Design and simulation of a controller for an active suspension system of a rail car

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Abstract: The quest to increase the performance of rail car in terms of stability, safety and ride comfort has triggered research into the design, modelling and dynamic simulation of the suspension system of a rail car. Advance control and automation techniques can be used to optimize the performance of a rail car suspension system. This work introduces the mathematical model of two-degree-of-freedom and the use of the conventional proportional integral as well as derivative (PID) controller for the active suspension system of a rail car. Dynamic simulation via the use of MATLAB Simulink environment was employed for the design and analysis of the effect of the controller with the control algorithm of PID controller used to reduce motion of rail car body and the damper. The dynamic model for the linear rail car suspension system was formulated while the PID controller type was used to test the system’s performance while the Ziegler–Nichols tuning rules were used to determine proportional gain, reset rate and derivative time of PID controllers. The results obtained show that the developed controller is highly efficient. The simulated results of the comparison between the passive suspension system and the active suspension system proved that, active suspension system with PID controller is superior.
control improves ride comfort, gives lower or almost no peak over shoot (amplitude) and faster settling time compared to the passive suspension system. This enhances safety, good comfort for drivers and passengers as well as isolation of railcar from noise, vibration and other disturbances.

Subjects: Industrial Design; Mechanical Engineering; Mechanical Engineering Design; Manufacturing Engineering; Manufacturing Technology

