EVS25
Shenzhen, China, Nov 5-9, 2010

Application of Linear Switched Reluctance Motor for Active Suspension System in Electric Vehicle

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Abstract

Electromagnetic active suspension system is considered to have improved stability and better dynamic response, compared to the hydraulic active suspension system. To investigate the influence of suspension parameters on system characteristics, the frequency response of quarter vehicle model is analyzed through Bode plots by varying the spring stiffness and damping coefficient. The sprung mass acceleration, suspension deflection and tire deflection are investigated respectively. This paper proposes a novel electromagnetic suspension system, comprising of a linear switched reluctance motor (LSRM) and a passive spring. The mechanical and electrical characteristics of the proposed linear motor are obtained and verified by using two-dimensional finite element method (FEM). The magnetic flux densities at specific translator positions are demonstrated. In order to study the feasibility and evaluate the performance of the proposed suspension system, a LQR optimal controller is developed and simulated with the quarter-vehicle model. The sprung mass acceleration, suspension deflection and related force applied by the actuator are investigated under different road disturbance. Both frequencies of disturbance are approximate to the suspension natural frequencies, which are the most severe working point of active suspension system. Simulation results demonstrate that good dynamic response and better ride comfort can be achieved by the proposed active suspension system.

Keywords: Active suspension system, Linear switched reluctance motor, LQR control

1 Introduction

The fundamental purpose of ground vehicle suspension system is to maintain continuous contact between the wheels and road surface, and to isolate passengers or cargo from the vibration induced by the road irregularities. These two purposes are responsible for the handling quality and ride comfort, respectively. However, these goals are generally contradictory. It is impossible for passive suspensions to achieve simultaneously a best performance of ride comfort and handling quality under all driving conditions.

In order to achieve better performance, active suspension systems have been proposed and applied over the past decades, as the development of industrial technology and control method. Currently, two types of active suspension are mainly used: hydraulic and electromagnetic. Hydraulic suspension systems offer higher force density, but have high system time constant. The limited bandwidth is insufficient for high frequency road irregularities. The shortcoming of hydraulic system can be overcome by the electromagnetic suspension systems. Better dynamic characteristics can be obtained by the electromagnetic actuator. Furthermore, it is more energy-efficient that no continuous power supply is required and the electromagnetic actuator can operate in both motoring and generating mode. The kinetic energy from road irregularities, which is dissipated to heat in dampers, can be regenerated and stored in the battery or applied for other devices in vehicle [1].

The electromagnetic suspension system comprises of an electromagnetic actuator and a mechanical spring. Several types of actuator have been proposed by earlier researchers and companies. Ismenio, et al. constructed a cylindrical linear actuator with axially magnetized NdFeB [2]. Bart, et al. proposed a slot-less brushless tubular permanent magnetic actuator [3], and Bose Corporation applied a multi-phase alternating current (AC) electric motor in their suspension system. However, the NdFeB magnet is quite costly which make this kind of suspension more expensive than other suspension systems. Furthermore, there is a big drawback in NdFeB magnet that it would lose magnetization around 150 ℃ . In view of this situation, a novel configuration of linear switched reluctance motor (LSRM) is proposed in this paper. The robust construction, low manufacturing and maintenance cost, less thermal problem, good fault tolerance capability and high reliability in harsh environments make it attractive alternative in the application of active suspension systems [4].

This paper is organized as follows. Section 2 presents the description of active suspension system, and analyzes the effect of system parameters. In section 3, a novel configuration of LSRM is proposed and the design is verified by finite element method (FEM). In section 4, the control methodology of the whole system is investigated, and the performances of proposed system are evaluated and compared with that of passive suspension systems by simulations. The conclusion is presented in section 5.

2 Suspension System Descriptions

Quarter-vehicle model is more extensively used to analyze and understand the influence of suspension parameters. It has simple structure with two degree-of-freedom in the vertical direction, which can be easily applied for the design and control of suspension systems. Although roll and pitch behaviors are eliminated in this kind of model, they can be simulated as external disturbance acting on the vehicle body.

