

Improvement of Heavy Vehicles Ride and Braking Performance via Combined Suspension and Braking Systems Control

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ABSTRACT

Due to the importance of the fast transportation under every circumstance, the transportation process may require a high speed heavy vehicle from time to time, which may turn the transportation process more unsafe. Due to that fact the truck safety during braking and the ride comfort during long distance travelling with high speeds should be improved. Therefore, the aim of this work is to develop a control system which combines the suspension and braking systems. The control system consists of three controllers; the first one for the active suspension system of the truck body and cab, the second one for the ABS and, the third for the integrated control system between the active suspension system and the ABS. The control strategy is also separated into two strategies. The first strategy is called the ride comfort control strategy for the improvement of the truck ride comfort during long distance travelling, and the second strategy is called the braking control strategy for the improvement of the truck braking performance via ABS and the integrated control system. The Fuzzy-Logic control theory is used to construct the controlled systems. The simulation model includes 3-axle long chassis truck suspension model with cab suspension, truck braking system model, road surface model, and tyre road interface model. The influence of the road roughness, tyre-road friction coefficient and truck speed on the performance of the control strategies is also studied. The simulation results demonstrated that using Fuzzy-Logic control with the controlled suspension systems and with the ABS provides a significant improvement in truck ride comfort and braking performance under different driving conditions. Furthermore, the integrated control system between the active suspension system and ABS improves the truck braking performance at different operating conditions in

comparison with the same truck without integrated control system.

INTRODUCTION

Recently, with the advance in microcontrollers a various electronic control systems and controllers have been developed and manufactured extensively for automotive application, such as the active suspension system, anti-lock braking system (ABS), electronic brake force distribution system (EBD), traction control system (TC), and so on. This microcontrollers or subsystems was greatly developed and applied widely to improve the ride comfort, safety and stability of the vehicles. However, each subsystem often fulfills a single detection function. For example, the active suspension system is designed to improve the vehicle ride comfort performance. On the other hand, the purpose of the ABS is to maintain the slip ratio around 15 to 25 percent for improving the truck stopping distance and stability during braking process. Therefore, different control theories and strategies of the ABS and active suspension systems for different types of road vehicles have been proposed individually by several literatures for decades [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11].

In fact, the control system of the vehicles is a complicated system also the performance of the vehicles depends not only on every separated subsystem but also on the interaction between these subsystems. For example, the main idea of the integrated control between the active suspension system and the ABS is using the active suspension actuator to provide an excessive active force. This force can be considered as a normal force to load onto wheel during braking for improving the vehicle stopping distance. Hence, the interaction between the suspension and braking systems is far too complex model

adequately. Therefore, few studies on this topic are available [12, 13, 14, 15, 16, 17]. A simple 2 degree of freedom quarter vehicle model with active suspension system integrated with ABS by employing the nonlinear back stepping design schemes is developed by [12]. In this study, the integrated control scheme has been used for improving the vehicle braking performance. However, this work can be considered as a preliminary investigation in integration of the particular subsystems of the vehicle, because the model ignored the effect of the pitch motion and load transfer from the rear axle to front axle during braking process. In more detailed studies [13, 14, 15, 16, 17], a half vehicle model was developed along with models of an ABS and an active suspension system. In [13, 14] the controllers were coordinated by having desired force profile of the suspension actuator being an open loop function of the brake actuators performance. In [13, 14] the main comments can be summarized in two points. The first point is using the open-loop control can be not safe for critical applications like that of the ABS, because it produces sometimes inherently unstable systems. The second point is the open-loop control was designed based on describing the road input as an impulse excitation. In [15] the active suspension controller is designed based on the optimal control theory. The ABS controller is designed to control the slip ratio by using proportional directional valve with time delay of 0.1 second, and the integrated closed-loop scheme of the ABS and active suspension system has been presented. In addition, the road surface has been modeled as a random input. They concluded that using the integrated control system reduces the stopping distance in comparison with a vehicle using ABS only, and they suggest using the Fuzzy-Logic control with the active suspension and ABS systems for increasing the computational efficiency of the control system. In [16, 17] the ABS and active suspension system controllers are designed based on Fuzzy-Logic control using different control strategies. The integrated control algorithm is constructed to coordinate two subsystems and the evaluation of the braking and ride performance has been presented. They concluded that the integrated control between the ABS and active suspension system improved the braking performance but the integrated controller has a negative effect on the vehicle ride comfort.

