

A PRELIMINARY STUDY INTO THE APPLICATION OF ACTIVE FORCE CONTROL STRATEGY TO AN AUTOMOTIVE SUSPENSION SYSTEM

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ABSTRACT

This paper describes a novel approach to control an automotive suspension system employing active force control (AFC) strategy. AFC has been demonstrated to be very robust and effective in controlling dynamical system subject to disturbances, uncertainties, parametric changes and operating conditions. In the study, AFC is explicitly implemented to provide a robust mechanism for the control of the suspension system that is subjected to a number of operating and loading conditions pertaining to road input disturbance and deliberately introduced external disturbances. The classical proportional and derivative (PD) controller is also incorporated into the control scheme that could further enhance the effectiveness of the proposed method. An iterative learning technique was embedded into the control loop to obtain the estimated mass required by the AFC strategy to effect the control action. A modelling and simulation study was carried out on a quarter car model system with the applied control scheme taken into account. A comparative study between the AFC and pure PD schemes was performed to demonstrate the differences in the performance.

1.0 INTRODUCTION

Suspension design requires a compromise between the passenger comfort and good vehicle handling. To provide a good ride comfort, the suspension should be soft enough, whereas good road holding requires stiff suspension. The main requirements of a vehicle suspension are that it should be able to minimize the vertical displacement and the acceleration of the body. These are to increase the passenger comfort, to minimize the dynamic tyre load to provide maximum road holding and react to the changes of mass payloads [1]. Due to these conflicting demands, suspension design has had to be something of a compromise, which is largely determined by the type of vehicle usages. Several approaches have been

proposed as the control model on the active suspension system. Palmeri et. al. [2] proposed a fairly robust control model-based method using H_∞ technique involving a multi-input-single-output (MISO) regulator. A comparative study was made between active and passive suspension systems with the former showing a very significant result. Ono et. al. [3] employed a distributed hierarchy control of active suspension system that showed the attenuation of the road vibration and improvement in vehicle attitude. In another work, a digital control vehicle active suspension system was performed incorporating a hydro-pneumatic slow active suspension with its analogue controller structure [4] embedding the classical PD and PI control elements plus preview control. Experimental results show satisfactory performance in line with the theoretical prediction. Wendel and Stecklein [5] came up with a regenerative pump concept as the active element of the system unlike the traditional hydraulic actuator that is almost universally used in the study of active suspension. However, this approach is still in its infancy stage due to some of the physical constraints encountered.

In this study, a robust method known as active force control (AFC) [6] is employed to control the performance of an automotive suspension system considering varied loading and operating conditions. The varied conditions of the car were simulated by applying several types of external disturbances in the forms of road input and the application of external forces. The preliminary study conducted here only considered the vertical displacements of the body during its operation. The paper is organised as follows. It starts with the definition of the problem statement and the description of the fundamental of the AFC method. The mathematical model governing the equations of motion of an automotive suspension system in the form of a quarter-car model is then described. This is followed by the narration of the assumptions made, the disturbance models and the control strategy used in the study. Consequently, a simulation study that includes a description of the parameters employed in the simulation process and the development of the SIMULINK model of the proposed control method are explained. Finally, the discussion of the results obtained is given and a conclusion with suggestions for future work is presented.

2.0 PROBLEM DEFINITION

In the conventional suspension system, the parameters for ride comfort and good handling are generally fixed, and they are chosen based on the design requirements of the vehicles. Practically, these parameters vary with the different road surfaces, vehicle speed, mass payloads and the disturbances. As the needs to control the behaviors of the automotive suspension on varied riding conditions arise, active suspension has been engineered. In this study, the control system introduced into the suspension system is known as the Active Force Control (AFC). By employing AFC, it is aimed that the system will remain robust and stable, even in the presence of known and unknown disturbances.

3.0 ACTIVE FORCE CONTROL

Hewit and Burdess [6] proposed an active force control (AFC) strategy to control a dynamic system in order to ensure the system will remain stable and robust in the presence of known and unknown disturbances. Musa [7] has demonstrated that AFC is superior compared to the conventional method in controlling a robot arm. The underlying concept of AFC is that it uses some measured and estimated values of the identified system parameters namely, the actuated force, acceleration and the estimated mass of the body.

Figure 1 illustrates the application of AFC to a dynamic translational system assuming a trajectory tracking control task. The physical quantities which need to be measured directly from the system are the actuated force, F_a and the linear acceleration, a which could be accomplished using some sensing elements such as a force sensor and an accelerometer. Then, the estimated mass, M required by the AFC strategy should be appropriately estimated using suitable methods such as the crude approximation technique and intelligent means [7]. By doing this, the compensated or estimated disturbance, F_d could be obtained indirectly through the following expression,

$$F_d = F_a - M a \quad (1)$$

The above equation, if properly obtained contribute to a very robust performance of the dynamic system even in the presence of known and unknown disturbances [6,7]. In the study, the dynamic system to be considered is an automotive suspension system that is subjected to external forces. The main drawback of the AFC method is the need to appropriately estimate the mass required by the strategy to be fed into the forward control loop for the compensation of the external disturbances. In this simulation study, an iterative learning method was used to approximate the estimated (sprung) mass of the system. The controller shown on the left of Figure 1 can be easily the conventional PD type.

