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Vibration control of an automotive engine using active mounts

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ABSTRACT

In this research, vibration control of national engine (EF7) using active mounts is studied for two different mounting layouts. The first layout is based on the current engine mounting of national engine (system of three passive mounts), but the hydraulic mount is substituted by an active mount. The other mounting layout contains 4 mounts, 2 of which are active ones and the others are passive mounts. In order to simulate the dynamic behavior of the engine, a 6-degree-of-freedom, one-way-coupled model is hired. In addition, a proper dynamic model for active mount with electromagnetic actuator is chosen.

In order to reduce transmitted force to the chassis through the mounting system, a feed-forward adaptive control method, FXLMS (Filtered-x Least Mean Squares) is employed which is one of the widely used methods in cancellation of acoustical and vibrational noises. Considering the characteristics of the transmitted force to the chassis, this method is used in a specific algorithm called narrow-band algorithm to cancel dominant harmonic frequency of the unwanted vibrations of the engine.

1) Introduction

The automotive engine is an important source of sounds and vibrations in vehicles. Therefore, engineers look forward to find new methods to vanish these unwanted effects of the engine. Motion of moving elements of the engine and also combustion pressure in the chambers generate forces in the frequency range of almost 20 Hz to 200 Hz. Reducing the effects of these forces can improve the vibration behavior of the engine and in addition, make the vehicle more silent and this can increase the durability of the engine components.

Many researchers have reported their studies on active control of engine vibration in the literature. Nakaji et al. [1] developed and tested an active control engine mount to reduce the idling vibration and booming noise in automobile engines. They employed an adaptive control strategy based on synchronized filtered-X least mean squares (SFXLMS). Aoki et al. [2] hired SFXLMS to improve the quietness of diesel engine vehicles. They employed a simplified adaptive filter, focusing on periodic components of vibrations induced by engine revolutions. Seba et al. [3] employed H_∞ feedback control and FXLMS feedforward control for engine vibration isolation and compared the performance results obtained from the mentioned controllers. They simulated the engine disturbance using electromechanical shaker and designed the controllers based on experimental identification. Hillis et al. [4] implemented the error-driven minimal controller synthesis (Er-MCSI) adaptive controller for the control of active engine mounts and compared the results to those of implementation of FXLMS adaptive controller. They applied both algorithms to an active mount fitted to a saloon car equipped with four-cylinder turbo-diesel engine without any prior knowledge of the system dynamics. Karimi [5] presented a method of designing optimal vibration control based on haar functions to control the bounce and pitch vibrations in engine-body system. Olsson [6] published the results of the studies on feedforward control of engine vibration using active mounts. A multi-input multi-output model containing the nonlinear behavior of engine mounts and the saturation in active mounts was used. Choi et al. [7] introduced a new type of three-axis active mount using piezoelectric actuators and investigated its dynamic characteristics. Moreover, they hired a Linear Quadratic Regulator (LQR) algorithm to control the vibration of system. Liang and Shi [8] used electromagnetically actuated active mounts to achieve better vibration performance than passive engine mounts. They employed fuzzy control method. Truong and Ahn [9] introduced a new type of semi-active hydraulic engine mount using controllable area of inertia track. They employed an optimal control algorithm to isolate the engine-related vibrations. Hillis [10] used FXLMS method to control a four-cylinder turbo-diesel engine using two active mounts. A system of two active mounts and one hydraulic passive mount was used as the mounting system. Mansour et al [11] designed an active decoupler hydraulic engine mount with application to variable displacement engine. They described the modeling and experimental analyses of the active engine mount and presented simulation and experimental results for different control signals. Fakhari and Ohadi [12] studied active vibration control of national engine using an electromagnetically actuated active mount and used robust control methods (H_2 and H_∞ schemes) in their studies.

In this research, vibration control of national engine (EF7) using active mounts for two different mounting layouts is studied and compared. To the best of the authors' knowledge, this matter has not been presented in any literature. In this regard, 3-mount layout (with one active mount) and 4-mount layout (with two active mounts) are considered. Effectiveness of the mentioned active mounting layouts in reducing transmitted force to the chassis is compared.

In section 2, the problem is described; how the engine induces unwanted vibrations in chassis and how the active system tries to cancel them. Then, the concepts of primary and secondary paths are presented. Also, the necessity of using adaptive digital control is discussed. In section 3, the model of engine which is used to simulate the vibration behavior on mounting system is introduced. Then, mounting system of national engine and another proposed 4-mount layout are introduced. Section 4 is dedicated to modeling of the active mount. In section 5, FXLMS control algorithm, identification of secondary path and the structure of adaptive notch filters in FXLMS are discussed. In section 6, the procedure for design of the controller for both mounting layouts is discussed. Section 7 is devoted to presenting the results of the research and comparing two mounting system layouts and the last section is dedicated to conclusion.

2) Problem statement

As shownin Figure 1, the combustion pressure in the chambers and the motions of various internal parts of the engine generate force F_e which excites the engine and is transmitted to the chassis through engine mounting system. Transmitted force to the chassis (F_t) make annoying sound and vibrations for the passengers in vehicle compartment. Therefore, the goal of this study is to reduce the effect of the engine force F_e on the chassis in vertical direction using active method. In other words, it is required to reduce or cancel transmitted force to the chassis F_t by adding active force F_a created by an actuator.

