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ABSTRACT

A suspension system is responsible for the safety of vehicle during its manoeuvre. It serves the dual purpose of providing stability to the vehicle while providing a comfortable ride quality to the occupants. Recent trends in suspension system have focused on improving comfort and handling of vehicles while keeping the cost, space and feasibility of manufacturing in the constraint. This paper proposes a method for improving handling characteristics of a vehicle by controlling camber and toe angle using variable length arms in an adaptive manner. In order to study the effect of dynamic characteristics of the suspension system, a simulation study has been done in this work. A quarter car physical model with double wishbone suspension geometry is modelled in SolidWorks. It is then imported and simulated using SimMechanics platform in MATLAB. The output characteristics of the passive system (without variable length arms) were validated on MSC ADAMS software. The adaptive system intends to improve vehicle handling characteristics by controlling the camber and toe angles. This is accomplished by two telescopic arms with an actuator which changes the camber and toe angle of the wheel dynamically to deliver best possible traction and manoeuvrability. Two PID controllers are employed to trigger the actuators based on the camber and toe angle from the sensors for reducing the error existing between the actual and desired value. The arms are driven by actuators in a closed loop feedback manner with help of a separate control system. Comparison between active and passive systems is carried out by analysing graphs of various parameters obtained from MATLAB simulation. From the results, it is observed that there is a reduction of 58% in the camber and 96% in toe gain. Hence, the system provides the scope of considerable adaptive strategy in controlling dynamic characteristics of the suspension system.

1. Introduction

Over the years, automobiles have been evolving continuously and went through a lot of developments. These developments are a result of advancements in technology, advanced manufacturing methods and the need to satisfy customer expectations. Technological advancements in various automobile systems has been made possible by the incorporation of numerous mechatronic systems which resulted in better performance output [1]. These include a myriad of changes from the incorporation of a basic windshield wiper to an exquisite interior with an air-conditioning and infotainment system. Apart from the physical appearance of the vehicle, an important feature people delve for is enhanced comfort and safety in vehicles. The system that is majorly responsible for a vehicles' comfort level is the suspension system. The suspension system is an integral unit responsible for maintaining the stability of a vehicle under static and dynamic conditions [2]. The suspension system plays a vital role in keeping the occupants comfortable by absorbing road shocks and vibrations and keeping the passenger cabin secluded. Without the suspension system, the vibrations and shocks would also be directly transferred to the steering, thereby making it extremely hard to control the vehicle [3]. From its introduction in horse carts in the form of iron chains and leather belts to the present form, it has

Abbreviations: KPI, King Pin Inclination; PID, Proportional Integral Derivative; FVSA, Front View Swing Arm; SLA, Short Long Arm; IC, Instantaneous Center; RC, Roll Center; RCH, Roll Center Height.

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been one of the most important systems which influenced the level of satisfaction of a consumer when he is inside the vehicle. Hence, the automotive industry strives to make efforts in improving it in every possible way. But any suspension system would face various challenges due to dynamic terrain conditions like uneven road surface, rolling, pitching, yawing, vehicle speed, load shifts and effect due to external forces like wind gusts that have always required the system to find the right balance between itself [4]. To meet these performance requirements in the conventional system there is always a compromise between ride quality and dynamic properties. If a suspension system the operating parameters are tuned for optimum dynamic conditions might be too soft and the ride quality would decrease while improving the ride quality to satisfy passengers would reduce the vehicle manoeuvrability characteristics [5]. Hence, the packaging parameters of the suspension systems such as camber, caster, toe etc. must be set-up accordingly as they are responsible for the response of the system. By varying these parameters in an adaptive manner, the dynamic characteristics of the vehicle can be varied on a real-time basis.

This lead to the development of advanced suspension systems that contain active components ranging from simple self-levelling suspensions to fully active systems. Active suspension is a type of automotive suspension that controls the vertical movement of the wheels relative to the chassis or vehicle body with an onboard data acquisition system and independent actuators. These active systems use the disturbances from the road condition/terrain as input to the electronic control unit (ECU) and the suspension system is tuned accordingly to achieve optimal performance in real time condition [6]. The advantage of active suspension system is that its performance is optimized according to the dynamic road conditions thereby enhancing manoeuvrability and comfort on a real time basis, while in passive suspension systems the behaviour of suspension system is determined entirely by the system parameters and road surface [7]. Implementation of electromagnetic controls to the suspension system gave engineers enhanced control over the vehicle dynamic characteristics [8]. Various types of active suspension systems have been employed in higher end vehicles like Delphi's active suspension or Mercedes's Magneto-Rheological fluid (MRF) technology [9]. However, these systems have high cost and are extremely complex because of the intricate technologies besides requiring very frequent maintenance. Hence, extensive research has been focused on developing active suspension systems that are economical but can adapt to dynamic road conditions. Adaptability to road conditions is achieved by varying the wheel parameters to suit the terrain and by varying them in real time, a dynamic control is achieved [10]. Among the several parameters, camber and toe angle are two important attributes that maximize lateral grip and stability to a great extent. By varying these parameters the reaction of the vehicle can be optimized to the dynamically changing terrain which has been analysed and researched by various people in the past years.

2. Literature review

Thacker et al. [11] have focused their research on suspension arm designs and proposed a design based on topology with material optimization for controlling the arms in finite element analysis to improve the performance of the system. The review work postulates camber and toe as the two important performance parameters affecting vehicle handling characteristics and also determines the camber extremities as 5.5 degrees to 5.5 degrees. Arana et al. [12] proposed a variable geometry suspension with an electro-mechanic actuator which controls the pitching of the chassis and the position of the upper-end eye of the strut system to improve suspension behaviour. The work successfully manages to reduce maximum squatting and diving angles during transients by at least 30%. Groenendijk [13] proposed an idea of active toe control based on signals from longitudinal and lateral acceleration, steering angle and yaw rate sensors in a 4-wheel individual steering control system. From the experiment, it has been concluded that toe-control improves vehicle handling behaviour and also decreases the vehicle side slip angle to achieve better dynamic behaviour of the vehicle. Shad, [14] came up with the idea of mechatronic suspension system with multiple active degrees of freedom to actively change the camber, along with active steer and suspension system to increase the vehicle's lateral forces. The work successfully improved vehicle's lateral tire force by 28% enhancing the vehicle's traction, leading to increased turning capabilities. Choudhery [15] proposed the idea of variable camber suspension system using electro-mechanical devices to sense the lateral forces acting on the vehicle, and employs the camber adjusters to provide necessary response to improve vehicle stability during turns and cornering. Pourshams et al. [16] came up with the idea of using a pneumatic system for providing the variations in camber angle of a double wishbone suspension system to improve traction and vehicle safety. The modelled system was able to adjust the camber angle from −5 to 5 degrees but their performance in dynamic conditions has not been evaluated. Esfahani et al. [17] proposed the idea of varying the camber angle using hydraulic actuators to vary the geometry of suspension system components for better traction and stability. The system was able to provide the camber adjustment of −5 to 5 degrees with a short response time for improved adaptability.