4.0 THE SUSPENSION SYSTEM MODEL

The mathematical model of the suspension system is developed from the fundamental principle of the dynamic system. Firstly, a suitable model of the suspension system with the exertion of all the related forces was considered. The model of the automotive suspension system is derived from a quarter car model as shown in Figure 2. The suspension spring with stiffness K_s and damper with damping coefficient, C are installed in parallel between the sprung (M_s) spacing required and unsprung (M_u) masses. The sprung mass of a car is defined as the mass that is carried by the suspension spring. It consists of the frame, body, engine, transmission and any other parts that moves directly with the frame and body.

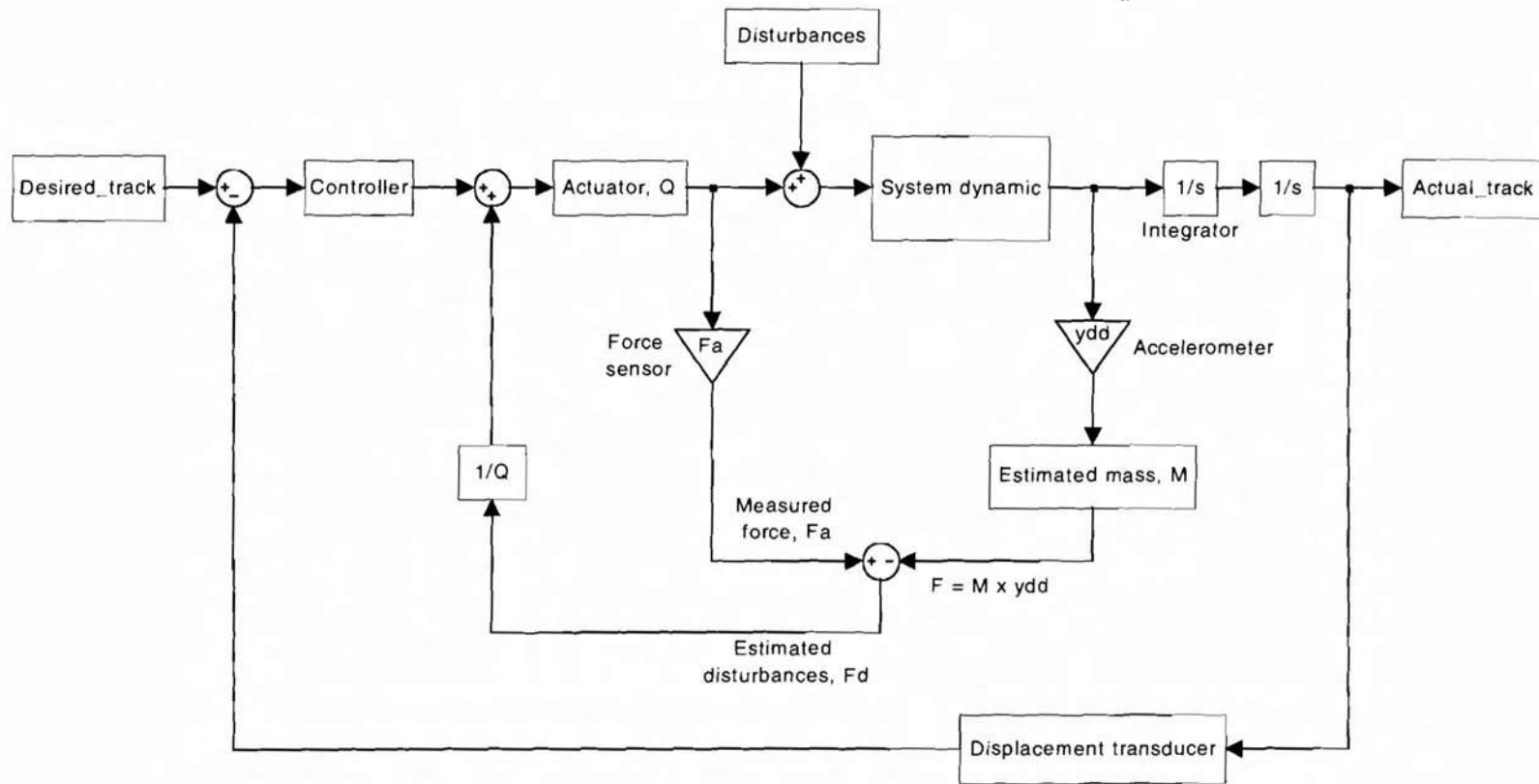


Figure 1 A schematic diagram of the AFC strategy applied to a translational system