According to the previous investigations in the integrated subsystems this conclusion can be written, most of the researchers concentrated their studies on passenger cars and they did not care about the trucks or truck-semi trailers. However, apply the idea of the integrated control between the suspension and braking systems is important for trucks, because the fast transportation process requires a high speed heavy vehicle, and due to that fact the truck safety during braking and ride comfort during long distance travelling with high speeds should be improved. Therefore, the aim of this work is to develop a control system which combines the suspension and braking systems with two control strategies to

improve the truck ride comfort during long distance travelling and the stopping distance and time during braking.

TRUCK MATHEMATICAL MODEL

The main goal of this study is to investigate the interaction between the vertical and longitudinal motions and its effects on the truck ride and braking performance. Hence, a suitable model for this study is a half truck model including suspension and braking systems dynamics for passive and active systems. The mathematical formulation of the road input and the tyre-road interface also included in the model. Therefore, this model enables the consideration of load transfer and pitch motion, which have a significant effect on the truck braking performance. On the other hand, for the aim of ride comfort this model enables the consideration of the cab pitch motion which has a considerable effect on the truck ride comfort. In addition, the tyre behavior on uneven roads can be included easily in this model by using empirical relationships.

SUSPENSION SYSTEM DYNAMICS MODEL

The seven degrees of freedom truck vertical dynamics model used in this paper is shown in [Figure 1](#). The passive suspension of the truck and cab modelled as linear spring and viscous damper, and denoted as K_s and C_s at each axle for truck suspension and K_c and C_c at the front and rear of the cab. On the other hand, the active suspension system of the truck and cab modelled as linear actuators, also it denoted simply as U_s at each axle for truck suspension and U_c at the front and rear of the cab. M_{w1} , M_{w2} and M_{w3} represent the steer axle, first drive axle and second drive axle masses respectively. M_t and M_c are the sprung mass of truck and cab. X_t and X_c represent the truck body and cab vertical displacement at C.G. The truck body, wheel, and road vertical displacements are denoted at each axle as X_b , X_w , and X_o respectively, also the cab vertical displacement and truck body vertical displacement under the cab are denoted as X_{cs} and X_{tc} at front and rear of the cab. θ_{tr} and θ_c are the pitch response of the truck body and cab. L , L_1 , L_2 , and L_3 are the wheel base and the distance from the truck C.G. to the steer axle, first drive axle and second drive axle respectively. L_4 , L_5 , L_{c1} , and L_{c2} are the distance from the truck C.G. and cab C.G. to the front and rear of the cab respectively. The equations of motion for the truck vertical and pitch responses can be written as the following:

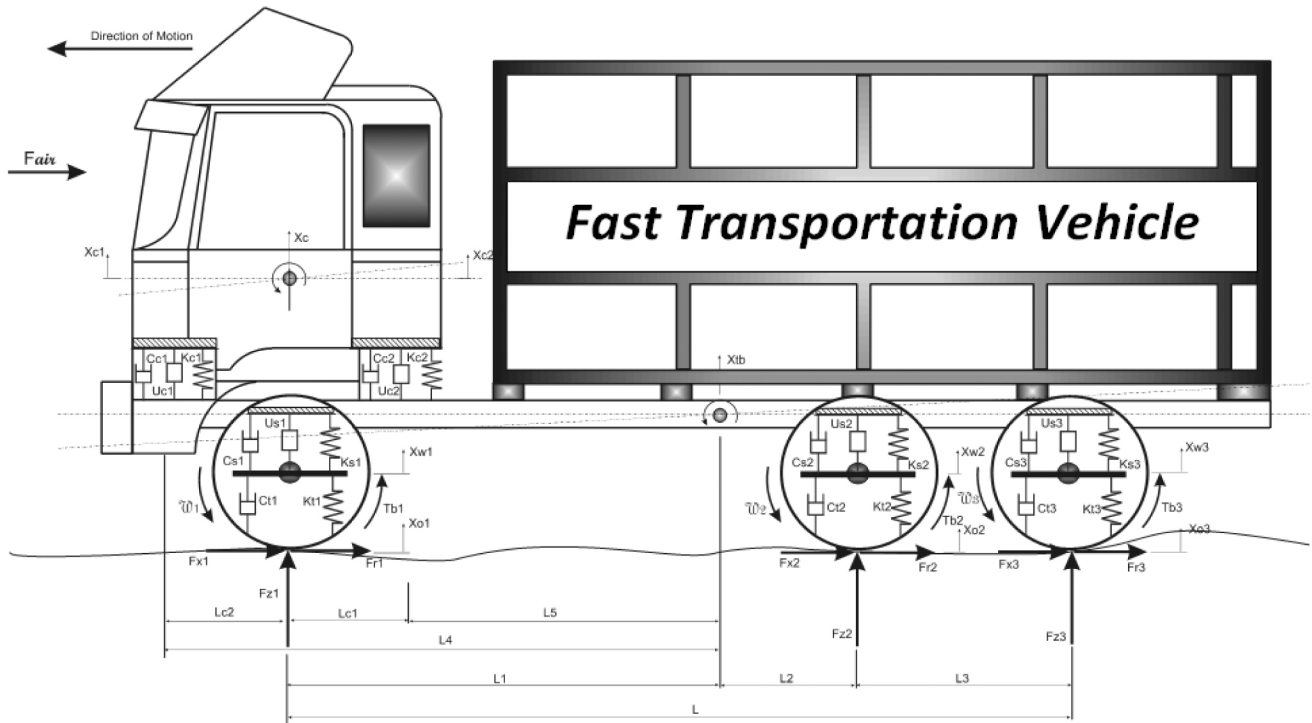


Figure 1. Truck dynamics model.

$$M_t \ddot{X}_t = -K_{s1}(X_{t1} - X_{w1}) - C_{s1}(\dot{X}_{t1} - \dot{X}_{w1}) - K_{s2}(X_{t2} - X_{w2}) - C_{s2}(\dot{X}_{t2} - \dot{X}_{w2}) - K_{s3}(X_{t3} - X_{w3}) - C_{s3}(\dot{X}_{t3} - \dot{X}_{w3}) + K_{c1}(X_{c1} - X_{tc1}) + C_{c1}(\dot{X}_{c1} - \dot{X}_{tc1}) + K_{c2}(X_{c2} - X_{tc2}) + C_{c2}(\dot{X}_{c2} - \dot{X}_{tc2}) + U_{s1} + U_{s2} + U_{s3} - U_{c1} - U_{c2} \quad (1)$$

$$I_t \ddot{\theta}_t = L_1 K_{s1}(X_{t1} - X_{w1}) + L_1 C_{s1}(\dot{X}_{t1} - \dot{X}_{w1}) - L_2 K_{s2}(X_{t2} - X_{w2}) - L_2 C_{s2}(\dot{X}_{t2} - \dot{X}_{w2}) - L_3 K_{s3}(X_{t3} - X_{w3}) - L_3 C_{s3}(\dot{X}_{t3} - \dot{X}_{w3}) + L_4 K_{c1}(X_{c1} - X_{tc1}) + L_4 C_{c1}(\dot{X}_{c1} - \dot{X}_{tc1}) - L_5 K_{c2}(X_{c2} - X_{tc2}) - L_5 C_{c2}(\dot{X}_{c2} - \dot{X}_{tc2}) - L_1 U_{s1} + L_2 U_{s2} + L_3 U_{s3} + L_4 U_{c1} + L_4 U_{c1} \quad (2)$$