Nemeth and Gaspar [18] presented the advantages of variable geometry suspension system, analysed the relation between steering and suspension and developed a control system to modify camber angle of front wheels during manuever. A change in camber angle is achieved through LPV methods and a change in yaw rate is also induced thereby improving vehicle stability. Nguyen et al. [19] presented the application of linear parameter varying (LPV) based control system to differential brake moment and the auxiliary front wheel steering angle to change the camber angles of the wheels with 4 semi-active dampers in order to improve the tracking of the road trajectory. Nemeth et al. [20] have proposed an LPV based control design for a variable geometry suspension system to reduce the lateral force during wheel tilting and the strategy incorporates the nonlinear tire characteristics. The tilting actuation of the wheel during cornering provides the additional lateral grip required to achieve better performance during manoeuvre. Tandel et al. [21] have studied the implementation of a Proportional Integral Derivative (PID) controller on a suspension system with various combinations of spring parameters and damping constants to reduce the vertical body acceleration. It has been found that after PID implementation to control suspension parameters, the vertical body acceleration reduced by almost 50%.

From various researches, it can be inferred that the active suspension control is accomplished by varying the camber, toe and damping coefficient of suspension system on a real time basis. By means of employing an electromechanical system, it is possible to achieve active camber control which results in improved vehicle stability and traction [22]. Also, it can be concluded from various researches that active toe control helps in decreasing wheel slip rate thereby improving a vehicle's dynamic behaviour.

3. Objective of the present work

Although there are several works in the field of active suspension system, very few work has been attempted to control the suspension system through a mechatronic system involving a PID controller and linear mechanical actuators. There have been several attempts to control toe and camber angles separately in an active manner using several mechanisms with the objective of improving the vehicle dynamic characteristics. Past researches have proven the significance of an independent adaptive control system. However, the active control of camber and toe angles in a simultaneous manner has not been reported in any research work.

The objective of our work is to develop a system that is capable of optimising both the camber and toe angles in a simultaneous manner to combine the advantages of both active toe control and active camber control systems. As a result, an integrated active camber and toe control system is proposed in this work for a double wishbone suspension system. The incorporation of mechatronic system with variable arms to adjust the wheel parameters in real time conditions monitored and optimized by a PID control system shows the novelty of the proposed idea. This paper is organised as follows: the next section covers the design description which will present the reason for choosing the double wishbone suspension system, it's advantages and drawbacks, the next section is the modelling section which portrays the modelling criteria and design parameters, followed by the description of the proposed system and its method of operation which is succeeded by design validation section. Then the simulation section provides the results of the simulation and its comparisons and followed by a conclusive summary.

4. Design description

In this work, a simple double wishbone suspension system has been chosen. The reason to choose this particular system is that it is an independent suspension system and it gives the designer the freedom to assign packaging parameters. It is easy to amend the output of the system for desired handling and comfort as mentioned in [23]. Hence it can be said that a double wishbone system fulfils the role that it has to play with the type of vehicle it is installed in by giving desirable handling characteristics [24]. It maintains good steering control during manoeuvring and also helps the vehicle to respond favourably to the control forces and provides isolation from high-frequency vibrations that are caused due to tire vibration in response to the road profile. Further, it is easier to change the wheel parameters of a double wishbone suspension under running conditions when compared to McPherson strut. The use of independent upper and lower arms provides more flexibility and freedom to adjust the parameters more precisely than a McPherson strut. This independent arrangement allows for controlling respective components of the system without affecting the entire system in a drastic scale.

In the process of designing, a designer can set various conditions such as ride height, roll centre height, king pin inclination, scrub radius and many other affecting factors including length of upper and lower arms for desired wheel travel path. He can also control the output behaviour such as camber, toe, and caster gain with respect to wheel travel and steering angle etc. [25]. Vehicles which use double wishbone have increased in negative camber with vertical movements of upper and lower arms, leading to better stability and handling performance as it translates to better stability properties for the car as the tires on the outside maintain more contact with the road surface. There are a few downsides to choosing a double wishbone system. Firstly it consumes more space as it is mounted horizontally causing an increase in vehicle width or reduced engine compartment area compared to conventional McPherson struts. It also proves to be expensive with respect to the conventional system because of the increased number of components. These along with the steering knuckle make it complex in designing.

The McPherson struts, on the other hand, are designed with more simplicity, and thus takes up less horizontal space. As a result, passenger compartment becomes spacious. They also have low un-sprung weight, an advantage that reduces the overall weight of the vehicle as well as increases the car's acceleration. Lower unsprung weight also makes the ride more comfortable. Nevertheless, the MacPherson struts come with their own drawbacks. It provides the designer very less freedom to set the parameters because of its simplicity and construction. It is a very tall assembly, making the system practically impossible on race cars with a low body. MacPherson Struts also have a problem with wider wheels as they can't be mounted without increasing the scrub radius. Hence, this paper focusses on the adaptive control of double wishbone system.

5. Modelling

The specification of the double wishbone system such as ride height, tire and rim size, spindle length and track width etc. that were chosen for the work are listed in Table 1. After finalizing the parameters mentioned in the table, the wheel offset is decided for the brake callipers and rotor assembly which reveals the outermost point for the lower ball joint is obtained and its height is defined based on the packaging factors as shown in Fig. 1.

The lower ball joint position which is adjusted with respect to the ground contact point is done to fix the kingpin inclination angle and it defines the scrub radius automatically. Then the upper ball joints position is decided on the kingpin axis and its height is decided based on the stress levels on upper arm and packaging ease. KPI in a combination of caster defines the camber gain characteristic while the vehicle is steered.

5.1. Front view geometry

The front view swing arm (FVSA) instant centre shown Fig. 2 is determined by the desired roll centre height and roll camber (RC) as follows. The corresponding equations are represented as follows (Eqs. (1)–(2)).

$$FVSA = \frac{t \times (1 - RC)}{2} \quad (1)$$

$t = track\ width$

$$RC = \frac{wheel\ camber\ angle}{chasis\ roll\ angle} \quad (2)$$

(With both measured relative to the road).