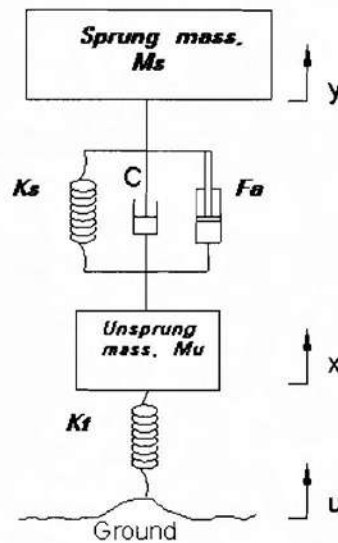


Figure 2 A quarter car model of the active suspension system

On the other hand, the unsprung mass is not carried by the spring but those components which include tyre, wheels, brake assemblies and any parts that travels directly with the tyres and wheels [8]. The ground serves as the road inputs (disturbances) to the control scheme and is represented by an excitation function, u that is exerted on the tyre and causes vertical displacements on both the sprung and unsprung masses of the system (denoted by x and y respectively). K_t is defined as the stiffness of the tyre.

The above configuration functions as a conventional passive suspension system. In this way, this model is very often used for suspension analysis and design because it is simple, yet able to capture many important characteristics of the full model [1]. In addition to the above, an actuator with force F_a is introduced into the system to provide the active element in conjunction with the implementation of the AFC scheme. The actuator will effectively and indirectly control the system by introducing some form of energy into the system when activated.

The governing equations of the suspension model were derived considering free body diagram analysis. These equations constituting the mathematical model of the system are to be used in constructing the simulation diagram via SIMULINK. Consequently, a control strategy is introduced into the suspension system to investigate the system's performance, particularly its robustness due to the presence of the external disturbances. A number of assumptions made throughout the modelling and simulation study are also clarified in the relevant section.

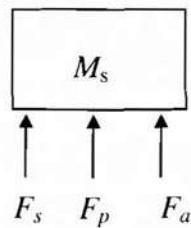


Figure 3 The forces acting on the sprung mass

Figure 3 is the free body diagram of the sprung mass showing the excitation forces. Based on Newton 2nd Law of Motion, the following relationship can be obtained,

$$F_s + F_p + F_a = M_s \ddot{y} \quad (2)$$

where F_s and F_p are the spring and damping forces and assuming that $x > y$, they can be expressed as

$$F_s = k_s (x - y) \quad (3)$$

$$F_p = C (x - y) \quad (4)$$

Substituting Equations (3) and (4) into Equation (2) yields

$$K_s(x - y) + C(\dot{x} - \dot{y}) + F_a = M_s \ddot{y} \quad (5)$$

Rearranging Equation (5) gives

$$M_s \ddot{y} + C(\dot{y} - \dot{x}) + k_s(y - x) - F_a = 0 \quad (6)$$

The forces acting on the unsprung mass are shown in Figure 4.

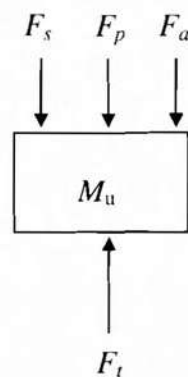


Figure 4 Forces acting on unsprung mass

The tyre force is represented by

$$F_t = K_t(u - x) \quad (7)$$

and equating the summation of forces gives

$$F_t - F_p - F_s - F_a = M_u \ddot{x} \quad (8)$$

Substituting Equations (3), (4) and (7) into Equation (8) yields

$$K_t(u - x) - C(\dot{x} - \dot{y}) - Fa = M_u \ddot{x} \quad (9)$$

Rearranging Equation (9), the equation becomes

$$M_u \ddot{x} + C(\dot{x} - \dot{y}) + K_t(x - u) + K_s(x - y) + Fa = 0 \quad (10)$$

4.1 The Actuator

The simulation study of the automotive suspension system which employed the AFC scheme is performed by employing an actuator that acts as one of the control elements in the system. The actuator has been modelled as a linear type with a suitable gain assigned which is being determined by performing a series of simulations. The actuator provides the controlled force to lift the sprung mass to a specified vertical displacement for effective position control.

4.2 Assumptions

A number of assumptions were made in the implementation of the tasks outlined in the objectives of this project. They were as follows,

- i. The quarter car model is configured and assumed reasonably accurate to represent the important characteristics of the full car model.
- ii. The value of the parameters adapted from [1] were used in the proposed study.
- iii. Only vertical movement of the suspension system (body) is considered in the study and that the suspension workspace is neglected.
- iv. The damping coefficient of the damper remains constant during operation.
- v. Unity gains were assumed for the sensors used in the study.

4.3 The Control Strategy

The proposed control strategy of the active suspension is schematically presented in Figure 5. The dotted line indicates the boundary of the control system and generally, the figure shows how the task is being implemented. By taking the desired point as the reference, the controller needs some measurements pertaining

to the acceleration and vertical displacement of the sprung mass in order to actuate the control signal via the actuator. These flow of information/data if appropriately and accurately conveyed will affect the controlling action of the proposed strategy. It is also important to note that the estimated mass of the system should also be appropriately acquired and later introduced into the AFC loop. If digital control is employed, most of the above parameters considered will be related to numerical data processing.

