MODELING AND SIMULATION OF SKYHOOK CONTROLLER FOR ACTIVE SUSPENSION SYSTEM

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Report submitted in partial fulfillment of the requirements for the award of the degree of Bachelor of Mechanical Engineering with Automotive Engineering

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SUPERVISOR’S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive Engineering.

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STUDENT’S DECLARATION

I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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This paper presents modeling and simulation force tracking control of a hydraulic actuator applied in a quarter car of the active suspension system using skyhook control system. The controller structure of the active suspension system was decomposed into two loops namely outer loop and inner loop controllers. Outer loop controller is used to calculate the optimum target force to reject the effects of road disturbances by using skyhook control and proportional-integral-derivative (PID) control system, while, the inner loop controller is used to keep the actual force close to this desired force. The results of the study show that the inner loop controller is able to track well the target force ranging from sinusoidal, square, saw-tooth and step functions of target force. The performance of outer loop controller also shows significant improvement in terms of body acceleration, body displacement and tire displacement, and spring deflection as compared to the passive suspension system.
ABSTRAK

Tesis ini membentangkan pemodelan dan simulasi pengesan kawalan kuasa dari aktuator hidraulik yang diaplikasikan dalam satu perempat struktur model kereta suspensi aktif menggunakan sistem kawalan “skyhook”. Struktur kawalan dari sistem suspensi aktif dibahagikan kepada dua bahagian iaitu kawalan pusingan luar dan kawalan pusingan dalam. Kawalan pusingan luar digunakan untuk mengira kuasa optimum yang disasarkan untuk menyingkirkan kesan daripada gangguan permukaan jalan dengan menggunakan sistem kawalan “skyhook” dan juga sistem kawalan “proportional-integral-derivative (PID)”, sementara kawalan pusingan dalam digunakan untuk memastikan kuasa sebenar menghampiri kuasa yang dikehendaki. Keputusan yang diperolehi berdasarkan analisis mendapati, sistem kawalan dalam mempunyai kebolehan untuk mengesan kepelbagaian jenis kuasa disasarkan merangkumi fungsi “sinusoidal”, “square”, “saw-tooth”, dan “step”. Begitu juga dengan prestasi sistem kawalan luar menunjukkan penambahbaikan dalam aspek pecutan badan, pemindahan badan, pemindahan tayar, dan pesongan pegas seperti yang dibandingkan dengan sistem suspensi pasif.
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\( A_p \)  
piston area, \( \text{m}^2 \)

\( C_d \)  
discharge coefficient

\( C_s \)  
damping constant, \( \text{N.s/m} \)

\( C_{tm} \)  
leakage coefficient

\( F_a \)  
actuator force, \( \text{N} \)

\( K_s \)  
spring stiffness, \( \text{N/m} \)

\( K_t \)  
tire stiffness, \( \text{N/m} \)

\( M_s \)  
sprung mass, \( \text{kg} \)

\( M_u \)  
unsprung mass, \( \text{kg} \)

\( P_L \)  
pressure induced by load

\( P_s \)  
supply pressure, \( \text{kN/m} \)

\( Z_r \)  
road profile

\( Z_s \)  
sprung mass displacement

\( Z_u \)  
unsprung mass displacement

\( \dot{Z}_s \)  
vertical velocity of the body

\( \dot{Z}_u \)  
vertical velocity of the wheel

\( u_1 \)  
spool valve position

\( u_2 \)  
bypass valve area

\( \alpha \)  
hydraulic coefficient, \( \text{N/m}^5 \)

\( \rho \)  
specific gravity of hydraulic fluid
\( \tau \)  
\( \text{time constant} \)

\( w \)  
\( \text{spool valve width, m} \)
LIST OF ABBREVIATIONS

AC  Automatic Control
PI  Proportional Integral
PID  Proportional Integral Derivative
CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

A car suspension system is the mechanism that physically separates the car body from the wheels of the car. The performance of the suspension system has been greatly increased due to increasing vehicle capabilities. Appleyard and Wellstead (1995) have proposed several performance characteristics to be considered in order to achieve a good suspension system. These characteristics deal with the regulation of body movement, the regulation of suspension movement and the force distribution. Ideally the suspension should isolate the body from road disturbances and inertial disturbances associated with cornering and braking or acceleration. The suspension must also be able to minimize the vertical force transmitted to the passengers for their comfort. This could be achieved by minimizing the vertical car body acceleration.

The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system. A passive suspension system is a conventional suspension system consists of a non-controlled spring and shock-absorbing damper. The commercial vehicles today use passive suspension system as means to control the dynamics of a vehicle’s vertical motion as well as pitch and roll. Passive indicates that the suspension elements cannot supply energy to the suspension system. The suspension spring and damper do not provide energy to the suspension system and control only the motion of the car body and wheel by limiting the suspension velocity according to
the rate determined by the designer. Hence, the performance of a passive suspension system is variable subject to the road profiles.

