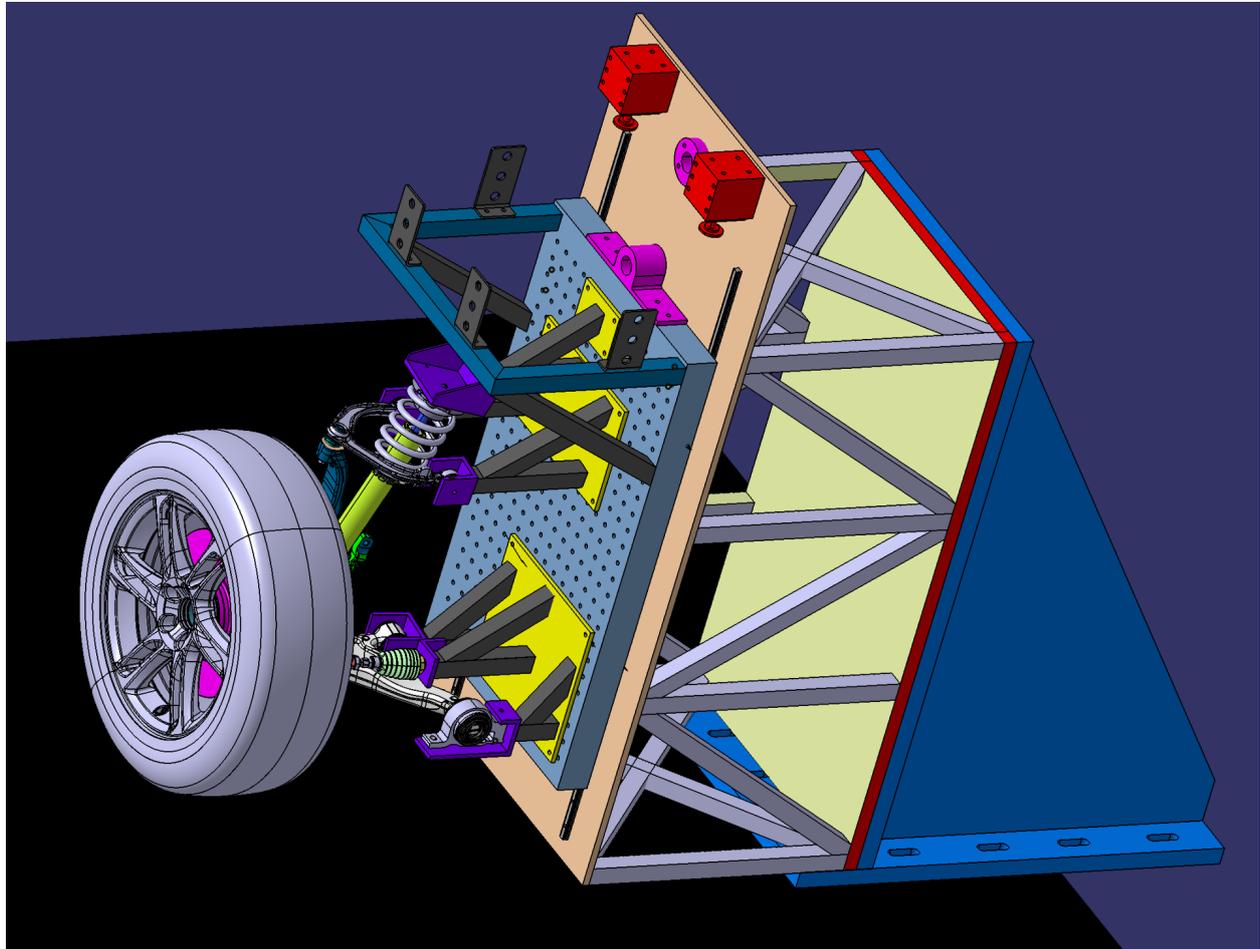




CHALMERS
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Quarter car rig for software function development of fully active suspension and continuously controlled dampers

Master's thesis in Mobility Engineering

SUYASH YASHWANT WAGH

DEPARTMENT OF MECHANICS AND MARITIME SCIENCES (M2)

CHALMERS UNIVERSITY OF TECHNOLOGY

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MASTER'S THESIS 2023

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Cover: Quarter Car model of a modular Testrig built in Catia showing SLA suspension attached to unsprung mass assembly which will move in vertical direction with the help of Linear motion guides attached to rigid structure.

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Abstract

Electric vehicles require a delicate balancing act by manufacturers. When engineers try to increase the ride comfort, ease of handling can be hurt and vice versa. Engineers in the automotive industry use computer-aided simulation extensively to find the right balance between comfort and handling. However, applying that requires training and extensive knowledge to get the best performance and results. Nowadays, fully active suspension (FAS) is a solution which can improve balance between ride comfort and handling. Currently, there's lot of competition going on in an automotive market, especially because of electric vehicles. Checking the effect of FAS in the early development phase of the car will be helpful to achieve the right balance between ride comfort and handling.

Early in the vehicle development projects, real vehicle prototypes are missing or lacking correct suspension geometry and/or FAS and CCD actuators. Virtual prototypes (simulation models) are also not available. Like the quarter car model, a test rig that takes approximately 1/4 of the sprung mass and unsprung mass where different suspension geometries can be used. Such test rig will be a good initial approach to understand the behavior of the vehicle. Therefore, this thesis involves the work to design a modular test rig for a quarter car that can be used on one poster in an existing 4-poster rig. The motion of the poster will be such that it will consider only the vertical dynamics of a car.

In the first step, a conceptual model of a rig has been designed by combining a tire, a suspension geometry, Linear Motion Guide (LM Guide), and all the necessary required components to fix the testrig to the ground. After performing analysis on Tire Testrig in Adams Car and considering the complexity, it has been decided to keep a tire stationary instead of rolling. Considering the motion dynamics, caged type LM guide has been chosen which can take optimum forces in radial direction and has a better rolling resistance than other models. Then, a CAD geometry of a test rig has been developed using Catia-Teamcenter Integration tool with the aim to build a rig that can be used for various suspension geometries. The modularity of a testrig is required as various platforms have different mounting points, characteristics of a wheel, etc. While performing different load cases and test run the testrig, various instrumentation as well as production sensors will be used to measure the different signals coming out from the test run.

The test rig which involves FAS and CCD can be validated either on a simulation level or by comparing with passive components on a 4-poster test rig. A simulation model which can exactly emulate the behavior or the functions of the physical quarter car testrig has been developed with the help of Adams software. Results from the simulation model accurately represents the real-world phenomenon.

There are many benefits of having a Quarter Car Testrig. Some of them involves tuning the controller of an active damper, to iterate springs with different stiffness values to achieve the required targets. Additionally, we can fix the inhouse built software package on the testrig to validate the system. In such a way we can have more mature products during development phase through various iterations, and testing. This might help us becoming more decisive in a way and reduce little bit of time required considering the overall planning.

Keywords: lorem, ipsum, dolor, sit, amet, consectetur, adipiscing, elit, sed, do.

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Suyash Yashwant Wagh, Gothenburg, June 2023

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

CCD	Continuously Controlled Damper
FAS	Fully Active Suspension
FF	Front Front
FLF	Front Lower Front
FLR	Front Lower Rear
FR	Front Rear
FUF	Front Upper Front
FUR	Front Upper Rear
LM	Linear Motion
MPU	Motor-pump Unit
RF	Rear Front
RL	Rear Lower
RLF	Rear Lower Front
RLR	Rear Lower Rear
RR	Rear Rear
RU	Rear Upper
RUF	Rear Upper Front
RUR	Rear Upper Rear
SLA	Short Long Arms
VCC	Volvo Car Corporation
DoF	Degree of Freedom
mm	Millimeter

Nomenclature

m_s	Sprung Mass
m_u	Unsprung Mass
k_s	Spring Stiffness
k_t	Tire Stiffness
z_s	Sprung mass travel
z_u	Unsprung mass travel
z_r	Road travel
F_z	Vertical wheel force
I_{xx}	Moment of inertia around the x axis as the object rotates around the x axis
I_{yy}	Moment of inertia around the y axis as the object rotates around the y axis
I_{zz}	Moment of inertia around the z axis as the object rotates around the z axis
M_{Total}	Total Corner mass of a car including unsprung mass
$X_{Unsprung}$	Unsprung mass for an axle
$M_{Unsprung}$	Unsprung mass for a corner
M_{Max}	Max sprung mass per corner

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1

Introduction

1.1 Background

Electric vehicles require a delicate balancing act by manufacturers. When engineers try to increase the ride comfort, ease of handling can be hurt and vice versa. Engineers in the automotive industry use computer-aided simulation extensively to find the right balance between comfort and handling. However, applying that requires training and extensive knowledge to get the best performance and results. Even with simulation tools, engineers still need to build a strategy that defines metrics and provides a comprehensive insight into performance. Nowadays, fully active suspension (FAS) is a solution which can improve balance between ride comfort and handling. FAS can add and dissipate energy in the suspension, pushing or pulling independently of deformation direction. The FAS system will adapt to the road conditions and the driver input with the aim to minimize the movement of the vehicle body which can eliminate motion sickness.

Checking the effect of FAS in the early development phase of the car will be helpful to achieve the right balance between ride comfort and handling. Similar to the quarter car model, a test rig that takes approximately 1/4 of the sprung mass and unsprung mass where different suspension geometries can be used. Such test rig will be a good initial approach to understand the behaviour of the vehicle. The test rig cannot model the whole dynamic behavior of the vehicle. However, it is envisioned to be sufficient for benchmarking and describing the fundamental conflict in the suspension system, i.e., the conflict between ride comfort and road holding.

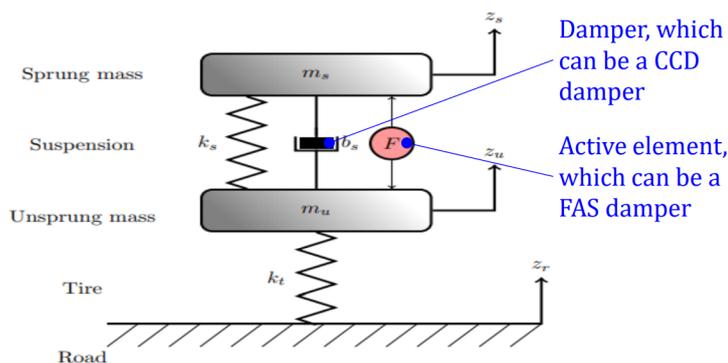


Figure 1.1: Active quarter car suspension model

1.2 Problem Statement

The overall problem which motivates this work is that, early in the vehicle development projects, real vehicle prototypes are missing or lacking correct suspension geometry and/or FAS and CCD actuators.

Virtual prototypes (simulation models) are also not available, because suppliers of FAS and CCD cannot deliver accurate enough virtual samples (submodels).

The envisioned solution for present thesis work is therefore a test rig. Because parts from different suppliers will be compiled, it is motivated that the vehicle manufacturer, as opposed to any of the suppliers, develops and uses such rig.

1.3 Objective

This thesis involves the work to design a modular test rig for a quarter car that can be used on one poster in an existing 4-poster rig. The purpose of the rig is to do software (SW) function development of the FAS system and continuously controlled dampers (CCD). The aim is to build a rig that can be used for various suspension geometries, such as short-long-arm (SLA) and MacPherson. It is also to propose a metric to rate isolation between wheel input and body movement by implying various load cases on the wheel in a test rig along with the sensors to fetch data to be able the perform analysis. The motion of the poster will be such that it will consider only the vertical dynamics of a car. The test rig which involves FAS and CCD can be validated either on a simulation level or by comparing with passive components on a 4-poster test rig. Different methods to capture the responses of the sprung quarter body will be explored and tested.

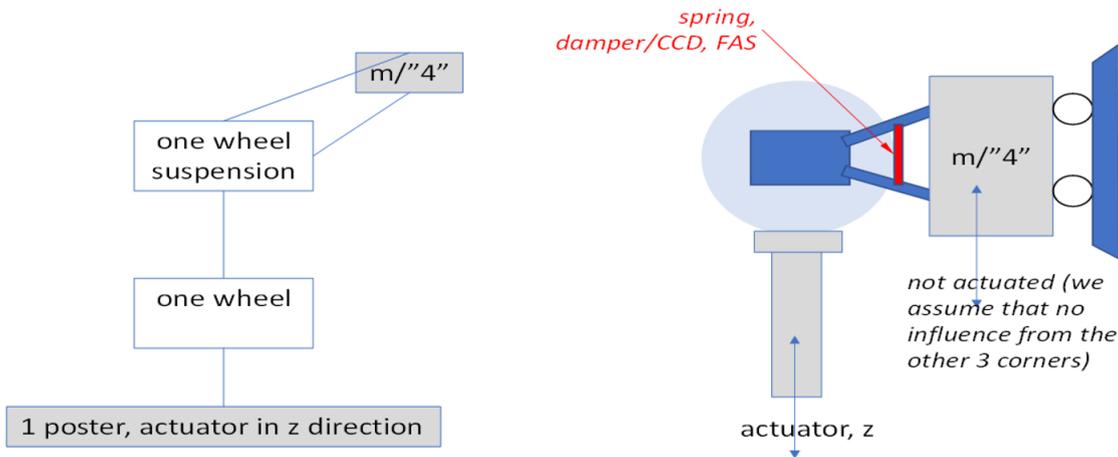


Figure 1.2: Conceptual drawing of the quarter car test rig

1.4 Deliverables

The main deliverables of the thesis are as given below:

- Design of rig (geometry models)
- Simulation model of rig
- Stretched target: Built and test run rig

1.5 Limitations

- No Anti-RollBar to be considered in the rig
- Rig to fit with independent suspension
- No subframe , E-motor, etc
- The rig will not emulate the effect from lateral and longitudinal tyre forces (only vertical)
- Zero steering rack position
- No solutions of how to connect the active suspension actuators electronically

1.6 Methodology

1. Literature study about the development of fully active suspension and semi-active dampers in the overall suspension development and understanding their effect on comfort and handling considering overall vehicle development.
 - Especially, the usage of the rig should be positioned side-by-side with the novel hard-point and bushing development methods which is under development in the project <https://research.chalmers.se/en/project/9987>.
 - Gather an understanding and exploration of various sensors used to control such systems, so that the computer and control solution can be represented in the rig.
2. Investigate different load cases that can be implemented on wheel/tire in the test rig, sensors needed for the test rig to be able to perform analysis of logged data which correlates wheel input and body motion in a vertical direction.
3. Design a quarter car test rig that can be integrated in the 4-poster rig available at VCC. Test rig fixture should be modular to allow independent suspension systems for one corner, e.g., SLA, MacPherson, 5-link, integral link, etc.
4. Validation of the model with the help of simulation data from Adams or kinematic simulation on Catia.
5. Create a metric of wheel input to (force: F_z) Road Variation to rate the isolation of wheel input and body movement based on rig input loads and signals from rig set up considering both primary and secondary rides.
6. Build a virtual prototype (simulation model) of the rig. Use the virtual prototype, before manufacturing the hardware parts, to check that it is credible that the rig will be used before manufacturing rig parts.

2

Theory

2.1 Four Link Suspension System (SLA)

The four link suspension system is similar to the double wishbone suspension system. Another name for this suspension system is short long arms suspension (SLA). The upper control arm is common in both the systems but the lower control arm is split into two separate two point links as shown in figure ?? . The degree of freedom is constrained as follows:

- Each two point link constrains one degree of freedom. Thus, 3 links constrain 3 DoF
- The upper control arm constrains 2 DoF

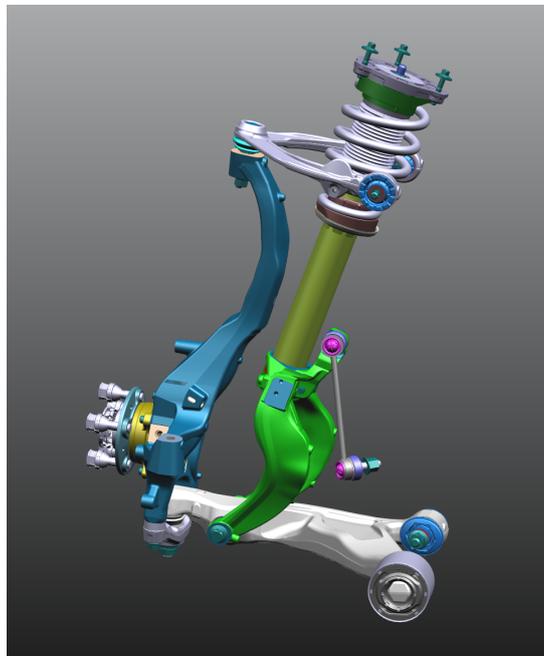


Figure 2.1: Four Link Suspension System

This suspension system can be more compact than a double wishbone system.

2.2 Five Link Suspension System

The five-link suspension is an independent type of multilink suspension system. This suspension is widely used in a modern premium or heavy cars for their good performance and cost. The design of the five-link suspension is derived from the double wishbone suspension system (see figure 2.2). The upper and the lower wishbones are split into two-point separate links. The idea of multilink suspensions is to use several links to ensure better control and tuning of the kinematic parameters.

Due to larger number of design parameters, it has the capability of fulfilling both complex kinematic and dynamic requirements imposed on the suspension systems. It is however, more difficult to integrate than any other suspension mechanisms, due to its general spatial configuration.

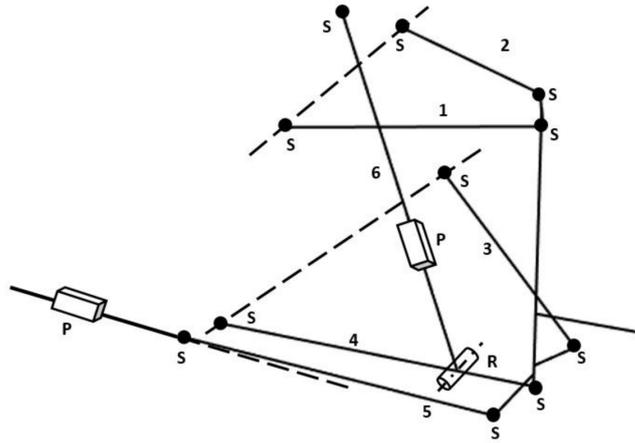


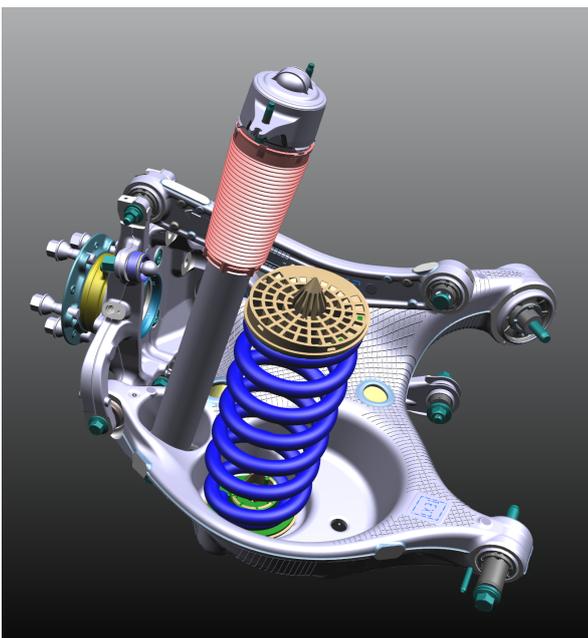
Figure 2.2: Five Link Suspension System

Commonly, these suspensions are used on a unibody vehicle which makes the subframe to be mandatory. The subframe is attached to the unibody through bushings to reduce the transmission of noise and vibrations from the wheel to the body.

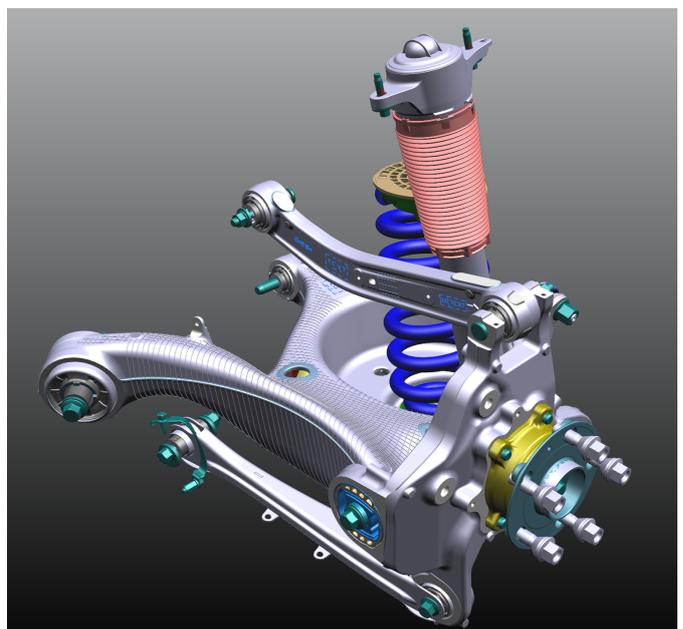
2.3 Integral Link Suspension System

Another form of the multilink suspension is the Integral link suspension. The suspension system comprises of a toe link to control the toe angle change and a camber link to control the camber angle change. What makes it unique is the presence of the trapezoidal shaped lower platform which poses a challenge in modelling the suspension system. The degree of freedom for the integral link is constrained as follows:

- Two point link constrains one degree of freedom. Thus, the toe and camber link each constrain 1 DoF
- The lower platform is a special case since it is connected to the integral link. This system constrains 3 DoF



(a) Front Isometric View



(b) Rear Isometric View

The integral link helps to control the torques on the knuckle during acceleration and braking and thus provides a high windup stiffness. The integral link suspension can be quite complex to understand and tuning the system can be difficult.

2.4 What is FAS?

FAS is that kind of suspension and damping systems in which the force at each wheel could be individually regulated. Hence, improving both the ride comfort and driving dynamics. In today's world of Vehicle Dynamics, there are various demands such as high ride comfort, full anti-roll and anti-pitch control even at highest accelerations, high interlocking with the air suspension, low energy requirements and flexible package. These objectives could be achieved by combining an air spring with an active hydropneumatic suspension. The air spring carries the basic load of the vehicle and regulates its level. If necessary, the hydropneumatic suspension generates dynamic forces that are superimposed on the forces of the air spring and are able to actively support and dampen the vehicle body.

The hydropneumatic suspension is identical at each wheel, figure 2.3. The damper has a continuously adjustable damping valve and a hydraulic accumulator on both working chambers. Over the hydraulic lines, each damper is connected to a Motor-pump Unit (MPU). The MPU can pump oil from one side of the damper to the other side so that a pressure drop is generated inside the damper, creating an active force. In the event of a disturbing excitation from the road, the hydraulic accumulators are working as spring elements and the MPU is decoupled from jerky movements of the wheel.

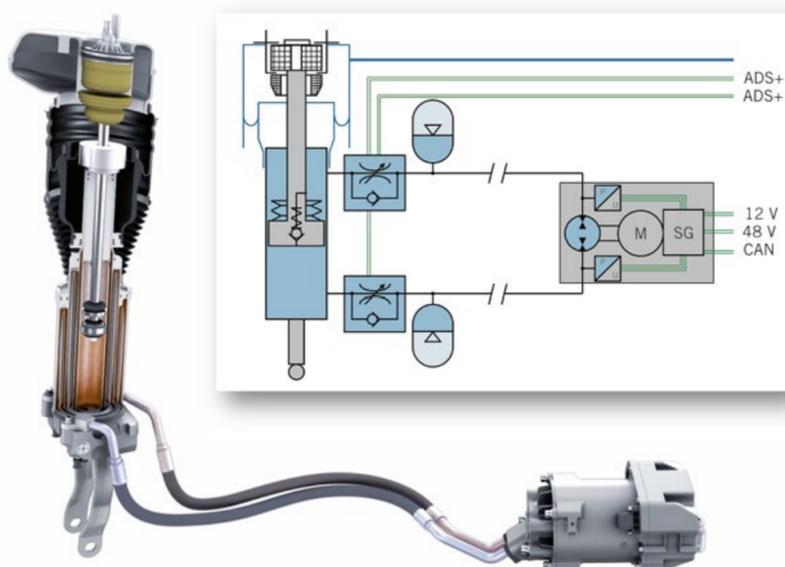


Figure 2.3: Schematic Representation of Fully Active Suspension

As shown in figure 2.4, the progressive characteristics of the hydraulic accumulators are ideally suited to the properties of the electric motor. In the case of low forces, the hydraulic accumulators have a low stiffness and offer a very good decoupling. However, the forces can be rapidly modulated because the electric motor can reach high speeds at low loads, figure 2.4 (operation point 1). With increasing forces, the stiffness is increasing simultaneously and thus compensates the declining dynamics of the electric motor, figure 2.4 (operation point 2). The power hyperbolas of the motor are thus fully utilized.