Keywords: advance control; controller; damper; rail car; suspension system

1. Introduction

In the past years, there has been widespread interest in the use of advance control and automation techniques to optimize the performance of the suspension system of a rail car (Al-Zughaibi and Davis, 2015; Bideleh & Berbyuk, 2016). Suspension systems have been widely applied to the modern automobile and rail cars with complex control algorithms to reduce the effects of vertical acceleration caused by road disturbances (Ulum, Affaf, & Suparwanto, 2017; Yao & Shi, 2013). Every moving rail car is exposed to vibration which results in passengers’ discomfort, decreased durability of the car itself and increased cost of maintenance. Different disturbances occur when a rail car leans over during driving, cornering, diving or braking. Also, unpleasant vertical movement, lateral, yawing and rolling vibrations of the rail car body can occur while driving over rail way irregularities (Hasbullah & Faris, 2017; Oh et al., 2016). These dynamic motions do not only have an adverse effect on comfort but can also be unsafe, because the tyres might wear out or lose their grip on the rail way. The main task of a rail car suspension is to ensure ride comfort by isolating the car from disturbances and enhance rail way holding capacity for a variety of rail way conditions and rail car manoeuvres (Colombo, Gialleonardo, Facchinetti, & Bruni, 2014; Zhang, Han, Zhang, & Yu, 2013). This in turn will directly contribute to the comfort and safety of the user. In general, suspension system connects the wheel and the vehicle body by springs, dampers and some other linkages (Shin, Lee, Yi, & Noh, 2015; Zong, Gong, Xuan, & Guo, 2013). A good suspension system should provide a comfortable ride and good handling within a reasonable range of deflection (Bideleh, Mei, & Berbyuk, 2016; Nguyen & Nguyen, 2017). From the system design perspective, there are two main categories of disturbances on a rail car, namely railway and load disturbances (Matamoros-Sanchez & Goodall, 2015; Nakajima, Shimokawa, Mizuno, & Sugiyama, 2014). Railway disturbances have the characteristics of large magnitude in low frequency such as hills and small magnitude in high frequency such as railway roughness. Load disturbances include the variation of load induced by accelerating, braking and cornering. Therefore, in the design of a good suspension system, importance is given to fairly reduce the disturbances via the use of suspension system with proper cushioning that will be soft against railway disturbances and hard against load disturbances. The vibration that occurs in most rail cars is undesirable, not only because of the resulting unpleasant motions, the noise and the dynamic stresses which may lead to failure of the suspension system but also because of energy losses and reduction in performance that accompany the vibration (Eris, Ergen, & Kurtulan, 2015; Gohrle, Schindler, Wagner, & Sawodny, 2014; Hasbullah, Faris, Darsivan, & Abdelrahman, 2015). The design of the rail car suspension system always involves a compromise between ride comfort and handling (Ahmed, Hazlina, & Rashid, 2015; Pathare & Nimbalkar, 2014; Wang, Chen, & Yu, 2017). For good ride comfort, a compliant suspension system is normally required while good handling demands a stiff suspension system to control body roll (Sun, Deng, & Li, 2013). With normal passive suspension system, the characteristics of the springs and dampers are fixed at the design stage and cannot be changed afterwards. By using controllable springs and dampers, these characteristics can be changed while the rail car is moving. It therefore becomes possible to have soft setting for good ride comfort whilst travelling in a straight line on a good railway, while the suspension can be changed to a hard setting moment later to give good handling when the rail car has to change direction as required for lane changing or even accident avoidance. Controllable suspension systems can therefore reduce or even eliminate ride comfort against handling compromise. Conventional suspension system is
also known as a passive suspension system consisting of spring and damper mounted at each wheels of the vehicle horizontally in parallel. According to Ha, Choi, and Lee (2012), the spring of the vehicle suspension supports the vehicle body, absorb and store energy. The damper or shock absorber is a component of the rail car suspension that is used to dissipate the vibration energy stored in the spring and to control the impulse from the rail that is transmitted to the car. Other purposes of suspension system are to isolate sprung mass from the unsprung mass vibration, to provide directional stability during cornering and to manoeuvre while providing damping for the high-frequency vibration-induced excitations (Herbst, 2013; Kumar, Sivakumar, Kanagarajan, & Kuberan, 2018). Due to the lack of attitude control of the vehicle body and demand for better ride comfort and stability, many researchers and academics in the automotive field are motivated to consider the use of active suspensions. The electronically controlled active suspension system can potentially improve ride comfort as well as the road handling and vehicle stability especially for racing car (Kim, Choi, & Lee, 2014; Oh et al., 2016). A better suspension system with soft and hard suspension should be controlled electronically in order to get “soft” against railway disturbance and “hard” against load disturbance (Hrovat, 1990). In order to increase the performance of ride handling and stability of the vehicle, a new suspension system has been developed in which the system is controlled electronically. There are two categories of advanced suspension system, namely semi-active and active suspensions. In the automotive industries, semi-active and active suspension systems are considered as new technologies that are used to improve ride quality and handling of the vehicle with the semi-active suspension system showing some similarities with the conventional suspension system. This kind of suspension has a spring and controllable damper in which the spring element is used to store the energy meanwhile the controllable damper is used to dissipate the energy. Some of the semi-active suspension systems use the passive damper and the controllable spring, the controllable damper usually acts with limited capability to produce a controlled force when dissipating energy (Hrovat, 1990; Nguyen, Choi, & Nguyen, 2014). The advantages of using semi-active suspension are the operational cost which is less than active suspension coupled with the fact that only small amount of energy is consumed (Jin, Xiao, Ling, Zhou, & Xiong, 2013; Marzbanrada, Ahmadib, Zohaorc, & Hojjatd, 2004). Meanwhile in active system, the components of spring or damper are replaced with an actuator controlled with the aid of the feedback mechanism from the vehicle body. Technically, active suspension system is used to control the movement of a vehicle using on-board controller by controlling the tire movement during cornering, braking and accelerating. The method of the controller for active suspension can be divided into four types based upon the control techniques, namely solenoid actuated, hydraulic actuated, electromagnetic recuperative and magneto-rheological damper. For semi-active with controllable damper, it acts like an actuator but with limited capability, to produce a controlled force when dissipating energy and when supplying energy, it switches to a zero damping state. The advantage of semi-active system is that the cost of operation is less than the other type of advanced suspension system (Oraby, Aly, El-Demerdash, & Selim, 2007).