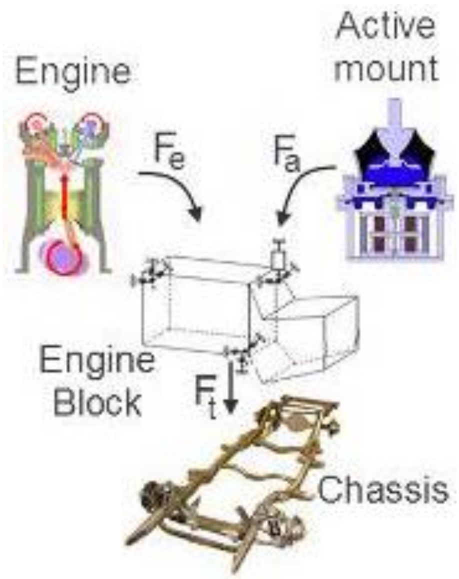

Figure 1: Transmission of the forces from the engine to the chassis and the effect of active mounts

F_a is generated based on a specific control law and affects the 3-dimentional motions of the engine block to cancel the effect of F_e and reduce F_t. In other words, vibration of the chassis might cause unwanted vibration or noise and the aim of an active control system is to reduce these annoying noises and vibrations. Vibration of the source is transmitted to the chassis through the mounting system (which is called primary paths) and the goal of the controller design is to cancel the transmitted vibration to the chassis. Therefore, the controller generates secondary vibrations (through secondary path) in opposite phase with the source vibration called anti-vibration. Finally, vibration and anti-vibration signals are combined together and cancel each other. It is desirable for the controller to be digital one, where the analog signals from electromechanical transducers are sampled and processed using digital signal processing (DSP) systems with enough speed and precision to execute sophisticated mathematical functions in real time [13].

Since vibration behavior of the engine is non-linear and excitation source of the engine has varying properties (because of varying engine speed), the active system should be adaptive. One of the most suitable tools in digital processing is adaptive filter which is fast, precise and efficient enough to be used for this problem. These filters continuously adjust some coefficients to minimize an error signal. In the considered problem in this study, the mentioned error would be the transmitted force to the chassis.

3) Dynamic modeling of the engine

In order to study performance of the active engine mount, a proper dynamic model of the engine is required. In this paper, a one-way-coupled model of the engine presented by Hoffman and Dowling [14] is employed.

In this model, it is presumed that the motion of the engine block does not affect the motion of the internal elements of the engine such as the crankshaft and pistons. A detailed derivation of the non-linear motion equations of crankshaft, connecting-rods and pistons, 3-dimentional motion equations of the engine block and specifications of the national engine used in this study can be found in reference [9].

Currently, the national engine is mounted fully passive that is shown in Figure 2. It consists of three engine mounts: two rubber mounts (transmission mount and torque strut mount) and one hydraulic mount. In this study, two different active mounting layouts are considered for national engine. In the first layout, the hydraulic mount is replaced with an active mount without any change in the locations of the transmission and torque strut mounts (3-mount layout). In other words, this layout contains one active and two passive mounts.

In the second layout, by keeping the locations of hydraulic and transmission mounts, the torque strut mount is replaced by two similar active mounts that are symmetrically positioned with respect to the crankshaft axis (4-mount layout). In other words, in this layout, two active and two passive mounts are considered. This 4-mount layout is not necessarily the best design and the optimization design for locations of the mounts can be subject of a separate study. However, the main purpose of this study is evaluation of the effectiveness of the active mount in the four-mount and three-mount layouts.

4) Modeling of the active mount

In this study, the active mount model presented by Lee and Lee [15] is used. In this model, active mount is the same as a hydraulic mount in structure, except for the decoupler which is replaced by an electromagnetic actuator; this actuator can control the pressure in the upper chamber and the force transmitted to the chassis. Figure 3 indicates the considered active mount.

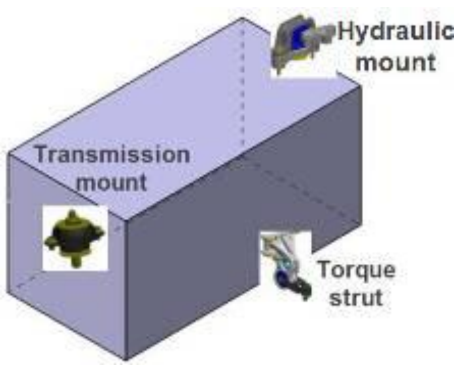

Figure 2: The current configuration of engine mounts in national engine (three passive mounts)

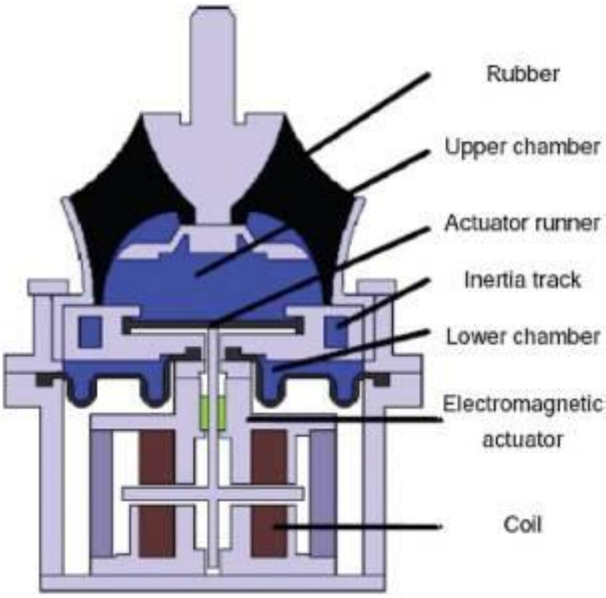

Figure 3: Active mount components [15]

Figure 4 indicates the model presented by Lee and Lee [15] schematically. The differential equations that describe the behavior of the engine mount are presented here, but the details can be found in references [15,12].

