Design and simulation of an active controller to reduce undesirable vertical vibrations in the 4DOF model of the seated human body in the vehicle

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Abstract

To decrease and dampen the vibrations of the seated human bodies in a variety of settings like automobiles and public transportation, control theories and advanced controller design techniques are demanded. In this paper, we approach to design an active multivariable controller for a seated human body model, and results are derived and simulated via MATLAB. After examining the pros and cons of the previous models and strategies for the human body model, it has been decided to design an active controller based on pole placement analysis. This controller is designed for a lumped mass model of the human body with 4 degrees of freedom (4DOF) and with respect to the road excitations' presence. In the suggested multivariable control strategy, the vertical movements of the four parts of the body are controlled via manipulation of an input force in the vertical direction that is presumed to be applied by a piezoelectric actuator. The dynamic behavior of the system around its natural frequencies is measured and the usefulness of the actuators is investigated.

Keywords: *Four degrees of freedom model of the human body; Vertical vibration; Active controller; Pole placement; Vibration suppression*

I. INTRODUCTION

Vehicles such as cars, ships, and airplanes cause vibrations when they are in motion, which can upset and even injure the passengers, [1] especially when the vertical vibration frequency falls to 4 to 8 Hz [2]. Researches' results show that these vibrations have adverse effects on the human mind. It is observed that this mental fatigue plays an important role in how a person drives and the risk of accidents [3]. An attenuator, therefore, seems necessary for these vibrations.

Simulating the passengers' situation, would shorten the work processes, save costs, and make the design easier. The sitting position of the human body, which is exposed to vibrations, is a very complex dynamic system whose dynamic characteristics are different for each person. Several biodynamic models have been proposed to study vibrations of the body, which are generally divided into two categories: lumped models and distributed parameter models.

In the distributed parameters method, the body is divided into a large number of small parts and this method is very accurate, but it is very complex.

In the lumped mass method, the body is modeled through differential equations and parameters such as the mass, spring, and damper. The mass model is used to predict body behavior under various stimuli. The advantages of this method are the ease of the production and analysis of the model and its low cost. Usually, this method is used to examine the stimulation in only one direction. In this article, a 4 degree of freedom model based on the lumped mass method is used to model the body [4].

To attenuate the vibrations, efficient and reliable control strategies are needed. In these control systems, conversion of electrical energy into mechanical energy and vice versa is required. So, piezoelectric materials are used simultaneously as sensors and actuators in intelligent systems. [5]

In this paper, a control system has been designed to suppress unwanted vertical vibrations of a seated passenger. The pole placement method has been used in this research because it is easier for a multivariate model, selected in the absence of nonlinear dynamic equations, to use a more straightforward and linear system such as pole placement. The dynamic behavior of the system around its natural frequencies is measured and the designed controller is tested.

II. 4DOF DYNAMIC MODEL OF THE SEATED HUMAN BODY

Fig.1 shows a 4DOF model with 18 parameters of a seated passenger. The body is modeled as a mass-spring-damper system in which the parts are considered as lumped masses that are internally connected by the springs and dampers. The realistic parameters of the system for simulation investigation are provided in Table 1. The lumped masses do not have length and the excitation from the road is transmitted to the body through the seat[6]. Mass 1 represents the human head, mass 2 and 3 represent the chest and upper limbs that are indirectly stimulated by the chair, and mass 4 represents the pelvis and limbs that are directly stimulated by the chair.