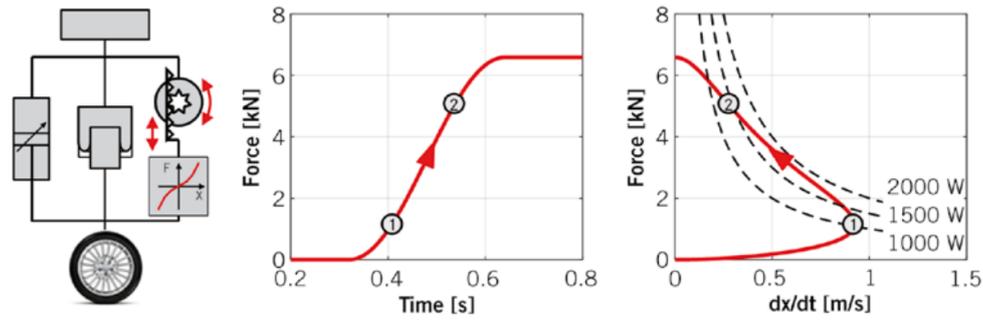


Figure 2.4: Progressive spring characteristic of the hydraulic accumulators

The electrohydraulic system offers additional advantages compared to electro-mechanical systems: The MPUs are free from forces to the outside, so they can be positioned freely in the vehicle and can be suspended very softly. The hydraulic system provides a very high force and power density, integrated lubrication and cooling, and simple overload protection.

As seen in figure 2.5, the hydraulic lines are directly assembled to the MPUs and the suspension struts without intermediate support. The MPUs are activated by the main control unit which also takes over the control of the damping valves and the air spring compressor. The hydraulic system is filled once and is then maintenance-free. There is only a single moving seal connecting it to its environment, namely the piston rod seal in the damper. The overall system has an additional weight of around 50 kg over the air spring suspension.

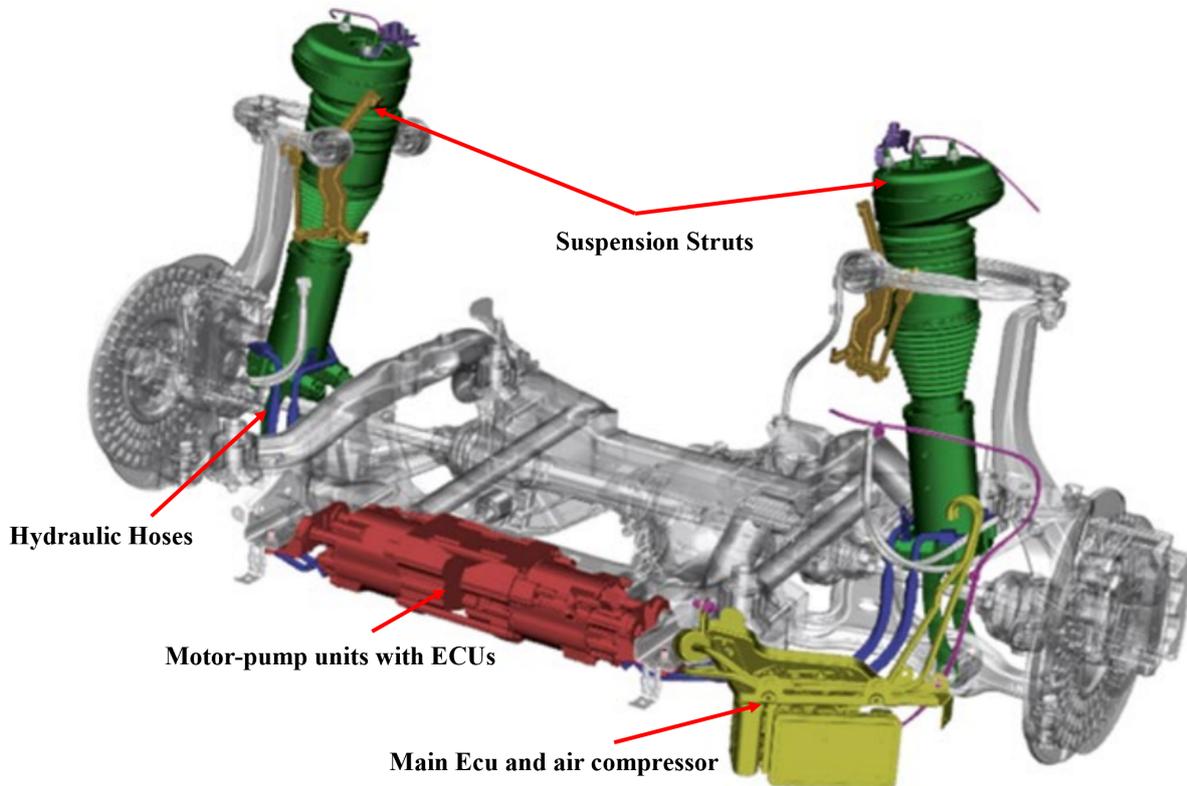


Figure 2.5: Fully Active Suspension System at the Front Axle

2.4.1 Active Suspension system vs Passive Suspension System

[Active Suspension system]

- Good Tunability, so we can achieve different attributes with the same HW
- It can adapt to different operating conditions
- It can adapt to driver inputs
- It can adapt to road conditions (Less primary/secondary ride tradeoff)
- It can achieve better attributes (Ride and handling)

[Passive Suspension system]

- It has limited bandwidth compared to HW solutions
- Greater Cost
- If not properly done, it can create a digital ride experience (The body motion is not predictable for the passengers)

2.5 What is Continuously Controlled Dampers (CCD)?

Dampers control the movement that's produced by vehicle's springs when you're on a bumpy road or brake suddenly. They soften this impact by converting kinetic energy from the springs into heat energy that dissipates through the hydraulic fluid. Important function of a damper is to ensure a tire is always in contact with the surface of the road. Additionally, dampers dissipate energy whereas springs just store energy. Continuously Controlled Dampers can control how much energy to dissipate for a given damper velocity, Figure 2.6.

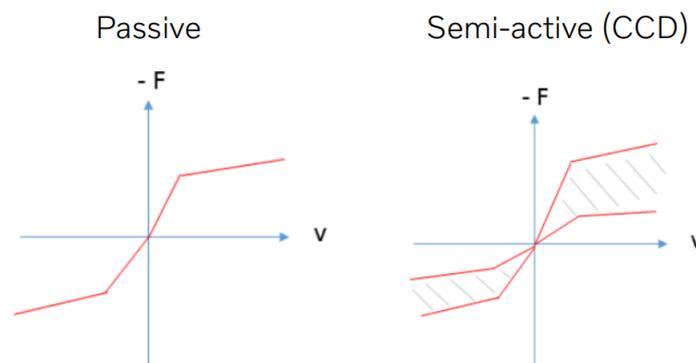


Figure 2.6: Force Vs Velocity Plot

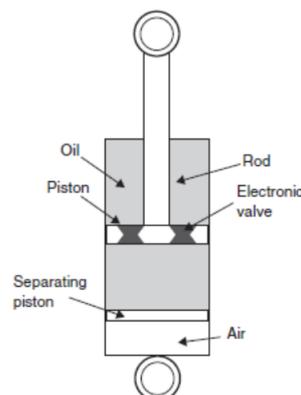


Figure 2.7: Schematic Representation of an Electrohydraulic Shock Absorber

In case of dampers, force is dependent on velocity of the body relative to the wheel. As seen in figure 2.6, passive dampers can only counteract damper motion. Whereas, semi active dampers can continuously adjust force levels by current modulation by changing flow resistance in the damper by electronic valve, Figure 2.7.

2.6 Four-Poster Test Rig

The four-post test rig can be used to investigate the vertical dynamics of the full vehicle and its suspension systems. We can then plot the time-based results and study them in the frequency domain to understand the various ride modes and their respective damping levels. The investigation will also help us learn more about the influences of the vehicle's vertical dynamics effects on handling behavior by studying the system's dynamic responses, which includes:

- Front-to-rear modal balance
- Suspension-to-body transfer function gain and phase
- Suspension-to-tire transfer function gain and phase
- Tire contact patch vertical load variation

The test involves assembling a standard full-vehicle model to a four-post test rig. The test rig is defined by four parts representing the tire pads that support the vehicle. The tire pads are constrained to move only in the vertical direction and a displacement actuator, or motion controller, controls their vertical motion. For this study, we are planning to use one of the four posters as it is a quarter car testrig consisting only one corner of the car.

Model	-086-45DI-A1W (MOOG)
Serial	101
Date	MAR 2012
Area	2799 MM ²
Stroke	254 MM
Force	50 KN
Max. PR	210 BAR

Table 2.1: Specifications

3

Overall Vehicle Development Process

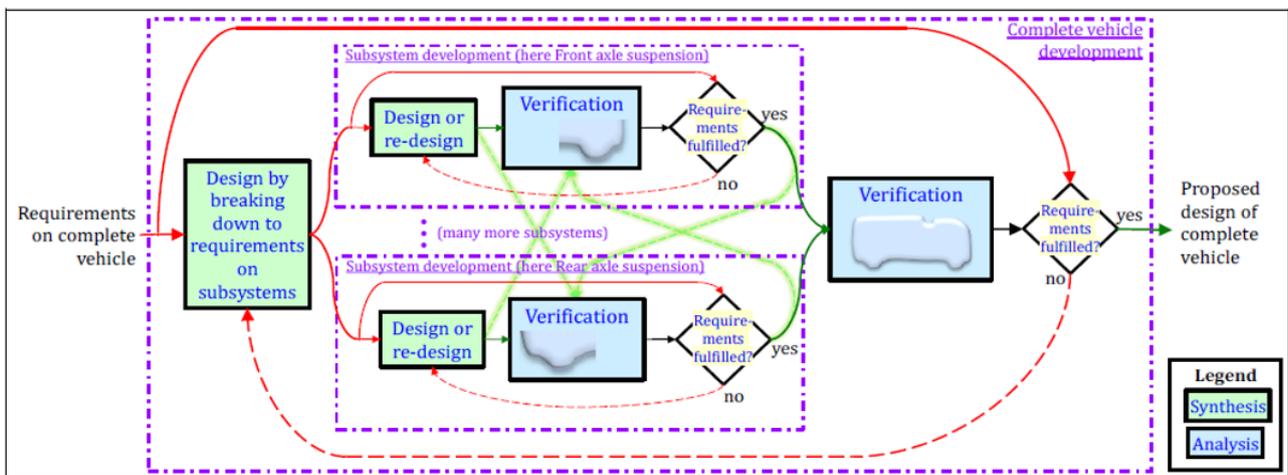


Figure 3.1: How V-model with development loops can be applied for a product decomposed in multiple subsystems

This figure 3.1 represents the overall vehicle development program. The requirements on complete vehicle are broken down into requirements on subsystems such as front axle suspension, rear axle suspension, steering, etc. After breaking down to requirements on subsystems, subsystem development process involves a design process followed by a verification on different simulation softwares or a physical testrig. Then, we check if the requirements on subsystems are fulfilled or not.

- If yes, a complete vehicle analysis is performed to check whether we have achieved the proposed design of a complete vehicle or not.
- If no, we look at the requirements again and redesign the product followed by the exact subsystem development process to see if it is ready to put on a complete vehicle.

It shows how the development loops can be applied for a product decomposed in multiple subsystems.

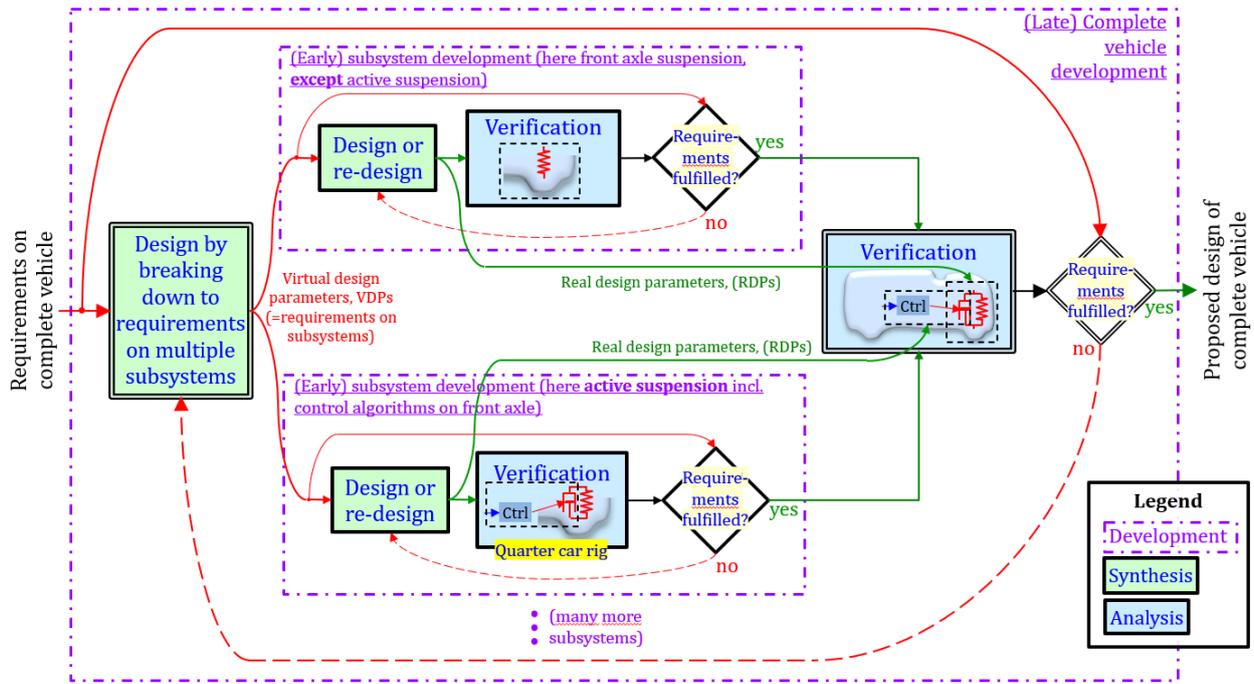


Figure 3.2: Parallel Development

The above figure 3.2 represents a complete process, including internal design loops for each subsystem considering this thesis project in mind. It basically shows the parallel development plan for various subsystems. This project demonstrates that, the front axle suspension system and the active suspension can be developed in parallel.

The geometric parameters are not usually available for the in-development platform. To bypass this hurdle we can use the data/ geometric parameters from the in-production platform and tune the active suspension parameters. This parallel design methodology gives us the upper hand by enabling us to work upon the active suspension even if the new front axle suspension design is not ready, resulting into more efficient and faster drawing-board to production processes.

3.1 Active Suspension

In the future, active suspension systems will replace conventional passive suspension systems since vehicle stability and passenger safety are high concerns for car designers. Cars tend to become smaller (SMART) and incorporate a higher center of gravity (SUV) and a reduced footprint, which increases the need for a suspension system with a rigid response when driving into turns while absorbing the road irregularities when driving under low-yaw circumstances (relatively straight). Currently, commercial systems consist of hydraulic or pneumatic actuators which offer a high force density and ease in design due to the commercial availability of the various parts.

The main purpose of this quarter car HIL rig is to optimize ride comfort and vertical dynamics plays a crucial role. Longitudinal and Lateral forces are a byproduct of any vertical motion such as small bump or droop. Therefore, road input and displacement are the important parameters to analyse under the use of this test rig.

3.1.1 Active Damper

In the damper, a differential cylinder converts the pressures generated from the damping valves and the MPUs into actuating forces. The two hydraulic accumulators are positioned concentrically

around the cylinders. The accumulators consist of so-called Gasbags that are filled with nitrogen, figure 3.3 Gasbags are sealed bags, which have a 0.2 mm thick foil, made of one layer of high-strength aluminum and several layers of Polyamide (PA). The aluminum provides a high gas sealing, the plastic is used for mechanical and chemical stability. The adjustable damping valves and the line connections are integrated into the shock absorber's foot.

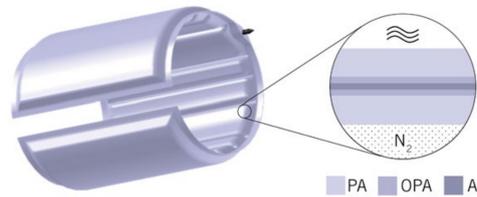


Figure 3.3: Gasbags as hydraulic accumulators

The active damper has the same spring travel as the semi-active damper of the air suspension. Due to the progressive spring characteristic of the hydraulic accumulators the buffers and pull stops are only required for very big wheel movements and have no influence on ride comfort. The mean system pressure is 30 bar, only slightly higher than for a conventional single-tube shock absorber, so friction is on a similarly low level. A pressure valve in the working piston ensures that the pressure is limited in the case of heavy compression. High-strength steel tubes enable a compact design and moderate weight.

3.1.2 Motor-Pump Unit

The MPU consists of three components: A bi-directional hydraulic pump, a brushless canned electric motor with single-tooth winding and a directly flange-mounted control unit, 3.4. The power- and the logic-boards are supplied by the 48-V on-board electrical system. Only the CAN transceiver is connected to a 12 V system and is therefore galvanically decoupled. The MPE generates volumetric flow rates of over 25 l/min and pressure differentials of 130 bar with a total efficiency of up to 70%. The design as a canned motor does not have any movable seals to the environment. The unit monitors itself, has its own sensors for pressure, temperature and rotary position and it communicates with the main chassis control unit via a CAN bus. A housing made of aluminum allows for good heat dissipation at low weight. Power is supplied via an integrated device connection and other electrical lines are routed via a six-pin plug. Apart from the cable lengths, all MPUs in the vehicle are identical.

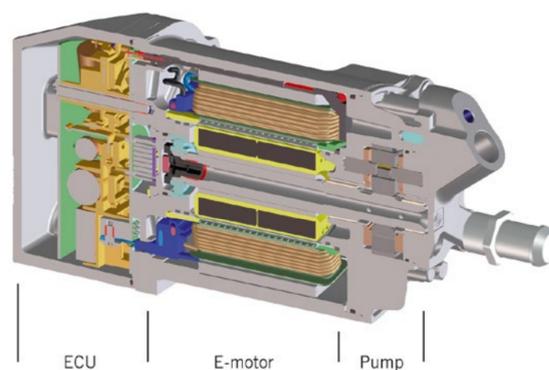


Figure 3.4: Motor-Pump Unit

3.1.3 Functions

The special feature of the active suspension systems is that the force on each wheel can be individually regulated. Therefore, the overall system has twice as many degrees of freedom as an active roll stabilization. While the latter only affects the roll angle and the roll moment distribution. FAS also acts on lifting and pitching of the vehicle body. The skyhook damping is therefore fully active and the vehicle has no pitching during braking and acceleration

3.1.4 Driving Behaviour

In this system, the active control can operate between 0.5 and 5 Hz. Thus, the driving dynamics and body movements can be influenced to a great degree. The behavior at higher frequencies, however, is dominated by the mechanical properties. Significant progress can be made mainly due to the good decoupling of the hydraulic accumulators at low loads, the low friction and the softer head bearings. Furthermore, the mechanical damping valves can now be controlled separately in the pushing and pulling directions.

4

Quarter Car Dynamic HIL Rig

The focus of this thesis is to build a quarter car dynamic HIL rig that could be used during development phase to study the influence of the system components, controllers on ride comfort and driving dynamics. Additionally, substantial part of the work is carried out on MSC Adams, which is a multi-body simulation software developed by Hexagon. It offers a convenient environment for analysing different automotive systems. A SLA suspension model is built which is used as a basis model, and is worked upon into a rig model to study the influence of different suspension components. This chapter covers the the flow of the executed work.

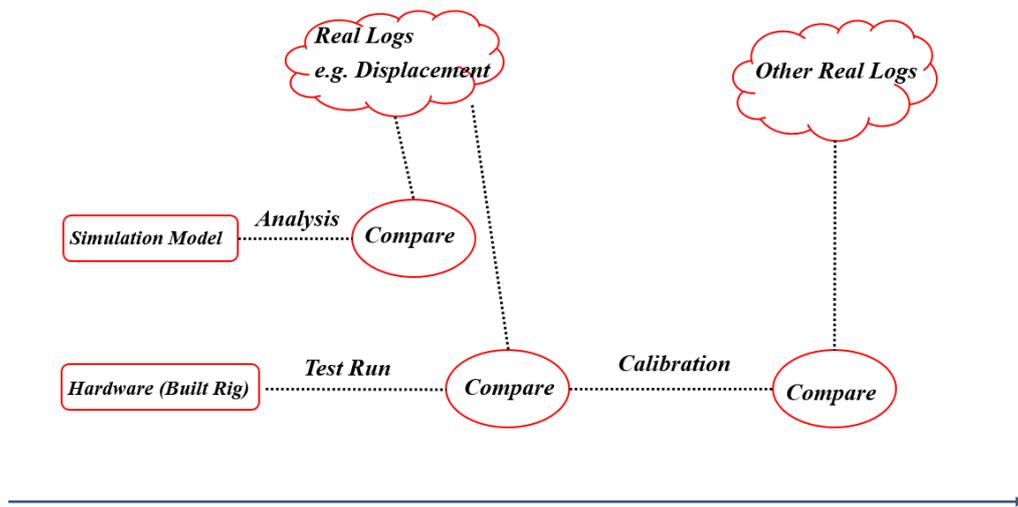


Figure 4.1: Flow chart of methodology

Figure 4.1 describes the work flow of the thesis. Once the problem was identified and henceforth the solution was found, data collection and analysis is considered as crucial part of any operational development. Further, a brainstorming session was done on various alternatives for a suspension systems and the study begins by considering SLA suspension with FAS system for the first test run as there is already a test vehicle in development phase with FAS system that could help us compare the results. Also, components required for this suspension system are easily available. It is important to understand what the influencing parameters are and how are these parameters influencing the design and performance. It is possible to develop a good suspension system which can be tuned for a large load range and required wheel travel if the sensitivity of the parameters can be understood. In this way, once the parameters are understood then we can make products more mature during development phase and to do this, certain test set up is required which was build with the help of certain tools such as Catia and MSC Adams. From the many conducted tests at Hällered, one good test results will be chosen with road displacement as an input for benchmarking.

Once every needed decision was taken, we built the CAD model of test rig to bring it into real world by manufacturing and assembling it on a four poster test rig. CAD modelling of the rig is explained in the section 5.1. Also, simulation model was built side by side with the help of Adams Car and Adams View which is explained in section 5.2. The manufacturing, assembling,

and testing of the designed rig could not be completed within the 20-week time frame. Once everything is built and assembled, next step would be to compare the test run data with real logs and calibrate it if needed to compare it with new set of real logs.

4.1 Geometry Design: CAD

During the initial phase of the thesis, as there are lot of different things involved in this test rig it was little bit difficult to find the starting point. That's where the engineering design approach is useful. The engineering method (also known as engineering design) is a systematic approach used to reach the desired solution to a problem. The engineering design process is critical because it ensures that the final product or solution meets the requirements.

Understanding a problem is the first step towards solving it. The most important consideration in this step is to understand the users' needs and how the product will fulfill them. In the next step, we conducted comprehensive research with the R&D which is essential for developing an effective plan. Understanding the problem and conducting comprehensive research inspires informed and practical ideas for an efficient product design. As such, it is important to consider the findings of the two steps above when brainstorming and conceptualizing ideas. So, we had a brainstorming session with technical experts within the field to start with the conceptual model as seen in figure 4.2. Concept phase might include the preparation of drawings and other studies, they are generally not considered to involve 'design'. The concept design represents the design team's initial response to the project brief, and articulates the broad outlines of function and form.

