# OPTIMIZATION OF PID CONTROLLER USING PSO ALGORITHM FOR ACTIVE SUSPENSION SYSTEM UNDER VARIOUS ROAD DISTURBANCES

**NOR HANI BINTI MD DESA**



**BACHELOR OF ELECTRICAL ENGINEERING WITH HONOURS**
**UNIVERSITI TEKNIKAL MALAYSIA MELAKA**

**2024**

# OPTIMIZATION OF PID CONTROLLER USING PSO ALGORITHM FOR ACTIVE SUSPENSION SYSTEM UNDER VARIOUS ROAD DISTURBANCES

**NOR HANI BINTI MD DESA**

**A report submitted**
**in partial fulfilment of the requirements for the degree of**
**Bachelor of Electrical Engineering with Honours**



**Faculty of Electrical Technology and Engineering**



**UNIVERSITI TEKNIKAL MALAYSIA MELAKA**

**2024**

# DECLARATION

I declare that this thesis entitled "OPTIMIZATION OF PID CONTROLLER USING PSO ALGORITHM FOR ACTIVE SUSPENSION SYSTEM UNDER VARIOUS ROAD DISTURBANCES" is the result of my own research except as cited in the references. The project report has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

Signature : Hani

Name : NOR HANI BINTI MD DESA

Date : 19/6/2024

## APPROVAL

I hereby declare that I have checked this report entitled "Optimization of PID Controller using PSO algorithm for Active Suspension System under Various Road Disturbances", and in my opinion, this thesis fulfils the partial requirement to be awarded the degree of Bachelor of Electrical Engineering with Honours.

Signature :

Supervisor Name : MOHD FAIRUS BIN ABDOLLAH

Date : 5/7/2024

## DEDICATIONS

To my beloved mother and father.

# ACKNOWLEDGEMENTS

I would like to express my special appreciation and thanks to my supervisor Mr. Mohd Fairus Bin Abdollah, you have been a tremendous mentor for me. I want to express my gratitude for supporting my research. Your guidance on my job and research has been helpful. Additionally, I would want to express my gratitude for allowing my defense to be a fun experience and for your insightful remarks and recommendations. I express my gratitude to Universiti Teknikal Malaysia Melaka (UTeM) for granting access to the research platform. A special thanks to my family and friends who gave their support to complete this project. Words cannot express how grateful I am to my family and friends for all the sacrifices that you have made on my behalf. Your prayer for me was what sustained me thus far. Finally, I thank my God, for letting me through all the difficulties. I have experienced Your guidance day by day. You are the one who let me finish my degree. I will keep on trusting You for my future.

## ABSTRACT

This report presents an active suspension system for a car to improve handling and comfort on the road in various road disturbance scenarios. The main limitation of passive suspension systems lies in their inherent compromise between ride comfort and safety, resulting from their inability to dynamically adjust to varying road conditions. Efforts to enhance ride comfort often led to trade-offs that may compromise safety, and vice versa. This duality necessitates a more adaptable and flexible solution. Active suspension systems emerge as a transformative methodology, allowing real-time adjustments and dynamic modifications to damping characteristics. This capability effectively separates the compromise between ride enjoyment and safety, enabling an optimal equilibrium by adaptively responding to fluctuations in road conditions. Our objectives include developing and analyzing a quarter-car model for the active suspension, optimizing the PID controller using the PSO algorithm, and comparing system performance with passive and Ziegler-Nichols-tuned active suspensions. To minimize the sprung mass acceleration when the vehicle is subjected to various road situations, a Proportional-Integral-Derivative (PID) controller based on particle swarm optimization (PSO) is presented in this research. MATLAB is used to generate quarter-cars and models of road bump disturbance for the dynamic investigation of the suspension system of the car. The advantages of the proposed PSO-based PID controller over the Ziegler-Nichols tuning method for the active suspension system are illustrated by simulation results in MATLAB, demonstrating significant improvements in gain values and tire-road contact, thereby enhancing both ride comfort and safety by adaptively responding to road conditions in real time.

## *ABSTRAK*

Laporan ini membentangkan sistem penggantungan aktif untuk kereta untuk meningkatkan pengendalian dan keselesaan di jalan raya dalam pelbagai senario gangguan jalan raya. Had utama sistem penggantungan pasif terletak pada kompromi yang wujud antara keselesaan dan keselamatan perjalanan, hasil daripada ketidakupayaan mereka untuk menyesuaikan diri secara dinamik dengan keadaan jalan yang berbeza-beza. Usaha untuk meningkatkan keselesaan perjalanan selalunya membawa kepada pertukaran yang boleh menjejaskan keselamatan, dan sebaliknya. Dualitas ini memerlukan penyelesaian yang lebih mudah disesuaikan dan fleksibel. Sistem penggantungan aktif muncul sebagai metodologi transformatif, membenarkan pelarasan masa nyata dan pengubahsuaian dinamik pada ciri redaman. Keupayaan ini secara berkesan memisahkan kompromi antara keseronokan perjalanan dan keselamatan, membolehkan keseimbangan optimum dengan bertindak balas secara adaptif terhadap turun naik dalam keadaan jalan raya. Objektif kami termasuk membangunkan dan menganalisis model suku kereta untuk penggantungan aktif, mengoptimumkan pengawal PID menggunakan algoritma PSO dan membandingkan prestasi sistem dengan penggantungan aktif pasif dan Ziegler-Nichols-tala. Untuk meminimumkan pecutan jisim sprung apabila kenderaan tertakluk kepada pelbagai situasi jalan raya, pengawal Proportional-Integral-Derivative (PID) berdasarkan pengoptimuman kawanan zarah (PSO) dibentangkan dalam penyelidikan ini. MATLAB digunakan untuk menjana suku-kereta dan model gangguan benjolan jalan untuk penyiasatan dinamik sistem penggantungan kereta. Kelebihan pengawal PID berasaskan PSO yang dicadangkan berbanding kaedah penalaan Ziegler-Nichols untuk sistem suspensi aktif digambarkan oleh hasil simulasi dalam MATLAB, menunjukkan peningkatan ketara dalam nilai keuntungan dan sentuhan jalan tayar, dengan itu meningkatkan keselesaan dan keselamatan pemanduan dengan bertindak balas secara adaptif kepada keadaan jalan dalam masa nyata.

**TABLE OF CONTENTS**

## LIST OF TABLES

## LIST OF FIGURES

## LIST OF SYMBOLS AND ABBREVIATIONS

| | | |
|---|---|---|
| PID | - | Proportional Integrated Derivative |
| PSO | - | Particle Swarm Optimization |
| Ki | - | Integral gain |
| Kp | - | Proportional gain |
| Kd | - | Derivative gain |
| ZN | - | Ziegler Nichols |
| LQR | - | Linear Quadratic Regulator |
| GA | - | Genetic Algorithm |
| ABC | - | Artificial Bee Colony |
| DOF | - | Degrees of Freedom |
| PL | - | Positive Large |
| PM | - | Positive Medium |
| PS | - | Positive Small |
| ZE | - | Zero |
| NS | - | Negative Small |
| NM | - | Negative Medium |
| NL | - | Negative Large |
| IMC | - | Internal Model Control |
| AVSS | - | Active Vehicle Suspension System |
| IAE | - | Integrated Absolute Error |
| ITAE | - | Integral of Time multiplied by the Absolute Error |
| ITSE | - | Integral of Time multiplied by the Squared Error |
| ISE | - | Integral Squared Error |

## LIST OF APPENDICES

# CHAPTER 1

# INTRODUCTION

## 1.1 Research Background

Active suspension systems strongly react towards changes in the road profile due to their ability to provide energy that can be utilized to develop relative motion. Sensors to monitor suspension fluctuations, such as body velocity, suspension displacement, wheel velocity, and wheel or body acceleration, are frequently included in active suspension systems. In the active suspension system, actuators are added to the passive components to provide extra forces dictated by a feedback control law. There are different control strategies, such as optimal state feedback and fuzzy control. In this paper, the research focuses on the PID controller as one alternative for designing a reliable active suspension.

A sound suspension system requires careful consideration of several performance aspects. These qualities have to do with force distribution, suspension movement regulation, and body movement regulation. The suspension should shield the body from inertial disturbances brought on by braking, acceleration, and corners. In addition, for the comfort of the passengers, the suspension should be able to reduce the vertical force transferred. It is important to discover a different method of managing the vibration of the car's suspension by utilising a PID controller since, as is well known, road roughness is the primary cause of vehicle vibration.

PID controller, on the other hand, is one method to obtain the feedback of specific systems. The feedback control design that is most often used is the PID controller. PID stands for Proportional-Integral-Derivative, which describes how the three terms work together to create a control signal from an error signal. The desired closed-loop dynamics are obtained by adjusting the three parameters Kp, Ki, and Kd, often iteratively by "tuning" and without specific knowledge of a plant model. The proportional term is frequently sufficient to provide stability. A step disturbance, which is frequently a startling specification in process control, can be rejected thanks to the integral term. A response's shaping or dampening is accomplished via the derivative term. PID controllers represent the most popular category of control systems.

The active suspension system performs better when the Particle Swarm Optimization (PSO) algorithm is used since it makes it easier to alter the PID parameters iteratively. The suspension system is guaranteed to adjust smoothly to changing road conditions and disturbances thanks to this dynamic optimization process. The active suspension system, which demonstrates the adaptability and efficiency of PID controllers in the field of control systems engineering, strikes an ideal balance between stability, disturbance rejection, and response shaping thanks to the cooperative synergy of PID control and the PSO algorithm.

## 1.2 Motivation

The study of active suspension systems in a situation of various road disturbances shows considerable potential to enhance the comfort and performance of automobiles using active suspension systems that can adjust to the dynamics of the vehicle and the road. To optimize the settings of the Proportional-Integral-Derivative (PID) controller, which controls the suspension force, the research will employ the Particle Swarm Optimization (PSO) technique. Under diverse road disturbances, the study will evaluate the PID controller based on PSO with alternative approaches and methodologies. There is a need to resolve the trade-off between road holding ability, handling stability, and ride quality for passenger and driver satisfaction. Additionally, the study will amplify the PSO algorithm and active suspension technology, which are critical to Malaysia's automobile sector. By improving handling and stability, these technologies go beyond simple comfort and are essential in creating a remarkable driving experience. With the US expected to be at the forefront of implementing this automotive innovation, these technologies have the potential to completely transform a variety of vehicle models, from luxury cars to general-purpose vehicles. The main objective is to rethink travel, making it more technologically sophisticated, safer, and comfortable.

## 1.3 Problem Statement

The main limitation of passive suspension systems is evident in the inherent compromise between ride comfort and safety considerations that occur concurrently. The inherent immutability of passive systems results in inferior performance due to their inability to dynamically adjust to varying road conditions. It is worth noting that endeavors to improve the comfort of rides frequently involve making trade-offs that may compromise safety considerations, and conversely. The existence of this duality highlights the necessity for a solution that is more adaptable and flexible. In response, the introduction of active suspension systems has developed as a transformative methodology. Active suspension systems allow for real-time adjustments, which facilitate dynamic modifications to damping characteristics. This capability effectively separates the compromise between ride enjoyment and safety. This invention exhibits the potential to achieve optimal equilibrium by adaptively responding to fluctuations in road conditions and presenting a transformative departure from the traditional trade-offs associated with passive systems.

## 1.4 Objectives

The lists of objectives were determined as follows:

a) To develop and analyze the quarter car model for active suspension system to specific road disturbances to give comfort to the passengers.
b) To investigate and optimize PID controller using PSO algorithm.
c) To compare the system performance between the optimized PID controller and classical PID controller under various road disturbances.

### 1.5 Scope of the Research

The scopes of the research are as follows:

1. In this study, simulation of two degree of freedom (DOF) quarter vehicle suspension system is developed via MATLAB Simulink environment. The parameters of the model are selected based on previous research.
2. The performance of the active suspension system is evaluated based on two road profiles which are single bump and two bumps.
3. A PID controller for active suspension system is designed and implemented to regulate the suspension travel using the PSO algorithm.
4. A ride comfort comparison is made between the performance of the optimized PID based on PSO algorithm and the classical PID controller.

## 1.6 Thesis Organization

This thesis is organized into five chapters, including the introduction. Each chapter is different from the others, and each one is discussed along with the theory needed to understand it. The research background, problem statement, objective, and scope of the research are all defined in the chapter introduction to better define specific aspects of the active suspension system.

The background and basic literature review will be covered in Chapter 2, whichwill include previous study or research material on the problem of secure active suspension system, the method that has typically been used in previous research, and the distinction between the PSO algorithm and MATLAB Simulink.

The methodology involved in active suspension system using PSO algorithm and MATLAB Simulink, which is all the identified objective function with the related formulation, will be described in detail in Chapter 3. The overall MATLAB Simulink and PSO algorithm and process were also discussed in the chapter.

Chapter 4 shows the simulation and the resulting case using MATLAB Simulink and the PSO algorithm. A solution is found using the process. A reliability study,comparison tables, and convergence characteristics are also studied.



Chapter 5 concludes the work performed with significant results. It also providesan insight into the future studies that can be done on this subject.

All costs related to the completed report such as printing, binding, typing, and photocopying will be fully borne by the student.

# CHAPTER 2

## LITERATURE REVIEW

### 2.1. Introduction

This chapter primarily describes previous research on suspension systems that has been done and defined over the last few years. The past and present situation of the issue is also covered in this chapter, along with a summary of earlier research on closely related issues.

### 2.2. Research Paper

In research paper [1], focusing to maintain ride comfort for the passengers and maintain tire contact with the road in the event of a vehicle disturbance by minimizing the acceleration of the vehicle body that is transferred to them. Three fundamental methods are passive, semi-active, and vehicle suspension systems can be controlled with active suspension systems. Active suspension systems use actuators to provide the desired control action between the vehicle's sprung body and wheel axle, reducing the impact of road disturbances on passengers. As a result, they are more effective than semi-active and passive suspension systems. This paper proposes an optimal proportional-integral-derivative (PID) controller based on particle swarm optimization to reduce vertical body acceleration and enhance vehicle active suspension systems' capacity for road handling. A PSO-based PID controller is suggested in the current work to enhance the active suspension system of the car's dynamic performance. The highest gains are achieved by taking into consideration the goal of minimizing vehicle body acceleration. Nonetheless, two kinds of road bump disturbances are examined: one includes a single bump, and the other features two bumps with varying heights and durations. The outcomes clearly show that by according to the simulation results, the suggested PSO-based PID controller performs more effectively in terms of position, velocity, and acceleration as well as for keeping better road-tire contact than the ZN approach based on the passive system and active system under situations of uneven road disturbance.

In another research paper [2], vehicle shock absorption is taken into account by the suspension system. In addition to supporting the vehicle's weight, it works to minimize or eliminate vibrations that could be caused by several things, including uneven road surfaces, aerodynamic forces, and uneven tire/wheel assemblies. The quarter car passive and semi-active suspension systems are presented in this research using Simulink models. In the suspension displacement domain, the fuzzy system that has been optimized performs better than any other system. Consequently, out of all the control algorithms, the optimized fuzzy system provides the most comfortable ride. A passive system experiences no stabilization at all when the suspension is displaced. The techniques of ground hook and skyhook control algorithms are combined in fuzzy hybrid algorithms. The optimized fuzzy logic system shows the best stabilizing times in terms of tire displacement, while the ground hook provides the least amount of overshoot. However, in terms of vehicle handling, the optimized fuzzy system is still comparable to the ground hook system. As a result, to design a semi-active suspension system, the research successfully applies hybrid artificial intelligence techniques. Semi-active and passive suspension systems have been developed in Simulink. Regarding road handling and comfort during the ride, the optimized fuzzy logic-controlled system performs significantly better than the passive system and alternative control schemes.

In their paper [3] a suspension system is to obtain ride comfort and maintain the tires in contact with the road during road disturbances, and to minimize the vertical acceleration of the vehicle's body that is experienced by the occupants. Moreover, in this research, the ideal damping force for the vehicle's active suspension system is suggested by optimization of particle swarms (PSO) using optimal 2PID controllers. This leads to improvements in road handling, passenger comfort, and ride stability. MATLAB Simulink was utilized to assist with the modeling, simulation, and optimization processes. Furthermore, the suggested approach has a good improvement in ride comfort and a high level of vibration suppression on the vehicle. Based on the simulation results, the results clearly show that active suspension outperformed passive suspension in terms of rising time, settling time, and overshooting.