- The front swing arm length is first established (line A-A),
- Then the roll centre location is established and projected from ground contact point through RC to line AA, establishing IC.
- Then, the lines are projected from outer ball joints to IC and control arm lengths and the inner pivot point locations are obtained depending on the chassis requirements.

Table 1
Design parameters of the suspension system design.

Parameter	Value
Ride Height	205 mm
Toe Angle	0 degree
Camber Angle	0 degree
Caster Angle	4 degree
King Pin Angle	4 degree
Wheel	255/40R18
Track width	1398 mm

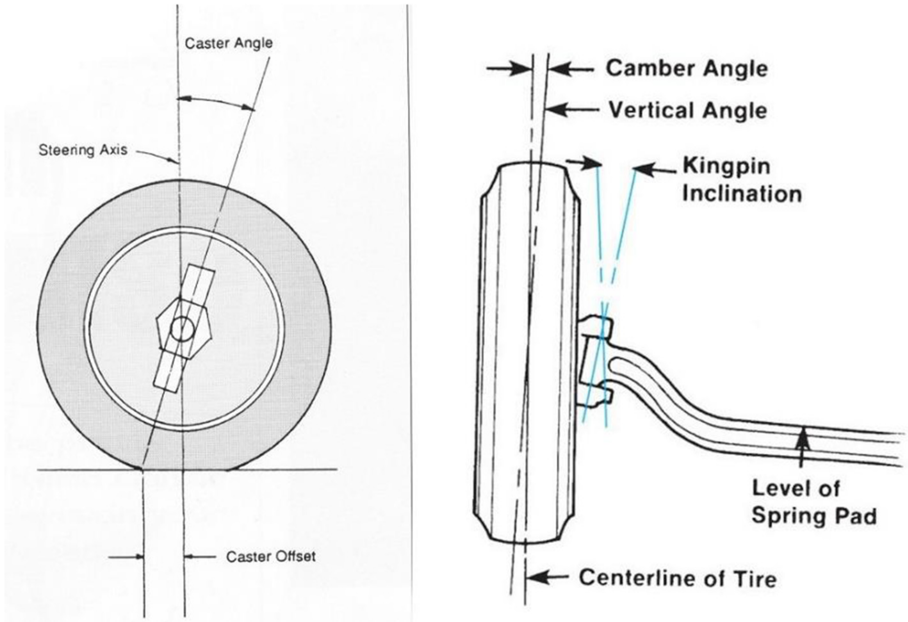

Fig. 1. Front suspension packaging.

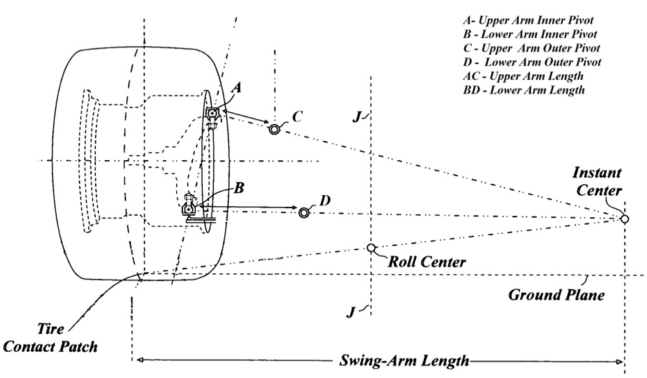

Fig. 2. SLA front view geometry.

- Then tie rod outer pivot is connected to IC keeping the outer pivot point in wheel centre plane to avoid Bump steer conditions, and then the tie rod length is established based on the rack position and length (Fig. 3).

5.2. Side view geometry

After construction, if front view geometry side view geometry is then constructed.

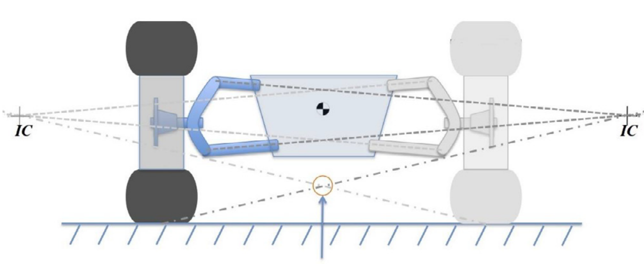

Fig. 3. Location of roll centre by connecting IC with ground contact patch.

- First, the anti-squat and anti-dive characteristics are obtained and hence angle ϕ is established.
- Then the side view swing arm length is obtained by line BB and the intersection point gives the side view IC.S

Front anti-dive is calculated as in the following Eqs. (3)–(4)

$$Front\ anti\ dive = \frac{\tan\phi f}{\left(\frac{h}{l}\right) \times (\%front\ braking)} \times 100 \quad (3)$$

$$Front\ anti\ lift = \frac{\tan\theta}{h/l} \times 100 \quad (4)$$

And thus have been put into equations to get IC.

6. Model and working of the proposed adaptive suspension system

To analyse the active camber model a simulation model is needed to simulate or measure various conditions and draw conclusions. In this case, a standard simulation model is not available. Therefore a model is developed. The model is modelled on SolidWorks (Fig. 4) and simulated on Matlab Simulink-SimMechanics. SimMechanics formulates and solves the equations of motion for the complete mechanical system. Models from CAD systems, including mass, inertia, joint, constraint, and 3D geometry, can be imported into SimMechanics. An automatically generated 3D animation allows visualization of system dynamics.

6.1. Model interpretation in SimMechanics

Two quarter car double wishbone suspensions are modelled in SolidWorks which is then imported into SimMechanics using the SimMechanics Link in SolidWorks. Using the 2nd generation SimMechanics-MATLAB joints and assemblies are employment to the imported models. The (Figs. 5, 6) below shows the various modules of the geometry that is being simulated. The schematic of the suspension system in Matlab is shown in Figs. 5 and 6 for passive and active suspension system respectively. Using various

Fig. 4. SOLIDWORKS model of the setup with variable length upper arm and tie rod.

joints that are available in the Simulink Library the connections of the models are defined. The rigid transforms present in every element of the model describes the location of the joints that are made within. The degrees of freedom to the connection is given as per its actual joining, this is facilitated by the functional blocks which are available in the Simulink library. Different kinds of joints are available based on the type of the motion i.e. rotational and translation which a component is going through with respect to the other component to which it is attached. These models are purely physical based compared to usual mathematical models that Simulink offers. The Simulink P-S converter block is used to convert Simulink signals into Physical signals. The input is given through this following block. The input given to the system is a sine wave of force which is given at the base of the cylinder–jack joint to simulate a vertical motion which will cause the system to undergo wheel travel. With the help of transform sensors, the values that have been sensed are read and are attached to the P-S Simulink converter block. This block helps in converting physical signals into Simulink signals. These signals are further directed into sinks called scope which help in viewing the output values of the system. The blocks indicate the components of the system like the tire, spindle, knuckle, tie rod, lower arm, upper arm, the test rig and the jack. In the case of the adaptive model actuators and sensors are added. The sensors used in passive model are just for the acquisition of data and they do not modify the system. The input to the system is a sine wave road profile having bumps and potholes which are of 150 mm. The camber and toe angles are sensed using transform sensors in both the model. The PID controllers used in the adaptive model receive input signals from the sensors, process the data and provide response signals to the actuators so that the camber and toe angles are brought to the desired values. The Proportional, Integral and derivative gain is tuned manually through Ziegler- Nichols method.