According to Arunachalam, Mannar Jawahar, and Tamilporai (2003), for the full active suspension system, energy is supplied into the system. It requires several components such as the accelerometer, actuators, control unit, ACs, servo valves, high pressure tanks for the control fluid, sensors for detecting the system (Allen, 2008). According to Ebrahimi (2009), they are expensive but have better performance than the passive suspension system. There are two types of configurations which are dependent upon the link of passive part (spring and damper) and active part (controller). If these two parts are linked in parallel, they are known as high-bandwidth configuration. The advantages of high-bandwidth to low-bandwidth lies in the ability to exercise control at the higher frequency while the suspension system is working continuously as a passive suspension in the event of an actuator not working as an active part. Many researchers have carried out early works and researches on advanced suspension in the past few decades in order to improve the stability and ride handling performance of the rail car using advanced methods such as: proportional–integrator–derivative (PID) controller plus linear quadratic reg-ulator (LQR) vibration controller (Minakaldas & Arefsoliman, 2014); LQR control design (Arefsoliman, 2011; Nguyen, Nguyen, & Kim, 2000); fuzzy scheme (Abdelhady, 2003; Shao-Juan,
Zhong-Hua, & Chen, 2014); impedance control system (Fateh & Alavi, 2009); neural network control system (Eski and Yildirim, 2009); spatial vehicle model (Demic, 2003); fuzzy sliding mode controller (Lin, Lian, Chung-Neng, & Wun-Tong, 2009); fuzzy reasoning and a disturbance observer (Semiha & Huseyin, 2008; Yoshimura & Takagi, 2004); fuzzy logic controller (Chen, Liu, Tong, & Lin, 2008; Chiou & Liu, 2009; Du & Zhang, 2009; Qiu, Feng, & Yang, 2009). Another control strategy has been done by Yamamoto, Sugai, Kanda, and Buma (2014) using PID controller. This new control strategy is proposed for half car model that has four-degree-of-freedom (DOF) and as a result, the PID controller is capable to minimize the vehicle body acceleration. Although, from most of the research work on advanced suspension either active or semi-active suspension, it was found that an advanced active suspension system is able to increase the performance of ride and handling stability as well as safety of the vehicle especially the rail car. Recent study by Hany, Al Adl, Samir, and Younes (2008), regarding an active suspension system of a quarter car model using MATLAB Simulink and IPG Car Maker software based on PID control scheme also confirms this fact. Recent works on improving the dynamic performance of the vehicle have been conducted by Nurkan, Sakman, and Guclu (2008) on quarter car model of an active suspension system of light passenger vehicle using PID controller in order to improve ride comfort. Jinzhi, Fan, and Jun (2002) have performed the controller design for active vehicle suspension by using GA-based PID and fuzzy logic in controlling four DOF ground vehicles based on pitch plane of half car model. In their work, two loops controller are able to minimize the vertical acceleration and pitch acceleration of the vehicle body as well as to improve vehicle ride comfort performance with better system robustness. The study for roll plane of half car model with four DOF has also been performed by Yoshimura and Watanabe (2003). The controller is proposed in order to prevent the rollover situation and as a result, there were significant performances as compared with the passive system without rollover prevention. Zhu, Du, Zhang, and Wang (2014) developed a more sophisticated mathematical model for a roll-plane active hydraulically interconnected suspension system for verification of the new model. Data comparisons between simulations and experiments indicate high consistent response of the model and the real system thus validating the robustness and accuracy of the new mathematical model.

The aim of this work is to study and control the behaviour of a rail car suspension system when cornering or hitting railway disturbances. Although many researchers have reported the design and simulation of active suspension systems for both automobile and rail vehicles but the dynamic simulation and control of the effects of rail vehicle acceleration, suspension travel and rail holding ability on the active suspension system have not been sufficiently highlighted. The work focuses on the design and dynamic simulation of a controller for the active suspension system of a rail car in order to improve ride comfort, safety, performance of rail cars and minimize frequency of maintenance of rail car and railways via the isolation or compensation for unwanted disturbances as well as unwanted vibrations.