$$m_t\ddot{x}_t + R_t\dot{x}_t + K_t x_t = -A_t P \quad (1)$$

$$m_c\ddot{x}_c + R_c\dot{x}_c + K_c x_c = -A_c P + f_c \quad (2)$$

$$A_e\dot{x}_e + C_b\dot{P} = A_t\dot{x}_t + A_c\dot{x}_c \quad (3)$$

and the transmitted force to the chassis can be presented as:

$$F_{trans} = K_r x_e - P(A_e - A_c) + R_c\dot{x}_c + K_c x_c - f_c \quad (4)$$

In equations 1-4, K_r, C_b, A_e, A_C, A_t, m_t, R_t, K_t, m_c, R_c and K_c represent stiffness of the mount in the main direction, volumetric compliance of the rubber, equivalent cross-sectional area of the upper chamber, actuator runner and the inertia track, equivalent mass of the fluid in the inertia track, equivalent damping coefficient of the inertia track, volumetric compliance of the lower chamber, runner mass, damping coefficient and stiffness of mechanical spring of the actuator, respectively. Also, P, x_e, x_c and x_t denote the pressure of the upper chamber, engine vibration in the main direction of the active mount, actuator runner displacement and the equivalent displacement of the equivalent mass in the inertia track, respectively. In addition, f_c represents the force generated by the actuator that is applied to the engine and chassis.

For further information on the modeling of the active engine mount and the numerical values used in the simulation, one can refer to the references [15, 12].

In order to consider the behavior of the actuator, it is simulated based on the model presented by Lee and Lee in [16]. In this actuator, two coils are employed to control the position of the actuator runner using control current *i*. Figure 5 demonstrates a schematic structure of the actuator.

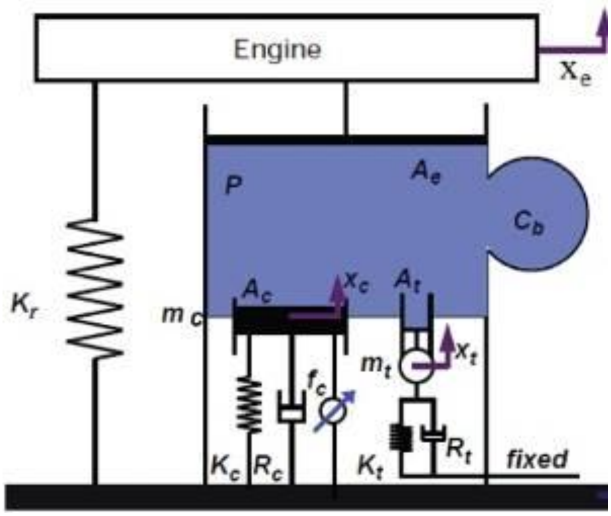

Figure 4: Active engine mount model [12,15]

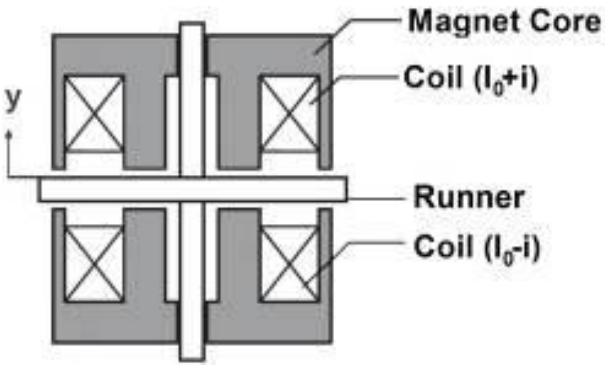

Figure 5: Electromagnetic actuator model [16]

A linearized model of the generated force by the actuator is represented in [16] and it can be assumed to be a linear function of the control current i and the runner displacement x_c as below:

$$f_c = K_i i + K_{x_c} x_c \quad (5)$$

The coefficients K_i and K_{xc} can be calculated using Coil turns (N), offset current (I_0), pole face area (A_p) and nominal gap (g_0) [16].

5) FXLMS method

An active vibration system is, in fact, an electromechanical system which has to cancel the unwanted vibration by generating an appropriate signal and using the superposition principle. Since the vibration source does not necessarily perform in steady state condition and so the frequency, amplitude and the phase of the unwanted signal are variable, the controller must be adaptive so that it can be adjustable to the source changes. Adaptive filters are one of the most commonly used tools in active cancellation and have the ability to adjust their coefficients in order to optimize an error signal based on the Least Squares method.

Figure 6 indicates a schematic diagram for an active control system. In this method, using a reference signal $x(n)$ and the control algorithm, the canceller signal $y(n)$ is generated and combined by the unwanted vibration in the physical domain and cancel it. The aim of the adaptive filter $W(z)$ is to minimize the error signal and in the ideal case reduce it to zero. So, the adaptive algorithm tries to adjust the digital filter coefficients using LMS[1] algorithm to leave small amount of error signal $e(n)$ and the convergence occurs when the filter $W(z)$ estimates the primary path $P(z)$ ($W(z)=P(z)$). In other words, the convergence is occurred when the primary and secondary vibration are equal ($d(n)=y(n)$) and this means that the unwanted vibration is cancelled in the physical domain.

Figure 6 demonstrates a general algorithm of active control, but the effect of secondary path is not considered. When the secondary path is considered, the use of FXLMS algorithm is inevitable; in which the reference signal $x(n)$ is filtered by an identified model of secondary path $\hat{S}(z)$ and the filtered signal $x'(n)$ is fed to the LMS unit instead of $x(n)$ to guarantee the convergence of the LMS algorithm [13]. Figure 7 indicates the FXLMS algorithm diagram.

The filter $W(z)$ is a Finite Impulse Response (FIR) filter with L adjustable coefficients. The following vector is the coefficient vector of $W(z)$ in the sample time n:

$$w(n) = [w_0(n)\ \ w_1(n)\ \ \cdots\ \ w_{L-1}(n)]^T \tag{6}$$

and a tapped delayed line of reference signal $x(n)$ is used to generate the output of the filter.

$$x(n) = [x(n)\ \ x(n-1)\ \ \cdots\ \ x(n-L+1)]^T \tag{7}$$

$$y(n) = w^T(n)x(n) = w(n) * x(n) \tag{8}$$

and $x'(n)$ is generated by linear convolution combination of vectors $x(n)$ and $s(n)$. Note that $s(n)$ is the representative of $\hat{S}(z)$. In fact, using the off-line identification of secondary path, a pulse transfer function $s(n)$, a vector of length L, is derived to model the secondary path $S(z)$ which is the same for all samples.