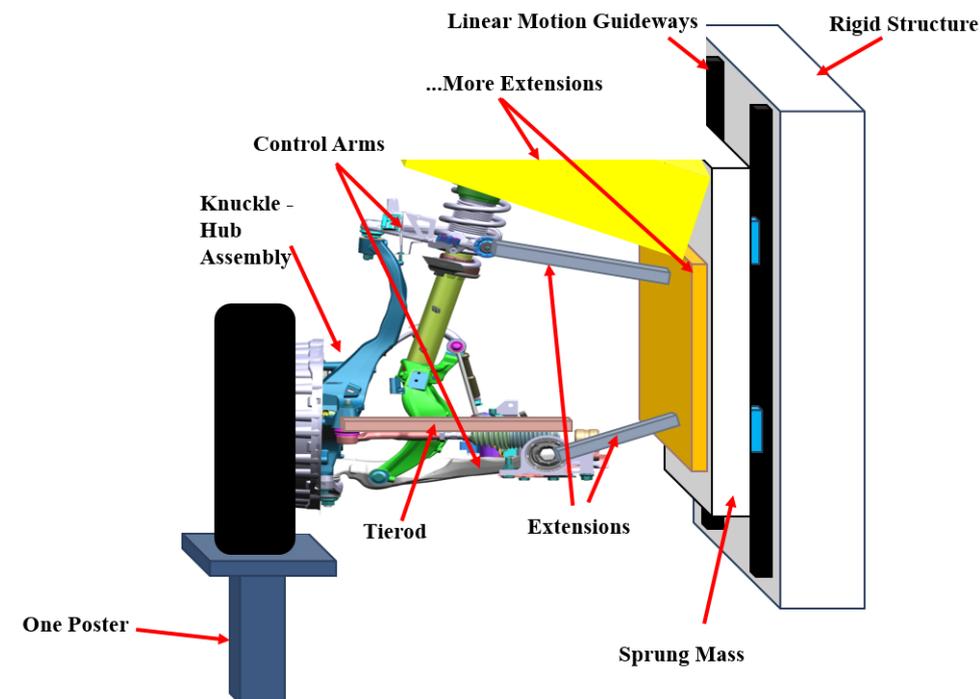


Figure 4.2: Conceptual Testrig

4.1.1 Modular Testrig

Concept phase is followed by 'detailed design' or 'developed design' during which all the main components of the building and how they fit together are described. By the end of the detailed design process, the design should be dimensionally correct and coordinated. However, technical aspects of the design may require further development.

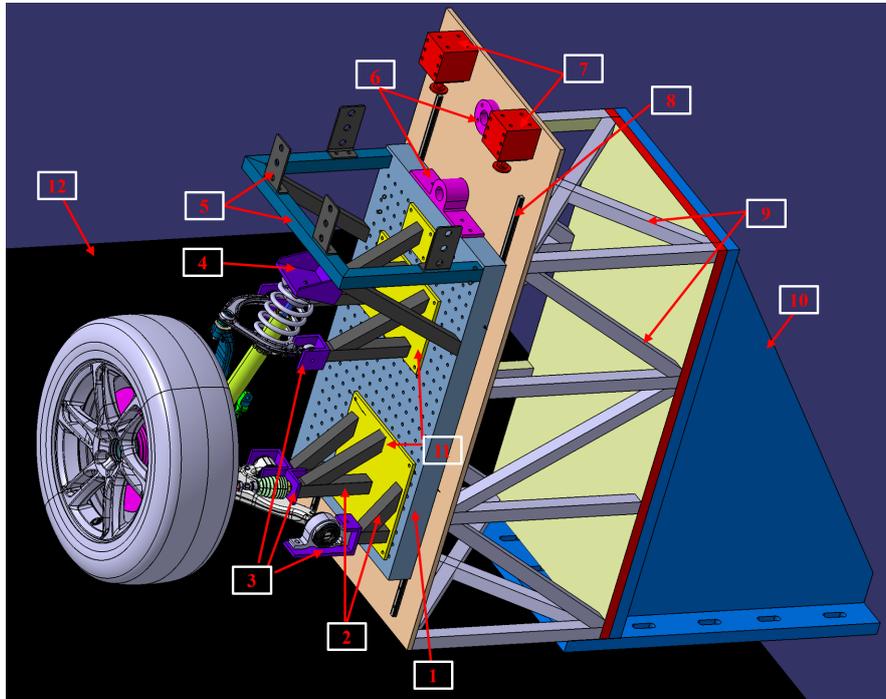


Figure 4.3: Modular Testrig

Sr. No.	Name
1	Sprung Mass Plate
2	Extensions (To mount the shorter control arms)
3	C Brackets
4	Strut Top Mount
5	Extra Sprung Mass Holder
6	Manual Holder (To mount Sprung Mass Plate)
7	Bump Stopper
8	Linear Motion Guideways
9	Bridging Connections
10	Rigid Plate
11	Adaptor Plates
12	Road

Table 4.1: Names of different components in the Modular Testrig

4.1.2 Modularity?

Broadly, modularity is the degree to which a system's components may be separated and recombined, often with the benefit of flexibility and variety in use. A famous example of modularity is LEGO. These plastic toys contain elements that can easily be assembled and reused to develop different finished products. The concept of modularity is used primarily to reduce complexity by breaking a system into varying degrees of interdependence and independence across and "hide the complexity of each part behind an abstraction and interface".

As explained earlier, many suspension geometries such as SLA Suspension, Five Link Suspension, and Integral Link Suspension are used in Volvo Cars manufactured on various platforms having different mounting points, characteristics of a wheel, etc. In case of making a testrig modular so

that it can be used for various purposes, different suspension geometries could be easily assembled, we have to think about packaging of components along all 3 X, Y, and Z global directions. For example, a certain ‘A’ suspension geometry has longest control arm along Y direction, ‘B’ suspension geometry has longest strut assembly along Z direction, etc. Following are the various factors that were considered while designing this test rig to act as an modular test rig:

- Maximum Dimension along Y Axis
- Maximum Dimension along Z Axis
- Biggest Span along X Axis

4.1.2.1 Maximum Dimension along Y & Z Axes

To find out the maximum dimension along both Y & Z axes, we need to review all the suspension geometries. Therefore, we collected all the dimensions along YZ plane. We measured the distance from Tire Center to Subframe Hardpoint of Control Arms in Y direction. Similarly, in Z direction, we measured the distance from Tire Contact Patch to the Strut Top Mount Hardpoint, Figure 4.4.

As seen in the figure 4.4, dimensions for all 4 platforms are collected. From those platforms, CMA platform will be excluded because of following two reasons:

- Suspension geometries used on Platform A might not be used in further projects.
- Maximum dimension in the Y direction was limited by the space available at the test area where the test rig will be mounted.

Platform A	FF	414.48
	FR	384.87
	RF	417.86
	RR	650
	Damper Front (z)	956.11(593.06+363.05)
	Damper Rear (z)	895.99(536.03+359.96)
Platform B	FUF	319.89
	FUR	319.22
	FLF	419.14
	FLR	428.66
	RF	496.20
	RR	540.30
	RU	418.69
	RL	470.75
	Damper Front (z)	964.82(590.44+374.38)
	Damper Rear (z)	720.98(346.60+374.38)
Platform C	FUF	329.89
	FUR	329.21
	FLF	430.34
	FLR	438.66
	RF	478.40
	RR	551.80
	RU	421.05
	RL	413.12
	Damper Front (z)	1007.711(621.87+385.84)
	Damper Rear (z)	774.46(388.62+385.84)
Platform D	FUF	298
	FUR	298
	FLF	438.98
	FLR	395.59
	RUF	305.38
	RUR	377.91
	RLF	419.78
	RLR	534.10
	Damper Front (z)	931.05(544.18+386.87)
	Damper Rear (z)	774.45(388.61+385.84)

Figure 4.4: Dimensions along Y & Z axes

Assumptions:

- Add 100 mm extra length to each dimension as some of the distances taken are eye to eye.
- Distances are in mm.

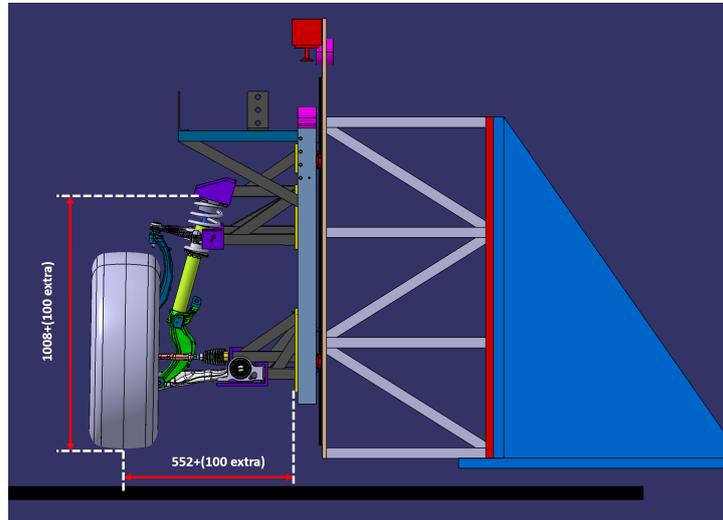


Figure 4.5: Maximum Dimensions along Y & Z axes on actual Testrig

After analysing all the dimensions, it can be conclude that 'Platform C' has the maximum dimensions as highlighted in figure 4.5.

- **Maximum dimension along Y axis = 552+(100)**
- **Maximum dimension along Z direction = 1008+(100)**

4.1.2.2 Biggest Span along X Axis

To find out the biggest span along X axis, we again reviewed all the suspension geometries and collected all the dimensions. We measured the distances of all the Subframe Hardpoints of Control Arms from Tire Center in X direction, Figure 4.6.

Platform B	FUF	56.13
	FUR	-184.34
	FLF	12.06
	FLR	-360.13
	RF	178.35
	RR	-181.10
	RU	-21.14
	RL	154.57
Platform C	FUF	59.79
	FUR	-186.42
	FLF	6.10
	FLR	-371.90
	RF	231.18
	RR	-168.90
	RU	2.26
	RL	144.94
Platform D	FUF	120.81
	FUR	-167.90
	FLF	32.48
	FLR	-377.38
	RUF	159.66
	RUR	-10.44
	RLF	207.49
	RLR	215.72

Figure 4.6: Dimensions along x axis

As seen in the figure 4.6, dimensions for all 3 platforms are collected. Platform A is excluded because of the same reasons as mentioned above.

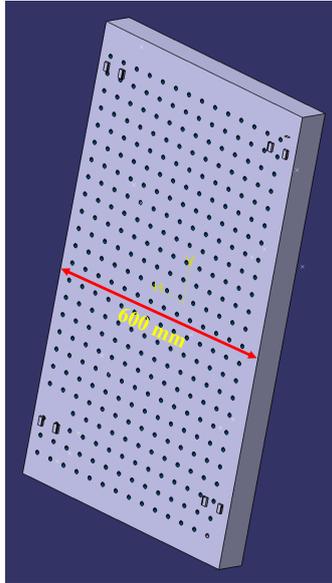


Figure 4.7: Maximum Dimension along x axis on actual Testrig

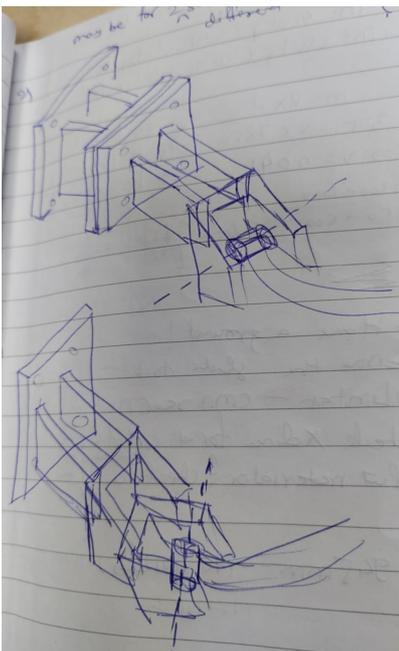
After analysing all the dimensions, it can be concluded that the span created between the RF hardpoint (Rear suspension geometry and front control arm) on Platform C and FLR hardpoint (Front suspension geometry and Lower Rear control arm) on Platform D is the biggest one amongst other span created by different hardpoints, figure 4.7.

- RF distance from Tire Center = 231
- FLR distance from Tire Center = -377

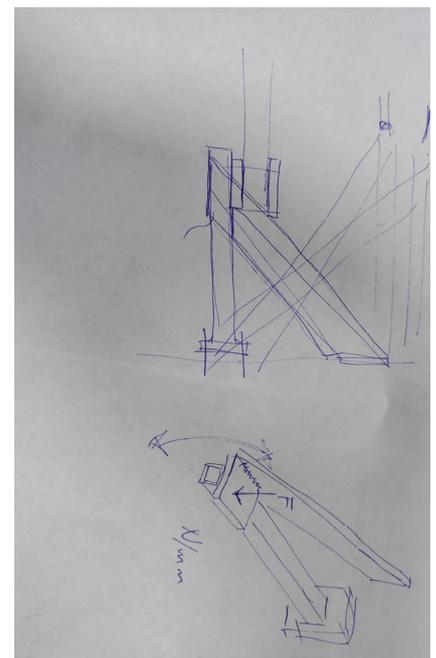
In total the span becomes, $231 + 377 = 608 \approx 600$ mm.

4.1.2.3 Extension Design Ideas

As the testrig being modular, it has been designed for the longest control arm. But there are suspension geometries with shorter control arm. So, we need to think about a solution to mount them in a rigid way. After brainstorming about different extension ideas, following design ideas were picked out.



(a) Design 1 and 2



(b) Design 3

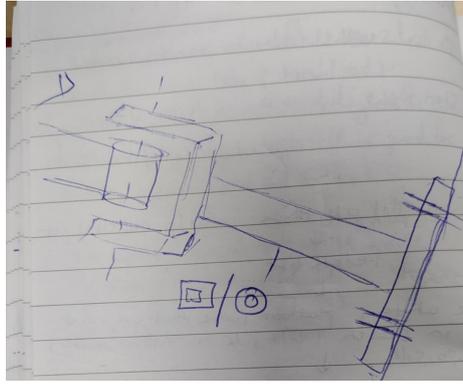


Figure 4.8: Design 4

Considering the concept of DFMA and the availability of material in house, we finalized the extension design 3, which uses steel square tube of 40*40*2 (All in mm) which is easily available in house. The design is very simple that one side of square tube will be welded to C type mount (which will be mounted to control arms with the help of bolts) while, the other side will also be welded to an adaptor plate which will be bolted to sprung mass plate.

4.1.2.4 The Idea Behind Bridging Connections

As seen in figure 4.9, we are limited by the distance between Tire Center to the starting point of a Rigid Plate. We cannot have smaller distance than 1260 mm. Also, on the other hand, we are limited by the distance between Tire Center and an Adaptor Plate. That's why we had to design a connecting path between sprung mass plate and a rigid plate in the form of bridging connections. Those connections are made of steel square tube and the material is easily available in the workshop at Volvo Cars.

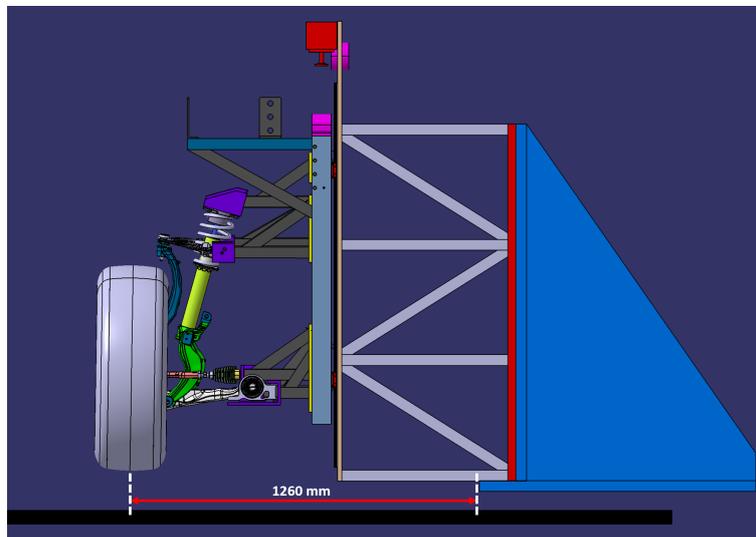


Figure 4.9: Overall Length Limitation

4.1.2.5 Vehicle Weights

Being a modular testrig, we need to make sure that this testrig could be useful for various models of the car having different weights. Therefore, after discussion with technical experts within the same field about range of weights for future platforms including current ongoing projects, we finalized on following two different weight sections (sprung mass of the car), one being the heavy car and other being the lightweight car.

- **Vehicle Weights (including unsprung mass):** Maximum : X kg & Minimum : Y kg
- **Weight Distribution = 50/50**

This is how we performed the calculation for a maximum sprung mass per corner.
Let's say, [Heavy Car]

$(X \times 50\%) / 2 = M_{Total}$ [This will be a heavy car corner mass.]

Unsprung mass : $X_{Unsprung} / 2 = M_{Unsprung}$

Max sprung mass per corner : $M_{Total} - M_{Unsprung} = M_{Max}$

For example,

1800 kg x 50% / 2 = 450

Unsprung mass: 100 / 2 = 50

Max sprung mass per corner = 450 - 50 = 400 kg

Similarly, we did the calculations for a lightweight car. To make the assembly process more easier, we decided to build the testrig for lightweight car. The test rig is designed in such a way that we can add extra sprung mass as we require, e.g., if we decide to testrun the heavy vehicle or in between lightweight and heavy vehicle.

From figure 4.10 and Table 4.2, you can see the weights of different sprung mass components.

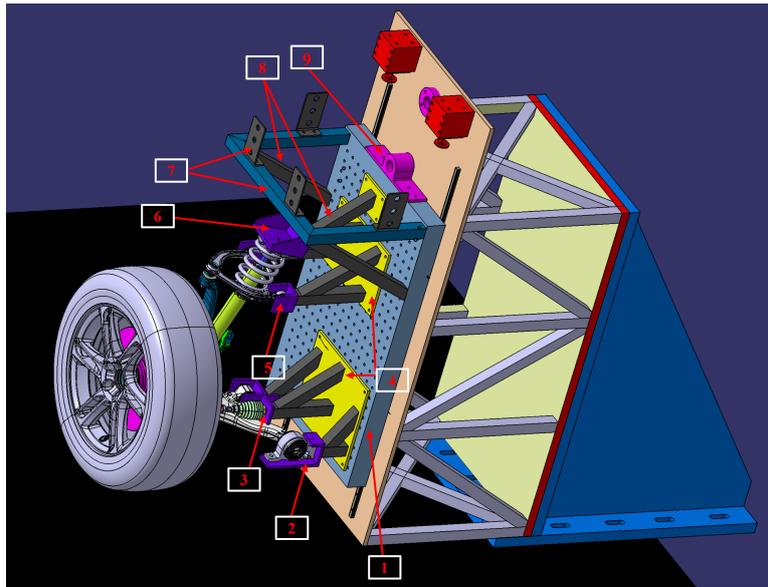


Figure 4.10: Sprung Mass Components

Sr. No.	Component Name	Weight (kg)
1	Sprung Mass Plate	332.64
2	Lower Rear Arm Mount	1.676
3	Tierod Mount	0.66
4	Extension Plates	16.987
5	C type Mounts (Quantity=3)	3.693
6	Damper Mount	2.507
7	Extra Sprung Mass Structure	10.531
8	Extension Square Tubes	7.078
9	Sprung Mass Plate Holder	3.754
10	Total	379.526

Table 4.2: Weights of Sprung Mass Components according to Figure 4.10

4.1.3 What is the Maximum Displacement of Sprung Mass Plate in Jounce and Rebound?

In Vehicle Dynamics, there is a term called Motion Ratio. Mathematically, it is the ratio of shock travel and wheel travel. A motion ratio is a way of showing how much leverage the wheel has over the spring and damper. For example, if wheel travel is 100 mm and motion ratio is 0.8 then shock travel is 80 mm. Similarly, we tried to analyse the ratio between wheel travel and body travel. Therefore, we performed a suspension analysis by running parallel wheel travel baseline simulation on Adams Car. As seen in figure 4.11, we plotted the travel of wheel center in vertical Z direction against the travel of a certain hardpoint on subframe in same direction to get an approximate idea about the ratio between both of them.

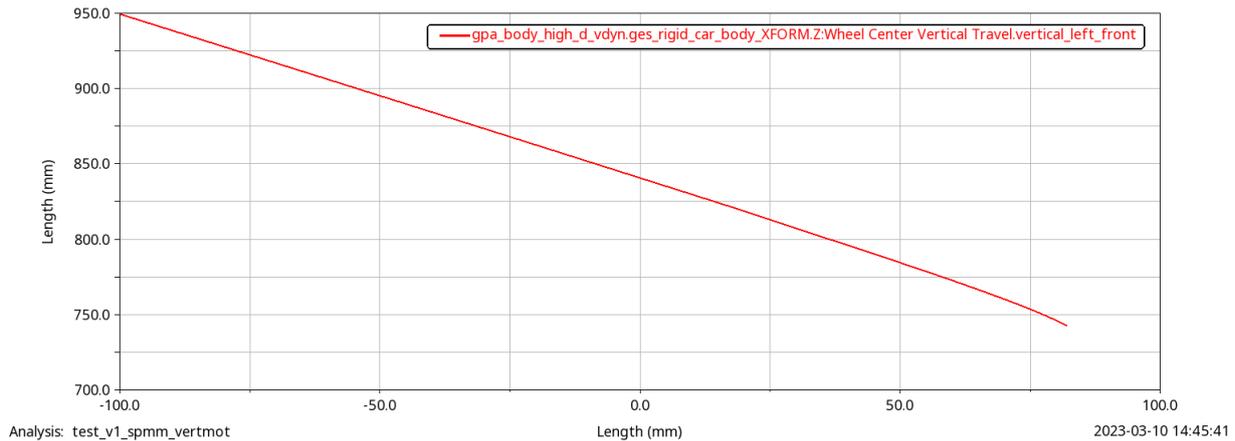


Figure 4.11: Wheel Travel vs Body Travel

Above figure represents the displacement of the body with wheel travel as an input within the range of +80 to -100 mm. From the figure 4.11, it can be seen that the maximum body displacement at maximum jounce is approximately 80 mm. Similarly, the maximum body displacement at maximum rebound is close to 100 mm. So, we can conclude that the ratio between wheel travel and body travel is 1:1.

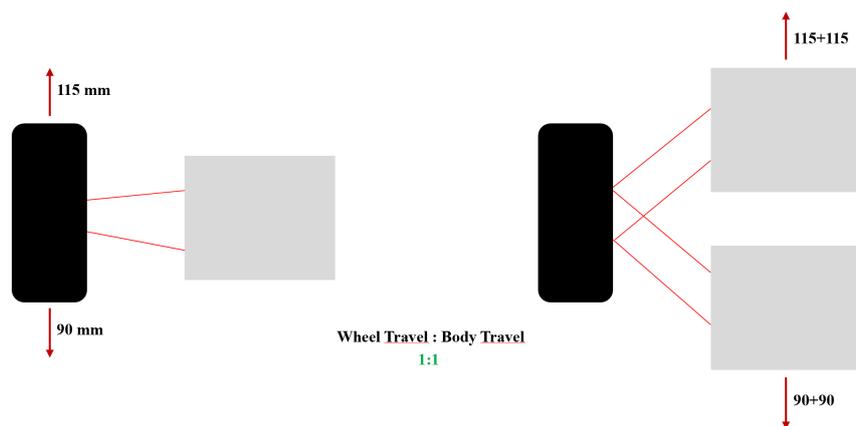


Figure 4.12: Concept Diagram behind Maximum Displacement of the Body during Jounce and Rebound

In real world phenomenon, when the wheel travels, the body can have maximum displacement in either jounce or rebound direction. While designing the test rig, we have to keep it mind the

maximum displacement of body in either directions. From the concept diagram 4.12, one can see that if a wheel travels 115 mm in jounce direction then the maximum body displacement in jounce will be $115+115=230$ mm considering the ratio between wheel and body travel is 1:1. Similarly, the maximum body displacement in rebound will be $90+90=180$ mm.