2. Materials and method

This research methodology utilizes a simulation design via the use of MATLAB-Simulink to investigate the suspension system behaviour. The main reason for employing dynamic simulation is that it can easily and adequately study the behaviour of a system during the design stage before development of the real system. The dynamic simulation involves several steps. The first step is the modelling process where a schematic diagram of suspension is sketched based upon the real system of suspension. Then, the system is represented in MATLAB-Simulink based on the equation of motion created from the schematic diagram. The bogie of a rail car is shown in Figure 1.

2.1. Mathematical model of the active suspension system of a rail car

The equations for the model motion were found by adding the vertical forces on the sprung mass and unspring masses. The sprung mass is represented by \( m_s \), while the tire and axles are represented by the unspring mass \( m_u \). The spring, damper and a variable force generating element placed between the sprung and unspring masses constitute suspension. From this model, the design was expanded into a full car model. The goals of designing a suspension system are to
enhance driving comfort and good rail handling with the suspension system able to reduce the
vertical body acceleration and suspension travel. The passive suspension system cannot imple-
ment these goals because in order to have a good rail handling ability, more stiff suspension is
needed meanwhile for the stability and comfort, less stiff suspension is needed. To solve this
problem, an active suspension system was introduced in which an actuator is used to control
system. This report concentrates more on an active suspension in which an actuator is fixed
parallel with the spring and damper between sprung and unsprung mass of the vehicle. The
actuator is used to take a necessary action to supply and modulate the energy in order to improve
the performance of driving comfort and ride handling. The actuator is fixed parallel with the spring
and damper. In case the actuator is not functioning well or damaged, the suspension system will
still work as the passive system without a controller. The dynamic model is aimed at obtaining the
full state feedback response of non-linear quarter car suspension model. It presents the simulation
of the full state feedback response of a quarter car active suspension model with the suspension
deflection as the output and step inputs from the road disturbance and the actuator. Figure 2
shows a two- DOF system representing a free body diagram for the car rail model. The model
consists of the sprung mass \( m_s \) and the unsprung mass \( m_u \). The tire is modelled as a spring with

Figure 2. The free body diagram for the rail car model (Ling et al., 2014).
stiffness $F_k$. The suspension system consists of a passive spring $F_k$ and a damper $F_b$ in parallel with an active control force $F$.

From Figure 2 and Newton’s law, the dynamic equation was obtained as follows:

Let,

$$F_{m_s} = m_s \ddot{x}_1 \quad (1)$$

$$F_{m_u} = m_u \ddot{x}_2 \quad (2)$$

$$F_k = k_s^l(x_2 - x_1) + k_s^m(x_2 - x_1)^3 \quad (3)$$

$$F_k = k_l(x_2 - w) \quad (4)$$

$$F_b = b_s^l(\dot{x}_2 - \dot{x}_1) \quad (5)$$

Let

$b_l = 0$ then $F_b = 0$

\[+ \sum F = mx\]

Considering

$m_s$

This implies that,

$$F_{m_s} = F_k - F + F_b \quad (6)$$

$$m_s \ddot{x}_1 = k_s^l(x_2 - x_1) + k_s^m(x_2 - x_1)^3 - F + b_s^l(\dot{x}_2 - \dot{x}_1) \quad (7)$$

$$\ddot{x}_1 = \frac{1}{m_s} \left[ k_s^l(x_2 - x_1) + k_s^m(x_2 - x_1)^3 - F + b_s^l(\dot{x}_2 - \dot{x}_1) \right] \quad (8)$$

Considering

$m_u$

$$F_{m_u} = -F_k + F - F_b + F_h \quad (9)$$

$$m_u \ddot{x}_2 = -k_s^l(x_2 - x_1) - k_s^m(x_2 - x_1)^3 + F - b_s^l(\dot{x}_2 - \dot{x}_1) + k_l(x_2 - w) \quad (10)$$

$$\ddot{x}_2 = \frac{1}{m_u} \left[ -k_s^l(x_2 - x_1) - k_s^m(x_2 - x_1)^3 + F - b_s^l(\dot{x}_2 - \dot{x}_1) + k_l(x_2 - w) \right] \quad (11)$$