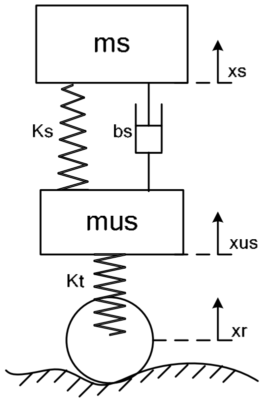


Figure 1: Passive suspension mode

2.1 Passive Suspension System

Conventional passive suspension, as shown in Figure 1, is modeled by a linear spring and damper. The spring is considered to support the sprung mass, which is valued at one fourth of the total vehicle body mass. The damper is used to dissipate the energy generated by the vibration. The tire is modeled as a spring of high stiffness without damping, which acting on both unsprung and sprung mass. The parameters of the quarter vehicle suspension are presented in Table 1.

Table 1: Suspension system parameters

Parameter	Value
Sprung mass	400kg
Unsprung mass	50kg
Spring stiffness	20000N/m
Damper coefficient	1000N/m/s
Tire stiffness	180000N/m

The dynamic motion can be represented by the following equations:

$$m_s\ddot{x}_s + k_s(x_s - x_{us}) + b_s(\dot{x}_s - \dot{x}_{us}) = 0 \qquad (1)$$

$$m_{us}\ddot{x}_{us} - k_s(x_s - x_{us}) - b_s(\dot{x}_s - \dot{x}_{us}) + k_t(x_{us} - x_r) = 0 \qquad (2)$$

where m_s and m_{us} are the sprung mass and unsprung mass, x_s and x_{us} are the displacements of respective masses, k_s and k_t are the spring stiffness, b_s is the damper coefficient and x_r represents the road disturbance.

2.2 Active Suspension System

The quarter vehicle model of active suspension system is shown in Figure 2. The hydraulic active suspension is modeled by a conventional passive suspension with an addition of active actuator between the sprung and unsprung masses [5], while in the electromagnetic active suspension system the passive damper is replaced by the active actuator. The damping effect of tire is also negligible in the active suspension model.

The dynamic equations of active suspensions are:

$$m_s \ddot{x}_s + k_s(x_s - x_{us}) + b_s(\dot{x}_s - \dot{x}_{us}) = u \tag{3}$$

$$m_{us}\ddot{x}_{us} - k_s(x_s - x_{us}) - b_s(\dot{x}_s - \dot{x}_{us}) + k_t(x_{us} - x_r) = -u \tag{4}$$

where u is the active force generated by the actuator. The damping coefficient is set to zero when analyzing the electromagnetic suspension.

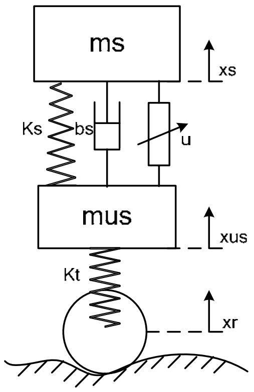


Figure 2: Active suspension model

Considering the following state variables

$x_1 = x_s - x_{us}$ Suspension deflection

$x_2 = \dot{x}_s$ Sprung mass velocity

$x_3 = x_{us} - x_r$ Tire deflection

$x_4 = \dot{x}_{us}$ Unsprung mass velocity

We can obtain the state space equation

$$\dot{X} = AX + Bu + L\dot{x}_r \tag{5}$$

$$\text{where } A = \begin{bmatrix} 0 & 1 & 0 & -1 \\ -\frac{k_s}{m_s} & -\frac{b_s}{m_s} & 0 & \frac{b_s}{m_s} \\ 0 & 0 & 0 & 1 \\ \frac{k_s}{m_{us}} & \frac{b_s}{m_{us}} & -\frac{k_t}{m_{us}} & -\frac{b_s}{m_{us}} \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & \frac{1}{m_s} & 0 & -\frac{1}{m_{us}} \end{bmatrix}^T,$$

and $L = \begin{bmatrix} 0 & 0 & -1 & 0 \end{bmatrix}^T$.

