DESIGN, DEVELOPMENT AND ANALYSIS BY BONDGRAPH SIMULATION TECHNIQUE FOR COMFORT RIDE ON VEHICLES

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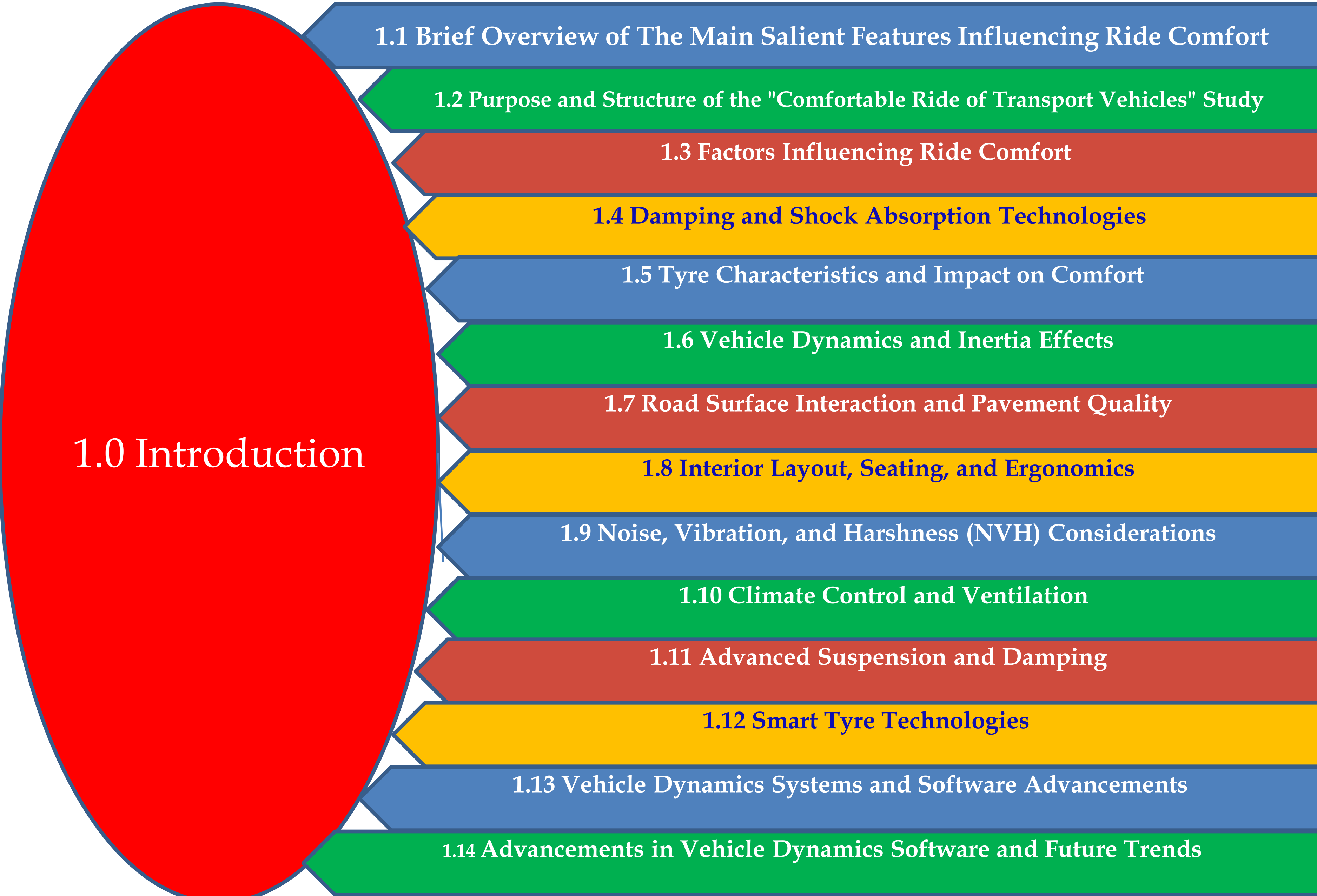
OBJECTIVES

Development of Vehicles for Rural and District Roads in India

To analyze the distinctions between rural and urban road conditions and develop vehicles optimized for rural roads. This research aims to *enhance operational efficiency, vehicle lifespan,* and *passenger comfort*, addressing *economic factors specific to India's extensive (more than 50%)* ***rural and district roads*** *against total 62,15,797 km networks*.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1.1 Brief Overview of The Main Salient Features Influencing Ride Comfort
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1.10 Climate Control and Ventilation
1.11 Advanced Suspension and Damping
1.12 Smart Tyre Technologies
1.13 Vehicle Dynamics Systems and Software Advancements
1.14 Advancements in Vehicle Dynamics Software and Future Trends

1. INTRODUCTION

Passenger comfort in modern transportation is crucial, impacting safety, well-being, and satisfaction. Essential for public transit adoption, it reduces congestion and emissions. India’s vast rural road network underscores infrastructure challenges. Thus it is needed to focus on optimizing ride comfort through advanced modeling, particularly the Bond Graph method, to address diverse terrains and enhance the overall travel experience.

Contd…1. INTRODUCTION

Moreover, in a global context where sustainability and environmental consciousness are paramount, ride comfort has gained prominence as a critical factor in the adoption of public transit and shared mobility solutions.

A comfortable ride can super way individuals towards choosing collective modes of transport over private vehicles, contributing to reduced congestion, lower emissions, and improved overall urban livability.

India possesses the **world's *second-largest* road network, spanning *62,15,797 km***, trailing only the **United States** with ***68,53,024 km*** as on March 2023.

Contd…**1. INTRODUCTION**

The country has hardly constructed national highways, expressways, state highways, contributing to a total length of ***3,15,349 km. remarkably; more than 50% of this extensive road network is situated in rural areas***, consisting of surfaces not fortified with cement pavement or bituminous materials of requisite strength.

This research focuses on optimizing ride comfort through advanced modeling, particularly the Bond Graph method, as a cornerstone of safe, sustainable transport, comfort addresses diverse passenger needs especially on rural roads.

1.1 Key Features Influencing Ride Comfort

A vehicle's suspension system is vital for performance, safety, and comfort. It manages tyre-road interaction, absorbs shocks, and protects the structure while balancing stiffness and compliance for optimal comfort and stability under various conditions.

1.2 Purpose of the Study

The study examines factors affecting passenger comfort across transport modes, focusing on design, materials, and technology. It aims to provide insights to enhance vehicle design, engineering, and passenger satisfaction.

1.3 Factors Affecting Ride Comfort

Suspension systems play a crucial role in ensuring comfort, stability, and handling, which are essential to ride quality.

1.4 Damping and Shock Absorption Technologies

Damping and shock absorption are essential in suspension systems, controlling wheel oscillations on uneven surfaces. These technologies ensure ride comfort, enhance stability, and improve vehicle handling.

1.5 Tyre Characteristics and Impact on Comfort

Tyre attributes significantly influence comfort, absorbing shocks, reducing noise, and providing a smooth ride. Proper tyre selection enhances overall ride quality.

1.6 Vehicle Dynamics and Inertia Effects

Vehicle dynamics analyze motion under forces like acceleration and braking. Inertia impacts vehicle behavior, influencing stability and response during various driving conditions.

1.7 Road Surface Interaction and Pavement Quality

Road surface conditions greatly impact vehicle dynamics, ride comfort, and operational efficiency. High-quality pavements ensure better handling and smoother rides.

1.8 Interior Layout, Seating, and Ergonomics

Vehicle interiors prioritize comfort and safety through ergonomic seating, intuitive controls, and user-friendly layouts, enhancing convenience for passengers.

1.9 Noise, Vibration, and Harshness (NVH) Considerations

Addressing NVH ensures quieter, smoother rides, improving passenger comfort and overall driving experience.

1.10 Climate Control and Ventilation

Efficient climate control and ventilation systems enhance passenger comfort while promoting sustainability and minimizing environmental impact.

1.11 Advanced Suspension and Damping

Dynamic suspension and active damping technologies improve comfort, handling, and performance, showcasing innovation in automotive engineering.

1.12 Smart Tyre Technologies

Smart tyres and adaptive pressure systems boost safety, efficiency, and ride comfort by optimizing tyre pressure management.

1.13 Vehicle Dynamics Systems and Software Advancements

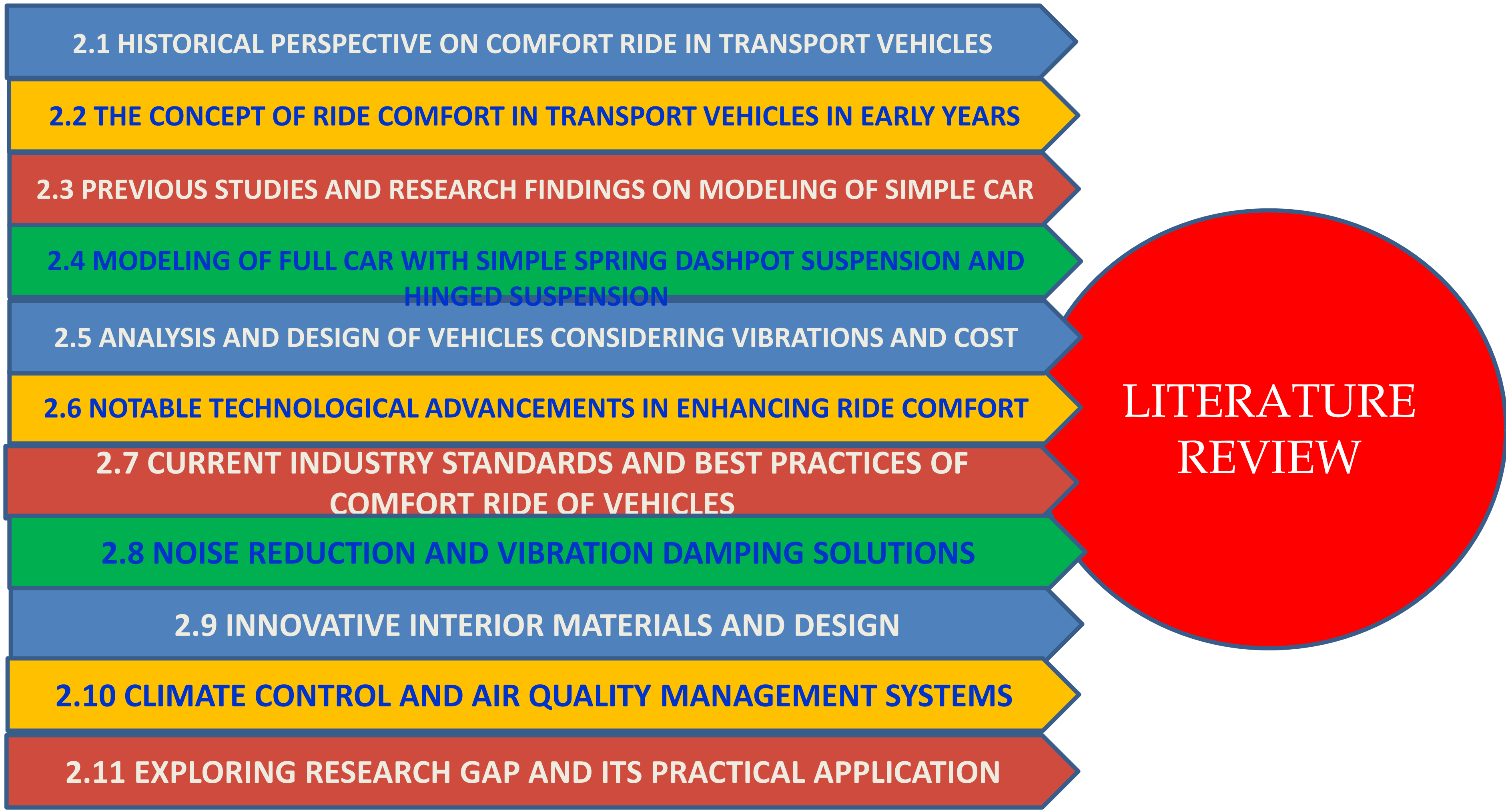
Vehicle dynamics control systems enhance safety and comfort by improving brake, steering, and suspension technologies. Advanced vehicle dynamics software, such as Dymola and Modelica, aids in design and performance analysis.

Real-time simulations support ADAS and autonomous vehicle development, enabling engineers to predict behavior, refine designs, and ensure rapid, safety-critical responses under various conditions.

1.14 Advancements in Vehicle Dynamics Software and Future Trends

This research emphasizes using Symbol Shakti and MATLAB to enhance vehicle dynamics understanding and engineering innovation. A suspension damping coefficient ≤8 kNs/m minimizes sprung-mass displacement while enabling speeds of 50–75 km/h, optimizing performance on rural roads.

This balance supports cost-effective, eco-friendly vehicles tailored to India’s rural sector. Addressing unique road conditions, the study advances sustainable transportation, aligning with economic and environmental goals.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2.2 THE CONCEPT OF RIDE COMFORT IN TRANSPORT VEHICLES IN EARLY YEARS
2.3 PREVIOUS STUDIES AND RESEARCH FINDINGS ON MODELING OF SIMPLE CAR
2.4 MODELING OF FULL CAR WITH SIMPLE SPRING DASHPOT SUSPENSION AND HINGED SUSPENSION
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2.8 NOISE REDUCTION AND VIBRATION DAMPING SOLUTIONS
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2.10 CLIMATE CONTROL AND AIR QUALITY MANAGEMENT SYSTEMS
2.11 EXPLORING RESEARCH GAP AND ITS PRACTICAL APPLICATION
LITERATURE REVIEW

2.0 LITERATURE REVIEW

2.1 Historical Perspective on Comfort Ride In Transport Vehicles

There has always been trade, human interaction, and transport for almost no society has ever been purely subsistence in character. Expanded trade and the exertion of political power have asked for more, and there have been surges of transport improvements associated with the expansion of empires. Overland and river routes served the trade of Mesopotamia five millennia ago. Roman roads supported Roman hegemony as roads did for Persian, Chinese, and New World rulers.

2.0 LITERATURE REVIEW

2.2 The Concept of Ride Comfort In Transport Vehicles In Early Years:

It has evolved significantly over the years, influenced by technological advancements, societal changes, and a growing emphasis on passenger well-being. Here is a brief historical perspective on the evolution of ride comfort.

2.3 Previous Studies and Research Findings on Modeling of Simple Car:

Andradottir et al. (1997) offer a comprehensive exposition on simulation modeling and analysis, addressing numerous pivotal inquiries. These include elucidations on the nature of modeling, simulation, and the amalgamation of both in simulation modeling and analysis.

Contd…2.3 Previous Studies and Research Findings on Modeling of Simple Car

Tseng and Ashrafi (1999) laid the foundation for addressing practical challenges in enhancing technology within vehicle stability control systems. They conducted a thorough examination of diverse approaches to designing and developing automotive systems.

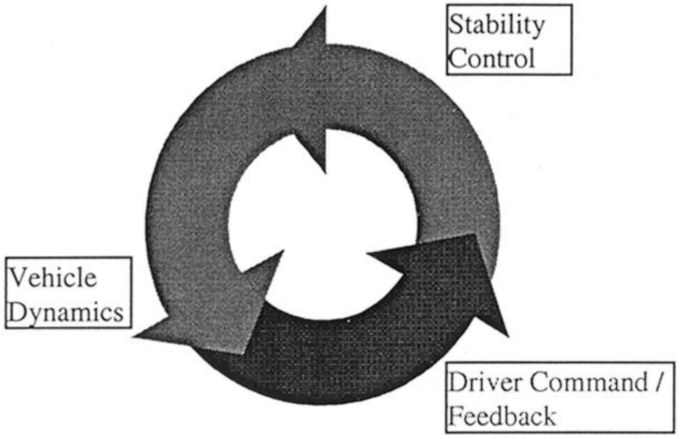


Fig. 2.1 Driver/system/vehicle interaction

Contd…2.3 Previous Studies and Research Findings on Modeling of Simple Car

Louca et al. (2000) emphasized the importance of establishing a comprehensive model that accurately encompasses the key components of an automobile: the engine, drive train, and vehicle drive systems.

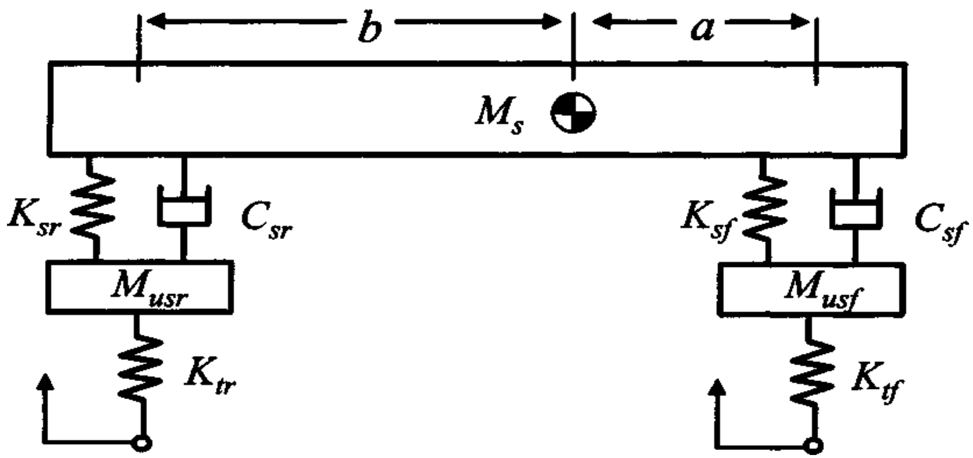


Fig.2.2 Half-car model

Contd…2.3 Previous Studies and Research Findings on Modeling of Simple Car

Kim et al. (2003) established that target cascading in product development is a systematic approach aimed at systematically transmitting the desired top-level system design goals to appropriate specifications for subsystems and components, done in a coherent and efficient manner.

Kim et al. (2003) outline a comprehensive approach involving hydraulic system design and vehicle dynamic modeling for the advancement of tyre roller traction a crucial aspect in the analysis of tyre roller systems.

Contd…**2.3 Previous Studies and Research Findings on Modeling of Simple Car**

Maxim and Nguyen (2006) explored automobile suspension modeling, focusing on vehicle ride quality over uneven terrain and body motion control—key considerations for automotive and vibration engineers.