For the state-space representation,

Let

$$\dot{x}_1 = x_3' \quad (12)$$
\[ \dot{x}_2 = x_4' \] 
\[ \dot{x}_1 = \frac{1}{m_2} \left[ k_1^l (x_2 - x_1) + k_2^l (x_2 - x_1)^3 - F + b_2^l (x_2 - x_1) \right] \]  
\[ \dot{x}_4 = \frac{1}{m_2} \left[ -k_1^l (x_2 - x_1) - k_2^l (x_2 - x_1)^3 + F - b_2^l (x_2 - x_1) + k_4 (x_2 - w) \right] \]

Using the state-space representation,
\[ \ddot{x} = f(x) + gu \]

The state, input and output vector are expressed as Equations (17)–(19), respectively.
\[ x = [x_1 \ x_2 \ x_3 \ x_4]^T \]  
\[ u = [u \ w] \]  
\[ y = [y] \text{ i.e. } y = x_2 - x_1 \]

where \( y_m \) is the lateral displacement (m), \( \psi_m \) is the yawing angle (deg.), \( \phi_m \) is the rolling angle (deg.), \( Y_{h,R} \) is the lateral displacement along the horizontal axis (m), \( Z_{h,R} \) is the vertical displacement (m), \( \phi_{vL,R} \) is the torsional angle (deg.), \( X, Y, Z \) are the system axes with coordinates \( x^I, y^I \) and \( z^I \).

2.2. Design using Simulink
Designing an automatic suspension system for a rail car is a control problem. The suspension system is designed and a model is used to simplify the problem to a one-dimensional spring-damper system. In this case, \( m_1, m_2 \) and \( w \) are the sprung mass, unsprung mass as well as the vertical displacement of rail, respectively.

The model parameters are given as follows:

(1) Sprung or body mass \( (m_1) = 120,000 \) kg
(2) Unsprung or wheel assembly mass \( (m_2) = 1125 \) kg
(3) Spring constant or stiffness of suspension system \( (k_1) = 10,0000 \) N/m
(4) Spring constant or stiffness of wheel assembly \( (k_2) = 800,000 \) N/m
(5) Damping constant or coefficient of suspension system \( (b_1) = 3050 \) Ns/m
(6) Damping constant or coefficient of wheel assembly \( (b_2) = 25,000 \) Ns/m

The simulated model is shown in Figure 3.

2.3. The description of model for rail disturbance input
In this case, rail bump is declared as the rail disturbance input for the model. For a rail car, only a single wheel is declared to hit the rail bump. The rail bump is assumed to have 0.05 m height presented as a step input \( w \), in MATLAB-Simulink.
2.4. Controller design using proportional–integral–derivative (PID)

PID control stands for proportional, integral and derivative control. The controls are designed to automatically eliminate steady-state errors and adjust to the system's requirements. In order to avoid the small variation of the output at the steady-state, the PID controller is also designed so that it reduces the errors by the derivative nature of the controller. A PID controller is depicted in Figure 4. The set-point is the reference point where the measurement is pre-set to be and the error generated is the difference between the set-point and measurement representing deviation from the pre-determined values (Error = set-point − measurement) while the variable being adjusted to the desired range is called the manipulated variable which is usually equal to the output of the controller. The output signal of the PID controllers often responds to changes in the measurements and reference points over time. Increasing the controller gain will make the loop go unstable. Integral action was included in controllers to eliminate this offset. With integral action, the controller output is proportional to the amount of time the error is present. Integral action eliminates offset. Controller Output = (1/Integral) ∫ Error dt. With derivative action, the controller output is proportional to the rate of change of the measurement or error calculated by the rate of change of the measurement with time hence the derivative action compensates for changes in measurement by preventing further changes in the measurement when compared to the proportional action. When a load or set-point change occurs, the derivative action causes the controller gain to move the “wrong” way when the measurement gets near the set-point hence it is often used to avoid overshoot. For the purpose of this study, the error is taken to be the difference between the measured acceleration and desired acceleration. The block diagram of the PID controller is shown in Figure 4.

The parameters for the design of PID controller for active suspension system are: body acceleration, suspension travel and rail holding.