2.3 System Parameters Effect

From the dynamic equations, the open loop transfer functions are obtained. The effect of system parameters is investigated by looking at the bode plot. The transfer function from road vertical velocity to sprung mass acceleration, suspension deflection and tire deflection are:

$$\frac{\ddot{x}_s}{\dot{x}_r} = \frac{k_t s(b_s s + k_s)}{d} \tag{6}$$

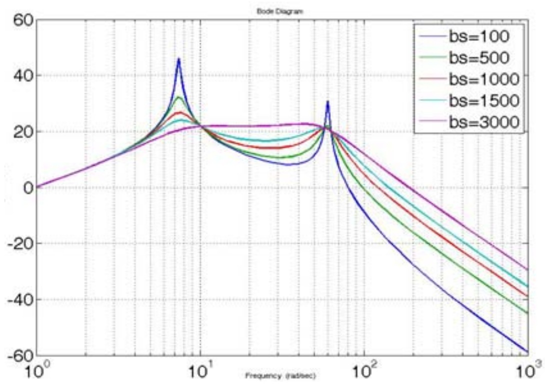
$$\frac{x_s - x_{us}}{\dot{x}_r} = -\frac{k_t m_s s}{d} \tag{7}$$

$$\frac{x_{us} - x_r}{\dot{x}_r} = -\frac{m_{us} m_s s^3 + (m_{us} + m_s) b_s s^2 + (m_{us} + m_s) k_s s}{d} \tag{8}$$

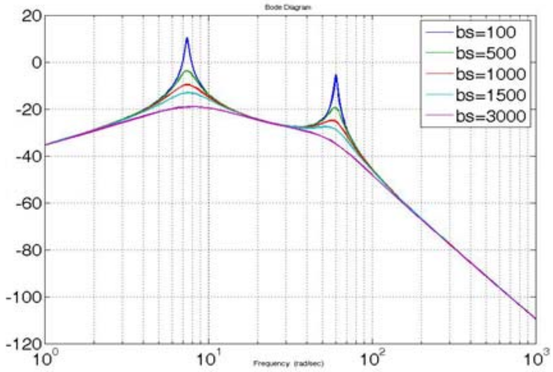
where d is the system characteristic polynomial.

$$d = m_{us} m_s s^4 + (m_{us} + m_s) b_s s^3 + [(m_{us} + m_s) k_s + m_s k_t] s^2 + b_s k_t s + k_s k_t \tag{9}$$

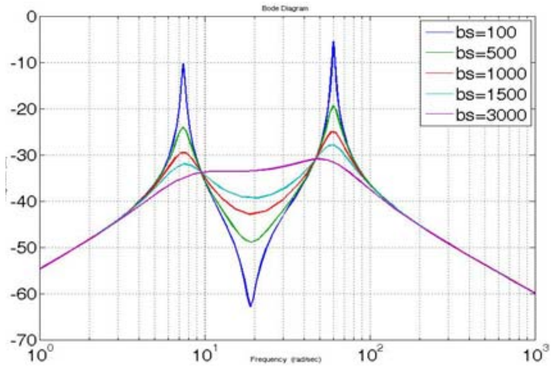
In order to demonstrate the effect of damper, the resultant Bode plots for five damping coefficients are shown in Figure 3. In each plot, two peaks occur at the body natural frequency and wheel natural frequency [6]. It can be seen from Figure 3(a), the sprung mass acceleration response is deteriorated at low frequencies as the suspension damping coefficient decrease. The suspension deflection is improved obviously around two natural frequencies with increased damping coefficient, as is shown in Figure 3(b). The effect on tire deflection is illustrated in Figure 3(c) that reduced tire deflection is obtained between the two natural frequencies with smaller damper, while the responses at two frequencies become worse.