The proposed control scheme applied to the suspension system is a combination of AFC and PD controller with an iterative learning technique employed to estimate M . Figure 6 shows a block diagram of the proposed control scheme based on Equations (1), (6) and (10). It should be noted that the disturbances comprise the road input (bump and hole) and also deliberately introduced external disturbances in the form of constant and harmonic forces, all of which serve as a mechanism to test the robustness of the proposed method. For the AFC loop, the force sensor and accelerometer are assumed to have unity gains, while the proportional and derivative gains, K_p and K_d for the PD controller have been pre-determined through a number of trial runs. The estimated mass M necessary for the AFC implementation is approximated using an iterative learning algorithm. The outermost loop represents the conventional PD control loop involving the measurement of the actual displacement of the body (sprung mass).

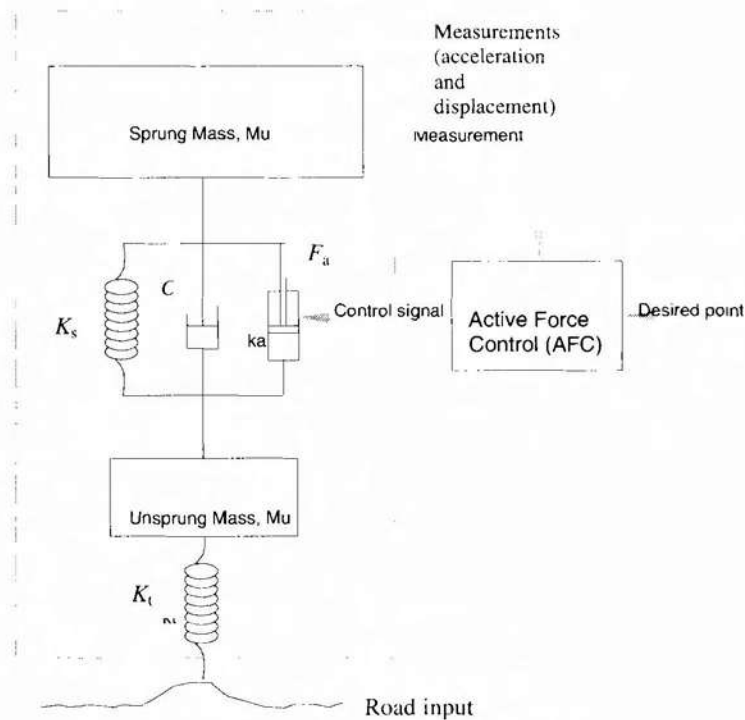


Figure 5 A diagram of the control strategy

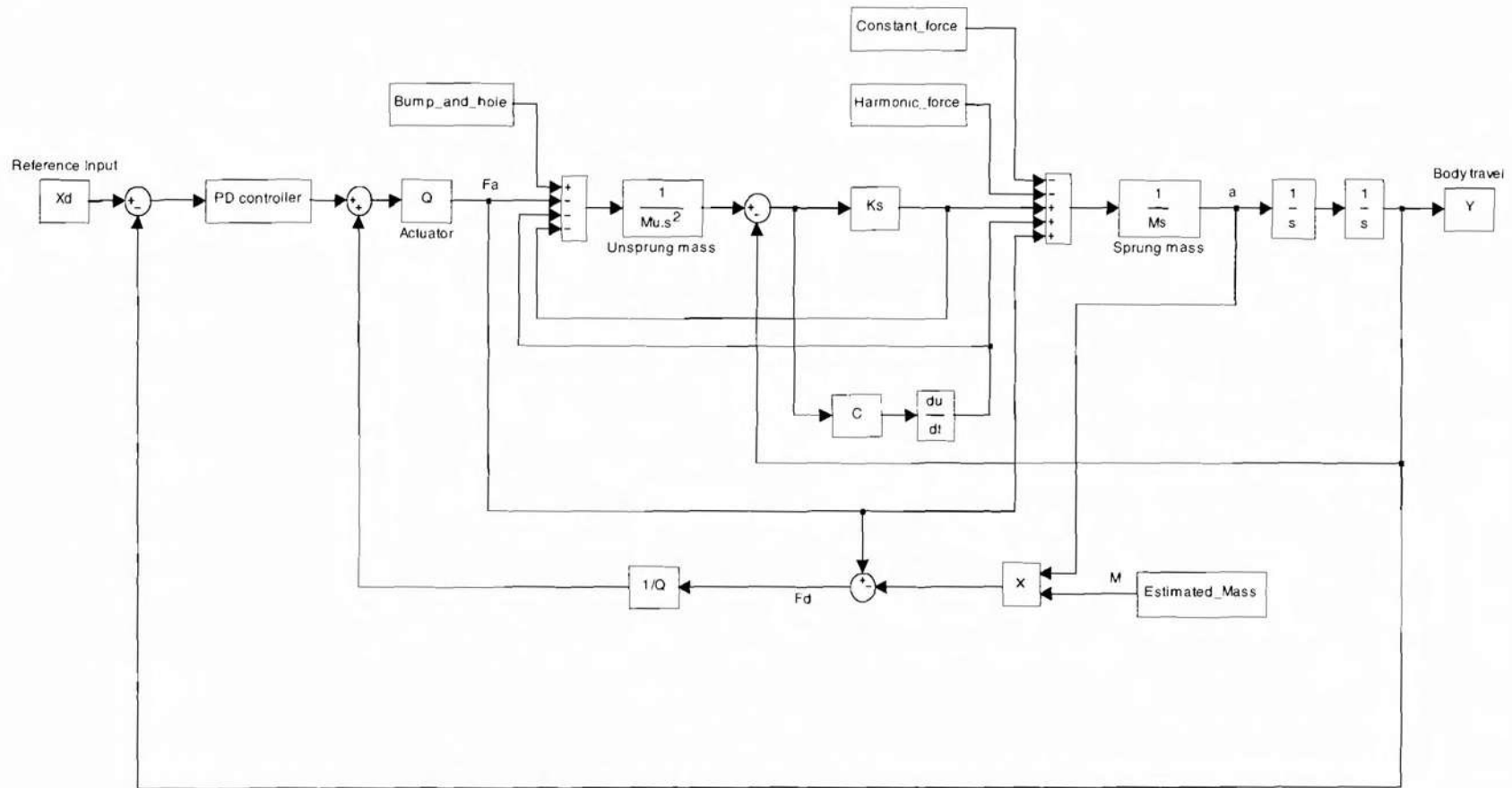


Figure 6 Proposed A.F.C control scheme applied to the suspension system

4.4 Disturbances

Several types of disturbances have been specifically introduced to the system to examine the robustness of the proposed control system in the presence of the external disturbances. There are three types of disturbances considered in the study and they take the following forms:

- i. A constant force of magnitude 3500 N acting downward on the sprung mass.
- ii. A harmonic force acting downward on the sprung mass (amplitude equals 100 N).
- iii. Road input disturbance in the form of a ‘bump and hole’ for a duration of 4s and with peak displacements of ± 8 cm about the zero datum (ground).