6.2. Working procedure of the adaptive control mechanism

The proposed active suspension system includes a PID controller, sensors, and linear actuators. The reason for choosing PID method is because of its simplicity and robust nature [26]. The PID system is economical, less complicated and easier to design and operate. Further, the system has been in use in the automobile industry for various purposes. The first sensor is located on the knuckle for measuring the camber angle and the second sensor is located in the upper arm fulcrum axis for sensing the toe angle. The sensors monitor the camber and toe angles and transmit the data continuously to the PID control system. Based on these real time sensor values, the PID system calculates the error i.e. the change in camber or toe angle that has occurred during that instant of time. To bring back the camber and toe angles to the specified initial values, the bi-directional actuator must be actuated. The actuators vary the position of upper A-arm and tie rod dynamically to control camber and toe angles respectively based on the control signal provided from the PID controller. The PID Controller calculates the length to which the linear actuators must be extended or contracted based on equations provided below in Eqs. (5) and (6) and also in (Fig. 7).

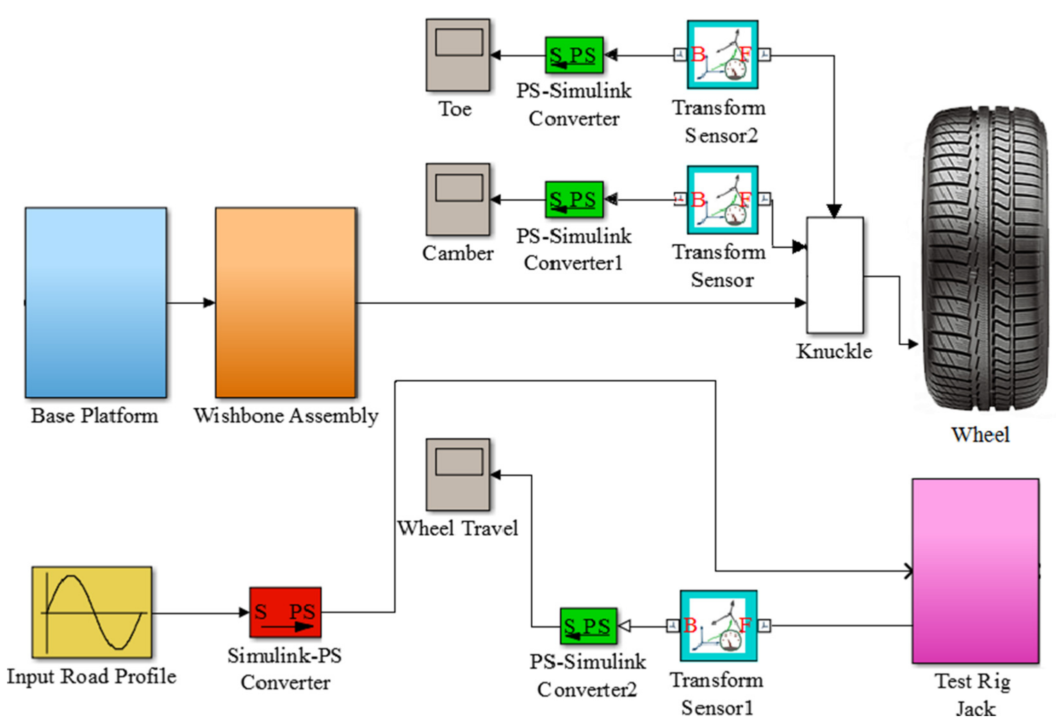

Fig. 5. Simulink block diagram of passive suspension system.

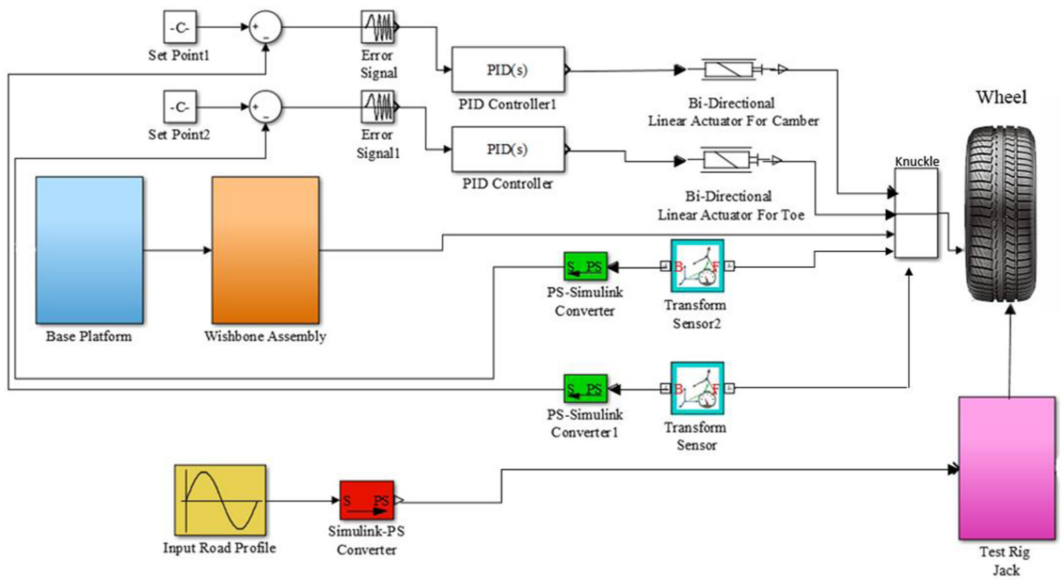

Fig. 6. Simulink block diagram of adaptive suspension system.