$$X_{t1} = X_t - L_1 \theta_t, \quad X_{t2} = X_t + L_2 \theta_t, \quad X_{t3} = X_t + L_3 \theta_t \quad (3)$$

$$h_i = h + (X_{ti} - X_{wi}) + (X_{wi} - X_{oi}), \quad i = 1, 2, 3 \quad (4)$$

The equations of motion for the cab vertical and pitch responses can be written as:

$$M_c \ddot{X}_c = -K_{c1}(X_{c1} - X_{tc1}) - C_{c1}(\dot{X}_{c1} - \dot{X}_{tc1}) - K_{c2}(X_{c2} - X_{tc2}) - C_{c2}(\dot{X}_{c2} - \dot{X}_{tc2}) + U_{c1} + U_{c2} \quad (5)$$

$$I_c \ddot{\theta}_c = L_{c1} K_{c1}(X_{c1} - X_{tc1}) + L_{c1} C_{c1}(\dot{X}_{c1} - \dot{X}_{tc1}) - L_{c2} K_{c2}(X_{c2} - X_{tc2}) - L_{c2} C_{c2}(\dot{X}_{c2} - \dot{X}_{tc2}) + L_{c1} U_{c1} + L_{c2} U_{c2} \quad (6)$$

$$\begin{aligned} X_{c1} &= X_c - L_{c1} \theta_c, & X_{c2} &= X_c + L_{c2} \theta_c, \\ X_{tc1} &= X_t - L_4 \theta_t, & X_{tc2} &= X_t - L_5 \theta_t \end{aligned} \quad (7)$$

The equations of motion for the steer axle and drive axles can be written as:

$$M_{w1} \ddot{X}_{w1} = -K_{t1}(X_{w1} - X_{o1}) - C_{t1}(\dot{X}_{w1} - \dot{X}_{o1}) + K_{s1}(X_{t1} - X_{w1}) + C_{s1}(\dot{X}_{t1} - \dot{X}_{w1}) - U_{s1} \quad (8)$$

$$M_{w2} \ddot{X}_{w2} = -K_{t2}(X_{w2} - X_{o2}) - C_{t2}(\dot{X}_{w2} - \dot{X}_{o2}) + K_{s2}(X_{t2} - X_{w2}) + C_{s2}(\dot{X}_{t2} - \dot{X}_{w2}) - U_{s2} \quad (9)$$

$$M_{w3}\ddot{X}_{w3} = -K_{t3}(X_{w3} - X_{o3}) - C_{t3}(\dot{X}_{w3} - \dot{X}_{o3}) + K_{s3}(X_{t3} - X_{w3}) + C_{s3}(\dot{X}_{t3} - \dot{X}_{w3}) - U_{s3} \quad (10)$$

BRAKING SYSTEM DYNAMICS MODEL

With the forces shown on the truck in [Figure 1](#), the equation of motion for the truck braking dynamics in longitudinal direction can be described as the following [18]:

$$M_{tot}\dot{v} = -F_{x1} - F_{x2} - F_{x3} - F_{r1} - F_{r2} - F_{r3} - F_d \quad (11)$$

where M_{tot} is the total mass of the truck, which $M_{tot} = M_t + M_c + M_{w1} + M_{w2} + M_{w3}$. F_d , F_{ri} , and F_{xi} are the aerodynamic drag force, rolling resistance force and tyre-road friction force respectively. These forces can be calculated from the following formulas and according to operating conditions.