Figures 7 and 8 graphically show the harmonic force function and road input disturbances respectively.

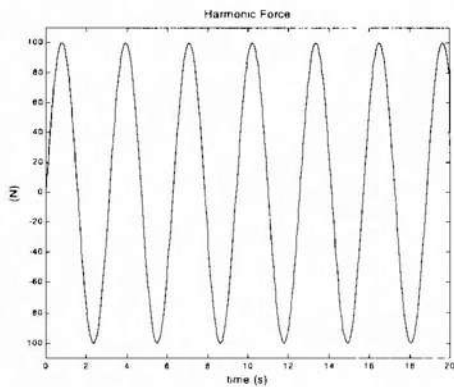


Figure 7 Applied harmonic force

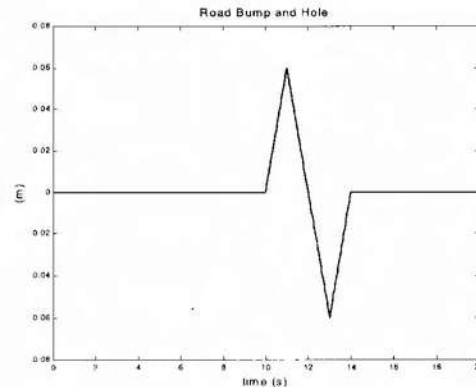


Figure 8 A ‘bump and hole’

4.5 Iterative Learning Algorithm

The estimated mass, M is obtained using the iterative learning technique as proposed in [7]. The concept of iterative learning is such that as time approaches infinity, the track error of a dynamical system should converge to zero datum. In other words, the system is said to be learning progressively with the error gradually decreases as time increases.

The equation governing the algorithm is expressed as,

$$M_{k+1} = M_k + Ae + Be \quad (11)$$

where

- k is the current value
- $k+1$ is the next step value
- A is a proportional constant
- B is a derivative constant
- e is the track error of the system
- \dot{e} is the derivative of the track error

5.0 SIMULATION

The major advantage of analyzing the response of the modelling system through the simulation is that it allows the designer to predict the behavior of the system before it is built. In this study, the MATLAB/SIMULINK software package was chosen as a tool to perform the simulation task because it offers the most powerful desirable features. Figure 9 shows the overall SIMULINK diagram comprising five sub-systems related to the suspension system model, the AFC control scheme, the iterative learning algorithm, the disturbance models and the system response. The diagrams were constructed based on the governing equations of the mathematical modelling of the sub-systems involved. Note that the input and output variables of each sub-system are also clearly indicated in the figure.