$$(U_{PID})_C = K_P(\theta c_{req} - \theta c_{actual}) + K_I \int_0^t (\theta_{Creq} - \theta_{Cactual})dt + K_D \frac{d}{dt}(\theta_{Creq} - \theta_{Cactual}) \tag{5}$$

$$(U_{PID})_T = K_P(\theta_{Treq} - \theta_{Tactual}) + K_I \int_0^t (\theta_{Treq} - \theta_{Tactual})dt + K_D \frac{d}{dt}(\theta_{Treq} - \theta_{Tactual}) \tag{6}$$

The final control system output voltage signals $(U_{PID})_C$ for camber given by Eq. (5) and $(U_{PID})_T$ for toe given by Eq. (6) are provided to the respective linear actuators for varying the length of upper A-arm and tie rod. The gain values in PID controller such as K_P, K_I and K_D are tuned through the Ziegler-Nichols method, in order to give better stability in the control system and have been fixed at 10, 5 and 5 respectively after numerous iterations. These values can be further tuned for better accuracy and efficiency.

7. Design validation

Inorder to verify the authenticity of the developed model, it must be validated with a standard one. This verification is also required in order to determine the trustworthiness of the results and also to identify the level of its accuracy. ADAMS CAR software has built in templates for various vehicle systems. As part of the package a standard suspension system is also available. The user can specify the system parameters like camber, caster, toe, steering

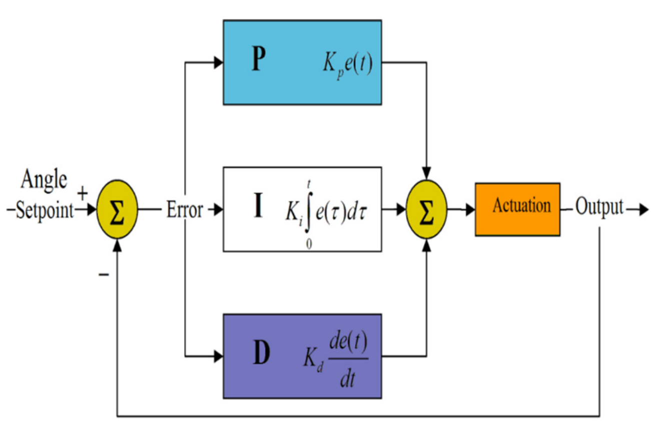

Fig. 7. Schematic of the PID control system.

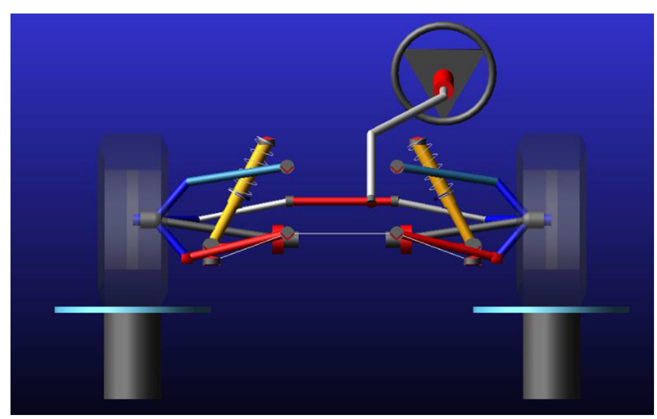

Fig. 8. SLA front view geometry model in ADAMS CAR.

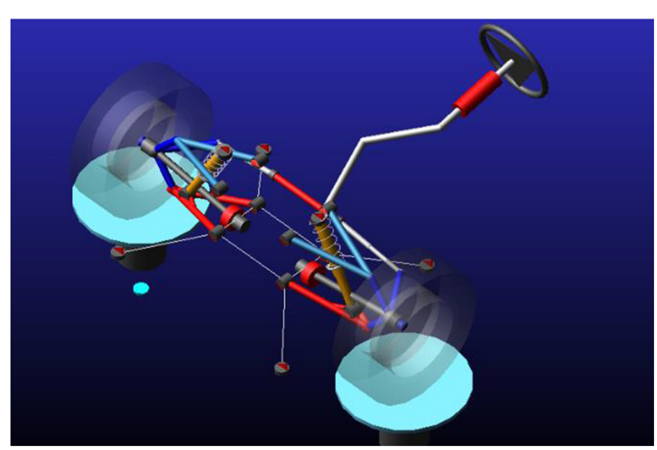

Fig. 9. SLA isometric view geometry model in ADAMS CAR.

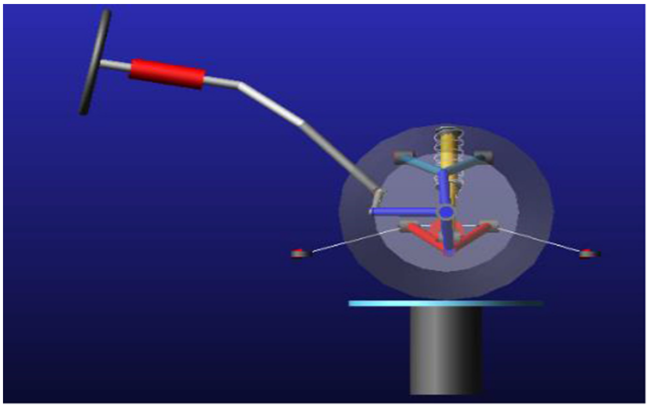

Fig. 10. SLA side view geometry model in ADAMS CAR.

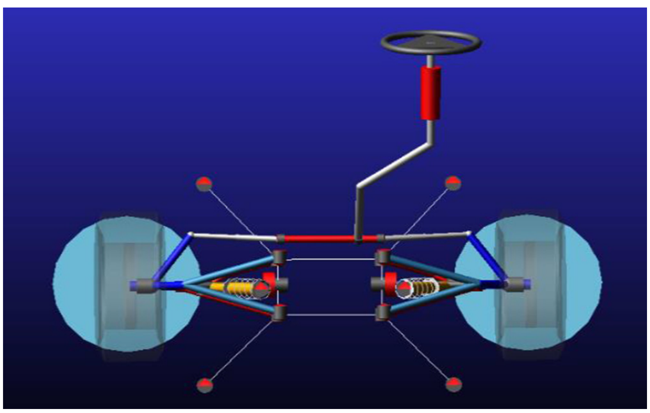

Fig. 11. SLA top view geometry model in ADAMS CAR.

axis inclination damping coefficient etc., to meet their requirements. It also provides features to test the suspension system through built in simulation techniques. The data provided by the software is validated and approved by most companies around the world. By utilizing the facilities of the software, a model of passive suspension system has been developed and simulated. By verifying and validating the accuracy level of the passive suspension system developed in Matlab-Simulink with the ADAMS suspension system, it can be confirmed that the models in Matlab-Simulink are accurate and they provide valid simulation results. A quarter car model of double wishbone passive system was built in ADAMS (Figs. 8–11) including the front steering geometry, using the same parameters used to develop the MATLAB model and is simulated for wheel travel from +120 mm to −120 mm and its characteristic curves such as camber and toe were plotted with respect to wheel travel. The results reveal the behaviour in which the camber and toe gain occurs and to the extent they occur.

