Investigation on Parameters of Automotive Electromagnetic Active Suspensions

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Abstract–This paper presents the model of the electromagnetic active suspension consisting of an electromagnetic actuator and a mechanical spring. Furthermore, the authors investigate the effects of two crucial parameters, which are the spring stiffness and actuator force, on performances of the suspension. The study shows that the large spring stiffness results in the small initial displacement and requires the large actuator force, the small spring stiffness leads to the large initial displacement and needs the small force, and the large actuator force improves steady-state and dynamic performances and results in large volume, more weight and high cost of the actuator. Thus, this paper is considerable valuable for better designs of the electromagnetic active suspension and the actuator.

Keywords–Active suspensions, linear actuators, spring stiffness, vehicles.

I. Introduction

For the automotive industry, a major consideration is safety of vehicles and passengers. Hence, the main objective of automotive suspension systems is to isolate the vehicle body from road irregularities in order to maximize passenger ride comfort, to produce continuous road-wheel contact, and to improve the handling quality of vehicle. The ideal automotive suspension systems would rapidly independently absorb road shocks and would slowly return to its normal position for maintaining optimal tire-to-road contact. However, this is difficult to be achieved by using the passive suspension systems, where a soft spring allows for too much movement and a hard spring causes passenger discomfort due to road irregularities [1]-[3].

Significant improvement in suspension performance is achieved by the electromagnetic active suspension systems, which can generate control forces to absorb road shocks rapidly, suppress the roll and pitch motions, maintain the vehicle at a horizontal level, and ameliorate both safety and comfort. Hence, research and development of the electromagnetic active suspension systems is needed for conventional internal combustion engine vehicles and electrical vehicles.

A typical electromagnetic active suspension consists of an electromagnetic actuator and a mechanical spring [1]-[5]. Both are in parallel. Fig. 1 shows the schematic diagram of the single-wheel electromagnetic active suspension, where m_s represents the quarter of sprung vehicle mass, F_s represents the spring, and F_A represents the electromagnetic actuator. An automotive active suspension system includes four single-wheel active suspensions. Such an electromagnetic active suspension possesses the following features: (a) performance of the suspension is contributed by both the spring and the actuator, (b) the actuator force must be upward if the sprung mass is required to move upward, (c) the actuator force must be downward if the sprung mass needs to move downward, and (d) the spring still provides the basic performance of the suspension but the suspension performance will be debased if the actuator fails.

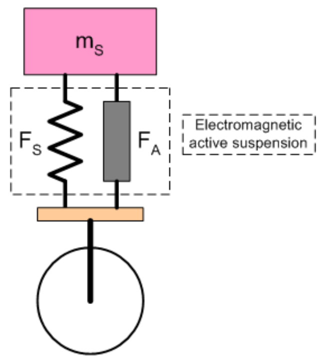

Fig. 1: Schematic diagram of a single-wheel active suspension

For the single-wheel active suspension, the spring stiffness, the actuator force, and the displacement are three crucial parameters and the design of the active suspension depends on them. Consequently, investigation on the effects of the spring stiffness, the actuator force, and the displacement on the active suspension is valuable for the design of the electromagnetic active suspension. This paper is focused on this issue.

II. Models of Electromagnetic Active Suspensions

1. Steady-State Model

1.1. Initial position (x_0)

At steady-state, the sprung mass is still. The initial position (x_0) means that there are only two forces on the sprung mass, which are the spring force and the weight of the sprung mass. At this position, the actuator force is zero. The spring has an initial displacement (Δx_0) due to the weight of the sprung mass. The steady-state model at the initial position is illustrated in Fig. 2.

The model can be described by the equations, given as

$$F_s - F_g = 0 \tag{1}$$

$$F_s = -k_s x_0 \tag{2}$$

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$$F_g = m_s g \tag{3}$$

$$\Delta x_0 = x_0 \tag{4}$$

where F_g represents the weight of the sprung mass, k_s represents the spring stiffness, and g represents the weight acceleration.

