Semi-Active Electromagnetic Suspension System Under Varying Control Algorithms

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Abstract—**Vehicle suspension system plays a crucial role in deciding a car's performance in terms of stability, handling and comfort. Vehicle suspension systems can be grouped into three categories: passive, semi-active and active. Under these categories, the most famous and utilised car suspension system is the hydraulic actuators system. However, it is inefficient in the transfer of energy and due to its non-linear force distribution, it makes it unreliable in practice. Therefore, magnetic levitation system can be applied to the car's suspension to improve its stability on bumpy roads while providing optimal comfort. But, it comes at a greater cost compared to other systems such as hydraulics. Hence, this paper tried to integrate these two systems into a semi-active hybrid suspension system. Furthermore, to ensure optimal quality of ride and overall performance, the system was tested on three controllers namely PID, PD and LQR.**

Index Terms—**vehicle suspension system, control algorithms, magnetic suspension, PID, PD, LQR**

I. INTRODUCTION

The vehicle suspension system plays a major role in determining a ride's comfort and ease. Hence, designing an optimal suspension system results in better car performance and safety. A suspension system consists of springs, dampers and connectors that link the wheels to the vehicle. This system is responsible for ensuring maximum wheel traction with the road profile [1]. Hence, different types of suspension systems propose numerous advantages and disadvantages.

Suspension systems can be categorised into three categories passive, active and semi-active suspension. Firstly, passive systems do not need a power source to operate making them simple to implement. But, they cannot adapt to sudden excitations caused by the road fluctuations. Consequently, this will result in a lower car stability and lower safety measures [2]. In contrast, semi-active suspension systems can automatically adjust their spring stiffness and damping coefficient. This allows for better car stability under more unpredictable and uneven ground. Moreover, it helps to significantly mitigate car vibrations caused by the ground surface [3]. Furthermore, this system can be greatly optimised by the use of proper control algorithms. As for active suspension systems, they can generate a force or torque to compensate for the roll or pitch of the car as a result of a hump or uneven ground [4]. Nevertheless, utilising such suspension requires extensive energy consumption compared to passive and semi-active suspension, making it both expensive and unsustainable.

The most viable choice for vehicle suspension are hydraulic actuators. This is due to their high power-to-weight ratio, economical cost and the ability to generate force over an extended period of time without overheating. However, this system is highly inefficient as it requires constant pressurised system to operate [5]. Additionally, hydraulic actuators have a high response time due to flexible tubes. Therefore, these disadvantages makes it unrealistic to rely on a pure hydraulic system.

On the other hand, magnetic suspension systems are becoming widely available and they propose numerous advantages. This technology allows a vehicle to move at higher top speeds, have better tyre grip to the road surface and a smaller turning radius [6]. This magnetic system works by replacing traditional shock absorbers with electromagnetic actuators to overcome vibrations caused by uneven ground [7].

Thus, the proposed solution to offer the best stability and safety is a semi-active hybrid suspension system between hydraulic and electromagnetic suspensions. This allows for better wheel-ground traction, higher top speeds and better overall car stability. This paper will further investigate which controller is the optimal choice to operate the suspension system to provide the best comfort and car stability.

II. Literature Review

The evolution and advancements of a car's suspension system is due to recent needs of higher top speed, enhanced car stability and increased car ride comfort. Previously, the suspension system of automobiles in the 20th century used simple spring leaf systems [8]. Although such systems were sufficient back then, today's vehicles need to operate on different terrains and higher top speeds. Subsequently, it became evident that a need of a more complex suspension system is needed.

So, advancements and development of suspension system has lead to three broad categories: passive, active and semi-active. In order to compare each system, their stability, cost and ride comfort needs to be analysed. Passive systems are the simplest configuration of suspension with constant stiffness and damping. They are considered simple to build and maintain. Their simplicity make them cost effective relative to other systems. However, their trade-off is that they offer limited flexibility [9]. Subsequently, developments in suspension systems lead to the creation of semi-active systems. These systems offer electronically controlled damping that helps to improve the overall stability and handling of cars [9]. Lastly, active systems utilise sensors, actuators and controllers to offer the best performance, stability and comfort to a car [10]. Consequently, they increase the complexity and cost of the suspension system. This leads to sophisticated systems. Hence, it requires more energy consumption to operate compared to passive and semi-active systems.

