Optimum Control Forces</td>
<td>$Q_{c, opt} = -A^{-1} b$</td>
<td></td>
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The minimum cost function at the error sensing locations can be obtained by substituting the expression for optimum control forces into the expression for the error criterion in Table 1.

NUMERICAL SIMULATION

Initially, off-line control was done to evaluate only the physical system configuration without any limiting effects due to the control system, so that the maximum achievable control results could be achieved. This involved taking measurements of the transfer functions from the control sources to the error sensors and measurements of the primary disturbance at the error sensors. To evaluate the control performance at a set of monitor sensors, the transfer functions from the control actuators to the monitor sensors were measured as well. The optimum control inputs, corresponding to the minimum cost function at the error sensors, were calculated and the vibratory energy reduction at the monitor sensors was then calculated using these optimum control inputs.

Table 2 shows the numerical results using both control strategies. Both sets of control actuators located on the top and bottom of the intermediate mass were used.

Table 2 Predicted overall vibratory energy level reduction at the monitor sensors

<table>
<thead>
<tr>
<th>Harmonic order of engine speed</th>
<th>fundamental</th>
<th>1.5&lt;sup&gt;th&lt;/sup&gt;</th>
<th>2&lt;sup&gt;nd&lt;/sup&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall reduction at monitor sensors (dB)</td>
<td>Kinetic energy control</td>
<td>40.9</td>
<td>28.6</td>
</tr>
<tr>
<td></td>
<td>Modal control</td>
<td>40.2</td>
<td>34.3</td>
</tr>
</tbody>
</table>

From Table 2, one can see that the vibratory energy reductions at the fundamental frequency and the frequency corresponding to the 1.5<sup>th</sup> order are 40.9 dB and 28.6 dB, respectively, for kinetic energy control and 40.2 dB and 34.3 dB, respectively, for modal control. As the supported mass may be assumed rigid in the frequency range of interest, the seven monitor sensors ($M_1$ – $M_7$), which are mapped in Figure 2, can measure all degrees of freedom of the mass. Therefore the vibratory energy level reduction at the monitor sensors is representative of the total vibratory energy level reduction of the mass.
REAL TIME CONTROL

To verify the predicted results, real time control was carried out with an EZ-ANC II multi-channel controller. A ‘simulated’ primary disturbance was generated by feeding the signals that were recorded on a ship into the primary inertial actuators attached to the top mass. Both kinetic energy and modal control strategies were evaluated.

Experimental Set-up

For kinetic energy control, seven control actuators and seven error sensors were used. The signals from the error sensors were input into the EZ-ANC II controller and also into a Brüel & Kjær PULSE system (multi-data acquisition system). The average reduction of the sum of the squared signals at the error sensors was measured using the PULSE system. Monitor sensors were not used for this part of the work as the intermediate mass was sufficiently rigid that the measured overall vibration level reductions at the error sensors were representative of the total vibration reduction of the rigid intermediate mass. Control adaptation ceased once the overall vibration level reductions at the error sensors reached a maximum. The power spectra at the error sensors were then recorded. For modal control, electronic summers between the controller and control actuators were used to construct the modal actuators. Figure 3 shows a block diagram of the experimental set-up for real time control. In the figure the summers are used only for modal control.

Control Results

For kinetic energy control, the cost function to be minimised was the sum of the squared error signals at the 7 error sensor locations. This quantity is proportional to the kinetic energy of the intermediate mass; thus minimisation of the cost function implies minimisation of the vibratory energy. Figure 4 shows acceleration levels before and after kinetic energy control. Only the set of control actuators with a resonance frequency around the fundamental of the engine speed were used. From the figure it can be seen that the large acceleration level reductions of 30.1 dB and 30.4 dB were achieved at frequencies corresponding to the fundamental and 1.5th order respectively, but poor control performance was obtained at the 2nd order frequency.
The reason for this phenomenon is that the control actuators resonated at a frequency corresponding approximately to the engine running speed, so that a large portion of the energy used to model the cancellation path was distributed around the fundamental and 1.5th orders, shown as the dotted line in Figure 5. On the other hand, poor cancellation path modelling as a result of the low S/N ratio at the second order frequency resulted in poor control performance at this frequency. To improve control performance at the second order frequency, an additional set of seven actuators, all of which were tuned to resonate around the 2.5th order engine operating speed, were introduced into the system. The configuration of the control actuators for this case was described previously. The response of the intermediate mass with both sets of actuators present is shown as the solid line in Figure 5.