$$F_d = 0.5 * C_d * \rho * A_f * v^2 \quad (12)$$

where C_d , ρ , A_f , and v are the truck aerodynamic coefficient, air density, frontal projected area, and truck speed respectively. At the same time, for truck speed up to 150 km/hr the rolling resistance forces can be described using the following equation [18].

$$F_{ri} = (0.006 + 0.23 * 10^{-6} * v^2) F_{zi} \quad (13)$$

where F_{zi} is the normal forces due to truck weight and suspension system. The dynamic modes of the truck wheels due to braking process can be calculated from the following formula:

$$J_{wi} = (F_{xi} + F_{ri}) r_{di} - T_{bi} \quad (14)$$

where J_{wi} , r_{di} and T_{bi} are the wheel inertia including any drive train effect, effective dynamic tyre radius and braking torque at each wheel respectively.

TYRE-ROAD INTERFACE MODEL

The tyre-road friction force is the main problem in the ABS design. However, the relation between longitudinal wheel slip ratio and tyre-road friction coefficient is physically unknown to day knowledge. Therefore, this relation is usually determined from empirical models as the Dugoff and Magic Formula tyre models [7, 10, 19, 20]. In this study, a widely used semi empirical model is used to calculate the relation between tyre-road friction coefficient and longitudinal slip ratio at different operating condition [20]. The obtained relationship between the tyre-road friction coefficient and the

wheel slip ratio is shown in [Figure 2](#) for different road condition.

$$\mu_i(\lambda_i) = D \cdot \sin \left(C \cdot \arctan \left(B\lambda_i - E(B\lambda_i - \arctan(B\lambda_i)) \right) \right) \quad (15)$$

where μ_i and λ_i are the tyre-road friction coefficient and the wheel slip ratio. Which $\lambda_i = 1 - [(\omega \cdot r_{di})/v]$ and B, C, D, and E are the Magic formula tyre model coefficients as shown in [Table 1](#). As a result of obtaining the tyre-road friction coefficient, the tyre-road friction force can be calculated using the following equation:

$$F_{xi} = \mu_i(\lambda_i) F_{zi} \quad (16)$$

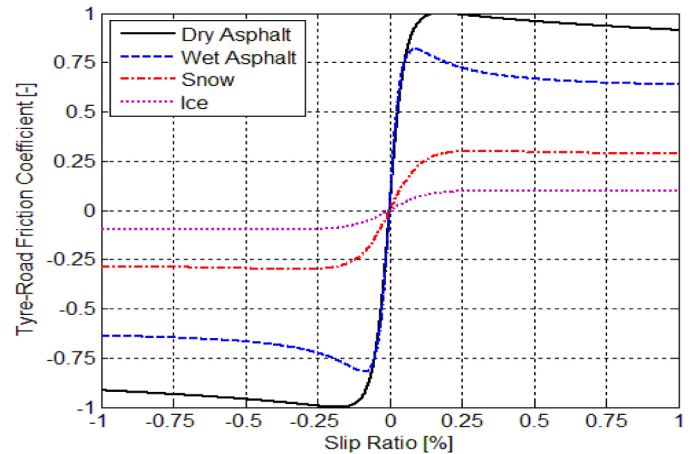


Figure 2. The relationship between tyre-road friction coefficient and slip ratio.

Table 1. Tyre-road interface model parameters.

Road condition	B	C	D	E
Dry Asphalt	10	1.9	1	0.97
Wet Asphalt	12	2.3	0.82	1
Snow	5	2	0.3	1
Ice	4	2	0.1	1

ROAD INPUT MODEL

In this work, a time domain random road profile is generated as a filtered white noise signal considering the road roughness and the change in truck speed [21]. The random road profile is shown in [Figure 3](#).

$$\dot{X}_o(t) = -2\pi \cdot f_o \cdot X_o(t) + \sqrt{G \cdot G_o(t) \cdot v(t)} w(t) \quad (17)$$

where X_o is the road input displacement, G_o is the road roughness coefficient, v is the truck speed and $w(t)$ is zero-

mean Gaussian white noise with intensity 1. f_o is the low cut-off frequency, which is assumed to be 0.01 Hz in this study.

