ACTIVE ENGINE MOUNT TECHNOLOGY FOR AUTOMOBILES

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ABSTRACT

We present a narrow-band tracking control using a variant of the Least Mean Square (LMS) algorithm [1,2,3] for suppressing automobile engine/drive-train vibration disturbances. The algorithm presented here has a simple structure and may be implemented in a low cost micro controller. We also present the results of a laboratory experiment to validate the algorithm. The experimental results show over 20 dB attenuation of the first nine harmonics and subharmonics of the engine/drive-train disturbance.

1. INTRODUCTION

Noise and vibration in an automobile is generated by many sources, such as engine/drive-train, wind and aerodynamic effects, tire-road contact, exhaust system, etc. The presence of such noise and vibration makes automobile ride less comfortable for the riders. Discomfort level of drivers and occupants increases with longer exposure time. This paper addresses the issue of engine/drive-train vibration, and presents a low cost solution to the problem.

Variety of techniques are available to address the problem. The most popular method today is to put passive mounts (passive isolators) between the chassis and the engine/drive-train of automobiles, A passive mount is essentially a soft spring and damper combination that can significantly reduce vibration propagation from the engine/drive-train to the chassis of a automobile. While soft passive mounts are effective in reducing broadband vibration, the presence of soft connections results in substantial deterioration of the automobile stability during cornering and acceleration, and is unacceptable specially for high performance cars. Another method would be to use broadband [4,5,6] active control to suppress the vibration disturbance at the point where engine/drive-train is mounted to the chassis. However, such broadband control fails to provide adequate performance since it does not focus its energy at the disturbance frequencies where needed. Since engine/drive-train vibration disturbance is primarily composed of discrete harmonic tones that are highly correlated to the engine speed (RPM), a preferred way would be to actively suppress the most annoying harmonic tones by self tuning notch filter bank. This method does not cause any general softening of the connection between the engine/drive-train and the chassis. In most cases, the first few harmonic tones that are also correlated to the engine RPM are targeted. Due to the tracking nature, the method easily accommodates normal variation of the crank shaft speed (RPM) and continually adjusts the center frequency of each notch filter in the bank

The Jet Propulsion Laboratory (JPL) has been developing this active engine mount technology with an objective of reducing narrow-band vibration by at least 20dB (tenfold) in the neighborhood of the normal cruising speed (i.e., 3000 RPM). One of the most severe constraints in developing such technology is the cost per unit at production, It turns out that with the advent of low cost DSP (Digital Signal Processor) technology and the low cost rare-earth magnets, such systems can be implemented well below $100.00 per unit.

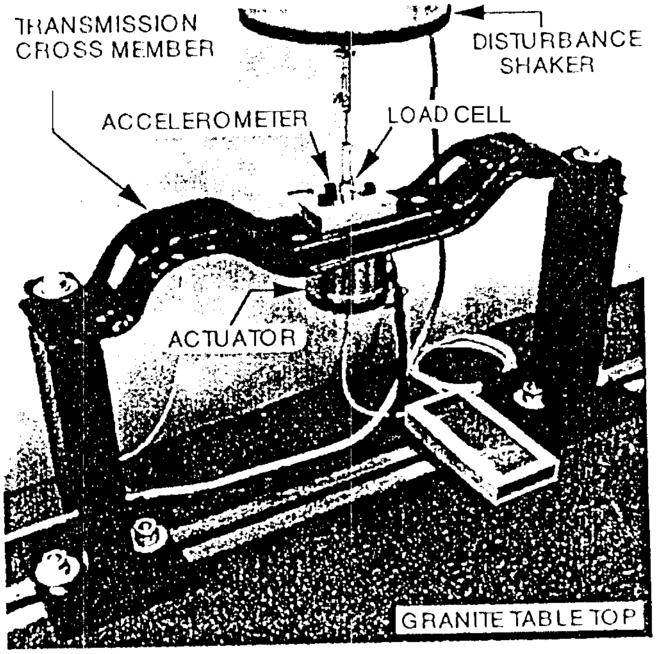

Figure 1. JPL active engine mount technology experiment.

2. SYSTEM HARDWARE

The proposed engine/drive-train noise and vibration suppression system employs one accelerometer attached to the drive-train mounting cross member, a proof mass actuator rigidly attached to the cross member, a magnetic pickup installed near a toothed flywheel, onc power amplifier, and a DSP (Digital Signal Processor) equipped with one A/D (Analog to Digital) converter, onc D/A (Digital to Analog) converter and onc programmable digital counter. The proof mass actuator is capable of generating forces as large as 40N for frequencies above 40Hz and weighs about 1 Kg. The vibration control system hardware arc chosen to suppress the drive-tl ain vibration disturbance for a Lincoln Mark VI 11 automobile.