Among the numerous suspension techniques used, hydraulic suspension systems have been widely adapted in the industry. Hydraulic control systems adjust the pressure, flow direction and flow rate of the oil in order to meet the requirements of the system in terms of direction, speed and torque [11]. Although it seems like a robust system, it suffers from flaws. To illustrate, its non-linear force due to leakage and contaminants leads to inaccuracies in the hydraulic control system [12]. Additionally, these systems ultimately convert the vibrational mechanical energy into wasted energy as heat further showcasing their inefficiencies as a suspension system [13]. Due to these factors, an alternative solution must exist to achieve higher efficiency and better car handling on different floor profiles.

Relating to the rapid developments in railway transportations, we can try to interpret this achievements and apply them to a car's suspension system instead. To further explore this topic, magnetic suspension trains (maglev) works on the concept of levitating above the track rather than the usage of wheels. The benefits of such system is the fact that due to the absence of a mechanical linkage between the tracks and train body, the frictional forces can be neglected [14]. Although it can be claimed that such system is expensive, in the long term, it requires less maintenance, longer life span and can achieve higher top speeds compared to ordinary systems [15]. Magnetic levitation also increases the handling and the smoothness of a ride making it a superior choice.

The workings of this system is to apply a repulsion or attraction force with the aid of electromagnets and applying linear motors for driving. The primary components of this system are levitation, guidance, input power transfer, propulsion, and control systems [16]. In this paper, we will discuss the performance of different controllers on a hybrid system between hydraulic and magnetic suspension system. The most known control algorithm is the Proportional, derivative and integral (PID) controller. Different variations of this controller exist such as PD neglecting the integral controller. This controller's primary goal is to eliminate errors by setting three constants proportional gain (K_p), derivative gain (K_d) and integral gain (K_i). These factors helps to control the stability of the control to successfully mitigate errors [17]. Additionally, other controllers exist such as linear quadratic regulator (LQR). It has various uses such as in inverted pendulum systems [18].

However, which is the superior controller in mitigating the most vibrations, providing the best ride comfort and stability of a car. So far, most studies either focus on magnetic or hydraulics alone in terms of control. In this paper, the hybrid combination between them will be investigated on different control algorithms like PID, PD and LQR.

III. Methodology

The study examines control algorithms for semi-active electromagnetic hybrid dampers in light vehicles, aiming to enhance comfort, minimize body acceleration, and conserve energy. These dampers utilize feedback from the control algorithm to adjust the magnetic field, thus altering the damping coefficient [19]. The objective is to evaluate how various control algorithms influence metrics of RMS acceleration, suspension travel, and performance when subjected to road disturbances. The dynamic representation of the quarter-car suspension system is based on Newton's second law of motion, which incorporates the effects of spring forces, viscous damping, and electromagnetic forces. This framework is then transformed into a state-space representation, facilitating simulations for control designs that govern the electromagnetic actuator. PD, PID, and LQR controllers are utilized to continuously adjust the damping coefficient of the electromagnetic actuators, using feedback from acceleration and suspension travel to minimize overshoot and settling time. The parameters are manually tuned. The complete model is implemented in MATLAB, and the system's performance is compared to that of a traditional coil damper suspension.
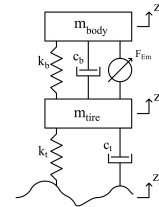
A. Dynamics of the quarter-car model

Fig. 1. Free body diagram of a two-degree-of-freedom quarter-car model.

Variable	Description
m_{body}	Mass of body supported by suspension
k_b	Stiffness of the suspension spring
c_b	Suspension damping coefficient
z_b	Body vertical displacement
F_{em}	Electromagnetic actuator force
m_{tire}	Mass of tire
k_t	Tire vertical stiffness
c_t	Tire damping coefficient
z_t	Tire vertical displacement
z_g	Road profile input

The dynamics of the quarter-car model are governed by Newton's second law of motion and are described using linear relationships. The suspension spring generates a restoring force that is proportional to the displacement of both the body and the tire. Meanwhile, the mechanical damper produces a force that is proportional to its velocity. The electromagnetic actuator is modelled as an active force that operates in parallel with the spring and damper of the body. This actuator generates a force represented by a control input term, supplementing the effectiveness of the mechanical damper. The system formulates two coupled second-order differential equations that control the vertical motion of the body and tire, represented by Equation (1) and (2).