(a) Acceleration response



(b) Suspension deflection response



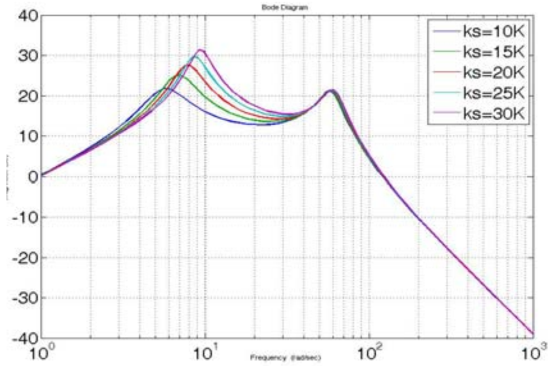
(c) Tire deflection response
Figure 3: Effect of damping coefficient

The effect of spring stiffness is examined in Figure 4, by comparing five curves of increasing stiffness. Figure 4(a) shows the response of sprung mass acceleration that the isolation of vibration is increasingly improved as the spring stiffness is decreased. However, the suspension deflection at low frequencies is becoming severe, as shown in Figure 4(b) and (c).

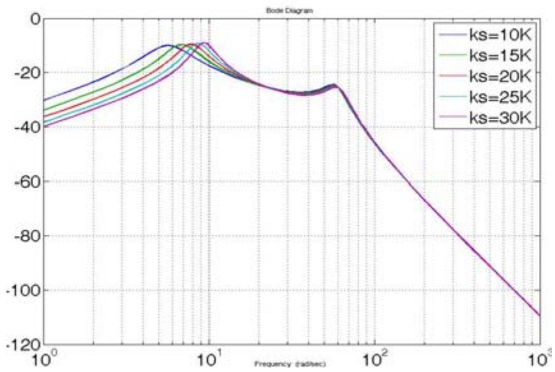
3 LSRM

3.1 System Specification

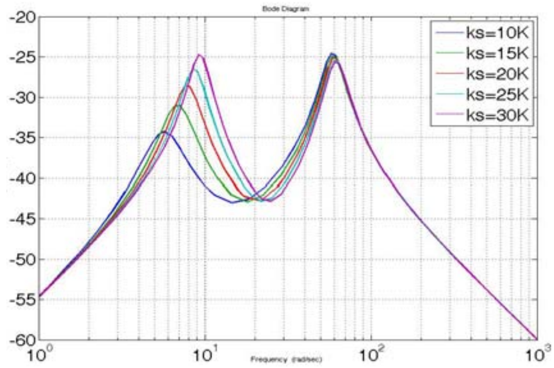
Before the design of LSRA for active suspension, there are several parameters need to be identified, such as peak and continuous force, maximum stroke and velocity. Each parameter has significant influence on the performance of active suspension. The required active force, stroke and velocity are mainly dependent on the vehicle body weight, road irregularities and the expected performance. The vibration magnitude of sprung mass should be controlled within an acceptable range for passenger comfort. Approximate indications that human react to the magnitudes of vibration are presented in ISO2631. The value varies with the duration and the type of activities under vibration by passengers.



(a) Acceleration response



(b) Suspension deflection response



(c) Tire deflection response
Figure 4: Effect of spring stiffness

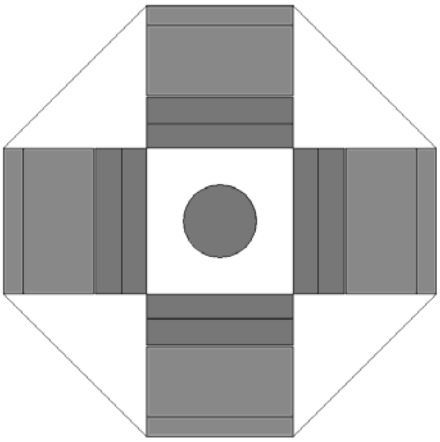
Maximum stroke is the available travel distance between the sprung and unsprung mass. The value is selected not only to meet the requirements for roll and pitch behavior, but also to absorb the road irregularities. Movement that exceeds the maximum stroke can lead to serious damage to the actuator and cause extreme uncomfortable feeling to passengers, thus, longer stroke is preferred to ensure the function and safety of the suspension. However, too much margin in stroke will extend the length and increase the weight of actuator. By considering the above design aspects, the specification of suspension systems are determined and listed in Table 2.