5.1 Simulation Parameters

The parameters used in this simulation are described as follows:

Suspension system (adapted from [1]):

Sprung mass,	$M_s = 290$ kg
Unsprung mass,	$M_u = 59$ kg
Suspension spring stiffness,	$K_s = 18812$ N/m
Damping coefficient,	$C = 3000$ Ns/m
Tyre stiffness,	$K_r = 250$ kN/m

Iterative learning algorithm:

Proportional constant,	$A = 600$
Derivative constant,	$B = 550$

Actuator:

Gain,	$K_a = 29500$ N/m
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PD Controller:

Proportional gain,	$K_p = 0.3$
Derivative gain,	$K_d = 0.05$

Reference input function:

Reference displacement – a step function, final value = 0.04 m

Simulation parameters:

Integration algorithm,	: Ode 45 (Dormand-Prince)
Start time,	: 0.0 s
Stop time,	: 20.0 s
Minimum step size,	: auto
Maximum step size,	: auto

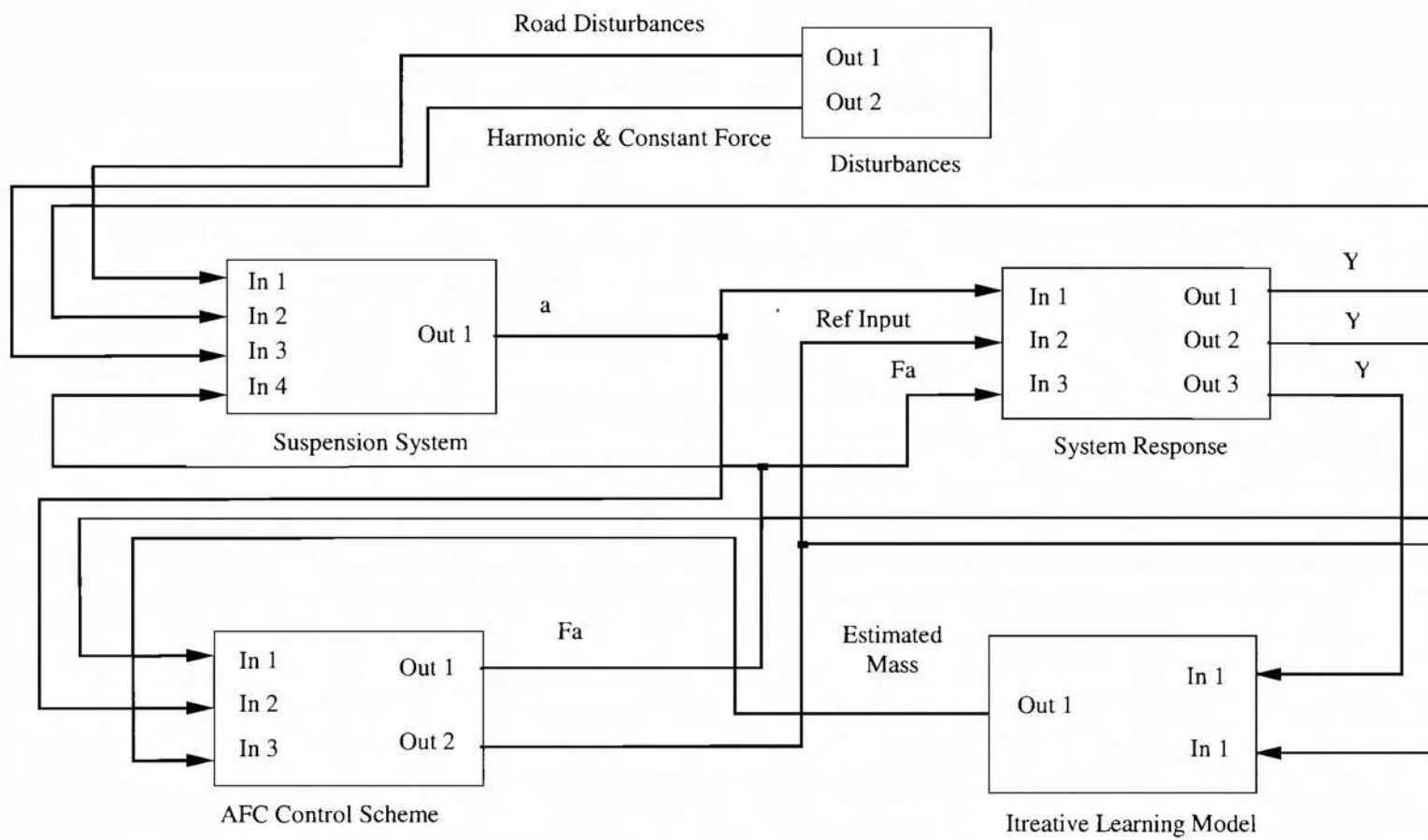


Figure 9 A SIMULINK diagram of the proposed control scheme

6.0 RESULTS AND DISCUSSION

The results obtained from the AFC scheme is compared to the classical PD controller. Both control schemes were simulated using the same values of the parameters such as K_p , K_d and K_a . The reference input is also the same. The plotted graphs of Figure 10 through 14 present both results of AFC strategy and PD controllers applied on an automotive suspension system. In Figure 10, it is obvious that the AFC strategy has showed that the response of the body travel follows its reference input until the end of the simulation period. On the other hand, for the PD controller, the response is totally unacceptable since it goes far beyond the reference input. It was noted that the car body was pushed extremely far down by a displacement of about 17 cm. This seems unrealistic since the large amount of body travel may exceed the suspension travel limit.