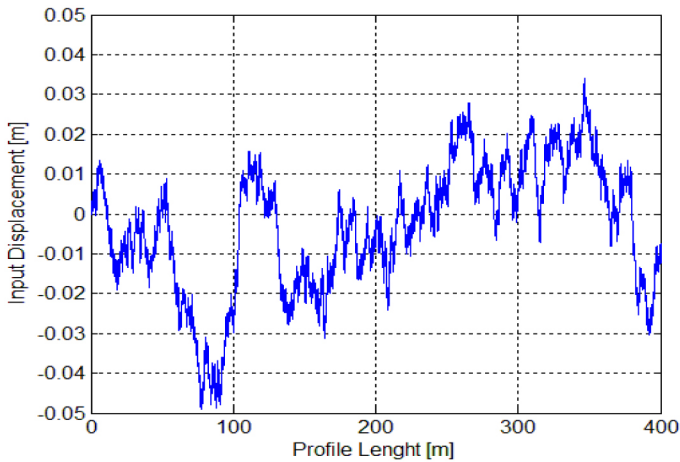


Figure 3. The random road profile.

CONTROL THEORY AND STRATEGIES

CONTROL THEORY

In this investigation, the Fuzzy-Logic control theory is used, or in other words one of the intelligent control methods, which offers several unique features that make it a particularly good choice for many control problems. In addition it can use to control the nonlinear systems that would be difficult or impossible to model mathematically. Moreover, it does not require precise, noise-free inputs and can be programmed to fail safely if a feedback sensor quits or destroyed. The output control is a smooth control function despite a wide range of input variations. Therefore, any sensor data provides some indication of a system actions and reactions is sufficient. Hence, it allows the sensors to be inexpensive and imprecise thus keeping the overall system cost and complexity low [22, 23]. For these advantages of the Fuzzy-Logic control, a lot of researchers concentrated their attention to apply the Fuzzy-Logic control theories in the controlled systems of the vehicle.

ACTIVE SUSPENSION SYSTEMS CONTROLLERS

TRUCK SUSPENSION SYSTEM CONTROLLER

The active suspension system controller has been built by using Fuzzy-Logic control theory. This controller uses the truck body vertical acceleration and the differentiation of the body vertical acceleration as a controller input signals. As a result of the control process, the suspension actuators desired forces have been generated as an output signals. The rule base and the interface engine are based on Mamdani-Type of fuzzy inference, while the defuzzification process is based on

the center of area method. The gradient-based optimization algorithm is used to optimize the controller input and output gains. The optimization goal is to minimize the body vertical acceleration and the dynamic tyre load whilst at the same time maintains the suspension working space in the allowable limits. Therefore, the following performance index is selected for the optimization process.

$$PI = 0.5 \int_{t_0}^{\infty} q_1 (X_{b1} - X_{w1})^2 + q_2 (X_{w1} - X_{o1})^2 + q_3 \ddot{X}_{b1}^2 + q_4 (X_{b2} - X_{w2})^2 + q_5 (X_{w2} - X_{o2})^2 + q_6 \ddot{X}_{b2}^2 + q_7 (X_{b3} - X_{w3})^2 + q_8 (X_{w3} - X_{o3})^2 + q_9 \ddot{X}_{b3}^2 + \rho_1 U_1^2 + \rho_2 U_2^2 + \rho_3 U_3^2 \quad (18)$$