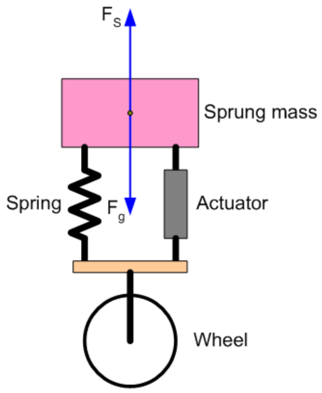


Fig. 2: Steady-state model at the initial position (x_0)

1.2. Position (x_1) where upward moving by Δx ($\Delta x > 0$)

In this case, there are three forces on the sprung mass and the sprung mass moves upward by Δx, from x_0 to x_1. The displacement is Δx. The model of the suspension can be depicted in Fig. 3.

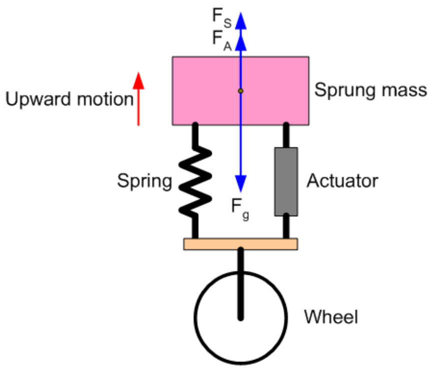


Fig. 3: Steady-state model at the position (x_1)

The steady-state model at the position (x_1) can be expressed as

$$F_s + F_A - F_g = 0 \tag{5}$$

$$F_s = -k_s x_1 \tag{6}$$

$$x_1 = x_0 + \Delta x \tag{7}$$

1.3. Position (x_2) where downward moving by Δx ($\Delta x < 0$)

For this position, the sprung mass moves downward by Δx ($\Delta x < 0$), from x_0 to x_2. The displacement is Δx. The steady-state model can be illustrated in Fig. 4. There are three forces on the sprung mass.

The steady-state model at the position (x_2) can be described as

$$F_s - F_g - F_A = 0 \tag{8}$$

$$F_s = -k_s x_2 \tag{9}$$

$$x_2 = x_0 + \Delta x \tag{10}$$

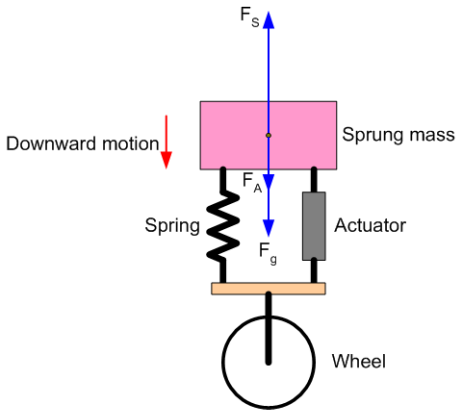


Fig. 4: Steady-state model at the position (x_2)

2. Dynamic Model

At the dynamic state, the resultant force on the sprung mass can be expressed as

$$F_R = \begin{cases} F_s + F_A - F_g & (upward \quad motion) \\ F_s - F_A - F_g & (downward \quad motion) \end{cases} \tag{11}$$

$$x = x_0 + \Delta x \tag{12}$$

$$F_s = -k_s x \tag{13}$$

where F_R denotes the resultant force on the sprung mass.

Consequently, the vertical acceleration can be computed as

$$a_v = \frac{F_R}{m_s} \tag{14}$$

where a_v denotes the vertical acceleration of the sprung mass.

Hence, the displacement equation can be given as

$$\Delta x = v_0 t + \frac{1}{2} a_{ave} t^2 \tag{15}$$

where t is the time, v_0 is the initial velocity when t is zero, and a_{ave} is the average acceleration (m/s^2) from x_0 to x.

If v_0 is assumed as 0, the estimated time from x_0 to x is calculated as

$$t = \sqrt{\frac{2|x - x_0|}{a_{ave}}} \tag{16}$$

The velocity equation is expressed as

$$v = v_0 + a_{ave} t \tag{17}$$

The average acceleration from x_0 to x is computed as

$$a_{ave}(x) = \frac{\sum_{k=1}^{N} a_v(x_k)}{N} \tag{18}$$

where N is the integer.

III. EFFECT ON STEADY-STATE PERFORMANCE

In this paper, it is assumed that the quarter of the sprung vehicle mass is 400 kg. At steady-state, the sprung mass is still and the resultant force on the sprung mass is zero. Using the proposed models in the last section, the steady-state performance can be computed. Fig. 5 show the change of the spring force and the actuator force with the displacement when the sprung mass moves upward, and Fig. 6 illustrates the actuator force and the spring force when the sprung mass moves downward. The initial displacement of the spring due to the sprung mass is shown in Fig. 7.