Figures 11 and 12 show the body acceleration that was achieved by AFC and PD control strategies. In the AFC strategy, the car body is undergoing a normal acceleration and deceleration. This is because the body accelerates when moving in the downward direction and decelerates in the upward direction. For the PD controller, the car body initially decelerates and then gradually shifts to a very small acceleration before diminishing. The acceleration of the car body while hitting the road disturbance is shown in Figure 13.

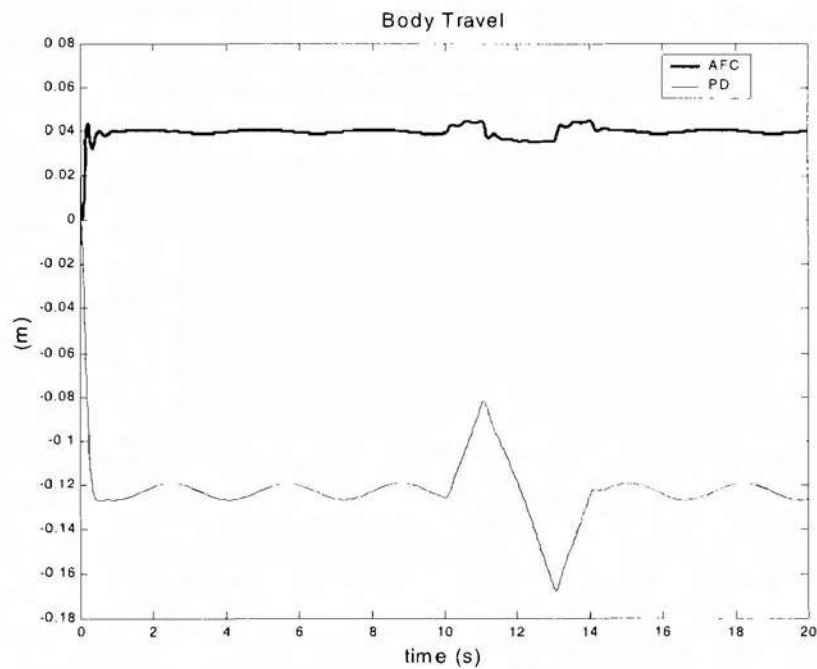


Figure 10 The response of the body travel for both control schemes

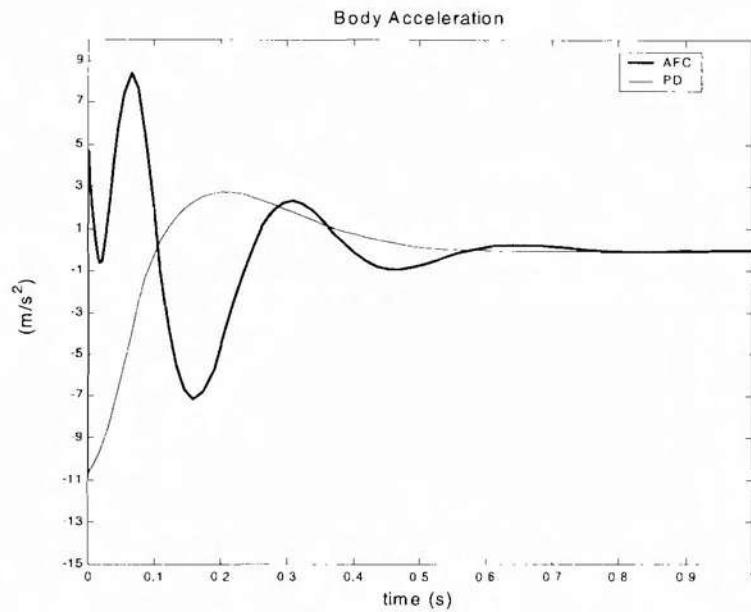


Figure 11 A close-up view of the body acceleration for both control schemes

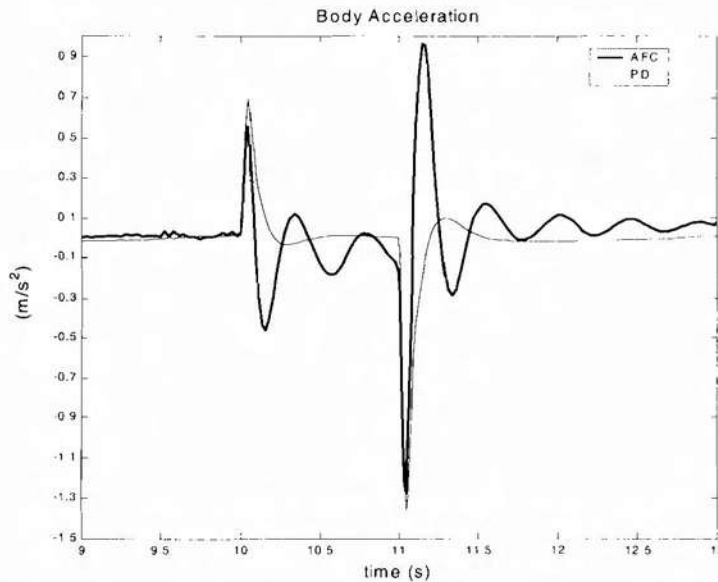


Figure 12 Body acceleration on rough road surface for both control schemes

The actuated force is shown in Figure 13. In order to react against the disturbances, a larger amount of force is required by the AFC strategy than the amount needed by the PD controller. The inability of the PD controller to ensure the system follows the reference input is resulting from the insufficient amount of force generated. This is as noticed in Figure 13 that the PD controller has actuated only a comparatively small force than the force actuated by the AFC strategy.

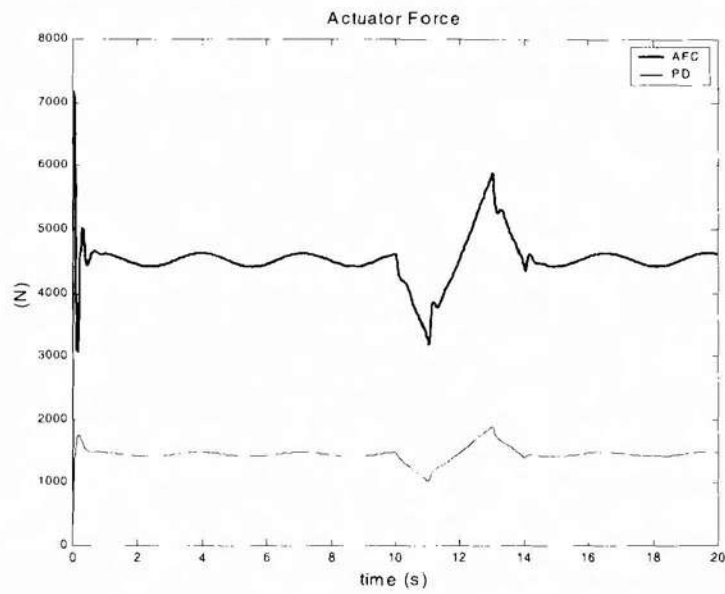


Figure 13 The actuated force for both control schemes