The semi-active suspension has the same elements but the damper has two or more selectable damping rate. In early semi-active suspension system, the regulating of the damping force can be achieved by utilizing the controlled dampers under closed loop control, and such is only capable of dissipating energy (Williams, 1994). Two types of dampers are used in the semi-active suspension namely the two state dampers and the continuous variable dampers. The disadvantage of these dampers is difficulties to find devices that are capable in generating a high force at low velocities and a low force at high velocities, and be able to move rapidly between the two.

An active suspension is one in which the passive components are augmented by hydraulic actuators that supply additional force. Active suspensions differ from the conventional passive suspensions in their ability to inject energy into the system, as well as store and dissipate it. The active suspension is characterized by the hydraulic actuator that placed in parallel with the damper and the spring. Since the hydraulic actuator connects the unsprung mass to the body, it can control both the wheel hop motion as well as the body motion. Thus, the active suspension now can improve both the ride comfort and ride handling simultaneously.

Although various control laws have been proposed to control the active suspension system, the methods were successful applied in computer simulations based only but not in real applications. Therefore, a real active suspension system is needed to implement and test the developed control strategy. A quarter car models are chosen as an initial model of controlling the active suspension system due to the simplicity of the model. Modeling of the quarter car suspension as well as the non-linear hydraulic actuator including its force tracking controller for an active suspension system is investigated in this study.
1.2 PROBLEM STATEMENT

The statement of the problem of this project is expressed as follow:

To model and simulate skyhook controller for active suspension system that can improved car performance on various condition of road profile. For an active suspension hydraulic actuators is apply to supply additional force to the system. In this case, the control systems need to be develop and the force that needs to be injected to the hydraulic need to be determined.

1.3 OBJECTIVES

The objectives of this project are as follows:

i. To develop hydraulic model.
ii. To develop force tracking controller.
iii. To develop skyhook controller to an active quarter car suspension using hydraulic actuator.

1.4 PROJECT SCOPE

This project is about modeling and simulation of skyhook controller for active suspension system. The quarter car modeling was based on passive suspension system. MATLAB software is being used in purpose to develop program for modeling and analyzing the system controller created. In this software, block diagram will be form based on the required equation using SIMULINK and then being analyzed by giving variable target value to get the desired force match with the various input from road profiles.
CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system and/or control bandwidth (Appleyard and Wellstead, 1995). A passive suspension system is a conventional suspension system consists of a non-controlled spring and shock-absorbing damper as shown in Figure 2.1. The semi-active suspension has the same elements but the damper has two or more selectable damping rate as shown in Figure 2.2. An active suspension is one in which the passive components are augmented by actuators that supply additional force as shown in Figure 2.3.

2.2 PASSIVE SUSPENSION SYSTEM

The commercial vehicles today use passive suspension system to control the dynamics of a vehicle’s vertical motion as well as pitch and roll. Passive indicates that the suspension elements cannot supply energy to the suspension system. The passive suspension system controls the motion of the body and wheel by limiting their relative velocities to a rate that gives the desired ride characteristics. This is achieved by using some type of damping element placed between the body and the wheels of the vehicle, such as hydraulic shock absorber. Properties of the conventional shock absorber establish the tradeoff between minimizing the body vertical acceleration and maintaining good tire-road contact force. These parameters are coupled. That is, for a comfortable ride, it is desirable
to limit the body acceleration by using a soft absorber, but this allows more variation in the
tire-road contact force that in turn reduces the handling performance. Also, the suspension
travel, commonly called the suspension displacement, limits allowable deflection, which in
turn limits the amount of relative velocity of the absorber that can be permitted. By
comparison, it is desirable to reduce the relative velocity to improve handling by designing
a stiffer or higher rate shock absorber. This stiffness decreases the ride quality performance
at the same time increases the body acceleration, detract what is considered being good ride
characteristics.

An early design for automobile suspension systems focused on unconstrained
optimizations for passive suspension system which indicate the desirability of low
suspension stiffness, reduced unsprung mass, and an optimum damping ratio for the best
controllability (Thompson, 1971). Thus the passive suspension systems, which approach
optimal characteristics, had offered an attractive choice for a vehicle suspension system and
had been widely used for car. However, the suspension spring and damper do not provide
energy to the suspension system and control only the motion of the car body and wheel by
limiting the suspension velocity according to the rate determined by the designer. Hence,
the performance of a passive suspension system is variable subject to the road profiles.
2.3 SEMI-ACTIVE SUSPENSION SYSTEM

In early semi-active suspension system, the regulating of the damper force can be achieved by utilizing the controlled dampers under closed loop control, and such is only capable of dissipating energy (Williams, 1994). Two types of dampers are used in the semi-active suspension namely the two state dampers and the continuous variable dampers.