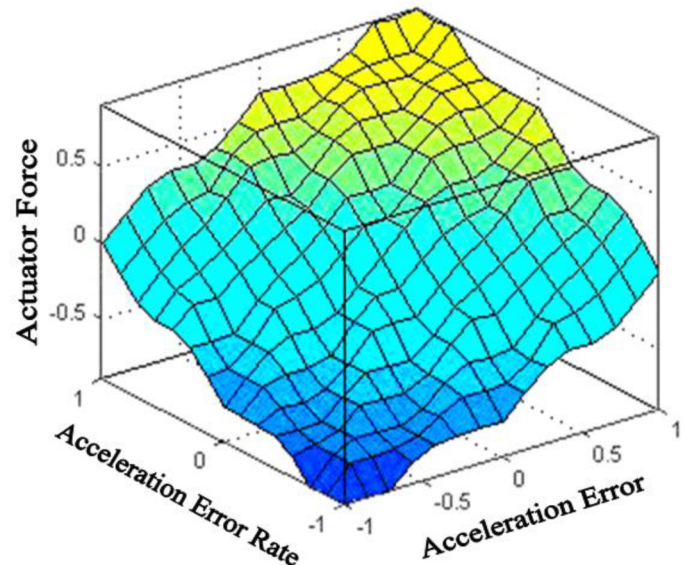


Figure 4. The active suspension system controller surface map.

where $q_1, q_2, \dots, q_9, \rho_1, \rho_2$ and ρ_3 are the weighting parameters. Surface map of the active suspension system controller is shown in Figure 4.

CAB SUSPENSION SYSTEM CONTROLLER

The cab suspension system controller is suggested to reduce the transmitted vibration from the truck body to cab body, also to reduce the excessive vibration during braking process when the integrated control system is used. From this point of view, the cab vertical and pitch accelerations are selected as input signals. The controller uses the cab vertical acceleration and the error in the cab vertical acceleration, also uses the cab pitch acceleration and the error in the cab pitch acceleration as a controller inputs. The output signals of the controller are the cab front and rear actuators desired forces. The rule base and interface engine are based on Mamdani-Type of fuzzy inference, while the defuzzification process is based on the center of area method. The gradient-based optimization algorithm is used to optimize the controller input and output gains. The optimization purpose is to minimize the cab body

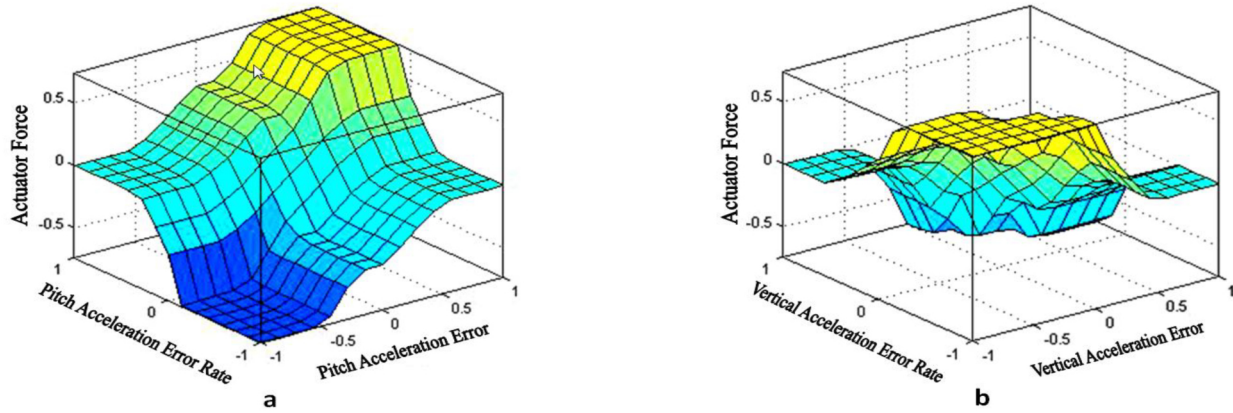


Figure 5. The cab suspension system controller surface map a - pitch motion part and b- vertical motion part.

accelerations, also at the same time maintain the suspension working space of the cab in the allowable limits. Therefore, the following performance index is selected for the optimization goal.

$$PI_c = 0.5 \int_{t_0}^{\infty} q_{c1}(X_{c1} - X_{bc1})^2 + q_{c2}(X_{c2} - X_{bc2})^2 + q_{c3}\dot{X}_c^2 + q_{c4}\ddot{X}_c^2 + \rho_{c1}U_{c1}^2 + \rho_{c2}U_{c2}^2 \quad (19)$$

where q_{c1} , q_{c2} , q_{c3} , ρ_{c1} and ρ_{c2} are the weighting parameters. The surface map of the cab suspension system controller is shown in Figure 5.