The track error of the body travel is shown in Figure 14. The figure clearly demonstrate the effectiveness of the AFC control strategy applied to the suspension system. As the system with the PD controller is unable to follow the reference input, a high magnitude of steady state error occurs in the system as can be seen in the figure.

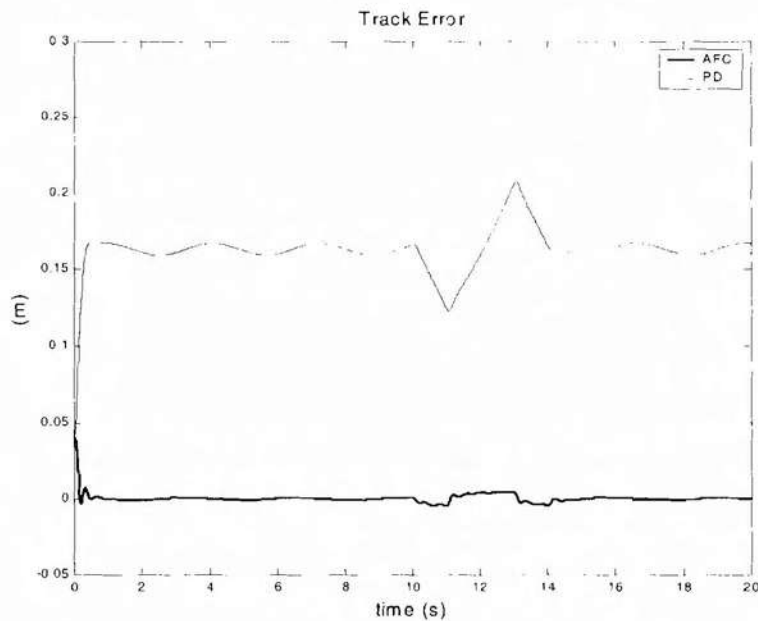


Figure 14 The track error of body travel for both control schemes.

6.0 CONCLUSION

In this preliminary study, the proposed AFC scheme which has been applied to an automotive suspension system has been shown to be very robust and effective in compensating the error due to the specified road input profile and the introduced external disturbances. The AFC scheme has shown that the response of the system follows satisfactorily the given reference input, whereas for the PD controller, this task cannot be achieved satisfactorily. Furthermore, the response of the system employing only classical PD controller goes far beyond the acceptable limit of the suspension workspace. The appropriate selection of the AFC parameters such as the estimated mass, actuator gain and controller gains is vital for the effectiveness of the control system. Further comprehensive and rigorous study should include the riding performance involving more degrees-of-freedom, full suspension workspace, frequency response analysis and the application of other intelligent methods to approximate the estimated mass using the AFC strategy.

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