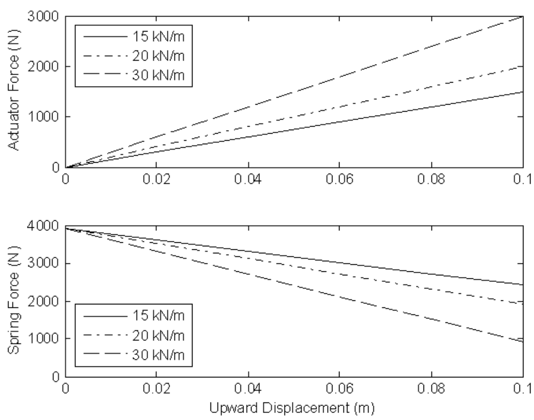


Fig. 5: Upward motion at steady-state

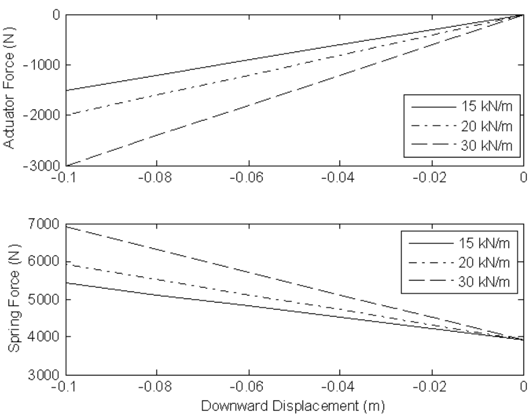


Fig. 6: Downward motion at steady-state

It can be seen from Fig. 5 and Fig. 6 that (i) the direction of the spring force is always upward due to the weight of the sprung mass, (ii) the actuator force is zero if the sprung mass is at the initial position since the spring force balances the weight of the sprung mass, (iii) the direction of the actuator force must be upward and the sum of the actuator force and the spring force balances the weight of the sprung mass if the sprung mass moves upward, (iv) the direction of the actuator force must be downward and the sum of the actuator force and the weight of the sprung mass balances the spring force if the sprung mass moves downward, (v) the large displacement of the sprung mass requires the large actuator force, (vi) the actuator force is always smaller than the weight of the sprung mass due to the spring force and hence the output power of the actuator can be saved, (vii) the large spring stiffness results in the large actuator force and the small initial displacement, (viii) the small spring stiffness results in the small actuator force and the large initial displacement, and (ix) large sprung mass results in large initial displacement.

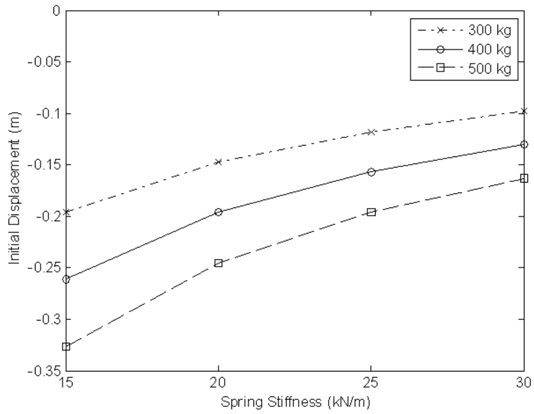


Fig. 7: Initial displacement of the spring due to the sprung mass

IV. EFFECT ON DYNAMIC PERFORMANCE

The vertical acceleration of the sprung mass is the key performance index of the dynamic performance of the active suspension. The bandwidth of the active suspension depends on it. Using the proposed dynamic model of the active suspension, the effect on the dynamic performance can be investigated. For the various actuator force values, the relationships between the vertical acceleration and the displacement are illustrated in Figs. 8-10 at three spring stiffness values, respectively.