In this research paper [4], bump analysis is performed on a 3-DOF quarter vehicle model, considering the direction of vibrations operating on the human body in the coordinate system's translational vertical direction. By following ISO-2631-1:1997(E), a review is conducted of the ride comfort levels, performance metrics, seat acceleration, chassis acceleration, and seat displacement. The International Organization for Standardization's ride comfort standards are used to compare the weighted accelerations for active and passive suspension systems. As the results, its demonstrate that the fuzzy logic controlled active suspension system, whose accelerations are within the comfortable zone (less than 0.315 m/sec$^2$) is shown in Table 2.1, has a higher degree of ride comfort than the passive suspension system, whose accelerations are within the extremely uncomfortable zone (greater than 2m/sec$^2$). This analysis shows that the occupant's peak-to-peak displacement is 76.9% less in the fuzzy-controlled system than in the passive suspension system. The fuzzy-controlled system is also more stable. Furthermore, compared to passive suspension systems, it is found that nearly 84% of accelerations are dampened out before reaching the human body with fuzzy active systems, improving passenger comfort during the ride.

| **R.M.S accelerations** | **Comfort Zones** |
|---|---|
| Less than 0.315m/sec$^2$ | Not uncomfortable |
| 0.315m/sec$^2$ to 0.63m/sec$^2$ | A little uncomfortable |
| 0.5m/sec$^2$ to 1m/sec$^2$ | Fairly uncomfortable |
| 0.8m/sec$^2$ to 1.6m/sec$^2$ | Uncomfortable |
| 1.25m/sec$^2$ to 2.5m/sec$^2$ | Very uncomfortable |
| Greater than 2m/sec$^2$ | Extremely uncomfortable |

Table 2.1: Comfort reactions to vibration environments [4].

Case study in [5], a new proportional integral-derivative (PID) controller design is to balance the two competing requirements of passenger ride comfort and road handling for the active vehicle suspension system (AVSS) of a quarter-car model. This design is based on the widely recognized Internal Model Control (IMC) theory. Researchers have categorized control methods into several categories: robust control, adaptive control, fuzzy logic control, nonlinear optimum control, robust control, and PID control. Using the IMC-PID approach to build the A VSS's feedback system, it seeks to produce good vehicle handling, stability, and comfort performance that achieves a desirable level. Nonetheless, data shows that the recommended active suspension system has lowered all peak, overshoot, and settling times for road holding and passenger comfort when compared to the passive system. As a result, it showed a notable enhancement in ride comfort and road-holding capability compared to the passive suspension system.

The research article by [6]. In this study, an observer design is utilized to predict the road profile, and the sliding mode is selected as the control technique. The principal objective behind developing an active suspension system is to enhance ride comfort by reducing the disturbance caused by bumps and uneven terrain. So, from the result, the passive suspension system's response indicates that it is stable but takes some time to stabilize for a unit step input and it also fails to reduce the applied force in the event of two bumpy road disturbances. This leads to the system's sprung mass deflection. While for the active suspension for unit step and two bump road profile is successful to reduce the sprung mass's deflection and acceleration, indicating that it is effective in enhancing ride comfort. The outcomes clearly show that active suspension responds far better with a disturbance observer than it does with neither the observer nor the passive suspension system. However, the least effective suspension to absorb any disruption to the system is passive suspension. In addition, the boundary layer, which is modifiable by changing the sliding gain in the controller, and the continuous switching function have been used to solve the chattering issue.

Another case study in [7], automotive suspension systems are vital to maintaining a vehicle's safety and comfort. The article explains the fundamental requirements of a suspension system and how semiactive, active, and passive suspensions can be used to meet those objectives. Passive suspensions are unable to adjust their responses because spring constants and damping coefficients have set values. Semiactive suspensions are limited to generating force in a single direction based on the suspension's inherent movement in response to a road disturbance, whereas active suspensions can generate force in any direction and of any magnitude, as shown by their range in all four quadrants on the Figure 2.1. This demonstrates the adaptability of active suspensions; but, due to the closed-loop feedback control necessary for their functioning and the increased expense of energy requirements as previously mentioned, they are more complex. So, the outcomes clearly show that a simple and affordable control method for reducing oscillations brought on by road disturbances is presented by semiactive suspension systems because all closed-loop feedback systems include some danger of instability.

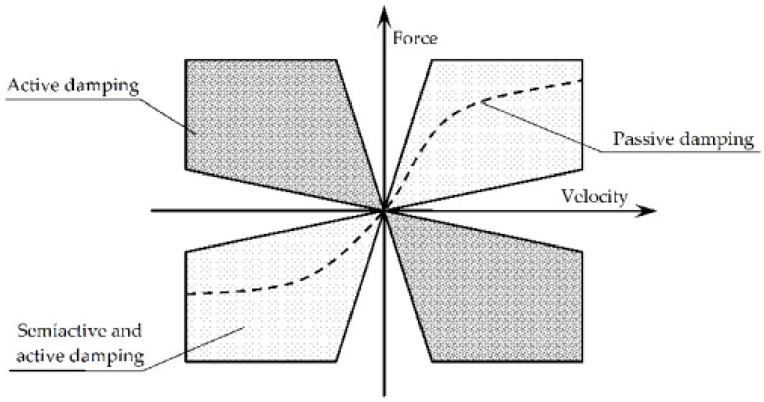



Figure 2.1: The comparison of Force-Velocity responses between Passive, Active & Semiactive suspensions [7].

The researchers in article [8], the active suspension is used to combine the two control algorithms to investigate and demonstrate the performance benefits of each control technique. MATLAB Simulink is used to compare the output of the two controllers after trial and error were used to optimize the LQR's settings based on the half-car model illustrated in Figure 2.2. The performance of LQR and PID is examined in this work using a half-car model. Half of the car mass is usually spread over one front wheel, one rear wheel, and one half-car model, as Figure 2.2 illustrates. According to the study's findings, LQR outperforms PID in terms of overshoot and instantaneous response. Additionally, as evidenced by a variety of input types, it has improved settling characteristics about the smoothness of the settling curve. With additional improvement, the LQR could perform noticeably better than conventional controllers. One way to get considerably better results out of the LQR is to employ extended genetic algorithm optimization to optimize the weight matrices that control its performance.

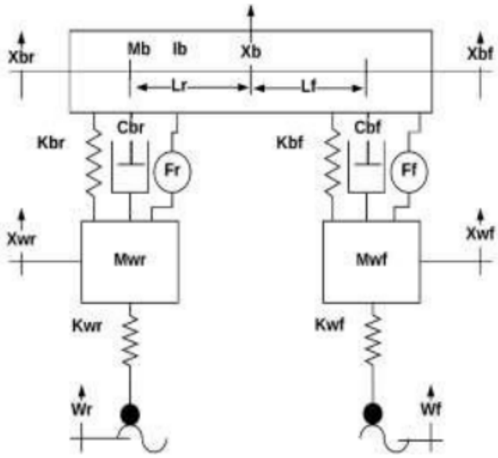



Figure 2.2: The half car model [8].

## 2.3. Active Suspension system

Active suspensions differ from conventional passive suspensions in their ability to inject energy into the system and store and has divided active suspensions into two categories: low-bandwidth or soft active suspensions, characterized by an actuator in series with a damper and the spring in Figure 2.3 and 2.4. Wheel hop motion is controlled passively by the damper so that the active function of the suspension can be restricted to body motion. Therefore, such a type of suspension can only improve the ride comfort. A high-bandwidth or stiff active suspension is characterized by an actuator paralleling the damper and the spring. Since the actuator links the unsprung mass to the body, it can control both the wheel hop and body motions. The high bandwidth with active suspension can now simultaneously enhance handling and riding comfort. Therefore, almost all studies on the active suspension system utilized the high bandwidth type.

Different active suspension models are reported in the literature, either modelled linearly (used most) or non-linear such as are Macpherson strut suspension systems [20].

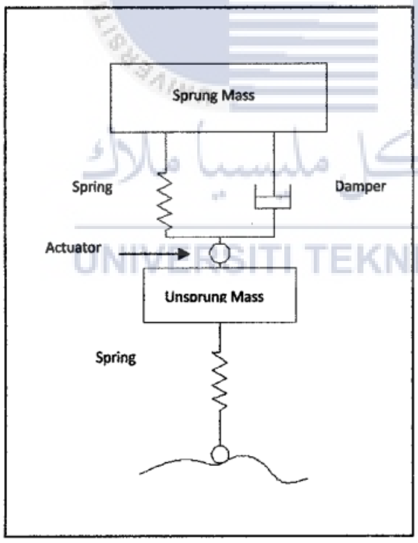



Figure 2.3: A low bandwidth or soft active suspension

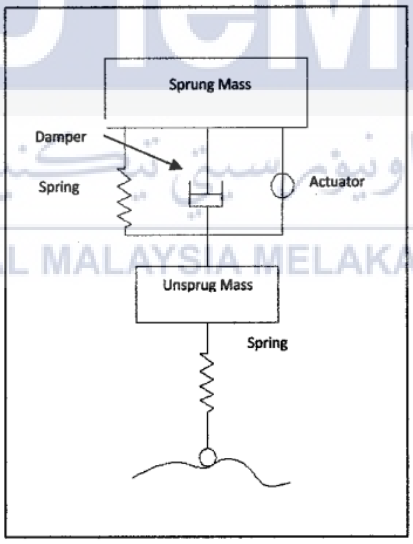



Figure 2.4: A high bandwidth or stiff active suspension

## 2.4. Types of controllers

Controllers are parts or equipment that control and govern how different systems operate. Controllers come in various forms, each intended for a particular need. Here's an overview of some popular multiple types:

### 2.4.1. Proportional Integral Derivative (PID) Controller

Due to its effectiveness in various applications, PID type controllers remain the industry's most extensively employed control algorithms despite recent advancements in the control period. The availability of well-established rules for tuning the parameters of the controller and their relatively simple structures that can be easily implemented are key reasons why they are desirable in real time applications. A PID controller is a feedback controller that continuously determines the appropriate adjustment whenever the actual conditions of a model contrast with a set point or desired value. In general, the PID controller comprises proportional, integral, and derivative components, and the general equations of the PID are shown below.

$$G(s) = K_p + \frac{K_i}{s} + K_d s$$

Where $K_p$, $K_i$ and $K_d$ are proportional, integral, and derivative gain constants of the PID controller.

The study presented in [1], [3], [5], [8-13] using PID Controller to control the suspension for quarter car model and MATLAB Simulink simulation is used. In [1], road disturbances are modeled using MATLAB Simulink to analyze the effectiveness of the active suspension system. In the present research, two types of road disturbances which is one with a single bump and another one is with two bumps are modeled using Simulink. A PSO-based PID controller is suggested in the present research to experience the active suspension system of the car's dynamic performance. As a result, the position, velocity, and acceleration of the vehicle's sprung body subjected to single bump disturbance are significantly improved in the case of an active suspension system with a PSO-based proposed PID controller, as compared to passive suspension systems and active suspension systems using the conventional Ziegler-Nichols method. According to the simulation results, the suggested PSO-based PID controller performs significantly better in position, velocity, and acceleration and maintains good road-tire contact under conditions of bumpy road disturbances than the passive system and active system based on ZN-method.

### 2.4.2. Fuzzy Logic Controller

A particular area of focus in the research of artificial intelligence is fuzzy logic, which is predicated on the usefulness of information that is neither definitively true nor false. The study presented in [4],[13], [16-17], are focusing on Fuzzy Logic controller tuning with PSO algorithm. This paper demonstrates fuzzy logic techniques to design an active car suspension system controller that enhances suspension system performance. The controller design makes use of the triangular membership function, and the linguistic variables are employed in seven regions for the input variable: Positive Large (PL), Positive Medium (PM), Positive Small (PS), Zero (ZE), Negative Small (NS), Negative Medium (NM), and Negative Large (NL) as shown in Table 2.2. Five output variables are set as ZE, PS, PM, and PL for the output [16]. Based on the simulation results, it is evident that all three controllers, which is PID, LQR and Fuzzy Controller have been constructed correctly. However, the fuzzy control scheme performs better than the other controller and the passive system. Performance improves with better tuning even when the controller structures remain the same.

| | | error (e) | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | PB | PM | PS | ZE | NS | NM | NB |
| error rate (er) | PB | PB | PB | PB | PB | PM | PS | ZE |
| | PM | PB | PB | PB | PM | PS | ZE | NS |
| | PS | PB | PB | PM | PS | ZE | NS | NM |
| | ZE | PB | PM | PS | ZE | NS | NM | NB |
| | NS | PM | PS | ZE | NS | NM | NB | NB |
| | NM | PS | ZE | NS | NM | NB | NB | NB |
| | NB | ZE | NS | NM | NB | NB | NB | NB |

Table 2.2: Fuzzy Controller Rules [17]

### 2.4.3. Linear Quadratic Regulator (LQR) Controller

The Linear Quadratic Regulator (LQR) is a prominent optimum control method employed in control systems. The objective is to minimize a cost function that captures the trade-off between actuator power and system performance [8]. This is achieved by optimizing the gain matrices to attain the highest level of performance while simultaneously minimizing the amount of actuator force needed. The LQR controller is specifically intended to minimize a cost function that captures the balance between control effort and system performance [8][9]. The system utilizes state feedback to ascertain the control input and is calibrated to optimize performance while adhering to the constraints imposed by actual and achievable force values. However, further improvements can be implemented to the LQR controller through the utilization of genetic algorithm optimization, resulting in further advancements in performance. Using genetic algorithm optimization is a common way to improve the performance of the LQR controller by making the weight matrices work better [8][9]. Features of the LQR controller's performance might potentially be enhanced through the application of genetic algorithm optimization techniques. This fine-tuning process may result in increased system response, reduced settling time, and minimized actuator power usage. Genetic algorithm optimization can also be used to strategically place the poles of the Linear Quadratic Regulator (LQR) controller. This achieves the best performance of the controller and makes it even better at controlling system dynamics. In the end, the study looks at an active suspension system and compares the LQR controller to other control methods to see how well it works at improving both ride comfort and vehicle handling [9].



## 2.5 Metaheuristics algorithm

The optimization of suspension systems is a complex task due to the involvement of multiple parameters and the need to balance conflicting objectives. In recent years, metaheuristic algorithms have emerged as powerful tools for solving combinatorial optimization problems. These algorithms are inspired by natural phenomena and exhibit the ability to explore large solution spaces efficiently. It aims to explore the application of metaheuristic algorithms in optimizing suspension systems.

### 2.5.1 Particle Swarm Optimization Algorithm (PSO)

The study presented in [1-2], focuses on PID controller tuning with the Particle Swarm Optimization algorithm, first developed by Kennedy and Eberhart, is one of the modern heuristic algorithms. It was developed inspired by the social behavior of fish and birds in schools and has proven effective in resolving continuous nonlinear optimization issues. The following situation is the basis for this algorithm: a group of birds is randomly hunting for food in a given region, and there is only one food item. Although no bird knows the exact location of the meal, they are all aware of its distance at any given moment. The best and most efficient way to locate the food is to follow the bird that is closest to it. The PSO algorithm solves the optimization problem based on such a scenario.

In the research [10], the active suspension system is intended for use with the PID controller. Two-degree-of-freedom quarter-car vehicle models have been created. During simulation, hydraulic dynamics are also considered. PID controllers' derivative time, reset rate, and proportional gain are all determined by setting the PSO algorithm. The technology is designed to handle inputs from random roads, potholes, and rough roads. The simulation findings demonstrate how an active suspension system with PID control enhances ride comfort. It merely requires a smaller rattling space concurrently. On the other hand, no appreciable gain in road-holding ability has been seen, particularly for random road surfaces. In addition to having a very straightforward design and using well-known standard gear, the PID controller has been shown to be a useful tool for creating active suspension systems.

### 2.5.2 Genetic Algorithm (GA)

The genetic algorithm is a technique to find the solution for both constrained and unconstrained optimization problems that is based on natural selection which is the process that drives biological evolution. A population of unique solutions undergoes recurrent modifications using the genetic algorithm. The genetic algorithm chooses members of the existing population to be parents at each phase, using them to create offspring for the following generation. The people "evolve" towards an ideal solution throughout subsequent generations.

In the research [11], the mathematical models of passive suspension and active suspension will be identified in this study. The last one is managed by the traditional PID Ziegler-Nichols technique, which yields the best results for PID parameter optimization by implementing genetic algorithms. The outcomes clearly show that using a PID controller optimized by genetic algorithms based on the IAE fitness function, the simulation results improve performance. In the future, greater attention will be paid to alternative active suspension management strategies and their comparison to determine which best provides adequate suspension stability and improved driver and passenger comfort.

### 2.5.3 Artificial Bee Colony Algorithm (ABC)

Dervis Karaboga created Artificial Bee Colony (ABC) in 2005, inspired by bee behavior. ABC uses parameters like colony size and cycle count as parameters for a population-based search process. The goal is to locate food sources with high nectar content. Bees in ABC fly in a multidimensional search space, making decisions based on their experiences and nest mates' experiences. Some bees randomly select food sources without prior knowledge, remembering new positions if nectar amount is significant. The ABC system balances exploration and exploitation processes by blending local search techniques with global search techniques, overseen by scouts and bystanders.

