Active Isolation of a Vibrating Mass

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ABSTRACT: Active vibration isolation can offer improved performance at low frequencies compared with passive isolation. Active control methods require the selection of a suitable cost function to be minimised. Experimental results are presented of the active vibration isolation of a simulated rotating machine mounted to a beam which uses two types of cost function, the force and acceleration respectively at the base of the isolator. The results show that minimisation of one type of cost function does not necessarily lead to the minimisation of the other cost function.

1. INTRODUCTION

Conventional passive vibration isolation of machinery from supporting structures, typically utilises springs, air mounts, rubber blocks or sheets of neoprene. An isolation system is usually selected based on the operating frequency of the machine. For example, an electric motor rotating at 1440 rpm has an operating frequency of 24 Hz. The machine mounted on the vibration isolators must have a resonance frequency less than the driving frequency for vibration isolation to occur. More specifically, the driving frequency of the machine should be at least $\sqrt{2}$ times the resonance frequency of the system. However, by selecting a vibration isolator of low stiffness, so that greater isolation occurs, the deflection of the mount will be large and this may result in unacceptable instability of the system. Consequently a compromise has to be made to balance the vibration reduction with the deflection.

By using active vibration control combined with passive isolation, deflections can be reduced compared with those required for passive vibration isolation and vibration levels can be reduced compared to those achieved only with a passive isolation system. Active vibration control utilises a control shaker (magnetic, piezo-electric or hydraulic) which provides a counteracting force to reduce vibration levels as well as one or more sensors to evaluate the performance of the control and to provide error signals to the controller.

As an illustration of the potential of active vibration isolation, an 8 kg mass supported by an active vibration isolator was excited using a shaker. The isolator was attached to a simply supported beam, 1.5 m long. Figure 1 shows the experimental set up.

This arrangement can be thought of as a vibrating machine mounted on an isolator. The top mass represents the machine to be isolated and the upper shaker provides a simulated out of balance which disturbs the system. The spring and control shaker are the passive and active isolation systems while the lower mass and beam represent the foundation, including its flexibility and mass. The intention of the active vibration isolator in this case was to reduce the vertical vibration transmitted into the supporting beam.

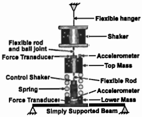

Figure 1. Beam experiment set up.

2. INSTRUMENTATION SETUP

An accelerometer and force transducer were placed on the top of the mass and between the isolator and the beam to measure the vibration transmitted into and out of the vibration isolator. Charge amplifiers were used to convert the piezo-electric transducer signals into voltage signals, which were recorded by the digital signal analyser.

In the example discussed here, feedforward active vibration control was used which employs a digital controller generating an appropriate control signal based on error and reference signals. The controller was basically an adaptive electronic filter, which had the objective of minimising the error signal. A Causal Systems EZ-ANC digital controller was used to provide a control signal which was amplified to drive the control shaker inside the active vibration isolator. The reference signal was provided by an electronic oscillator on the digital signal analyser. In practice, a suitable reference signal could be obtained from a tachometer attached to the rotating shaft of the vibrating machine, which can be used with additional electronics to generate a reference signal containing the shaft rotational frequency and its harmonics. The same reference signal was also used by the power amplifiers connected to the primary shaker to provide a disturbing force, which simulates the harmonic vibration from a rotating machine. The vibration isolation performance was compared for two different error signals. The first was the acceleration at the base of the isolator, and the second was the force between the lower mass and the beam. Figure 2 shows the instrumentation set up.

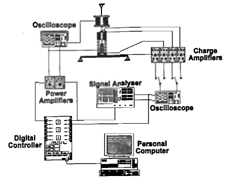

Figure 2. Instrumentation set up.

3. EXPERIMENTAL RESULTS

The driving shaker excited the mass at a single frequency and the amplitude was adjusted so that under passive vibration isolation (without the control shaker operating) the force into the top mass was 1 N.

Figure 3 shows the acceleration levels of the beam when the accelerometer in the base of the isolator was used as the error sensor.