Figure 6 shows the control results achieved with the additional seven control actuators. By comparing the results shown in Figure 6 with those in Figure 4, one can see that the vibration level reduction at the 2nd order frequency is slightly increased but a further 18 dB reduction is achieved at the 2.5th order. This is partly because the

\[
\Delta_1 = 30.1 \text{ dB}; \quad \Delta_{1.5} = 30.4 \text{ dB}; \quad \Delta_2 = 3.1 \text{ dB}; \quad \Delta_{2.5} = 1.6 \text{ dB}; \quad \Delta_3 = 6.8 \text{ dB}
\]

Figure 4: Overall acceleration levels before and after kinetic control; the resonance frequency of the control actuators was around fundamental of engine speed.

Figure 5: Response of the intermediate mass during cancellation path modeling.
additional set of control actuators resonated at a frequency that was closer to the 2.5th order frequency than the 2nd order. Another reason for the continued poor performance at the second order frequency is the presence of electronic noise in the system around the 2nd order frequency due to an earth loop problem. Thus, there is no doubt that good control performance can be achieved at the 2nd order frequency if properly tuned actuators are used and the ground loop problem is eliminated by proper insulation of the accelerometers.

\[ \Delta_1 = 31.1 \text{ dB}; \Delta_{1.5} = 17.6 \text{ dB}; \Delta_2 = 5.2 \text{ dB}; \Delta_{2.5} = 19.9 \text{ dB}; \Delta_3 = 6.5 \text{ dB} \]

Figure 6: Overall acceleration levels before and after kinetic control; the resonance frequencies of control actuators were around 1st and 2.5th order engine speed.

For modal control, electronic summers were used to combine the controller outputs so that the modal control actuators, which controlled each of six degrees of freedom, could be constructed. The overall vibration levels before and after modal control are shown in Figure 7 when both sets of actuators were used.

\[ \Delta_1 = 29.3 \text{ dB}; \Delta_{1.5} = 14.3 \text{ dB}; \Delta_2 = 9.7 \text{ dB}; \Delta_{2.5} = 15.4 \text{ dB}; \Delta_3 = 1.6 \text{ dB} \]

Figure 7: Overall acceleration levels before and after modal control; both sets of control actuators located on the top and bottom of the intermediate mass.

As expected, overall vibration levels were reduced at the required frequencies and especially good results were achieved at the frequencies that were close to the
resonance frequencies of the control actuators. These results are similar to those achieved using kinetic energy control, as shown in Figure 6.

CONCLUSIONS

The feasibility of using active control to minimise the vibration levels of an intermediate mass of an existing two-stage passive isolation mount used in a ship has been investigated. The control objective was to minimise the overall vibration levels of the intermediate mass for all six degrees of freedom. To achieve control, seven error sensors and seven control actuators were used. Two control strategies were evaluated; namely, kinetic energy control and modal control. Numerical simulations were conducted using the measured disturbance at the error sensors and the measured transfer functions from the control actuators to the error sensors. To verify the predicted results, real time control was carried out using an EZ-ANC II ten-channel controller. The real time results demonstrated that overall vibration levels of the intermediate mass could be reduced by 31.1 dB, 17.6 dB and 5.2 dB for kinetic energy control, and the overall vibration levels of the mass could be reduced by 29.3 dB, 14.3 dB and 9.7 dB for modal control at frequencies corresponding to the fundamental, 1.5th and 2nd orders respectively.

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REFERENCES