The accelerometer senses the en,gillc/drive-train vibration fed to the DSP through the A/D converter. The DSP implements the compensator using a modified form of the Least Mean Square (LMS) algorithm and generates a control signal through the D/A converter and amplifier. The proof mass actuator responds by generating an opposing force to the vibration, The digital counter counts the number of teeth of the flywheel passed at any instant of time. This number is used by the DSP to detect the relative flywheel location and to generate the reference signals for the compensator for all harmonic tones being suppressed. All the reference signals and the necessary orthogonal quadrature signals are generated using only one lookup table. The generation of the reference and orthogonal quadrature signals requires little computation even for a large number of harmonics (both multiple and fractional of crank shaft speed).

3. EXPERIMENT SETUP

In our laboratory verification test at JPL, the drive-train mounting cross member was mounted on two rigid posts clamped to a granite block as shown in Figure 1 and Figure 2. The compensator is implemented in a TI C-$0 DSP to attenuate the first five harmonics and four in-between half harmonics. The engine/drive-train disturbance was generated by a large shaker attached to the cross member from the top, The data for acceleration of the cross member and magnetic sensor pickup of the flywheel motion at various engine speeds (rpm) were measured off-line by the FORD Motor Co. The two-channel data was recorded simultaneously on a TEAC RD-135T Data Recorder and was delivered to JPL for this vibration control experiment. During the laboratory test, the recorded data is played back to generate the similar cross member acceleration by the attached shaker and to feed the digital counter of the DSP with the measured pulse train signal. The recorded acceleration signal is passed through an inverse filter and then is used to drive the disturbance shaker amplifier. This is done in order to approximate the open loop cross member acceleration to that of the measured value.

4. COMPENSATOR DESIGN

A modified form of the Least Mean Square (LMS) algorithm [1,2,3] is implemented to realize a set of tracking type notch filter banks covering all harmonic tones of interest, As all LMS filter blocks in the compensator arc driven by an