In research paper [15], proposed a control strategy for the active suspension system that uses fuzzy logic, the sliding mode concept, and a PID controller optimized by the ABC algorithm. The fuzzy system evaluates the system's unknown characteristics, the PID controller regulates the force generated by the actuator, and the sliding mode controller controls the location of the sprung mass. The controller's parameters are optimized through the application of the ABC algorithm. As a result, by comparing the new control approach to a passive suspension system, the former decreases body displacement and suspension deflection to extremely modest levels. Additionally, the suggested control system sufficiently lowers the body's acceleration. The results show that the regulated active suspension system responses have improved by approximately 90% when compared to the passive suspension system.

### 2.6 Summary of the suspension system

Table 2.3 shows the explanation and differences among the suspension systems such as active, semi-active and passive suspension system. Regarding enhancing ride comfort, active suspension systems outperform semi-active and passive ones, particularly when handling road disturbances. An active suspension system utilizes sensors and actuators to dynamically respond to road conditions in real time, in contrast to semi-active and passive systems, which remain static. This implies that it may lessen the effect of vibrations and bumps for a more comfortable and smoother ride. Moreover, active suspension is the best option for improving riding comfort on various types of roads because of its rapid adaptability.

Table 2.3: Different categories of the suspension system

| No. | Suspension system | Description |
|---|---|---|
| 1. | Active suspension system | - Active suspension system has improved ride the comfort [21]. |
| 2. | Active suspension system | - It has been demonstrated that when the PID parameters are set correctly, the active suspension system outperforms the passive suspension system in performance [5]. |
| 3. | Active suspension system | - The findings show that active suspension performs significantly better when a disturbance observer is used than when neither the observer nor the system of passive suspension is used [6]. |
| 4. | Active suspension system | - Simulation results indicated that active suspension had better rising time, settling time, and overshooting than passive suspension [3]. |
| 5. | Semi-active suspension system | - Due to its lack of force actuators and modest resonant response with improved high-frequency isolation, semi-active suspension systems offer an affordable and energy-efficient alternative [7]. |
| 6. | Passive suspension system | - Passive suspension is the least effective suspension to absorb any disruption to the system [6]. |
| 7. | Passive suspension system | - However, to assure comfort or handling, passive suspension is recommended to active suspension when using a FUZZY or PID controller. [22]. |

## 2.7 Summary of controller design for active suspension system

Table 2.4 shows the explanation and differences between active suspension systems such as PID, LQR and Fuzzy Logic controller. Regarding enhancing ride comfort, PID controller is better than LQR and Fuzzy Logic Controller.

Table 2.4: Different types of controllers of active suspension system

| No. | Controller | Description |
|---|---|---|
| 1. | PID Controller | - The PID controller has proven to be a viable instrument in developing active suspension systems, in addition to its widely recognized standard hardware and relatively simple architecture [23]. |
| 2. | PID Controller | - Many types of vehicle suspension controls use the PID controller, the most famous controller in the industrial, and it also has a high application value to address the PID controller's inadequacies [24]. |
| 3. | PID Controller | - Simulation results demonstrate that ride comfort is enhanced by active suspension systems with PID control [10]. |
| 4. | LQR Controller | - According to the study, the LQR outperforms the PID regarding overshoot and instantaneous response [8]. |
| 5. | LQR Controller | - With the PSO, the LQR controller enhances the system's key performance parameters, including tire deflection, acceleration of the car body, suspension movement, and dynamic load on tire [8]. |
| 6. | Fuzzy Logic Controller | - The fuzzy control system outperforms other control systems in terms of performance measures such as suspension travel, seat velocities, sprung mass displacement, and seat acceleration [9]. |
| 7. | Fuzzy Logic Controller | - It can be focused on the improved body vehicle for the suspension system, which has superior performance compared to PIDSO and passive systems. The capacity of the FLCPSO to control the suspension system has been proven [16]. |

# CHAPTER 3

# METHODOLOGY

## 3.1 Introduction

This chapter outlines the proposal for this study, which includes the fundamentals of the approach that will be used to carry it out. After carefully considering the norms and specifics of the earlier studies, the choice of planning, processing, and testing will be offered. The key methodological tenet is recommending the best approaches and strategies for finishing this study. This section provides a cursory description of the strategies adopted to meet the study's objectives. To facilitate comprehension, several flowcharts are utilized to explicate the methodological details, with accompanying elaborations for each segment. The method applied throughout the project is segmented into three distinct phases, each corresponding to a specific objective. As illustrated in Figure 3.1, the initial segment focuses on objective 1, which entails developing and analyzing the quarter car model for active suspension to specific road disturbances to enhance passenger comfort. The investigation and optimization of the PID controller, determined via the PSO algorithm, comprise the second segment. Finally, the third segment compares the performance of the optimized PID controller against the classical PID controller under various road disturbances, employing suitable error performance metrics.

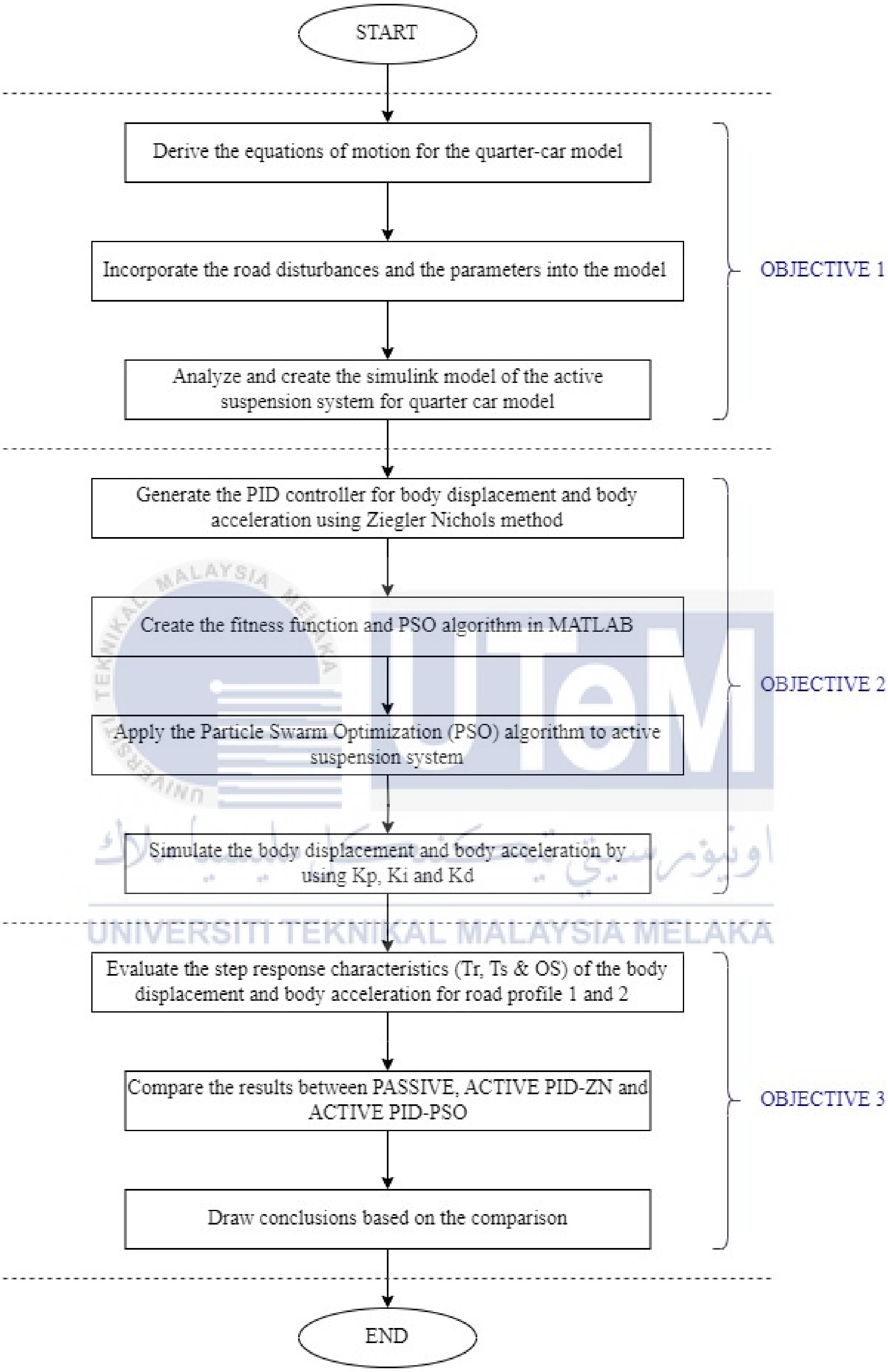



Figure 3.1: Flowchart of methodology to achieve all the objectives in this project

## 3.2 Method for achieving objective 1

The objective 1 in this project is to develop and analyze the quarter car model for active suspension to specific road disturbances to give comfort to the passengers. So, the method to achieve objective 1 begins with choosing the quarter car model to develop in active suspension system and express the system dynamics using differential equations. The equations of motion will relate the acceleration, and displacement of the sprung mass to the forces acting on it. To make the model more realistic, introduce road disturbances (road profile) such as Road Profile 1: one bump and Road Profile 2: two bumps into the system. Next, simulate the quarter car response by analyzing how the suspension reacts to different road profiles.

Evaluate various metrics related to suspension performance:

- Suspension Displacement: Measure how much the suspension compresses or extends during motion.
- Suspension Acceleration: Assess the acceleration of the sprung mass caused by road disturbances.
- Comfort and Stability: Consider how well the suspension system absorbs shocks (comfort) and maintains stability during driving.

By analyzing these metrics, it can optimize the suspension design for both ride comfort and car stability.



### 3.2.1 Mathematical Modelling for the quarter car model

The quarter car model can be said to be the simplest model among all the models. Most quarter car models are analyzed on two degrees of freedom (DOF) which are the vertical displacement of the sprung mass and unsprung mass. Although it has the less variables thus make it less complicated to be analyze, it is enough to develop the active suspension based on this model as it will still shows the differences between passive and active suspension, in most of the past active suspension designs were create based on this model. The quarter car model shown in Figure 3.2 with the highlighted features is used to create the active suspension system.

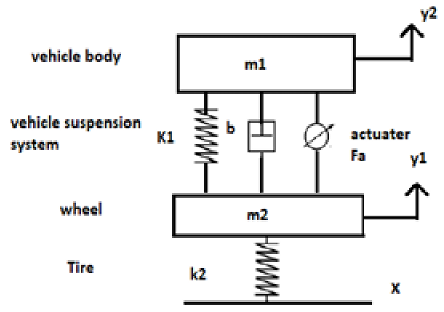



Figure 3.2: Graphic of quarter car model for active suspension system

In this work, it is focusing on using PID controller as the feedback controller. A PID controller is one of the most used feedback controllers. To design the PID controller for this model first it must come up with the transfer function for the model.

Where,

$m_1$ : Sprung mass (vehicle body)

$m_2$ : Unsprung mass (wheel)

$k_1$ : Spring constant of suspension system

$\boldsymbol{k_2}$ : Spring constant of wheel and tire

$\boldsymbol{b}$ : Damping constant of suspension system

$f_a$ : Force actuator



The following equations may be used to generate the quarter car model using Newton's second law.

$$\sum F = ma \tag{1}$$

$$m_1\ddot{y}_1 = -k_1(y_1 - y_2) - b(\dot{y}_1 - \dot{y}_2) \tag{2}$$

$$m_2\ddot{y}_2 = k_1(y_1 - y_2) + b(\dot{y}_1 - \dot{y}_2) + k_2(y_2 - x) \tag{3}$$

By using Laplace transform for equation (3.2) and (3.3),

$$m_1s^2Y_1(s) = -k_1\big(Y_1(s) - Y_2(s)\big) - b(sY_1(s) - sY_2(s)) \tag{4}$$

$$m_2s^2Y_2(s) = k_1\big(Y_1(s) - Y_2(s)\big) + b\big(sY_1(s) - sY_2(s)\big) + k_2(Y_2(s) - X(s)) \tag{5}$$

Rewrite the equations to isolate terms involving $Y_1(s)$ and $Y_2(s)$:

For $Y_1(s)$:

$$m_1s^2Y_1(s) + k_1Y_1(s) + bsY_1(s) = k_1Y_2(s) + bsY_2(s)) \tag{6}$$

$$(m_1s^2+k_1+bs)Y_1(s) = (k_1+bs)\,Y_2(s) \tag{7}$$

For $Y_2(s)$:

$$m_2s^2Y_2(s) = k_1Y_1(s) + bsY_1(s) - (k_1+k_2)Y_2(s) - bsY_2(s) + k_2\,X(s) \tag{8}$$

$$(m_2s^2+k_1+k_2+bs)Y_2(s) = k_1Y_1(s) + bsY_1(s) + k_2\,X(s) \tag{9}$$

Express these equations in matrix form to solve for $Y_1(s)$ and $Y_2(s)$:

$$\begin{bmatrix} m_1s^2+k_1+bs & -(k_1+bs) \\ -(k_1+bs) & m_2s^2+k_1+k_2+bs \end{bmatrix}\begin{bmatrix} Y_1(s) \\ Y_2(s) \end{bmatrix} = \begin{bmatrix} 0 \\ k_2\,X(s) \end{bmatrix}$$

Solve for $Y_1(s)$ and $Y_2(s)$

From the equation (3.7):

$$Y_1(s) = \frac{(k_1+bs)Y_2(s)}{m_1s^2+k_1+bs}$$

Substitute $Y_1(s)$ into the equation (3.9):

$$(m_2s^2+k_1+k_2+bs)Y_2(s) = k_1\frac{(k_1+bs)Y_2(s)}{m_1s^2+k_1+bs} + bs\frac{(k_1+bs)Y_2(s)}{m_1s^2+k_1+bs} + k_2\,X(s) \tag{10}$$

Simplify and solve for $Y_2(s)$:

$$(m_2s^2+k_1+k_2+bs)Y_2(s) = k_1\frac{(k_1+bs)^2Y_2(s)}{m_1s^2+k_1+bs} + k_2\,X(s) \tag{11}$$

$$Y_2(s)[(m_2s^2+k_1+k_2+bs)\,(m_1s^2+k_1+bs) - (k_1+bs)^2] = k_2\,X(s)(\,m_1s^2+k_1+bs) \tag{12}$$

$$Y_2(s) = \frac{k_2\,X(s)(\,m_1s^2+k_1+bs)}{(m_2s^2+k_1+k_2+bs)\,(m_1s^2+k_1+bs)-(k_1+bs)^2} \tag{13}$$

Substitute $Y_2(s)$ back into the equation for $Y_1(s)$:

$$Y_1(s) = \frac{(k_1+bs)k_2\,X(s)(m_1s^2+k_1+bs)}{(m_2s^2+k_1+bs)[\,(m_1s^2+k_1+k_2+bs)(m_2s^2+k_1+bs)-(k_1+bs)^2]} \tag{14}$$

$$Y_1(s) = \frac{(k_1+bs)k_2\,X(s)}{(m_1s^2+k_1+k_2+bs)(m_2s^2+k_1+bs)-(k_1+bs)^2} \tag{15}$$

Transfer function for Body Displacement, $H(s)$

$$H(s) = \frac{Y_1(s)}{X(s)} = \frac{(k_1+bs)k_2}{(m_1s^2+k_1+bs)\left[(m_2s^2+k_1+k_2+bs)-\frac{(k_1+bs)^2}{m_1s^2+k_1+bs}\right]} \tag{16}$$

This can be simplified as:

$$H(s) = \frac{(k_1+bs)k_2}{(m_1s^2+ k_1+bs)(m_2s^2+k_1+k_2+bs)-(k_1+bs)^2} \tag{17}$$

$$H(s) = \frac{240000000s+2560000000}{12500s^4+4675000s^2+375000s-256000000} \tag{18}$$

Transfer function for Body Acceleration, $G(s)$

Body acceleration $\ddot{y}_1$ is $s^2Y_1(s)$ :

$$G(s) = \frac{s^2Y_1(s)}{X(s)} = s^2.\frac{(k_1+bs)k_2\ X(s)}{(m_1s^2+ k_1+k_2+bs)(m_2s^2+k_1+bs)- (k_1+bs)^2} \tag{19}$$

$$G(s) = \frac{s^2Y_1(s)}{X(s)} = s^2.\frac{240000000s+2560000000}{12500s^4+4675000s^2+375000s-256000000} \tag{20}$$

In general, transfer function of PID Controller is

$Controller = C(s)$

$$K_p + \frac{K_i}{s} + K_d s$$

$$C(s) = \frac{K_d s^2 + K_p s + K_i}{s}$$

For the closed-loop system, the transfer function $T(s)\ is$:

$$T(s) = \frac{C(s)G(s)}{1 + C(s)G(s)}$$

Where $G(s)$ is the transfer function of the system (i.e., $s^2Y_1(s)/\ X(s)$).