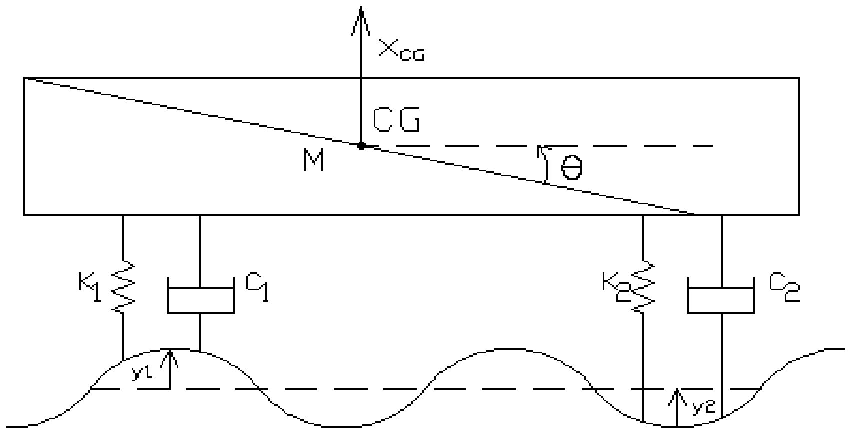


Fig. 2.3 Spring-mass-damper model

Contd…2.3 Previous Studies and Research Findings on Modeling of Simple Car

Wakeham and Rideout (2011) examine the optimal model complexity for designing vehicle active suspension controllers using the Linear Quadratic Regulator (LQR) method. Their approach balances suspension travel, ride quality, and road holding through a weighted performance index.

Mitra A. C. and Banerjee Nilotpal (2013) highlight the complexities of suspension system design, driven by numerous control parameters, diverse objectives, and stochastic disturbances.

2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension

Glass (2001) conducted an experimental assessment of the Volvo Optimized Air Suspension-2 (VOAS-2), a prototype trailing-arm suspension for heavy trucks Fig. 2.4.

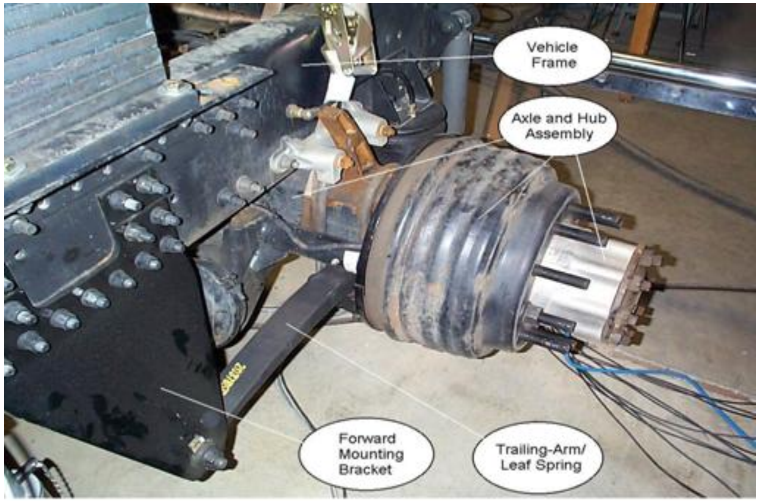


Fig. 2.4 VOAS-2 Suspension on test vehicle

Contd. 2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension

Rideout et al. (2007) propose a method to systematically analyze and quantify decoupling in dynamic system models and segment them accordingly.

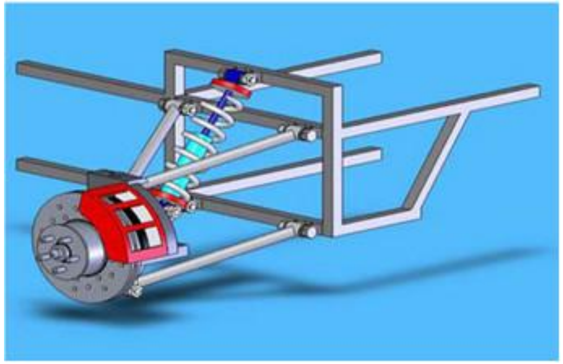


Fig. 2.5 Vehicle systems models

Granda (2008) established core theoretical principles of vehicle dynamics and design, integrating computer-aided techniques for constructing and analyzing vehicle dynamics and mechatronics systems as shown in Fig. 2.5.

Contd. **2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension**

Zoroofi (2008) highlighted the need for alternative energy sources in transportation due to fossil fuel limitations, advocating electric and hybrid vehicles as viable solutions.

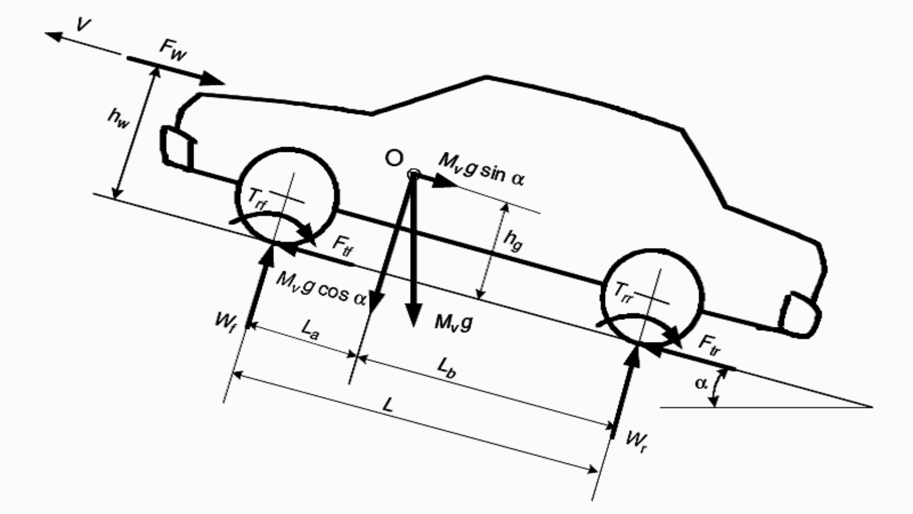


Fig. 2.6 Vehicle longitudinal forces

Contd. **2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension**

Silva et al. (2008) used the model-based Analytical Reduction Relationships (ARR) technique with *Diagnostic Bond Graphs* to detect and isolate faults in vehicle suspensions.

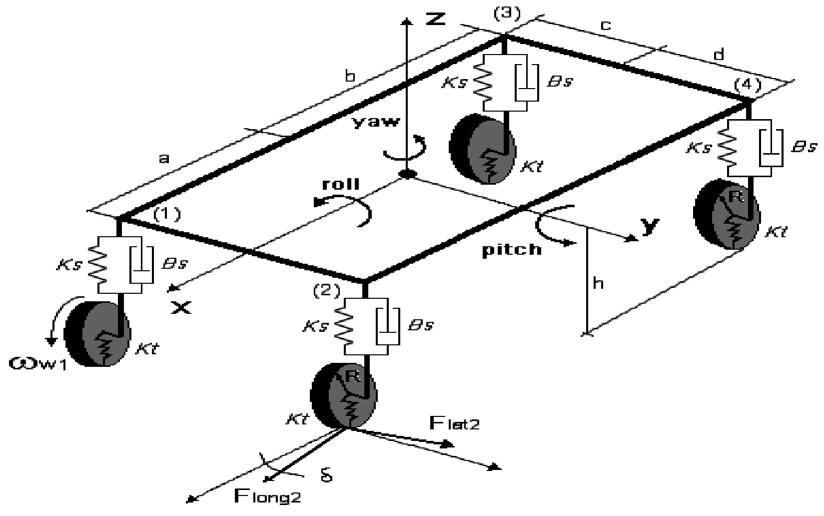


Fig. 2.7 Full car suspension model

Contd. 2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension

Adibi and Rideout (2009) showcased active suspension systems' effectiveness in improving comfort using bond graph modeling.

Milner et al. (2009) developed a validated six-degree-of-freedom model for autonomous vehicle design.

Gauchia and Sanz (2010) highlighted risks of fossil fuel dependence, emphasizing challenges from depleting reserves, uncertain resources, and geopolitical and economic issues in distribution.

Contd. 2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension

Creed et al. (2010) developed a detailed car model, equipped with suspension force actuators, to support future advancements in active suspension control systems for improving ride quality in standard road vehicles as shown in Fig. 2.8.

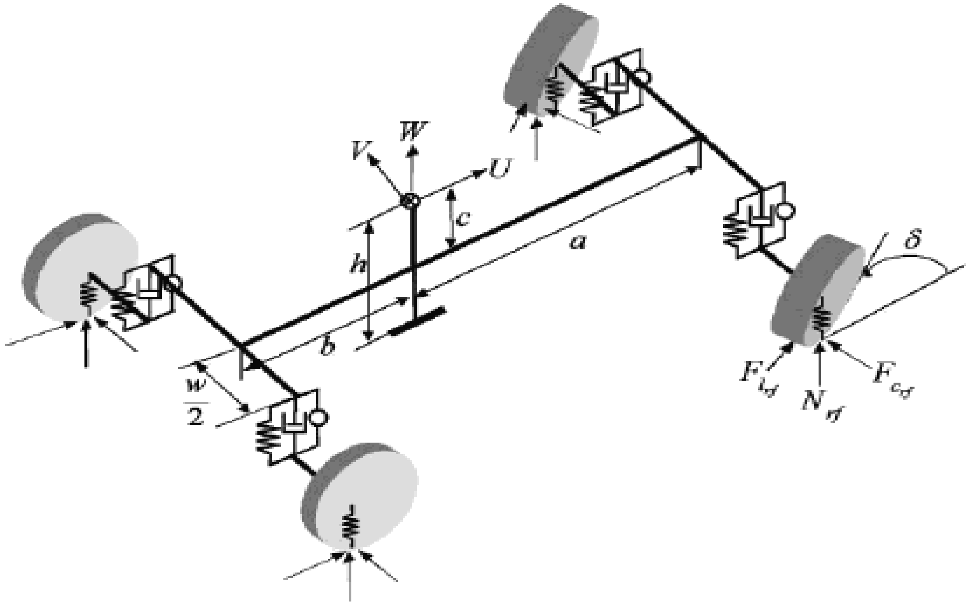


Fig. 2.8 Full car model

Contd. 2.4 Modeling of Full Car With Simple Spring Dashpot Suspension and Hinged Suspension

Lyons et al. (2011) emphasized hands-on lab work for mastering mechanical systems.

Phalke and Mitra (2016) highlighted suspension systems' role in vehicle comfort by dampening vibrations.

Dridi et al. (2017) introduced Bond Graph modeling for suspension systems using TPMLSA technology.

Al Rawashdeh et al. (2019) tackled robust active suspension design with uncertain parameters.

Dacova et al. (2021) explored vibration and noise impacts on health and comfort.

2.5 Analysis and Design of Vehicles Considering Vibrations and Cost

The bond graph concept, introduced by Paynter (1961), marked a pivotal advancement in physical system modeling by treating objects as interconnected elements via 0 and 1 junctions.

Karnop et al. (1975) expanded this approach using Kirchhoff's bond graphs for electrical networks, while Mukherjee et al. (2000) introduced Symbols2000 software for modeling and simulation, enhancing design efficiency.

Loukas et al. (2001) applied bond graphs to vehicle subsystems, emphasizing energy-based modeling for performance improvement. Filippini et al. (2005) used bond graphs to evaluate dynamic four-wheel models, integrating components like chassis and tires via 20sim software.

Contd.. 2.5 Analysis and Design of Vehicles Considering Vibrations and Cost

Further applications, such as **Brendan J. Chan et al.'s (2003)** MATLAB simulations, analyzed ride control through acceleration and displacement data.

Comfort enhancements have been a significant focus, with **Hong et al. (2003)** optimizing pressure distribution using air cells, and **Gao et al. (2011)** utilizing frequency-domain analysis for ride comfort assessment.

Avesh and Srivastava (2012) proposed active suspension systems to reduce vibrations and improve stability, while **Hamed et al. (2018)** developed a seven-degree-of-freedom model addressing durability on uneven rural roads.

Contd.. 2.5 Analysis and Design of Vehicles Considering Vibrations and Cost

Recent studies introduced innovative methods like **Sihem Dridi et al.'s (2017)** TPMLSA actuator for minimizing wheel vibrations and **Cheng Cheng et al.'s (2019)** Active Variable Geometry Suspension for improved ride comfort and road holding. MATLAB/Simulink and fuzzy PID controllers continue to facilitate precise modeling and simulation.

These approaches underscore the importance of cost-effective designs that balance ride comfort, vehicle stability, and durability while addressing energy conservation and reduced carbon emissions, offering significant economic and environmental benefits.

2.6 Notable Technological Advancements In Enhancing Ride Comfort

Technological advancements have significantly improved public / commercial vehicles, enhancing their performance, comfort, and safety. Key innovations include the shift from manual to power steering, diesel to electric engines, and conventional to ABS braking systems. Moreover, autonomous vehicles have become a promising future technology, though challenges remain.

In **2006, Shunichi** emphasized the importance of driving support systems that account for human behavior, aiming for driver-vehicle symbiosis. By analyzing driver behavior, these systems could reduce driving workload and improve safety.

Contd.. 2.6 Notable Technological Advancements In Enhancing Ride Comfort

Chen Zhengke (2015) focused on nonlinear suspension systems, which are vital for ride comfort and vehicle stability. His study highlighted the importance of suspension system design in balancing comfort and handling stability. He developed a model for vehicle suspension with two degrees of freedom, using statistical linearization for better ride comfort prediction.

In 2017, Adam A. and Sakdirat K. explored using smartphones to assess ride comfort in trains, allowing passengers to provide real-time feedback on vibrations. This study introduced artificial neural networks to process smartphone data, showing their effectiveness in evaluating ride quality.

Contd.. **2.6 Notable Technological Advancements In Enhancing Ride Comfort**

Paliwal V. et al. (2021) optimized suspension parameters in a quarter car model using a genetic algorithm. Their research showed that higher damping and lower spring stiffness improve passenger comfort while balancing suspension performance.

Lastly, **Ferhath A.A. and Kamalakkannan K. (2023)** reviewed control strategies in vehicle suspension systems, highlighting their role in improving safety and comfort. They identified a gap in understanding the various control algorithms, signaling room for further research in suspension system advancements.

2.7 Current Industry Standards and Best Practices of Comfort Ride of Vehicles

Best practices ensure efficiency and high standards across industries. In automotive design, they focus on enhancing ride comfort by addressing temperature, air quality, noise, vibration, light, and ergonomics.

In 2002, ***Manuel Carlos Gameiro da Silva*** underscored the importance of measuring ride comfort due to rising mobility and consumer expectations. His study introduced comfort indices and highlighted the role of physical factors like temperature and vibration, using mannequins to simulate human functions for better comfort evaluation.

Contd.. **2.7 Current Industry Standards and Best Practices of Comfort Ride of Vehicles**

In 2005, ***Schalk Els*** examined ride comfort in military off-road vehicles, comparing objective methods like ISO 2631 and VDI 2057 with driver feedback. He found all methods effective but noted variations in acceptable limits, with vertical acceleration being the key factor.

In 2011, ***Aihua Tang and colleagues*** addressed inconsistencies in ride comfort standards across vehicles and terrains. They proposed two systems: ISO 2631 for regular vehicles and a tailored approach for military vehicles using parameters like vertical acceleration and absorbed power, improving standardization across different conditions.

2.8 Noise Reduction and Vibration Damping Solutions

Efforts to reduce vehicle weight have been successful in optimizing fuel efficiency without compromising performance. However, the weight reduction often leads to increased noise and vibrations due to changes in the vehicle's structure.

This poses a challenge for manufacturers to strike a balance between weight reduction and maintaining acceptable noise and vibration levels.

2.9 Innovative Interior Materials and Design

It is therefore observed that the evolution of automotive interior design reflects a dynamic interplay between consumer expectations, technological advancements, and environmental consciousness.

The integration of advanced materials, smart surfaces, and sustainable options demonstrates the industry's commitment to delivering interiors that not only meet but exceed the evolving expectations of consumers.

2.10 Climate Control and Air Quality Management Systems

Indoor air quality is vital for health, as indoor pollution can be more harmful than outdoor pollution. Effective climate control is crucial in vehicles where people spend significant time. This section discusses maintaining optimal conditions and introduces a system using an Arduino microcontroller and fuzzy logic to improve climate control.

Poor ventilation, material emissions, and airborne pollutants can cause respiratory issues and allergies, highlighting the need for efficient systems to ensure clean air and healthy conditions in vehicles.

2.11 Exploring Research Gap and Its Practical Application

The literature review highlights the development of advanced sub-models, known as Capsules, designed to reduce road-induced vibrations such as bumps and uneven surfaces. These Capsules are automatically generated from system equations, minimizing vibrations effectively.

They can resolve differential causalities and algebraic loops using a robust symbolic solution engine, generating high-level C code and integrating external code. Simulations are typically carried out using tools like Bond graph, Simulink, and MATLAB.

Contd..

2.11 Exploring Research Gap and Its Practical Application

The review studies fail to address differences between rural and urban road conditions, especially the distinct impacts of high-speed travel on rural roads, including costs, vehicle lifespan, and comfort.

This ***highlights a critical research gap*** in designing vehicles specifically for India’s rural roads. With the world’s second-largest (62,15,797 kms) road network, India requires tailored solutions that consider economic factors, durability, and passenger comfort, ensuring vehicles are optimized for the unique challenges and conditions of rural transportation.

3. METHODOLOGY

3.1 PROCESS OF VEHICLE RIDE COMFORT

3.2 OBJECTIVE OF VEHICLE RIDE COMFORT

3.3 ASSUMPTIONS FOR VEHICLE RIDE COMFORT

3.4 Symbol Shakti Software Tool For Modeling and Simulation

3.5 Bond Graph Tools for Dynamic Systems

3.6 Concept of Bond Graph Elements

3.7 Development of The Bond Graph Model

3. METHODOLOGY

3.1 Process of Vehicle Ride Comfort

Vehicle ride comfort is a critical aspect of automotive design, directly influencing passenger satisfaction and vehicle performance. Achieving optimal ride comfort involves a systematic process that encompasses vehicle modeling, simulation, testing, and iterative refinement.

This process ensures that vehicles can effectively absorb road irregularities, minimize vibrations, and provide a smooth driving experience.

3.2 Objective of Vehicle Ride Comfort

Vehicle ride comfort aims to enhance passenger experience and vehicle performance by reducing discomfort from road irregularities, vibrations, and disturbances while maintaining safety and handling.