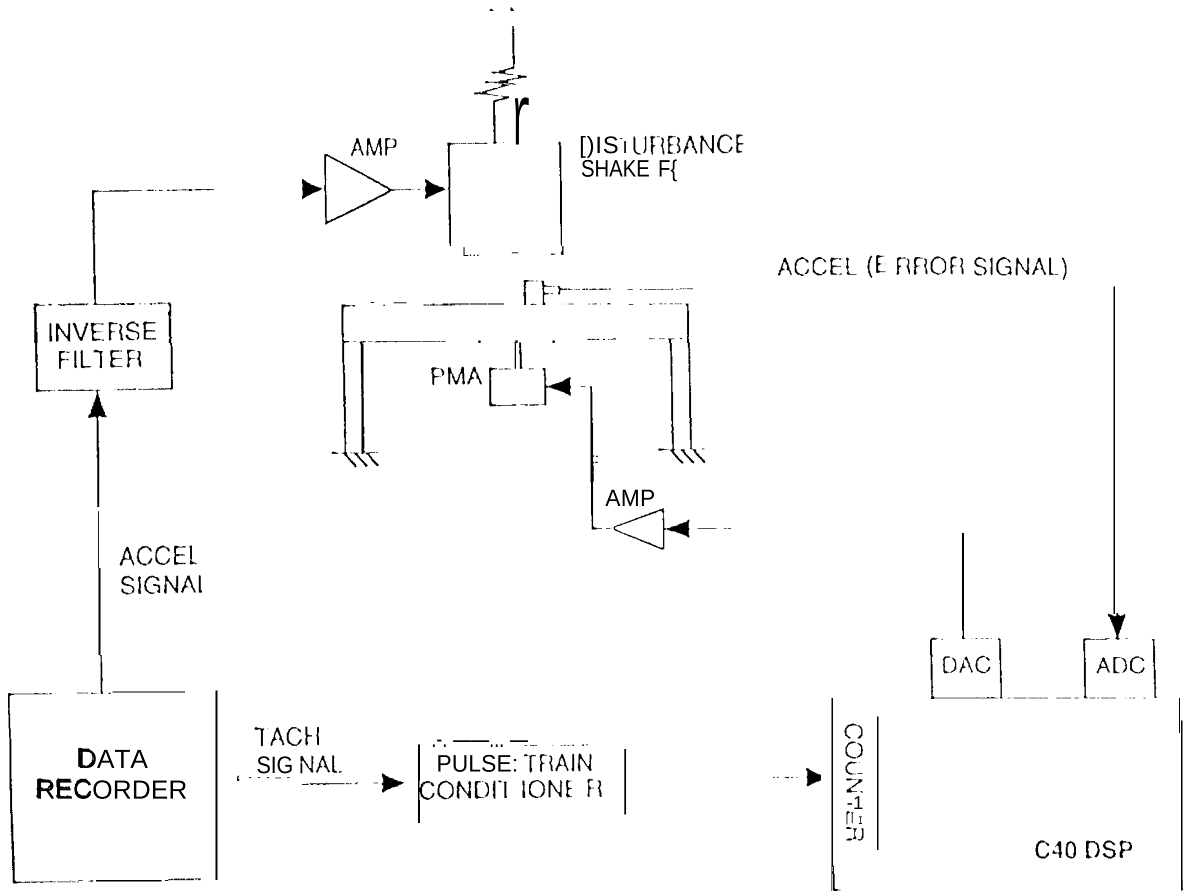

Figure 2. JPL active engine mount experiment setup schematic.

orthogonal set of reference signals of magnitude C_k and frequency ω_k for all harmonic tones k, the compensator converges to the following notch filter bank:

$$G(s) = \sum_k 2C_k^2 \mu_k \frac{s}{s^2 + \omega_k^2}$$

where, μ_k is the convergence parameter for the kth filter. Note that all of the notch filters have no damping to limit their gains to finite values. This is necessary for many practical reasons such as for identification of the implemented filters and for more stable digital implementation. To introduce damping, the pole of the integrator of each notch filter is slightly perturbed from the origin to the left half real axis of the complex plane.

The most important part in implementing the compensator is generation of orthogonal reference signals for each filter without using much computational power. The magnetic pickup sensor installed near the fly wheel and the programmable counter of the DSP are used to generate all necessary orthogonal reference signals. A pulse is generated as each gear tooth of the flywheel passes the magnetic pickup sensor. A flywheel with n teeth generates n pulses and each pulse represents a $2\pi/n$ radian rotation of the engine's crank shaft. Obviously, a larger n will result in finer resolution. The flywheel of the Lincoln Mark VIII automobile had 164 teeth. A total pulse count at any instant represents the crank shaft relative angular position which is strongly correlated to the engine/power train vibration disturbance. The DSP counter is set to count upward and resets (to zero) when the count reaches nm, where m is the smallest integer such that km is an integer for every harmonic tone k. For example, when k= 1/3, 1/2, 2/3, 1, 4/3, 3/2, 5/3, 2, ... m must be equal to 6 to satisfy this condition, i.e. 6 is the smallest number that is divisible by both 2 and 3.

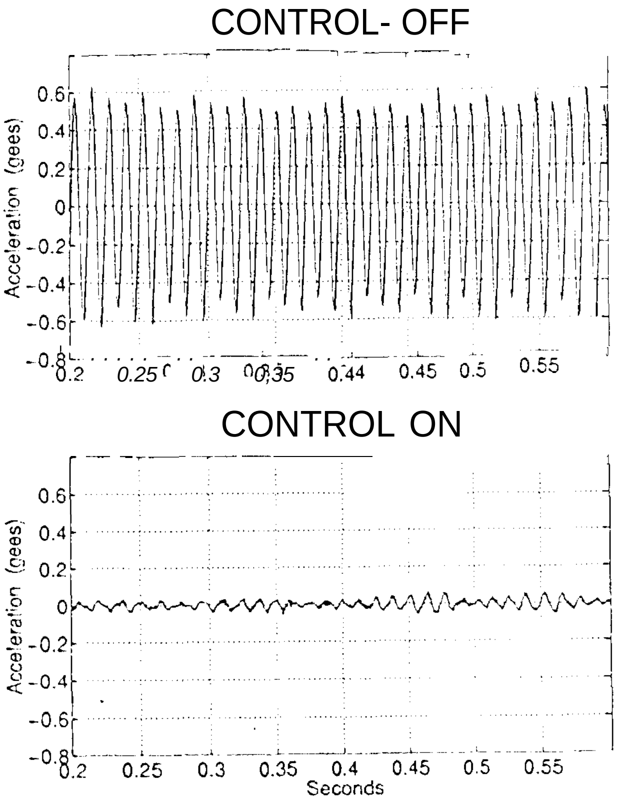

Figure 4. Open loop and closed loop time domain acceleration disturbance **data.**