### 3.2.2 Design Requirements

The objective is to design a car suspension system that strikes a balance between road-holding ability and passenger comfort. When encountering road disturbances (such as potholes or uneven pavement), the car body should exhibit minimal oscillations that dissipate rapidly. This is parameter of the design for suspension system modelling:

Table 3.1: The parameters of the quarter car model.

| Parameter | Variable | Value | Unit |
|---|---|---|---|
| Sprung mass (body) | $m_1$ | 250 | kg |
| Unsprung mass (wheel) | $m_2$ | 50 | kg |
| Spring constant of suspension system | $k_1$ | 16000 | N/m |
| Spring constant of wheel and tire | $k_2$ | 160000 | N/m |
| Damping constant of suspension system | $b_s$ | 1500 | N.s/m |

**Source**: S. A. Al-Khafaji, A. H. Saleh, and S. M. Shaheed, "Optimization of PID Controllers in Active Suspension Systems: A Comparative Study of the Firefly Algorithm and the Particle Swarm Optimization," Math. Model. Eng. Probl., vol. 10, no. 6, pp. 2023–2030, 2023, doi: 10.18280/mmep.100612.



### 3.2.3 Modelling of Road Profile

Various types of road profiles are engaged in the study of the suspension dynamics in relation to the vehicle's handling ability and ride comfort. Overall, road disturbances that last for a short time and have a strong consequence have a negative effect on passenger ride comfort and may cause the road surface to lose its hold. Uneven roads are an essential cause of disturbances on the road. The road disturbances are represented with basic sinusoidal functions for the dynamic response study of the vehicle suspension system on rough roads. In this study, road disturbances with step disturbance, single and two bumps are determined for the car suspension system's performance examination. MATLAB Simulink models were created for road disturbances to make research using Simulink modelling easier.

### 3.2.3.1 Road Profile 1 (RP1)

The following equation can be used to simulate a single bump in the road.

$$R_{P1}(t) = \begin{cases} \frac{A}{2}\left(1 - \cos\left(2\pi\left(\frac{t}{T_{b1}}\right)\right)\right), & for\ T_{b1} \le t \le 2T_{b1} \\ 0 \qquad\qquad\qquad\qquad\ , & Otherwise \end{cases} \tag{3}$$

In Eq. (3), A is the height of the bump and $T_{b1}$ is its duration. The duration of the bump is determined by the ratio of the bump length (L) to the vehicle velocity (V). It is believed that the road bump disturbance forms between $T_{b1}$ and $2T_{b1}$ seconds, and that the rest of the time the road profile is plain.

In the first scenario, a road disturbance with a single bump of height (A) 0.05m has been taken into consideration. It is assumed that the vehicle is moving at a magnitude of 36 km/h and that the bump's length (L) is 5 m. The vehicle is thought to be crossing the bump between the time interval of 0.5 and 1.0 seconds, as shown by the bump duration $T_{b1}$, which is computed as (L/V) and equals 0.5 seconds. Fig. 3.3 shows the road bump disturbance with a single bump.



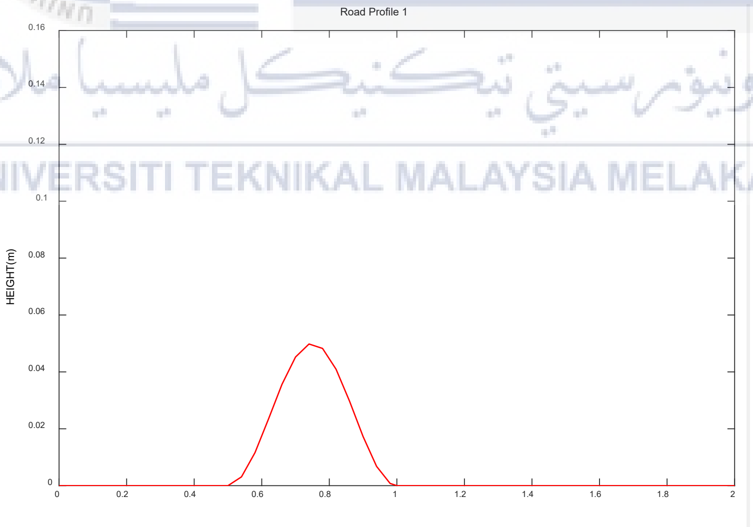



Figure 3.3: Road disturbances with a single bump

**<u>Simulink model of the active suspension system with Road Profile 1</u>**

Simulink model of road profile road 1 which is one bump is developed and shown in Figure 3.4. In Figure 3.4, the values $T_1$ and $T_2$ are assumed to be equal to the start time $T_{b1}$ and reflect the end time $2T_{b1}$ as described in Eq. (3).

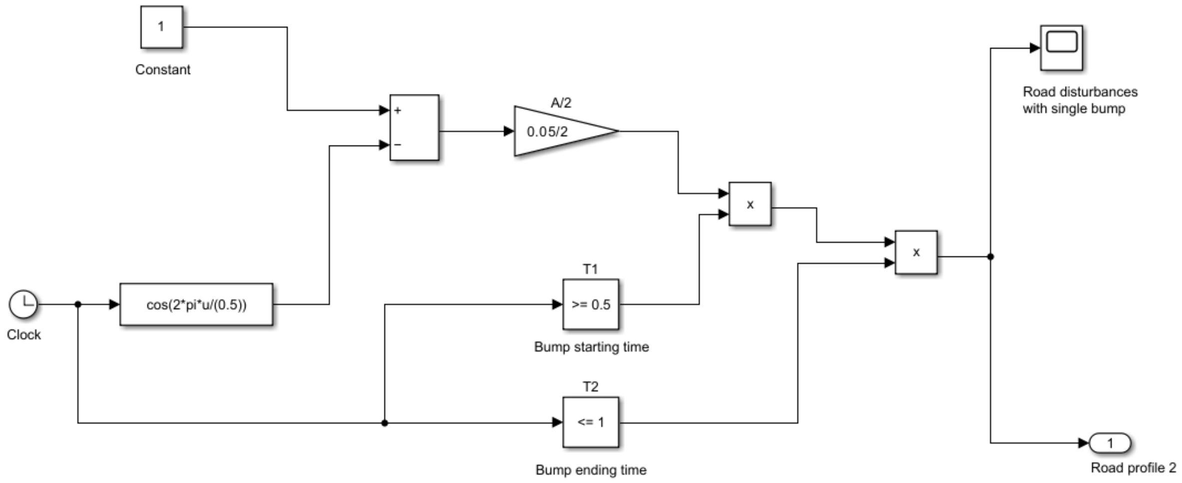



Figure 3.4: Simulink model of road profile 1

### 3.2.3.2 Road Profile 2 (RP2)

The road disturbance with two bumps of different magnitudes is designed using the following equation.

$$\boldsymbol{R_{P2}(t)} = \begin{cases} \frac{A}{2}\left(\mathbf{1} - \mathbf{cos}\left(2\pi\left(\frac{t}{T_{b1}}\right)\right)\right), & \boldsymbol{for\ T_{b1} \le t \le 2T_{b1}} \\ \frac{B}{2}\left(\mathbf{1} - \mathbf{cos}\left(2\pi\left(\frac{t}{T_{b2}}\right)\right)\right), & \boldsymbol{for\ 8T_{b2} \le t \le 9T_{b2}} \\ \mathbf{0}, & \boldsymbol{Otherwise} \end{cases} \tag{4}$$

According to Eq. (4), the first road bump disturbance during the time interval between $T_{b1}$ and $2T_{b1}$ seconds has a height of A, and the second bump disturbance of height occurs during the time period between $8T_{b2}$ and $9T_{b2}$ seconds.

The two bumps in the road are 0.05m in height (A) and 0.075m in height (B), are taken into consideration in the second case, as shown in Figure 3.5. It is considered that the car is travelling at a magnitude of 36 km/h and that the first and second bumps are 5m and 6m long, respectively. For the first bump, the time Tb1 is equal to the bump length divided by the velocity; in the case of the second bump, this equals 0.75 seconds. The first bump is being crossed by the car between the intervals of 0.5 and 1.0 seconds and 6.0 and 6.75 seconds.

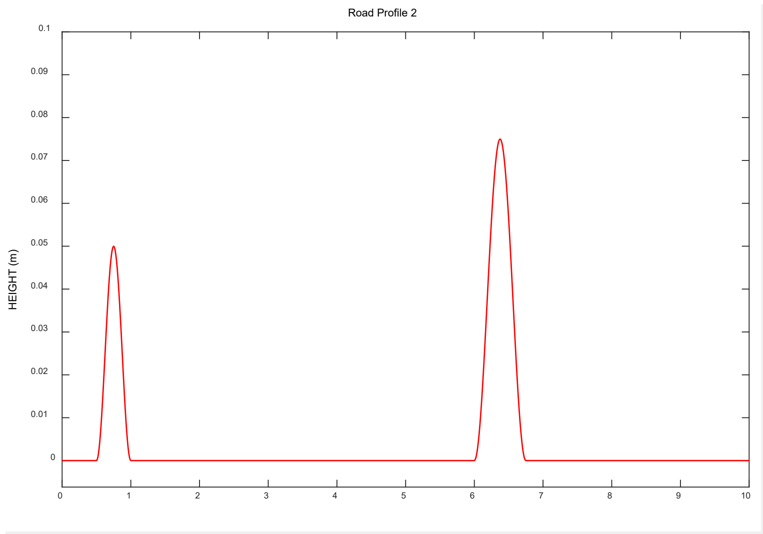



Figure 3.5: Road disturbances with two bumps

**Simulink model of active suspension system with Road Profile 2**

Simulink model of road profile road 1 which is two bump is developed and shown in Figure 3.6. In Figure 3.6, the values of $T_1$ and $T_2$ are equal to the starting time of the bump $T_{b1}$ and the end time of $2T_{b1}$, as described in Eq. (4). In the second method involving two bumps, the values $T_3$ and $T_4$ are considered equal to $8T_{b2}$ and $9T_{b2}$, respectively.



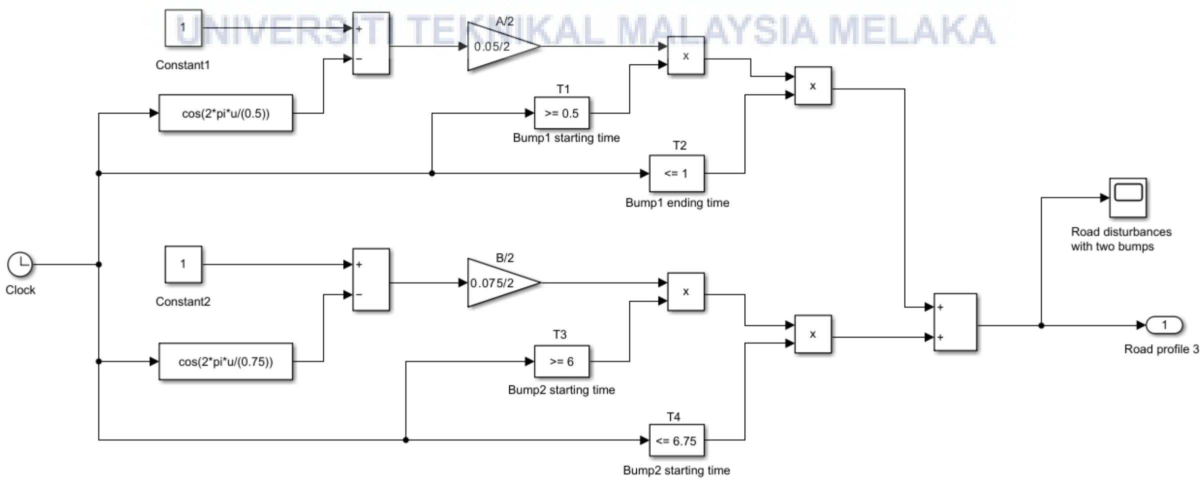



Figure 3.6: Simulink model of road profile 2

## 3.3 Method for achieving objective 2

In this project, the objective 2 is to investigate and optimize PID controller using PSO algorithm in active suspension system. To evaluate the performance of PSO algorithm in active suspension system, the technique must start with tuning the PID Controller using conventional method and continue with PSO algorithm.

In this section, the Ziegler Nichols method and PSO algorithm are chosen to be used in this tuning the PID Controller. The step by step of how the tuning PID controller is explaining in this section. After obtaining the value of proportional gain (Kp), integral gain (Ki) and derivative gain (Kd) from the test that has been done, the specific performance metrics such as rise time, settling time and overshoot has been tabulated in the table to see the different between passive system and active PID-ZN and active PID-PSO.

### 3.3.1 Tuning the PID Controller

This chapter mainly describes the tuning method to find the value of Kp, Ki and Kd. Moreover, each solution will show the result respect to single objective functions that run using MATLAB.

#### 3.3.1.1 Ziegler Nichols Tuning Method



PID tuning basically can be done by using a few methods, one of the commonly used is Ziegler-Nichols method. It can be used in either original form or in modification. The methods are based on determination of some features of process dynamics. The controller parameters are then expressed in terms of features by simple formulas. In Ziegler Nichols Tuning Method, the procedures are:

The Ziegler-Nichols method is a heuristic method of tuning a PID controller. It involves the following steps:

1. Increase the proportional gain, Kp, until the system reaches the ultimate gain, Ku, at which the output of the control system starts to oscillate.

2. Measure the oscillation period, Tu, of the output.

3. Use the Ku and Tu values to set the PID parameters according to the Ziegler-Nichols tuning rules. The classic Ziegler-Nichols tuning rules are:

- P controller: Kp = 0.5*Ku
- PI controller: Kp = 0.45*Ku, Ki = 1.2*Kp/Tu
- PID controller: Kp = 0.6*Ku, Ki = 2*Kp/Tu, Kd = Kp*Tu/8

Table 3.2: Ziegler-Nichols calculation for closed-loop.

| Control | Kp | Ti | Td |
|---|---|---|---|
| P only | 0.5Ku | ∞ | 0 |
| PI | 0.45Ku | Tu/1.2 | 0 |
| PID | 0.6Ku | Tu/2 | Tu/8 |

### 3.3.1.2 PSO Algorithm Tuning Method

The Particle Swarm Optimization (PSO) algorithm is like a dance of nature, where each participant, or "particle," learns from both their own experience and the group's wisdom. Imagine a flock of birds or a school of fish gliding gracefully, each one adjusting its path by learning from its own best moves and the top performers of the group. This harmony allows them to home in on their target, much like how PSO particles fine-tune their positions to zero in on the ultimate solution. It's a beautiful blend of individual insight and collective knowledge that guides them through a maze of possibilities to the most efficient answer. PSO's brilliance shines across various fields, harnessing this shared intelligence to tackle complex challenges with finesse. In the world of Particle Swarm Optimization (PSO), each particle carries with it two essential qualities: velocity and position. Think of velocity as the particle's inner compass, recalibrated through its personal journey and the shared stories of the swarm. Position, on the other hand, is the particle's footprint in the vast dance hall of possibilities, shifted by the rhythm of its velocity. Together, these particles waltz through the many dimensions of the search space, united by a common quest to uncover the most exceptional solution—the global best—hidden within the swarm's collective embrace.

The step-by-step algorithm for PSO based optimization for vehicle suspension system is given below as shown in Figure 3.7:

1. First, the randomly generated population initializes the swarm's particles for the PID controller gains, setting their velocity positions to a constant value.
2. Set the iteration count to one.
3. Calculate the fitness function values for each particle.
4. Set the current value to $P_{best}$. if the particle's fitness value from the current population is higher than that of the matching previous population.
5. Set the best fitness value from all previous iterations as $G_{best}$.
6. Determine velocity and position of each particle for the next iteration.
7. If the iteration count is less than the maximum number, simply proceed to step 3. Otherwise, increase the iteration count by one.