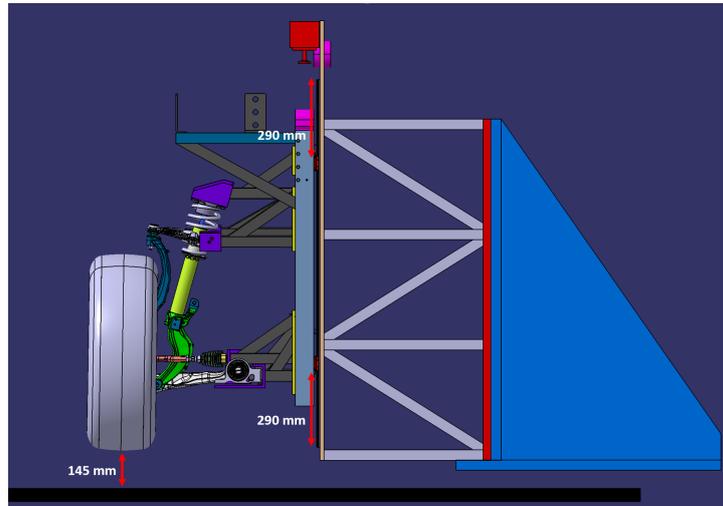


Figure 4.13: Testrig with Maximum Displacement for the Sprung Mass Body

Therefore, in this case, as seen in the figure 4.13, the maximum travel of an actuator of one poster is ± 145 mm, the guideway blocks have the freedom to move upto $145+145=290$ mm in both opposite directions.

4.1.4 Guideways

A linear guideways is a machine element that utilizes bearings, which were developed for rotary motion, in order to move heavy objects easily in a straight line. It is referred to as a “recirculating linear ball bearing” by ISO and JIS, and “linear guideway” by the Japan Machine Tool Builders’ Association.

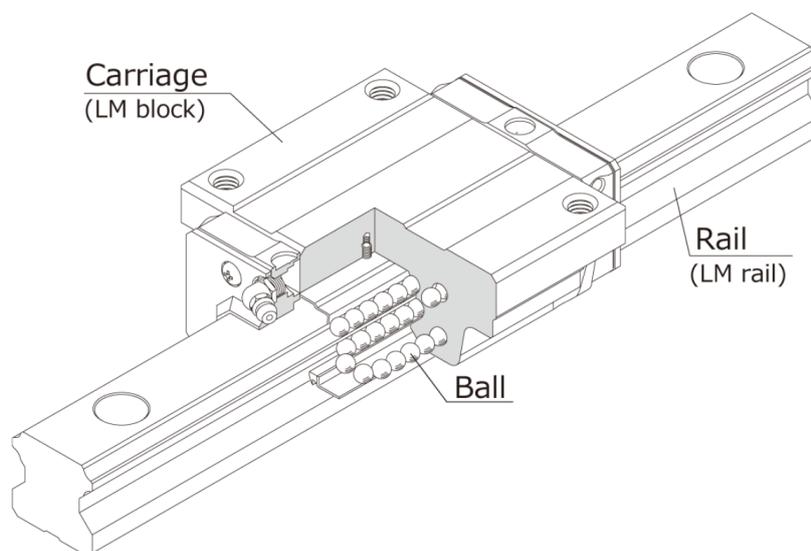


Figure 4.14: Structure of the LM guide

The linear guide largely comprises three components: a mobile carriage, a rail that supports the movements of the carriage, and balls. Linear motion is enabled by attaching a mechanism for recirculating the balls.

4.1.4.1 Flowchart for Selecting an LM Guide

The following flowchart can be used as reference for selecting an LM Guide.

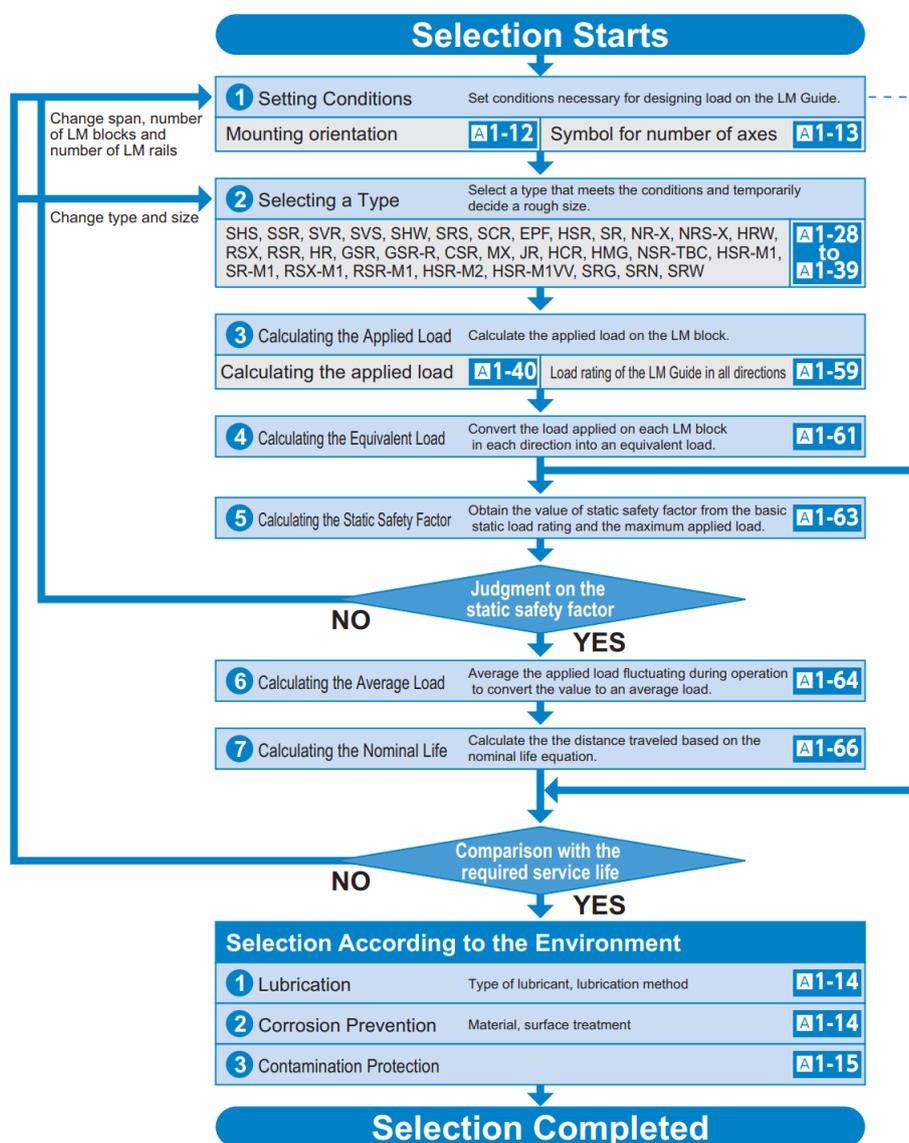


Figure 4.15: Steps for selecting an LM Guide

4.1.4.2 Setting Conditions of the LM Guide

Mounting Orientation

The LM Guide can be mounted in the following five orientations.

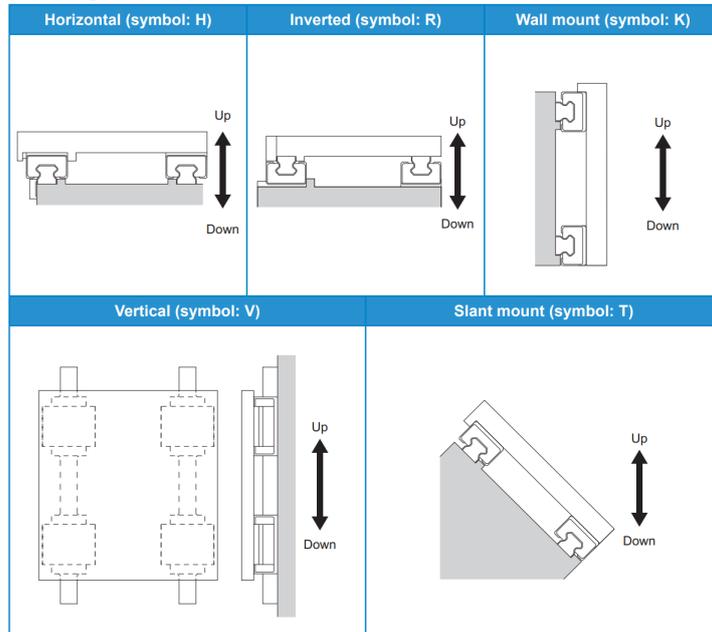


Figure 4.16: Mounting orientation

Symbol for Number of Axes

For this test rig, two units of the LM Guide are parallelly used in Combination on the same plane.

Model number coding		
SHS25C2SSCO+1000LP - II		
Model number (details are given on the corresponding page of the model)		Symbol for number of axes ("II" indicates 2 axes. No symbol for a single axis)
[Symbol for Number of Axes]		
Symbol for number of axes: none	Symbol for number of axes: II	Symbol for number of axes: II
Required number of axes: 1	Required number of axes: 2	Required number of axes: 2
	 Note: When placing an order, specify the number in multiple of 2 axes.	 Note: When placing an order, specify the number in multiple of 2 axes.
Symbol for number of axes: III	Symbol for number of axes: IV	Other
Required number of axes: 3	Required number of axes: 4	Required number of axes: 2
 Note: When placing an order, specify the number in multiple of 3 axes.	 Note: When placing an order, specify the number in multiple of 4 axes.	 Using 2 axes opposed to each other

Figure 4.17: Symbol for Number of Axes

Service Environment

• Lubrication

When using an LM system, it is necessary to provide effective lubrication. Without lubrication, the rolling elements or the raceway may be worn faster and the service life may be shortened.

A lubricant has effects such as the following:

- Minimizes friction in moving elements to prevent seizure and reduce wear.
- Forms an oil film on the raceway to decrease stress acting on the surface and extend rolling fatigue life.
- Covers the metal surface to prevent rust formation.

To fully bring out the LM Guide's functions, it is necessary to provide lubrication according to the conditions. E.g., If the mounting orientation is other than horizontal use, the lubricant may not reach the raceway completely. Additionally, Even with an LM Guide with seals, the internal lubricant gradually seeps out during operation. Therefore, the system needs to be lubricated at an appropriate interval according to the service conditions.

• Corrosion Prevention

Any LM system requires a material that meets the environments. For use in environments where corrosion resistance is required, some LM system models can use martensite stainless steel. Also, the surfaces of the rails and shafts of LM systems can be treated for anti-corrosive or aesthetic purposes.

• Contamination Protection

When foreign material enters an LM system, it will cause abnormal wear or shorten the service life, and it is necessary to prevent foreign material from entering the system. When entrance of foreign material is predicted, it is important to select an effective sealing device or dust-control device that meets the environment conditions. There are various contamination protection accessories for LM Guides such as end seals made of special synthetic rubber with high wear resistance, and side seals and inner seals for further increasing dust-prevention effect. In addition, for locations with adverse environment, Laminated Contact Scraper LaCS and dedicated bellows can also be used.

4.1.4.3 Calculating the applied load

The LM Guide is capable of receiving loads and moments in all directions that are generated due to the mounting orientation, alignment, gravity center position of a traveling object, thrust position and cutting resistance.

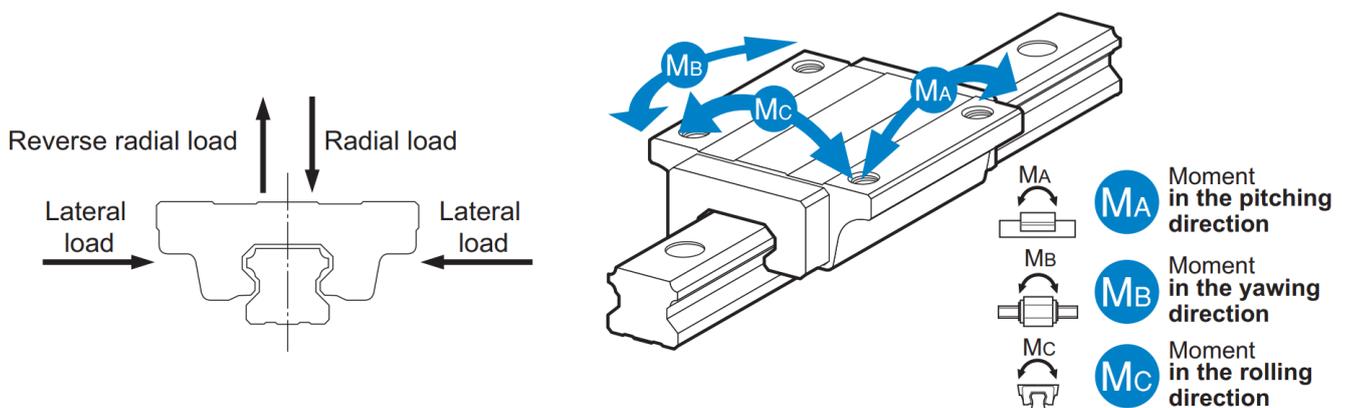


Figure 4.18: Directions of the Loads Applied on the LM Guide

4.1.4.4 Double-axis Use

Setting Conditions

Set the conditions needed to calculate the LM system's applied load and service life in hours.

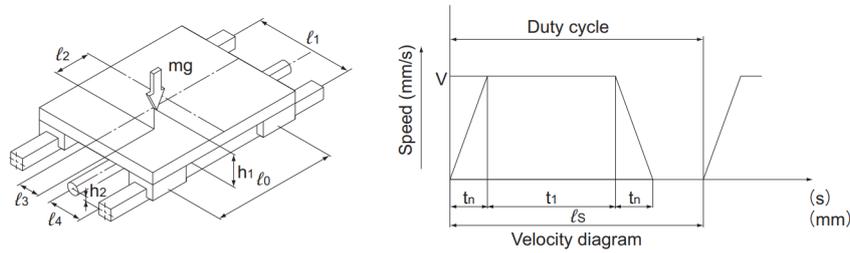


Figure 4.19: Condition

The conditions consist of the following items:

- Mass: m (kg)
- Direction of the working load
- Position of the working point (e.g., center of gravity): l_2, l_3, h_1 (mm)
- Thrust position: l_4, h_2 (mm)
- LM system arrangement: l_0, l_1 (mm) (No. of units and axes)
- Velocity diagram:
 - Speed: V (mm/s)
 - Time constant: t_n (s)
 - Acceleration: α_n (mm/s^2)
- Duty cycle Number of reciprocations per minute: N_1 (min^{-1})
- Stroke length: l_s (mm)
- Average speed: V_m (m/s)
- Required service life in hours: L_h (h)

Applied Load Equations

The load applied to the LM Guide varies with the external force, such as the position of the gravity center of an object, thrust position, inertia generated from acceleration/deceleration during start or stop, and cutting force. In selecting an LM Guide, it is necessary to obtain the value of the applied load while taking into account these conditions.

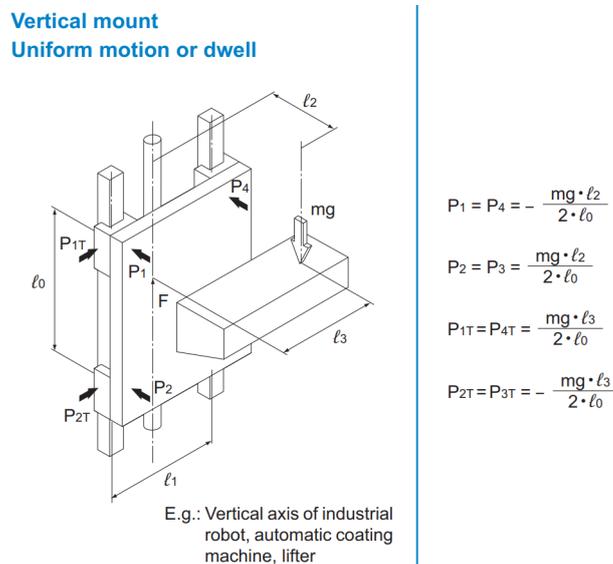


Figure 4.20: Condition and applied load equations

4.1.5 Features and Dimensions of selected Model

4.1.5.1 Structure and Features of the Caged Ball LM Guide

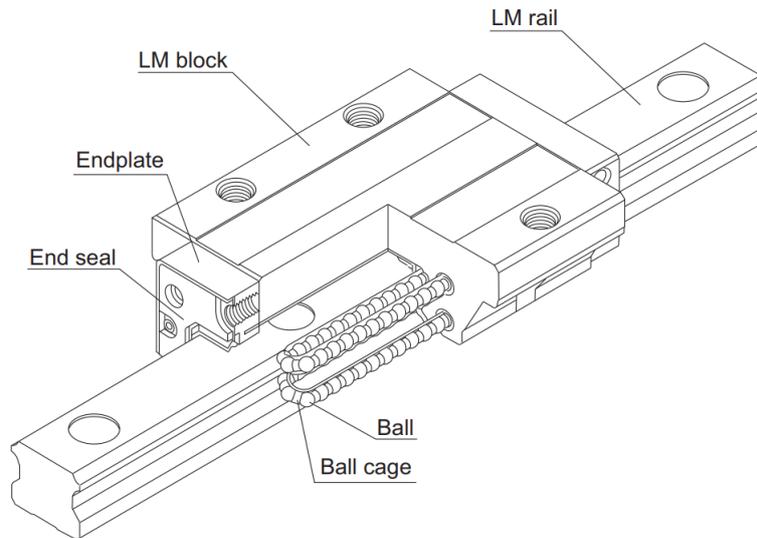


Fig.1 Structural Drawing of the Caged Ball LM Guide Model SHS

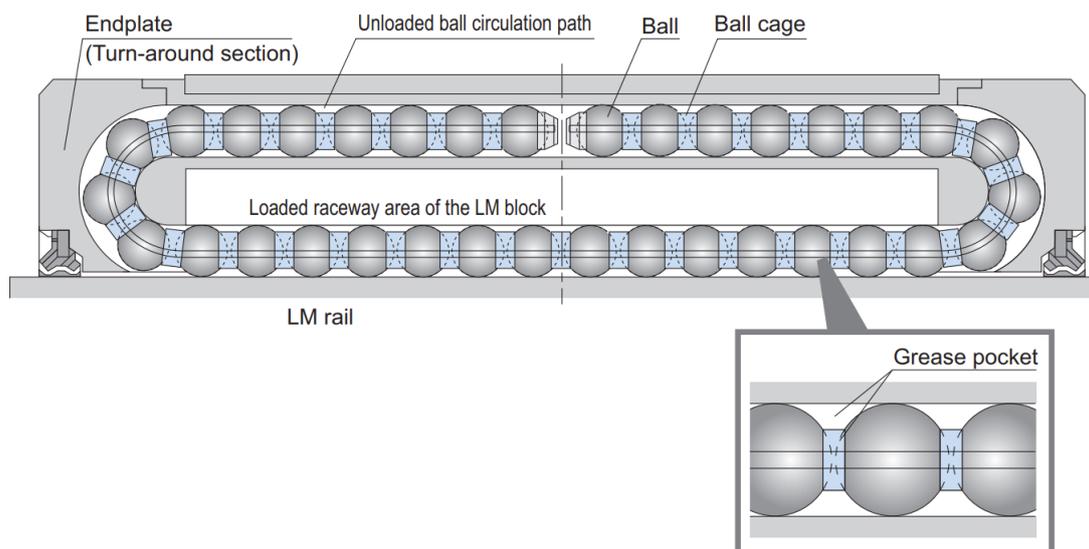


Figure 4.21: Circulation Structure inside the LM Block of the Caged Ball LM Guide

With the Caged Ball LM Guide, the use of a ball cage allows lines of evenly spaced balls to circulate, thus to eliminate friction between the balls. In addition, grease held in a space between the ball circulation path and the ball cage (grease pocket) is applied on the contact surface between each ball and the ball cage as the ball rotates, forming an oil film on the ball surface. As a result, an oil film is not easily broken.

Advantages of Ball Cage Technology

- The absence of friction between balls, together with increased grease retention, achieves long service life and long-term maintenance-free (lubrication-free) operation.
- The absence of ball-to-ball collision achieves low noise and acceptable running sound.
- The absence of friction between balls achieves low heat generation and high speed operation.
- The circulation of lines of evenly spaced balls ensures smooth ball rotation.
- The absence of friction between balls allows high grease retention and low dust generation.

[Low Noise, Acceptable Running Sound]

- Noise Level Data

Since the ball circulation path inside the LM block is made of resin, metallic noise between balls and the LM block is eliminated. In addition, use of a ball cage eliminates metallic noise of ball-to-ball collision, allowing a low noise level to be maintained even at high speed.

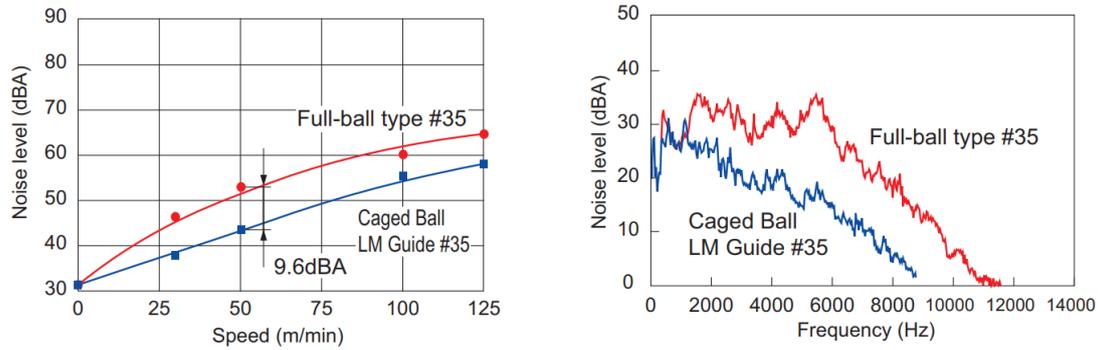
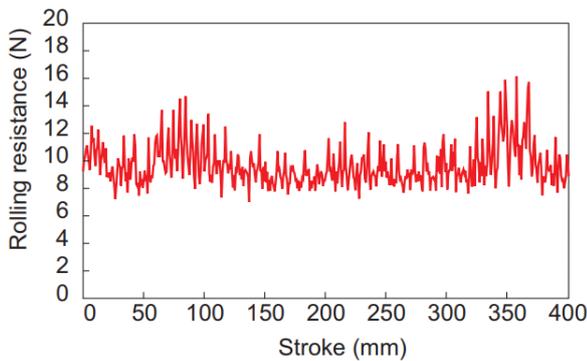


Figure 4.22: Comparison of Noise Levels between Caged Ball LM Guide and Full-Ball Type (at speed of 50 m/min)

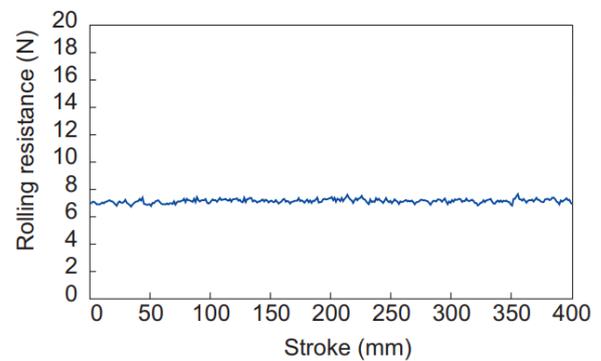
[Smooth Motion]

- Rolling Resistance Data

Use of a ball cage allows the balls to be uniformly aligned and prevents a line of balls from meandering as they enter the LM block. This enables smooth and stable motion to be achieved, minimizes fluctuations in rolling resistance, and ensures high accuracy, in any mounting orientation.