$$
\begin{aligned}
m_{\text{tire}}\ddot{z}_t + c_t(\dot{z}_t - \dot{z}_g) + k_t(z_t - z_g) \\
- c_b(\dot{z}_b - \dot{z}_t) - k_b(z_b - z_t) = -F_{\text{em}}(t)
\end{aligned} \tag{1}
$$

$$
m_{\text{body}}\ddot{z}_b + c_b(\dot{z}_b - \dot{z}_t) + k_b(z_b - z_t) = F_{\text{em}}(t) \tag{2}
$$

The traditional passive suspension is derived from the sixth-generation Shelby Mustang GT350, released in 2016, which features a double-ball-joint MacPherson strut suspension. According to Ford Motor Company (2016), this model exhibits a weight distribution of approximately 54% at the front and 46% at the rear, with a total weight of 3,705 pounds (1,705 kg). Furthermore, the 2016 Shelby Mustang is fitted with 295/35 R19 tires, each weighing 8.16 kg.

Constant parameters (Unit)	Value
m_{body} (kg)	460.5
k_b (N/m)	34000
c_b (Ns/m)	3660
m_{tire} (kg)	13
k_t (N/m)	30000
c_t (Ns/m)	451
State Variables (Unit)	—
z_b (m)	—
z_t (m)	—
z_g (m)	—

The manufacturer does not provide comprehensive dynamic data regarding tire parameters and the mechanical damper; thus, the values were derived through an estimation approach. The tire mass was estimated using typical values for a 19-inch ultra-high-performance tire, which range from 12 to 14 kg, selecting a representative value of 13 kg. Given the need for high vertical rigidity, a vertical stiffness value of 300 kN/m was chosen, resulting in physically realistic tire deflections. For high-performance radial tires, a damping coefficient of 451 Ns/m was selected. According to [20], for a mechanical damper utilized in sports cars, using a representative damping ratio of $\zeta = 0.4$ yields a resultant damping coefficient of

$$
c_b = 2\zeta\sqrt{k_b m_b} = 2(0.4)\sqrt{34\,000 \times 460.5} \approx 3660\,\text{Ns/m}. \tag{3}
$$

This value aligns with the performance expectations for such high-performance vehicles, where optimal damping characteristics are crucial for enhancing handling and stability.

The dynamics of the quarter-car equations were formulated into a state-space representation, facilitating systematic analysis, the implementation of diverse control strategies, and comprehensive simulation. This approach enhances the ability to explore the system's behaviour and optimize control responses in a structured manner. The state variables are

$$
x = [z_b,\ z_t,\ \dot{z}_b,\ \dot{z}_t]^T \tag{4}
$$

, used to capture the displacement and velocity of the body and tire, while the input vector include the road profile displacement and the electromagnetic force

$$
u = [F_{em},\ z_g,\ \dot{z}_g]^T \tag{5}
$$

. The standard state space form $\dot{x} = Ax + Bu$ and $y = Cx + Du$, where

$$
A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_b}{m_b} & \frac{k_b}{m_b} & -\frac{c_b}{m_b} & \frac{c_b}{m_b} \\ \frac{k_b}{m_t} & -\frac{k_b}{m_t} & \frac{c_b}{m_t} & -\frac{c_b}{m_t} \end{bmatrix}, \quad B = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ \frac{1}{m_b} & 0 & 0 \\ -\frac{1}{m_t} & \frac{k_t}{m_t} & \frac{c_t}{m_t} \end{bmatrix}
$$

$$
C = \begin{bmatrix} -\frac{k_b}{m_b} & \frac{k_b}{m_b} & -\frac{c_b}{m_b} & \frac{c_b}{m_b} \end{bmatrix}, \quad D = \begin{bmatrix} \frac{1}{m_b} & 0 & 0 \end{bmatrix}
$$

The state space formulation has several assumptions, primarily the assumption of linearity within the system. This formulation does not account for roll, pitch, or lateral dynamics, potentially limiting its applicability in scenarios where these factors may be significant.