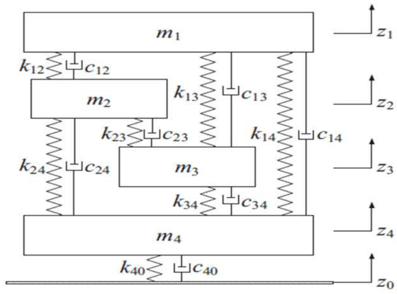

Fig. 1. *Four degrees of freedom model of the seated human body [4]*

TABLE I. DESCRIPTIONS AND AMOUNTS OF THE PARAMETERS IN FIGURE1 (SOME PARAMETERS ARE ADOPTED FROM [4])

Parameter	***Description***	***Amount(units)***
m1	Mass1	6.342 (Kg)
m2	Mass2	16.345 (Kg)
m3	Mass3	8.754 (Kg)
m4	Mass4	24.056 (Kg)
k12	Spring between mass 1 and 2	33158 (N/m)
k13	Spring between mass 1 and 3	20301 (N/m)
k14	Spring between mass 1 and 4	13245 (N/m)
k23	Spring between mass 2 and 3	21182 (N/m)
k24	Spring between mass 2 and 4	11971 (N/m)
k34	Spring between mass 3 and 4	6084 (N/m)
k40	Spring between mass 0 and 4	54751 (N/m)
c12	Damper between mass 1 and 2	1183.0(N.s/m)
c13	Damper between mass 1 and 3	1164.0(N.s/m)
c14	Damper between mass 1 and 4	1604.7(N.s/m)
c23	Damper between mass 2 and 3	1022.6(N.s/m)
c24	Damper between mass 2 and 4	2905.7(N.s/m)
c34	Damper between mass 3 and 4	2533.9(N.s/m)
c40	Damper between mass 0 and 4	1858.8(N.s/m)
z1	Mass1 movement	-
z2	Mass2 movement	-
z3	Mass3 movement	-
z4	Mass4 movement	-
z0	Road excitation	0.1.sin(ω.t)

As can be seen in Table 1, the excitation from the side of the road (surface 0) is considered to be sinusoidal with an amplitude of 10 cm. The spring and damper between member 4 and surface 0 are essentially representatives of car seats.

III. VIBRATION RESPONSE OF THE HUMAN BODY MODEL WITHOUT CONTROLLER

In this section, the relationships governing each mass are described and the behavior of the system is modeled in the Simulink environment. The equation governing each member is as follows:

$$m_1\ddot{z}_1 + \dot{z}_1(c_{12} + c_{13} + c_{14}) - (\dot{z}_2c_{12} + \dot{z}_3c_{13} + \dot{z}_4c_{14}) + z_1(k_{12} + k_{13} + k_{14}) - (z_2k_{12} + z_3k_{13} + z_4k_{14}) = 0 \quad (1)$$

$$m_2\ddot{z}_2 + \dot{z}_2(c_{12} + c_{23} + c_{24}) - (\dot{z}_1c_{12} + \dot{z}_3c_{23} + \dot{z}_4c_{24}) + z_2(k_{12} + k_{23} + k_{24}) - (z_1k_{12} + z_3k_{23} + z_4k_{24}) = 0 \quad (2)$$

$$m_3\ddot{z}_3 + \dot{z}_3(c_{13} + c_{23} + c_{34}) - (\dot{z}_1c_{13} + \dot{z}_2c_{23} + \dot{z}_4c_{34}) + z_3(k_{13} + k_{23} + k_{34}) - (z_1k_{13} + z_2k_{23} + z_4k_{34}) = 0 \quad (3)$$

$$m_4\ddot{z}_4 + \dot{z}_4(c_{14} + c_{24} + c_{34} + c_{40}) - (\dot{z}_1c_{14} + \dot{z}_2c_{24} + \dot{z}_3c_{34} + \dot{z}_0c_{40}) + z_4(k_{14} + k_{24} + k_{34} + k_{40}) - (z_1k_{14} + z_2k_{24} + z_3k_{34} + z_0k_{40}) = 0 \quad (4)$$

$$Mz^{\cdot\cdot} + Cz^{\cdot} + Kz = f_z \quad (5)$$

Due to the importance of the resonant conditions for obtaining the system's worst conditions, it is needed to study the behavior of the system around its natural frequencies. By solving $|K - M\omega^2| = 0$, the natural frequencies of the system are obtained as follows:

$$\omega_n = [23.6 \quad 62.7 \quad 81.4 \quad 111.7](\text{rad/s}) \quad (6)$$

The dynamic system is simulated in Simulink and to examine its resonant behavior, the road excitations sine wave frequency is set equal to the natural frequencies of the system and the vibration amplitude of each member is examined. The information of this study is given in Table 2. As it is indicated in Table 2, the worst case occurred around the first natural frequency. The vibration behavior of each mass around the first natural frequency is shown in Figs. 2 to 5.