(a) Rolling Resistance Fluctuation Data with Full-Ball (Vertical-use feeding speed: 1 mm/s)



(b) Rolling Resistance Fluctuation Data with Caged Ball (Vertical-use feeding speed: 1 mm/s)

4.1.5.2 Caged Ball LM Guide Radial Type Model SSR

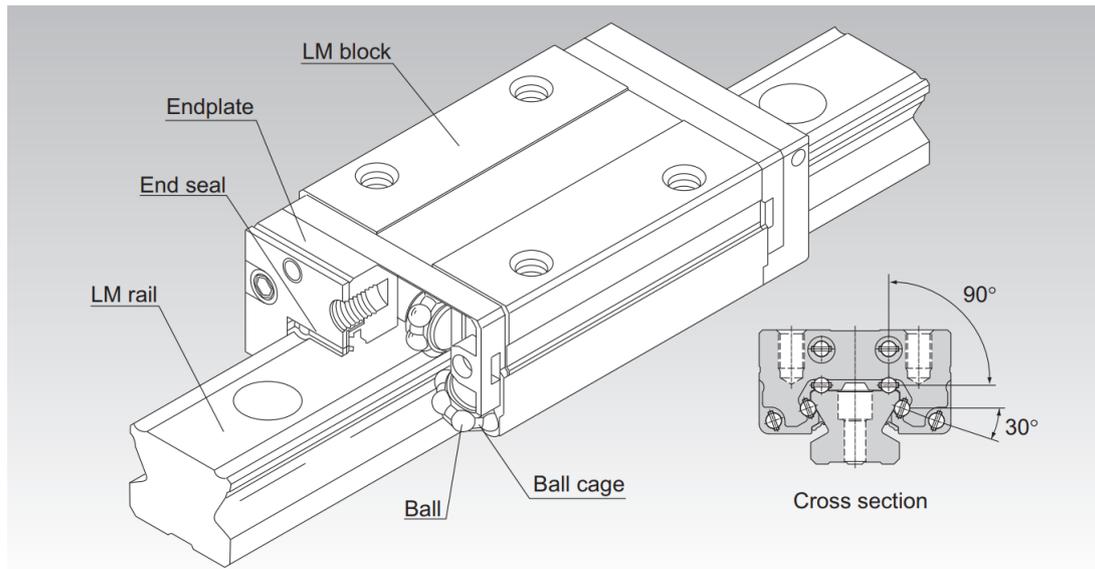


Figure 4.23: Model SSR structure

Balls roll in four rows of raceways precision-ground on an LM rail and an LM block, and ball cages and endplates incorporated in the LM block allow the balls to circulate.

Use of the ball cage eliminates friction between balls and increases grease retention, thus to achieve low noise, high speed and long-term maintenance-free operation.

[Compact, Radial Type]

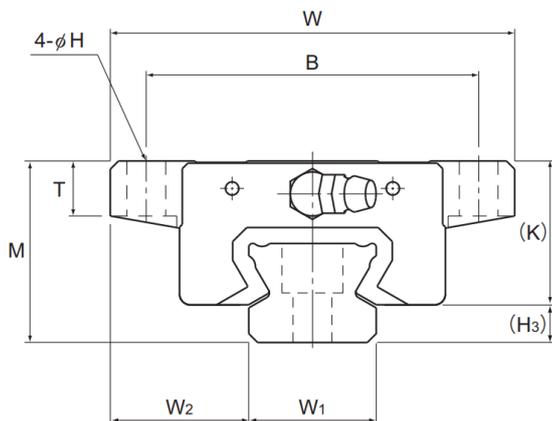
It is a compactly designed model that has a low sectional height and a ball contact structure in the radial direction.

[Superb Planar Running Accuracy]

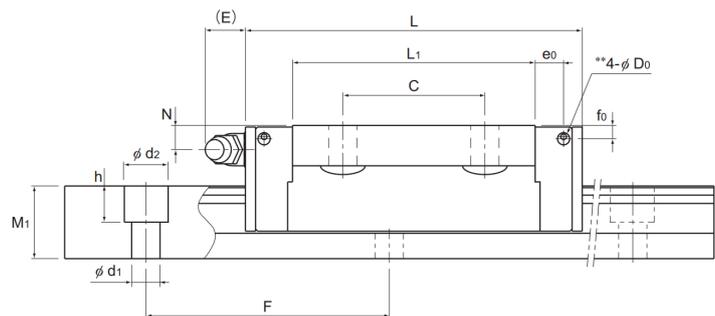
Use of a ball contact structure that is highly resistant to loads in the radial direction minimizes radial displacement under radial loads and provides stable, highly accurate motion.

[Self-adjustment Capability]

The self-adjustment capability through front-to-front configuration of unique circular-arc grooves (DF set) enables a mounting error to be absorbed even under a preload, thus to achieve highly accurate, smooth straight motion.



(a) LM Model Drawing 1



(b) LM Model Drawing 2

Unit: mm

Model No.	LM block dimensions															LM rail dimensions					Basic load rating		Static permissible moment kN·m*					Mass			
	Outer dimensions															Width W ₁ ±0.05	Height W ₂	Pitch M ₁	Pitch F	Length* d ₁ ×d ₂ ×h	Length* Max	C kN	C ₀ kN	M _A			M _B		M _C	LM block kg	LM rail kg/m
	Height M	Width W	Length L	B	C	H	L ₁	T	K	N	E	f _s	e _s	D _s	Grease nipple									H _s	1 block	Double blocks	1 block	Double blocks	1 block		
SSR 15XTB	24	52	56.9	41	26	4.5	39.9	7	19.5	4.5	5.5	2.7	4.5	3	PB1021B	4.5	15	18.5	12.5	60	4.5×7.5×5.3	3000 (1240)	14.7	16.5	0.0792	0.44	0.0486	0.274	0.0962	0.19	1.2
SSR 20XTB	28	59	66.5	49	32	5.5	46.6	9	22	5.5	12	2.9	5.2	3	B-M6F	6	20	19.5	15.5	60	6×9.5×8.5	3000 (1480)	19.6	23.4	0.138	0.723	0.0847	0.448	0.18	0.31	2.1
SSR 25XTB	33	73	83	60	35	7	59.8	10	26.2	6	12	3.3	6.8	3	B-M6F	6.8	23	25	18	60	7×11×9	3000 (2020)	31.5	36.4	0.258	1.42	0.158	0.884	0.33	0.53	2.7
SSR 30XTB	42	90	97	72	40	9	70.7	10	32.5	8	12	4.5	7.6	4	B-M6F	9.5	28	31	23	80	7×11×9	3000 (2020)	46.5	52.7	0.446	2.4	0.274	1.49	0.571	0.87	4.3
SSR 35XTB	48	100	110.9	82	50	9	80.5	13	36.5	8.5	12	4.7	8.8	4	B-M6F	11.5	34	33	27.5	80	9×14×12	3000	64.6	71.6	0.711	3.72	0.437	2.31	0.936	1.33	6.4

(c) LM Block

(d) LM Rail

4.1.5.3 Examples of Arrangements of the Guide System

The following are representative guide systems and arrangements when installing the LM Guide.

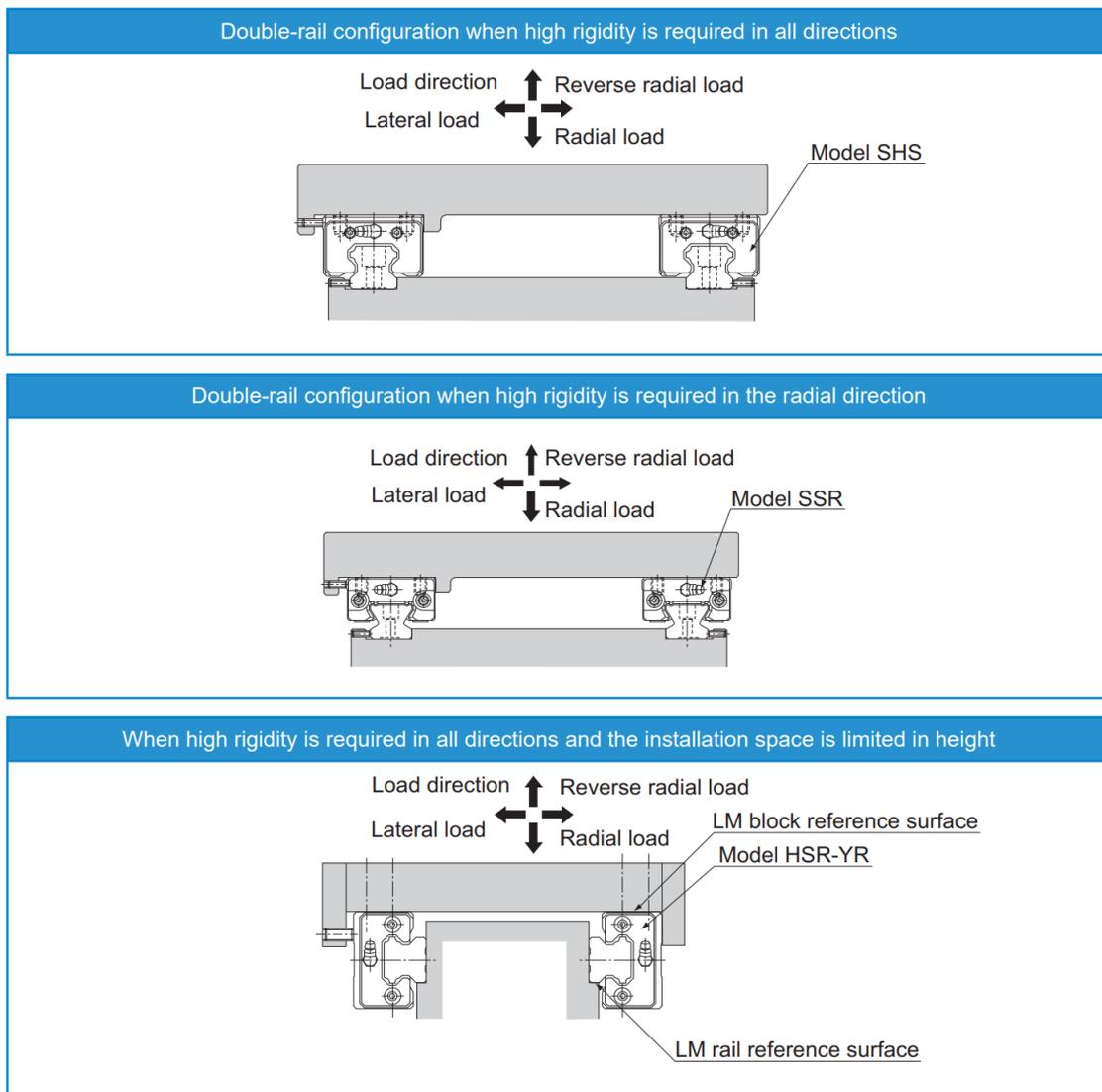


Figure 4.24: SSR Model Arrangement

4.1.6 Mounting Surface

4.1.6.1 Designing a Mounting Surface

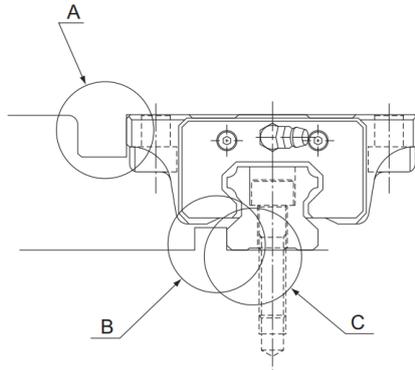


Figure 4.25: Designing a Mounting Surface

If particularly high accuracy is required for the machine to which an LM Guide is to be mounted, it is necessary to mount the LM rail with high accuracy. To achieve the desired accuracy, we have to be sure to design the mounting surface while taking the following points into account.

[Corner Shape]

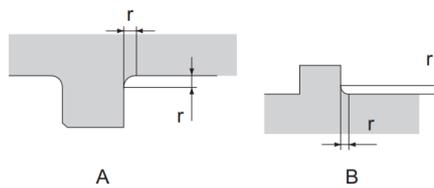


Figure 4.26: Designing a Mounting Surface

If the corner on the surface on which the LM rail or LM block is to be mounted is machined to be shaped R, which is greater than the chamfer dimension of the LM rail or LM block, then the rail or the block may not closely contact its reference surface. Therefore, when designing a mounting surface, it is important to carefully read the description on the “corner shape” of the subject model.

[Perpendicularity with the Reference Surface]

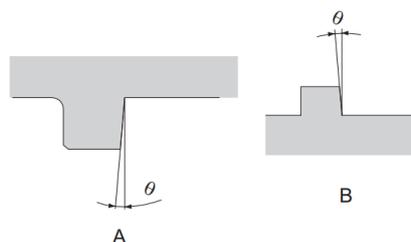


Figure 4.27: Designing a Mounting Surface

If the perpendicularity between the base mounting surface for the LM rail or the LM block and the reference surface is not accurate, the rail or the block may not closely contact the reference surface. Therefore, it is important to take into account an error of the perpendicularity between the mounting surface and the reference surface.

[Dimensions of the Reference Surface]

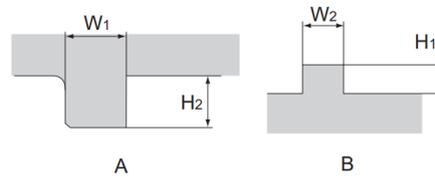


Figure 4.28: Designing a Mounting Surface

When designing the reference surface, be sure to take into account the height and the thickness of the datum area. If the datum area is too high, it may interfere with the LM block. If it is too low, the LM rail or the LM block may not closely contact the reference-surface depending on the chamfer of the rail or the block. Additionally, if the datum area is too thin, the desired accuracy may not be obtained due to poor rigidity of the datum area when a lateral load is applied or when performing positioning using a lateral mounting bolt.

[Dimensional Tolerance between the Reference Surface and the Mounting Hole]

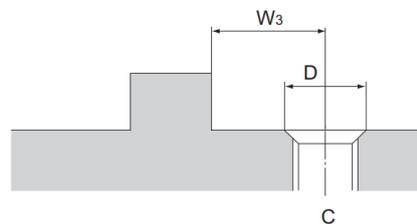


Figure 4.29: Designing a Mounting Surface

If the dimensional tolerance between the reference surface of the LM rail or the LM block and the mounting hole is too large, the rail or the block may not closely contact the reference surface when mounted on the base. Normally, the tolerance should be within ± 0.1 mm depending on the model.

[Chamfer of the Tapped Mounting Hole]

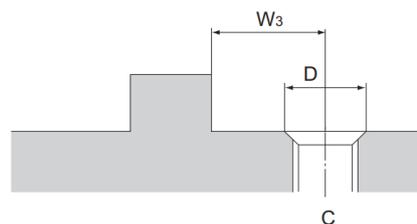


Figure 4.30: Designing a Mounting Surface

To mount the LM rail, the mounting surface needs to be tapped and the tapped hole has to be chamfered. If the chamfer of the tapped hole is too large or too small, it may affect the accuracy. Guidelines for the chamfer dimension: Chamfer diameter $D = \text{nominal diameter of the bolt} + \text{pitch}$

Example: Chamfer diameter D with M6 (pitch): $D = 6 + 1 = 7$

4.1.6.2 Shoulder Height of the Mounting Base and the Corner Radius

Normally, the mounting base for the LM rail and the LM block has a reference-surface on the side face of the shoulder of the base in order to allow easy installation and highly accurate positioning. The corner of the mounting shoulder must be machined to have a recess, or machined to be smaller than the corner radius “ r ,” to prevent interference with the chamfer of the LM rail or the LM block. [Model SSR]

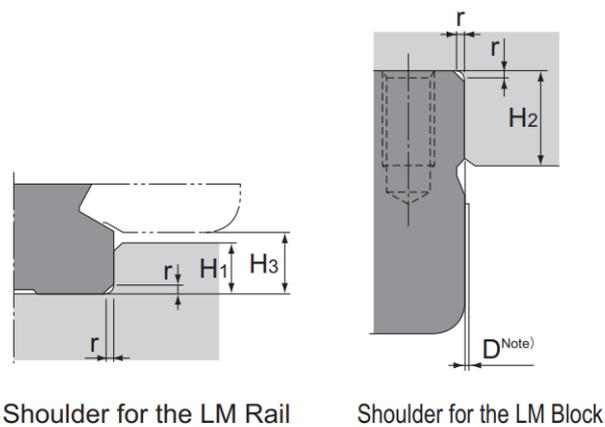


Figure 4.31: SSR Shoulder Height Drawing

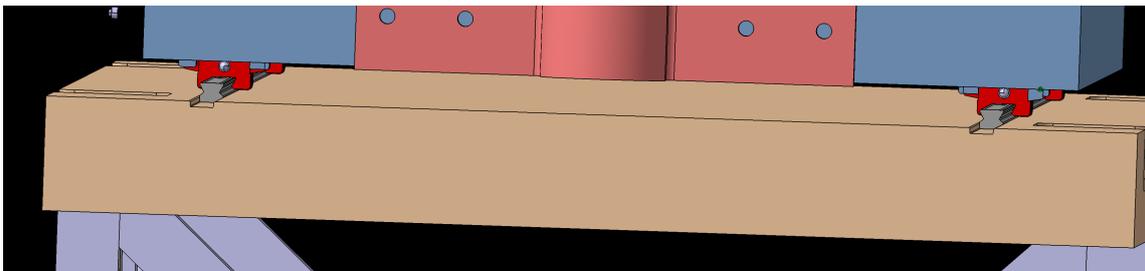


Figure 4.32: Guideway with shoulder heights on actual test rig

Unit: mm

Model No.	Corner radius $r(\text{max})$	Shoulder height for the LM rail H_1	Maximum shoulder height for the LM block H_2	H_3	D
15 X	0.5	3.8	5.5	4.5	0.3
20 X	0.5	5	7.5	6	0.3
25 X	1	5.5	8	6.8	0.4
30 X	1	8	11.5	9.5	0.4
35 X	1	9	16	11.5	0.4

Figure 4.33: SSR Shoulder Height Dimensions

4.1.6.3 Example of Mounting the LM Guide When the Master LM Rail Does not Have a Reference Surface

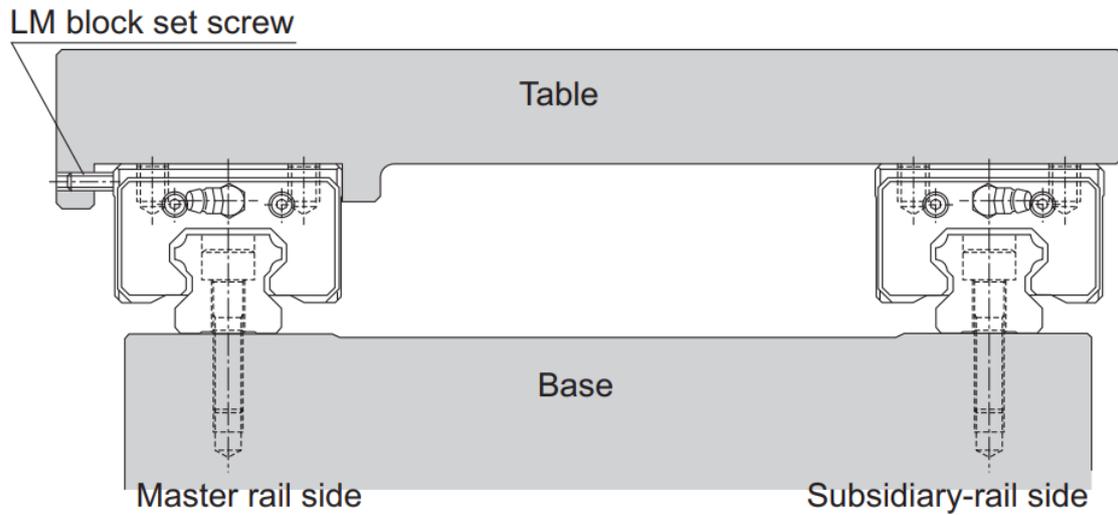


Figure 4.34: Mounting of Master and Subsidiary-rail

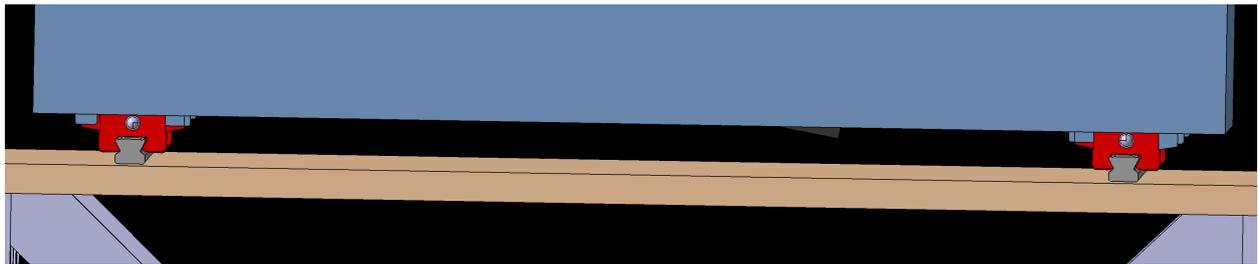


Figure 4.35: Guideway without reference surface on actual test rig

- Mounting the Master LM Rail

- Using a Temporary Reference Surface

You can temporarily set a reference surface near the LM rail mounting position on the base to achieve straightness of the LM rail from the rail end. In this method, two LM blocks must be joined together and attached to a measurement plate.

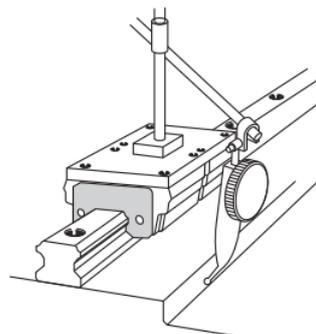


Figure 4.36: Using a Temporary Reference Surface

– **Using a Straight-edge**

After temporarily fastening the mounting bolts, use a dial gauge to check the straightness of the side reference surface of the LM rail from the rail end, and at the same time, fully fasten the mounting bolts

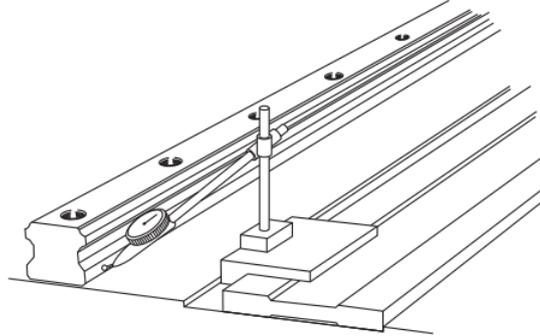


Figure 4.37: Using a Straight-edge

• **Mounting the Subsidiary LM Rail**

To mount the subsidiary LM rail in parallel with the master LM rail, which has been correctly installed, following methods are recommended:

– **Using a Straight-edge**

Place straight-edges between the two rails, and arrange the straight-edges in parallel with the side reference surface of the master LM rail using a dial gauge. Then, secure the mounting bolts in order while achieving straightness of the subsidiary rail with the straight edge as the reference by using the dial gauge.

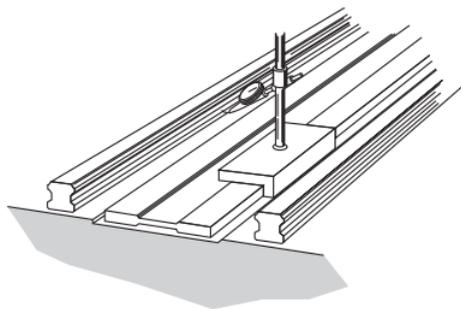


Figure 4.38: Using a Straight-edge

– **Using Parallelism of the Table**

Secure the two LM blocks on the master LM rail with the table (or a temporary table for measurement), and temporarily fasten the LM rail and the LM block on the subsidiary LM rail with the table. Place a dial gauge to the side face of the LM block on the subsidiary rail from the dial stand fixed on the table top, then fasten the bolts in order while achieving parallelism of the subsidiary LM rail by moving the table from the rail end.