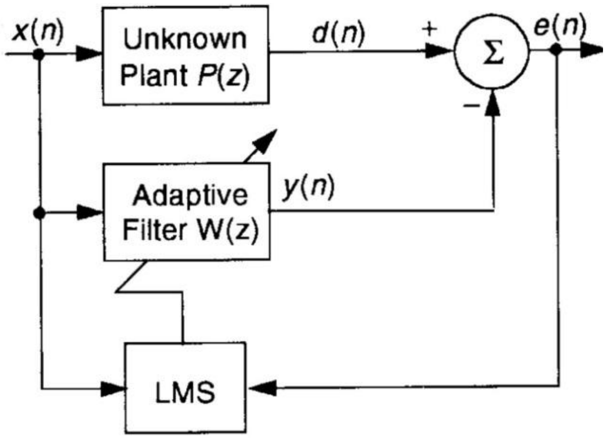

Figure 6: A schematic diagram for an active control system [13, 17]

$$x'(n) = s(n) * x(n) \tag{9}$$

The coefficient vector w(n) is updated using the following equation which is derived to minimize the MSE[2] surface using the steepest descent algorithm.

$$w(n+1) = w(n) + \mu x'(n)e(n) \tag{10}$$

The equation 10 shows what happens in the LMS box in Figure 7. In this equation, $x'(n)$ is a tapped delayed line of signal $x'(n)$ and μ is a coefficient which is determined to control the convergence process of the filter coefficients.

The secondary path can be identified using on-line or off-line system identification methods based on the characteristics of the secondary path. In the engine vibration problem, the secondary path consists of the active mount model and also depends on the orientation of active mount. So, the secondary path is time-invariant for our problem and identification using off-line method is sufficient. Figure 8 demonstrates the structure of off-line identifier used to model the secondary path $S(z)$. The concept of the system identification using FIR filters and LMS algorithm can be found in [13, 17].

In active control methods, based on the characteristics of the vibration, a broad-band or narrow-band method may be hired. When the vibration source generates primary excitations with a large range frequency content and it is possible to gather data from the primary excitation (for example in active noise control systems) the broad-band algorithms are used; however, if the primary excitation is a narrow-band signal or if it is not possible to gather data directly from this signal, narrow-band algorithms should be hired. For example, in engine vibration reduction problem, it is not possible to sense the excitation signal directly; instead, the rpm signal has enough information to make an appropriate signal for FXLMS algorithm. Figure 9 indicates a schematic view of a narrow-band active controller.

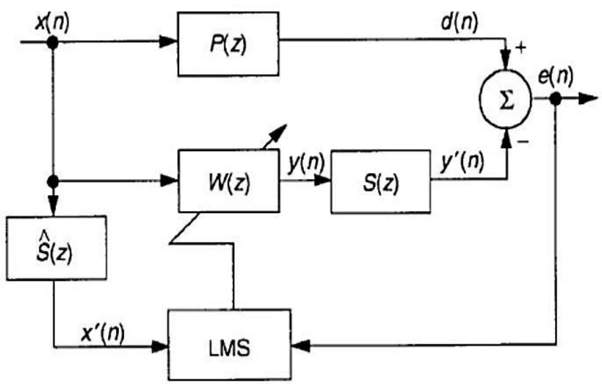

Figure 7: FXLMS algorithm [13, 17]

[1] Least mean squares

[2] Mean square error

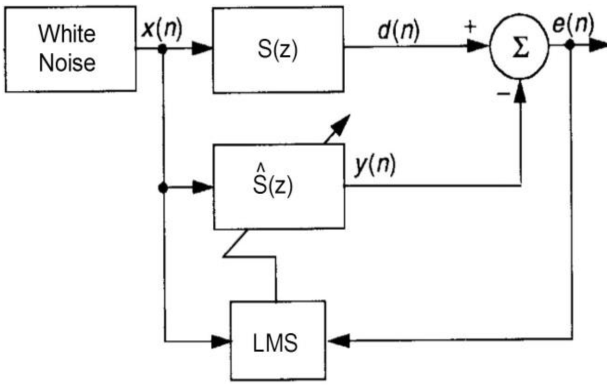

Figure 8: Identification of secondary path using FIR filters [13, 17]

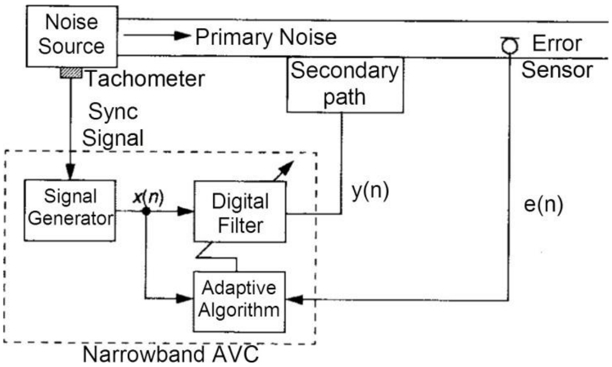

Figure 9: The narrow-band active vibration control system [13, 17]

In this method, reference signal is derived from a non-structural sensor such as tachometer and sinusoidal signals are generated with frequencies which are wanted to be canceled from the desired signal. This method is called adaptive control using adaptive notch filters.

Adaptive notch filters use the same algorithms as broad-band systems, but by using this method, specific frequencies can be cancelled from the desired signal.