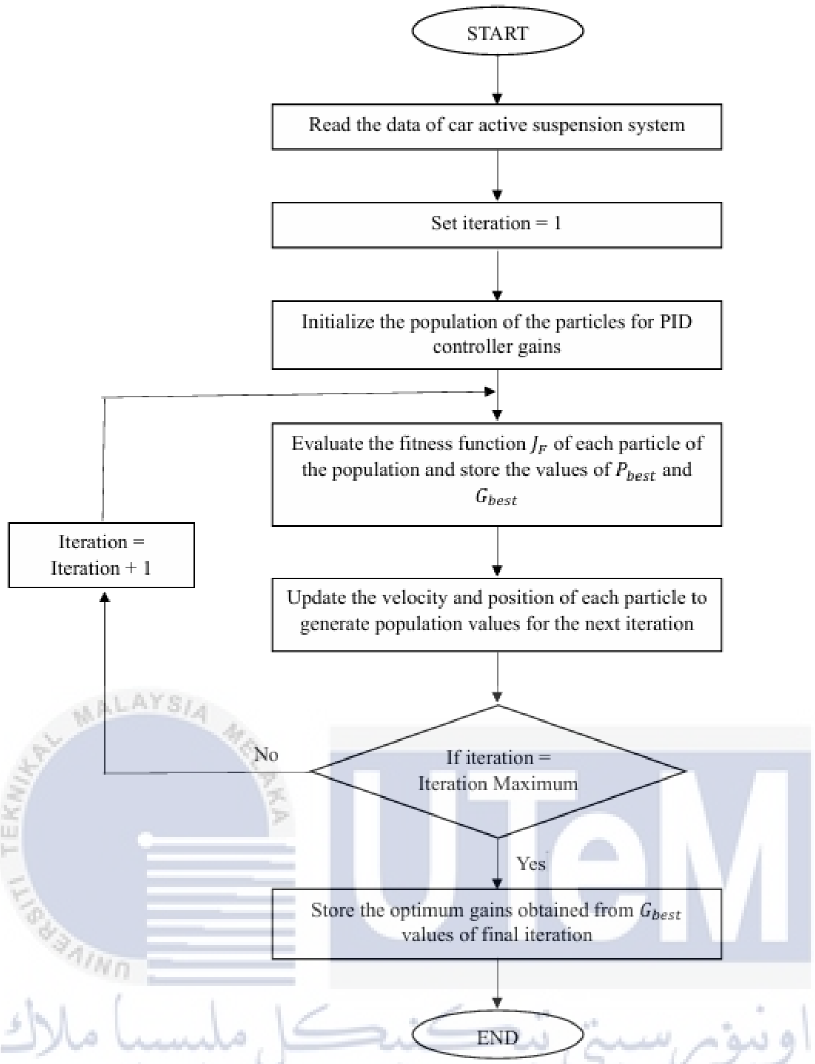



Figure 3.7: Flowchart for obtaining optimum PID gains using PSO algorithm for active suspension system

The optimization a population of particles representing potential solutions, the Particle Swarm Optimization (PSO) algorithm is commonly employed to tackle optimization problems wherein the objective function is either analytically unknown or computationally demanding to evaluate. These particles traverse the search space with the goal of identifying the optimal solution. The incorporation of performance criteria such as ITAE into the PSO (Particle Swarm Optimization) algorithm guides the optimization process towards identifying the optimal set of controller parameters that enhance control system performance. This enables the fine tuning and optimization of control systems to achieve specific performance objectives based on the chosen criterion. Figure 3.8 is a block diagram of PSO Algorithm with performance index.

- For ITAE performance,

  ITAE (Integral of Time multiplied by Absolute Error): ITAE = $\int|e\ (t) * t|dt$

  Where: e(t) is the error at time t.

- For ITSE performance,

  ITSE (Integral of Time multiplied by Squared Error): ITSE = $\int(e(t)2{*}t)\ dt$

  Where: e(t) is the error at time t.

- For IAE performance, IAE (Integral of Absolute Error): IAE = $\int|e(t)|dt$

  Where: e(t) is the error at time t.

- For ISE performance, ISE (Integral of Squared Error): ISE = $\int(e(t)\text{^}2)\ dt$

  Where: e(t) is the error at time t.

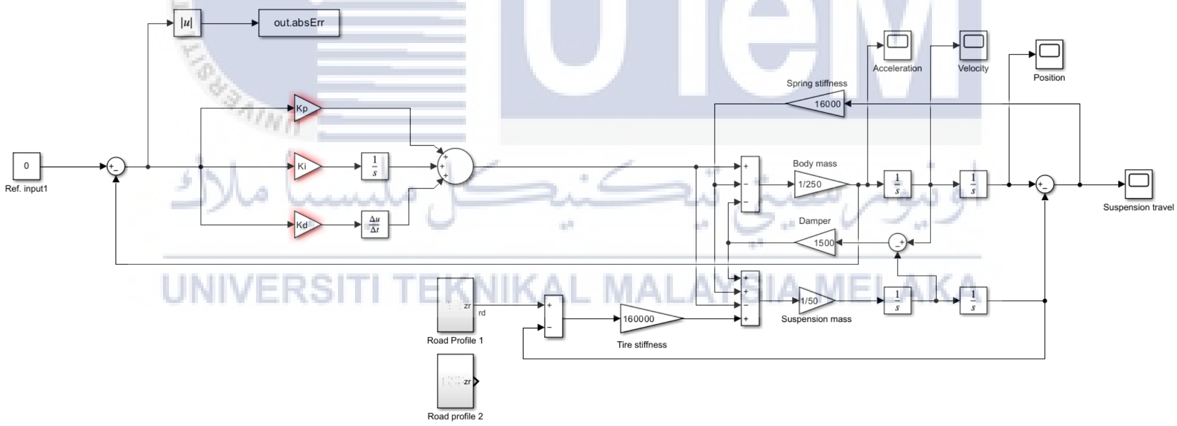



Figure 3.8: Block diagram of PSO Algorithm for Active Suspension System with performance index.

### 3.4 Method for achieving objective 3

In this project, the objective 3 is to compare the system performance between the optimized PID controller and classical PID controller under various road disturbances. This method focuses on system performance based on three systems which are passive, active PID-ZN and active PID-PSO. This section clearly shows that all the data from the road profile 1 and 2 has been compared and draw a conclusion based on these results. The Simulink model for 3 types of suspension system has been shown in Figure 3.9 for road profile 1 and Figure 3.10 for road profile 2.

- **Road Profile 1**

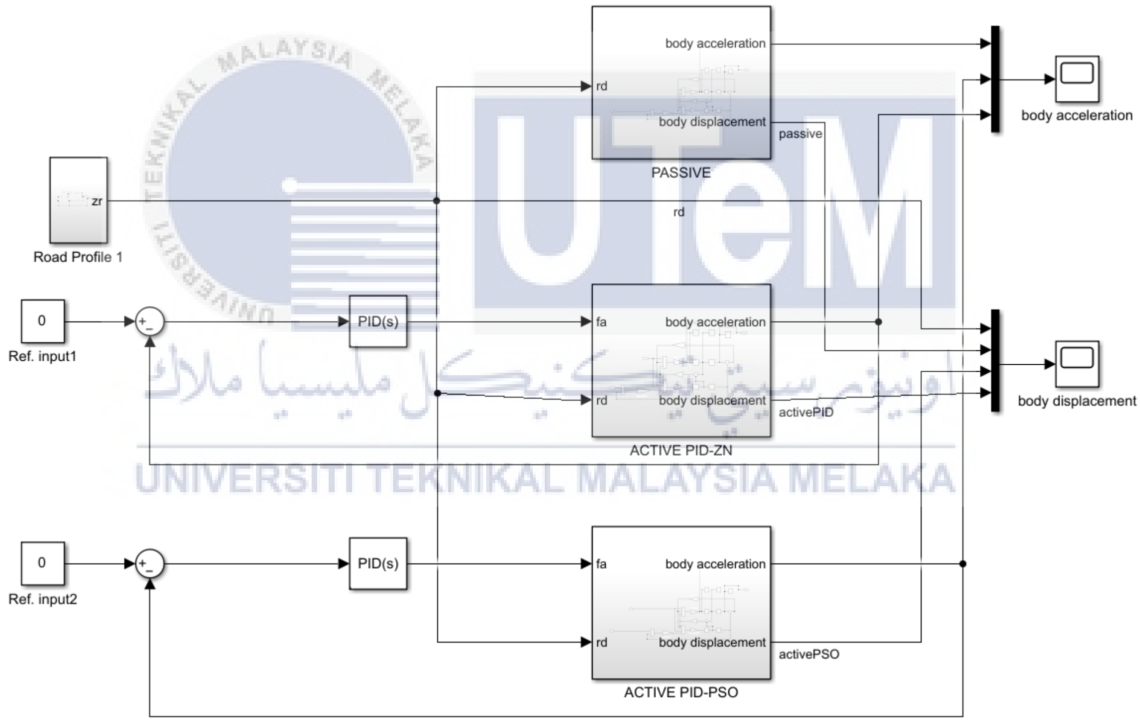



Figure 3.9: Simulink model to compare the performance between PASSIVE, ACTIVE PID-ZN and ACTIVE PID-PSO for road profile 1.

- **Road Profile 2**

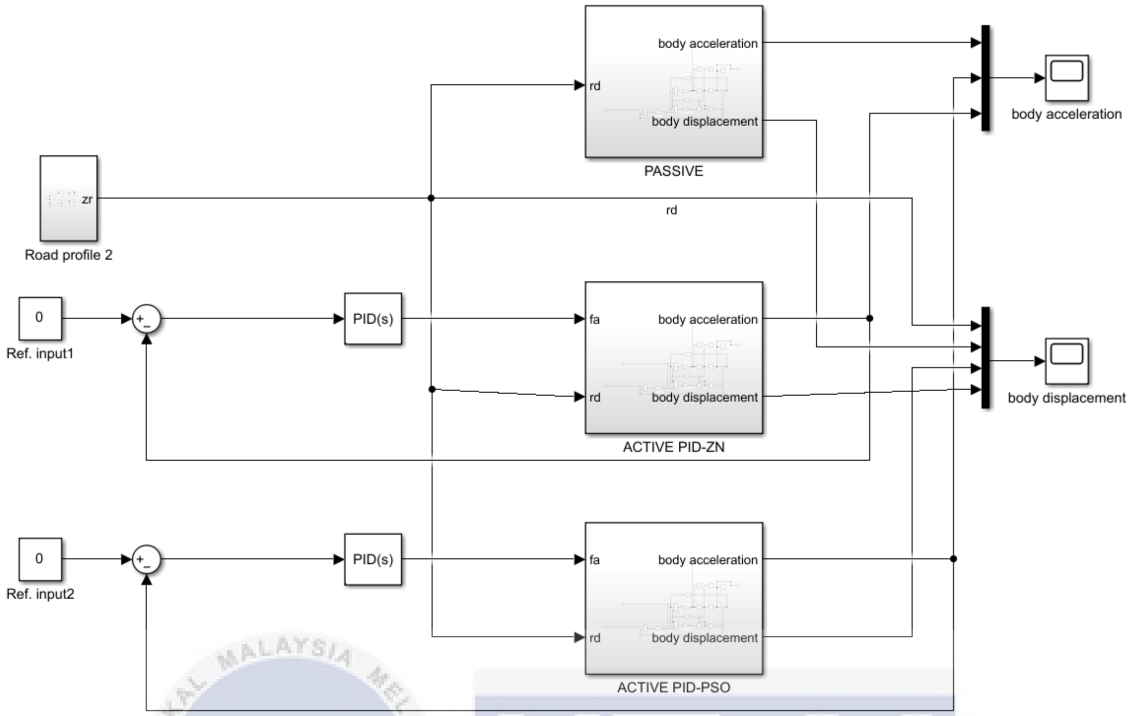



Figure 3.10: Simulink model to compare the performance between PASSIVE, ACTIVE PID-ZN and ACTIVE PID-PSO for road profile 2

## 3.5 Summary

The method that is proposed in this chapter is fully utilized to achieve the objectives of this project and overcome the problems that face as discussed in problem statements. All of the methods have been fully explained clearly for each objective. The next chapter will present the results and discussions of the project which was obtained by using the method as discussed in this chapter.

# CHAPTER 4

# RESULTS AND DISCUSSIONS

## 4.1 Introduction

This chapter mainly described the result using classical PID controller using Ziegler Nichols method and the optimized PID controller using PSO Algorithm. Moreover, each solution will show the result respect to single objective functions that run using MATLAB.

## 4.2 PID Controller

The PID controller simulation involves utilizing the gain obtained from the Ziegler-Nichols method and PSO algorithm. This method enables the determination of optimal control gains by minimizing the body displacement and body acceleration. By incorporating the gains optimized by PSO into the PID Controller, the system's response can be enhanced, resulting in improved stability, performance, and tracking accuracy.

### 4.2.1 Classical PID controller using Ziegler Nichols Method

The response of the active suspension (closed loop) system of the quarter car model for the ultimate gain value (Ku = 5800) and the period of the sustained oscillation for this value of ultimate gain is called ultimate time Tu, which is determined from the step response of the closed loop system and is found to be Tu = 1.2655sec. The ultimate gain (Ku) and ultimate time (Tu) determined above are used to set the tuning rules for the quarter car model using the Zeigler-Nichols tuning rules. As discussed, the values of the P, PI and PID controller are obtained and are tabulated in Table 4.1.

Table 4.1: Ziegler Nichols tuning values

| Control | Kp | Ti | Td |
|---|---|---|---|
| P only | 2900 | ∞ | 0 |
| PI | 2610 | 2062.4259 | 0 |
| PID | 3480 | 5500 | 550.4925 |

- **Road Profile 1**

Based on the value of Kp = 3480, Ki = 5500 and Kd = 550.4925 that being tuned from Ziegler Nichols method, these are the results that have been generated.

For body displacement, as shown in Figure 4.1, the passive system exhibits a quicker rise time of 0.1192 seconds compared to the active PID-ZN system's rise time of 0.4575 seconds. However, the active PID-ZN system shows a better performance in terms of overshoot and settling time. The passive system has an overshoot of 5.823%, while the active PID-ZN system significantly reduces this to 1.779%. Similarly, the settling time is considerably longer for the active system at 9.466 seconds compared to 4.486 seconds for the passive system. This indicates that while the active PID-ZN system responds more slowly initially, it ultimately provides a smoother and more controlled displacement response.

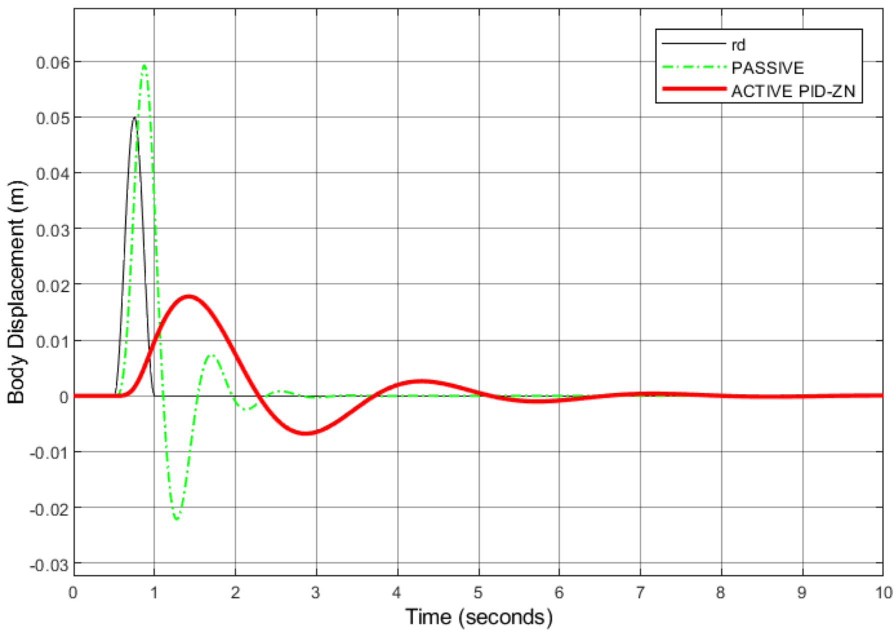



Figure 4.1: Body displacement for one bump disturbance

Table 4.2: The value of body displacement using Ziegler Nichols method for road profile 1

| | **PASSIVE** | **ACTIVE PID-ZN** |
|---|---|---|
| Rise Time (sec) | 0.1192 | 0.4575 |
| Settling Time (sec) | 4.486 | 9.466 |
| Overshoot (%) | 5.823 | 1.779 |

Regarding body acceleration, depicted in Figure 4.2, the active PID-ZN system again shows an improvement in performance over the passive system. The rise time for the passive system is 0.0846 seconds, while the active system is slightly slower at 0.1314 seconds. The passive system experiences an overshoot of 2.141%, whereas the active PID-ZN system reduces this to 1.624%. Additionally, the settling time for the passive system is 4.513 seconds compared to 5.150 seconds for the active system. This indicates that the active PID-ZN system, although slightly slower in initial response, provides a more stable acceleration profile with less overshoot and slightly longer settling time, resulting in improved ride comfort and stability.