An accurate road profile that adheres to the UAE's technical standards for speed bumps is essential for evaluating the performance of the suspension system and simulating real-world conditions. In the UAE, speed bumps have a height of approximately 3 to 4 inches (7.6 to 10.2 cm) and a length ranging from 500 mm to 250 mm. The road input utilized a half-sine wave with a length of 0.375 m and a height of 0.089 m, accompanied by a vehicle travelling at a speed of 7.2 km/h, $z_g(t) = h\sin^2(\pi v t/2L) = 0.089\sin^2(2\pi t/0.75)$.
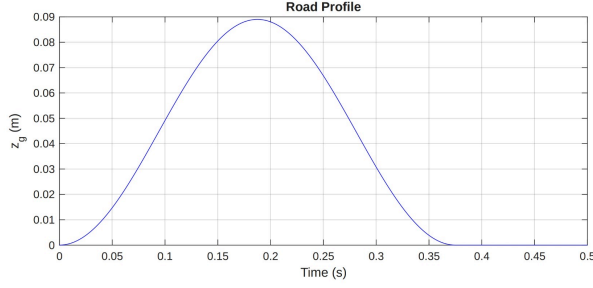
Fig. 2. Road profile input wave.

B. Controller

The controller developed enhanced ride comfort while accounting for the physical limits of the suspension system by defining constraints and validating the system's stability through simulations. The objective was to reduce jerk, acceleration, and excess energy from the electromagnetic actuator, which highlighted the stability and effectiveness of the controllers. The PID controller uses a closed-loop feedback mechanism, utilizing proportional gain to generate an output based on the difference between the current system value and the desired set point. The integral gain addresses accumulated errors, while the derivative gain reduces oscillations by anticipating errors based on the rate of change [21]. According to [22], the Linear Quadratic Regulator (LQR) controller is a strategy designed to minimize a cost function expressed as a quadratic equation. This function represents the sum of deviations from the desired values, containing weighting factors Q and R that penalize both the deviations and the control effort, respectively. The system was simulated using MATLAB to allow implementation of the state-space model and controllers. The MATLAB script quantified the ride comfort by using RMS acceleration as an indicator of sustained vibration and shock intensity, suspension travel to ensure the system remained within mechanical limits, and the energy consumption to ensure the reliability of the semi-active hybrid electromagnetic suspension.

IV. RESULTS

Based on the MATLAB simulation software, the controlled semi-active electromagnetic suspension is compared with the passive suspension system under identical conditions. Several evaluation indicators are analysed including the displacement, acceleration, and the suspension deflection.
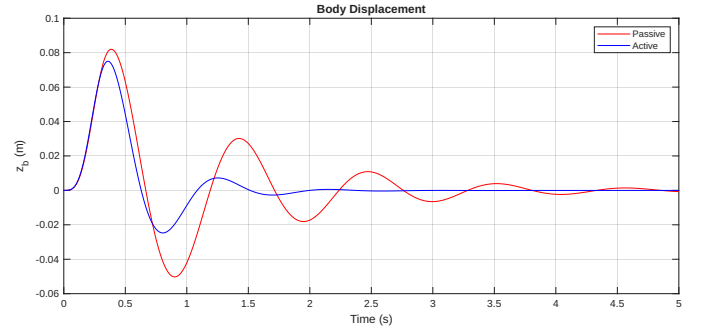

Fig. 3. PD controlled semi-active electromagnetic suspension body displacement vs time graph.
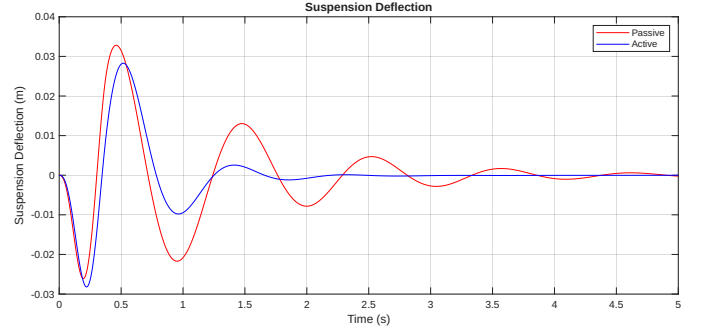

Fig. 4. PD controlled semi-active electromagnetic suspension's suspension deflection vs time graph.
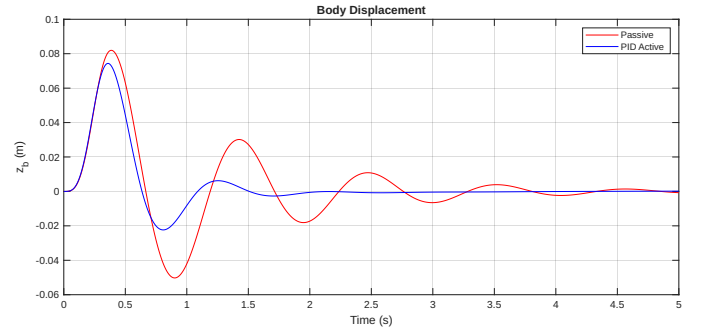