3.3 Assumptions for Vehicle Ride Comfort

Ride comfort design relies on assumptions about the vehicle, environment, and human perception. These provide a testing framework and may evolve with advancements in materials, suspension technologies, and comfort standards.

3.4 Symbol Shakti Software Tool For Modeling and Simulation

Symbol Shakti is a robust software tool for modeling, simulation, and control analysis. It excels in symbolic and numeric computation, making it invaluable for research and industrial modeling of complex systems.

3.5 Bond Graph Tools for Dynamic Systems

Bond graphs are graphical representations of dynamic systems, showing bi-directional energy exchange, unlike block diagrams and signal-flow graphs, which depict uni-directional information flow.

3.0 METHODOLOGY

Table 3.1 Efforts and Flow Variables

Systems	Effort (e)	Flow (f)
Mechanical	Force (F)	Velocity (v)
	Torque (τ)	Angular velocity (ω)
Electrical	Voltage (V)	Current (i)
Hydraulic	Pressure (P)	Volume flow rate (dQ/dt)
Thermal	Temperature (T)	Entropy change rate (ds/dt)
	Pressure (P)	Volume change rate (dV/dt)
Chemical	Chemical potential (μ)	Mole flow rate (dN/dt)
	Enthalpy (h)	Mass flow rate (dm/dt)
Magnetic	Magneto-motive force (e_m)	Magnetic flux (φ)

3.6 Concept of Bond Graph Elements

Source of Effort (SE)
Source of Flow (SF)
The Inertia Element (I)
The Complaint Element (C)
The Resistive Element (R)
Transformer (TF)

Contd.. **Concept of Bond Graph Elements**

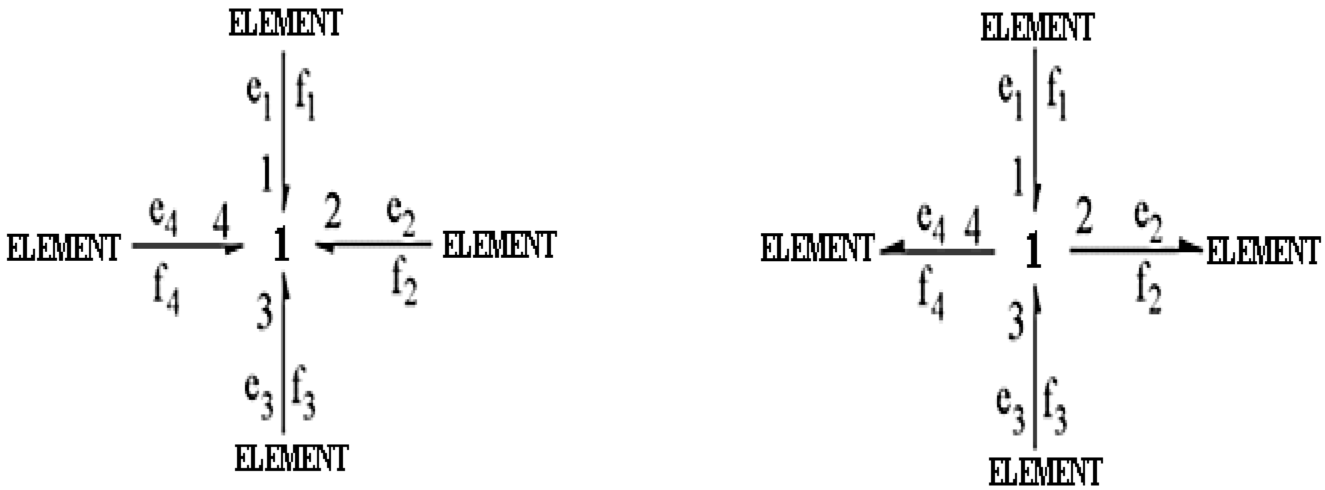


Fig.3.1 '1' Junction Element Models

Contd.. **Concept of Bond Graph Elements**

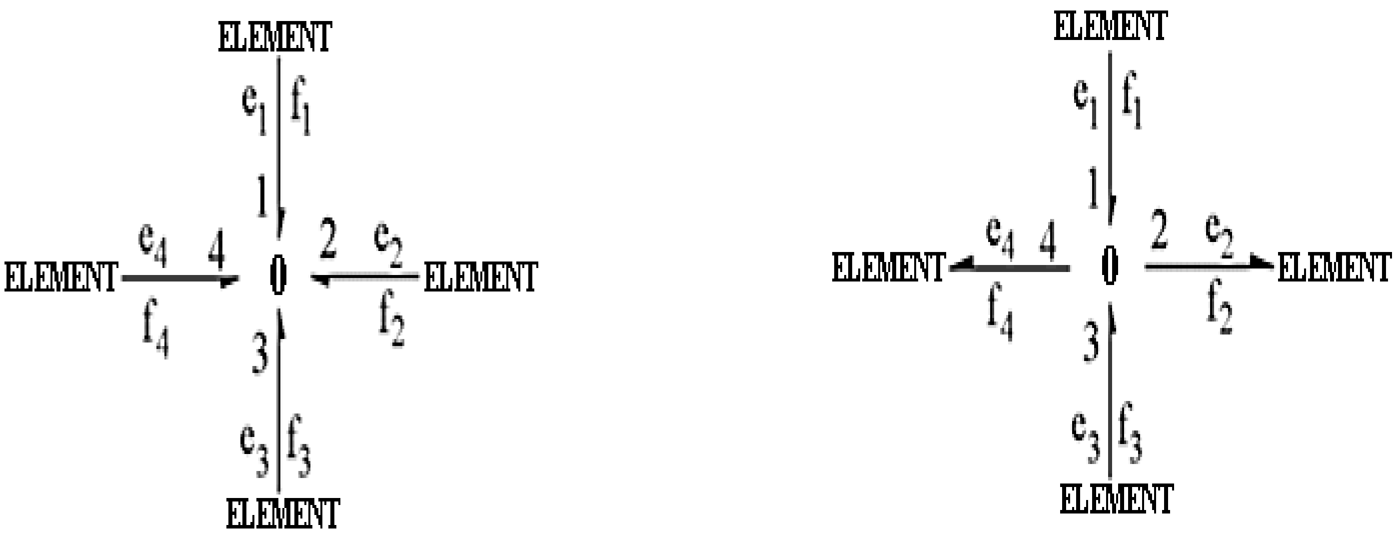
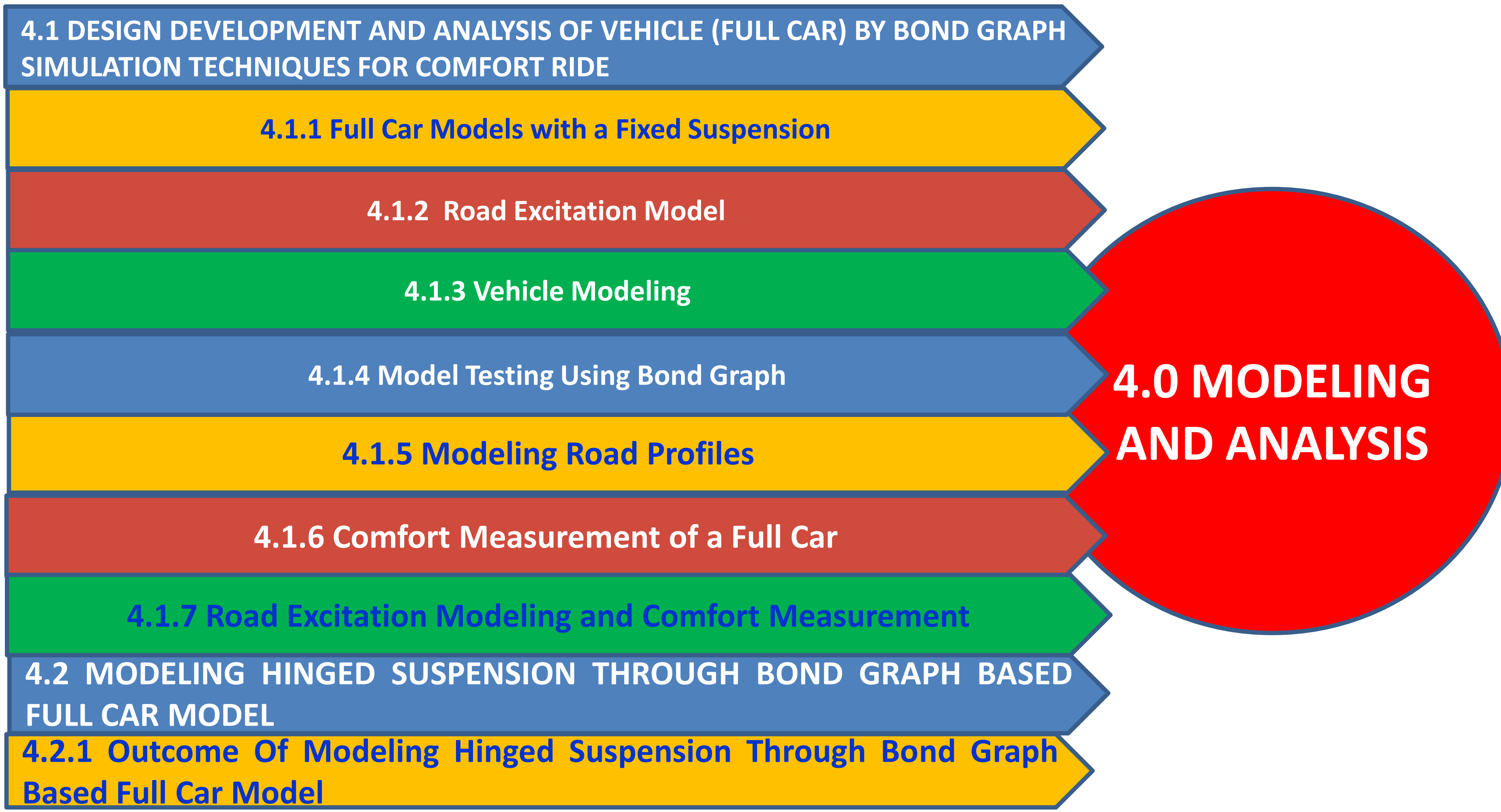


Fig.3.2 '0' Junction Element Models

3.7 Development of The Bond Graph Model

Every model reflects human understanding and is limited by knowledge, purpose, or tools used. In bond graph systems, each graph is unique, representing system dynamics through junctions.

The framework from **Mukherjee et al. (2006)** aids in developing these models, where 1 and 0 junctions are crucial. These junctions conserve power and are reversible.


4.1 DESIGN DEVELOPMENT AND ANALYSIS OF VEHICLE (FULL CAR) BY BOND GRAPH SIMULATION TECHNIQUES FOR COMFORT RIDE
4.1.1 Full Car Models with a Fixed Suspension
4.1.2 Road Excitation Model
4.1.3 Vehicle Modeling
4.1.4 Model Testing Using Bond Graph
4.1.5 Modeling Road Profiles
4.1.6 Comfort Measurement of a Full Car
4.1.7 Road Excitation Modeling and Comfort Measurement
4.2 MODELING HINGED SUSPENSION THROUGH BOND GRAPH BASED FULL CAR MODEL
4.2.1 Outcome Of Modeling Hinged Suspension Through Bond Graph Based Full Car Model
4.0 MODELING AND ANALYSIS

4.0 MODELING AND ANALYSIS

4.1 Design Development and Analysis of Vehicle (Full Car) by Bond Graph Simulation Techniques for Comfort Ride

The studies here have been divided into two parts. The first one deal with the modeling of full car with simple spring dashpot suspension on the front end rear wheels as it goes over a road bump; various responses are studies through a bond graph model.

The second study deals with development of a more elaborate suspension model with a hinged arm between the wheels and the chassis with spring and dash pots on both sides of the pivot.

Contd.. **4.1 Design Development and Analysis of Vehicle (Full Car) by Bond Graph Simulation Techniques for Comfort Ride**

The first model, based on standard assumptions like chassis rigidity and linear tire stiffness, will be verified against the bond graph in Mukherjee et al. (2006) by using the parameters they reported.

The rocking and heaving motions of the vehicle will be compared with their results. The full car model will then be used to analyze vehicle performance over bumps. The second model, a hinged arm suspension commonly used for active control, will be developed next, and its performance will be evaluated as the vehicle passes over a bump.

4.1.1 Full Car Models with a Fixed Suspension

The distance between the center of gravity and the front and rear suspensions is labeled as 'b' and 'a,' respectively. The vehicle's vertical movement is modeled with spring and dashpot suspensions at the front and rear wheels (**Fig. 4.1**). A velocity step input of 15 m/s, sustained for 10 seconds, is applied to simulate the vehicle's behavior. The model enables the study of both heave and pitching motions.

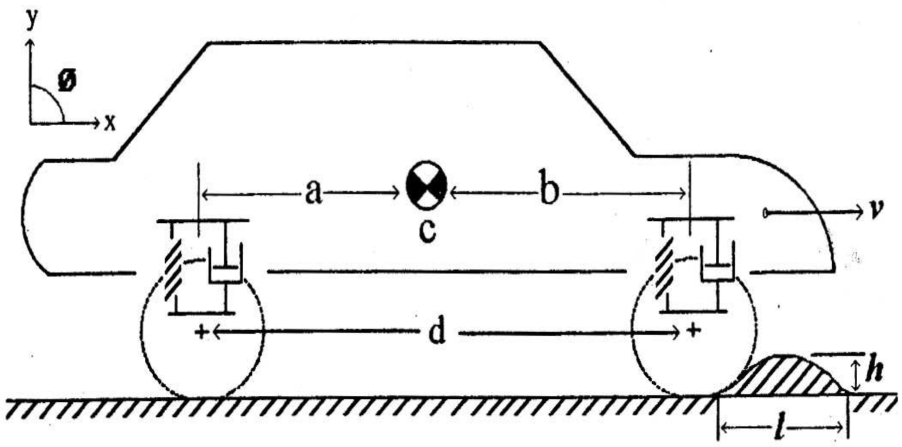


Fig. 4.1 Full car model with a fixed suspension

Contd..4.1.1 Description of the Elements of the Bond Graph

Table 4.1 The description of the elements of the Bond graph

Name of the element	Symbol
Flow equalizing junction	1
Effort equalizing junction	0
Inertial element	I
Compliant element	C
Resistive element	R
Source of effort	SE
Source of flow	SF
Transformer	TF

Contd..4.1.1 Parameters of a Full Car Model

Table 4.2 Parameters of a full car model with fixed and hinged suspension

Name of the element		Symbol
Flow equalizing junction		1
Effort equalizing junction		0
Inertial element		I
Description	Parameter name	Values used
Velocity of the full car	v	15 m/s
Height of ground excitation	h	0.1 m
Length of ground excitation	l	0.3 m
Rear damper	REAR_DM	100 N.s/m
Rear stiffness	REAR_ST	20000 N/m
Front damper	FRONT_DM	100 N.s/m
Front stiffness	FRONT_ST	20000 N/m
Mass of the full car	CAR_MASS	1080 kg
Distance of rear wheel from C.G	a	1.1 m
Distance of front wheel from C.G.	b	0.9 m
Moment of inertia of the full car	J_c_CAR	250 kgm^2

4.1.2 Road Excitation Model

A road excitation model is a mathematical representation used to characterize the unevenness and irregularities of road surfaces that induce vibrations in vehicles. This model is crucial in vehicle dynamics, as it enables the simulation of road-induced forces and their effects on vehicle components, ride comfort, and durability.

Road Classifications: Roads are typically classified into categories based on their roughness, such as ISO 8608 road classes. These classifications use a power spectral density (PSD) function to quantify the road surface irregularities.

4.1.3 Vehicle Modeling

Vehicle modeling involves creating mathematical and computational representations to simulate vehicle behavior under various conditions. These models aid in vehicle design, performance evaluation, and control system development, offering insights into dynamics, energy efficiency, safety, and environmental impact.

4.1.4 Model Testing Using Bond Graph

Bond graphs serve as a multidisciplinary tool for modeling and analyzing dynamic systems by representing energy flow across mechanical, electrical, hydraulic, and other domains. They enable systematic model testing and validation against expected results, ensuring accurate system behavior analysis.

4.1.5 Modeling Road Profiles

Road profile modeling captures elevation changes and surface irregularities, crucial for analyzing vehicle-road interaction, enhancing comfort and safety, and ensuring infrastructure durability.

4.1.6 Comfort Measurement of a Full Car

Full car comfort involves ride quality, noise, vibration, seat ergonomics, climate control, and passenger experience. Accurate measurement is key to vehicle design, customer satisfaction, and competitiveness.

4.1.7 Road Excitation Modeling and Comfort Measurement

Road excitation modeling and comfort measurement are vital for vehicle dynamics, aiding in suspension design, ride quality improvement, and minimizing fatigue-inducing vibrations.

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

i). Creating Bond Graph Model: To create the bond graph model for the full car (Fig. 4.2), a scheme is developed using literature and bond graph logic with two ‘0’ junctions, four ‘1’ junctions, and relevant transformers. The model is implemented in the Symbol Shakti software via its Bond Pad entry module.

The model entry begins with the flow-equalizing ‘1’ junction structure (Fig. 4.2). The software aids activities like numbering and causality checks. In bond graphs, efforts on bonds connected to a 0-junction are equal, while the algebraic sum of flows is zero, with signs determined by bond half-arrow directions.