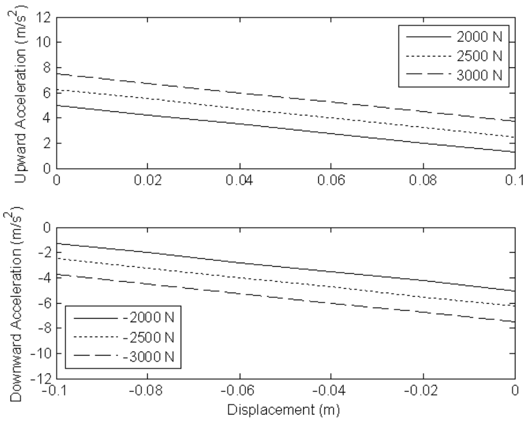


Fig. 8: Acceleration (spring stiffness = 15 kN/m)

It can be observed that (a) the acceleration has the maximum value at the initial position; (b) for the same actuator force, the acceleration has the same value at the initial position, regardless of the spring stiffness; (c) the increase in the displacement leads to the decrease in the acceleration; (d) for the same displacement, the large actuator force results in the large acceleration; and (e) for the same actuator force, the large spring stiffness gives rise to the small acceleration, except the initial position.

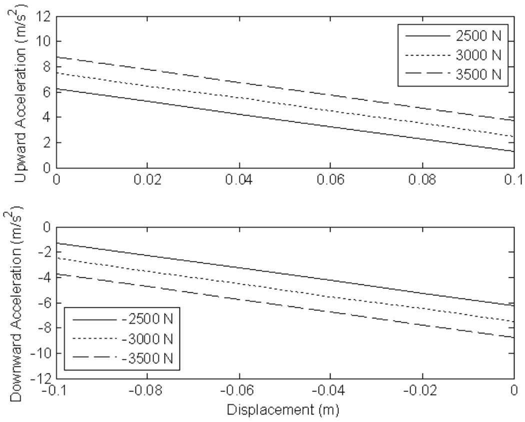


Fig. 9: Acceleration (spring stiffness = 20 kN/m)

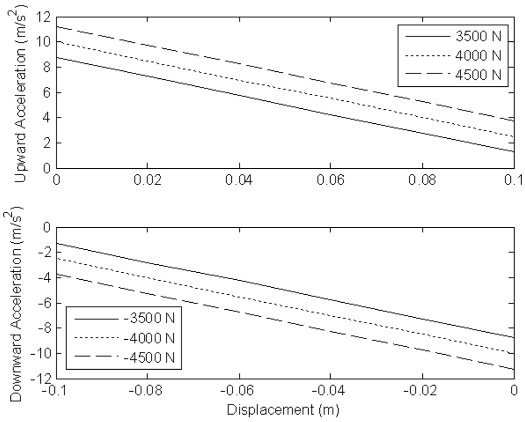
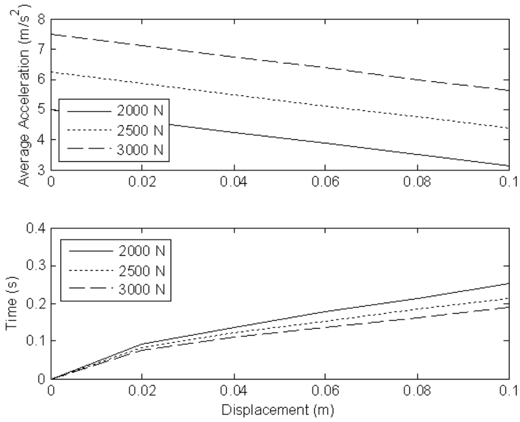
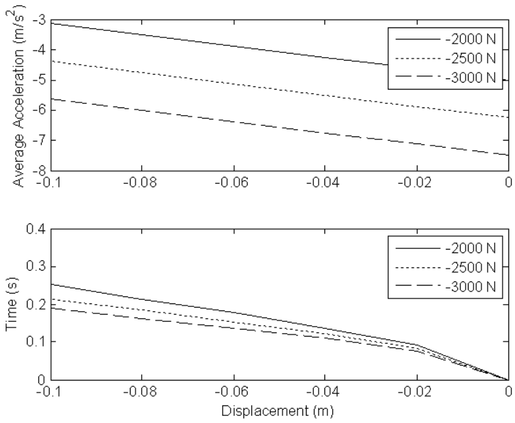