Fig. 5. PID controlled semi-active electromagnetic suspension body displacement vs time graph.
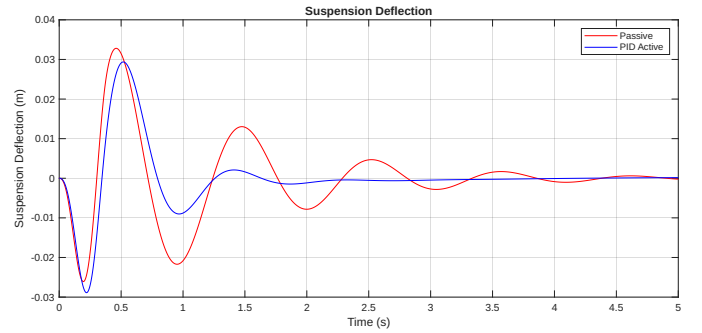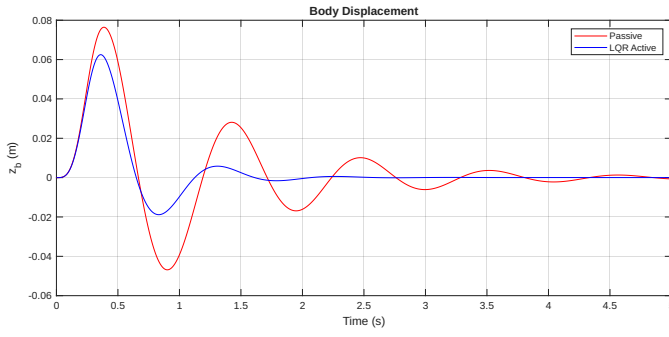

Fig. 6. PID controlled semi-active electromagnetic suspension's suspension deflection vs time graph.

Fig. 7.   LQR controlled semi-active electromagnetic suspension body displacement vs time graph.
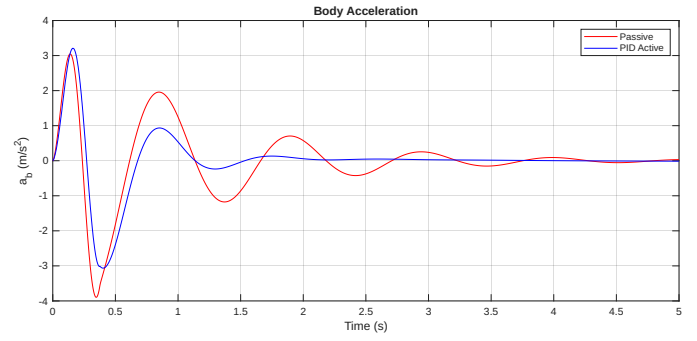

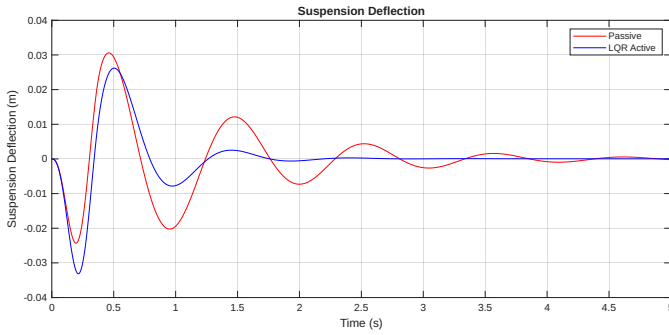Fig. 10.   PID controlled semi-active electromagnetic suspension's acceleration vs time graph.

Fig. 8.   LQR controlled semi-active electromagnetic suspension's suspension deflection vs time graph.
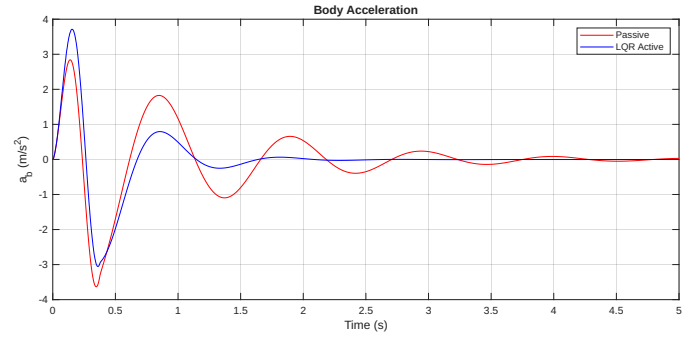

Fig. 11.   LQR controlled semi-active electromagnetic suspension's acceleration vs time graph.