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

Bond Graph Model for Full car

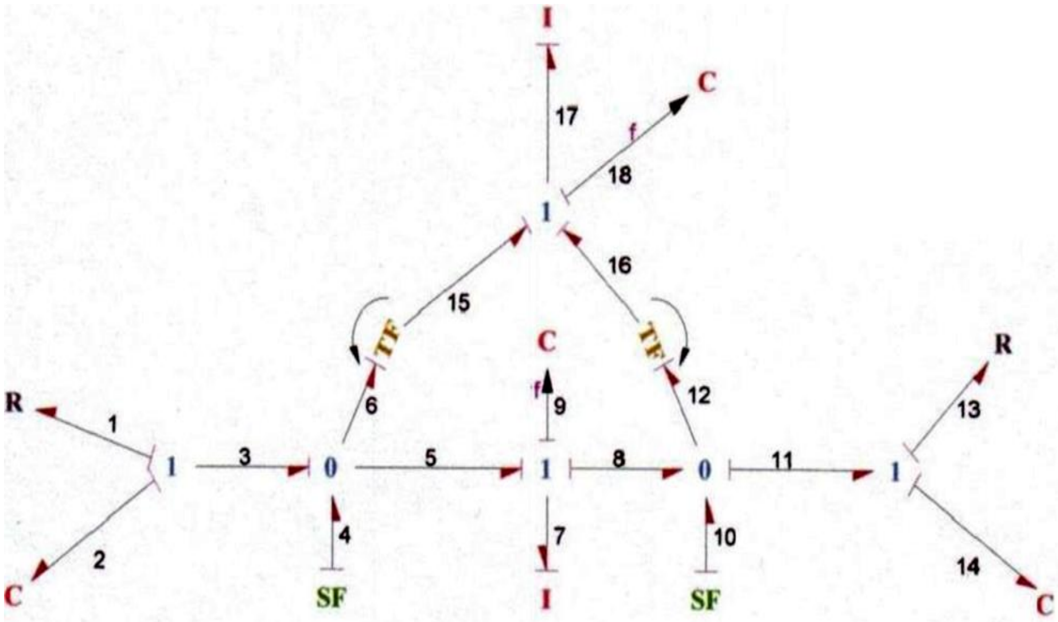


Fig. 4.2 Bond graph model of full car model with a fixed suspension

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

ii) Running the Model on Symbol Shakti: To run the model is started simulator by pressing simulate item from the process menu. Symbol Shakti simulator window opens with active document. “Parameters” item is chosen from the view menu, A box appears showing all system parameters followed by a small square box to the left of each parameter. Here, model parameter are set, Time range is set by selecting “Simulation properties” item from the “View” menu.

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

iii). Simulation and the Results Obtained: The Full car model with fixed suspension has been tested with the parameters shown in the Table 4.2, along with the road bump.

Fig.4.3, shows the rocking motion of the full car when the full car is driven at 15 m/s speed. These results match almost exactly with the results for this reported in Mukherjee et al (2006).

Fig 4.4 shows the heaving motion of the full car under the same conditions. These results also match the ones reported in Mukherjee et al (2006). These two validate the model and also proper appreciation and the use of the Software Symbol Shakti.

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

Contd..

Further studies have been carried out using the Bond graph Model created. Motion of only the front wheel is simulated in terms of rocking and heaving as it passes over the bump in terms of time and shown in Fig. 4.5, and Fig. 4.6. Results show that the behavior of the individual wheels may be significantly different when taken alone, as compared to over all model.

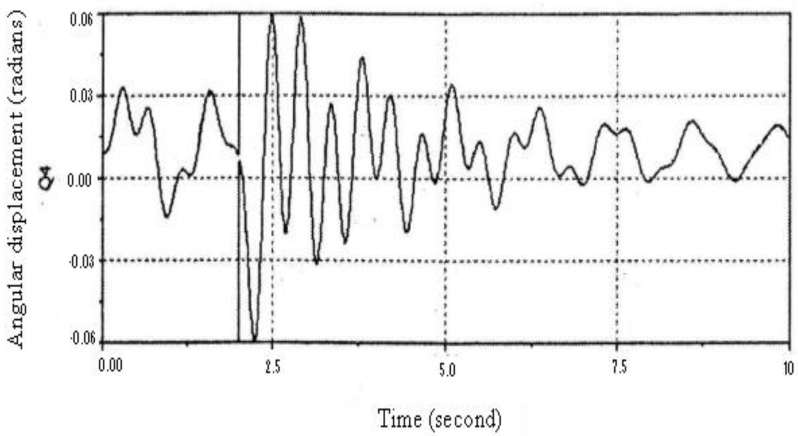


Fig. 4.3: The full car model's rocking motion at the front suspension is analyzed using vertical angular displacement at the C.G. Input parameters for Symbol Shakti include angular displacement (Q4 radians), speed (15 m/s), and time (10 seconds).

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

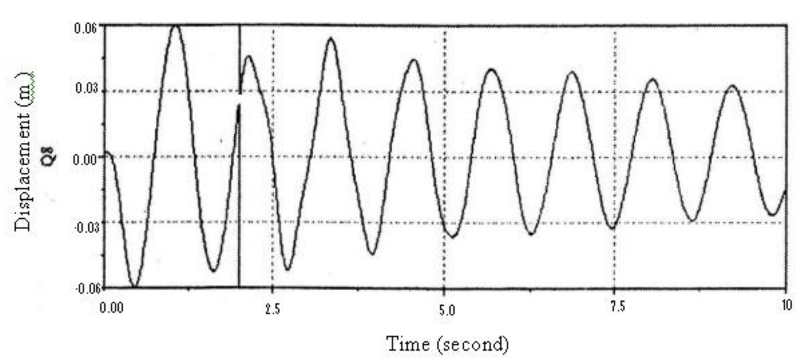


Fig. 4.4: Heaving motion of the full car model at rear suspension with vertical displacement at C.G., Input parameters to Symbol Shakti, Displacement- (Q8 m), Speed 15m/s, Time 10 seconds

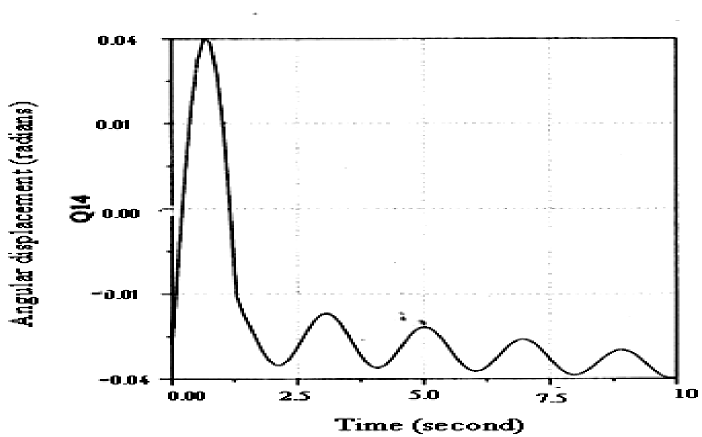


Fig.4.5 Rocking motion of the full car model at front suspension with vertical Angular displacement , Input parameters to Symbol Shakti, Angular displacement- (Q14 radians), Speed 15 m/s, Time 10 seconds

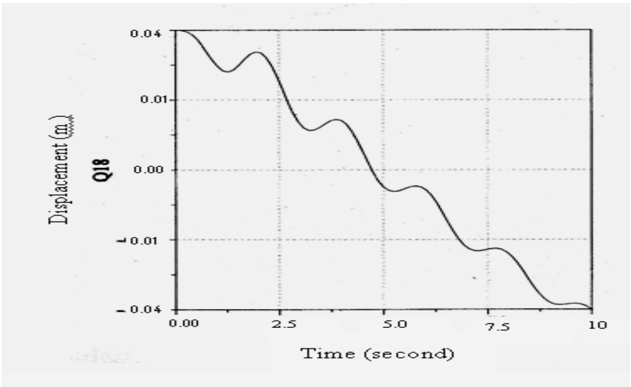


Fig. 4.6 Heaving motion of the full car model at rear suspension with vertical displacement, Input parameters to Symbol Shakti, Displacement (Q18 m), Speed 15 m/s, Time 10 seconds.

Contd..**4.1.7 Road Excitation Modeling and Comfort Measurement**

iv).Validation of Results and Possibilities of further Studies: The results were validated using data from Mukherjee et al. (2006). The first two graphs, showing rocking and heaving motion of the full car, yielded identical values, confirming the correct use of the software, units, and values.

4.2 Modeling Hinged Suspension through Bond Graph Based Full Car Model

i). Development of the Vehicle Model: The vehicle model, shown in Fig 4.7, incorporates additional elements compared to the earlier design. Two hinged platforms have been added, connected to the vehicle chassis, with springs and dashpots between the hinged arms and the vehicle, as proposed by Mukherjee et al. (2006).

Contd..**4.2 Modeling Hinged Suspension through Bond Graph Based Full Car Model**

i). Development of the Vehicle Model:

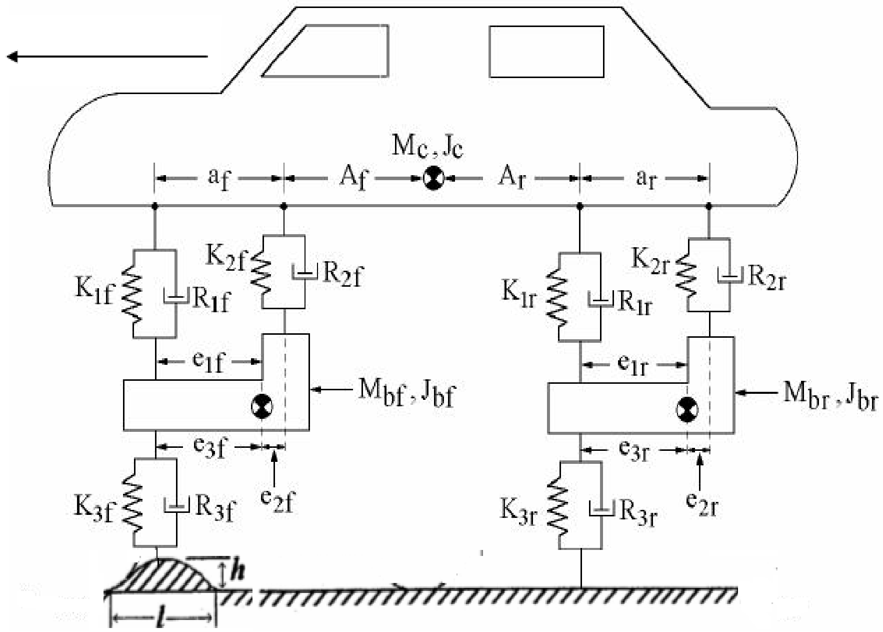


Fig. 4.7: Full car model with hinged suspension

Contd..**4.2 Modeling Hinged Suspension through Bond Graph Based Full Car Model**

ii).Development of the Bond Graph Model:

The development of the Bond graph model begins similarly to the non-hinged suspension vehicle, using the same road bump details.

For the proposed configuration, two additional hinges with associated masses and moments of inertia were added, along with two sets of springs and dashpots between the hinged platform and the vehicle body. Transformers were introduced to create a feasible system, as depicted in Fig. 4.8.

Contd..**4.2 Modeling Hinged Suspension through Bond Graph Based Full Car Model**

Development of the Bond Graph Model

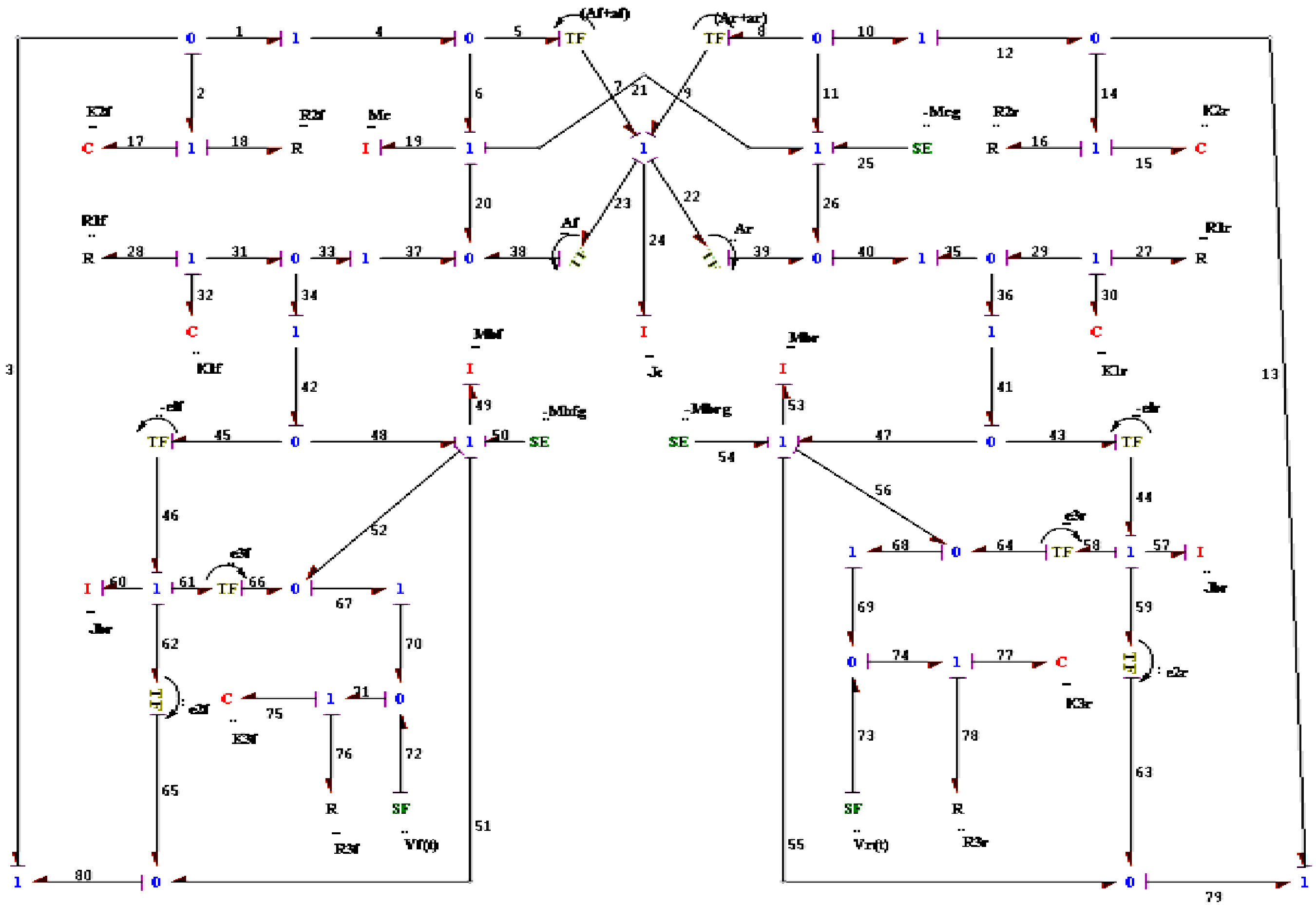


Fig. 4.8: Bond graph model of full car model with a hinged suspension

Contd..4.2 Modeling Hinged Suspension through Bond Graph Based Full Car Model

iii).Running the Model on Symbol Shakti: A compiled model in Symbol Shakti can be executed to generate results and plots by specifying relationships. Bond activation allows for detailed observation of value variations. The system's equations for the full car model with hinged suspension are automatically solved by the Symbol Shakti solver, as shown in **Fig. 4.9**.

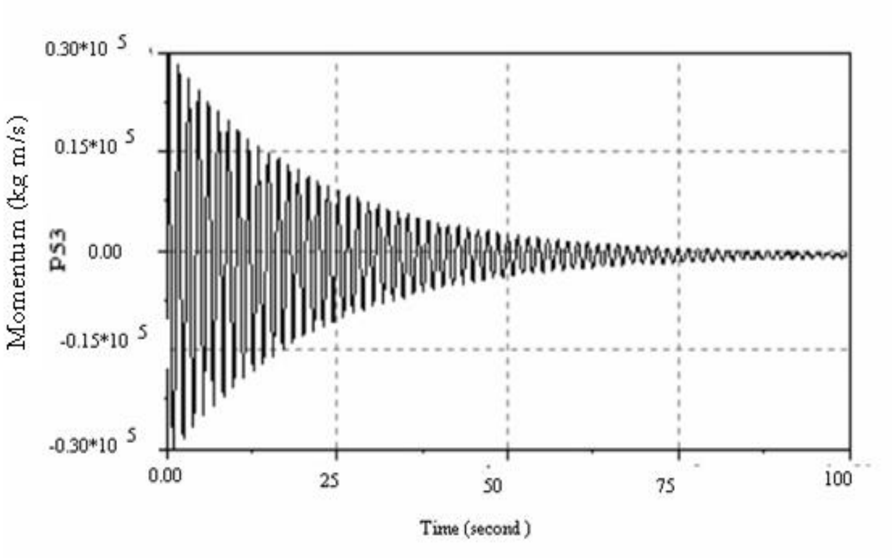


Fig.4.9 Heaving motion of the full car model at rear suspension. Observed through Momentum in bonds, Speed 15 m/s, Time 100 seconds, Numerical data is taken from Table 6.3

4.2.1 Outcome of Modeling Hinged Suspension through Bond Graph Based Full Car Model

Outcome: This study explores bond graph-based modeling of automotive systems, focusing on a full-car suspension model. A hinged arm suspension was modeled for a vehicle traversing a sine-shaped bump.

The study comprised two parts:
First part, a full car with a simple suspension was modeled, and simulation results were verified with literature data.

Second part, a hinged suspension was integrated into the full-car model. The bond graph model was successfully compiled, yielding preliminary results for specific scenarios.

5.0 RIDE COMFORT AND QUALITY ASSESMENT

- 5.1 RIDE QUALITY
- 5.2 RIDE COMFORT
- 5.3 Vehicle Stability
- 5.4 Measuring and Improving Road Vehicle Comfort
- 5.5 Vehicle Comfort Phenomena
- 5.6 Vehicle Indices Used To Evaluate Comfort
- 5.7 Vehicle Dynamics Performance
- 5.8 Vehicle Performance…………………………….
- 5.15 Commercial Motor Vehicles (CMVs) on Indian Rural Roads

5.0 RIDE COMFORT AND QUALITY ASSESMENT

5.1 Ride Quality

Ride quality measures the comfort experienced by passengers or drivers during travel, considering factors like vibration, noise, smoothness, and stability. Ensuring optimal ride quality is essential for user satisfaction, safety, and infrastructure durability.

5.2 Ride Comfort

Ride comfort is a key factor in vehicle design, affecting passenger satisfaction, safety, and usability. It includes both physical and psychological experiences, influenced by vibration, noise, and the vehicle's ability to absorb road irregularities. Manufacturers aim to improve ride comfort while balancing other performance aspects like handling, efficiency, and cost.

5.0 RIDE COMFORT AND QUALITY ASSESEMENT

5.3 Vehicle Stability

Vehicle stability is vital in automotive engineering, ensuring the vehicle maintains its path and remains controllable in different driving conditions. It is key to both passenger safety and optimal vehicle performance.

5.4 Measuring and Improving Road Vehicle Comfort Under Various Running Conditions

Vehicle comfort is a critical factor influencing passenger satisfaction and driving experience. It encompasses various parameters such as ride quality, noise, vibration, thermal comfort, and seating ergonomics.

5.0 RIDE COMFORT AND QUALITY ASSESMENT

5.5 Vehicle Comfort Phenomena

Vehicle comfort refers to the overall sense of ease, relaxation, and safety experienced by passengers or drivers. It includes physical, sensory, and psychological factors such as ride quality, noise, ergonomics, climate control, and seat comfort, all of which work together to define the comfort perception in a vehicle.