Table 2: Specification of suspension system

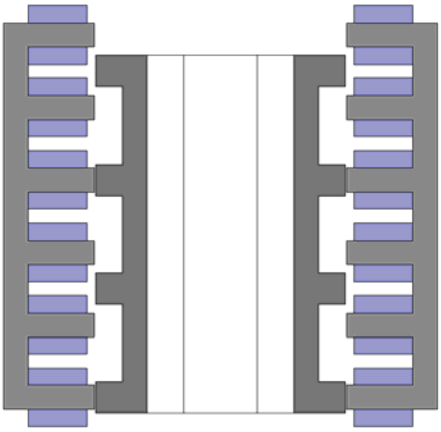
Specification	Value
Maximum Force	4000N
Continuous Force	1000N
Maximum Stroke	0.1m
Maximum speed	1m/s

3.2 LSRM Design

The proposed LSRM consists of four identical three-phase linear switched reluctance actuators, as shown in Figure 5. The stator and translator are laminated with silicon steel plates, and connected to the vehicle body and the wheel, respectively. In order to reduce the weight of translator, the phase windings are installed on the stator. Thus the translator is free of coils and permanent magnet that would not add too much weight on sprung mass. The windings of the same phase are connected in series. Only one converter is required, and it is relatively stationary to the stator. The mechanical parameters are listed in Table 3.



(a)



(b)

Figure 5: Configuration of TLSRA

Table 3: LSRM parameters

Specification	Value(mm)
Stator pole width	16.6
Stator slot width	33.4
Translator pole width	27.6
Translator slot width	47.4
Yoke thickness	16.6
Stator pole height	45
Translator pole height	18
Stack length	80
Air-gap	0.8

3.3 Design Verification

To verify the design of LSRM, two-dimensional finite element analysis (FEA) is used. Since the LSRM is composed of four identical modules, the FEA can be simplified to analyze only one module. Figure 6 shows the flux linkage versus current at different translator positions between unaligned and aligned positions. The force profile at different currents and positions is shown in Figure 7. The position of 0mm represents the unaligned position, and the force reaches the largest value at an intermediate position.

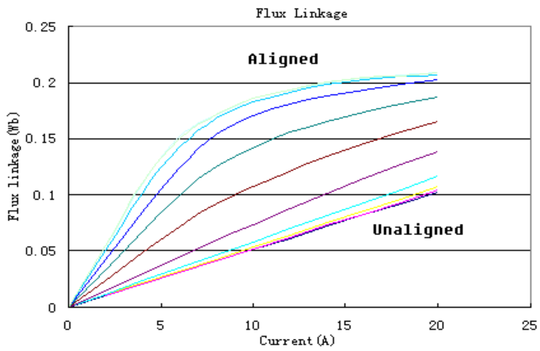


Figure 6: Flux linkage characteristics

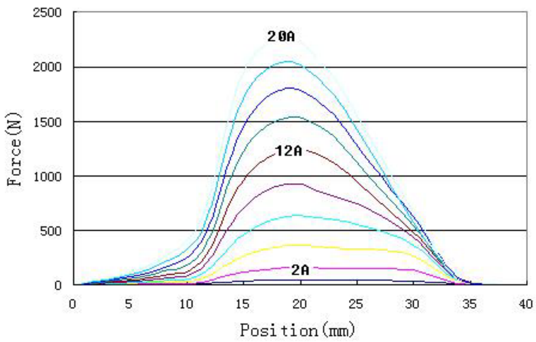
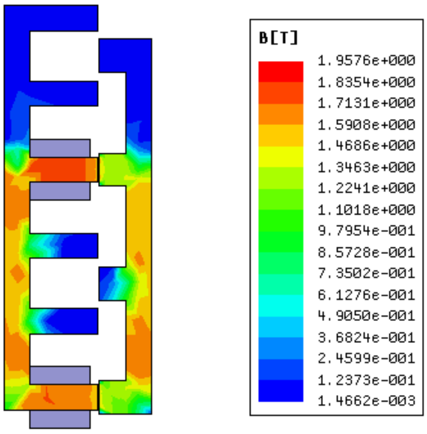


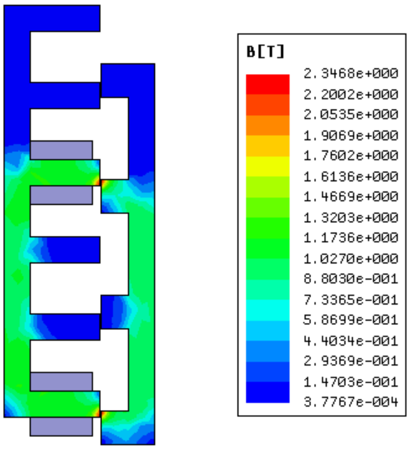
Figure 7: Force characteristics

Figure 8 shows the magnetic flux density distributions at position that generate largest force and the aligned positions. It can be seen from

Figure 8(b) that there is severe local saturation in both pole corner in the low overlap position.