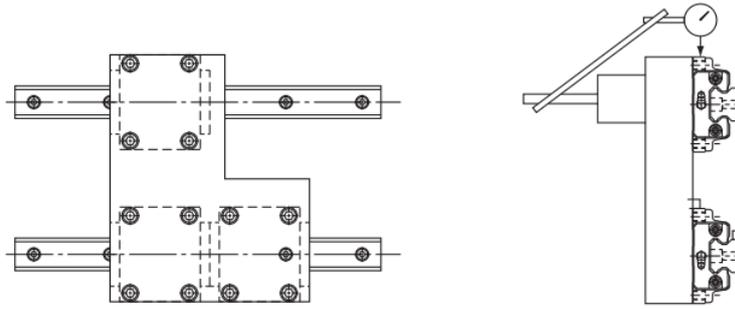


Figure 4.39: Using Parallelism of the Table

– **Having the Subsidiary LM Rail Follow the Master LM Rail**

Place the table on the blocks of the correctly mounted master LM rail and the temporarily fastened subsidiary LM rail, and fully fasten the two LM blocks on the master rail and one of the two LM blocks on the subsidiary rail with bolts. Fully tighten the mounting bolts on the subsidiary LM rail in order while temporarily fastening the remaining LM block on the subsidiary LM rail.

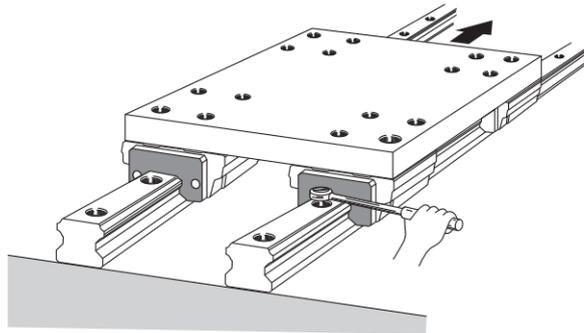


Figure 4.40: Having the Subsidiary LM Rail Follow the Master LM Rail

– **Using a Jig**

Use a jig as shown in figure 4.41, to achieve parallelism of the reference surface on the subsidiary side against the side reference surface of the master side from one end of the rail by the mounting pitch, and at the same time, fully fasten the mounting bolts in order.

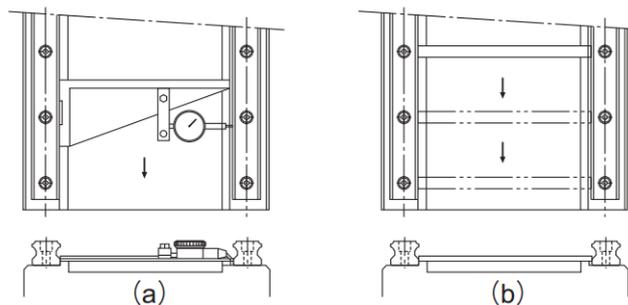
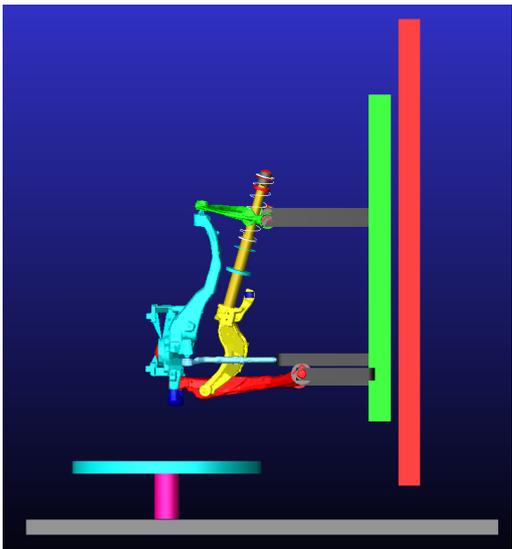


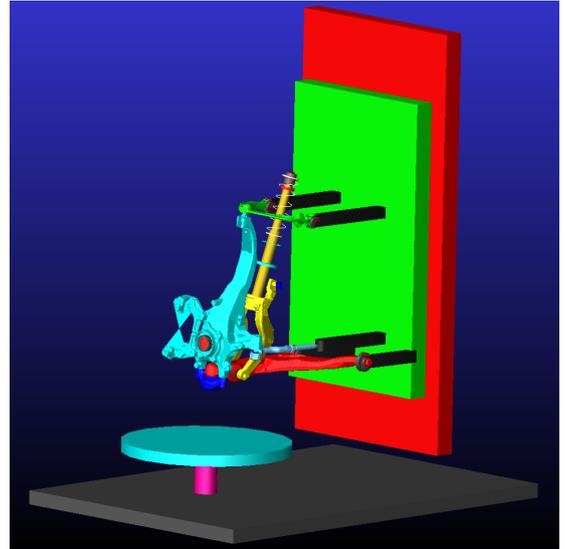
Figure 4.41: Using a Jig

4.2 Simulation Model: MSC Adams

Building a simulation model of the whole rig was the next stage in the thesis. The suspension system model was build using Adams Car while other components of the rig setup were modelled in Adams View. As mentioned, Short-long arms suspension system is built from scratch which will be considered as a base model and on top of that the model will be improvised to build into a quarter car dynamic rig by adding components such as extensions, sprung mass plate, rigid plate along with corresponding required joints which is more feasible in Adams view to make the whole system function in vertical direction, as seen in the following figure.



(a) Front View



(b) Isometric View

4.2.1 Joints

The links are connected to the knuckle with the help of spherical joints and are connected to the sprung mass plate (subframe) by a bushing. Additionally, the steering link is connected to chassis side by a fixed joint because of limitations as mentioned earlier. The spring damper system is connected to the sprung mass plate by installing fixed joint at the top mount and the lower mount is connected to the lower control arm by a spherical joint, figure 4.42

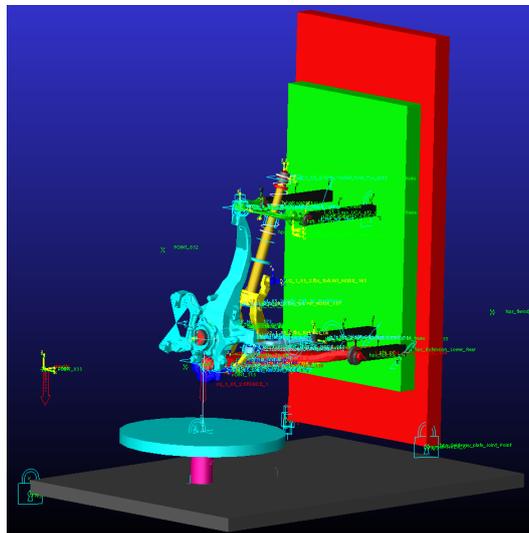
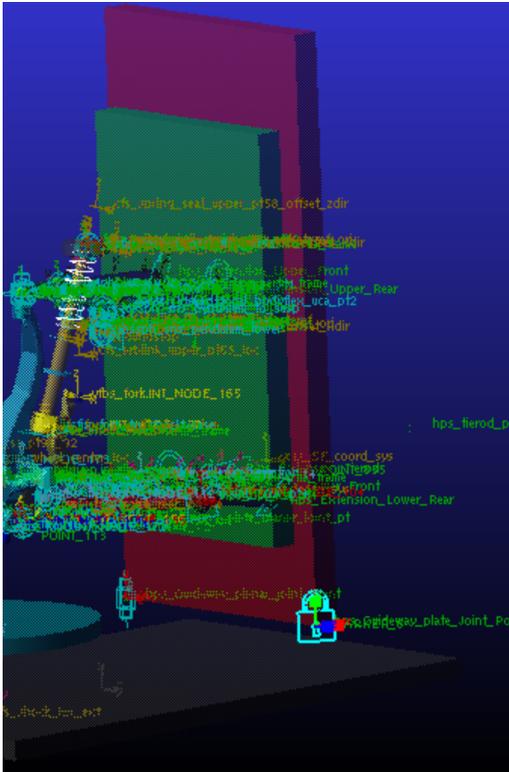


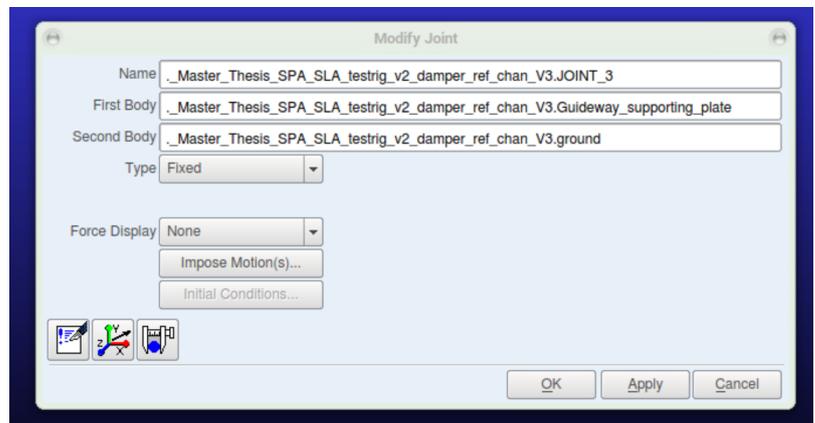
Figure 4.42: Quarter Car Rig with Joints

Fixed Joint

Similarly, with the help of Adams view, as the rigid plate will be bolted to the ground, we have used fixed joint in this simulation model, figure 'a'. As seen in figure 'b', the fixed joint is defined between the guideway supporting plate (Rigid Plate) and the ground.



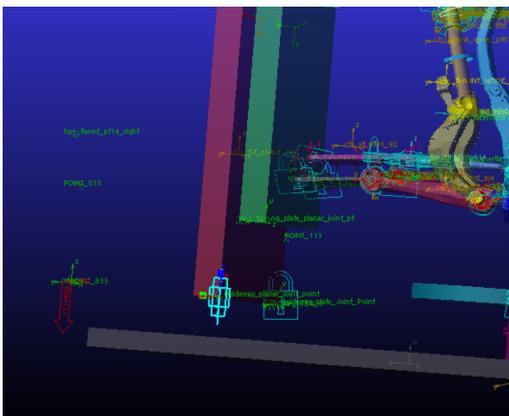
(a) Fixed Joint



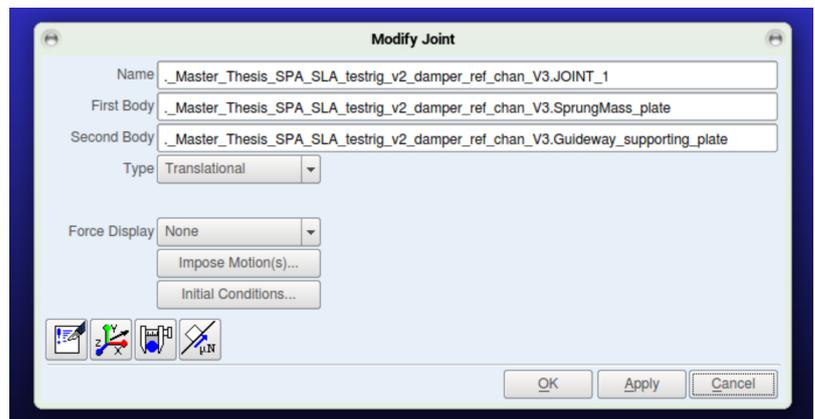
(b) Fixed Joint Definition

Translational Joint

In the physical rig, the sprung mass plate will be sliding vertically on the rigid plate to imitate the motion of a sprung body. The sliding vertical motion will be guides by LM guides. In Adams View, LM guides could be well represented by modelling it as a translational joint in the virtual model, figure 'c'. As you can see from the Translation Joint Definition, figure 'd', sliding vertical motion is defined between sprung mass plate and the guideway supporting plate (Rigid plate).



(c) Translational Joint



(d) Translational Joint Definition

4.2.2 Tire

Tires are very important for all vehicles including passenger and commercial vehicles. They Carry the load resulting from the weight of the vehicle and all the overloads linked to dynamic movements of the vehicle, together with any aerodynamic overloads at high speed, at the same time absorbing the irregularities in the road.

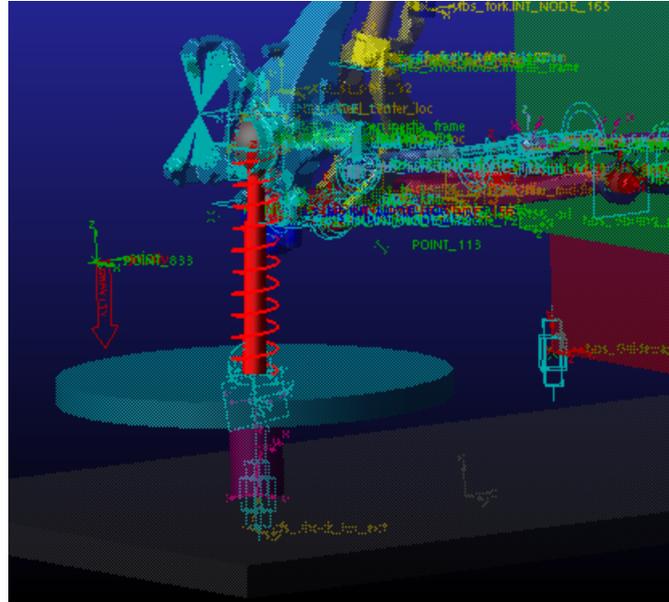


Figure 4.43: Tire as a Spring-Damper System

Initially, as everyone knows tire was modelled as a spring damper system, figure 4.43. Unfortunately, when we performed the general actuation analysis, there was some unknown error due to which we were forced to find an another solution to install the effect of a tire.

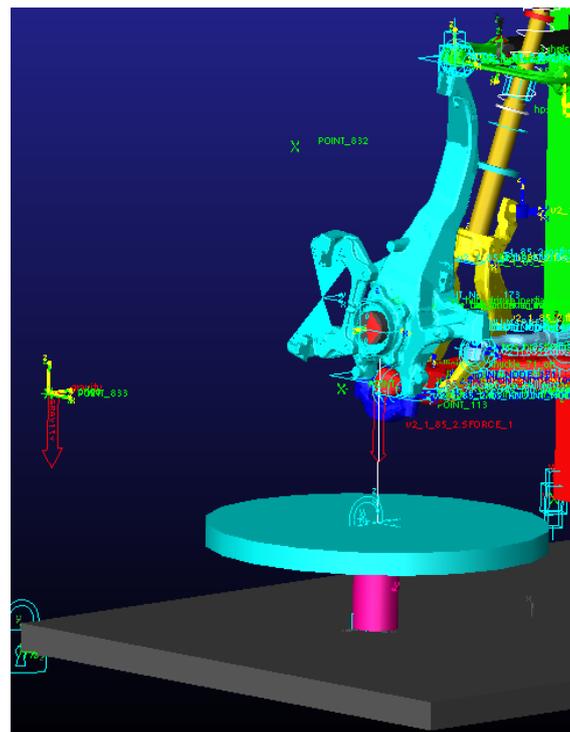


Figure 4.44: Tire as a Single Component Force

There are few commands in Adams to represent a tire such as importing a tire graphical property file or using single component force as a tire by putting tire equation in the form of function. As the main aim was to build a simple working model and avoid complexity, we decided to go for later option, figure 4.44. Single-component force (SFORCE) returns a force or torque applied by a specified single-component force on one or two bodies directly affected by the single-component force.

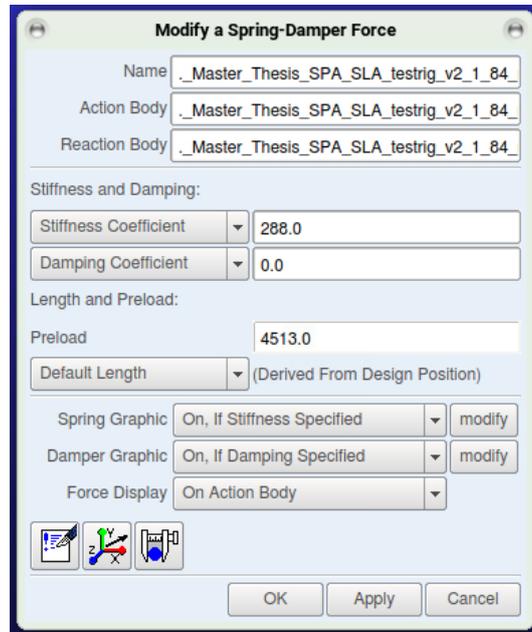


Figure 4.45: Tire properties

Figure 4.45 represents a tire as a Spring-Damper Force, while figure 4.46 shows a tire as a single component force. If you compare both the figure, even though they are two different methods to represent a tire, all the tire parameters are same. In the figure 4.46, we just create a function by taking magnitude between two different points in vertical direction along with stiffness coefficient and preload acting on a tire.

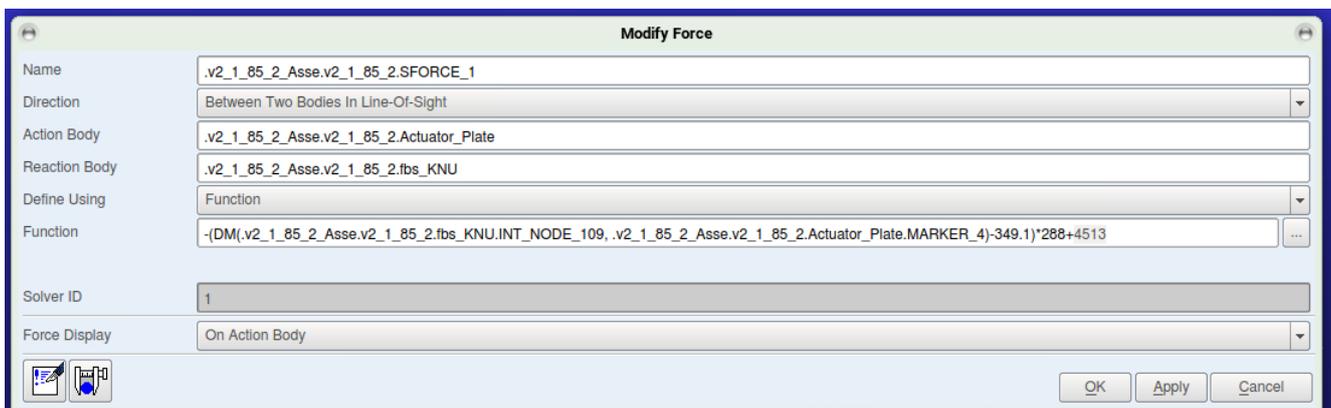


Figure 4.46: SFORCE

4.2.3 Will there be a huge effect of vertical stiffness and damping of having a rolling tire while performing a test run on physical quarter car test rig compared to stationary tire?

There were lots of good discussions happened to find a solution to the above issue but still it was not possible to find an exact solution by doing any physical testing. I had couple of meetings with wheel suspension technical expert and people with the background of Tire Mechanics within VCC. Even the experts within VCC are trying to investigate this, because in Hällered shake rig (SR10) they face this potential "issue" (vehicle in stand still in rig, but they actually want to evaluate vehicles moving with rotating wheels). So they would also like to know more about how big difference that is expected. One way to study this is by performing Tire test rig simulation in Adams car.

Tire Test Rig

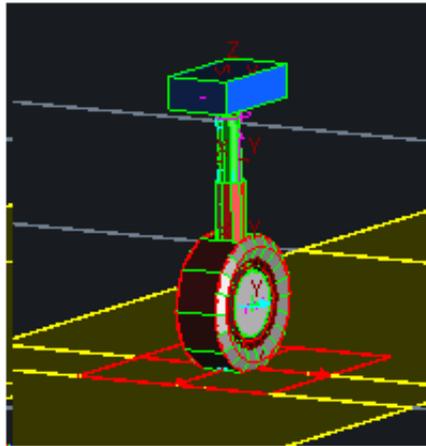


Figure 4.47: Tire Test Rig Simulation in Adams Car

The Tire Test Rig application offers the possibility to analyze specific tire responses in a test rig with a single tire under any conditions with any excitations. Tire load or wheel axle height can be defined, as well as a tire forward velocity, rotational velocity, steering and camber angle. The roads can be flat or provided with a cleat, can be fixed to ground and can be moving according to the user's specification.

Starting the Tire Test Rig in Adams Car:

- Select **Simulate** → **Component Analysis** → **Tire Testrig**. This will open the Tire Testrig dialog box as shown below.

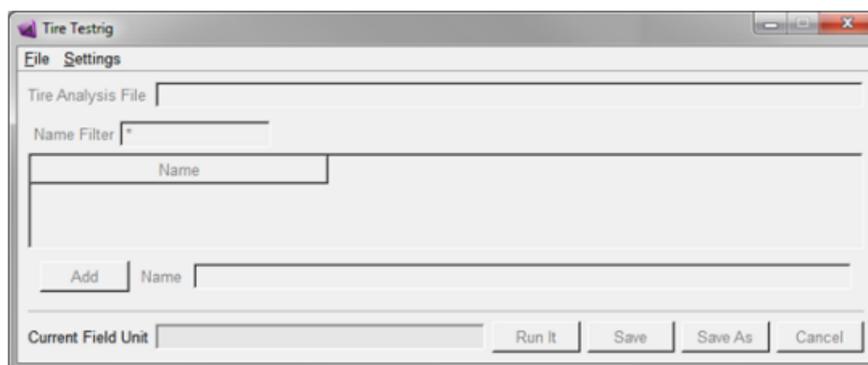


Figure 4.48: Tire Test Rig Simulation Dialogue Box

- Predefined Test Rig analyses database files can be selected by clicking on **File** → **Open**.

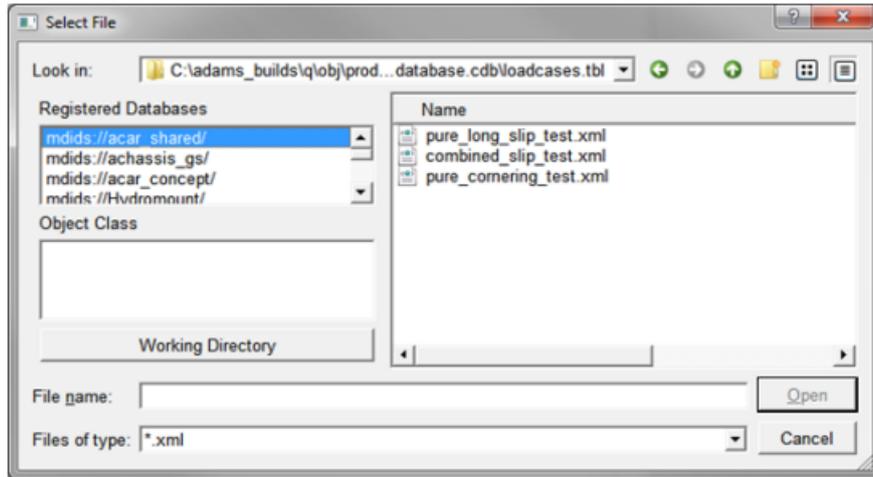


Figure 4.49: Tire Test Rig Simulation Dialogue Box

- In the Registered Databases select any suitable .xml file from the folder and then click **Open**.

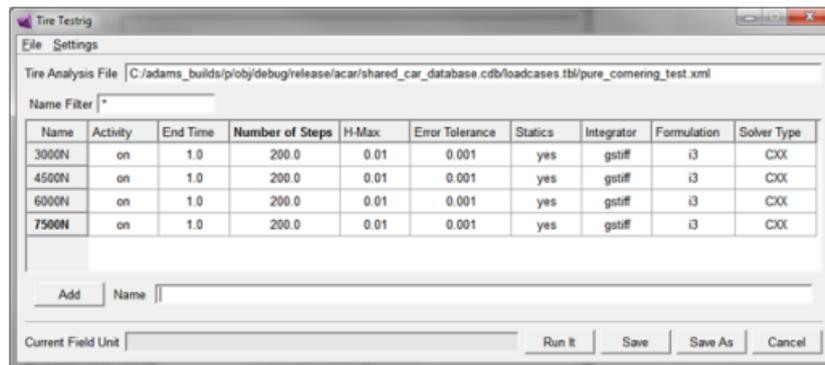


Figure 4.50: Tire Test Rig Simulation Dialogue Box

- Each line represents one simulation with the Tire Test Rig. Clicking on one of the cells in the **Name** column will allow you to define the simulation specifications in detail. For example clicking on '3000 N' will bring you into the **Tire** tab.
- By clicking the **Run It** button, all specified simulations will be run with the solver and simulation settings as shown. After each simulation the simulation results will be imported into the Adams Postprocessor and a set of standard plots will be created.
- By clicking the **Add** button a new analysis will be added with a name specified in the **Name** box.
- The set of simulations can be saved in an .xml database file and re-loaded for another session.