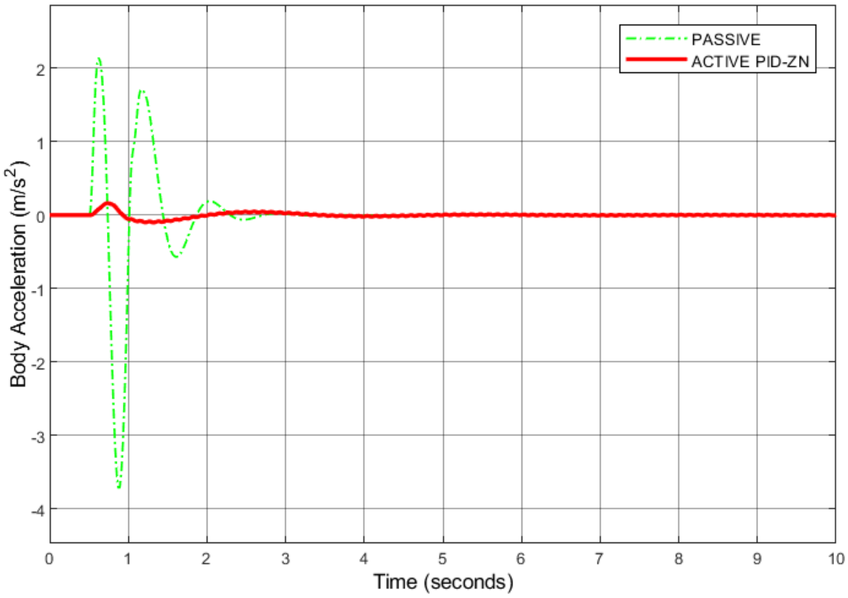



Figure 4.2: Body acceleration for one bump disturbance

Table 4.3: The value of body acceleration using Ziegler Nichols method for road profile 1

| | **PASSIVE** | **ACTIVE PID-ZN** |
|---|---|---|
| Rise Time (sec) | 0.0846 | 0.1314 |
| Settling Time (sec) | 4.513 | 5.150 |
| Overshoot (%) | 2.141 | 1.624 |

In conclusion, the active PID-ZN system offers better performance in terms of reducing overshoot and providing smoother responses in both body displacement and acceleration, although it has slightly longer rise and settling times compared to the passive system.

- **Road Profile 2**

For body displacement, as illustrated in Figure 4.3, the passive system exhibits a rise time of 0.1301 seconds, which is quicker than the 0.2935 seconds observed in the active PID-ZN system. However, the active system demonstrates superior performance in terms of overshoot and settling time. Specifically, the passive system's overshoot is considerably higher at 9.739%, whereas the active PID-ZN system reduces this to 3.943%. Additionally, the settling time is shorter for the passive system at 3.141 seconds, while the active system has a longer settling time of 5.277 seconds. This suggests that although the active PID-ZN system responds more slowly at first, it ultimately provides a more controlled and stable displacement response, with significantly less overshoot.

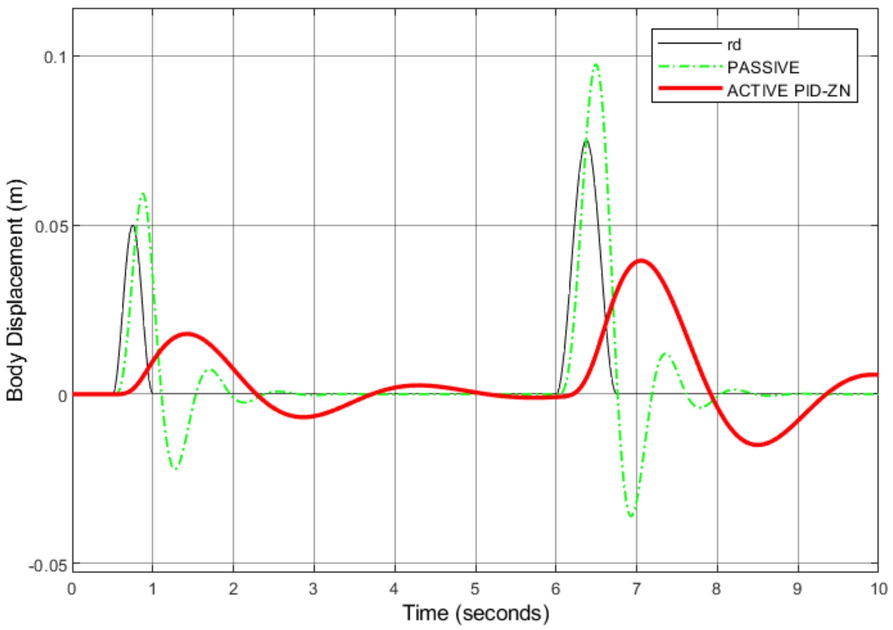


Figure 4.3: Body displacement for two bumps disturbance

Table 4.4: The value of body displacement using Ziegler Nichols method for road profile 2

| | **PASSIVE** | **ACTIVE PID-ZN** |
|---|---|---|
| Rise Time (sec) | 0.1301 | 0.2935 |
| Settling Time (sec) | 3.141 | 5.277 |
| Overshoot (%) | 9.739 | 3.943 |



Regarding body acceleration, illustrated in Figure 4.4, the active PID-ZN system shows a much more significant improvement over the passive system. The passive system has a rise time of 0.0772 seconds, whereas the active PID-ZN system achieves a quicker rise time of 0.02288 seconds. More importantly, the active PID-ZN system drastically reduces the overshoot to 0.2369%, compared to 2.758% in the passive system. The settling time for the passive system is 3.876 seconds, slightly shorter than the active system's 4.101 seconds. These results indicate that the active PID-ZN system not only responds more rapidly but also ensures a much smoother acceleration profile with minimal overshoot.

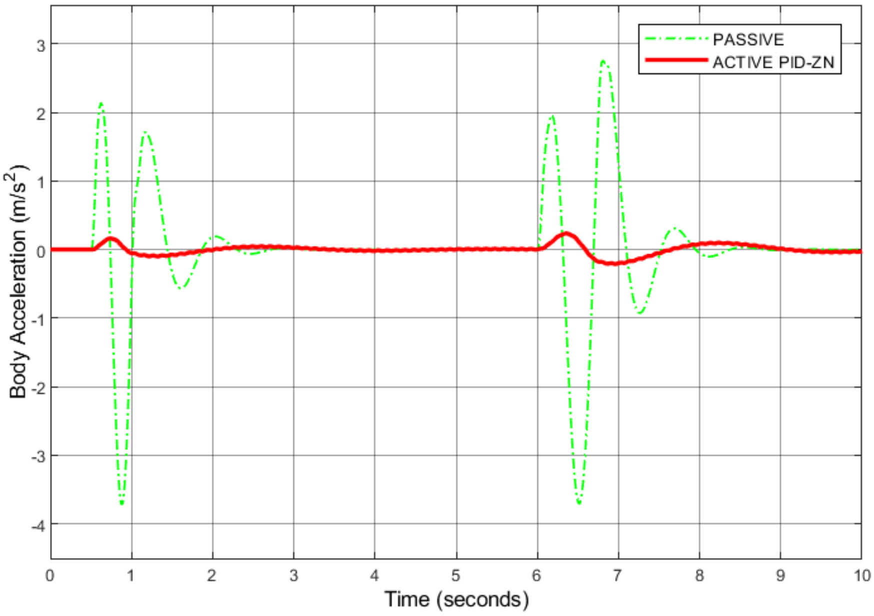



Figure 4.4: Body acceleration for two bumps disturbance

Table 4.5: The value of body acceleration using Ziegler Nichols method for road profile 2

| | **PASSIVE** | **ACTIVE PID-ZN** |
|---|---|---|
| Rise Time (sec) | 0.0772 | 0.02288 |
| Settling Time (sec) | 3.876 | 4.101 |
| Overshoot (%) | 2.758 | 0.2369 |

In conclusion, the active PID-ZN control system offers substantial enhancements in terms of reducing overshoot and providing a smoother response in both body displacement and acceleration for road profile 2. Despite a slightly longer rise and settling time in displacement, the active system significantly improves ride comfort and stability, particularly by minimizing overshoot in both displacement and acceleration responses.

### 4.2.2 Optimized PID controller using PSO Algorithm

In this section, the PID controller has been optimized using the PSO algorithm with ITAE, ITSE, IAE and ISE performance.

Table 4.6: The parameter that is set in the evaluation of PSO algorithm performance for road profile 1 and 2

| Parameter | Values |
|---|---|
| Number of iterations | 100 |
| Number of particles | 30 |
| Number of variables | 3 |
| Upper bound | 3500 5500 25 |
| Lower bound | 0.1 0.1 0.1 |



Table 4.6 shows the value of the gain controller that is proportional gain, integral gain and derivative gain with the variable error performance index to evaluate the performance of control systems. Each of these indices has a specific way of weighing the error over time to provide a measure of the system's performance. The best value of the gain controller will be put in the active suspension system for road profiles 1 and 2 with the optimal error performance index.

- **Road Profile 1**

Table 4.7: The optimized PID controller parameters with the different error performance index for road profile 1.

| Parameter | Proportion gain, Kp | Integral gain, Ki | Derivative gain, Kd |
|---|---|---|---|
| ITAE | 3499.6205 | 5499.6109 | 16.253461 |
| ITSE | 3495.5099 | 5499.1393 | 7.6733795 |
| IAE | 3499.6399 | 5499.9912 | 0.42103393 |
| ISE | 3499.6607 | 5499.9251 | 0.10258545 |

Table 4.7 shows the optimized PID controller parameters with the different error performance index for road profile 1. The elements of the step response characteristic include Rise Time (Tr), Settling Time (Ts) and Overshoot (OS). The results in Table 4.8 show that the choice of error performance index significantly influences these step response characteristics. For body displacement, all indices result in similar rise times around 0.367 to 0.368 seconds, but the settling time and overshoot vary more noticeably. Specifically, the ITAE-optimized controller has the shortest settling time at 5.553 seconds, whereas the ISE-optimized controller has the longest at 7.500 seconds and overshoot remains consistent across all indices, hovering around 1.48%.

In contrast, Table 4.9 reveals more pronounced differences in the step response characteristics of acceleration. The ITAE index yields the fastest rise time at 0.05759 seconds and the shortest settling time at 3.709 seconds but results in a high overshoot of 23.59%. The ITSE, IAE, and ISE indices produce longer settling times and slightly higher overshoot percentages, with the ISE index showing the longest settling time of 6.973 seconds. These variations suggest that while ITAE provides a quicker response and stabilization for body displacement, it may lead to higher overshoot and less damping in the acceleration response, highlighting the trade-offs involved in optimizing PID parameters for different performance criteria.

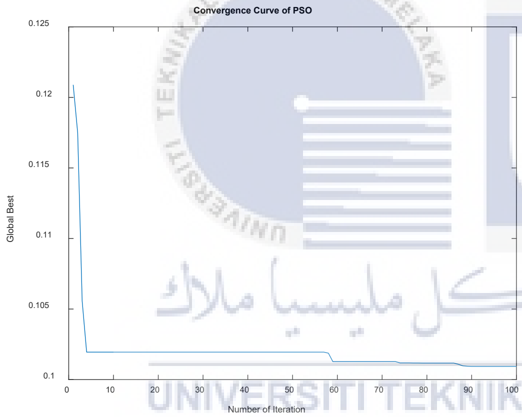
Table 4.8: The step response characteristic of body displacement of road profile 1 with different error performance index.

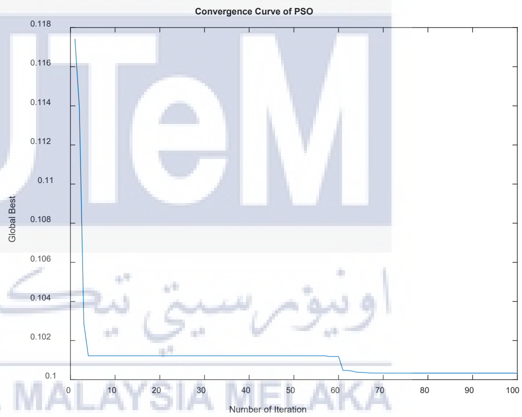
| Parameter | Rise Time (sec) | Settling Time (sec) | Overshoot (%) |
|---|---|---|---|
| ITAE | 0.3685 | 5.553 | 1.482 |
| ITSE | 0.3678 | 7.490 | 1.487 |
| IAE | 0.3674 | 7.500 | 1.483 |
| ISE | 0.3675 | 7.500 | 1.482 |

Table 4.9: The step response characteristic of acceleration of road profile 1 with different error performance index.

| Parameter | Rise Time (sec) | Settling Time (sec) | Overshoot (%) |
|---|---|---|---|
| ITAE | 0.05759 | 3.709 | 23.59 |
| ITSE | 0.05644 | 4.902 | 23.86 |
| IAE | 0.05278 | 4.902 | 23.96 |
| ISE | 0.05401 | 6.973 | 23.80 |

Figure 4.5 clearly shows the convergence curve of PSO for each error performance index. The data for the body displacement and body acceleration is represented as the step response characteristic had been tabulated in Table 4.8 and Table 4.9 and clearly explain the results. Figure 4.6 shows that the better graph has been optimized using ITAE.

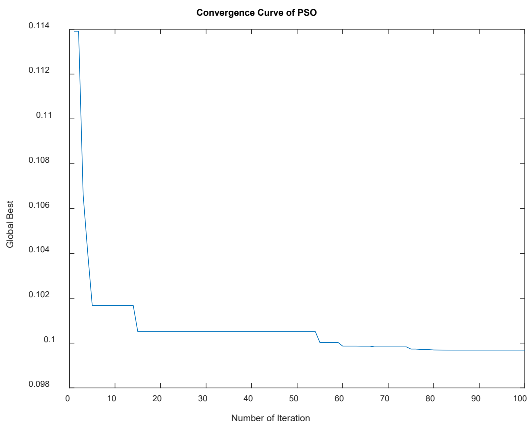


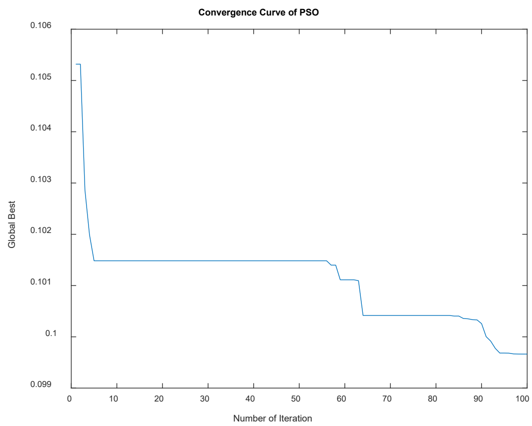




(a)

(b)






(c)

(d)

Figure 4.5: The convergence curve of PSO with difference error

performance index, (a) ITAE, (b) ITSE, (c) IAE and (d) ISE for road profile 1

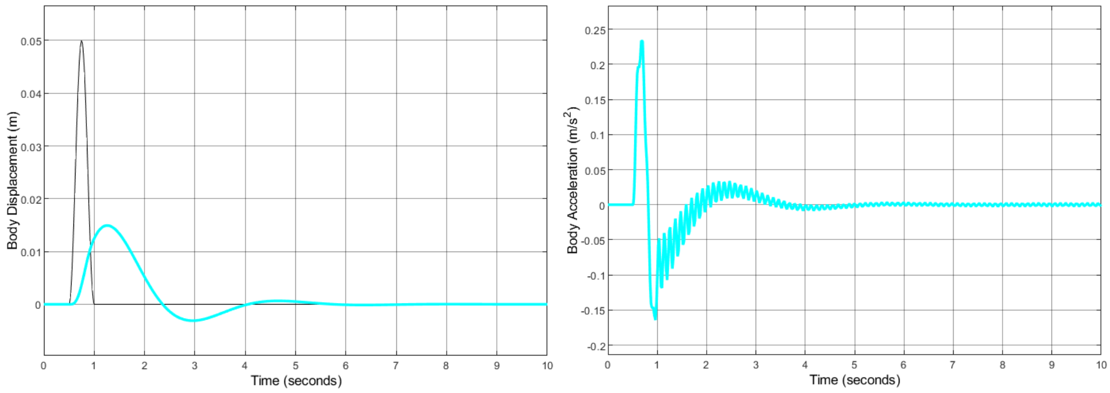



Figure 4.6: The graph of the body displacement and body acceleration for road profile 1 with the ITAE.

- **Road Profile 2**

Table 4.10: The optimized PID controller parameters with the different error performance index for road profile 2.

| Parameter | Proportion gain, Kp | Integral gain, Ki | Derivative gain, Kd |
|---|---|---|---|
| ITAE | 3499.8251 | 5497.0601 | 2.4958769 |
| ITSE | 3498.5405 | 5493.7692 | 0.11575301 |
| IAE | 3499.4231 | 5499.8675 | 4.7654438 |
| ISE | 3499.8814 | 5498.2554 | 0.10339797 |

Table 4.10 shows the optimized PID controller parameters with the different error performance index for road profile 2. The elements of the step response characteristic include Rise Time (Tr), Settling Time (Ts) and Overshoot (OS). The results in Table 4.11 show that the step response characteristics for body displacement of road profile 2 exhibit minimal variation in rise time across different error indices, all approximately 0.2915 seconds. The settling times are also quite similar, with the ITAE and ITSE indices resulting in the shortest settling times of 5.593 and 5.553 seconds, respectively, while IAE has a slightly longer settling time of 5.711 seconds and overshoot percentages are consistent across all indices, which is around 3.26%.