The results from simulation of the suspension model are shown in (Figs. 12, 13) and they depict that for a wheel travel of +120 to −120 the camber varies from +2.6 degrees to −3.2 degrees and toe varies from −0.3 to 0.45 degrees. The difference in the upper and lower extreme values of camber results from the slight difference in upper and lower arm length and the toe change is quite low due to the outer tie rod point lying at the wheel centre plane and the unsymmetrical behaviour of toe with upward and lower wheel travel results due to the initial positioning of the arms at an angle with wheel centre maintaining almost zero camber at ride height. Since both the left and right wheels have the same geometry and wheel parameters and also the simulation test being identical for both the wheels, both the wheels behave in the same way. As a result, the graphs of both the left and right wheel are identical and they overlap each other for the entire range of operation.

The results of this validated data are chosen as a benchmark for the passive suspension model developed in Matlab. These results verify and validate the results of simulation of the passive model done on MATLAB Simulink (explained in further sections). Comparing both the data over the entire operating range, their data corresponds up to 90% accuracy with some minor deviations. Through this comparison, the passive model developed is validated and is used for comparison with active suspension model.

8. Simulation of the proposed adaptive system and results

In order to evaluate the performance of the active suspension model and to benchmark its performance with the existing passive suspension system, a simulation study has been performed on both the systems. A road profile with bumps and potholes is given to the system using Simulink to simulate wheel travel and analyse the camber and toe characteristics of both the systems. In this work, the ideal camber and toe angles for best performance are assumed to be 0 degrees (mentioned as deg in short-form hereafter).

8.1. System input characteristics

In the following study, the input that was given to the system is a sine road profile with an amplitude of 150 mm and the time

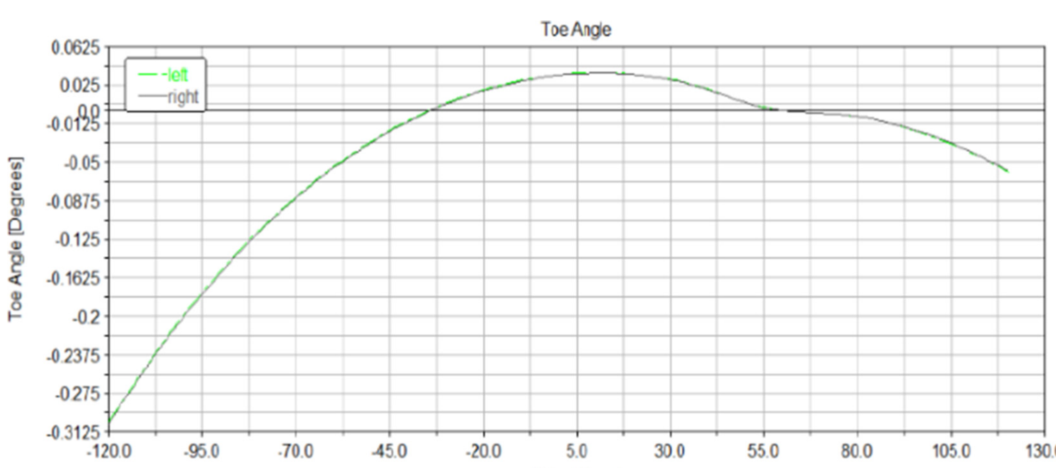

Fig. 12. Toe vs wheel travel of passive system in ADAMS CAR.

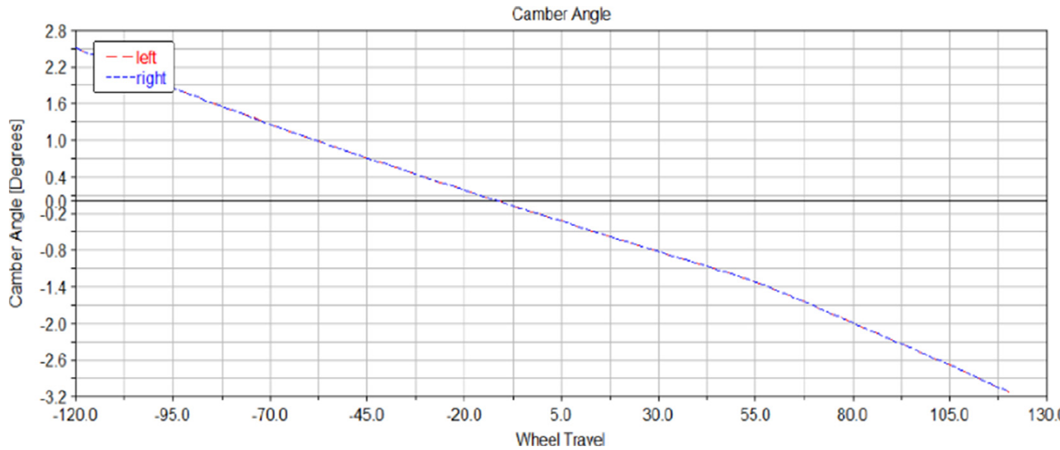

Fig. 13. Camber vs wheel travel of passive system in ADAMS CAR.

period 1 s. This is given to the system through a jack that is attached to the test rig, over which the tire rests. The simulation period is 10 s (Fig. 14).

8.2. System output characteristics

The output data is recorded through attaching sensors that are available in the Simulink library. The data that is mainly recorded through these sensors are camber, toe and wheel travel and the data of passive and adaptive systems are put to evaluation.

As inferred from the above graphs, (Figs. 15, 16) the camber and toe characteristics varying with time. The camber characteristic varies in a sinusoidal fashion in response to the sinusoidal input at the wheel through the hydraulic lift. The toe variations which have been observed are asymmetric in nature due to the position of the arms and their length. This often seen in off road vehicles. The camber angle obtained is 2.5 deg. The toe angle obtained is -0.6 deg.