The design of controller is expressed by Equation (20):
The signal (U) will pass through the controller that computes the derivative and integral of error signal. The signal U after passing the controller is equal to the summation of proportional gain (K_p) multiplied by the magnitude of the error and integral gain (K_i) time the integral of the error and derivative gain (K_d) multiplied by the derivative of the error. The signal U will be sent to the system and the output of Y will be obtained. The signal then will be sent again to the sensor in order to check an error occurs in the system. The PID controller was designed in the MATLAB environment using the transfer function directly and employed applied to generate a continuous-time control. Thus, the tuning of PID controller was done using following steps: generation of transfer function; importation of the obtained parameters to linear time-invariant system to confirm the tuning base and tuning the PID controller.

The PID controller is shown in Figure 5.

In designing a controller for active suspension, it is necessary to know the standard form of the control structure. The standard form of control structure consists of three items: reference, controller (PID) and subsystem. The controller reference is used as a target of the control design to keep the rail car body stable when in contact with external disturbance. It is necessary to tune the PID iteratively in order to obtain better control action for the system. In order to achieve the system requirements, it is necessary to make sure the proposed controller working properly by.
tuning the controller properly. In other words, this tuning process must be fine-tuned in order to achieve the required set such as decreasing rise time and settling time, and eliminating the steady-state error.

In case of PID controller, there are several methods used in tuning this three-term controller. Those methods are process reaction curve method and ultimate cycle method. There is another method used for tuning the PID controller, namely the “trial and error” method to get the value of $K_p$, $K_i$ and $K_d$. The potential for improved ride comfort and better road holding using PID controller design was examined.

3. Results and discussion
In this report, the trial and error method was used in order to clearly determine the function and the characteristics of proportional (P), integral (I) and derivative. Based on the characteristic PID controller as shown in Table 1, it can be concluded that the $K_p$, $K_i$ and $K_d$ are dependent of each other.

Table 1 also shows the summarization of the PID controller characteristic. It can be noted that there is a relationship between these three-term controllers because if one of the controllers is changed, it gives an effect to the other two. Increasing and decreasing the value of the proportional gain ($K_p$) only give an effect to the decreasing a rise time and the steady-state error. Tuning an integral gain ($K_i$) can also reduce the rise time of the system and able to eliminate the steady-state error. Meanwhile, a derivative gain ($K_d$) gives an effect of decreasing an overshoot and settling time, and has a small change for rise time and steady-state error. The combination of these controllers increases the stability of the system, improves the transient response as well as eliminates the steady-state error. The way to finding the values of $K_p$, $K_i$ and $K_d$ is from the trial and error method according to Zeigler–Nichols tuning rules. Zeigler–Nichols proposed the rules for determining the proportional gain $K_p$, integral time $T_i$, and derivative time $T_d$ based on the transient response characteristics of a given system. The tuning values of PID controller using Zeigler–Nichols are presented in Table 2.

3.1. Analysis of the suspension system using dynamic simulation
Simulation based on the mathematical model for quarter car using dynamic simulation was performed. The rail disturbances produce a push on the rail car constituting additional weight on the rail car body. According to the Newton’s third law of motion, a reaction force to this push produces an equivalent but opposite weight on the rail car body. This force is transmitted through the entire rail car body causing mechanical stresses to build up. The performances of the suspension system in terms ride comfort and ride handling were observed, where road disturbance is assumed as the input for the system. Parameters observed are the car body travel, wheel deflection and the car body acceleration for quarter car. Force generated by damper in case of active PID controlled system was also observed for controller accuracy with small amplitude value for vehicle body acceleration (Figure 6). The excitation force gets cancelled by the generated force by the actuator indicating the efficiency of the PID controller. By comparing the performance of the passive and active suspension system using PID control technique, it is clearly realized that the active suspension gives lower or almost no peak over shoot (amplitude) and faster settling time.

| S/N | Controller response | Rise time | Over-shoot | Setting time |
|-----|---------------------|-----------|------------|-------------|
| 1   | $K_p$               | Decrease  | Increase   | Small change|
| 2   | $K_i$               | Decrease  | Increase   | Increase    |
| 3   | $K_d$               | Small change | Decrease | Decrease    |

Table 1. Effects of controller parameters on closed loop system
Body travel can reduce the amplitude and settling time compared to passive suspension system as shown in Figure 6. The nature of the wave motion is that of a damped system.