In addition, the step response characteristics for acceleration, shown in Table 4.12, demonstrate more significant differences. The ITAE-optimized controller results in the fastest rise time of 0.01906 seconds but also exhibits the highest overshoot at 26.85%. The settling times vary, with the ISE-optimized controller achieving the shortest settling time of 4.213 seconds, whereas the other indices have settling times slightly above 5 seconds. The overshoot percentages for acceleration remain high across all indices, slightly below or above 26.6%. This data suggests that while all indices perform similarly for body displacement, with minor variations in settling time, the ITAE index provides the quickest response for acceleration at the cost of higher overshoot. In contrast, the ISE index offers a more balanced performance with shorter settling times and slightly lower overshoot.

Table 4.11: The step response characteristic of body displacement of road profile 2 with different error performance index.

| **Parameter** | **Rise Time (sec)** | **Settling Time (sec)** | **Overshoot (%)** |
|---|---|---|---|
| ITAE | 0.2915 | 5.593 | 3.261 |
| ITSE | 0.2914 | 5.553 | 3.259 |
| IAE | 0.2916 | 5.711 | 3.263 |
| ISE | 0.2914 | 5.553 | 3.257 |



Table 4.12: The step response characteristic of body acceleration of road profile 2 with different error performance index.

| **Parameter** | **Rise Time (sec)** | **Settling Time (sec)** | **Overshoot (%)** |
|---|---|---|---|
| ITAE | 0.01906 | 5.490 | 26.85 |
| ITSE | 0.03634 | 5.416 | 26.60 |
| IAE | 0.04649 | 5.481 | 26.68 |
| ISE | 0.03629 | 4.213 | 26.58 |

Figure 4.7 clearly shows the convergence curve of PSO for each error performance index. The data for the body displacement and body acceleration is represented as the step response characteristic had been tabulated in Table 4.11 and Table 4.12 and clearly explain the results. Figure 4.8 shows that the better graph has been optimized using ISE.

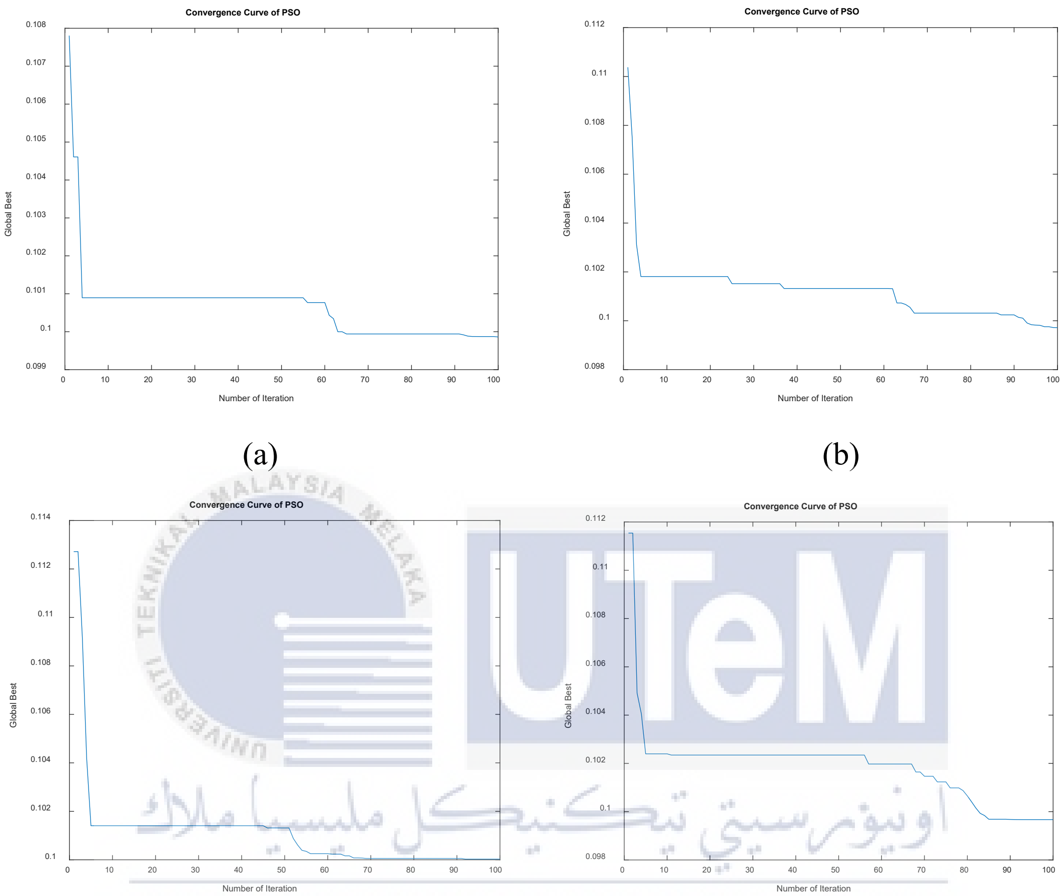





Figure 4.7: The convergence curve of PSO with difference error performance index, (a) ITAE, (b) ITSE, (c) IAE and (d) ISE for road profile 2

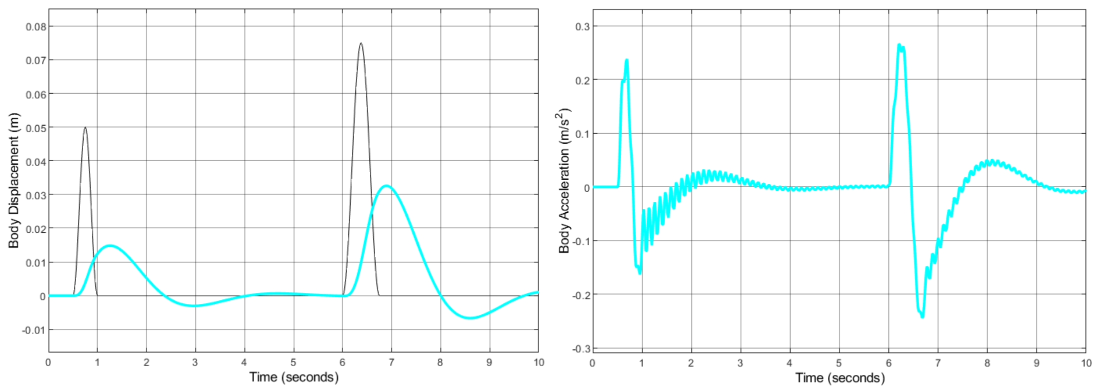



Figure 4.8: The graph of the body displacement and body acceleration for road profile 2 with the ISE.

## 4.3 Comparison results between PASSIVE, ACTIVE PID-ZN and ACTIVE PID-PSO

The optimization of the PID controller using the PSO algorithm for road profile 1, characterized by a single bump disturbance, reveals substantial improvements in both body displacement and acceleration metrics. From Figure 4.13, the Active PID-PSO achieves a rise time of 0.3685 seconds, which is faster than the Active PID-ZN which is 0.4575 seconds but slower than the passive system's 0.1192 seconds. However, the Active PID-PSO significantly reduces overshoot to 1.492%, compared to 1.779% for the Active PID-ZN and 5.823% for the passive system. Additionally, the Active PID-PSO decreases the settling time to 5.553 seconds, which is shorter than the Active PID-ZN's 9.466 seconds but longer than the passive system's 4.486 seconds. This indicates that while the PSO-optimized controller is slightly slower to respond initially, it provides a more stable and controlled displacement response with reduced overshoot.

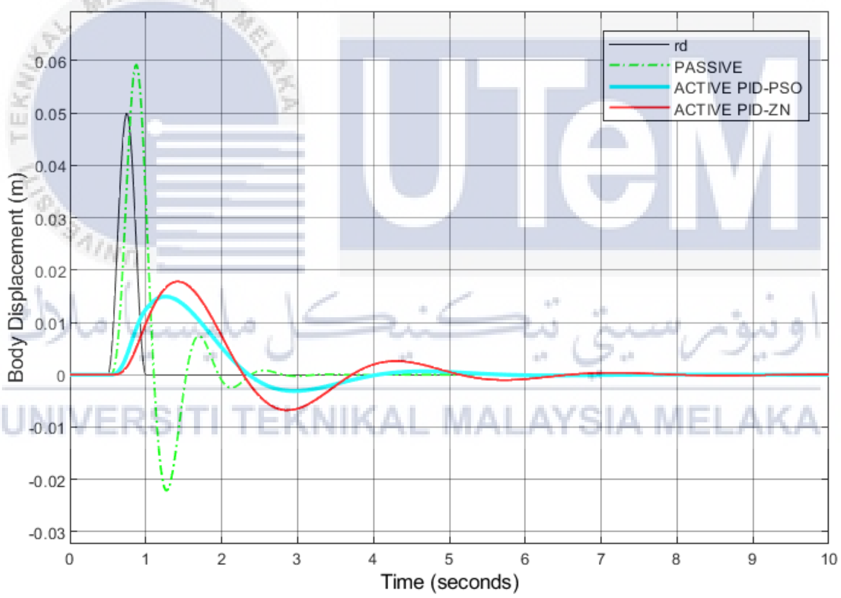



Figure 4.9: Body displacement for one bump

Table 4.13: The value of body displacement using PSO Algorithm for road profile 1

| | PASSIVE | ACTIVE PID-ZN | ACTIVE PID-PSO |
|---|---|---|---|
| Rise Time (sec) | 0.1192 | 0.4575 | 0.3685 |
| Settling Time (sec) | 4.486 | 9.466 | 5.553 |
| Overshoot (%) | 5.823 | 1.779 | 1.482 |

In terms of body acceleration as shown in Figure 4.14, the PSO-optimized PID controller shows even more pronounced benefits. The rise time for the Active PID-PSO is 0.05759 seconds, faster than both the Active PID-ZN which is 0.1314 seconds and the passive system's 0.0846 seconds. The overshoot for the Active PID-PSO is 23.59%, slightly higher than the Active PID-ZN's 16.24% but still comparable to the passive system's 21.41%. Importantly, the settling time for the Active PID-PSO is 3.709 seconds, which is shorter than both the Active PID-ZN's 5.150 seconds and the passive system's 4.513 seconds. These results highlight that the PSO algorithm effectively fine-tunes the PID controller parameters, resulting in quicker response times and reduced settling times for body acceleration, although with a marginal increase in overshoot.

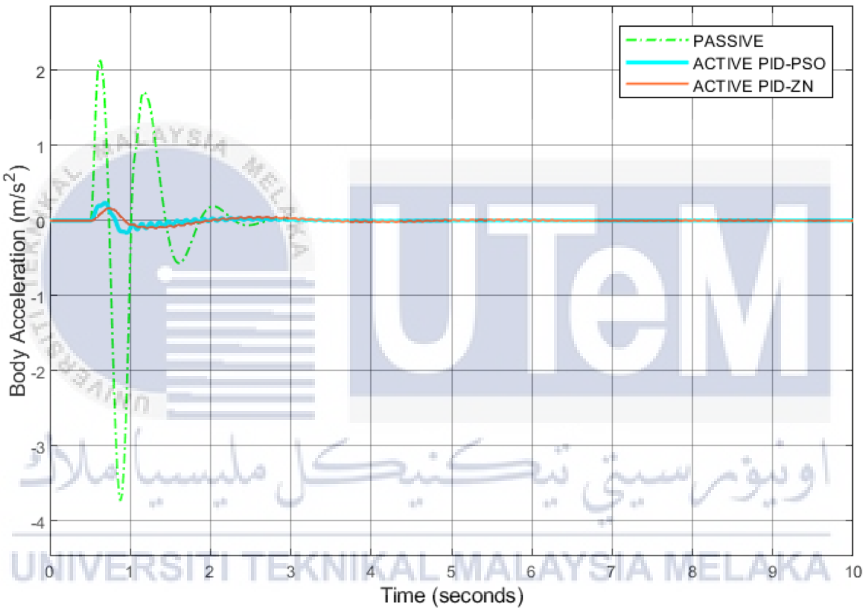



Figure 4.10: Body acceleration for one bump

Table 4.14: The value of body acceleration using PSO Algorithm for road profile 1

| | PASSIVE | ACTIVE PID-ZN | ACTIVE PID-PSO |
|---|---|---|---|
| Rise Time (sec) | 0.0846 | 0.1314 | 0.05759 |
| Settling Time (sec) | 4.513 | 5.150 | 3.709 |
| Overshoot (%) | 21.41 | 16.24 | 23.59 |

In conclusion, the optimized PID controller using PSO algorithm offers a significant enhancement in car performance over road profile 1. By providing quicker and more stable responses in both body displacement and acceleration, the PSO optimization improves ride comfort and stability, making it a superior choice compared to both the passive system and the conventional Active PID-ZN controller.

- **Road Profile 2**

The results from Tables 4.15 and 4.16, along with Figures 4.11 and 4.12, illustrate the performance of the optimized PID controller using the PSO algorithm for body displacement and acceleration in Road Profile 2. For body displacement, the PSO-optimized PID controller exhibits a rise time of 0.2914 seconds, a settling time of 5.553 seconds, and an overshoot of 3.257%. These values indicate a well-tuned system with a balance between quick response and stability. Compared to the active PID-ZN controller, which has a rise time of 0.2935 seconds, settling time of 5.277 seconds, and overshoot of 3.943%, the PSO-optimized controller demonstrates slightly faster response and significantly reduced overshoot. The passive system, while showing the quickest rise time of 0.1301 seconds and shortest settling time of 3.141 seconds, suffers from a much higher overshoot of 9.739%, highlighting its instability.

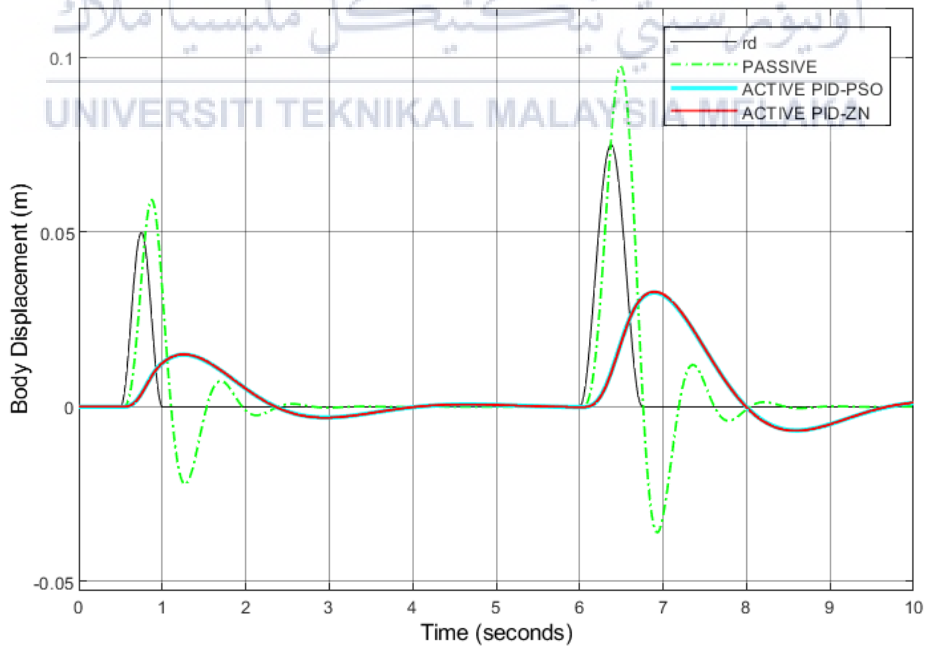



Figure 4.11: Body displacement for two bumps

Table 4.15: The value of body displacement using PSO Algorithm for road profile 2

| | PASSIVE | ACTIVE PID-ZN | ACTIVE PID-PSO |
|---|---|---|---|
| Rise Time (sec) | 0.1301 | 0.2935 | 0.2914 |
| Settling Time (sec) | 3.141 | 5.277 | 5.553 |
| Overshoot (%) | 9.739 | 3.943 | 3.257 |

For body acceleration, the PSO-optimized PID controller achieves a rise time of 0.03629 seconds, a settling time of 4.213 seconds, and an overshoot of 26.58%. This indicates a moderate improvement over the active PID-ZN controller, which has a rise time of 0.02288 seconds, settling time of 4.101 seconds, and overshoot of 23.69%. The passive system again shows a faster rise time of 0.0772 seconds but at the cost of a significantly higher overshoot of 27.58% and a settling time of 3.876 seconds. The PSO-optimized controller, therefore, provides a more controlled and stable response, reducing excessive oscillations and improving the overall performance for road profile 2. This balanced performance, especially in managing overshoot, underscores the effectiveness of the PSO algorithm in optimizing PID controller parameters for varying road conditions. In conclusion, these results clearly show that the Active PID-PSO controller provides a superior driving experience, with faster response times, less overshoot, and quicker stabilization, making it the best choice for enhancing car stability and passenger comfort under various road disturbances.