The input reference signal is a cosine function $x(n)=x_0(n)=A\cos(\omega_0 n)$ and a 90 degrees phase deference generator is used to generate a sinusoidal function $x_1(n)=A\sin(\omega_0 n)$. Figure 10 shows the FXLMS algorithm diagram which uses adaptive notch filters to cancel specific frequency from the desired signal $d(n)$.

Two orthogonal signals x_0 and x_1 are generated using the tachometer signal and are considered as the input reference signal to the algorithm. These two signals are linearly combined to build the output signal $y(n)$. LMS algorithm uses the filtered input signals by estimated secondary path $\hat{S}(z)$ to adjust the coefficients W_1 and W_2 and reduce error signal. Also, parallel structure of multiple adaptive notch filters can be hired to cancel multiple frequencies from the desired signal.

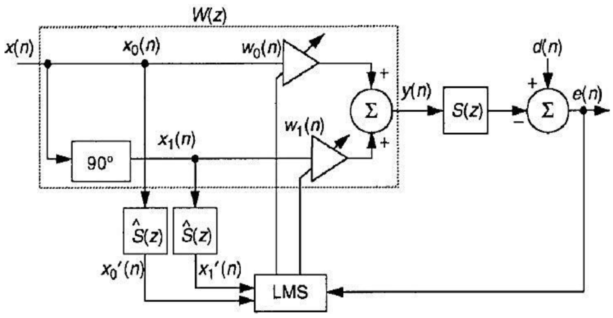

Figure 10: Filtered-X LMS algorithm using adaptive notch filters [13, 17]

6) Controller design

In this section, the details of design of the controller are discussed. As mentioned in previous sections, the main source of the unwanted vibration which is transferred to the chassis is the combustion and inertia forces in the engine. These forces are represented as the excitation signals in Figure 11. Since it is not possible to measure these forces easily to be used in the control algorithm, it is necessary to use a reference signal $x(n)$ which is in correlation with these forces (The details of selection of the appropriate reference signal are discussed in the following paragraphs).

The engine vibration is transmitted to the chassis through passive mounts and the passive section of the active mount which build the primary path $P(z)$. Control signal $y(n)=V_c$ is applied to the actuator and is transmitted to the chassis as control force $y'(n)=F_a$ through the secondary path $S(z)$ (which consists of the actuator and the active part of the active mounts). $\hat{S}(z)$ is the finite impulse transfer function of $S(z)$ which is derived using off-line identification of $S(z)$. $e(n)$ is the total force transmitted to chassis in the vertical direction which is determined using sensors implemented in the locations of engine mounts. $e(n)$ consists of both unwanted vibration and the effect of the actuator on the chassis.

Signal $e(n)$ and $x'(n)$ are processed using the LMS algorithm to adjust the pulse transfer function $W(z)$ to optimize the error signal $e(n)$. Figure 11 indicates the signal communication between components of the active control system of engine.

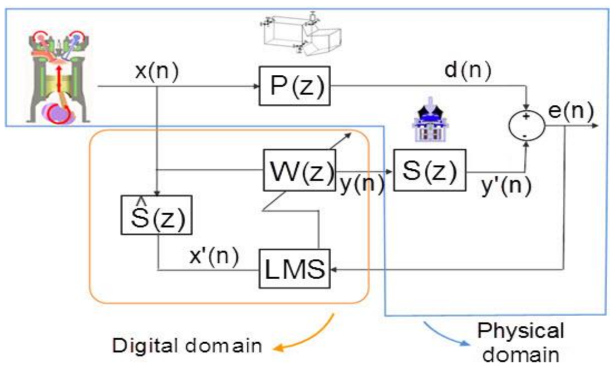

Figure 11: signal communication between components of the AVC system

As discussed before, appropriate reference signal should be chosen for FXLMS controller using narrow-band algorithm. An appropriate signal can be a harmonic signal with the frequency of the dominant frequency of the transmitted force to chassis because this signal has sufficient correlation to the transmitted force to chassis.

The dominant frequency of the transmitted force to chassis F_t depends on the number of the cylinders of the engine [18], the period of the combustion cycle and generated torque. For the national engine which is a four-cylinder engine, the dominant frequency in transmitted force to the chassis is 2 times engine speed (2*rpm). In other words, it can be seen in the pattern of the transmitted force to chassis that the frequency of 2*rpm constructs the most portion of F_t. Generally, almost all harmonics of the rpm appear in the transmitted force to chassis, but the second harmonic is the dominant frequency and cancelling this frequency from F_t might lead to effective reduction of transmitted force to chassis.

Figure 12 indicates the FFT[3] of the transmitted force to the chassis when the simulation is running in 5500 rpm ≈ 91.7 Hz for 5 seconds in steady state. It can be seen in this figure that the second harmonic of the engine speed is the dominant signal on the induced vibration on chassis. There are also other harmonics of the engine speed in the transmitted force to chassis which are strongly dominated by the second harmonic so are not considered as the target of the cancellation algorithm. Since the dominant frequency of vibration induced on the chassis is 2*rpm, it is reasonable to consider the reference signal to be the second harmonic of engine speed.

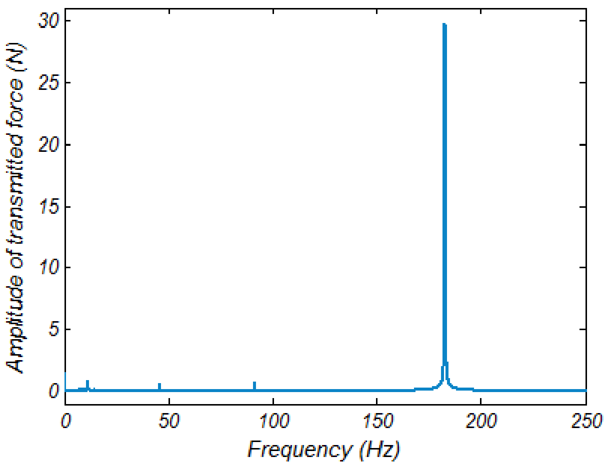

Figure 12: Amplitude of transmitted force to the chassis for simulation in 5500 rpm

For the 3-mount layout which was described in section 3, the controller includes an adaptive notch filter. Using the speed of the engine and a signal generator, two harmonic signals are generated and fed to the controller as reference signals. These signals are the sine and cosine functions with frequency of 2*rpm. Figure 13 indicates the structure of the adaptive controller.