TABLE II. THE RESONANT VIBRATION AMPLITUDES WITHOUT A CONTROLLER AT NATURAL FREQUENCIES

Frequency(rad/s)	***Mass1***	***Mass2***	***Mass3***	***Mass4***
23.6	13.76 cm	13.86 cm	14.01 cm	13.96 cm
62.7	7.9 cm	7.79 cm	7.86 cm	7.84 cm
81.4	6.7 cm	6.53 cm	6.57 cm	6.57 cm
111.7	5.4 cm	5.15 cm	5.12 cm	5.17 cm

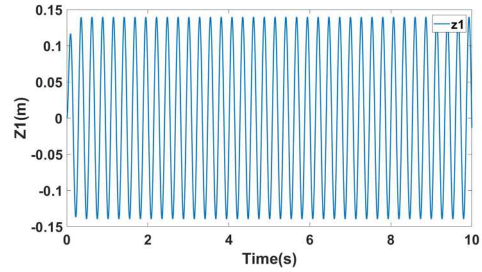

Fig. 2. *Mass1 vibration without controller at the first natural frequency*

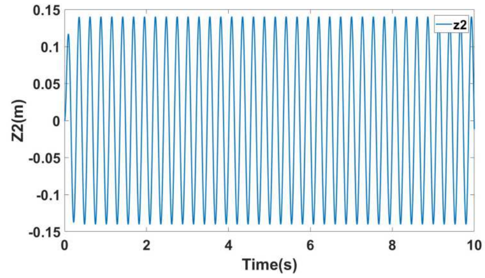

Fig. 3. *Mass2 vibration without controller at the first natural frequency*

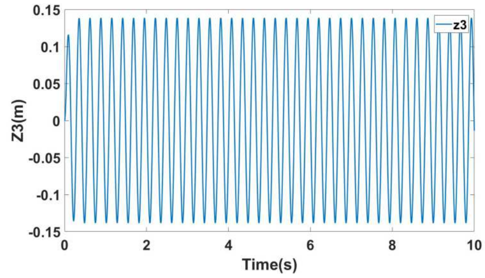

Fig. 4. *Mass3 vibration without controller at the first natural frequency*

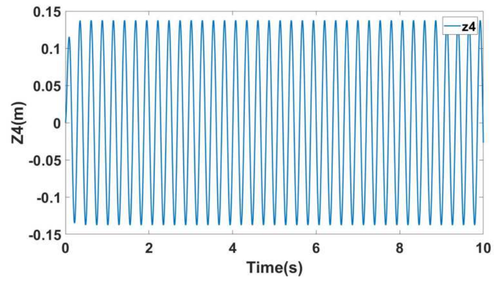

Fig. 5. *Mass4 vibration without controller at the first natural frequency*

IV. STATE-SPACE CONFIGURATION OF THE HUMAN BODY MODEL

To design the controller by the method of pole placement, it is necessary to remodel the system in the state space configuration. State variables are introduced as below:

$$X_1 = Z_1 . X_2 = \dot{Z}_1 . X_3 = Z_2 . X_4 = \dot{Z}_2 . X_5 = Z_3 . X_6 = \dot{Z}_3 . X_7 = Z_4 . X_8 = \dot{Z}_4 \quad (7)$$