Similarly, the graphs shown in Figs. 17 and 18 present the camber and toe characteristics of adaptive system. While comparing the adaptive and passive systems, a substantial reduction of the camber and toe angle can be interpreted (Figs. 15–18). The adaptive suspension system successfully manages to reduce the camber angle to 1.151 deg from 2.8 deg and toe to 0.023 deg from 0.6 deg.

Comparing the camber characteristics (Figs. 19, 20) of both the passive and adaptive system, it can be observed that there is a reduction in the in the camber angle over the whole span of wheel travel. At some points in the graph, two or more values of camber can be observed for given wheel travel. These vibrations are caused due to the combined effect of inertial imbalance about the wheel centre plane and due to counter forces produced by the wheel when it is travelling in different directions.

In case of the toe characteristics (Figs. 21, 22) a vertical shift has been observed due to the same inertial effects which were

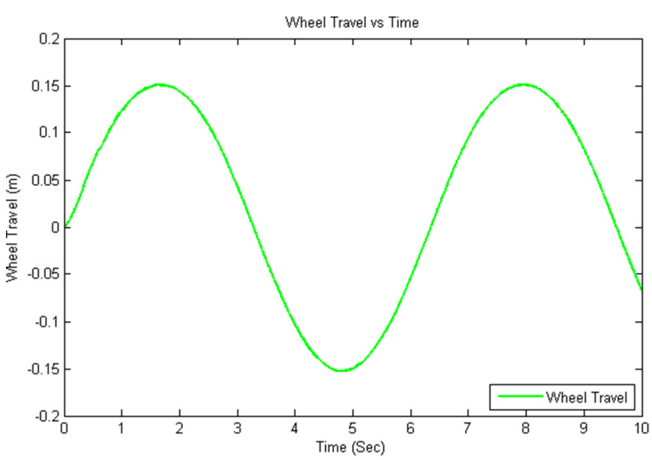

Fig. 14. Graph of wheel travel vs time as input road profile in MATLAB.

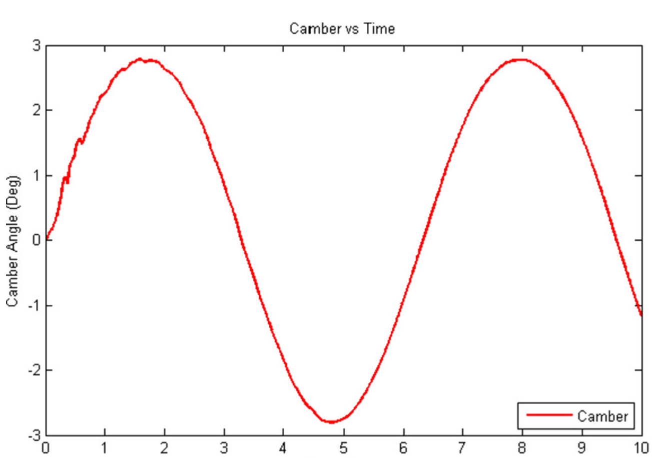

Fig. 15. Graph of camber vs time of passive system.

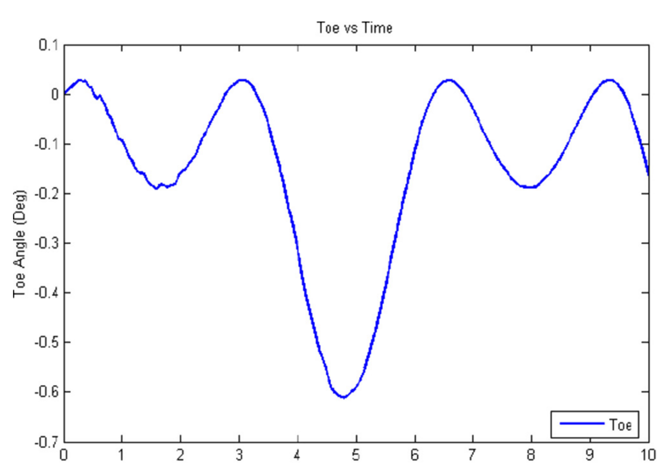

Fig. 16. Graph of toe vs time of passive system.

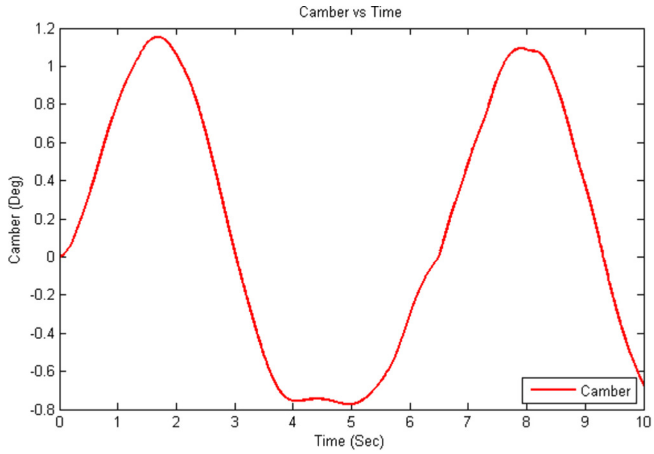

Fig. 17. Graph of camber vs time of adaptive system.

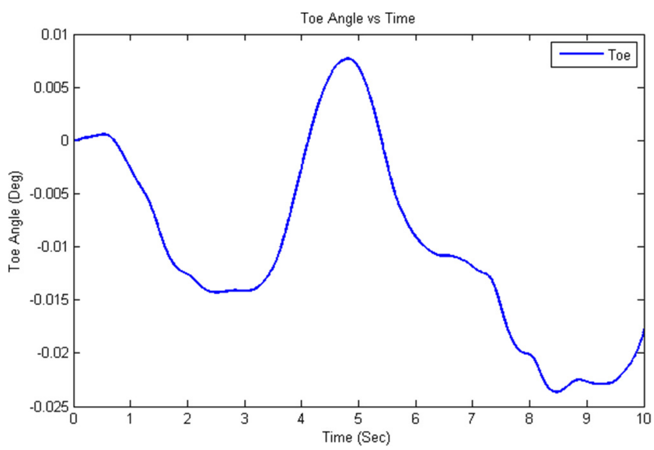

Fig. 18. Graph of toe vs time of adaptive system.