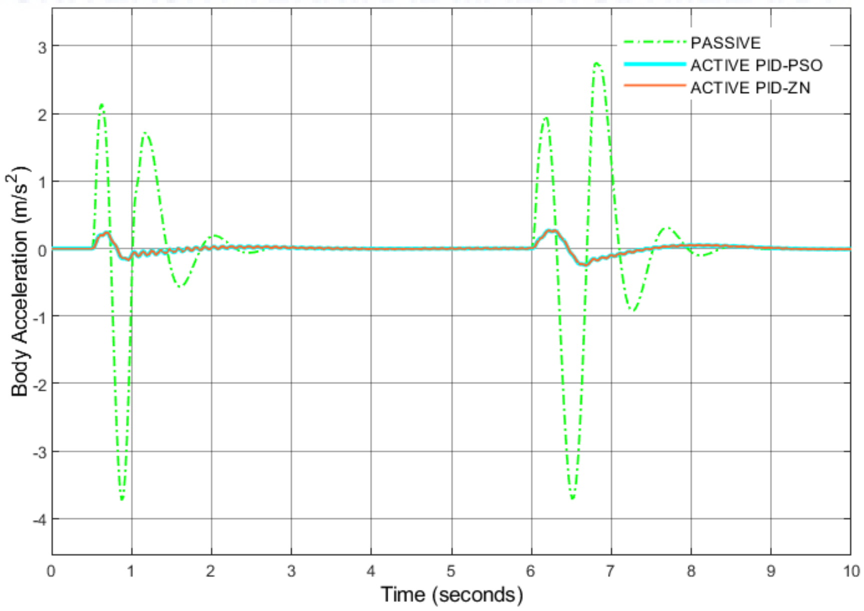



Figure 4.12: Body acceleration for two bumps

Table 4.16: The value of body acceleration using PSO Algorithm for road profile 2

| | **PASSIVE** | **ACTIVE PID-ZN** | **ACTIVE PID-PSO** |
|---|---|---|---|
| Rise Time (sec) | 0.0772 | 0.02288 | 0.03629 |
| Settling Time (sec) | 3.876 | 4.101 | 4.213 |
| Overshoot (%) | 27.58 | 23.69 | 26.58 |

# CHAPTER 5

## CONCLUSION AND RECOMMENDATIONS

### 5.1 Conclusion

In this work, a mathematical model of a quarter car has been derived, the governing equations have been simulated using Simulink MATLAB. The study consisted of applying different road disturbances to the model, first the model was given a one bump and two bumps. Results showed that not all the optimized PID using PSO Algorithm shows better results and performed much better than the classical PID using Ziegler Nichols method and passive model. By fine-tuning the parameters (Kp, Ki, Kd) using the PSO algorithm, the active suspension system demonstrates improved characteristics. Based on the performance index considered (ITAE, ITSE, IAE, and ISE), the PSO algorithm consistently outperforms classical methods. It achieves lower settling time, faster rise time, and a reasonable level of overshoot, which are desirable qualities for an active suspension system. The PSO algorithm, with its ability to search for optimal solutions in a large parameter space, proves effective in optimizing the PID controller for the active suspension system. This demonstrates the algorithm's potential for optimizing control systems in other applications as well. In conclusion, the optimization PID controller using the PSO algorithm is a promising and effective approach for improving the performance of the active suspension system.



### 5.2 Future work

To enhance the performance of the PID controller for the active suspension system, the focus will be on experimental validation by implementing the optimized controller on a physical vehicle or a high-fidelity simulator. This real-world testing will allow fine-tuning and adjustments based on practical considerations. Additionally, unforeseen challenges or limitations of the optimized controller will be identified through this validation process. Furthermore, exploration of advanced optimization algorithms, such as genetic algorithms, PSO variants, or reinforcement learning, aims to further improve the controller's performance. Nonetheless, additional in-depth research should be carried out and to enhance the number of studies to be included in the PSO algorithm, it is advised that the database source be expanded.

In addition, the mathematical model using MATLAB can be further improved and validated using experimental results. Furthermore, comparative analysis of outcomes from different optimization approaches provides valuable insights into their strengths and limitations in optimizing active suspension systems.

# REFERENCES


[1] D. D. D. Das, M. S. Kumar, and S. R. Gampa, "Optimum PID Controller Design using PSO for Vehicle Active Suspension System Considering MATLAB Simulink Modeling based Road Profiles," J. Electr. Eng., no. July, pp. 286–295, 2017, [Online]. Available: http://www.jee.ro/index.php/jee/article/view/WW1467987284W577fb5541cf9e%0Ahttp://www.jee.ro/index.php/jee/article/download/WW1467987284W577fb5541cf9e/1521

[2] A. J. Qazi, U. A. Farooqui, A. Khan, M. T. Khan, F. Mazhar, and A. Fiaz, "Optimization of Semi-active Suspension System Using Particle Swarm Optimization Algorithm," *AASRI Procedia*, vol. 4, pp. 160–166, 2013, doi: 10.1016/j.aasri.2013.10.025.

[3] H. Emad, A. Mary, and M. M. Ali, "Control of Active Car Suspension System Based on 2PID and PSO Optimization," Texas J. Eng. Technol., vol. 7, pp. 155–170, 2022, [Online]. Available: htps://zienjounals.com

[4] T. R. Rao and P. Anusha, "Active suspension system of a 3 DOF quarter car using fuzzy logic control for ride comfort," *CARE 2013 - 2013 IEEE Int. Conf. Control. Autom. Robot. Embed. Syst. Proc.*, pp. 1–6, 2013, doi: 10.1109/CARE.2013.6733771.

[5] T. N. L. Vu, D. Van Dung, N. Van Trang, and P. T. Hai, "Analytical design of PID controller for enhancing ride comfort of active vehicle suspension system," in *Proceedings - 2017 International Conference on System Science and Engineering, ICSSE 2017*, IEEE, 2017, pp. 305–308. doi: 10.1109/ICSSE.2017.8030886.

[6] F. A. Ansari and R. Taparia, "Modeling, Analysis and Control of Active Suspension System using Sliding Mode Control and Disturbance Observer," *Int. J. Sci. Res. Publ.*, vol. 3, no. 1, pp. 1–6, 2013, [Online]. Available: www.ijsrp.org

[7] S. S. Eligar and R. M. Banakar, "A Survey on Passive, Active and Semiactive Automotive Suspension Systems and Analyzing Tradeoffs in Design of Suspension Systems," 2018 Int. Conf. Recent Innov. Electr. Electron. Commun. Eng. ICRIEECE 2018, pp. 2908–2913, 2018, doi: 10.1109/ICRIEECE44171.2018.9008620.

[8] A. H. Mohamed, D. Abidou, and S. A. Maged, “LQR and PID Controllers Performance on a Half Car Active Suspension System,” 2021 Int. Mobile, Intelligent, Ubiquitous Comput. Conf. MIUCC 2021, pp. 48–53, 2021, doi: 10.1109/MIUCC52538.2021.9447609.

[9] J. Bharali and M. Buragohain, “A comparative analysis of PID, LQR and Fuzzy logic controller for active suspension system using 3 Degree of Freedom quarter car model,” *1st IEEE Int. Conf. Power Electron. Intell. Control Energy Syst. ICPEICES 2016*, pp. 1–5, 2017, doi: 10.1109/ICPEICES.2016.7853152.

[10] S. Davoodi Moghadam, S. Balochian, and O. Salehyan Rad, “PSO Base Tuning of the PID Control of Active Suspension System,” *Int. J. Artif. Intell. Mechatronics*, vol. 3, no. 1, pp. 6–11, 2014

[11] Z. Boulaaras, A. Aouiche, and K. Chafaa, “PID controller optimized by using Genetic Algorithm for Active Suspension System of a Quarter car,” *2022 19th IEEE Int. Multi-Conference Syst. Signals Devices, SSD 2022*, pp. 779–784, 2022, doi: 10.1109/SSD54932.2022.9955858.

[12] M. H. Mat and I. Z. Mat Darns, “Self-tuning PID controller for active suspension system with hydraulic actuator,” *IEEE Symp. Comput. Informatics, Isc. 2013*, no. 4, pp. 86–91, 2013, doi: 10.1109/ISCI.2013.6612381.

[13] M. S. Sadeghi, S. Varzandian, and A. Barzegar, “Optimization of Classical PID and Fuzzy PID Controllers of a nonlinear quarter car suspension system using PSO algorithm,” *2011 1st Int. eConference Comput. Knowl. Eng. ICCKE 2011*, pp. 172–176, 2011, doi: 10.1109/ICCKE.2011.6413346.

[14] Y. Chu, K. Bao, and M. Bian, “Research on the Control Strategy of Tracked Vehicle Semi-Active Suspension Based on Improved Particle Swarm Algorithm,” *IET Conf. Proc.*, vol. 2022, no. 13, pp. 797–801, 2022, doi: 10.1049/ICP.2022.1917.

[15] A. K. Abdulzahra and T. Y. Abdalla, “Fuzzy Sliding Mode Control Scheme for Vehicle Active Suspension System Optimized by ABC Algorithm,” *Int. J. Intell. Syst. Appl.*, vol. 11, no. 12, pp. 1–10, 2019, doi: 10.5815/ijisa.2019.12.01.

[16] M. H. A. Talib and I. Z. Mat Darus, "Development of fuzzy logic controller by particle swarm optimization algorithm for semi-active suspension system using magneto-rheological damper," *WSEAS Trans. Syst. Control*, vol. 9, no. 1, pp. 77–85, 2014.

[17] J. Hurel, J. Amaya, J. Peralta, D. Alvarado, and F. Flores, "Particle Swarm Optimization applied on Fuzzy Control: Comparative analysis for an Quarter-car Active Suspension Model," *Proc. IEEE Int. Conf. Ind. Technol.*, vol. 2022-Augus, 2022, doi: 10.1109/ICIT48603.2022.10002809.

[18] "Control Tutorial for MATLAB & Simulink." [Online]. Available: https://ctms.engin.umich.edu/CTMS/index.php?example=Suspension&section=ControlPID

[19] M. Jamil, S. Zafar, and S. O. Gilani, "Designing PID controller based semi-active suspension system using MATLAB simulink," *Lect. Notes Inst. Comput. Sci. Soc. Telecommun. Eng. LNICST*, vol. 224, pp. 282–295, 2018, doi: 10.1007/978-3-319-94180-6_27.

[20] N. Al-Holou, T. Lahdhiri, D. S. Joo, J. Weaver, and F. Al-Abbas, "Sliding mode neural network inference fuzzy logic control for active suspension systems," *IEEE Trans. Fuzzy Syst.*, vol. 10, no. 2, pp. 234–246, 2002, doi: 10.1109/91.995124.

[21] M. Ranade and J. Zhou, "Modelling and Simulation of an Active Vehicle Suspension System," pp. 1–15, 2022, doi: 10.18260/1-2-620-39067.

[22] S. M. Salah, A. Abdelaziz, C. Kheireddine, A. Elmoundher, T. Mounia, and F. Zaamouche, "Comparison between a Passive and Active Suspension Vehicle using PID and Fuzzy Controllers with Two Entries Applied on Quarter Vehicle Model," *Proc. - 2019 1st Int. Conf. Sustain. Renew. Energy Syst. Appl. ICSRESA 2019*, 2019, doi: 10.1109/ICSRESA49121.2019.9182349.

[23] S. Mouleeswaran, "Design and Development of PID Controller-Based Active Suspension System for Automobiles," *PID Controll. Des. Approaches - Theory, Tuning Appl. to Front. Areas*, 2012, doi: 10.5772/32611.

[24] S. Y. Han, J. F. Dong, J. Zhou, and Y. H. Chen, "Adaptive Fuzzy PID Control Strategy for Vehicle Active Suspension Based on Road Evaluation," *Electron.*, vol. 11, no. 6, 2022, doi: 10.3390/electronics11060921.

# APPENDICES

## APPENDIX A THE M. FILE USE TO TUNE PID CONTROLLER BY USING PSO ALGORITHM

```
%% Standard Particle Swarm Optimization (PSO) algorithm
clc;
close all;
clear variables;
clear global;

fitFunc = @FitnessFunction;

%% Define the details of the particles
noP = 30; % Increased number of particles
nVar = 3; % Number of variables
ub = [3500 5500 25]; % Upper bound
lb = [0.1 0.1 0.1]; % Lower bound

%% Define the PSO's parameters
maxIter = 100; % Maximum iteration
c1_initial = 2; % Initial Cognitive Component
c2_initial = 2; % Initial Social Component

wMax = 0.9; % Maximum value of Inertia Weight
wMin = 0.2; % Minimum value of Inertia Weight

vMax = (ub - lb) .* 0.2;
vMin  = -vMax;

%% Define variable for plotting
convergenceCurve = zeros(1, maxIter);

%% ====== THE PSO ALGORITHM ======

%% 1) Initialize the particles
Swarm.GBEST.X = zeros(1, nVar);
Swarm.GBEST.F = inf;

for k = 1:noP % Loop for noP number of times to create the particles
    Swarm.Particles(k).X = (ub - lb) .* rand(1, nVar) + lb;
    Swarm.Particles(k).F = 0;
    Swarm.Particles(k).V = zeros(1, nVar);
    Swarm.Particles(k).PBEST.X = zeros(1, nVar);
    Swarm.Particles(k).PBEST.F = inf;
end

%% 2) Main loop of the PSO algorithm
for t = 1:maxIter % Repeat for maxIter iterations
    % Update the inertia weight and coefficients dynamically
    w = wMax - t * ((wMax - wMin) / maxIter);
    c1 = c1_initial - 1.5 * (t / maxIter); % Decrease c1 over time
    c2 = c2_initial + 1.5 * (t / maxIter); % Increase c2 over time

    for k = 1:noP
        currentX = Swarm.Particles(k).X;
        Swarm.Particles(k).F = fitFunc(currentX, k);
```

```
        % Update the PBEST.X and PBEST.F (if necessary)
        if Swarm.Particles(k).F < Swarm.Particles(k).PBEST.F
            Swarm.Particles(k).PBEST.X = currentX;
            Swarm.Particles(k).PBEST.F = Swarm.Particles(k).F;
        end

        % Update the GBEST.X and GBEST.F (if necessary)
        if Swarm.Particles(k).F < Swarm.GBEST.F
            Swarm.GBEST.X = currentX;
            Swarm.GBEST.F = Swarm.Particles(k).F;
        end
    end

    % Update gains (X) and velocity (V)
    for k = 1:noP
        Swarm.Particles(k).V = w .* Swarm.Particles(k).V ...
            + c1 .* rand(1, nVar) .* (Swarm.Particles(k).PBEST.X - 
Swarm.Particles(k).X) ...
            + c2 .* rand(1, nVar) .* (Swarm.GBEST.X - Swarm.Particles(k).X);

        % Velocity clamping
        index1 = find(Swarm.Particles(k).V > vMax);
        index2 = find(Swarm.Particles(k).V < vMin);
        Swarm.Particles(k).V(index1) = vMax(index1);
        Swarm.Particles(k).V(index2) = vMin(index2);

        % Update position
        Swarm.Particles(k).X = Swarm.Particles(k).X + Swarm.Particles(k).V;

        % Position clamping with reflection
        index1 = find(Swarm.Particles(k).X > ub);
        index2 = find(Swarm.Particles(k).X < lb);
        Swarm.Particles(k).X(index1) = ub(index1) - (Swarm.Particles(k).X(index1) 
- ub(index1));
        Swarm.Particles(k).X(index2) = lb(index2) + (lb(index2) - 
Swarm.Particles(k).X(index2));
    end

    % Randomly reinitialize some particles if no improvement
    if mod(t, 10) == 0  % Every 10 iterations, for example
        for k = 1:ceil(noP / 5)  % Reinitialize 20% of particles
            Swarm.Particles(k).X = (ub - lb) .* rand(1, nVar) + lb;
            Swarm.Particles(k).V = zeros(1, nVar);
        end
    end

    outmsg = ['Iteration #', num2str(t), ' Swarm.GBEST.F = ', 
num2str(Swarm.GBEST.F)];
    disp(outmsg);

    convergenceCurve(t) = Swarm.GBEST.F;
end

% End of ====== THE PSO ALGORITHM ======

%% Announce the optimized value
outmsg = ['The Global Best variables = ', num2str(Swarm.GBEST.X)];
disp(outmsg);
```

```
%% Plot the convergence curve
figure;
plot(convergenceCurve);
title('Convergence Curve of PSO');
xlabel('Number of Iteration');
ylabel('Global Best');

figure;
semilogy(convergenceCurve);
title('Convergence Curve of PSO');
xlabel('Number of Iteration');
ylabel('Global Best (in log scale)');

end
```

# APPENDIX B FITNESS FUNCTION

```
% Mass-Spring-Damper system
% a single output objective function
% the input of this function is x and x is a vector

function [FitVal] = FitnessFunction(x,~)

assignin('base','Kp',x(1));
assignin('base','Ki',x(2));
assignin('base','Kd',x(3));

simOut = sim('MassSpringDamper_WVC');

t = simOut.tout;
absErr = simOut.absErr;

itae = trapz(t,t.*absErr);

FitVal = itae;

end
```