The state-space equations are described as:

$$Y = CX + Du \quad , \quad \dot{X} = \text{AX} + \text{Bu} \quad (8)$$

In equation 13, the input or u is $c_{40}\dot{Z}_0 + k_{40}Z_0$ where $Z_0 = 0.1\sin(\omega t)$ is defined. According to these explanations and equations, and according to Equation 13, the matrices are found from the below equations:

$$\dot{X}_1 = X_2 \; \dot{X}_3 = X_4 \; \dot{X}_5 = X_6 \; \dot{X}_7 = X_8 \quad (9)$$

$$\dot{X}_2 = \frac{-1}{m_1}((c_{12} + c_{13} + c_{14})X_2 - c_{12}X_4 - c_{13}X_6 - c_{14}X_8 + (k_{12} + k_{13} + k_{14})X_1 - k_{12}X_3 - k_{13}X_5 - k_{14}X_7) \quad (10)$$

$$\dot{X}_4 = \frac{-1}{m_2}((c_{12} + c_{23} + c_{24})X_4 - c_{12}X_2 - c_{23}X_6 - c_{24}X_8 + (k_{12} + k_{23} + k_{24})X_3 - k_{12}X_1 - k_{23}X_5 - k_{24}X_7) \quad (11)$$

$$\dot{X}_6 = \frac{-1}{m_3}((c_{13} + c_{23} + c_{34})X_6 - c_{13}X_2 - c_{23}X_4 - c_{34}X_8 + (k_{13} + k_{23} + k_{34})X_5 - k_{13}X_1 - k_{23}X_3 - k_{34}X_7) \quad (12)$$

$$\dot{X}_8 = \frac{-1}{m_4}((c_{14} + c_{24} + c_{34} + c_{40})X_8 - c_{14}X_2 - c_{24}X_4 - c_{34}X_6 + (k_{14} + k_{24} + k_{34} + k_{40})X_7 - k_{14}X_1 - k_{24}X_3 - k_{34}X_5) \quad (13)$$

To evaluate the system response, as an example, the response at the first resonance condition, i.e., the worst case related to the first natural frequency of 23.6 (rad / s), is studied. The vibration diagrams of the system are obtained and compared with those in the previous section. Results demonstrate the exact construction of the state space model. For the sake of brevity, responses of the state space model are not presented again.

V. CONTROLLER DESIGN BASED ON POLE PLACEMENT METHOD IN MULTIVARIABLE PROBLEM

To design the controller, first, the controllability of the dynamic system must be evaluated. For a linear system (LTI), the entire system is controllable only if the controllability matrix (S) has a complete rank of order n.. The controllability matrix is obtained as [7][8]:

$$S = [B \; AB \; \ldots \; A^{n-1}B] \quad (14)$$

The rank of the S is 8 and it is full rank; so the system is completely controllable. Poles of the system without controller are calculated through the following characteristic equation:

$$|\lambda I - A| = 0 \quad (15)$$

According to the above relation, poles λ are found as:

$$\lambda_{1-8} = [\,-731.7 \; -576.8 \; -423.5 \; -18.7 \; -12.7 \; -4.3 \; 0 \; 0\,] \quad (16)$$

To design the controller, we consider the actuator force below level zero (F), as the controller input.

For this system, equation 1 to 5 is also valid and equation 17 would be added to them. Also, equations 9 to 12 are established in the state space, and only equation 13 changes as follows:

$$c_{40}(\dot{z}_0 - \dot{z}_4) + k_{40}(z_0 - z_4) = F \quad (17)$$

$$\dot{X}_8 = \frac{-1}{m_4}((c_{14} + c_{24} + c_{34})X_8 - c_{14}X_2 - c_{24}X_4 - c_{34}X_6 + (k_{14} + k_{24} + k_{34})X_7 - k_{14}X_1 - k_{24}X_3 - k_{34}X_5 - F) \quad (18)$$

Obviously, for the state space matrices of the new system, matrices B, C, D would remain the same, and matrix A changes slightly. Equation 19 is also added to the set of Equations 8 and then Equation 8 can be rewritten as follows:

$$u = F = -KX \quad (19)$$

$$\dot{X} = (A - BK)X \quad (20)$$

. In equation 20, matrix K is the state feedback coefficient that is used to adjust the desired location of the system poles and acts as a controller. The characteristic equation of the new closed-loop control system is as follows. The roots of this equation must be at the location of the desired poles, and the K matrix is designed accordingly.

$$|sI - A + BK| = 0 \quad (21)$$

According to the above relations, the matrix K can be obtained by having the desired poles. To obtain the desired poles, a trial and error process is required to reach the desired states and to keep the system stable. After designing a servo system, we select the desired poles with trial and error and K would be obtained.