Tire Test Rig Inputs

The Test Rig different tab's and their respective options are explained below:

- **Tire**

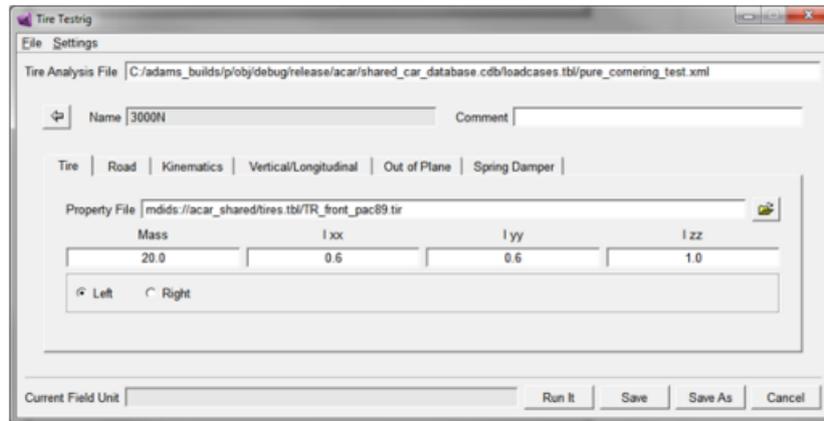


Figure 4.51: Tire Test Rig Inputs

The '**Property File**' cell defines the property file to be used in this simulation, while the '**Mass**', '**Ixx**', '**Iyy**' and '**Izz**' define the mass and inertia of the wheel and tire.

With '**Left**' and '**Right**' the tire characteristics may be mirrored depending on the mounted side of the tire defined in the tire property file.

- **Road**

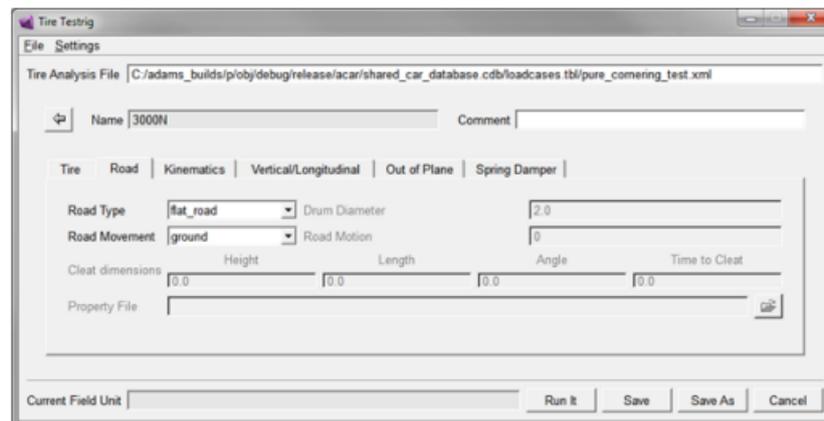


Figure 4.52: Tire Test Rig Inputs

- **Road Type**

- * **flat road:** A 2D flat road will be used
- * **flat with cleat:** A 2D flat road with a cleat (plank) positioned at a location (Time to Cleat) x (Initial long. Velocity) is used. The '**Height**', '**Length**' and the '**Angle**' of the cleat can be specified. The '**Angle**' is the angle in between the plank's x-axis with the road's x-axis.
- * **drum with cleat:** A 2D drum road with a cleat (plank) positioned at a location (Time to Cleat) x (Initial long. Velocity) is used. The '**Height**', '**Length**' and

the '**Angle**' of the cleat can be specified. The '**Angle**' is the angle in between the plank's x-axis with the road's x-axis.

- * **user defined:** Any Adams road property file can be selected

– Road Movement

- * **ground:** The road will be fixed to ground, no movements of the road.
- * **cambered road:** In the '**Road Motion**' cell an Adams (Motion displacement) function expression can be defined to specify a rotational movement of the road around the Test Rig's x-axis. The rotation axis of the camber motion is located at zero road height.
- * **forward moving road:** In the '**Road Motion**' cell an Adams (Motion displacement) function expression can be defined to specify a forward movement of the road around the Test Rig's x-axis.
- * **lateral moving road:** In the '**Road Motion**' cell an Adams (Motion displacement) function expression can be defined to specify a lateral movement of the road around the Test Rig's y-axis.
- * **upward moving road:** In the '**Road Motion**' cell an Adams (Motion displacement) function expression can be defined to specify an upward movement of the road around the Test Rig's z-axis.

• Kinematics

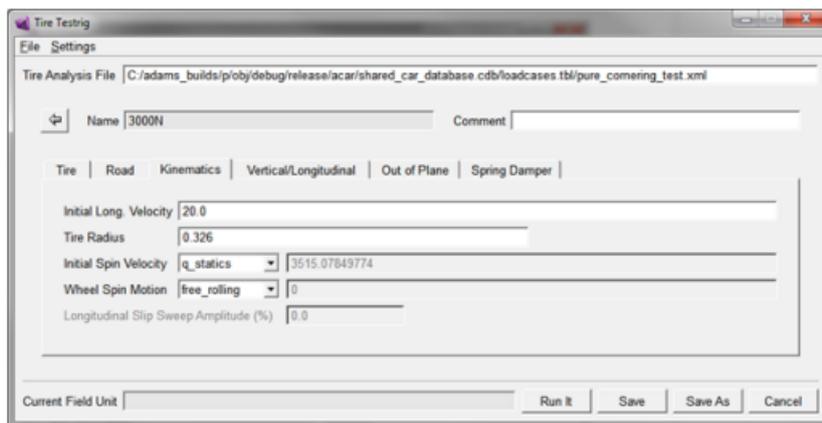


Figure 4.53: Tire Test Rig Inputs

- **Initial Long. Velocity:** The initial forward velocity of the wheel with respect to ground at the start of the simulation
- **Tire Radius:** The tire unloaded radius, required to position the wheel exact at the tire radius distance above the road at the beginning of the simulation.
- **Initial Spin Velocity**
 - * **q statics:** The initial rotational velocity (used at the start of the simulation) of the wheel will be calculated using the 'Tire Radius' and the 'Initial Long. Velocity' by a quasi-statics setup.
 - * **user defined:** The user can specify the initial rotational velocity of the wheel in the cell at the right of this drop down box.
- **Wheel Spin Motion**
 - * **free rolling:** No rotational motion will be applied to the wheel (free rolling).
 - * **long slip sweep:** A rotational motion will be applied to the wheel starting at the negative 'Longitudinal Slip Sweep Amplitude' value towards the positive 'Longitudinal Slip Sweep Amplitude' at the end of the simulation. The slip value is based on the '**Initial Long. Velocity**' and the '**Tire Radius**'.

- * **disp motion:** A rotational displacement motion that is applied to the wheel can be defined at the right of the drop down box.
- * **vel motion:** A rotational velocity motion that is applied to the wheel can be defined at the right of the drop down box.
- **Vertical/Longitudinal**

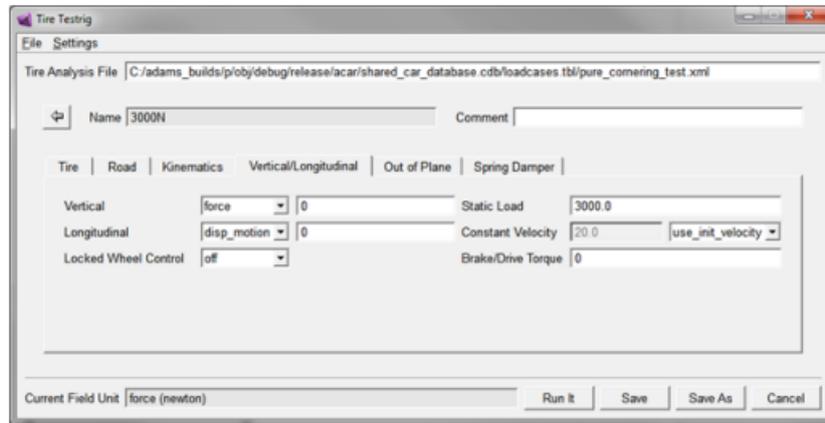


Figure 4.54: Tire Test Rig Inputs

– Vertical

- * **force:** A vertical load/force expression, for a vertical force on the wheel center, can be defined in the cell at the right of this drop down box. For a constant vertical load value the '**Static Load**' input box can be used. When the '**force**' option is used the wheel can move without any constraints in vertical direction.
- * **disp motion:** A vertical displacement motion is applied to the wheel center based on the function expression defined at the right side of the drop down box. A zero motion displacement means that the distance in between the wheel center and the road is equal to the Tire Radius defined under tab Kinematics. A positive displacement causes a tire deflection.
- * **vel motion:** A vertical velocity motion is applied to the wheel center based on the function expression defined at the right side of the drop down box. In the initial position, the distance of the wheel center to the road is equal to the unloaded tire radius.

– Longitudinal

- * **force:** A force along the x-axis of the tire Test Rig on the wheel center can be defined in the box at the right of this drop down box.
- * **disp motion:** A displacement motion along the x-axis on the wheel center will be applied as defined at the right of this drop down box. For a displacement equal to a constant velocity, the '**Constant Velocity**' input box can be use. The **Use Initial Velocity** button will import the Initial Long. Velocity value defined under the Kinematics tab.
- * **vel motion:** A velocity motion along the x-axis on the wheel center will be applied as defined at the right of this drop down box. For a velocity equal to a constant velocity, the '**Constant Velocity**' input box can be used. The **Use Initial Velocity** button will import the Initial Long. Velocity value defined under the Kinematics tab.

– Locked Wheel Controller

- * **off:** The anti-lock controller is not applied.
- * **on:** A simple anti-lock (torque) controller is applied to the wheel.

- **Brake/Drive Torque:** An expression can be defined for a rotational torque along the wheel spin axis.
- **Out of Plane**

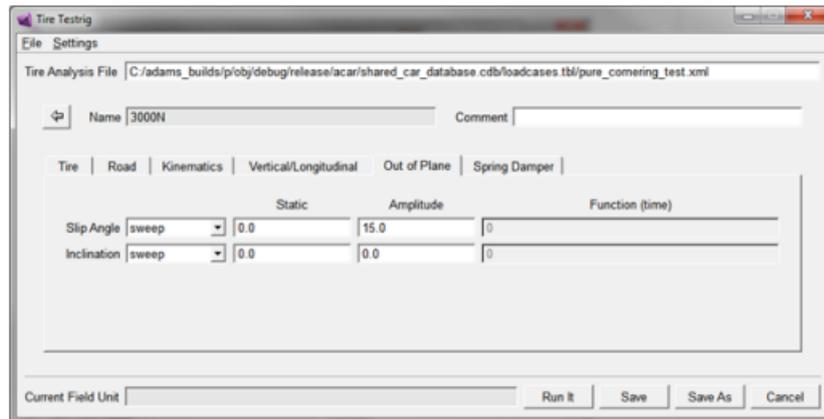


Figure 4.55: Tire Test Rig Inputs

- **Slip Angle**
 - * **sweep:** A sine rotational motion is applied along the vertical axis of the wheel (at the wheel center). A static steer angle and the amplitude of the sweep can be specified. One complete sine period will be performed over the duration of the simulation.
 - * **user function:** An Adams expression can be specified for the rotational motion of the wheel around the vertical axis.
- **Inclination**
 - * **sweep:** A sine rotational motion is applied along the longitudinal axis (x-axis) of the wheel (at the wheel center). A static steer angle and the amplitude of the sweep can be specified. One complete sine period will be performed over the duration of the simulation.
 - * **user function:** An Adams expression can be specified for the rotational motion of the wheel around the longitudinal axis.
- **Spring/Damper**

The Spring Damper tab specifies the properties of a spring in between the wheel center and the ground.

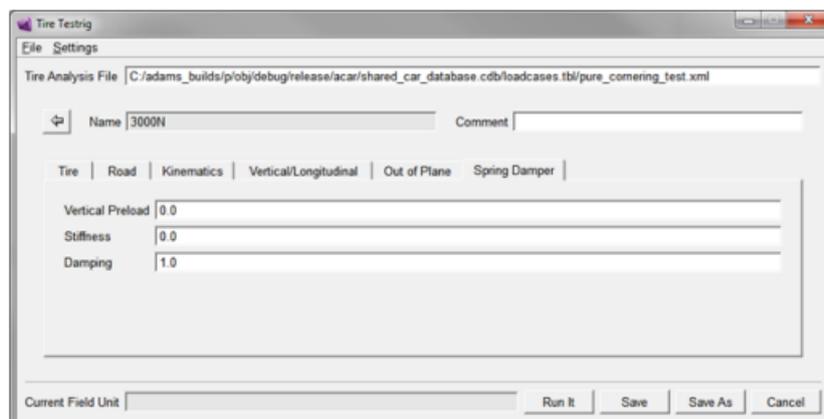


Figure 4.56: Tire Test Rig Inputs

- **Vertical Preload:** The preload force of the spring in between the wheel center and the ground
- **Stiffness:** The stiffness rate of the spring in between the wheel center and the ground
- **Damping:** The damping rate of the spring in between the wheel center and the ground

To get an approximate solution and somewhat clear idea about the effect of vertical stiffness and damping, we ran sine sweep function while performing Tire Testrig Simulation. We Plotted force versus Deflection graph as below in fig 4.57.

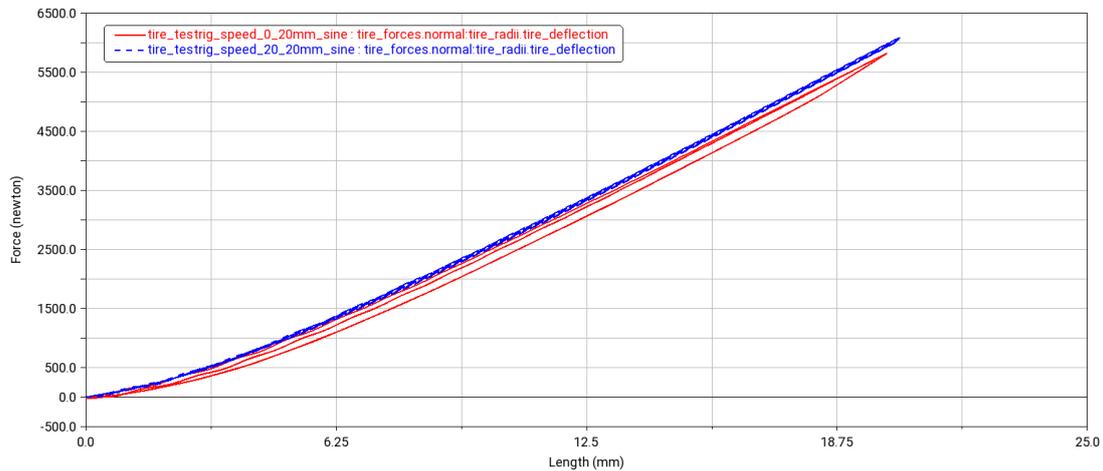


Figure 4.57: Force vs Deflection graph using Tire Testrig Analysis in Adams Car

From the above graph, we can conclude that, vertical stiffness increases if we use rotating tire instead of stationary tire whereas, graph shows hysteresis when a tire is in stand still position.

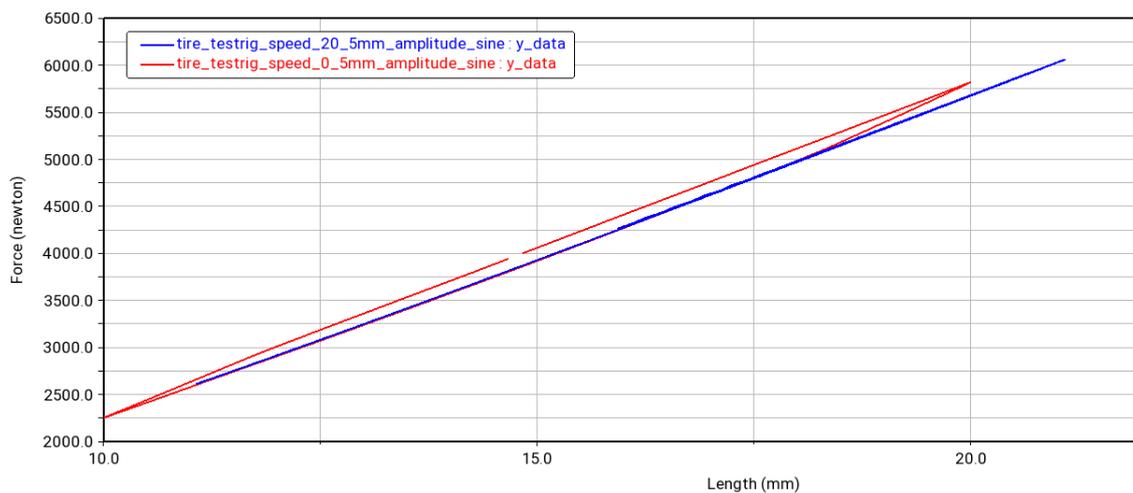


Figure 4.58: +/- 5 mm at 15mm initial displacement

To get more clear idea about hysteresis, we changed the scale and plotted the graph at 15 mm initial displacement. Then, we analysed the performance of a tire between the range of 10 to 20 mm displacement as shown in the figure 4.58. The exact reason behind the hysteresis is still under investigation but after discussion with some experts, rolling resistance phenomenon of tire might be the cause of hysteresis.

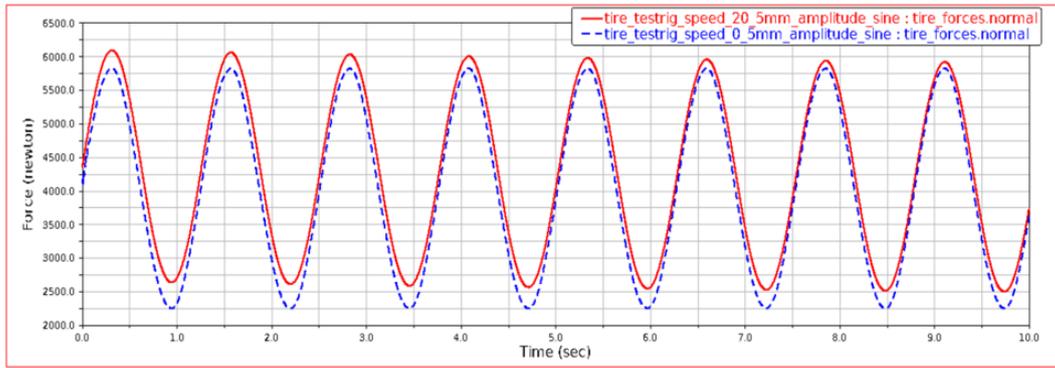


Figure 4.59: Normal force vs Time for both Rotating and Stationary tire

We also plotted Normal Force versus Time to see the difference between vertical forces during certain time range acting on a rotating tire compared to tire in stand still. From the graph 4.59, we concluded that, there is not much big difference in forces assuming the load cases that we will running on actual testrig. Also, considering the complexity added by having a rotating tire on a testrig and limitation of time, we decided to keep a tire in a stand still position.

4.2.4 Input to the actuator

Once the model is properly build with all the required joints and bushings, next step is to make sure it works dynamically. This has been done in Adams View by installing Translational Joint Motion Actuator at the location of Actuator Pole/one poster as seen in figure 4.61 which takes displacement as an input. However, we can use forces from road profiles as an input if we install a force actuator instead of motion actuator.

We can apply the translational joint motion using the Interpolation function with the help of following steps:

- Creating a Spline
- Defining a Translational Joint Motion
- Simulating Your Model

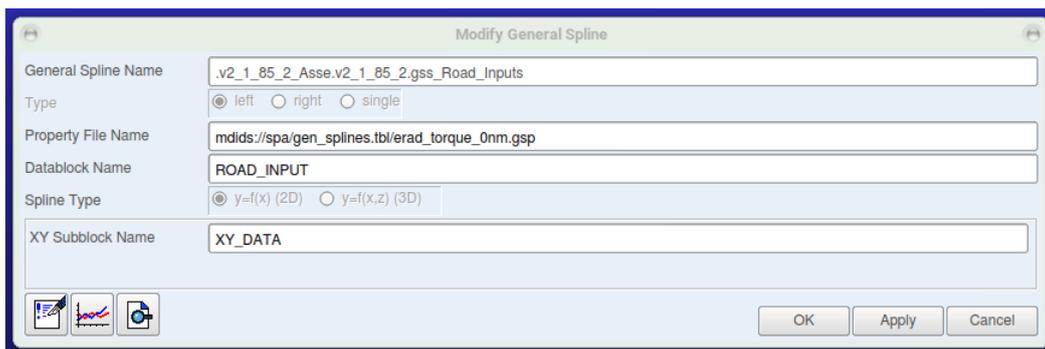


Figure 4.60: Create a spline

In this section, you'll create a spline statement to reference the file and channel arguments from a test performed on a physical model in a test lab, figure 4.60. A spline creates a continuous function from a set of data points. Splines are useful when you have test data or manufacturer specifications that specify the value of a function at several points. The spline can define a curve (two-dimensional, x, y) or a surface (three-dimensional, x, y, z).

You can use splines to create nonlinear functions for motions, forces, or other elements that use functions. In the case of a motion, the points define the displacement, velocity, or acceleration as

a function of time, displacement, velocity, or another Adams quantity.

Defining a Translational Joint Motion

The Adams View spline element contains the x, y or x, y, z data points that you want to interpolate. To use the spline element, you must write a function expression [figure 4.61] that includes Adams spline functions (such as AKISPL or CUBSPL). The functions or subroutines interpolate the discrete data.

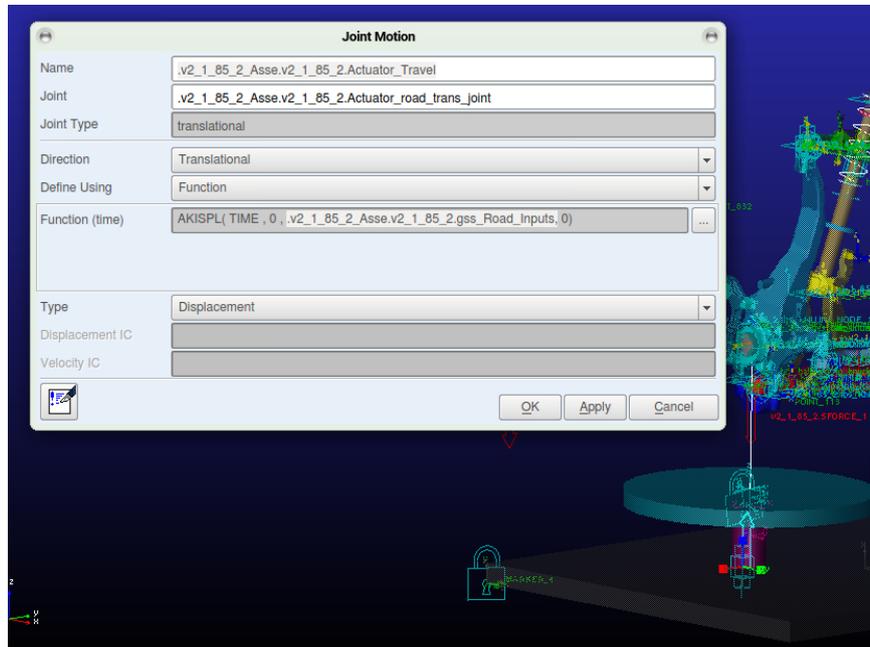


Figure 4.61: Akima cubic curve fitting function as an input to the translational joint motion

AKISPL accesses data in a SPLINE statement (C++ or FORTRAN), using the Akima cubic-curve fitting method to fit a spline to the data (that is, to add interpolated points), and returns one of the following:

- A value for the dependent variable (y) for each value it receives for the independent variable or variables.
- The first partial derivatives of the dependent variable.
- The second partial derivatives of the dependent variable.

If the spline data input with the SPLINE statement has one independent variable, Adams Solver uses a cubic polynomial to interpolate between points. If the spline data has two independent variables, Adams Solver first uses a cubic interpolation method to interpolate between points of the first (x) independent variable, and then uses a linear method to interpolate between curves of the second (z) independent variable.