The two state dampers switched rapidly between states under closed-loop control. In order to damp the body motion, it is necessary to apply a force that is proportional to the body velocity. Therefore, when the body velocity is in the same direction as the damper velocity, the damper is switched to the high state. When the body velocity is in the opposite direction to the damper velocity, it is switched to the low state as the damper is transmitting the input force rather than dissipating energy. The disadvantage of this system is that while it controls the body frequencies effectively, the rapid switching, particularly when there are
high velocities across the dampers, generates high-frequency harmonics which makes the suspension feel harsh, and leads to the generation of unacceptable noise.

The continuous variable dampers have a characteristic that can be rapidly varied over a wide range. When the body velocity and damper velocity are in the same direction, the damper force is controlled to emulate the skyhook damper. When they are in the opposite directions, the damper is switched to its lower rate, this being the closest it can get to the ideal skyhook force. The disadvantage of the continuous variable damper is that it is difficult to find devices that are capable in generating a high force at low velocities and a low force at high velocities, and be able to move rapidly between the two.

![Semi-active suspension system](image)

**Figure 2.2:** The semi-active suspension system

Source: Williams (1994)

### 2.4 Active Suspension System

Active suspensions differ from the conventional passive suspensions in their ability to inject energy into the system, as well as store and dissipate it. (Crolla, 1988) has divided
the active suspensions into two categories; the low-bandwidth or soft active suspension and the high-bandwidth or stiff active suspension. Low bandwidth or soft active suspensions are characterized by an actuator that is in series with a damper and the spring as shown in Figure 2.3(a). Wheel hop motion is controlled passively by the damper, so that the active function of the suspension can be restricted to body motion. Therefore, such type of suspension can only improve the ride comfort. A high-bandwidth or stiff active suspension is characterized by an actuator placed in parallel with the damper and the spring as illustrated in Figure 2.3(b). Since the actuator connects the unsprung mass to the body, it can control both the wheel hop motion as well as the body motion. The high-bandwidth active suspension now can improve both the ride comfort and ride handling simultaneously. Therefore, almost all studies on the active suspension system utilized the high-bandwidth type. Various types of active suspension model are reported in the literature either modeled linearly (used most) or non-linear; examples are Macpherson strut suspension system (Al-Holou et al., 1999, Hong et al., 2002).

![Figure 2.3: Types of active suspension system](source: Crolla (1988))
2.5 **QUARTER CAR MODEL**

The vehicle model considered in this study is a quarter car models. The quarter car model for passive suspension system consists of one-fourth of the body mass, suspension components and one wheel as shown in Figure 2.4(a). The quarter car model for active suspension system, where the hydraulic actuator is installed in parallel with the spring, is shown in Figure 2.4(b).

The assumptions of a quarter car modeling are as follows: the tire is modeled as a linear spring without damping, there is no rotational motion in wheel and body, the behavior of spring and damper are linear, the tire is always in contact with the road surface and effect of friction is neglected so that the residual structural damping is not considered into vehicle modeling. The equations of motion for the sprung and unsprung masses of the passive quarter car model are given by:

![Diagram of Quarter Car Model](image)

**Figure 2.4:** The quarter car model
Whereas, the equations of motion for the sprung and unsprung masses of the active quarter-car model are given by:

\[ M_s \ddot{Z}_s + K_s (Z_s - Z_u) + C_s (\dot{Z}_s - \dot{Z}_u) = 0 \]
\[ M_u \ddot{Z}_u + K_t (Z_u - Z_r) + K_s (Z_u - Z_s) + C_s (\dot{Z}_u - \dot{Z}_s) = 0 \]  \hspace{1cm} (2.1)

\[ M_s \ddot{Z}_s + K_s (Z_s - Z_u) + C_s (\dot{Z}_s - \dot{Z}_u) + F_a = 0 \]
\[ M_u \ddot{Z}_u + K_t (Z_u - Z_r) + K_s (Z_u - Z_s) + C_s (\dot{Z}_u - \dot{Z}_s) - F_a = 0 \]  \hspace{1cm} (2.2)

Where,

\[ M_s = \text{sprung mass} \]
\[ K_s = \text{spring stiffness} \]
\[ K_t = \text{tire stiffness} \]
\[ M_u = \text{unsprung mass} \]
\[ C_s = \text{damping constant} \]
\[ Z_r = \text{road profile} \]
\[ Z_s = \text{sprung mass displacement} \]
\[ Z_u = \text{unsprung mass displacement} \]
\[ F_a = \text{actuator force} \]

Due to the tire stiffness, vertical force acting on the contact point between tire and the road will be created when the tire hits a certain road profile. Then, the vertical force is transferred to the wheel resulting in vertical acceleration of the wheel. Part of the vertical force is damped out by the suspension elements, whereas, the rest is transferred to the vehicle body via the suspension elements. The vehicle body will move vertically in response to the vertical force of the suspension elements. The performance criteria of the suspension system to be investigated in this study are body acceleration (\( \ddot{Z}_s \)), body displacement (\( Z_s \)), suspension working space (\( Z_u - Z_s \)) and wheel displacement (\( Z_u \)).