The system does have two polarities at origin (zero), so a type 1 servo system should be used in a situation where the plant has an integrator. The block diagram of this servo system is shown in Fig. 6. It is assumed t> 0 and the system is completely controllable, which are the correct assumptions and would apply to our system.

According to the designed servo system and the trial and error process, the matrix of the desired poles (m) and the gain matrix (K) are obtained as follows:

m = [-731.71 -576.81 -423.57 -18.76 -12.7 -4.31 -0.2 -0.1] (22)

K=[0.0416 1.9025 0.4048 4.9037 0.1755 2.6262 0.4881 7.2168] (23)

As can be seen, the dominant poles of the first system have been changed by trial and error to have a stable and not noisy answer with reasonable control effort, and the distant and non-dominant poles of the system have been kept constant.

VI. CONTROLLER EFFECT IN VIBRATION REDUCTION OF THE HUMAN BODY MODEL, RESULTS & DISCUSSION

At the first natural frequency, where the worst case of resonant occurs, the output diagrams of the vibrations of mass 1 to 4 with the designed controller are shown through Figs. 8 to 10. The two conditions of with and without controller can be compared to see the efficiency and performance of the designed controller.

As can be seen in Tables 2 and 3, the vibration amplitude of the masses are greatly reduced with the controller. The question that may arise is about the applied force of the piezoelectric actuator. Given that the input of our system, u, is the same as the force of action, according to Equation 19, we can multiply any state variable in the corresponding gain from the K and sum of these values is equal to the applied force. Fig. 11 shows the applied force of the actuator at the system's first natural frequency.

Due to this much reduction in the amplitude of the masses' vibrations, the force is as expected and is not an abnormal.. In Table 3, the results are presented in more detail for the four resonant conditions.