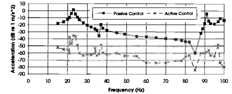

Figure 3. Acceleration level using passive and active vibration control - acceleration error sensor.

It took approximately 30 seconds for the system to stabilise and achieve the maximum attenuation using active control. When the measurements were taken, the adaptation was turned off.

Figure 4 shows the force levels exerted by the base of the isolator on the beam when the accelerometer in the base of the isolator was used as the error sensor.

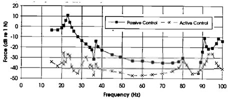

Figure 4. Force level using passive and active vibration control - acceleration error sensor.

Figure 5 shows the acceleration levels of the beam when the force transducer in the base of the isolator was used as the error sensor.

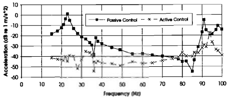

Figure 5. Acceleration level using passive and active vibration control - force error sensor.

Figure 6 shows the force levels of the beam when the force transducer in the base of the isolator was used as the error sensor.

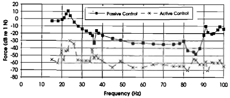

Figure 6. Force level using passive and active vibration control - force error sensor

These results show that the vibration isolation performance is dependent on the chosen error signal. When the acceleration at the base of the isolator was chosen as the error signal, the acceleration at the base of the isolator was reduced by between 20-65 dB. However, the force at the base of the isolator was reduced by between 0 and 35 dB. When the force at the base of the isolator was chosen as the error signal, the force was reduced between 5-55 dB. However, the acceleration signal was reduced by at most 40 dB and at some frequencies was actually worse than passive isolation. In each case, the controller tried to minimise the chosen error signal which is shown in Figures 3 and 6. These two figures show there is always some improvement in vibration isolation for the controlled parameter. What the results demonstrate is that controlling one parameter (say force) does not guarantee that the other parameter will also be reduced (say acceleration).

To improve the vibration isolation with active control, an error signal must be used which combines both acceleration and force. There are two methods which can use both acceleration and force signals. The first is to use two error signals (force and acceleration) and minimise both error signals. Intuitively this seems a reasonable solution and it is possible to improve the results by placing greater weighting on the error signal which would provide the greatest vibration isolation. This would give improved performance at one frequency, but would not necessarily give improved performance for the operating frequency range. Clearly this method is not suitable for general active vibration isolation applications as some manual adjustment is required to ensure that the best performance is obtained. The second method is to combine the force and acceleration signals into a vibration intensity or power transmission signal. This method has been used by Pan et al (1993) but the experimental work required manual adjustment of the amplitude and phase of the control signal. Recently an algorithm was developed by Kang and Kim (1997) which was used with an adaptive controller for controlling the acoustic intensity in a duct. This method could be applied to the reduction of vibrational power transmission in an active isolation system.

The rigid mass in the experimental setup described here was excited with harmonic vibration. This has computational advantages for a feedforward controller in that the repetitive nature of the signals means there is no need to calculate the required control signal faster than the time taken for the vibration signal to travel from the vibration source to the control source location. In other words, the system does not need to be causal. If the experimental setup were to be excited with random vibration, then unless the system were causal, the controller would never be able to generate an appropriate control force to prevent the disturbing vibration from reaching the support structure. Instead, the task of the controller would change to the minimisation of the modal response of the support structure once the vibration had disturbed the support structure. Alternatively if the system were causal, then feedforward control is likely to be successful over a frequency range of about two octaves. The actual frequency bands for which the controller would be effective would depend on the sampling rate used for the reference and error signals. For further information on active vibration isolation, see Hansen and Snyder (1997) and Fuller et al (1996).

The work described here has indicated that it is possible to reduce the vibration transmission from a harmonically vibrating machine by using active vibration control techniques. The results indicate that the degree of vibration isolation depends on the cost function chosen to be minimised. The acceleration and force cost functions give different results because their minima do not coincide, as a result of structural damping. Current work is underway to continue the investigation using vibrational power transmission as a cost function and to investigate the effect of moment excitation on the vibration isolation performance.

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