The LQR controller achieved the smallest body displacement amplitude among all tested systems, with a peak of just 0.0635 meters. In comparison, the PID and PD controllers yielded lesser reductions. The LQR controller exhibited better performance in minimizing vertical chassis motion, providing tighter control over suspension and tire deflection. This resulted in smoother displacements and a quicker decay of oscillations. In contrast, while the PD and PID controllers did manage to reduce displacement to some extent, they experienced larger overshoots and slower settling times.

The RMS body acceleration showed a reduction for all active controller systems compared to the passive suspension system. The PD controller demonstrated the most significant improvement, achieving a 20% reduction in RMS acceleration, followed closely by the PID controller at 17.9%, and the LQR controller at 10.5%. Additionally, both the PD and PID controllers reduced peak acceleration, while the LQR controller exhibited a slight increase in peak acceleration. The PD controller excelled in minimizing peak acceleration. The results indicate that the PD and PID offer better transient and comfort performance, especially in reducing high-frequency vibrations transmitted to the body.
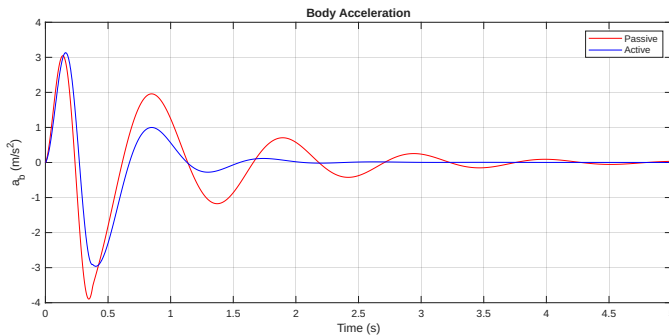

Fig. 9.   PD controlled semi-active electromagnetic suspension's acceleration vs time graph.
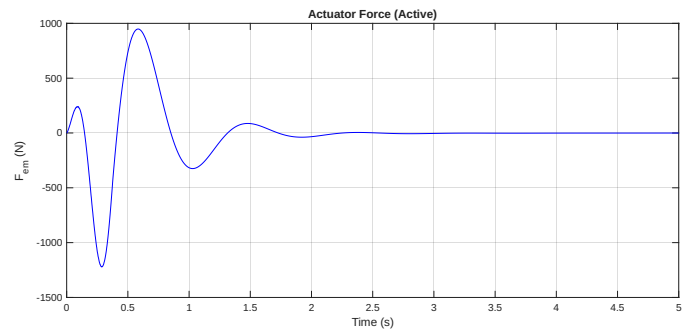

Fig. 12.   PD controlled semi-active electromagnetic suspension's actuator force vs time graph.
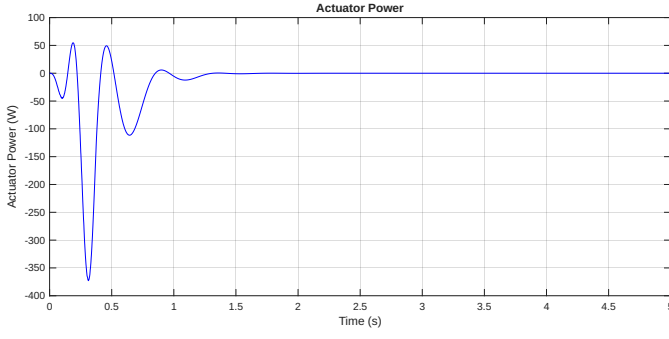
Fig. 13. PD controlled semi-active electromagnetic suspension's actuator power vs time graph.
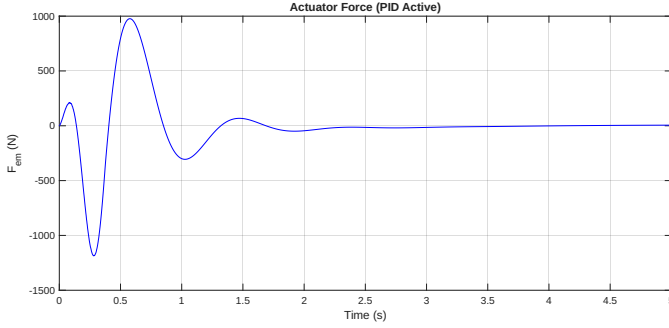