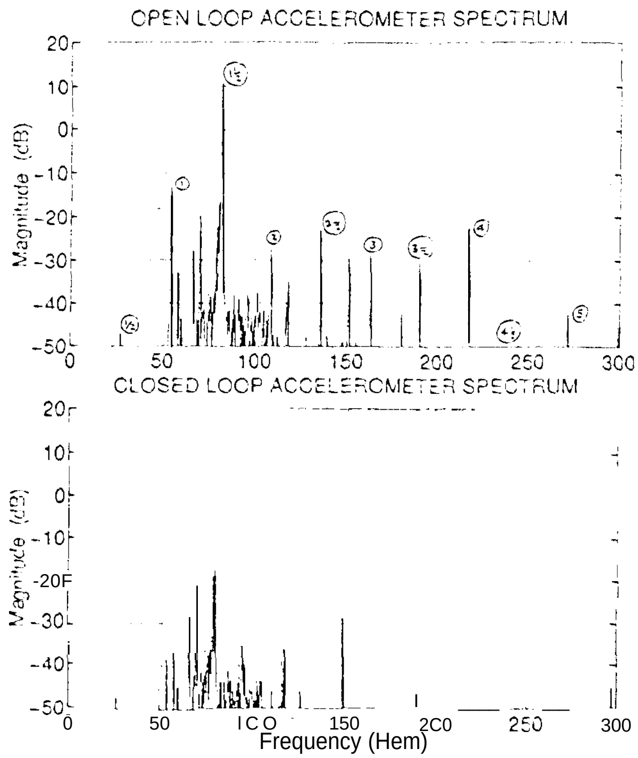

Figure 3. Open loop **and** closed loop frequency **domain acceleration** disturbance **data.**

A lookup table with *nm* elements is created outside the compensator loop and is filled with the values given below.

$$T(l) = \cos(\frac{2\pi}{nm} l);\ l = 0, \cdots, nm-1,$$

With this setup, orthogonal reference signals, X_k and Y_k, for each hamonic tone k are easily generated as follows

$$X_k = T(\{mki\}\%\{nm\}),$$
$$Y_k = T(\{mki + nm/4\}\%\{nm\})$$

where, i and % represent the instantaneous value of the counter and the modulus operator respectively. A modulus operation yields the remainder when the first integer is divided by the second integer. Note that for this setup, C_k is equal to 1 for each harmonic k and this simplifies the notch filter bank equation given previously. Therefore the filter bank gain is completely independent of the pulse train signal amplitude and frequency. The center frequencies of the notch filter bank continually adjust with the pulse train signal frequency only.

For each harmonic tone, only two modulus, one addition and one division operations are needed for orthogonal reference signals generation at every update of the compensator. One additional advantage of the current implementation is that the orthogonal reference signals, X_k and Y_k, instantaneously tracks any change in engine crank shaft speed and thus results in a quicker and improved suppression response. It is important to mention here that as *n/k* becomes smaller with higher *k*, the corresponding orthogonal reference signals become less and less smooth and insert high frequency noise to the proof mass actuator. A smoothing lowpass analog filter placed between the D/A converter and the power amplifier greatly alleviates the problem.

5. EXPERIMENT

The compensator for the first 9 harmonic tones (1, 3/2, 2, 5/2, 3, 7/2, 4, 9/2, and 5) is implemented on a TI C40 board with an update rate of 10 KHz.

Figure 3 and Figure 4 respectively show the time and the frequency domain cross bar acceleration data for the cruising speed of about 3200 *rpm*. Comparison of the data for the open loop and the closed loop tests show a reduction of 25 dB of the fundamental harmonic ($k=1$) and 30 dB of the strongest one-and-a-half harmonic ($k=1\%$). The over all reduction was better than 20 dB for all harmonics in the frequency range of 0-500 Hz.

6. CONCLUS1ON

We presented a control algorithm and a hardware architecture for implementing engine/power-train vibration suppression. 'I'he results showed suppression of disturbance harmonics and sub harmonics in excess of 20 dB. The algorithm that is implemented in the TI C40 DSP is simple and can easily be implemented in a inexpensive dedicated DSP. Newer, faster and cheaper S-bit RISC-based micro-controllers show promise for low cost implementation. Our future research will focus on reducing number of parts count in an effort to bring down the overall cost for volume production.

7. ACKNOWLEDGMENT

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