5.6 Vehicle Indices Used To Evaluate Comfort

Comfort in vehicles is an essential factor for both the driver and passengers. The experience of comfort involves not only physical aspects, such as seat quality and cabin space, but also psychological factors, such as noise levels and ride smoothness.

5.0 RIDE COMFORT AND QUALITY ASSESEMENT

5.7 Vehicle Dynamics Performance

Vehicle dynamics studies the forces and motion involved in a vehicle's movement, stability, and control, focusing on its interaction with the road and how the driver's inputs influence vehicle behavior.

5.8 Vehicle Performance

Vehicle performance encompasses a vehicle's efficiency, handling, speed, acceleration, and safety across various conditions. It is influenced by factors such as engine design, drivetrain, aerodynamics, suspension, tires, and onboard technology.

5.9 Effect of Vertical Vibration Frequency in Vertical Dynamics Analysis

Vertical vibration frequency is crucial in analyzing mechanical systems, including structural engineering and vehicle dynamics. Understanding its impact ensures structural integrity, comfort, and efficiency.

5.10 Effect of Vertical Vibration Frequency in Full Car

Vertical vibration frequencies significantly affect a vehicle's comfort, performance, and durability, arising mainly from road irregularities and suspension-road interactions.

5.0 RIDE COMFORT AND QUALITY ASSESMENT

5.11 Suspension Deflection in a Full Car

Suspension deflection in vehicle dynamics refers to the movement of suspension components under load, crucial for understanding a car's response to road irregularities and driving conditions. It directly impacts ride comfort and handling performance.

5.12 RMS Vertical Body Acceleration

RMS vertical body acceleration is vital in vehicle dynamics, assessing the forces affecting vehicles and occupants. It significantly influences ride quality and stability.

5.0 RIDE COMFORT AND QUALITY ASSESMENT

5.13 Road Holding in a Full Car

Road holding refers to a vehicle's ability to maintain stability and control on diverse road surfaces, crucial for ensuring safety, comfort, and effective driving performance, particularly during turns or on uneven terrain.

5.14 Overall Vehicle Performance

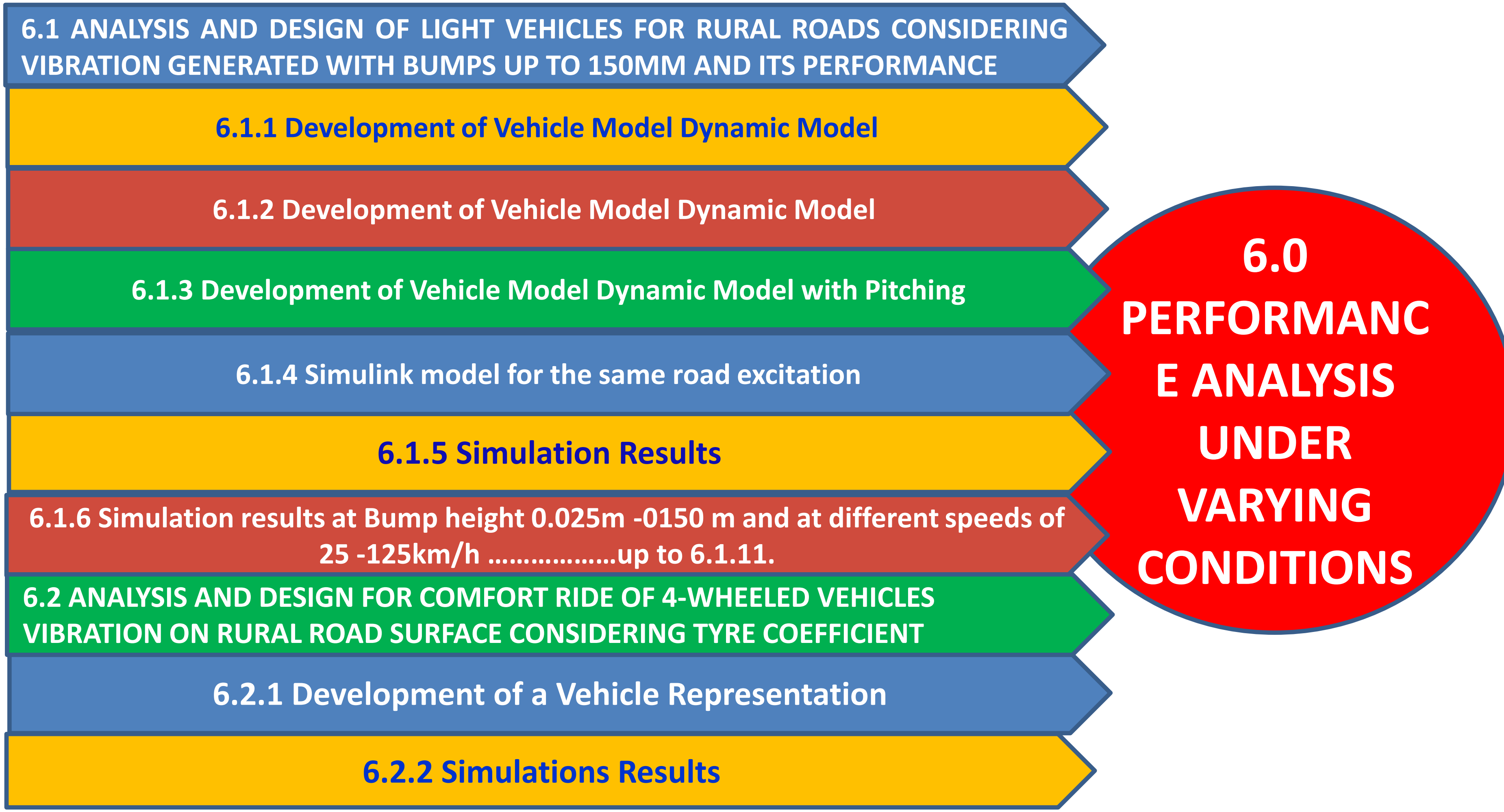
Overall vehicle performance includes factors like engine efficiency, handling, braking, comfort, and safety. It reflects the integration of mechanical design, technology, and user experience to deliver optimal operation and driving satisfaction.

5.0 RIDE COMFORT AND QUALITY ASSESEMENT

5.15 Commercial Motor Vehicles (CMVs) on Indian Rural Roads

India’s extensive road network connects rural areas, home to 65% of the population. CMVs are essential for goods and passenger transport in these regions, driving economic growth.

However, challenges like poor infrastructure and regulatory gaps demand a comprehensive approach, including reforms, infrastructure upgrades, and community participation.


6.1 ANALYSIS AND DESIGN OF LIGHT VEHICLES FOR RURAL ROADS CONSIDERING VIBRATION GENERATED WITH BUMPS UP TO 150MM AND ITS PERFORMANCE
6.1.1 Development of Vehicle Model Dynamic Model
6.1.2 Development of Vehicle Model Dynamic Model
6.1.3 Development of Vehicle Model Dynamic Model with Pitching
6.1.4 Simulink model for the same road excitation
6.1.5 Simulation Results
6.1.6 Simulation results at Bump height 0.025m -0150 m and at different speeds of 25 -125km/h ………………up to 6.1.11.
6.2 ANALYSIS AND DESIGN FOR COMFORT RIDE OF 4-WHEELED VEHICLES VIBRATION ON RURAL ROAD SURFACE CONSIDERING TYRE COEFFICIENT
6.2.1 Development of a Vehicle Representation
6.2.2 Simulations Results
6.0 PERFORMANCE ANALYSIS UNDER VARYING CONDITIONS

6.0 PERFORMANCE ANALYSIS UNDER VARYING CONDITIONS

The operation of high-speed vehicles on rural roads poses unique challenges compared to highways and expressways, affecting costs, vehicle lifespan, and comfort.

A ***notable research gap*** exists in designing vehicles suited for India's rural and district roads.

With *India's road network spanning 6.2 million km—the second-largest globally after the U.S. network of 6.8 millions*—addressing this gap is vital for improving transportation efficiency and durability.

6.1 Analysis and Design of Light Vehicles For Rural Roads Considering Vibration Generated With Bumps Up To 150mm and Its Performance

As of March 31, 2020, India possesses the world's second-largest road network, spanning 62,15,797 km, trailing only the United States with 68,53,024 km. The country has constructed 1,38,531 km of National Highways and Expressways, 1,76,818 km of State Highways, contributing to a total length of 3,15,349 km.

Remarkably, approximately 50% of this extensive road network is situated in rural areas, consisting of surfaces not fortified with cement pavement or bituminous materials of requisite strength. Despite lacking these enhancements, these rural roads play a pivotal role in the transportation of the rural population.

Contd..**6.1 Analysis and Design of Light Vehicles For Rural Roads Considering Vibration Generated With Bumps Up To 150mm and Its Performance**

This study examines the impact of rural road vibrations on light vehicles, focusing on bumps, potholes, and irregularities that affect vehicle lifespan. A mathematical model with seven degrees of freedom is developed and simulated in Matlab/Simulink. Simulations analyze bump heights (0.025–0.150 m) and speeds (25–125 km/h), showing optimal performance and comfort at speeds above 65 km/h on challenging rural roads in India. The research highlights the importance of tyre stiffness, suspension systems, damping, and mass distribution in vehicle design.

6.1.1 Development of Vehicle Model Dynamic Model

The full car model uses linear equations of mass, spring, and damper with a seven-degree-of-freedom suspension system to ensure a comfortable ride. To analyze and optimize vehicle vibrations, a full car vibration model is essential. The model, shown in Fig. 6.1, incorporates body bounce and accounts for potential differences between the front and rear suspensions.

6.1.2 Development of Vehicle Model Dynamic Model

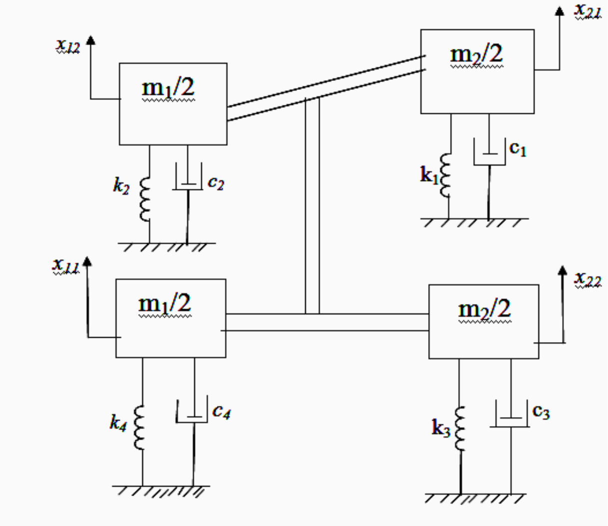


Fig. 6.1 Dynamic model

6.1.3 Development of Vehicle Model Dynamic Model with Pitching

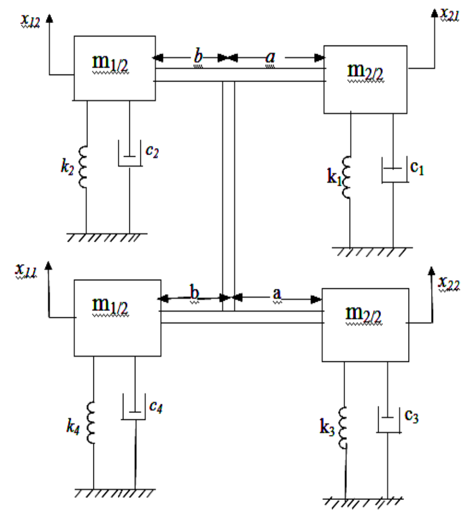


Fig. 6.2: Dynamic model with pitching

6.1.4 Simulink model for the same road excitation

The system simulates the suspension system, including sprung mass, unsprung mass, unsprung wheels, pitching, rolling, and bouncing. A mathematical model of a 4-wheeled vehicle with a driver seated on a cushioned seat is developed and simulated using Simulink software.

Table-6.1: Parameter of full car model

	Parameter Description	Value
m_s	Mass of the sprung mass	1300 kg
m_{uf}	Front mass of the wheel or unsprung mass	65 kg
m_{ur}	Rear mass of the wheel or unsprung mass	60 kg
Ip	Pitch moment of inertia	2391.08 kgm^2
Ir	Roll moment of inertia	391.08 kg m^2
k_f	Stiffness of vehicle for front	36300 N/m
k_r	Stiffness of vehicle for rear	19600 N/m
c_f	Front damping coefficient	4000 kNs/m
c_r	Rear damping coefficient	3000 kNs/m
a	Distance from centre of sprung mass to front wheel	1.6 m
b	Distance from centre of sprung mass to rear wheel	0.9 m

6.1.5 Simulation Results

Simulation results, presented in Table 6.1, are summarized and displayed graphically for better understanding. Speeds ranging from 25 km/h to 125 km/h were analyzed to evaluate suspension displacement.

Simulations using various bump heights (0.025 m, 0.050 m, 0.075 m, 0.100 m, 0.125 m, and 0.150 m) demonstrated vibrations over different time intervals, providing insights into achieving optimal comfort. The logical arrangement of results through Simulink made the findings easier to interpret.

6.1.6 Simulation results at Bump height 0.025m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below

These results, along with additional bump heights (0.050 m, 0.075 m, 0.100 m, 0.125 m, and 0.150 m), are analyzed for vibrations over different time intervals. The logical representation using Simulink ensures clarity, making the results easier to review and interpret for assessing comfort ride dynamic

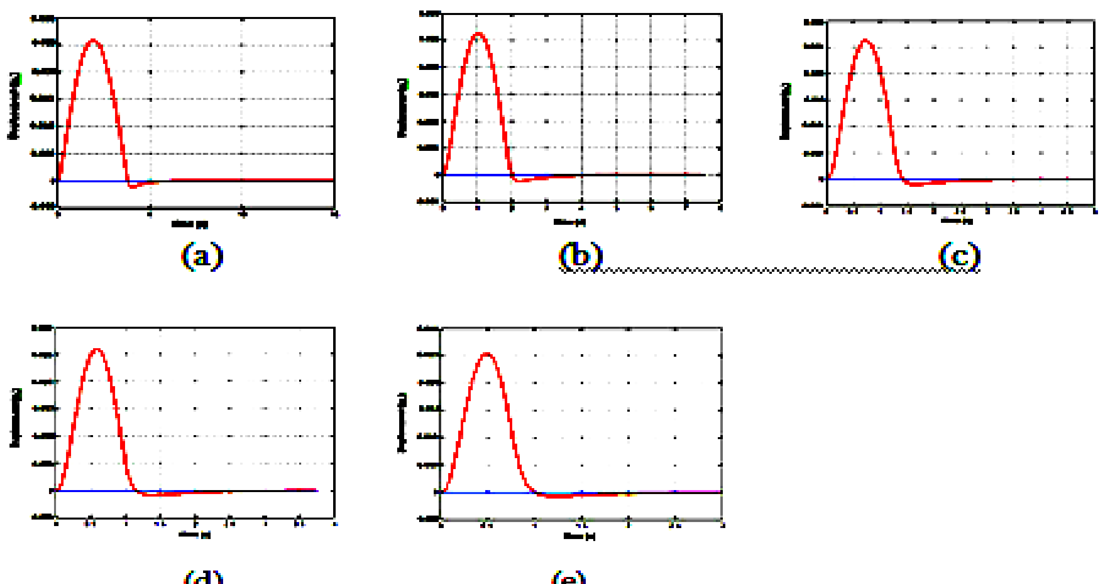


Fig 6.3: Sprung-Mass Displacement vs. Time at Bump height 0.025 m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

6.1.7 Simulation results at Bump height 0.050m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below

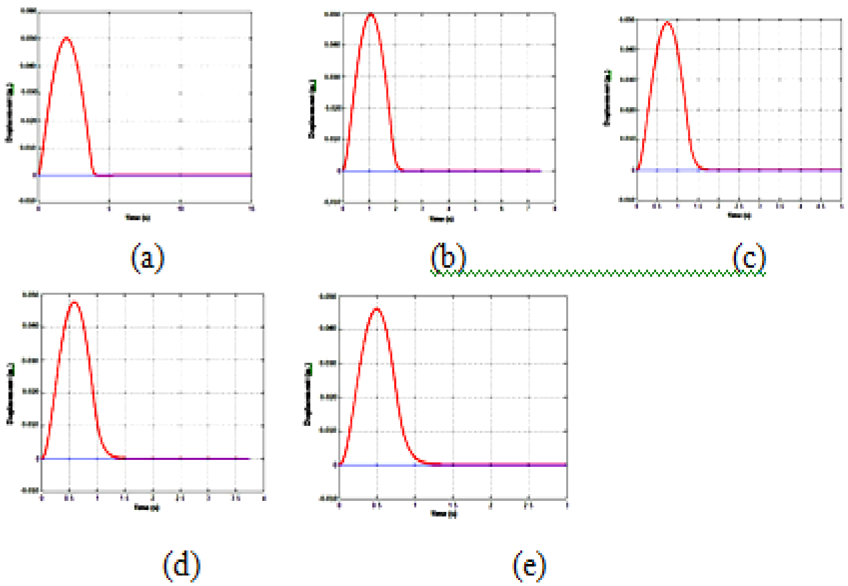


Fig 6.4: Sprung-Mass Displacement vs. Time at Bump height 0.050 m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