(a)



(b)

Figure 8: Magnetic flux density

4 Controller Design

In order to study the feasibility and evaluate the performance of the proposed suspension system, a control scheme is developed and simulated with the whole system. The control scheme has an outer loop to track the position reference of sprung mass and an inner loop to trace the required force for actuator. The controller block diagram is shown in Figure 9. A LQR optimal controller is designed to obtain the required force. The LQR is a full state feedback controller with an aim to minimize a quadratic cost function [8].

Considering the system state space model in equation (5), the quadratic cost function can be defined as:

$$\min J = \frac{1}{2}\int_0^{\infty}\left(x^T(t)Qx(t)+u^T(t)Ru(t)\right)dt \qquad (10)$$

then the feedback control that minimizes the cost is

$$u=-KX \qquad (11)$$

where K is given by

$$K=R^{-1}B^T P(t) \qquad (12)$$

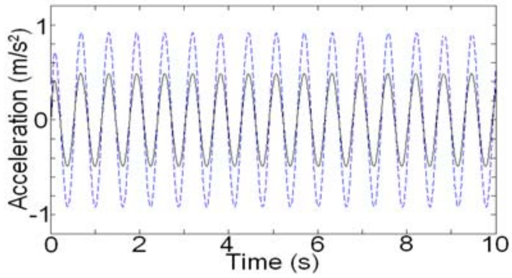
and P is found by solving the continuous time Riccati differential equation

$$A^T P(t)+P(t)A-P(t)BR^{-1}B^T P(t)+Q=-\dot{P}(t) \qquad (13)$$

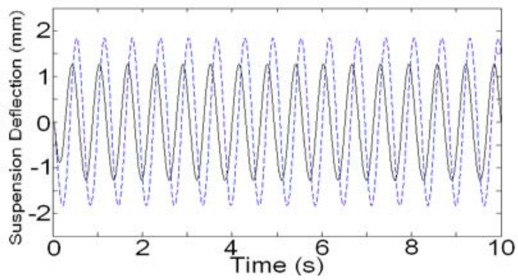
Since the mechanical time constant is much bigger than that of electrical current, the electromagnetic variables can be considered as constant when the mechanical variables are mainly discussed [9]. The electromagnetic force of LSRA can be approximately described as

$$F_k(x,i_k)=\frac{1}{2}\frac{dL_k}{dx}i_k^2 \qquad (14)$$

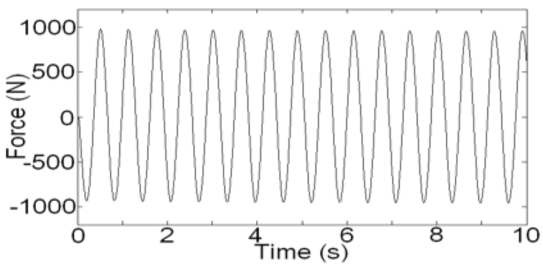
where F_k is the electromagnetic force generated by phase k, dL_k/dx is the inductance change rate of phase k.