In order to fully design the controller, sampling frequency, length of FIR filters and convergence coefficient should be set. In order to set the sampling frequency of the digital controller, it has to be considered that the sampling frequency of the controller should be at least 10 times of the most working frequency of the system which is 2*6000 rpm ≈ 200 Hz. So the sampling frequency is set to be 10*200 Hz = 2000 Hz.

Lengths of the filters and convergence coefficient have been set by trial and error method to reach the fastest and the most accurate cancellation of the transmitted force to the chassis. These parameters are presented in Table 1 for the 3-mount layout.

As mentioned before, our study has been extended to another mounting layout (4-mount) which was described in section 3. For the 4-mount layout, two sets of digital controllers, with similar structures to the controller which was used for the 3-mount layout, have been considered. Each of these digital controllers controls one of the active mounts. The characteristics of this controller are also listed in Table 1.

Figure 14 indicates how the controller interacts with other parts of the system and how the signals are transferred for the 4-mount layout.

Table 1: The characteristics of the adaptive filters for the 3-mount and 4-mount layouts

Characteristics	3-mount	4-mount
μ	10^{-6}	10^{-7}
L	10	10
F_s	2000 Hz	2000 Hz

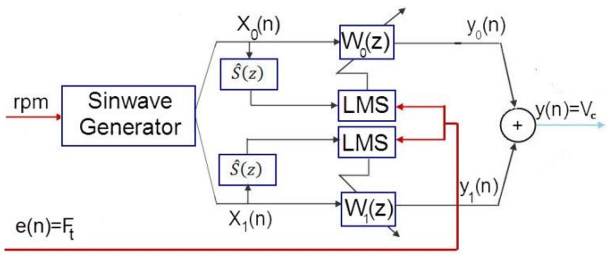

Figure 13: Structure of the FXLMS controller with adaptive notch filters

[3] Fast Fourier transform

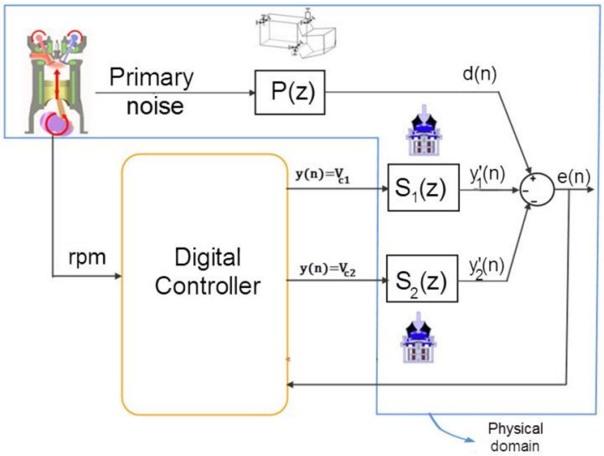

Figure 14: Interaction of system elements for the 4-mount layout

7) Results and discussion

In this section, the results obtained from the simulation of the models with 3 and 4 engine mounts are represented. In order to show the effect of the performance of the controllers, the simulation time is considered to be 15 seconds and in the first 5 seconds, the controller is set to be off. Then the controller starts to work and reduces the transmitted force to the chassis.

First 2-3 seconds expresses the transient response of the system in which the effect of the initial conditions is apparent, and then the steady state response of the system appears and continues until the end of the fifth second that the controller starts to work. The controller tries to reduce the transmitted force to the chassis and when the convergence is reached, the steady state response of the system with activated controller is appeared. Figure 15 shows the transmitted force to chassis when the engine is working in 4000 rpm and Figure 16 shows the control voltage generated by the controller to be applied on the active mount input voltage gate.

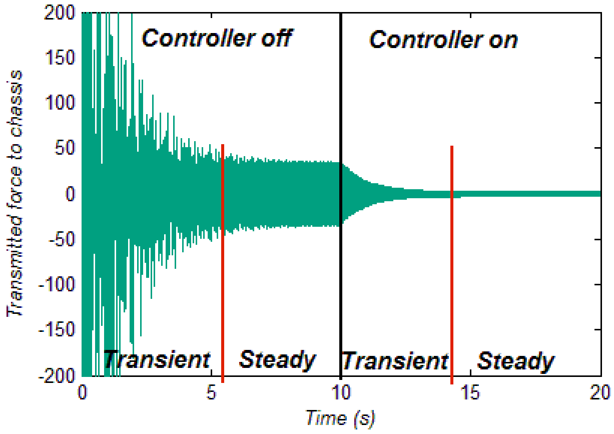

Figure 15: Total transmitted force to chassis for the 3-mount layout in 4000 rpm

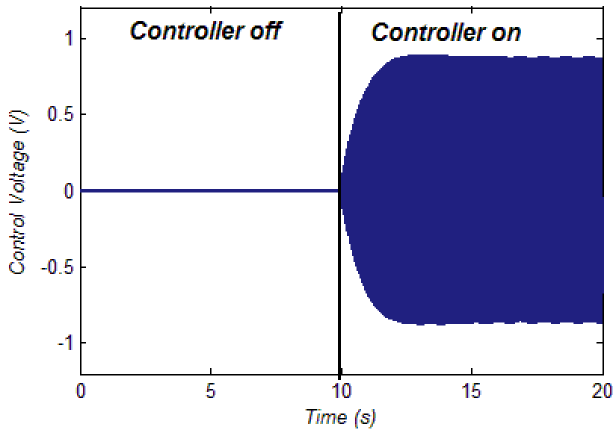

Figure 16: Control voltage generated by the controller for the 3-mount layout in 4000 rpm

By referring to the results presented in Figures 15 and 16, it can be understood that the controller does have appropriate performance in 4000 rpm; the reduction in amplitude of the transmitted force to the chassis is 85 percent and the control voltage has amplitude of 1 volt.