6.1.8 Simulation results at Bump height 0.075 m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below

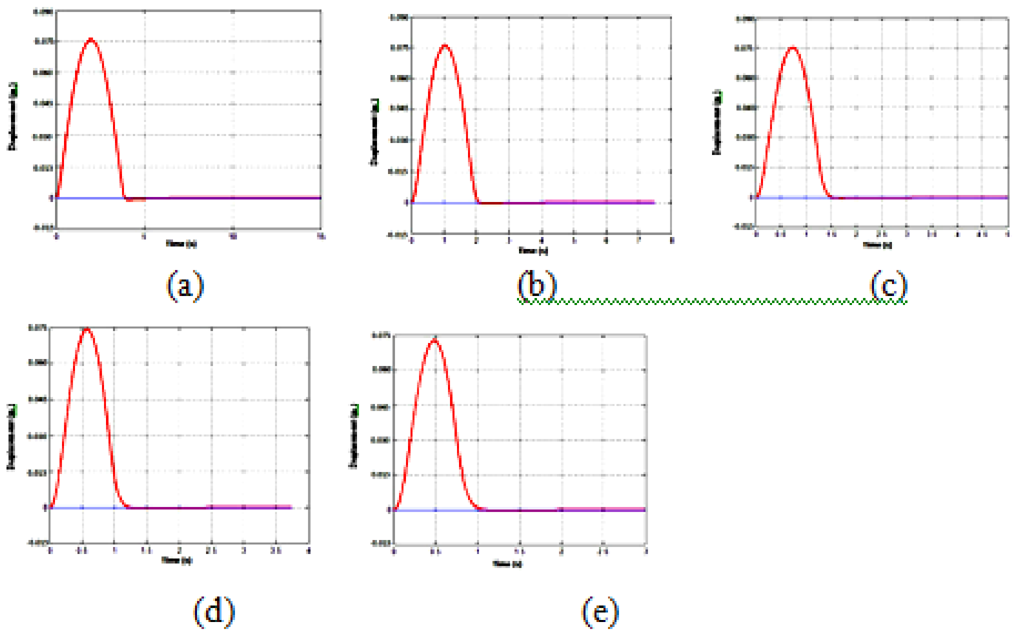


Fig 6.5: Sprung-Mass Displacement vs. Time at Bump height 0.075 m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

6.1.9 Simulation results at Bump height 0.100m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below

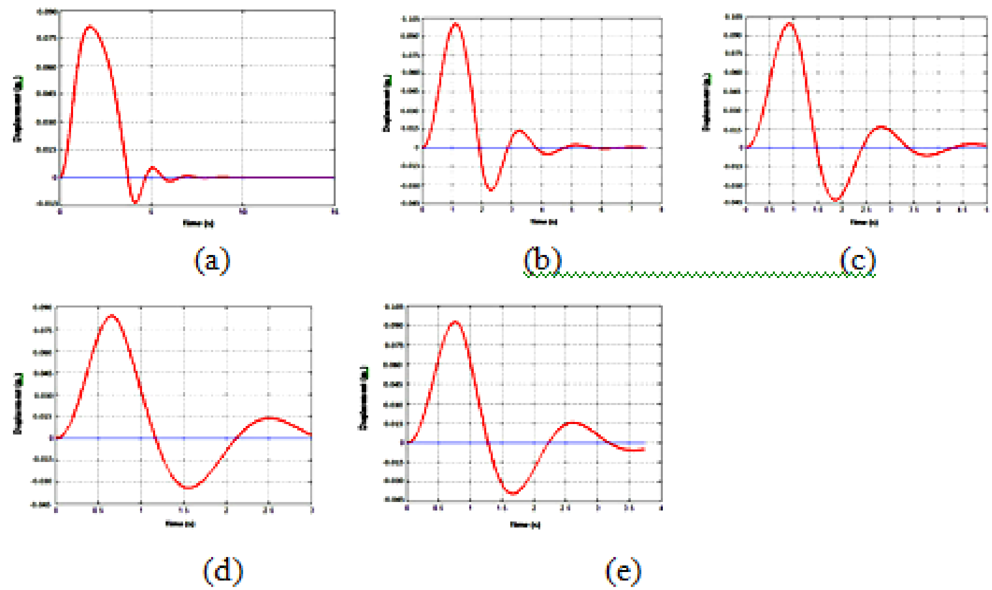


Fig 6.6: Sprung-Mass Displacement vs. Time at Bump height 0.100 m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

6.1.10 Simulation results at Bump height 0.125m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below

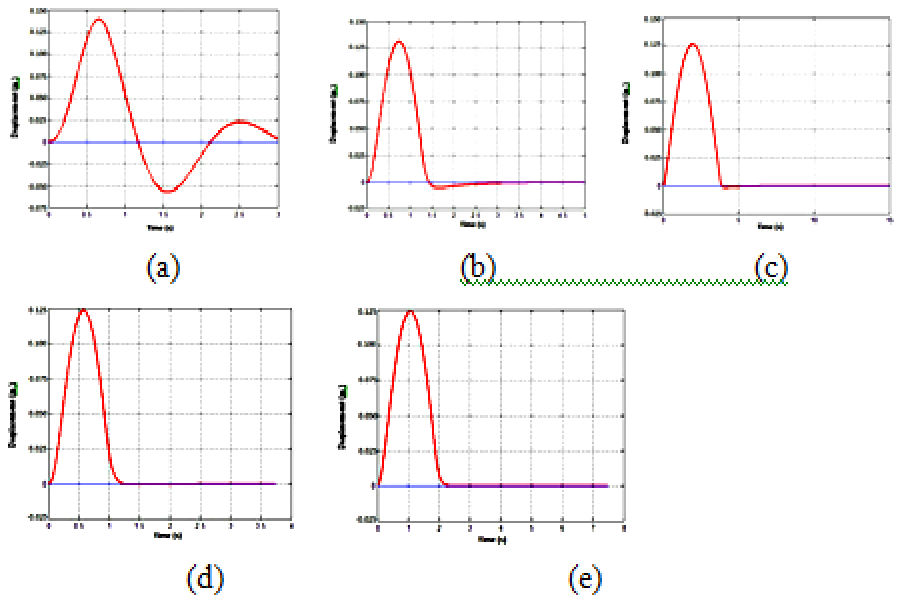


Fig 6.7: Sprung-Mass Displacement vs. Time at Bump height 0.125 m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

6.11 Simulation results at Bump height 0.150m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below

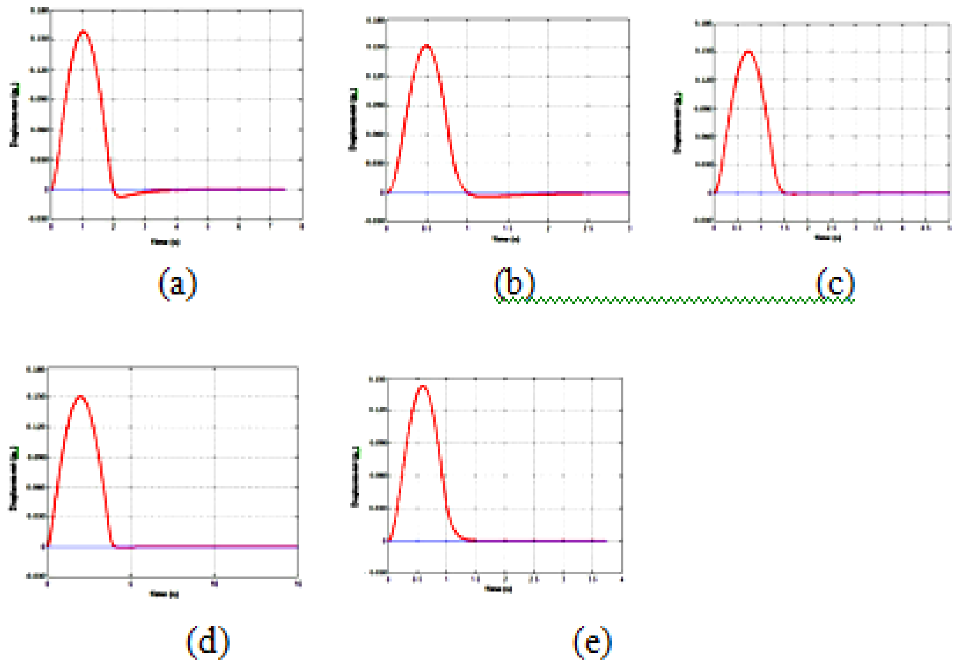


Fig. 6.8: Sprung-Mass Displacement vs. Time at Bump height 0.150 m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h

6.2 Analysis and Design For Comfort Ride of 4-wheeled Vehicles Vibration on Rural Road Surface Considering Tyre Coefficient and To Reduce Climate Impact

This study examines the effect of tyre damping coefficients up to 8 kN/m on vehicle vibrations. The results show that sprung mass displacement improves until a speed of 62.5 km/h, after which it decreases up to 125 km/h.

These findings are essential for optimizing ride comfort and understanding vehicle dynamics.

Contd..**6.2 Analysis and Design For Comfort Ride of 4-wheeled Vehicles Vibration on Rural Road Surface Considering Tyre Coefficient and To Reduce Climate Impact**

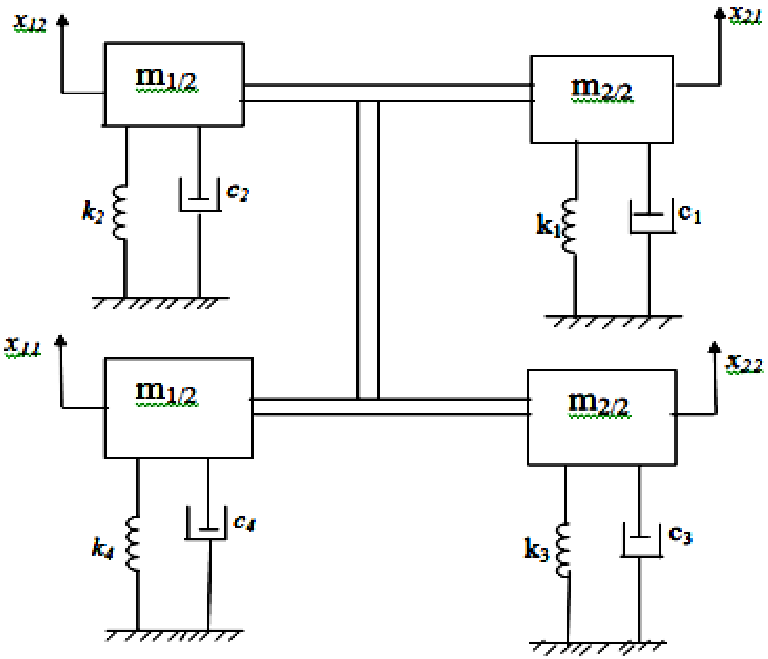


Fig. 6.9: Four wheeled car model when front and rear load are equal

Contd..**6.2 Analysis and Design For Comfort Ride of 4-wheeled Vehicles Vibration on Rural Road Surface Considering Tyre Coefficient and To Reduce Climate Impact**

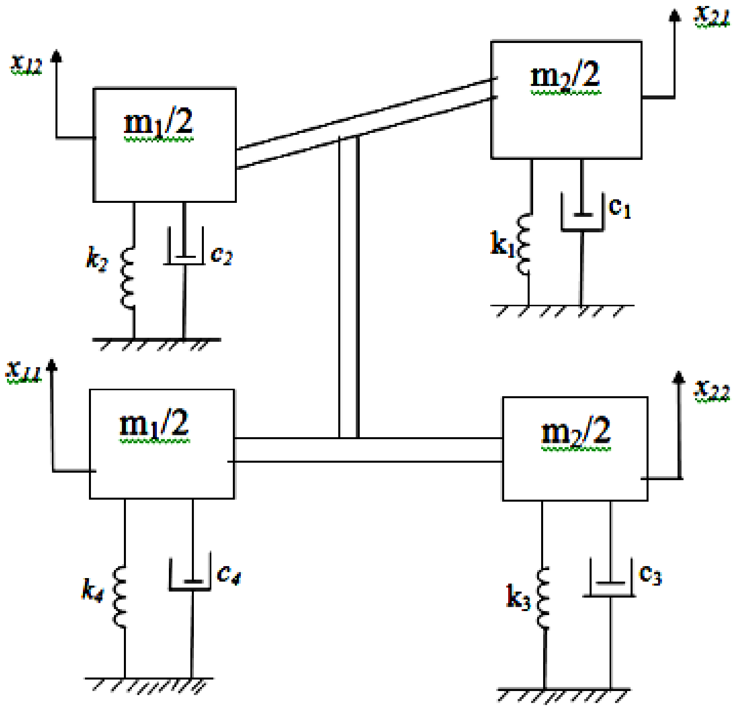


Fig. 6.10: Four wheeled car model having different load

6.2.1 Development of a Vehicle Representation

The development of a vehicle model for sprung mass focuses on heave, roll, and pitch dynamics, which complement rebound perpendicular to the sprung mass. Key parameters include:

ms: Sprung mass

muf: Unsprung mass of the front wheel

mur: Unsprung mass of the rear wheel

Ip: Moment of inertia for pitching

Ir: Moment of inertia for rolling

Zs: Displacement of the vehicle body

Zs1, Zs2, Zs3, Zs4: Displacement at each corner of the vehicle

Zu1, Zu2, Zu3, Zu4: Displacement of each wheel

These variables form the foundation for analyzing vehicle dynamics and performance.

Contd..6.2.1 Development of a Vehicle Representation

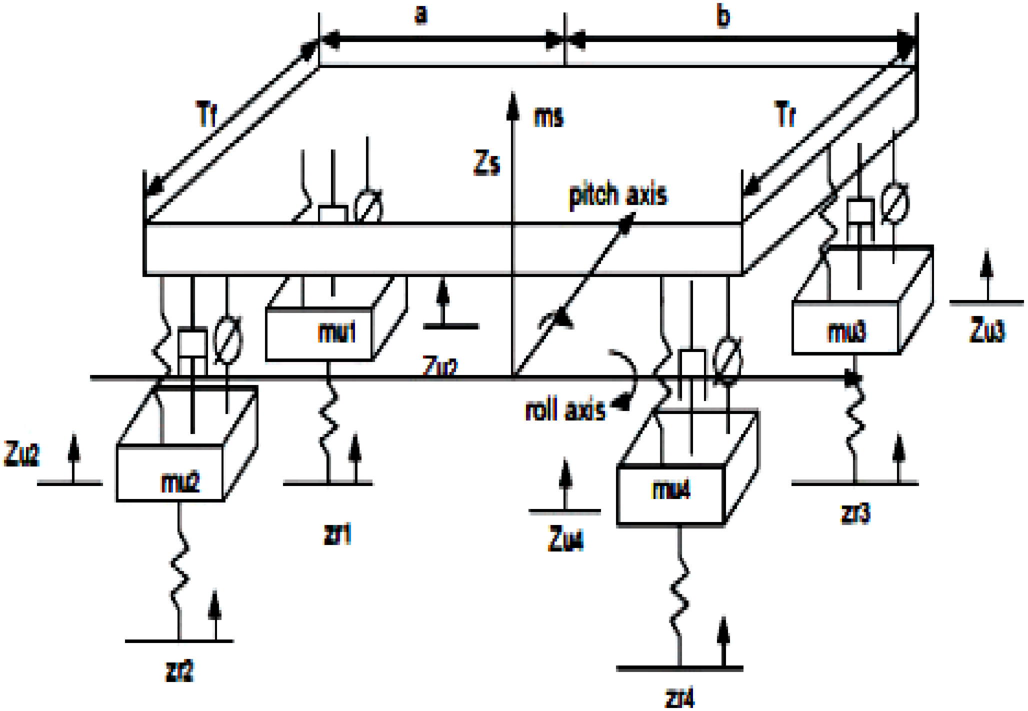


Fig. 6.11: Vehicle Representation

Contd..

6.2.1 Development of a Vehicle Representation

Table 6.2: Parameters considered for 4-Wheeled vehicle representation

SNo.	Description	Notations	Value	Units
	Sprung Mass of the vehicle	m_s	1125	kg
	Unsprung mass of front wheel	m_{uf}	65	kg
	Unsprung mass of rear wheel	m_{ur}	69	kg
	Moment of Inertia of pitching	Ip	2500	kgm^2
	Moment of Inertia of rolling	Ir	500	kgm^2
	Stiffness of vehicle at front axle	k_f	36,000	N/m
	Stiffness of vehicle at rear axle	k_r	20,000	N/m
	Front treat	T_f	0.505	m
	Rear treat	T_r	0.557	m
	Front tyre stiffness	k_{tf}	182500	N/m
	Rear tyre stiffness	k_{tr}	182500	N/m
	Front tyre coefficient	b_f	8 & 16	kNs/m
	Rear tyre coefficient	b_r	4 & 12	kNs/m
	Distance from centre of sprung mass to front wheel	a	1.15	m
	Distance from centre of sprung mass to rear wheel	b	1.65	m

Contd..**6.2.1 Development of a Vehicle Representation**

Simulation results are presented in this paper. Results are also presenting in summarized and graphical form in Table 7.2 is obtained after simulating the development model. The different figures in the simulated results obtained at various speeds are 25 km/h to 125 km/h taken for simulation of parameter analyzed suspension displacement. The simulation results are presented easier to read and review and found to understand with logical order through Simulink.

6.2.2 Simulations Result

Considering the sprung-mass, un-sprung mass, tyre damping coefficients at different bumps and speeds from 25 km/h, 50km/hr, 75km/hr, 100km/hr and 125 km/h) etc. for:

First stage; rear tyre (b_r) 4 kNs/m and front tyre (b_f) 8 kNs/m

Second stage; rear tyre (b_r) 12 kNs/m, and front tyre (b_f) 16 kNs/m,

6.0 PERFORMANCE ANALYSIS UNDER VARYING CONDITIONS

Contd..**6.2.2 Simulations Result**

These simulation results are shown in the form of graphs as under:

Evaluation of Time Lags for Achieving Comfort Zones at Various Speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h) with a Fixed Rear Tyre Damping Coefficient (b_r) of 4 kNs/m, Illustrated in Figs. 6.12 to 6.16

Contd..**6.2.2 Simulations Result**

Sprung-mass displacement vs. Time lag for comfort zone,

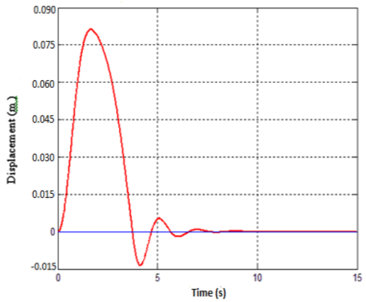


Fig. 6.12: when rear tyre *damping coefficient* (b_r) = *4 kNs/m* and speed = 25 km/h

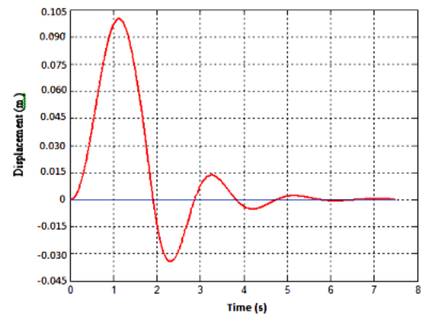


Fig. 6.13: when rear *tyre damping coefficient* (b_r) = *4 kNs/m* and speed = 50 km/h

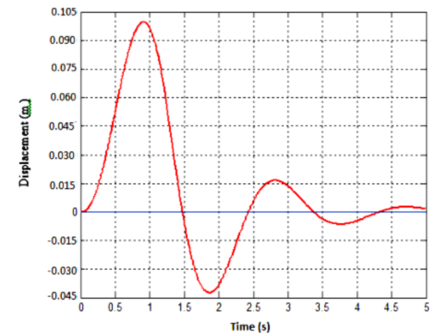


Fig.6.14: when rear *tyre damping coefficient* (b_r) = *4 kNs/m* and speed = 75 km/h

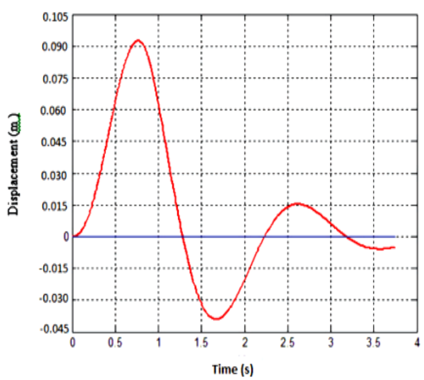


Fig. 6.15: when rear *tyre damping coefficient* (b_r) = *4 kNs/m* and speed = 100 km/h.