The PID controller design approach was examined for the active and passive system in terms of the suspension travel. From Figure 7, small amplitude value for suspension travel and wheel deflection was observed for the suspension travel for active suspension system. The suspension travel is the distance the suspension spring elongates or compresses when load is removed or applied, respectively. In other words, it is a function of the spring deflection as a result of rail disturbances and irregularities. Alternate application and removal of loads may cause stress to build up around the wheel which can produce unwanted wheel deflection. The suspension travel in the active suspension system was found to reduce to more than half or almost null of its value in the passive system (Figure 7). Initially there was variation in the deflected distances but over the period of 50 s, the suspension travel maintained a constant suspension travel distance of $-3 \text{ mm}$. In contract, the passive suspension system demonstrated large deflections upon the application and removal of load resulting from rail disturbances. By implication, the smaller the deflection, the higher the ride comfort and vice versa. By introducing an active element in the suspension, it is possible to reach a better compromise than using purely passive elements.

From Figure 8, there is significant improvement in road holding ability of the active suspension system compared to the passive system especially for random rail surface. The rail holding ability is the tendency of the rail car to move steadily along the designated track without sloshing. The value of displacement which is a function of the rail holding ability with respect to time was negligibly small at the starting point and maintained stability (0 mm) for the period of 50 s compared to passive system which shows variation throughout the time interval from $-2 \text{ to } 2 \text{ mm}$. This points to the fact that the controlled active suspension system could sufficiently prevent irregular movements of the rail car thus ensuring stability, smooth and vibration-free operation on the track as opposed to the passive system that demonstrated sloshing during operation.

| S/N | Type of controller | $K_p$   | $K_i$   | $K_d$   |
|-----|-------------------|---------|---------|---------|
| 1   | P                 | 0.5 $K_{cr}$ | $\infty$ | 0       |
| 2   | PI                | 0.45 $K_{cr}$ | 0.83 $P_{cr}$ | 0       |
| 3   | PID               | 0.6 $K_{cr}$ | 0.5 $P_{cr}$ | 0.125 $P_{cr}$ |
4. Conclusion and recommendations

4.1. Conclusion
The objective of this work was achieved. A quarter rail car model with two-DOF was modelled and a PID controller was designed for its active suspension system in order to minimize the effect of rail disturbances. Also, the hydraulic dynamics were considered and simulated leading to the formulation of a dynamic model for the linear quarter rail car suspension system. The PID controller was used to test the system's performance while the Ziegler–Nichols tuning rules were used to determine proportional gain, reset rate and derivative time of PID controllers. The simulated results proved that, active suspension system with PID control improves ride comfort requiring less space for movement. However, there is significant improvement in rail holding ability observed especially for random road surface. Besides its relative simplicity in design and the availability of well-known standard hardware, the viability of PID controller as an effective tool in developing active suspension system was established.

Figure 7. Effect of suspension travel on suspension system.

![Figure 7](image)

Figure 8. Effect of rail holding ability on the suspension system.

![Figure 8](image)
4.2. Recommendations

Work-study that has been done on controlled active suspension does not consider the dynamics of the force actuators. In this work, the mathematical modeling for quarter rail car does not include the dynamic model of the actuators. Thus, the study of these actuators is important so that it can give real-time performance. The parameters used also can affect the output performance either for active or passive suspension systems. Therefore, using different type of parameters can give different output performance. The consideration of the presence of non-linearities such as hardening spring, a quadratic damping force and the tyre lift-off phenomenon in a real suspension system is important to establish a proper non-linear quarter car suspension system. Fuzzy logic systems can also be used to approximate these non-linear systems. The development of novel direct fuzzy back-stepping control methods to handle the control design problems for the systems is also recommended.

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