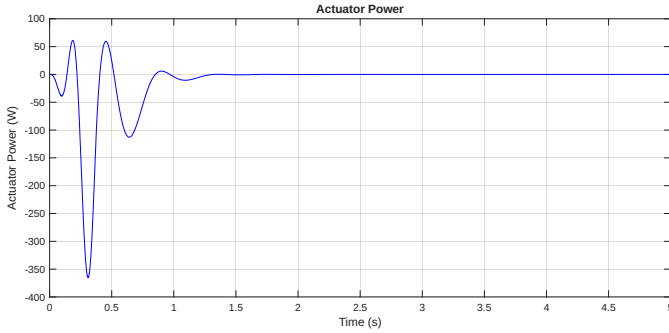Fig. 14. PID controlled semi-active electromagnetic suspension's actuator force vs time graph.

Fig. 15. PID controlled semi-active electromagnetic suspension's actuator power vs time graph.
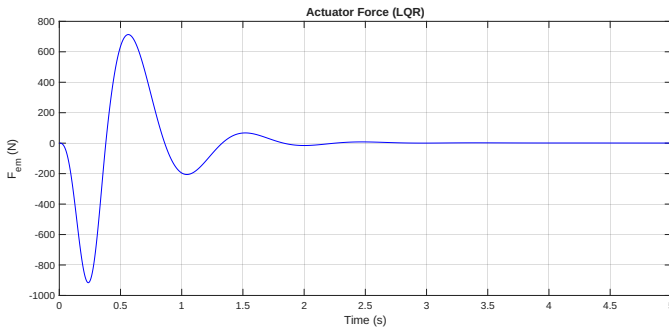

Fig. 16. LQR controlled semi-active electromagnetic suspension's actuator force vs time graph.
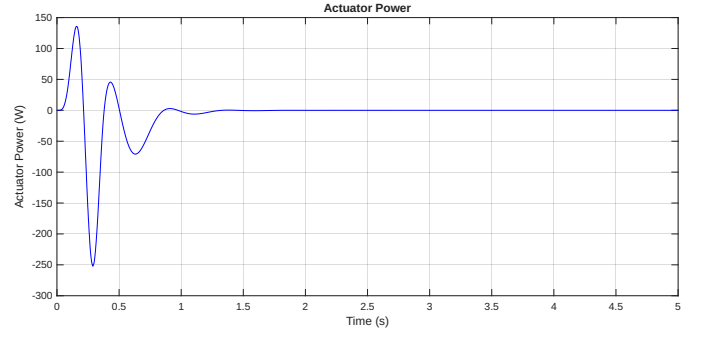

Fig. 17. LQR controlled semi-active electromagnetic suspension's actuator power vs time graph.

The LQR actuator force maintains a small force range, whereas the forces from the PD and PID controllers can reach up to 1500N. The PD and PID controllers exert greater strain on the actuator, while the LQR demonstrates greater efficiency. Additionally, the power consumption of the PD and PID controllers tends to be more erratic, in contrast to the structured usage of the LQR. Thus, the LQR controller is the more power-efficient, consistent, and effective option. The findings indicate that the Linear Quadratic Regulator (LQR) controller is more beneficial for systems constrained by thermal limits, energy consumption, and power availability.

The time-domain vibration behavior reflects a trade-off between different controllers. The PD and PID controllers significantly reduce vibration peaks and facilitate a quicker decrease in acceleration oscillations, consequently enhancing ride comfort. In contrast, LQR controllers deliver an improved displacement profile with minimal oscillatory behavior. While PD and PID focus on vibration control, LQR excels in state response and effectively reduces displacement. Therefore, PD and PID are superior for comfort-oriented vibration control, whereas LQR is more effective for positional regulation.

V. DISCUSSION

The objective of this study was to evaluate and analyse the impact of various control algorithms employing an electromagnetic actuator on the dynamic behavior of a quarter-car suspension, as compared to a passive suspension system. The effectiveness of the different control strategies for the semi-active hybrid suspension was assessed in terms of body displacement, body acceleration, settling behavior, and actuator demand. These metrics demonstrate the performance requirements for ride comfort and stability, aiming to reduce acceleration, attenuate disturbances, and ensure overall feasibility.