Performance study of the controller in all engine speeds show that the convergence is always reached for the designed controller.

In order to compare the frequency content of the transmitted force to the chassis when the controller is on to that of the controller when it is off, the FFT of the transmitted force to the chassis with and without control is presented in Figure 17. It is apparent that the second harmonic of engine speed is completely eliminated from the transmitted force to the chassis.

Similar results are obtained for the 4-mount layout.

In order to investigate the performance of the controller and compare two models (3-mount and 4-mount layouts), the steady state amplitude of transmitted force to the chassis and also the control voltage are presented in Figures 18 and 19. Results show that by activating the controller in 3-mount and 4-mount layouts, transmitted force to the chassis is considerably reduced.

It is also apparent that the transmitted force to the chassis is reduced about 85 % and 95 % for 3-mount and 4-mount layouts, respectively, in almost all engine speeds.

Referring to Figures 18 and 19, it is observed that the controller does not have a normal performance in almost 5000 rpm. Investigations in this study showed that this behavior is because of the resonance of the fluid in inertia track of active mount which prevents the controller to work properly. This inappropriate behavior can be eliminated by re-designing the active mounts so that the resonance does not happen in the effective frequency domain of the controller performance.

According to the obtained results, addition of the 4th mount (second active mount) to the mounting system reduces the force transmission in low engine speeds when the controller is inactive. In the case of activating the controller, transmitted force is more reduced in 4-mount layout compared with 3-mount layout. However, by addition of the 4th mount, transmitted force is reduced a little with relatively high control voltage.

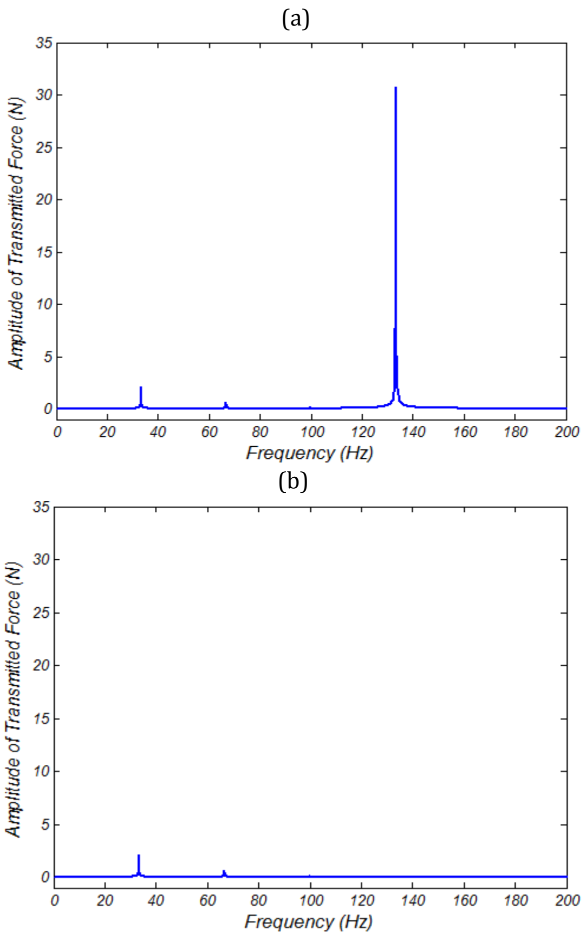

Figure 17: FFT of transmitted force to the chassis for 5-second simulation in 4000 rpm; (a) controller: off (b) controller: on

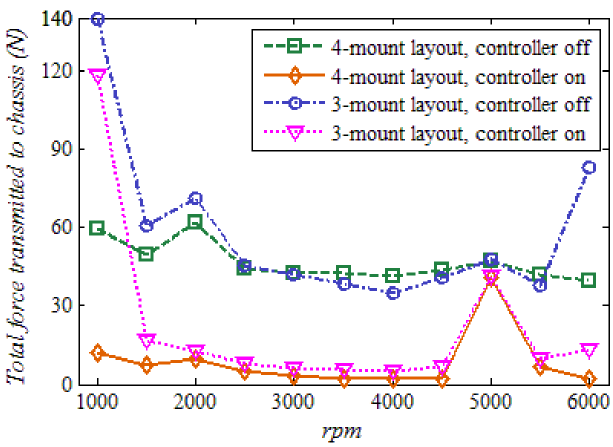

Figure 18: Steady state amplitude of transmitted force to the chassis in different engine speeds

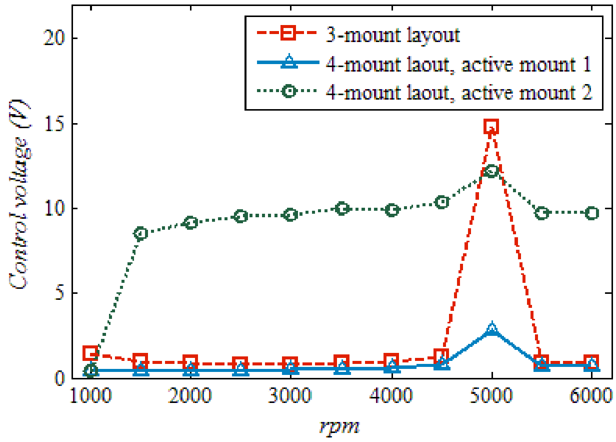

Figure 19: Steady state amplitude of control signal in different engine speeds

On the other hand, the ranges of the control voltage related to both active mounts in 4-mount layout are completely different. It is because of the fact that the locations of the mounts (especially 4th mount) are not optimized in this study and it is possible to achieve better improvements by finding better locations for the engine mounts. In other words, by optimizing the location of the 4th mount in 4-mount layout, transmitted force can be more reduced compared to 3-mount layout. Also, control voltage related to the second active mount in 4-mount layout can be reduced to the corresponding range of the first active mount by optimizing its location.