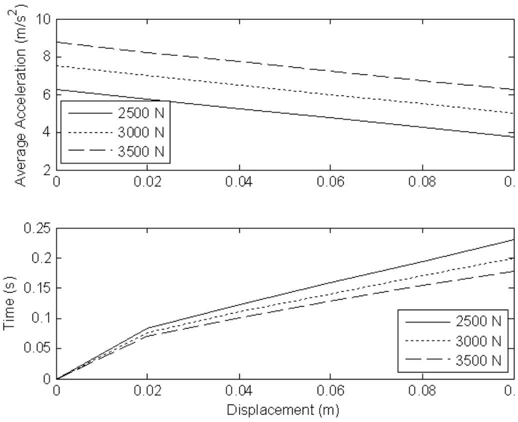
Fig. 10: Acceleration (spring stiffness = 30 kN/m)



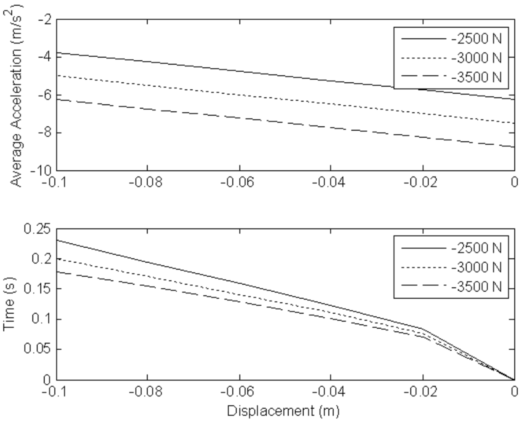
(a) Upward motion



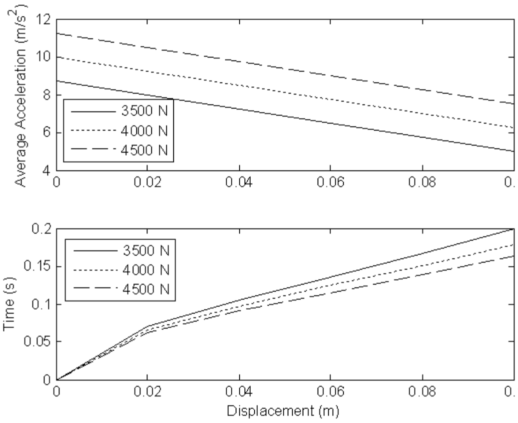
(b) Downward motion
Fig. 11: Spring stiffness = 15 kN/m



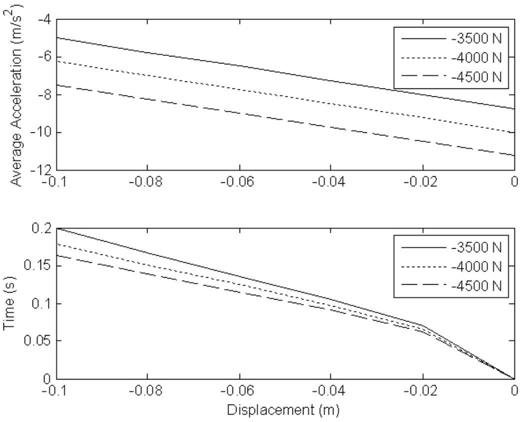
(a) Upward motion



(b) Downward motion
Fig. 12: Spring stiffness = 20 kN/m



(a) Upward motion



(b) Downward motion
Fig. 13: Spring stiffness = 30 kN/m

Figs. 11-13 show the average acceleration and the time taken by the displacement at different actuator force values and spring stiffness values.

It can be seen that (a) the large displacement leads to the small average acceleration and the long time at the same actuator force, (b) the large actuator force results in the large average acceleration and the short time at the same displacement, and (c) the small spring stiffness gives rise to the large average acceleration and the short time at the same displacement and the same actuator force.

V. CONCLUSION

The authors have proposed the steady-state and dynamic models of the electromagnetic active suspension. Using the models, the effects of the spring stiffness, the actuator force, and the sprung mass on the steady-state and dynamic performances can be investigated. The investigation in this paper shows that (a) on the one hand, the large actuator force can increase the displacement of the sprung mass and improve dynamic performance of the suspension; on the other hand, the large actuator force will result in the large volume and the high weight of the actuator; (b) the small spring stiffness requires the small actuator force and results in the large displacement of the sprung mass; (c) increase in the displacement leads to decrease in the acceleration; (d) the large spring stiffness gives rise to the small acceleration and the small displacement. Thus, this paper is considerable valuable for better designs of the electromagnetic active suspensions and the electromagnetic actuators.

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