The PD controller implemented for the electromagnetic actuator in the semi-active hybrid system demonstrates considerable progress over the passive system. It has decreased the initial peaks of body displacement and velocity while improving the rate of oscillation attenuation. This performance represents damping augmentation, where the increase in derivative gain enhances the system's damping ratio and

promotes greater energy dissipation. The overall RMS acceleration was reduced from 0.985 m/s² to 0.788 m/s², representing a decrease of 20%. This change reflects an enhancement in ride comfort throughout the measured duration. Although there was a slight increase in the peak acceleration, it decreased rapidly, indicating an overall improvement in dynamic performance.

The PID controller demonstrated a stronger resistance to deviations in the road profile compared to the PD controller, due to the presence of the integral gain. As shown in the displacement graphs above, the PID controller effectively reduced both steady-state error and oscillations. However, the integral gain can introduce some instability into the system, resulting in minor overshoots and a lower damping effect relative to the PD controller. While the dynamics of integral wind-up may enhance steady-state performance, they can also aggravate the response to initial conditions. The PID controller has proven to be more effective in reducing body acceleration compared to the passive system, although it is less effective than the performance of the PD-controlled system. The root mean square (RMS) acceleration improved from 0.919 m/s² to 0.877 m/s², reflecting a 17.9% reduction. However, the peak acceleration remained higher than that of the PD-controlled system. This indicates that while the PID controller is proficient at rejecting low-frequency disturbances, ride comfort is constrained by the integral gain, unless an anti-windup algorithm is implemented.

The improvements achieved by the LQR controller are significantly impacted by the selection of the weighting matrices Q and R; when chosen appropriately, the LQR controller can surpass the performance of PD and PID controllers. In this study, the selected weighting matrices delivered a reduction of 10.5% in the RMS acceleration, decreasing it from 0.919 m/s² to 0.822 m/s². This outcome suggests that the weighting matrices facilitated aggressive state corrections with minimal penalties on control effort, resulting in sharp and noticeable adjustments. This is evident in the peak acceleration, which increased to 3.7 m/s² compared to a passive peak acceleration of 3.63 m/s². While the LQR controller exhibited rapid settling and a favourable decay rate, it also indicated that the weighting matrices were not appropriately selected. As a result, the LQR-controller system did not prioritize ride comfort and under-penalized control effort, leading to increased jerk and acceleration while improving displacement.

In summary, this study confirmed that the semi-active hybrid electromagnetic suspension system can enhance performance and ride comfort compared to passive hydraulic suspensions; however, its effectiveness is influenced by the choice of control algorithms. The findings indicate that the PD controller stands out as a reliable solution, producing significant reductions in both displacement and acceleration, leading to improved ride comfort. Meanwhile, the PID controller offers enhancements in steady-state error but is prone to overshoot due to its integral gain, resulting in minimal damping and potential instability. The LQR controller demonstrates strong stability and ride comfort, exhibiting the quickest decay of oscillations; however, it relies heavily on the proper selection of weighting

matrices for precise modelling. Although it has the potential to outperform both the PD and PID controllers in terms of damping performance and comfort, poor tuning in this study produced unsatisfactory results. Overall, the PD controller delivered the most substantial improvements, followed by the PID and LQR controllers, which require further refinement.

VI. CONCLUSION

This paper discussed different types of suspensions namely hydraulics and magnetic suspensions. Additionally, different control algorithms like PID, PD and LQR were tested on the hybrid combination between hydraulic control systems and magnetic levitation. The purpose of this was to ensure maximum damping of vibrations caused by uneven terrain, enhance the car handling and stability while providing the optimal ride comfort.

As discussed in this paper, PD is the optimal choice as a controller for the semi-active hybrid suspension system. Although PID introduces the integral gain that reduces steady state errors, it can reduce the stability of the system hence becoming unreliable. As for LQR is heavily dependent on its tuning parameters. Even though it achieved the quickest decay of vibrations among the three controllers, that was not enough for it to outperform both PD and PID.

In conclusion, this paper covered different control algorithms and their performance on semi-active hybrid suspension system illustrating that the best performance is achieved by PD, PID and LQR in this specific order.

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