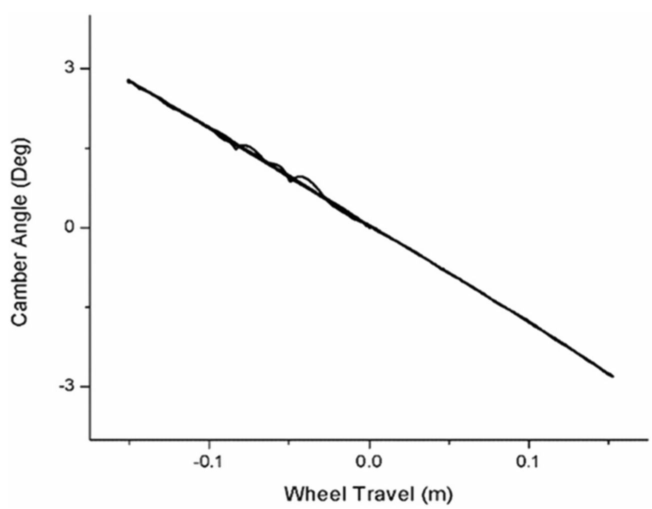

Fig. 19. Graph of camber vs wheel travel of passive system.

explained above. Data obtained from the sensors are directly taken from MATLAB and the minimum and maximum deflection in each case are recorded and shown in Table 2. Despite the vibration, an improvement of 58% in camber characteristics and a 96% improvement of Toe characteristics has been attained.

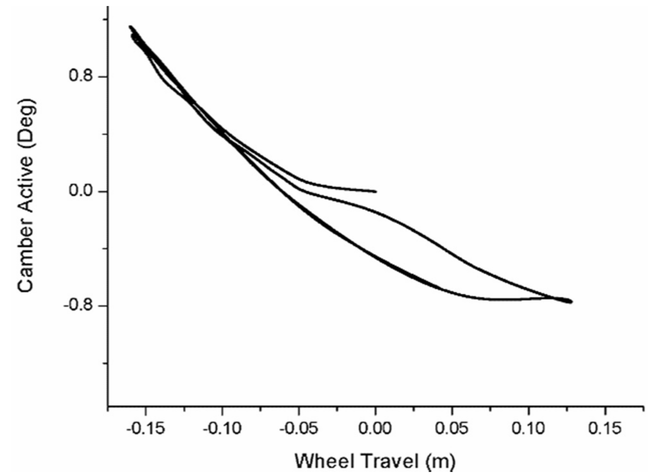

Fig. 20. Graph of camber vs wheel travel of adaptive system.

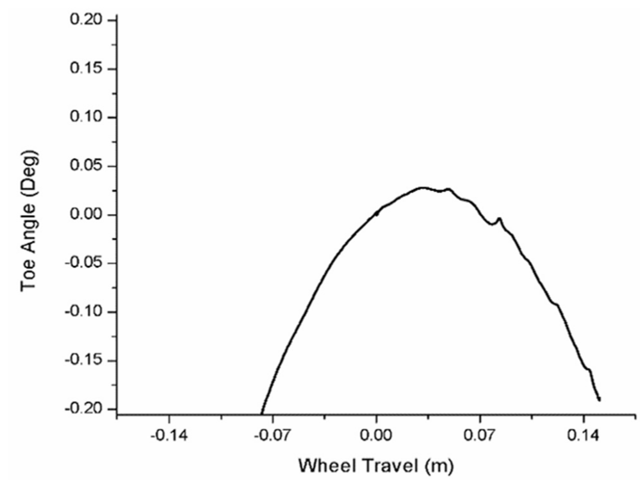

Fig. 21. Graph of toe vs wheel travel of passive system.

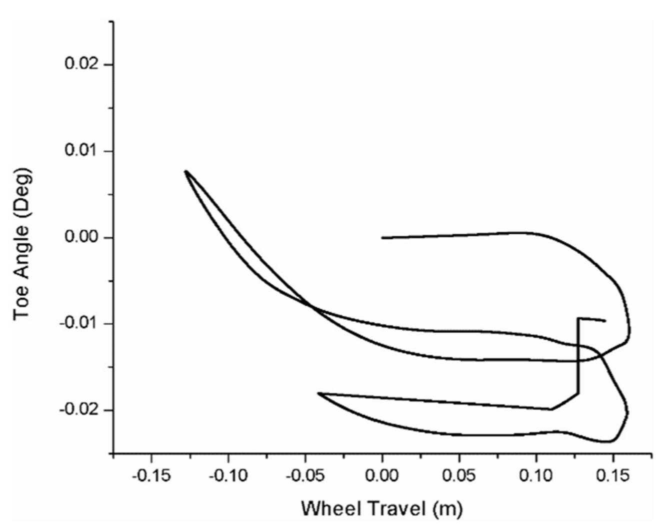

Fig. 22. Graph of toe vs wheel travel of adaptive system.

Table 2
Result comparison table.

Parameter (Deflection)	Passive (Deg)	Adaptive (Deg)	Reduction (%)
Camber Minimum	0	0	–
Camber Maximum	2.806	1.153	58.91
Toe Minimum	0	0	–
Toe Maximum	0.61	0.023	96.22

9. Conclusion

In the present work, an adaptive suspension system is developed to adjust camber and toe angles in real-time conditions to improve manoeuvrability and wheel traction. A passive suspension system model, as well as an active suspension model with linear actuators, were developed in SolidWorks and imported to Matlab. In the Matlab environment, sensors were added to both the systems while a PID control has been developed and tuned for an active system. A standard suspension model has been developed in ADAMS car and the results have been used for validating the performance parameters of the passive model developed in Matlab Simulink. The results from ADAMS Car and the Simulink passive suspension system corresponds to 90% accuracy, confirming the acceptability and correctness of Simulink study. The simulation results of the active suspension system have been compared with passive suspension system throughout the study. The performance of each system has been presented in the form of graph and the results of the comparison are provided in the form of a table. The proposed adaptive suspension system reduced the change in camber characteristics from 2.8 deg to 1.151 deg and also managed to reduce the change in toe angle from 0.6 to 0.023 deg. Overall, the simulation comparison of active and passive models show a 56% reduction in camber angle and 96% reduction in toe variation for corresponding wheel travels.

One of the advantages of this method is the robust and adaptable nature of the suspension setup at all running conditions in an economic way. This system would also help in increasing the wheel travel without compromising camber and toe angle characteristics. This can possibly change the face of vehicle dynamics. Further, this idea can also be incorporated into a four wheel steering and also a variable ratio power steering without any change in the design of the system. As mentioned, the system has some limitations due to vibrations caused by counteracting forces and external noise reception in sensors. In the future, research will be focused mostly on refining the method with nonlinearity and also to improve the smoothness in the response of the system. The focus will also be put on improving and optimising the frequency response of the system. At the same time, attempts will be made to develop the idea into a prototype.

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