(a) Sprung mass acceleration



(b)Suspension deflection



(c) Active force

Figure 10: System response at 10rad/s disturbance (Dashed line: passive, solid line: Active)

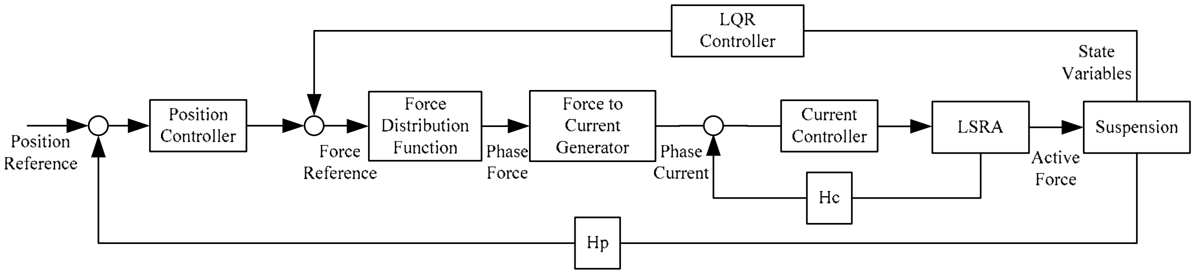


Figure: 9 Block diagram of suspension control

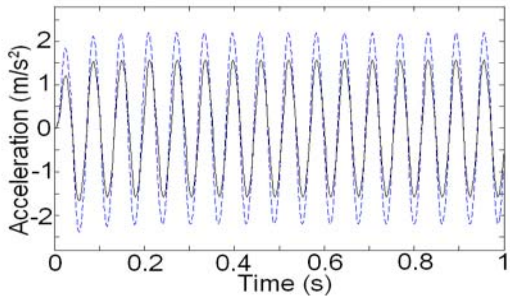
Based on this assumption, a winding excitation scheme for LSRA is followed to generate the active force. As shown in Figure 9, the scheme comprises a force distribution function (FDF) and force-current generations function. The FDF is applied to calculate the reference force for each phase at specific position, and the force-current generation function that derived from equation (14) is used to obtain the phase current reference.

Simulations are performed in the Matlab/Simulink environment with the system parameters shown in Table I. Responses of sprung mass and suspension deflection at 10rad/second and 100rad/second sinuous disturbance are demonstrated in Figure 10 and Figure 11, respectively. Both frequencies of disturbance are approximate to the suspension natural frequencies.

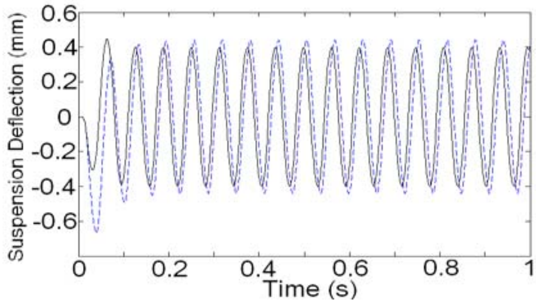
The corresponding control forces are shown to illustrate the feasibility of the actuator. The sprung mass accelerations are improved significantly compared to the passive suspension. However, under the optimal control, the suspension deflection can not be reduced considerably at the same time. It becomes even worse than passive suspension at low frequency disturbance.

5 Conclusion

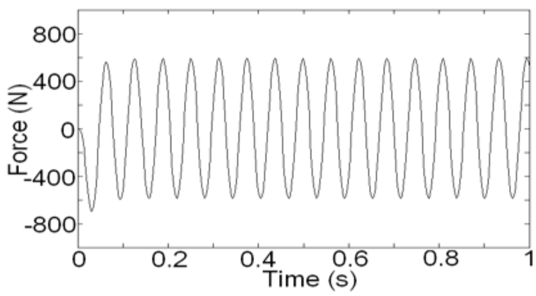
Improved ride quality and road holding capability are based on high performance suspension system. In this paper, the effects of suspension parameters on system performance are investigated, and an electromagnetic active suspension system that composed of linear switched reluctance actuator and mechanical spring is proposed. The design of LSRA is verified by two-dimensional FEA. Finally, a LQR optimal controller is developed and simulated with the quarter-vehicle model, in order to study the feasibility and evaluate the performance of the proposed suspension system. Enhanced performance is achieved by evaluating the response of sprung mass acceleration and suspension deflection.



(a) Sprung mass acceleration



(b) Suspension deflection



(c) Active force

Figure 11: System response at 100rad/s disturbance (Dashed line: passive, solid line: Active)

Acknowledgments

The authors grateful acknowledge the financial support of Innovation and Technology Fund of Hong Kong SAR under the project ITP/025/09AP

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