Altogether, it can be concluded that by employing one active mount (in 3-mount layout), transmitted force is considerably reduced. However, if more reduction in transmitted force is required, two active mounts (in 4-mount layout) should be used. But it is noted that the second active mount should be employed in an optimized location to achieve more effective reduction in transmitted force with lower control voltage. In other words, if the position of the second active mount is not optimized, less reduction in transmitted force with high control effort will be obtained.

8) Conclusions

In this article, vibration behavior of 4-cylinder engine (national engine) on active engine mount systems was simulated and studied by employing a proper dynamic model. In addition, vibration behavior of the engine with 3-mount and 4-mount layouts were investigated and compared.

It is concluded that vibration behavior of the engine with active mounts is more desired than that of the engine with current passive mounting system. In addition, in almost all engine speeds, the active system reduces the amplitude of the force transmitted to the chassis by 85 and 95 percent for the 3-mount and 4-mount layouts, respectively.

The obtained simulation results in this study show that the transmitted force is considerably reduced by employing one active mount (in 3-mount layout).

However, if more reduction in transmitted force is required, two active mounts (in 4-mount layout) should be used. But it is noted that the second active mount should be employed in an optimized location to achieve more effective reduction in transmitted force with lower control voltage.
As a future study, a general optimization problem can be performed to find the best locations of the engine mounts (especially active mounts) to achieve the best mounting system which is more able to cancel the unwanted vibration on the chassis with the optimized effort of the actuator.
It has to be mentioned that, in this article, the dominant frequency of the engine vibration which is 2*rpm was used for the cancellation of the unwanted vibration. Using parallel structure of multiple notch filters and considering different sets of frequencies to be cancelled, better cancellation results can be reached and is suggested for future studies. Also, performance of the controller should be studied under unsteady conditions of the engine (such as variable speed and gear changes) and is suggested for completing this study. Moreover, it is worth taking into consideration that in this research, the noises in sensors have not been simulated. In fact, the most important source of such noises is the force sensors that measure the transmitted force to the chassis and such noises could possibly influence the performance of the controller. So, another study can be performed to find the effect of these noises on the performance of such active control systems.

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List of Symbols

A_c	Decoupler area, m^2
A_e	Equivalent piston area of rubber,m^2
A_p	Pole face area,m^2
A_t	Cross-sectional area of inertia track,m^2
C_b	Volumetric Compliance, m^5/N
$d(n)$	Desired signal, N
$e(n)$	Error signal, N
F_a	Total force generated by active mounts, N
F_e	Induced force on the chassis due to performance of engine, N
F_s	Sampling frequency of digital systems, Hz
F_t	Total transmitted force to chassis in vertical direction, N
g_0	Nominal gap of electromagnetic actuator, m
I_0	Basic electric current in the coils of electromagnetic actuator, A
i_c	Control current input to electromagnetic actuator, A
K_c	Stiffness of actuator, N/m
K_r	Main stiffness of rubber part of active mount, N/m
K_t	Stiffness of diaphragm, N/m
L	Length of digital filters
m_c	Mass of actuator runner, Kg
m_t	Mass of fluid in inertia track, Kg
N	Number of coil turns
P	Primary path transfer function, N/V
S	Secondary path transfer function
V_c	Control voltage, V
W	Transfer function of adaptive filter
$w(n)$	Impulse transfer function of adaptive filter
$x(n)$	Reference signal
x_c	Displacement of actuator runner, m
x_e	Displacement of engine in location of active mount in main direction, m
x_t	Displacement of fluid in inertia track, m
$y(n)$	Adaptive filter output, V
μ	Adaptation coefficient of adaptive filters
$\hat{\xi}(n)$	Estimated mean square error

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پایش ارتعاشات موتور خودرو با استفاده از نگهدارنده‌های فعال

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چکیده

در این پژوهش، پایش ارتعاشات موتور ملی (EF7) با استفاده از نگهدارنده‌های فعال، با در نظر گرفتن دو سامانهٔ نگهدارندهٔ متفاوت بررسی می‌شود. در سامانهٔ نخست، نگهدارنده‌ها مطابق طرح کنونی سامانهٔ نگهدارنده‌های موتور ملی قرار گرفته‌اند (سه نگهدارندهٔ غیرفعال)، با این تفاوت که نگهدارندهٔ روغنی موتور با نگهدارندهٔ فعال جایگزین شده است. سامانهٔ دیگر دارای چهار نگهدارنده شامل دو نگهدارندهٔ فعال و دو نگهدارندهٔ غیرفعال است. حرکت پویای موتور با ۶ درجه آزادی و اتصال یک طرفهٔ موتور شبیه‌سازی می‌شود. همچنین، شبیه‌سازی مناسبی برای نگهدارندهٔ فعال با عملگر برقی-آهنربایی انتخاب می‌گردد.

برای کاهش نیروی نگهدارنده‌ها به کفی از روش پایش تطبیقی پیشخوراند FXLMS (کمترین میانگین مربعات با حذف علامتهای مرجع) که یکی از متداول‌ترین روش‌های پایشی بکار رفته در حذف نوفه‌های صدا و ارتعاش است، استفاده می‌شود. این روش با توجه به ماهیت نیروی منتقل شونده به شاسی، در قالب روند نوار باریک برای حذف بسامد غالب ارتعاشات نامطلوب موتور استفاده می‌شود.

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