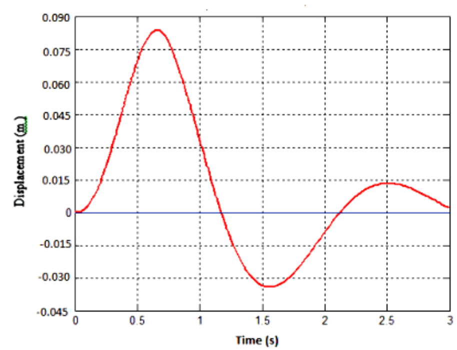


Fig. 6.16: when rear *tyre damping coefficient* (b_r) = *4 kNs/m* and speed = 125 km/h.

6.0 PERFORMANCE ANALYSIS UNDER VARYING CONDITIONS

Contd..**6.2.2 Simulations Result**

Evaluation of Time Lags for Achieving Comfort Zones at Various Speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h) with a Fixed front Tyre Damping Coefficient (b_f) of 8 kNs/m, Illustrated in Figs. 6.17 to 6.21

Contd..**6.2.2 Simulations Result**

Sprung-mass displacement vs. Time lag for comfort zone,

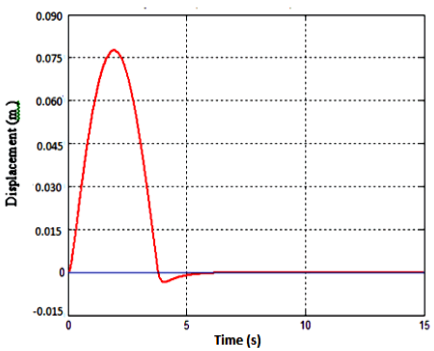


Fig. 6.17: when front *tyre damping coefficient (b_f) =8 kNs/m* and Speed = 25 km/h.

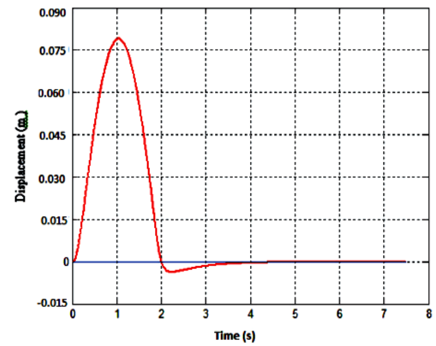


Fig. 6.18: when *front tyre damping coefficient (b_f) = 8 kNs/m* and speed = 50 km/h.

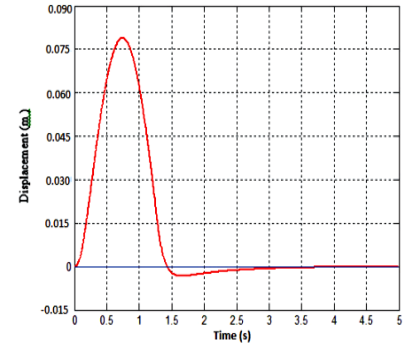


Fig. 6.19: when *front tyre damping coefficient (b_f) = 8 kNs/m* and speed = 75 km/h

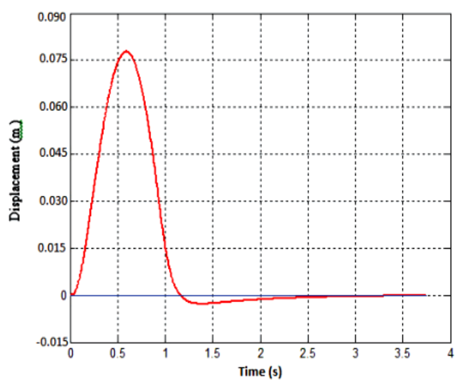


Fig. 6.20: when *front tyre damping coefficient (b_f) = 8 kNs/m* and speed = 100 km/h.

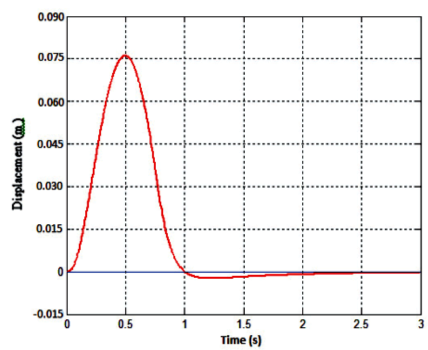


Fig. 6.21: when *front tyre damping coefficient (b_f) = 8 kNs/m* and speed = 125 km/h.

6.0 PERFORMANCE ANALYSIS UNDER VARYING CONDITIONS

Contd..**6.2.2 Simulations Result**

Evaluation of Time Lags for Achieving Comfort Zones at Various Speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h) with a Fixed Rear Tyre Damping Coefficient (b_r) of 12 kNs/m, Illustrated in Figs. 6.22 to Fig.6.26.

Contd..**6.2.2 Simulations Result**

Sprung-mass displacement vs. Time lag for comfort zone,

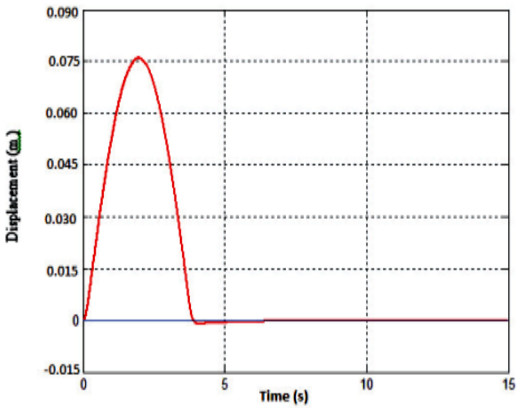


Fig. 6.22: when rear tyre damping coefficient (b_r) = 12 kNs/m and speed = 25 km/h.

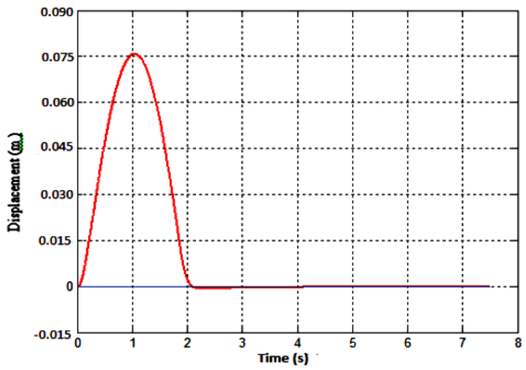


Fig. 6.23: when *rear tyre damping coefficient (b_r) =12 kNs/m* and speed = 50 km/h

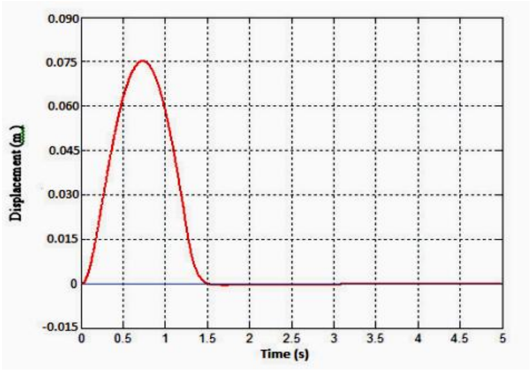


Fig. 6.24:when *rear tyre damping coefficient (b_r) = 12 kNs/m* and speed = 75 km/h

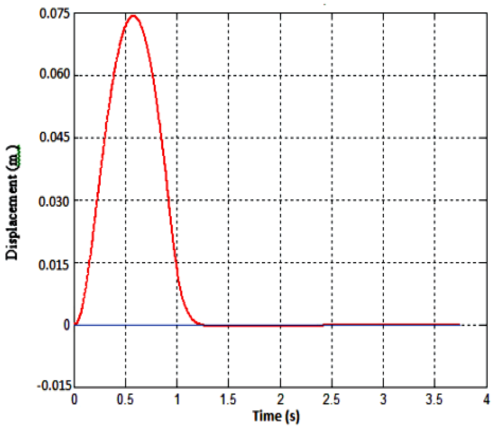


Fig. 6.25: when *rear tyre damping coefficient (b_r) =12 kNs/m* and speed = 100 km/h.

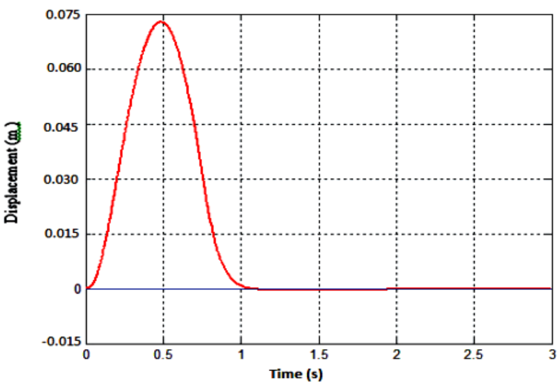


Fig. 6.26: when *rear tyre damping coefficient (b_r) =12 kNs/m* and speed = 125 km/h.

6.0 PERFORMANCE ANALYSIS UNDER VARYING CONDITIONS

Contd..**6.2.2 Simulations Result**

Evaluation of Time Lags for Achieving Comfort Zones at Various Speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h) with a Fixed Front Tyre Damping Coefficient (b_f) of 16 kNs/m, Illustrated in Fig.6.27 to Fig. 6.31

Contd..**6.2.2 Simulations Result**

Sprung-mass displacement vs. Time lag for comfort zone,

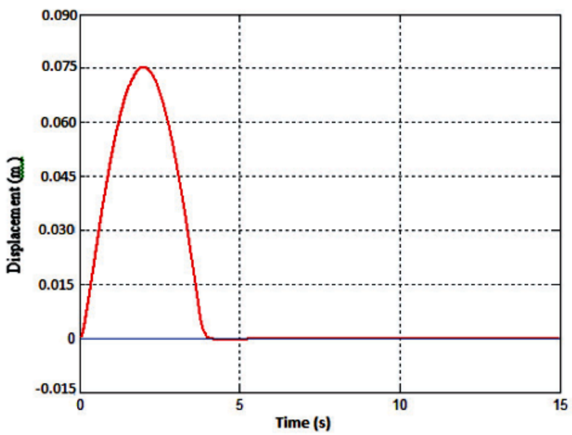


Fig. 6.27: when front tyre damping coefficient (b_f)=16 kNs/m and speed = 25 km/h

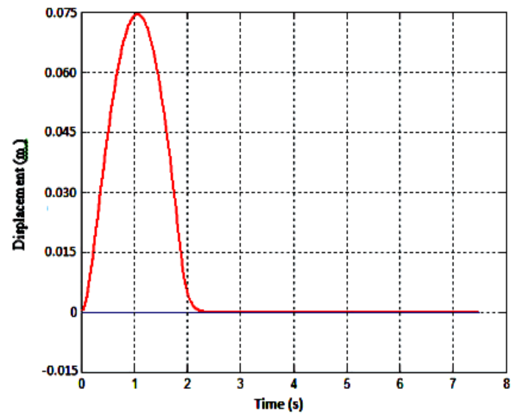


Fig. 6.28: when *front tyre damping coefficient (b_f)=16 kNs/m* and speed = 50 km/h.

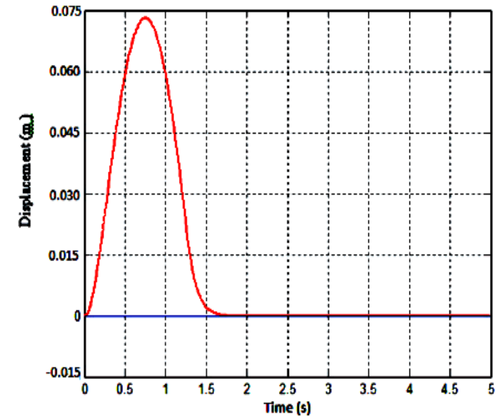


Fig. 6.29: when *front tyre damping coefficient, b_f =16 kNs/m* and speed = 75 km/h.

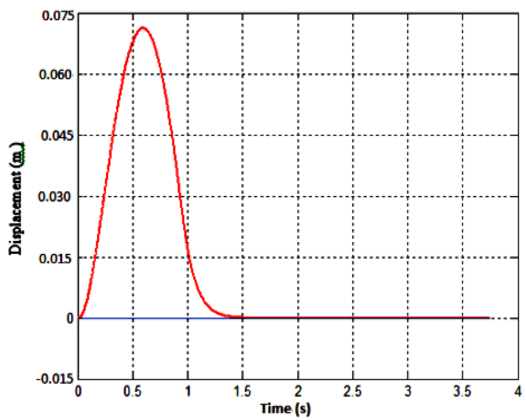


Fig. 6.30: when *front tyre damping coefficient, b_f =16 kNs/m* and speed = 100 km/h.

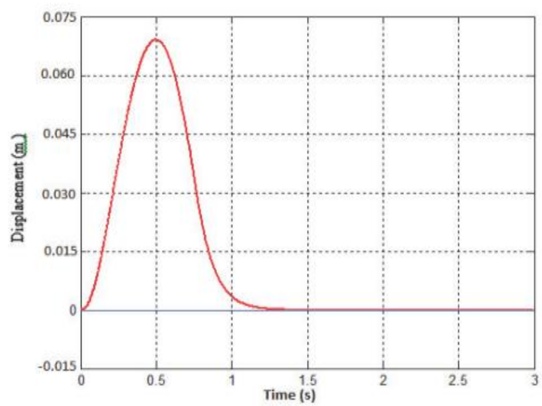


Fig. 6..31: when *front tyre damping coefficient, b_f =16 kNs/m* and speed = 125 km/h

Contd..

6.2.2 Outcome

Operating high-speed vehicles on rural roads poses unique challenges, particularly in terms of durability, comfort, and cost. Despite India's extensive rural road network, there is a significant research gap in designing vehicles specifically for these conditions.

To address this, research highlights the need for vehicles designed for rural roads with specific design criteria:

- ❑ Tyre damping coefficient: ≥4 kNs/m
- ❑Speed capability: ≤75 km/h
- ❑Suspension damping coefficient: ≤8 kNs/m for speeds between 50-75 km/h

This study is crucial in developing cost-effective, environmentally friendly vehicles tailored for rural roads, particularly in India.

7.0 RESULTS AND DISCUSSION

7.1 Analysis and Design of Light Vehicles For Rural Roads Considering Vibration With Bumps Up To 150 Mm and Its Performance

Analysis and Findings of Simulated Graphs Shown in Figs. 6.3(a) to 6.3(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.025 m, maximum amplitudes of vibration and its die out of time periods This situation creates comfort ride at low speed

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.025 m	25 km/hr	0.026 m	4 s
0.025 m	50 km/hr	0.027 m	2 s
0.025 m	75 km/hr	0.027 m	1.4 s
0.025 m	100 km/hr	0.026 m	1.1 s
0.025 m	125 km/hr	0.025 m	1 s

Table 7.1: Displacement vs. Time at 0.025 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Contd..**7.0 RESULTS AND DISCUSSION**

Analysis and Findings of Simulated Graphs Shown in Figs. 6.4(a) to 6.4(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.050m, maximum amplitudes of vibration and its die out of time periods are listed below in the Table 7.2.

This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.050 m	**25 km/hr**	**0.050 m**	**4 s**
0.050 m	**50 km/hr**	**0.050 m**	**2.2 s**
0.050 m	**75 km/hr**	**0.048 m**	**1.7 s**
0.050 m	**100 km/hr**	**0.047 m**	**1.5 s**
0.050 m	**125 km/hr**	**0.046 m**	**1.3 s**

Table 7.2: Displacement vs. Time at 0.050 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Contd..**7.0 RESULTS AND DISCUSSION**

Analysis and Findings of Simulated Graphs Shown in Figs. 6.5(a) to 6.5(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.075m, maximum amplitudes of vibration and its die out of time periods are listed below in the Table 7.3.

This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.075 m	**25 km/hr**	**0.075 m**	**4 s**
0.075 m	**50 km/hr**	**0.075 m**	**2 s**
0.075 m	**75 km/hr**	**0.075 m**	**1.5 s**
0.075 m	**100 km/hr**	**0.074 m**	**1.2s**
0.075 m	**125 km/hr**	**0.073 m**	**1 s**

Table 7.3: Displacement vs. Time at 0.075 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr.

7.0 RESULTS AND DISCUSSION

Analysis and Findings of Simulated Graphs Shown in Figs.6.6 (a) to 6.6(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.100m, maximum amplitudes of vibration and die out of time periods are listed below in the Table 7.4. This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.100 m	25 km/hr	0.080 m -0.001 m 0.003 m	3.9 s 4.1s 5.0s
0.100 m	50 km/hr	0.098 m -0.022m 0.015m -0.002m 0.000m	1.9 s 2.3 s 3.2 s 4.5 s 5.5s
0.100 m	75 km/hr	0.097 m	1.4 s
0.100 m	100 km/hr	0.082 m	1.2 s
0.100 m	125 km/hr	0.092 m	1.3 s

Table 7.4: Displacement vs. Time at 0.100 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Contd..**7.0 RESULTS AND DISCUSSION**

Analysis and Findings of Simulated Graphs shown in Figs. 6.7(a) to 6.7(e)

At different speeds of the 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr at bump height 0.125m; maximum amplitudes of vibration and die out of times are listed below in the Table 8.5.