4.3 Sensors

The need to test functionality without accessing a complete target system is increasing. . At the same time development times are often being shortened. The solution was a specially developed Hardware-In-the-Loop (HIL) - rig consisting of software and hardware that simulates parts of, or the entire, target system.

The rig consists of generic electrical interfaces for connecting ECUs and other hardware, as well as connecting analog and digital input and output for taking measurements and for simulated stimuli.

It has electronic loads to simulate engines, and a real-time PC running simulations and handling input and output. The capability to quickly switch, connect, or disconnect hardware facilitates the work considerably.

In addition to testing and verifying system requirements, the test system can be used for extended fault injections and testing scenarios that would be too risky to perform in a real target system.

In context of automotive applications "Hardware-in-the-loop simulation systems provide such a virtual vehicle for systems validation and verification. Since in-vehicle driving tests for evaluating performance and diagnostic functionalities are often time-consuming, expensive and not reproducible, HIL simulators allow developers to validate new hardware and software automotive solutions, respecting quality requirements and time to market restrictions. In addition, an I/O unit allows the connection of vehicle sensors and actuators (which usually present high degree of non-linearity). Finally, the Electronic Control Unit (ECU) under test is connected to the system and stimulated by a set of vehicle maneuvers executed by the simulator. At this point, HIL simulation also offers a high degree of repeatability during testing phase.

Once the test rig is manufactured and assembled, we would like to perform different load cases and test run the testrig. To measure the different signals coming out from the test run, following sensors will be used.

4.3.1 Production Sensors

Quarter Car rig being a dynamic HIL rig, some vehicle sensors (In Car sensors) as mentioned earlier will be used on this test setup. For a setup consisting a suspension system along with sprung body, usually there are three accelerometers, one on sprung mass, other on knuckle, and third on a wheel or a road platform which gives input creating a plane for the whole system. Additionally, few level sensors are also used to measure displacement between sprung and unsprung mass.

Following accelerometer module shows an example of various required specification parameters and their values.

Digital Accelerometer Module

Parameter	Value
Max. operating frequency range	120 Hz
Measurement Range	+/-1.6 g
Maximum Voltage	-0.3 - 16.5 V
Voltage Supply	5 - 11 V
Differential Measurement Signal	315 LSB/g
Noise (rms)	6.7 mg

Table 4.3: Accelerometer Specifications

4.3.2 Instrumentation Sensors

Once we get signals from the vehicle sensors, we need the set of instrumentation device which are more precise that detects and measures a physical parameter, and converts it into an electrical signal that can be read by an instrument or control system. Following table shows the set of instrumentation sensors that might be used to get those signals and help them reach to data

acquisition system so we can analyse the data and henceforth, tune the parameters using data driven engineering approach.

Sensor	Quantity	Position/Use
Accelerometer	1	knuckle/acceleration and translation on the knuckle
Accelerometer	1	body/acceleration and translation on the sprung mass
Accelerometer	1	On contact patch
String encoders	1	To find body position
String encoders	1	To find damper velocity
Strain gauge	1-2	Dependent on intended testing
Load Cell	1	Measure forces from Spring strut Top Mount to body
Load Cell	1	Measure contact force tire to ground (Poster plate)
Camera / Optical Sensors	1-2	To measure position of sprung and unsprung together

Table 4.4: Instrumentation Sensors

In this HIL rig, three accelerometers will be used in order to capture the acceleration of the chassis(body/sprung mass), knuckle and the contact patch.

Additionally, a strain gauge based load cells will be used, one below the wheel to measure the wheel load , while other between sprung body and spring strut top mount. For the measurement of the suspension deflection a wire rope actuated position transducer (WRAPT) with a digital incremental encoder might be used.

5

Results

In this section, the results of virtual model of the quarter car dynamic HIL rig are presented by performing general actuation analysis. The Analysis involves fourpost as simulation type as we will be using one poster for the physical rig. It involves running couple of interesting road profiles on Adams Car to understand the influence on ride comfort and driving dynamics by plotting the displacement of knuckle and body versus time.

5.1 Long Dip

Initially to check the effect on ride comfort, our first step is to see how model reacts to simple road profile. The analysis started by running a simple long dip road profile. As seen in figure 5.1, displacement versus time graph is plotted.

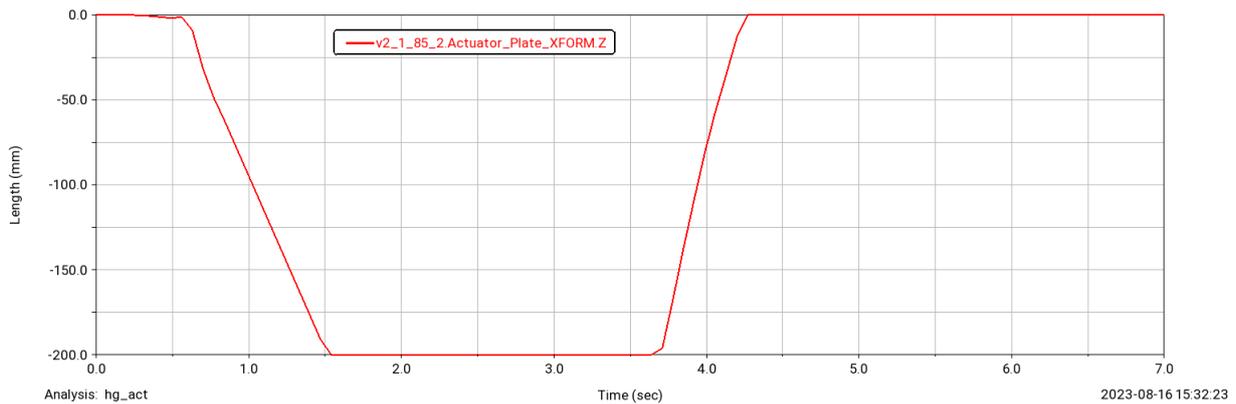


Figure 5.1: Displacement of an Actuator

Then, we plotted displacement of knuckle against time to check if the model is correctly working or not, figure 5.2. As knuckle is located relative to tire, we can assume that there should not be much difference in the displacement of knuckle. Figure 5.2 represents the similar behaviour as there is no huge difference between displacement of knuckle and tire.

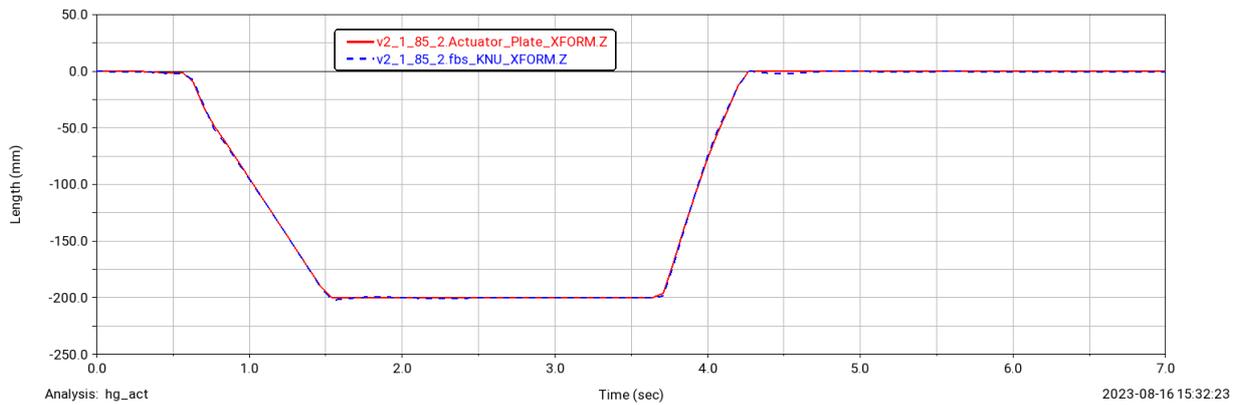


Figure 5.2: Displacement of Actuator and KNU

Similarly, to understand the influence of long dip on ride comfort, displacement of the sprung body is plotted against the time, figure 5.3. As you can observe during the starting phase of plot, there is small delay between the movement of a body and a tire. Also, between 1.5 sec to 2 sec, when a tire is on the ground, body is compressed and after some oscillations it again follows the road profile. During end phase of the plot when a tire reaches to the ground, you can observe the significant displacement of body imitating the real world phenomenon.

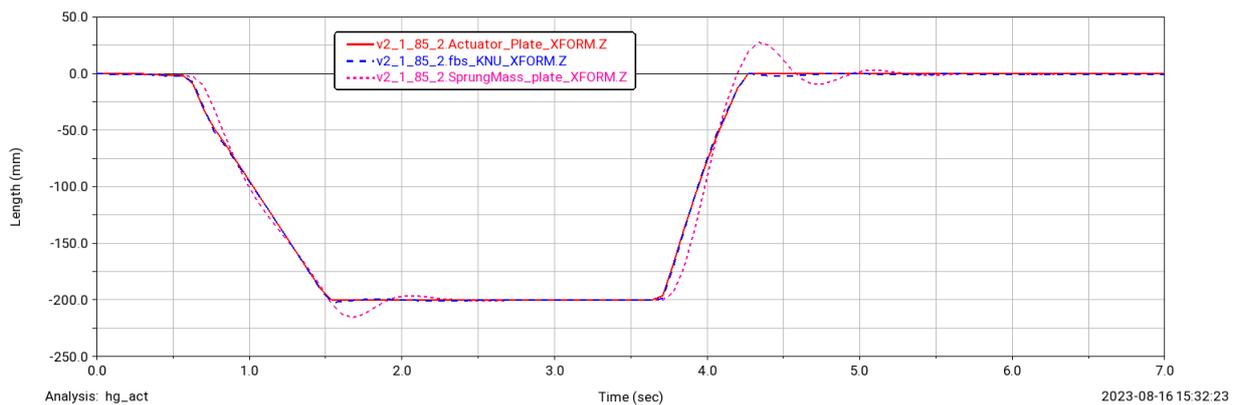


Figure 5.3: Displacement of Actuator, KNU, and Sprung Mass

To make sure that the model is adapting to different road profile changes and understand the influence more clearly, I tried to change the end slope of long dip by making it more steeper. As seen in figure 5.4, the amplitude of displacement is large for a steeper slope compared to previous slope which is similar case in real cars too.

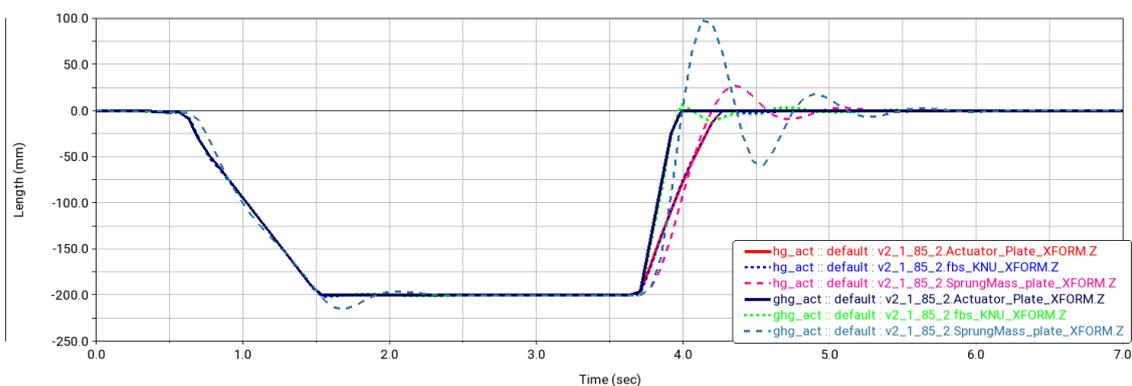


Figure 5.4: Displacement change due to steeper slope

5.2 Barfield Track

After making sure that the model is adapting to small changes in road profile as simple as long dip, our next step is to check the influence on ride comfort with more complex road profile named 'Barfield Track'. The displacement data received about barfield road profile is from the country side track from Hällered.

The analysis started by sending a displacement of barfield road profile as an input to the actuator. As seen in figure 5.5, displacement versus time graph is plotted.

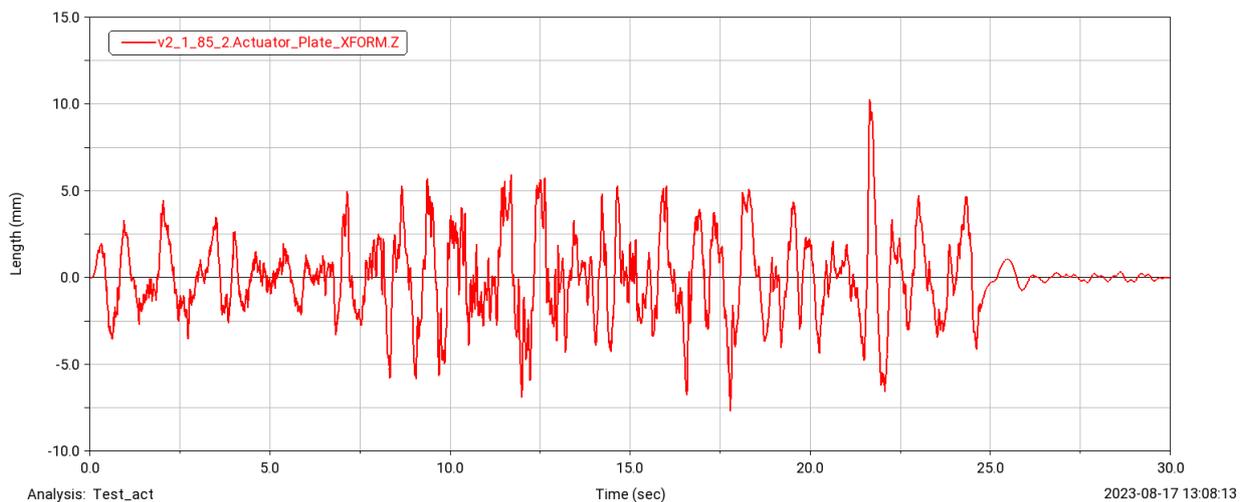


Figure 5.5: Displacement of an Actuator

As mentioned before, to check if the system is reacting to various excitations from road profile, displacement of knuckle is plotted, figure 5.6. As expected, a small variation is there in the knuckle movement when we compare displacements of both knuckle and tire. This actually shows that the system in the model is adapting to road behavioural changes.

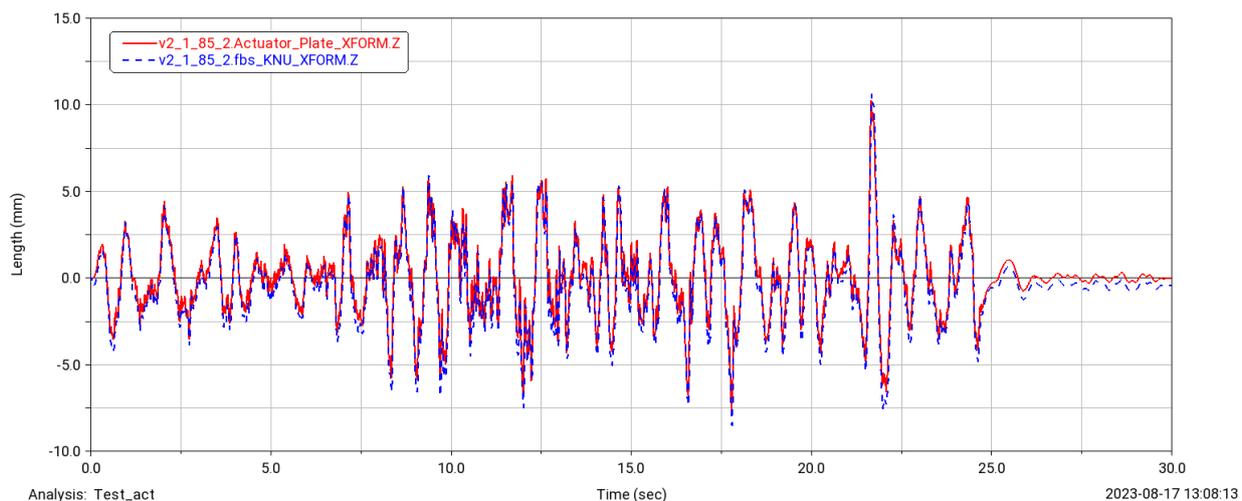


Figure 5.6: Displacement of Actuator and KNU

From figure 5.7, we can study the movement of sprung body to see how it reacts to this undulations from barfield track. As expected, the amplitude of sprung body goes beyond the certain values due to inertial effects compared to knuckle displacement which shows that the whole model is following the real world phenomenon.

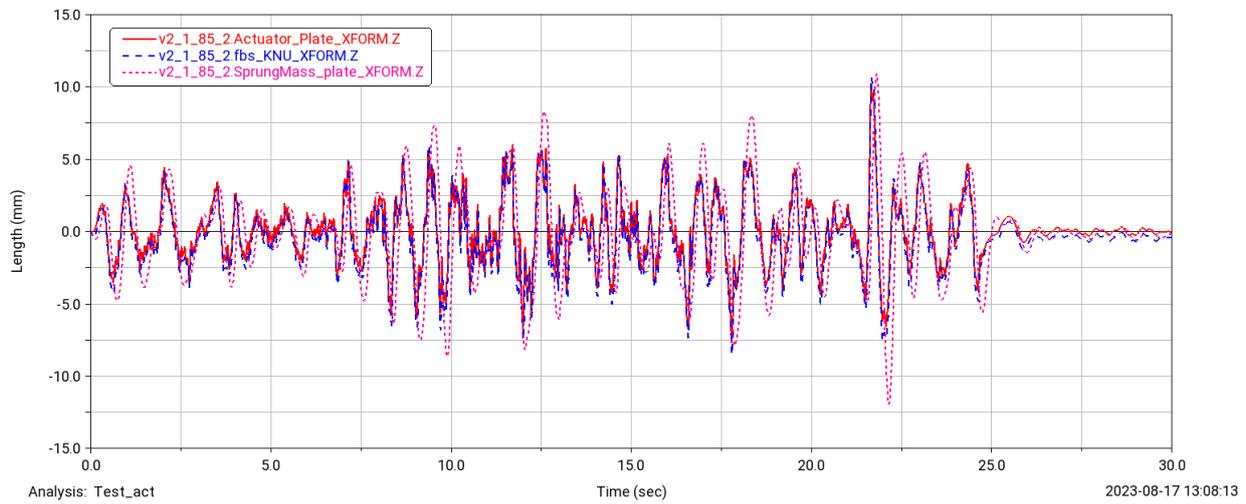


Figure 5.7: Displacement of Actuator, KNU, and Sprung Mass

Also, to get more clear understanding of how the spring-damper system is working in this model, we take a closer look at the above plot for first 5 seconds. From figure 5.8, we can observe that the sprung body displacement has isolated all small undulations/excitations coming from the road which proves that the spring-damper system is able to isolate all the unnecessary variations.

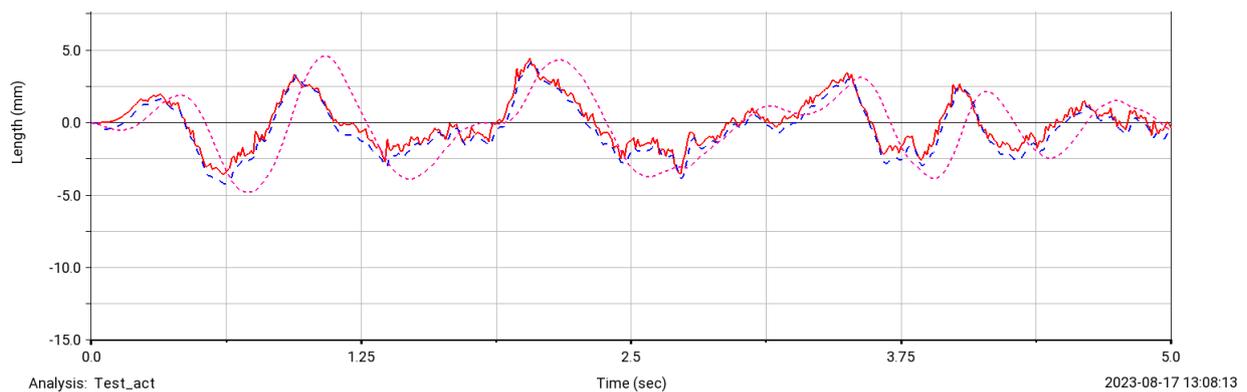


Figure 5.8: Displacement for first 5 seconds in figure 5.7

6

Conclusion

In conclusion, the results of this thesis highlight that the simulation model gives credible results and corresponds to real world phenomenon. We can perform different road profiles on quarter car dynamic HIL rig to help us achieve good balance between ride comfort and handling, if the (complete vehicle requirements on) comfort and handling are well broken down to requirements on how a single wheel should behave. This rig is the good way to make products more mature during development phase which might help us reduce bit of time in product development considering how fast paced the automotive world is!

As a first setup on quarter car dynamic HIL rig, a fully active suspension system will be used. Modularity is crucial in every system where we should be able to assemble and reuse some components to develop different finished product. CAD model shows that the rig is modular by designing unique extensions for each suspension application and addition of extra sprung mass holder incase there is a heavier vehicle in future.

Apart from In Car sensors, various instrumentation sensors used will measure accurate signals. With the help of calibration of control algorithms including calibrating parameter values such as control gains etc, we can tune the components to achieve a better performance and results. The simulation model on Adams could be used to tune various passive components of suspension system such as spring, damper. Graph 5.8 shows that the sprung body has isolated all the excitations from road profile. It shows that our model is working correctly, and spring-damper system is able to reduce all the undulations to achieve good ride comfort.

With the help of quarter car dynamic HIL rig, function development of the controller can be done for a FAS system. For instance, We can tune minimum damping value to minimize the wheel hop. On a controller side, we can add as much body control as we want. If we make sure that the wheel is on the ground and is not bouncing a lot, then we can tune how much body control we want on the car e.g. some customers or brand would like to have floaty feeling on the car but Volvo are known for more controlled feeling of the car. Additionally, we can tune the controller for CCD with 2 valves (Jounce and Rebound).

These findings highlight the significance of early phase testing of a product/component, system validation, and meticulous selection of requirement to make products more mature in early phases as well as achieve good balance between ride comfort and handling.

7

Future Work

Currently, the methodology for building a modular quarter car dynamic HIL rig is established for a Five link suspension, a Four link suspension, and an Integral link suspension system. This involves Adams simulation model which can help track the wheel center movement, knuckle as well as sprung body movement under the influence of various maneuvers, tune dampers and springs to minimize wheel hopping, etc.

This thesis work sheds light on the ride comfort and driving dynamics by running tests on a quarter car dynamic HIL rig for low range frequency, specifically up to 20 HZ. However, the challenges associated with operating the system at high range frequencies are evident, encompassing issues such wheel hopping, excess body movement during certain road maneuvers, and the need for component designs capable of meeting high-load requirements.

One of the limitations of this study is the rig not emulating the effect from longitudinal tire forces. A future work could be to further develop the test rig to understand the influence of longitudinal tire forces along with vertical movement on ride comfort as well as behavior and analyze how the wheel rate and frequency would affect the overall vehicle dynamics. Other dynamic parameters could be related to the ride comfort and different road profiles can be developed to tune the body controllers for better vehicle dynamics. Another possibility might be to see what it would be if we start adding the torque effect, remove air cushion between tire and ground if there's any so we can get the side forces and possibly longitudinal forces when you propel and brake.

Another interesting future work could be to emulate the steady state cornering by assuming the available wheel from quarter car rig either as inner wheel or outer wheel on an axle to understand the influence of lateral forces.

If we somehow manage to mimic the rest of the vehicle, either by cloning the rig to emulate the other axle or by adding a virtual dynamic model of a complete vehicle. Then we can understand the influence of pitch. Also, now we run the simulation as a function of longitudinal speed and if we try to incorporate wheelbase, then we can get the data related to front axle before even we go to rear axle. Additionally, a control of rear axle which utilizes sensed motion of the front axle would not be possible without mimicking the rest of the vehicle in some way. So, how well we can mimic signals from other axle towards controller could be a good work in future.

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