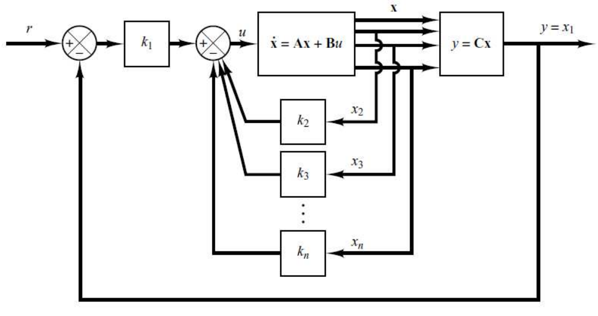

Fig. 6. *Type 1 servo system Block diagram when the plant has integrator[8]*

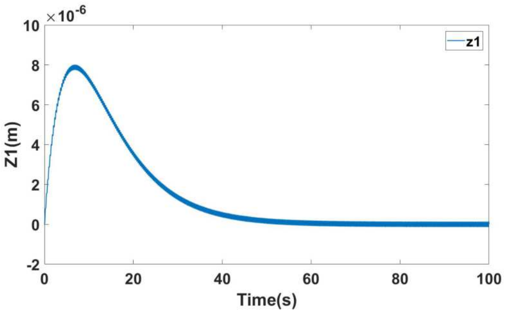

Fig. 7. *Mass 1 vibration with controller under the first resonant condition*

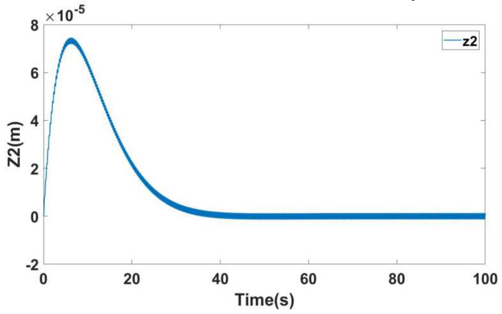

Fig. 8. *Mass 2 vibration with controller under the first resonant condition*

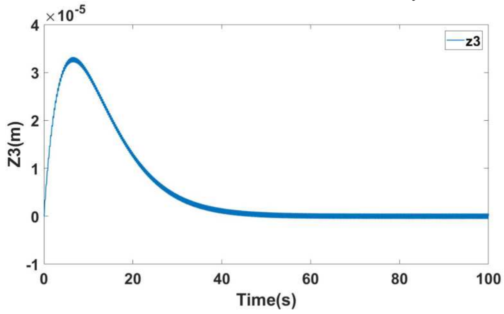

Fig. 9. *Mass 3 vibration with controller under the first resonant condition*

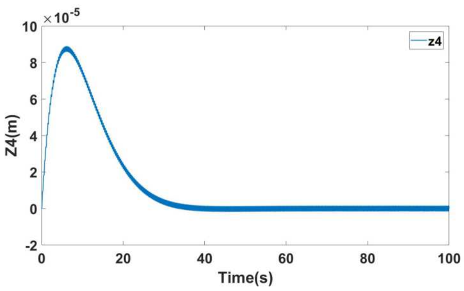

Fig. 10. *Mass 4 vibration with controller under the first resonant condition*

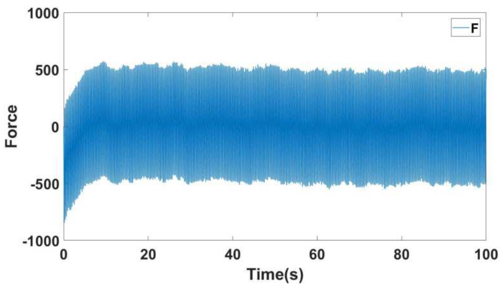

Fig. 11. *Actuator's force under the first resonant condition*

TABLE III. THE VIBRATION AMPLITUDES AND FORCES WITH CONTROLLER AT THE FIRST FOUR NATURAL FREQUENCIES (UNDER THE FIRST FOUR RESONANT CONDITIONS)

Frequencies (rad/s)	***Mass1 (peak) (mm)***	***Mass2 (peak) (mm)***	***Mass3 (peak) (mm)***	***Mass4 (peak) (mm)***	***Steady State Force***	***Max Force***
23.6	0.008	0.075	0.033	0.089	530N	846N
62.7	0.003	0.027	0.012	0.033	382 N	484 N
81.4	0.003	0.021	0.009	0.026	402 N	438 N
111.7	0.002	0.016	0.007	0.019	330 N	403 N

As it is observed, the vibrations are generally well attenuated and the forces are reasonable and available. At present, according to the trial and error method for selecting the desired poles, provided that all 4 objects have stable and desirable behavior, the designed controller is effective, while the most optimal conditions are achieved and realizable.

VII. CONCLUSIONS

This research examines the strategy of using an active controller to attenuate the vibrations transmitted from the road to the human body while driving. The nonlinear dynamics of the body model were considered as a linear model. After taking the problem into state space and using the related mathematical tools, the active controller was designed based on the pole placement approach.

In the target strategy, 4 output variables were controlled by an appropriate input variable.. The input variable is assumed to be applied by an actuator. By comparing the results with and without the controller in different situations, a significant reduction in the amplitude of system vibrations under resonance conditions was observed in resonant and non-resonant conditions.

In this study, the focus was on controller design based on control rules, not its application. In future studies, determining the actuator model and related items will be prioritized to complete the evaluation of the designed controller efficiency, and also further confirmation and practical tests are required.

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