7.0 RESULTS AND DISCUSSION

Table 7.5: Displacement vs. Time at 0.125 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.125 m	25 km/hr	0.137 m,	1.2 s,
		0.55 m,	-0.55s,
		0.025 m	2.5 sec
		0.000m	4.5sec
0.125 m	50 km/hr	0.131 m	1.4 s
		-0.016 m	1.6 s
		0.00 m	3.0s
		0.001m	5.0s
0.125 m	75 km/hr	0.126 m	4 s
		-0.040m	1.7s
		0.015m	2.7s
		-0.001m	3.7s
		0.001m	5.5s
0.125 m	100 km/hr	0.080 m	0.6 s
		-0.035m	1.6s
		0.015m	2.5s
		0.000m	3.6s
0.125 m	125 km/hr	0.095 m	0.7 s
		-0.040 m	1.6 s
		0.015m	2.6s
		-0.005m	3.5s
		0.000m	4.2s

This situation creates comfort ride at low speed and at larger speeds comfort in little less time.

Contd..**7.0 RESULTS AND DISCUSSION**

Analysis and Findings of Simulated Graphs Shown in Figs. 6.8(a) to 6.8(e)

At different speeds of the 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr at bump height 0.150m; maximum amplitudes of vibration and die out of times are listed below in the **Table 7.6**. This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.150 m	**25 km/hr**	**0.161 m**	**2 s**
0.150 m	**50 km/hr**	**0.151 m**	**1 s**
0.150 m	**75 km/hr**	**0.150 m**	**1.5 s**
0.150 m	**100 km/hr**	**0.150 m**	**4 s**
0.150 m	**125 km/hr**	**0.142 m**	**1.5 s**

Table 7.6: Displacement vs. Time at 0.150 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Contd..**7.0 RESULTS AND DISCUSSION**

7.2 Analysis and Design For Comfort Ride of 4-wheeled Vehicles Vibration on Rural Road Surface Considering Tyre Coefficient and To Reduce Climate Impact

Since vehicle passes through transient bumps, thus the comfort zone can be achieved by varying different tyre coefficients and speeds. **Figs. 6.12** to **Fig.6.31** show the various conditions of speeds and time of discomforts considering front and rear tyres damping coefficients.

Analysis and findings on fixed rear tyre damping coefficient (b_r) at 4kNs/m on time lags for achieving comfort zones across various speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h), illustrated in **Figs. 6.12 to 6.16**.

Contd..

7.0 RESULTS AND DISCUSSION

From Fig. 6.12, at a speed of 25 km/h with a rear tire damping coefficient of 4 kNs/m, the sprung mass reaches a peak displacement of +0.080 m at 2 seconds, drops to -0.015 m at 4 seconds, rises to +0.012 m at 5 seconds, oscillates with diminishing amplitudes, and stabilizes by 7 seconds. The comfort zone is achieved within 7 seconds, with a maximum upward displacement of +0.080 m.

From Fig. 6.13, at a speed of 50 km/h with a rear tire damping coefficient of 4 kNs/m, the sprung mass reaches a peak displacement of +0.100 m at 1.2 seconds, drops to -0.011 m at 4.2 seconds, oscillates with diminishing amplitudes, and stabilizes by 6 seconds. The comfort zone is achieved within 6 seconds, with a maximum upward displacement of +0.100 m.

Contd..**7.0 RESULTS AND DISCUSSION**

From Fig. 6.14, at a speed of 75 km/h with a rear tire damping coefficient of 4 kNs/m, the sprung mass reaches a peak displacement of +0.102 m at 0.9 seconds, drops to -0.044 m at 1.8 seconds, oscillates with diminishing amplitudes, and stabilizes by 5.5 seconds. The comfort zone is achieved within 5.5 seconds, with a maximum upward displacement of +0.102 m.

From Fig. 6.15, at a speed of 100 km/h with a rear tire damping coefficient **(**b_r**)** of 4 kNs/m, the sprung mass reaches a peak displacement of +0.091 m at 0.725 seconds. It then drops to -0.038 m at 1.6 seconds, rises to +0.015 m at 2.6 seconds, falls to -0.008 m at 3.7 seconds, and stabilizes by 4.2 seconds. The comfort zone is achieved within 4.2 seconds, with a maximum upward displacement of +0.091 m.

Contd..

7.0 RESULTS AND DISCUSSION

From Fig. 6.16, at a speed of 125 km/h with a rear tire damping coefficient of 4 kNs/m, the sprung mass reaches a peak displacement of +0.085 m at 0.6 seconds. It then drops to -0.035 m at 1.6 seconds, rises to +0.014 m at 2.5 seconds, oscillates further at 3.25 seconds, and stabilizes by 3.5 seconds. The comfort zone is achieved within 3.5 seconds, with a maximum upward displacement of +0.085 m.

Contd..
7.0 RESULTS AND DISCUSSION

Analysis and findings on fixed front tyre damping coefficient (b_f) at 8kNs/m on time lags for achieving comfort zones across various speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h), Illustrated in **Fig.6.17 to Fig.6.21.**

From Fig. 6.17, at a speed of 25 km/h with a front tire damping coefficient of 8 kNs/m, the sprung mass reaches a peak displacement of +0.077 m at 1.8 seconds, drops to -0.002 m at 3.5 seconds, and stabilizes by 5.5 seconds. The peak upward displacement of +0.077 m causes a single discomfort spike before vibrations cease within 5.5 seconds.

Contd..7.0 RESULTS AND DISCUSSION

From Fig. 6.18, at a speed of 50 km/h with a front tire damping coefficient of 8 kNs/m, the sprung mass reaches a peak displacement of +0.076 m at 1.0 second, drops to -0.002 m at 2.2 seconds, and stabilizes by 3.0 seconds. The peak upward displacement of +0.076 m causes a single discomfort spike before vibrations cease within 3.0 seconds.

From Fig. 6.19, at a speed of 75 km/h with a rear tire damping coefficient of 8 kNs/m, the sprung mass reaches a peak displacement of +0.076 m at 0.6 seconds, drops to -0.002 m at 1.3 seconds, and stabilizes by 2.5 seconds. The peak upward displacement of +0.076 m causes a single discomfort spike before vibrations cease within 2.5 seconds.

Contd..7.0 RESULTS AND DISCUSSION

From Fig. 6.20, at a speed of 100 km/h with a front tire damping coefficient of 8 kNs/m, the sprung mass reaches a peak displacement of +0.076 m at 0.55 seconds, drops to -0.002 m at 1.3 seconds, and stabilizes by 2.4 seconds. The peak upward displacement of +0.076 m causes a single discomfort spike before vibrations cease within 2.4 seconds.

From Fig. 6.21, at a speed of 125 km/h with a front tire damping coefficient of 8 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 0.5 seconds, drops to -0.001 m at 1.2 seconds, and stabilizes by 2.0 seconds. The peak upward displacement of +0.075 m causes a single discomfort spike before vibrations cease within 2.0 seconds.

Contd..7.0 RESULTS AND DISCUSSION

Analysis of fixed rear tire damping coefficient (br) at 12 kNs/m on time lags for achieving comfort zones across various speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h), as shown in **Figs. 6.22 to 6.26**.

From Fig. 6.22, at a speed of 25 km/h with a rear tire damping coefficient of 12 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 2 seconds and returns to 0.0 m by 4 seconds. The peak upward displacement of +0.075 m causes a single spike, leading to minor discomfort.

Contd..
7.0 RESULTS AND DISCUSSION

From Fig. 6.23, at a speed of 50 km/h with a rear tire damping coefficient of 12 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 1 second and returns to 0.0 m by 2.1 seconds. The peak upward displacement of +0.075 m causes a single spike, leading to discomfort.

From Fig. 6.24, at a speed of 75 km/h with a rear tire damping coefficient of 12 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 0.75 seconds and returns to 0.0 m by 1.5 seconds. The peak upward displacement of +0.075 m causes a single spike, leading to a higher degree of discomfort.

Contd..

7.0 RESULTS AND DISCUSSION

From Fig. 6.25, at a speed of 100 km/h with a rear tire damping coefficient of 12 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 0.6 seconds and returns to 0.0 m by 1.25 seconds. The peak upward displacement of +0.075 m causes a single spike, leading to significant discomfort.

From Fig. 6.26, at a speed of 125 km/h with a rear tire damping coefficient of 12 kNs/m, the sprung mass reaches a peak displacement of +0.074 m at 0.45 seconds and returns to 0.0 m by 1.1 seconds. The peak upward displacement of +0.074 m causes a single spike, resulting in severe discomfort.

Contd..

7.0 RESULTS AND DISCUSSION

Analysis of fixed Front Tire Damping Coefficient (bf) at 16 kNs/m for time lags to achieve comfort zones across speeds (25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h), as shown in **Figs. 6.27 to 6.31.**

From Fig. 6.27, at a speed of 25 km/h with a front tire damping coefficient of 16 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 2.5 seconds and returns to 0.0 m by 4.0 seconds. The peak upward displacement of +0.075 m causes a single spike, leading to discomfort.

Contd..**7.0 RESULTS AND DISCUSSION**

From Fig. 6.28, at a speed of 50 km/h with a front tire damping coefficient of 16 kNs/m, the sprung mass reaches a peak displacement of +0.075 m at 1.1 seconds and returns to 0.0 m by 2.2 seconds. The peak upward displacement of +0.075 m causes a single spike, leading to a higher degree of discomfort.

From Fig. 6.29, at a speed of 75 km/h with a front tire damping coefficient of 16 kNs/m, the sprung mass reaches a peak displacement of +0.074 m at 0.75 seconds and returns to 0.0 m by 1.75 seconds. The peak upward displacement of +0.074 m causes a single spike, leading to a higher degree of discomfort.

Contd..**7.0 RESULTS AND DISCUSSION**

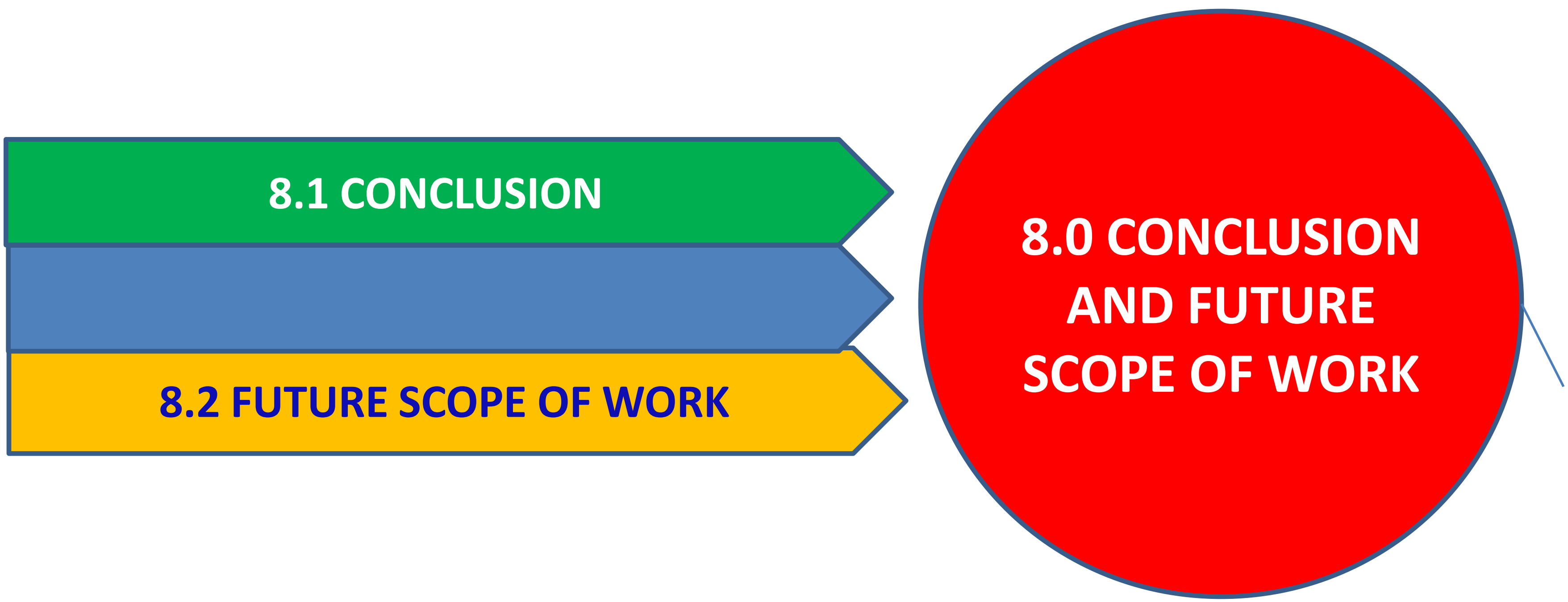
From Fig. 6.30, at a speed of 100 km/h with a front tire damping coefficient of 16 kNs/m, the sprung mass reaches a peak displacement of +0.074 m at 0.6 seconds and returns to 0.0 m by 1.5 seconds. The peak upward displacement of +0.074 m causes a single spike, resulting in severe discomfort.

From Fig. 6.31, at a speed of 125 km/h with a front tire damping coefficient of 16 kNs/m, the sprung mass reaches a peak displacement of +0.073 m at 0.5 seconds and returns to 0.0 m by 1.25 seconds. The peak upward displacement of +0.073 m causes a single spike, resulting in severe discomfort.

Contd…**7.0 RESULTS AND DISCUSSION**

Based on the findings in the analysis and design for a comfortable ride on rural road surfaces, focusing on the tire coefficient to minimize climate impact, it is observed that the vehicle model, when encountering transient bumps, stabilizes within comfort zones in 3.00 to 5.5 seconds.

This is under conditions with damping coefficient parameters set to (br) = 4 kNs/m to 12 kNs/m, and vehicle speeds ranging from 75 km/h to 125 km/h. These results are consistent across the sprung mass and other fixed parameters.


8.1 CONCLUSION
8.2 FUTURE SCOPE OF WORK
8.0 CONCLUSION AND FUTURE SCOPE OF WORK

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

8.1 Conclusion

Drawing upon the insights and discourse presented in the Analysis & Design for Ensuring a Comfortable Ride of 4-Wheeled Vehicles on Rural Road Surfaces, the simulation analysis of the model encompasses a comprehensive examination under diverse conditions. This includes an exploration of both fixed and variable parameters.

The Evaluation of a Vehicle's Response under the Conditions of Consistent Tyre Stiffness Coefficients, With Variations in both Speed and Road Bumps.

Contd..

8.1 CONCLUSION

The vehicle model was examined within the range of speeds from 25 km/h to 125 km/h on rural roads, with sprung mass displacement ranging from 0.025 m to 0.150 m, the ensuing results highlight the following predominant outcomes.

a) With bumps 0.025 m to 0.075 m, vibrations of vehicle are found die out with first spike in 2 seconds to 4 seconds at a speed of 25km/hr to 125 km/hr. This situation creates discomfort to the rider at low speed and at larger speeds too.

b) With bumps 0.100m to 0.125m, vibrations of vehicle are found die out in harmonic condition in 4 sec to 5 seconds at speeds from 25km/hr to 125km/hr. This situation creates comfort to the rider at 50km/hr to 75km/hr speeds and at lower and larger speeds gives little discomfort.

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

Contd..8.1 Conclusion

c) With bumps 0.150m, vibrations of vehicle are found die out in harmonic condition in 3 seconds to 3.5 seconds at speeds of 25km/hr and 50 km/hr. This situation creates comfort to the rider at 25km/hr to 50 km/hr vehicle speeds and at larger speeds between 75km/hr to 125km/hr no comfort situation is seen.

The evaluation of a vehicle's response to transient bumps and damping coefficients for both rear and front tyres, was examined within the range of tyre stiffness coefficients 4 kNs/m to 16 kNs/m with different speeds 25 km/h, 50 km/h, 75 km/h, 100 km/h, and 125 km/h.

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

Contd..8.1 Conclusion

The attainment of the vehicle's comfort zone on rural roads is elucidated through the following outcomes:

a) The maximum displacement is (+/-) 0.102 m when the model passes through vibration transients and stabilizes within 5.5 seconds, providing comfort at a speed of 75 km/h with a rear damping coefficient of 4 kNs/m.

b) The maximum displacement is (+/-) 0.077 m when the model experiences a spike before stabilizing within 5.5 seconds, causing discomfort even at a speed of 25 km/h with a front damping coefficient of 8 kNs/m.

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

Contd..**8.1 Conclusion**

c) The maximum displacement is found (+/-) 0.075m if model is not passing through vibration transient but gone to spike before it becomes under normal condition within 4.0 seconds and it creates higher degree of 146 discomfort even if speed is kept 25 km/h and rear damping coefficient considered 12 kNs/m.

d) The maximum displacement is found (+/-) 0.075m if model is not passing through vibration transient but gone to spike before it becomes under normal condition within 2.2 seconds and it creates severe discomfort even if speed is kept 25 km/h and front damping coefficient considered 16 kNs/m.

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

Contd..8.1 Conclusion

From above, it can be concluded that for rural road conditions where roads geometry cannot be ignored with bumps / pot-holes to the order of 100mm (0.100m) for sprung-mass displacement, the rural road vehicle should be designed to have tyre damping coefficient ≥4 kNs/m, which can attain speed ≤ 75km/h.

It is also seen that the sprung-mass displacement value decreases as the damping coefficient is increased. For a suspension damping coefficient ≤ 8 kNs/m, the sprung-mass displacement is found higher and vehicle speed can be kept above >50 km/h, but ≤ 75 km/h.

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

Contd..8.1 Conclusion

Therefore this research has the potential to contribute significantly to the development of cost-effective and environmentally friendly vehicles tailored for rural road transportation. The focus is particularly relevant in the context of India, which boasts the world's second-largest road network primarily situated in rural areas. The aim is to provide insights and solutions for designing vehicles that are both economically viable and environmentally sustainable, addressing the unique challenges and opportunities present in the rural transportation sector of India.

8.2 Future Scope of Work

Hinged arm suspensions, a longstanding concept, have gained significance with the advent of active computerized control in vehicles. In this study, a model devoid of active elements but comprising two sets of springs and dashpots was utilized. The successful compilation of the Bond graph using the Symbol Shakti software for this proposed model confirms a logically sound representation, though it does not guarantee the creation of a substantively valuable model. Further endeavors in that direction may be required for verification.

8.0 CONCLUSION AND FUTURE SCOPE OF WORK

Contd..8.2 Future Scope of Work

A more systematic investigation can be conducted to assess the advantages of hinged arm suspension under diverse conditions, utilizing the established model. This research can be expanded by incorporating control systems into the Bond graphs model, thereby progressing towards the development of an active suspension model.

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THANK YOU