ANTI-LOCK BRAKING SYSTEM CONTROLLER

The objective of the ABS system is to keep the slip ratio close to the peak value on the “tyre-road friction coefficient - slip ratio” curve in Figure 2. Furthermore, the currently used ABS sensors measure the wheel angular velocity which the wheel deceleration is determined by differentiation, and the tyre slip is estimated by comparing this measured wheel velocity with a memory of the wheel velocity before initiation of braking. Hence, the change of wheel velocity reflects the dynamic characteristic between the tyre and road. Therefore, in this study the slip ratio with the wheel deceleration control strategy of the ABS controller has been selected. The Fuzzy-Logic control theory is used in the construction of the ABS controller. Furthermore, the ABS controller uses the wheel slip ratio and the wheel deceleration as controller input signals. The wheel deceleration and the wheel slip ratio signals serve to alert the ABS controller to any locking tendency. Hence, the ABS controller responds to such an alert by sending a signal to the solenoid control valve to increase or decrease the braking pressure. In this study, the output signal generated by the ABS controller can be considered as the braking torque signal because the valve dynamics is neglected. The surface map of the ABS controller is shown in Figure 6.

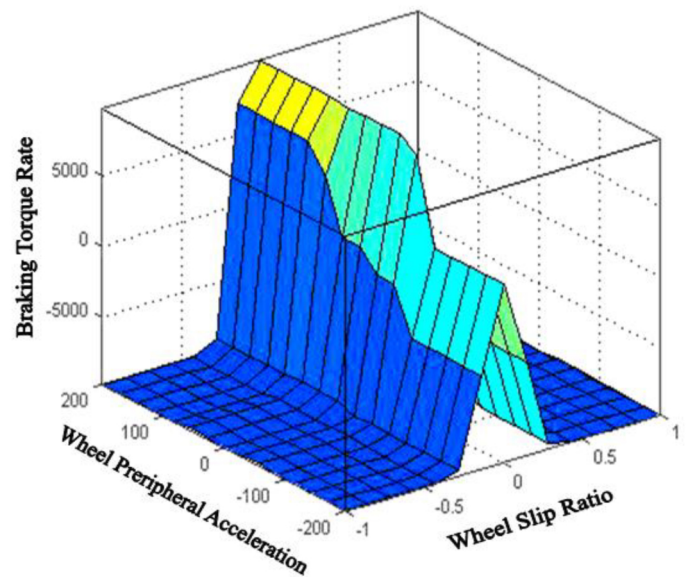


Figure 6. The ABS Fuzzy-Logic controller surface map.

INTEGRATED CONTROL ALGORITHM

The normal force at each wheel is the sum of the dynamic weight of the vehicle and the suspension forces. The dynamic weight of the vehicle is mainly determined by the vehicle longitudinal deceleration. Therefore, the integrated control system uses the active suspension system actuators as a tool to supply a normal force on the tyres change in phase with the brake torque being applied. When the brake torque is high, normal force would be high and vice versa [12, 13, 14, 15, 16, 17]. In this way, when the tyre-road friction coefficient reaches the peak value and the normal force also reaches the peak value, the maximal tyre-road friction force can be obtained. As a result, the maximal braking torque is obtained. Hence, provide the integrated control algorithm to the active suspension and ABS controllers is important to improve the braking performance. The integrated control algorithm can be written as the following: