Development of semi-active control system for hydraulic dampers

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Abstract

Hydraulic dampers are most widely used in the suspension system on trains and heavy vehicles. However, they suffer from a major drawback resulting from the change in characteristics of this device at high frequency, i.e. the equivalent stiffness of the damper increases considerably at high frequency along with a substantial decrease in the equivalent damping. This effect leads to increase in the transmitted force at high frequency excitations, which has a negative effect on the ride quality. To mitigate this effect, semi-active control strategies for hydraulic dampers have been developed and tested.

This study evaluates the dynamic response of four semi-active control strategies as tested on a numerical model of a quarter-train system and an experimental setup to test the performance of a single hydraulic damper. Skyhook control strategy, which is most commonly used control strategy in order to enhance comfort, is used as a reference for comparison. Each control strategy is evaluated for its performance under different base excitations and RMS acceleration on the chassis is used as a performance evaluation criterion as it is the most commonly used parameter to evaluate comfort.
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Introduction

Actively controlled suspensions have been widely studied theoretically and experimentally for automobiles and rail vehicles. These studies have demonstrated significant performance benefits. For railway vehicles, active suspensions that deal primarily with improvement in ride quality are now starting to be incorporated on a regular basis. Furthermore railway engineers are also interested in using active control to achieve a functionality that is not possible with a purely passive (mechanical) suspension. [1]

The present railway operators are facing intense pressure due to competition from the air and road transport industries. In order for them to operate more efficiently and effectively, they are investigating ways to reduce the traveling time, increase the number of passengers and reduce operating costs. The trend of faster and lighter vehicles are ways in which these objectives can be met. Therefore trains of the future will be designed with lighter bodies and bogies to enable them to operate at a much higher speed. But as the speed of the train is increased, it will lead to more high-frequency vibrations which will certainly affect the ride quality and generate unacceptable levels of internal noise.

The main objective of the project is to improve the high frequency characteristics of a railway secondary suspension system, by incorporating an electronic control system. By reducing the vibration at high frequencies, it is possible to enhance the ride comfort for the passengers. There are conventional control strategies, already available in literature to attain this objective, for example, a Skyhook control loop. In this project, other possible control strategies are studied and compared with a conventional skyhook strategy on the basis of the most relevant performance criterion to determine the best
solution for achieving our objective.

In the first chapter, "State of the art", we discuss about the design of a suspension system, the various constraints and limitations. Then we talk about how to manage these constraints using an actively controlled suspension, conventional control strategies i.e. skyhook, different types of active suspensions, their relative advantages and disadvantages, the latest technological advancements and research in the field of active suspensions and also about the most commonly used parameter to judge the performance of a suspension system.

Then in the second chapter, "Model of the passive damper", we describe the numerical model of the passive damper using various mathematical equations and formulas. This model is used for numerical simulation using Matlab and Simulink. Then we describe the Experimental setup used for characterization of the passive damper and also for validation of the numerical model.

In the third chapter, "Development of the semi-active damper", we discuss the numerical model for the semi-active damper and the Quarter-Train model used for analysis purpose. Then we introduce 4 different control strategies which we designed, as explained with the help of block diagrams and schematics showing how they can be implemented actually in a physical model. Then we discuss the Input excitation methods used and the results from the Quarter - Train model with semi-active damper. We also describe the method adopted for determination of the correlation curve between frequency and required valve opening.

In the fourth chapter, "Prototype of the semi-active damper", we describe the physical prototype of the semi-active damper which we created for physical testing of the proposed solution. We talk about the experimental setup created for testing and evaluation of the physical prototype and the results from the testing compared with numerical model.

Then in the last chapter, we summarize all our observations and findings from various numerical and physical tests conducted, comparison between different control strategies and try to find the best possible solution.
Chapter 1

State of the Art

The automotive suspension is a compromise between the conflicting demand of ride handling and vehicle stability [2]. The primary function of a suspension is to isolate the occupants of the vehicle from the accelerations generated in the vehicle by track irregularities. These irregularities arise as a result of construction tolerances, and the degradation which occurs in the track due to use and exposure to the elements. While the original need to isolate the vehicle from track irregularities is to enhance the ride comfort of the passengers, there is an equal need to protect the track from the impact of the vehicles using it as this can lead to rapid deterioration and high maintenance costs.

When a suspension is isolating a vehicle from track irregularities, it must also allow the vehicle to follow intentional long wavelength features in the track, where it is following contours in the landscape. Thus a suspension is a low pass filter which enables a vehicle to follow the long wavelength features that are designed into the track, while isolating it from the short wavelength features which arise as a result of construction tolerances and wear and tear.

In order to provide maximum isolation in a vehicle it is necessary to allow the wheels to follow the vertical profile of the road while the vehicle body remains at a fixed height in space. In practice the body cannot remain at a fixed height in space as this produces a requirement for infinite suspension travel. Consequently the degree of isolation produced by a suspension is a function of the amount of suspension travel available.
1.1 Design Constraints

When designing a suspension the two major variables available to the suspension designer are damping rate and spring rate. For estimation of comfort level of the passengers, body acceleration is considered the best indicator [3] [4]. Fig.1.1 shows the effect of varying the damping rate from the nominal values appropriate to a train, when it is driven over a track with a profile given in [2] [5]. At low damping rates there is a large peak in the body acceleration at the frequency of chassis mode (1 Hz), and a smaller one at the frequency of bogie mode (8 Hz). Increasing the damping rate, reduces the acceleration at frequencies of chassis mode while increasing it at high frequencies, creating an optimum damping rate for minimum acceleration. If the damping rate becomes very high, the acceleration at chassis mode reduces and at bogie mode becomes high. Consequently, in order to ensure a good ride, the damping rate has to be a compromise.

![Figure 1.1: Effect of damper rate on body acceleration](image)

Also, for a fixed damper with chosen optimized parameters, the stiffness and damping changes with the frequency of the vibration input. While stiffness increases with increase in frequency, damping rate reduces as shown in Fig 1.2, therefore the energy transmitted to the chassis increases at higher frequencies, so at higher train speeds. This can be quantified in form of a
1.1. DESIGN CONSTRAINTS

parameter called *Transmissibility*, defined by eq.1.1

\[ \tau = \sqrt{k^2 + \omega^2 r^2} \]  

where,

\begin{align*}
    k &= \text{Stiffness of the damper} \\
    \omega &= \text{Frequency of vibration} \\
    r &= \text{Damping}
\end{align*}

Transmissibility increases with increase in frequency as shown in Fig.1.3 The main aim of the project is to reduce the energy transmitted to the chassis at high frequencies, therefore to reduce the transmissibility.

![Figure 1.2: Effect of frequency on stiffness and damping rate](image)

![Figure 1.3: Variation of transmissibility with frequency in a passive damper](image)
This is where the need of an active suspension is realized. An active suspension provides us the flexibility of choosing the parameters which are best suited for the different situations rather than having a single optimized value. The different types of active suspension systems are discussed in detail in the next section along with their respective advantages and disadvantages.

1.2 Active Suspensions

An active suspension involves addition of sensors, a controller and actuators to an existing mechanical system, as shown in Fig.1.4 to obtain the desired performance characteristics in different conditions. In an uncontrolled or passive system, response to track inputs or other disturbances is defined by parameters such as masses, spring rates, damping constants, geometrical properties etc., however in a controlled or active system, these parameters are altered as per the control law, which ultimately governs the system response to input. Although it is not strictly necessary but usually the system is arranged to give feedback action and this can be used to give a fundamental modification to the basic dynamical properties of the passive system.

It is important to appreciate that, while active suspensions have often been portrayed as giving "better" performance (e.g., a more comfortable
1.3 Classification

Active Suspensions have a very vast field and there is a great variety of opportunities and possibilities, therefore it is useful to provide some sort of classification to identify the distinctness of some of the issues and also to emphasize the control options which are available. Three broad classification categories are introduced - the scope, the degree, and the technology of control. [1]

Scope of control identifies the modes of the suspension which are being controlled by the active suspension. Note that, due to the interactive dynamic nature of a railway vehicle, a number of modes may be influenced by a particular design, but this refers to those modes whose response is deliberately being addressed by the control action.

Degree of control defines the type and dynamic range of the control action. Two broad categories in this respect are Semi-Active and Full-Active control as shown in Fig. 1.5.
In semi-active control the characteristics of an otherwise passive element (mostly a damper, but not inevitably so) are varied rapidly on the basis of measurements of variables within the dynamic system and the control strategy employed. Closely related is an option known variously as semi-passive, adjustable passive and adaptive, in which the characteristics are varied on the basis of a variable which is not influenced by the dynamic system being controlled - a good example is varying the rate of the damper as a function of vehicle speed.

In full-active control, actuators are used which are either to completely replace or augment passive elements, and the force they produce is varied according to measurements from the system and control law. A full-active system must have an external source of power to supply the actuators (in addition to low level power supplies associated with the control electronics which are inevitable with any form of active suspension). There is another aspect to the degree of control, primarily applicable to full-active suspensions, which can broadly be subdivided into low bandwidth or high bandwidth. In low bandwidth systems there will be passive elements which dictate the fundamental dynamic response, and the function of the active element is associated with some low frequency activity such as leveling or centering. This restriction enables
some reduction in force and/or velocity requirement for the actuators. By contrast, high bandwidth active systems have a much enhanced capability, and the overall dynamic response will primarily be determined by the active control strategy, which will probably act throughout the frequency range which is relevant to the particular suspension function being controlled.

A semi active suspension is a valid engineering solution when it can reasonably approximate the performance of the active control because it requires a low power controller that can be easily realized at a lower cost than that of a fully active one. Note, however, that a semi active system clearly lacks other important secondary advantages of the fully active one, namely the ability to resist downward static forces due to passenger and baggage loads and to control the altitude of the vehicle. [6]

Technology of Control concerns the practicalities of implementation, and strictly covers the controller, the sensors and the actuators. The availability of remarkable quantities of computing power means that the controller is unlikely to be a limiting factor in the implementation, although issues such as reliability, ruggedness etc. cannot be ignored. The sensors are probably a significantly more important issue, but a large variety of suitable types is available, and the key aspects here relate to reliability and cost. The provision of high reliability actuation of sufficient performance is one of the main challenges in active suspensions. Cost of the total system (i.e., including power sources, etc.) is certainly important but ease of installation (compactness, etc.), maintainability and maintenance cost, reliability and failure modes must all have essential inputs into the process of choosing and procuring the actuator system. Actuator technologies which are possible for active suspensions are servo-hydraulic, servo-pneumatic, electro-mechanical and electro-magnetic. Servo-hydraulic actuators themselves are compact and easy to fit, but the whole system tends to be bulky and inefficient, and there are important questions relating to maintainability. Pneumatic actuators are a possibility, particularly since the airsprings fitted to many railway vehicles can form the basic actuator, but inefficiency and limited controllability are important restrictions. Electro-mechanical actuators offer a technology with which the railway is generally familiar, and the
availability of high performance servo-motors and high efficiency power electronics are favorable indicators. However they tend to be less compact and the reliability and life of the mechanical components needs careful consideration. Electro-magnetic actuators potentially offer an extremely high reliability and high performance solution, but they tend to be quite bulky and have a somewhat limited travel.

1.4 Types of Active/Semi-Active Dampers

1.4.1 Active Dampers

Hydraulic Actuated

Hydraulically actuated suspensions are controlled with the use of hydraulic servomechanisms. The hydraulic pressure to the servos is supplied by a high pressure radial piston hydraulic pump. Sensors continually monitor body movement and vehicle ride level, constantly supplying the computer with new data. As the computer receives and processes data, it operates the hydraulic servos, mounted beside each wheel. Almost instantly, the servo-regulated suspension generates counter forces to body vibrations. In practice, the system always have the desirable self-leveling and height adjusting features, with the latter now tied to vehicle speed for improved aerodynamic performance, as the vehicle lowers itself at high speed.

Benefits:

- Better ride quality specially on highly undulated surface as it is very effective in controlling Roll motion.
- High damping forces can be obtained.
- Ease of design and control.

Drawbacks
1.4. TYPES OF ACTIVE/SEMI-ACTIVE DAMPERS

- Usually operates in low frequency bandwidth and has large power requirements.
- Requirement of continuously pressurized system.
- Mass and intractable space requirements of the total system, including supply system, even though it mainly contributes to the sprung mass.

![Figure 1.6: Suspension system with hydraulically actuated dampers](image)

**Electromagnetic**

This is possibly the most advanced Active suspension system and a subject of latest research. This type of active suspension uses linear electromagnetic motors attached to each wheel. It consists of sets of permanent magnet and series of current coil. Each of them can either act as translator or static armature. The interaction between magnetic field of permanent magnet and magnetic field of electrically powered armature winding will cause movement or thrust on the translator. This thrust acts opposite to the body vibrations. It provides extremely fast response and also allows regeneration of power consumed, by using the motors as generators. This nearly surmounts the issues of slow response times and high power consumption of hydraulic systems. [7] [8] [9]
Benefits:

- Extremely fast response.
- Low power consumption due to regeneration and large frequency bandwidth.
- Higher efficiency
- Improved stability and dynamic behavior
- Accurate force control;

Drawbacks

- Relatively complex and expensive system.

Figure 1.7: Electro-mechanical Active damper

1.4.2 Semi-Active Dampers

Continuously Controlled Damping (CCD)

CDC is an electronic damping system, where a control unit takes the input from various sensors measuring body, wheel and lateral acceleration and use
1.4. TYPES OF ACTIVE/SEMI-ACTIVE DAMPERS

this data to continuously calculates the requisite damping forces for each individual wheel. The damper is adjusted as per this required value through a proportionating valve which opens or closes to alter the amount of damping force required. This system noticeably increases driving safety, comfort, and dynamics and also provides an option to the driver to choose between different driving modes such as *Comfort* (soft damping) and *Sport* (firm damping).

Benefits:

- Greater safety thanks to optimized wheel damping.
- Enhanced driving comfort and dynamics.
- Reduced roll, pitch, and vertical motion.
- Continuous adjustment in real time.

Figure 1.8: CDC Damper with external proportionating valve

**Magneto-Rheological Damper (MR)**

A magneto-rheological damper or magnetorheological shock absorber is a damper filled with magnetorheological fluid, which has a unique property that it’s viscosity changes under the effect of a magnetic field due to the presence
CHAPTER 1. STATE OF THE ART

of magnetic particles. The magnetic field is usually introduced by passing electric current through the coils inside the damper which acts as an electromagnet. This allows the damping characteristics of the shock absorber to be continuously controlled by varying the power of the electromagnet [10].

Magneto-rheological dampers are characterized by their extremely quick response, therefore they are ideal for applications where response time is of prime importance such as for improving comfort in a passenger car as damper characteristics can be varied quickly in response to road irregularities.

![Figure 1.9: Construction of a basic MR damper](image)

**Solenoid valve**

This is the most basic and economic type of Semi-Active damper. It consists of a solenoid valve which alters the flow of the hydraulic medium inside the shock absorber, therefore changing the damping characteristics of the suspension setup. The solenoids are wired to the electronic control unit, which sends them commands depending on the control algorithm.
1.5 Conventional control strategy

1.5.1 Skyhook Control

As the name suggests, the skyhook configuration shown in Fig 1.11. has a damper connected to some inertial reference in the sky. With the skyhook configuration, the trade off between resonance control and high-frequency isolation, common in passive suspensions, is eliminated. Notice that skyhook control focuses on the sprung mass; as $c_{\text{sky}}$ increases, the sprung mass motion decreases. This, of course, comes at a cost. The skyhook configuration excels at reducing the effect of base excitations on the sprung mass but at the expense of increased unsprung mass motion. [11]
CHAPTER 1. STATE OF THE ART

Figure 1.11: Skyhook damper configuration

Figure 1.12: Skyhook Configuration Transmissibility: (a) Sprung Mass Transmissibility; (b) Unsprung Mass Transmissibility
1.5. **COVENITIONAL CONTROL STRATEGY**

The transmissibility for this system is shown in Fig 1.12. for different values of the skyhook-damping coefficient, $c_{sky}$. Notice that as the skyhook damping ratio increases, the resonant transmissibility near $\omega_{n1}$ decreases, even to the point of isolation, but the transmissibility near $\omega_{n2}$ increases. In essence, this skyhook configuration is adding more damping to the sprung mass and taking away damping from the unsprung mass. The skyhook configuration is ideal if the primary goal is isolating the sprung mass from base excitations, even at the expense of excessive unsprung mass motion. An additional benefit is apparent in the frequency range between the two natural frequencies. With the skyhook configuration, isolation in this region actually increases with increasing $c_{sky}$.

Because this damper configuration is not possible in realistic automotive applications, a controllable damper is often used to achieve a similar response to the system modeled in Fig 1.13. The semi-active damper is commanded such that it acts like a damper connected to an inertial reference in the sky. Fig 1.13. shows the semi-active equivalent model with the use of a semi-active damper.

![Figure 1.13: Semi active equivalent model](image)
Several methods exist for representing the equivalent skyhook damping force with the configuration shown in Fig 1.13. Perhaps the most comprehensive way to arrive at the equivalent skyhook damping force is to examine the forces on the sprung mass under several conditions. First, let us define certain parameters and conventions that will be used throughout controller development. Referring to Fig 1.13., the relative velocity, \( v_{12} = (v_1 - v_2) \), is defined as the velocity of the sprung mass \( m_1 \) relative to the unsprung mass \( m_2 \). When the two masses are separating, \( v_{12} \) is positive. For all other cases, up is positive and down is negative.

Now, with these definitions, let us consider the case when the sprung mass is moving upwards and the two masses are separating. Under the ideal skyhook configuration we find that the force due to the skyhook damper is

\[
F_{sky} = -c_{sky} v_1
\]  

(1.2)

where \( F_{sky} \) is the skyhook damping force. Next we examine the semi-active equivalent model and find that the damper is in tension and the damping force due to the semi-active damper is

\[
F_{sa} = -c_{sa} v_{12}
\]  

(1.3)

where \( F_{sa} \) is the semi-active damping force. Now, in order for the semi-active equivalent model to perform like the skyhook model, the damping forces must be equal, or

\[
F_{sky} = -c_{sky} v_1 = -c_{sa} v_{12} = F_{sa}
\]  

(1.4)

We can solve for the semi-active damping in terms of the skyhook damping (eq 1.5.) and use this to find the semi-active damping force needed to represent skyhook damping when both \( v_1 \) and \( v_{12} \) are positive (eq 1.6.)

\[
c_{sa} = \frac{c_{sky} v_1}{v_{12}}
\]  

(1.5)

\[
F_{sa} = c_{sky} v_1
\]  

(1.6)

Next, let us consider the case when both \( v_1 \) and \( v_{12} \) are negative. Now the sprung mass is moving down and the two masses are coming together. In this scenario, the skyhook damping force would be in the positive direction, or

\[
F_{sky} = c_{sky} v_1
\]  

(1.7)
Likewise, because the semi-active damper is in compression, the force due to the semi-active damper is also positive, or

\[ F_{sa} = c_{sa}v_{12} \]  

(1.8)

Following the same procedure as the first case, equating the damping forces reveals the same semi-active damping force as the first case. Thus, we can conclude that when the product of the two velocities is positive, the semi-active force is defined by eq 1.6.

Now consider the case when the sprung mass is moving upwards and the two masses are coming together. The skyhook damper would again apply a force on the sprung mass in the negative direction. In this case, the semi-active damper is in compression and cannot apply a force in the same direction as the skyhook damper. For this reason, we would want to minimize the damping, thus minimizing the force on the sprung mass.

The final case to consider is the case when the sprung mass is moving downwards and the two masses are separating. Again, under this condition the skyhook damping force and the semi-active damping force are not in the same direction. The skyhook damping force would be in the positive direction, while the semi-active damping force would be in the negative direction. The best that can be achieved is to minimize the damping in the semi-active damper.

Summarizing these four conditions, we arrive at the well-known semi-active skyhook control policy:

\[
\begin{align*}
    v_1v_{12} &\geq 0, & F_{sa} &= c_{sky}v_1 \\
    v_1v_{12} &< 0, & F_{sa} &= 0
\end{align*}
\]  

(1.9)

It is worth emphasizing that when the product of the two velocities is positive that the semi-active damping force is proportional to the velocity of the sprung mass. Otherwise, the semi-active damping force is at a minimum.

It is preferable to add another component to the damping force provided by the actuator that is related to the passive behavior of the damper and is proportional to the relative speed of the chassis with respect to the chassis, described in eq 1.10. This component is added as a precautionary measure to provide damping force in case of a breakdown and to also damp
the vertical motion of the unsprung mass as skyhook only damps the chassis modes.

\[ F = r(v_1 - v_2) \]  \hspace{1cm} (1.10)

where,

\[ v_1, v_2 \] are velocity of chassis and bogie respectively

\[ r \] is the damping rate for the passive damper.

Therefore, the total damping force required from the actuator is

\[
\begin{cases} 
  v_1v_{12} \geq 0, & F_{sa} = r(v_1 - v_2) + c_{sky}v_1 \\
  v_1v_{12} < 0, & F_{sa} = r(v_1 - v_2) 
\end{cases}
\]  \hspace{1cm} (1.11)
Chapter 2

Model of Passive Damper

2.1 Numerical Model

A numerical model of the damper is required as it enables us to do our preliminary study about the damper characteristics, its behavior under different parameters such as displacement amplitude and frequency of input excitation. It also provides a feasible and more efficient way of testing and analyzing our control strategies through numerical simulation rather than depending upon experimental setup always. In this manner, we can predict errors at earlier stages and we can also perform more advanced analysis on a Quarter-train model which is complex to replicate experimentally.

The Hydraulic damper can have different configurations as described below [12]:-

1. Single Acting piston:

![Figure 2.1: Single-Acting piston type hydraulic damper](image)

Figure 2.1: Single-Acting piston type hydraulic damper
Here pressure is applied only from one side and return motion is because of a spring. Piston force $F$ is calculated as:-

$$F = pA - Kx \quad (2.1)$$

2. Double Acting piston - single rod: In this type, forward and reverse motions are provided by pressure on opposite faces of the piston but volume of chambers is asymmetric as rod is present on just one side of the piston.

$$F = p_1A_1 - p_2A_2 \quad (2.2)$$

![Double acting piston - single rod type hydraulic damper](image)

3. Double Acting piston - double rod: This is same as previous type but volume of chambers is also symmetric as rod is present on both sides of the piston.

$$F = (p_1 - p_2)A \quad (2.3)$$

![Double acting piston - double rod type hydraulic damper](image)

For our model we decided to use an Asymmetrical bidirectional hydraulic damper(double acting piston - single rod). This type of hydraulic damper offers asymmetric characteristics in compression and expansion
due to difference in the volume of the two main chambers, but is lighter in construction and more compact. Therefore, considering the need of a physical prototype at later stage, this is an effective solution.

A hydraulic damper can be broadly described as a cylindrical part as shown in figure 2.4, with two main chambers (chamber 1 and chamber 2) filled with a viscous fluid and separated by a piston with an internal valve (Valve A) to regulate the passage of fluid from one chamber to other. The flow of fluid through a valve is regulated as per the valve characteristics in compression and expansion and on the pressure difference between the 2 chambers.

Other than two main chambers, there is another chamber connected to main chamber by valve B which accounts for flow of fluid around the rod due to leakage and there is also a reserve chamber connected by valve C which holds extra reserve of fluid. All these valves have different characteristics depending upon their design. In this section we describe the numerical model of the hydraulic damper through mathematical equations and formula.

Figure 2.4: Model of a hydraulic damper
$V_1$, $V_2$ are the volumes of chamber 1 and 2

$p_1$, $p_2$ are the fluid pressures inside chamber 1 and 2

$V_{01}$ and $V_{02}$ are the initial volumes of the 2 chambers

$m_p$ is the mass of the piston

$m_r$ is the mass of the piston rod

$A_p$ is the area of piston

$A_r$ is the area of the piston rod

$z_G$, $z_E$ and $z_F$ are position of points G, E and F from reference

$h_p$ is the thickness of the piston

$Q_A$, $Q_B$ and $Q_C$ are fluid flow rates through valve A,B and C.

Pressure dynamics in compression chamber (chamber 1) and rebound chamber (chamber 2) can be explained by the equations below [13][12]:

\[
\frac{V_1}{\beta_1} \dot{p}_1 = -\dot{V}_1 - Q_A + Q_C 
\]  

(2.4)

\[
\frac{V_2}{\beta_2} \dot{p}_2 = -\dot{V}_2 + Q_A + Q_B
\]  

(2.5)

where,

$\beta_1$, $\beta_2 = \beta$ is isothermal bulk modulus of the fluid, this parameter defines compressibility of the fluid, higher the value of $\beta$, lower is the compressibility.

volumes of chambers are:

\[
V_1 = V_{01} + A_p(z_G - z_F - \frac{h_p}{2})
\]  

(2.6)

\[
V_2 = V_{02} + (A_p - A_r)(z_E - z_G + \frac{h_p}{2})
\]  

(2.7)
2.1. NUMERICAL MODEL

volumes time derivatives:

\[ \dot{V}_1 = A_p (\dot{z}_G - \dot{z}_F) \]  
\[ (2.8) \]

\[ \dot{V}_2 = (A_p - A_r) (\dot{z}_E - \dot{z}_G) \]  
\[ (2.9) \]

piston force

\[ F_{amm} = -(p_1 A_p - p_2 (A_p - A_r)) - (m_p + m_r) \ddot{z}_G \]  
\[ (2.10) \]

The flow through the valves is expected to be turbulent due to high viscosity of fluid and fast excitation due to road irregularities which increases flow speed. Turbulent flow A and C are evaluated as:

\[ Q = c_q A_v \sqrt{\frac{2|\Delta p|}{\rho}} \cdot \text{sign}(\Delta p) \]  
\[ (2.11) \]

where

\[ c_q = c_{q,max} \cdot \tanh\left(\frac{2\lambda}{\lambda_{\text{crit}}}\right) \]  
\[ (2.12) \]

\[ \lambda = \frac{h_d}{\nu} \sqrt{\frac{2\Delta p}{\rho}}, \quad h_d = \sqrt{\frac{4A_v}{\pi}} \]  
\[ (2.13) \]

\( \lambda_{\text{crit}} \) is the critical flow number at which transfer from laminar to turbulent characteristic occurs.

\( c_q \) is the discharge coefficient

\( A_v \) is the minimum area of the orifice cross section

2.1.1 Valve A

Valve A is the valve which allow the flow through the piston as shown in Figure 2.5. Inside the piston there are 2 holes 3 mm diameter for the flow passing from compression chamber to rebound chamber and 2 holes 3 mm diameter for the counter flow. There is a constant area which allows the flow in both directions, this area is represented by small grooves machines into the bottom washer. There are four grooves that are 1 mm wide and 0.3 mm deep. The flow area is assumed to be the sum of these rectangular grooves.

\[ A_{vA0} = n_g h_g w_g = 1.2 \text{mm}^2 \]  
\[ (2.14) \]
where,

\[ n_g \] is the number of grooves.

\[ h_g \] is the depth of the grooves.

\[ w_g \] is the width of the grooves.

When the damper is in compression stroke, the difference of pressure between compression chamber (chamber 1) and rebound chamber (chamber 2) makes the top washer move upward compressing the spring. In this case the flow area is a circular surface whose perimeter is the product of the circumference of flow diameter \( d_{\text{flow}} = 25 \text{ mm} \) times the lift of the valve which depends on pressure gap.

\[ A_{v_{Al}} = \pi d_{\text{flow}} l \]

(2.15)

where,

\[ l \] is the lift of the valve.

The maximum lift is equal to spring height which is 2 mm thus the maximum area is

\[ A_{v_{Al\max}} = \pi d_{\text{flowmax}} l_{\max} = 157 \text{mm}^2 \]

(2.16)

The lift of the valve depends on spring compression. The force on the spring is

\[ F = (p_1 - p_2) \frac{\pi (d_{\text{flow}}^2 - d_i^2)}{4} \]

(2.17)

where, \( F_0 = k l_0 \), which is the initial compression load of the spring \( d_i = 14.5 \text{mm} \) is the inner diameter of washer; while the lift is

\[ l = \frac{F - F_0}{k} = \frac{(p_1 - p_2)\pi (d_{\text{flow}}^2 - d_i^2)}{4k} - \frac{F_0}{k} \]

(2.18)

where \( k = 35 \text{kN/m} \) is the spring stiffness. Obviously the lift of the valve cannot be lower than zero, this mean that the valve start opening only when pressure force is higher than the static load.

\[ l = 0 \quad \Rightarrow \quad \frac{(p_1 - p_2)\pi (d_{\text{flow}}^2 - d_i^2)}{4} = k l_0 = 140 \text{N} \]

(2.19)

The free length of the spring is 19 mm while the working length is 13 mm, the initial compression is thus \( l_0 = 4 \text{ mm} \); the minimum pressure difference
to lift the valve is

\[(p_1 - p_2)_{\text{min}} = \frac{4kl_0}{\pi(d_{flow}^2 - d_i^2)} = 0.64\text{MPa}\]  \hspace{1cm} (2.20)

The pressure difference to which corresponds the maximum displacement of the spring is

\[(p_1 - p_2)_{\text{max}} = \frac{kl_{\text{max}}}{\pi(d_{flow}^2 - d_i^2)}\]  \hspace{1cm} (2.21)

Figure 2.5: Valve A: Piston mechanism
CHAPTER 2. MODEL OF PASSIVE DAMPER

For rebound stroke it is the same: the piston almost symmetric. The difference from compression stroke is that the spring which control the lift of the valve in rebound is stiffer than the compression spring. The geometry is exactly the same.

In fact there is a lower upper bound for flow area which is given by connection holes machined into the piston. This holes have a diameter of 2.4 mm and are two for each flow direction which means that the maximum flow area is

\[ A_{vA,max} = n_h \frac{\pi d^2}{4} \]  

(2.22)

The total flow area is

\[ A_{vA} = \begin{cases} 
A_{vA0}, & \text{if } (p_1 - p_2) < (p_1 - p_2)_{\text{min}} \\
A_{vA0} + \pi d_{\text{flow}} \left( \frac{\pi (d_{\text{flow}}^2 - d_i^2)}{4k} \right) (p_1 - p_2) - l - 0, & \text{if } (p_1 - p_2) < (p_1 - p_2)_{\text{max}} 
\end{cases} \]  

(2.23)

Figure 2.6: Valve A: Flow area versus pressure

2.1.2 Laminar leakage on piston valve

A laminar leakage is modeled on piston (Fig 2.7). The flow goes through the radial gap of the moving plate of the valve. The flow is

\[ Q_l = \frac{\delta^3 d \pi}{12 \nu l} (p_1 - p_2) \]  

(2.24)

where the diameter \( d = 14.5 \text{ mm} \), the gap is \( \delta = 0.1 \text{ mm} \).
2.1. NUMERICAL MODEL

2.1.3 Valve B

Valve B is made by obliging the oil flow around the rod. Then it accumulates in a small chamber linked to the reservoir chamber by two fixed diameter holes; it is possible to assume that the pressure inside this chamber is the same as the reservoir chamber. These holes have a diameter of about 3 mm. The flow from chamber 2 to reservoir one is thus laminar. Referring to the Fig 2.8, the flow across the rod is

\[
Q_B = \frac{\pi d_r \delta_B^3}{12 \mu l_B} (p_2 - p_r) + \frac{v_r}{2} \delta_B 2 \pi d_r \left[ m^3/s \right] \tag{2.25}
\]

where,

\[ d_r = 20 \text{mm}, \]

\[ \delta_B = 0.025 \text{mm}, \]

\[ l_B = 16 \text{mm} \]
2.1.4 Valve C

Valve C is made with a floating body as shown in Fig 2.9. When pressure in compression chamber (chamber 1) is higher than the pressure in reservoir chamber, there is a flow of oil passing through 4 holes in the floating body whose diameter is \( d_h = 1 \text{mm} \). These holes connect the compression chamber with a small chamber inside the floating body which is connected to the reservoir chamber through 2 rectangular grooves machined in the covering washer: the area of these grooves is \( h_g = 0.1 \text{mm} \) times \( l_g = 2 \text{mm} \).

\[
A_{vC0} = n_g h_g l_g \tag{2.26}
\]

where,

- \( n_g \) is the number of grooves.

Also, the difference of pressure causes a deformation of the washers. Considering the washer as an axial symmetric plate, from analytical solution we have that the displacement \( w \) as a function of the distance from the
revolution axes is

\[ w(r) = \frac{p_1 - p_r}{64D} \left[ r^4 - 4r_e^2r_i^2 - r_i^4 - 8r_i^2 \left( \frac{r_i^2}{2} - r_e^2 \ln \left( \frac{r_e}{r_i} \right) \right) \right] \] (2.27)

where

\[ D = \frac{E \ell^3}{12(1 - \eta^2)} \] (2.28)

with \( E \) and \( \eta \) respectively the Young modulus and the Poisson ratio of the material, \( p_1, p_r \) pressure in chamber 1 and reserve chamber, \( t \) is the washer thickness and \( r_i \) and \( r_e \), the inner and the outer radius respectively. For the valve lift we consider the displacement of the washer for \( r = r_e \) which is

\[ w(r_e) = \frac{p_1 - p_r}{64D} \left[ r_e^4 - 4r_e^2r_i^2 - r_i^4 - 8r_i^2 \left( \frac{r_i^2}{2} - r_e^2 \ln \left( \frac{r_e}{r_i} \right) \right) \right] \] (2.29)

Considering that there are four washers (\( n_w = 4 \)) of the same dimensions, the resulting compliance has to be divided by the washers number. The lift \( h \) of the valve is thus

\[ h(p_1 - p_r) = \frac{1}{n_w} \frac{p_1 - p_r}{64D} \left[ -3r_e^4 + 4r_e^2r_i^2 - r_i^4 - 4r_i^2 \left( \frac{r_i^2}{2} - r_e^2 \ln \left( \frac{r_e}{r_i} \right) \right) \right] \] (2.30)

the flow area is

\[ A_{vC} = h(p_1 - p_r) \cdot 2\pi r_e + A_{vC0} \] (2.31)

Instead when the pressure in compression chamber is lower than the reservoir chamber pressure, a plate moves to cover the holes in the floating body. The pressure difference on both side of the floating body makes the body move upwards winning the spring force. The valve is now opened and the reference area for this flow is the annulus between \( d_e = 17mm \) and \( d_i = 14.5mm \) as show in figure 2.9. In both cases a turbulent flow is assumed. The flow rate across valve C is

\[ Q_C = \begin{cases} c_q A_{vC} \sqrt{\frac{2[(p_R - p_1)]}{p}} \cdot \text{sign}(p_R - p_1), & \text{if } p_1 > p_R \\ c_q \left( \frac{\pi(d_e^2 - d_i^2)}{4} \right) \sqrt{\frac{2[(p_R - p_1)]}{p}} \cdot \text{sign}(p_R - p_1), & \text{if } p_1 < p_R \end{cases} \] (2.32)

Neglecting the transient when the valve opens, the flow area of the valve can be mapped as function of \((p_R - p_1)\) according to previous equations
Chapter 2. Model of Passive Damper

Figure 2.9: Valve C: Mechanism

Figure 2.10: Valve C: Area as function of pressure difference across the valve
2.2 Experiment

In this section we discuss about the experimental setup we use for physical testing of our numerical model. The main aim of the physical testing are as follows:

2. Validation of the numerical model.

2.2.1 Experimental Setup

For our study we use a hydraulic damper of secondary suspension system of a train which is mounted on a rigid support and given excitation through a hydraulic actuator. The experimental setup is shown in the figure below:

![Figure 2.11: Experimental setup for passive damper](image)

The main components of the system are:
• A computer controlled hydraulic actuator.
• A hydraulic damper
• Position measurement system with Lasers
• Load cell for Force measurement.
• Pressure hoses
• Connecting cables

The description of the equipments used is given in the Table 2.1:

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<th>Actuator</th>
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</thead>
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<td>MTS</td>
</tr>
<tr>
<td>Model</td>
<td>248.05</td>
</tr>
<tr>
<td>Maximum traction force</td>
<td>25 kN</td>
</tr>
<tr>
<td>Maximum compression force</td>
<td>25 kN</td>
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</tbody>
</table>

<table>
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<th>Load cell</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Name of manufacturer</td>
<td>MTS</td>
</tr>
<tr>
<td>Model</td>
<td></td>
</tr>
<tr>
<td>Range</td>
<td>kN</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Laser displacement transducer</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Name of manufacturer</td>
<td>MEL</td>
</tr>
<tr>
<td>Model</td>
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</tr>
<tr>
<td>Range</td>
<td>± 10 mm</td>
</tr>
</tbody>
</table>

Table 2.1: Experimental setup for passive damper

The hydraulic actuator can be used to give different types of input excitations like sinusoidal, ramp, sweep or random profile. The 2 lasers provide the displacement measurement and a load cell at the base provides the value of Force applied.

### 2.2.2 Experimental tests

**Characterization test**

For characterization of the passive damper, a sweep input with frequency from 1 Hz - 10 Hz is applied with different values of amplitude and the equivalent damping and stiffness of the damper are calculated at different frequencies.
2.2. EXPERIMENT

From the Force data, displacement of the piston and frequency of the input signal it is possible to estimate the equivalent stiffness and damping of the hydraulic damper.

\[ k = \frac{F}{x} \quad (2.33) \]

\[ r = \frac{1}{\omega} \left( \frac{F}{x} \right) \quad (2.34) \]

where,

\[ F = \text{Force on the piston} \]

\[ k = \text{equivalent stiffness of the damper} \]

\[ r = \text{equivalent damping of the damper} \]

\[ \omega = \text{frequency of the input} \]

Fig 2.12 shows the stiffness and damping curves for hydraulic damper in passive condition with external valve fully closed with sweep input (1Hz to 10Hz) and different amplitudes.

The stiffness and damping of a hydraulic damper depends on the force required and amount of energy absorbed during flowing of viscous fluid though the valve on piston from one chamber to another. From the graphs we can make the following inferences about the characteristics of the hydraulic damper.

1. The stiffness of the hydraulic damper at a constant amplitude increases with frequency as the oil has less time to pass from one chamber to another and compressibility of the fluid comes into effect, therefore the net force exerted on piston increases.

2. The damping of the hydraulic damper decreases with increase in frequency at constant amplitude for the same reason as less amount of fluid actually moves through the valve, the compression of fluid is like a spring phenomenon so less dissipation of energy, hence lower value of damping.
3. The stiffness of the hydraulic damper at a constant frequency but higher amplitude is lower as in this case time available for compression or expansion stroke remains same as frequency is same but displacement amplitude and velocity is high so pressure in cylinder rises, pushes the internal spring more which increase the lift of the valve and more fluid is pushed through the valve and less compressibility effect so force exerted on piston reduces.

4. The damping reduces with constant frequency and increase of amplitude as because of the same reason stated above with leads to higher lift of the valve, thus increasing the flow area which makes flow of fluid from one chamber to another easier, therefore, dissipating lesser amount of energy which means lower amount of damping.
2.2. EXPERIMENT

Comparison with numerical model

Here we compare the stiffness and damping curves for the numerical model and the physical model for the Hydraulic damper to check the accuracy and correctness of the numerical model. The comparison is done for the passive hydraulic damper. Fig 2.13 shows the results of the comparison.

![Comparison of results](image)

Figure 2.13: Comparison of results from numerical and experimental model for stiffness and damping of hydraulic damper in passive condition and external valve fully closed.

Here we observe good agreement between the numerical model and experimental model, therefore, we can say that the numerical model is validated and can be used for further evaluation of the semi-active system through a more appropriate quarter-train model.
Chapter 3

Development of the Semi-Active Damper

Based on our discussion in the chapter "State of the Art", about various types of Active/ Semi-Active solutions available for our application and also from their relative advantages and disadvantages, we decided to use a Semi-Active control system for our study. The main reason for this choice is that for this application for which we are designing such a system, a semi-active suspension offers a more cost effective, reliable and relatively less complex solution. An active system such as a hydraulic or an electromagnetic system is usually expensive and requires large modification in original component for implementation, therefore for our purpose of study, a semi-active suspension is a suitable choice.

In this chapter, we discuss the implementation method of Semi-Active control in the numerical model of the passive hydraulic damper. As described earlier, a semi active suspensions have similar components like a passive damper i.e a spring and a damper, however the difference is that here the properties of the damper can be varied. For this purpose, we have the available options of magneto rheological damper or using a solenoid valve or other external valve arrangement which could be controlled by our control strategies. Considering the cost effectiveness and amount of modification required during manufacturing of the physical prototype at later stage, we decide to introduce an external valve arrangement for varying the properties
of the damper.

3.1 Numerical Model

For implementation of Semi-Active control system we need to add an external valve arrangement (Fig 3.1) to the model described in the previous chapter. The opening angle of this valve is to be governed by the control strategies which we will discuss later in this chapter.

Fluid flow through the external valve is restricted because of the constriction in the passage of flow of the fluid through the valve and the flow rate depends upon the valve opening area, this can be described by making reference to the physical phenomenon of flow through an orifice (Fig 3.2)

Figure 3.1: Semi-Active damper model with external valve
where,

$A_0$ is the area of the orifice

$A_2$ is the minimum flow area = vena contracta

The fluid particles are accelerated between section 1 and 2.

Applying the continuity equation for the incompressible flow:

$$A_1 u_1 = A_2 u_2 = A_3 u_3 \quad (3.1)$$

$$A_2 = C_c A_0 \quad (3.2)$$

$C_c$ is the contraction coefficient

Applying Bernoulli’s equation:

$$\frac{1}{2} \rho u_2^2 + p_2 = \frac{1}{2} \rho u_1^2 + p_1 \quad (3.3)$$

where,

$\rho$ is the density of the viscous fluid of the damper

$p_1, p_2$ are pressures in chamber 1 and 2.

$u_1, u_2$ are velocity of fluid at section 1 and 2
Assuming that there is no pressure recovery during the expansion from section 2 to section 3.

\[ \Delta p = p_1 - p_2 = p_1 - p_3 \]  

(3.4)

\[ p_1 - p_2 = \Delta p = \frac{1}{2} \rho (u_2^2 - u_1^2) = \frac{1}{2} \rho (1 - \left( \frac{A_2}{A_1} \right)^2) u_2^2 \]  

(3.5)

\[ \Rightarrow u_2 = \sqrt{\frac{2\Delta p}{\rho}} \sqrt{1 - \left( \frac{A_2}{A_1} \right)^2} \]  

(3.6)

Volumetric flow rate \( Q_D \) (evaluated in minimum flow area):

\[ Q_D = C_c A_0 u_2 = \frac{C_c A_0}{\sqrt{1 - \left( \frac{A_2}{A_1} \right)^2}} \sqrt{\frac{2\Delta p}{\rho}} = C_d A_0 \sqrt{\frac{2\Delta p}{\rho}} \]  

(3.7)

where,

\[ C_d = \frac{C_c}{\sqrt{1 - C_c^2 \left( \frac{A_2}{A_1} \right)^2}} \Rightarrow \text{Discharge coefficient} \]  

(3.8)

The control strategy varies this valve area i.e. \( A_0 \) through an actuator to vary the flow rate \( Q_D \) through the valve, thus, altering the damper characteristics. To effectively study the effect of this Semi-Active damper on vibration characteristics in a train, this numerical model is used in the model of a Quarter-Train, described in the next section.
3.2 Quarter-Train Model Representation

The simplest representation of a train suspension is a 2 DOF quarter-train model with a secondary spring and damper connecting the chassis to the bogie, which is in turn connected to the wheel by primary spring and damper, as shown in Fig. 3.3. The mass representing the wheel, brakes and part of the suspension linkage mass, is referred to as the unsprung mass. In order to isolate the vehicle from irregularities of the track, the suspension is required to act as a filter. However, while acting as a filter it is also required to carry the static load of the vehicle. This creates a static deflection in the spring which has to be taken into account in the suspension design. It is also required to accommodate changes which occur in the static deflection due to changes in load, unless some form of leveling mechanism is employed.

The quarter train model shown in Fig. 3.3 can be described by a
pair of second-order differential equations:

\[ M_b \dddot{Z}_A + (r_1 + r_2) \ddot{Z}_A - r_2 \ddot{Z}_D + (K_1 + K_2)Z_A - K_2Z_D = r_1 \dot{Z}_W + K_1Z_W \]  
\[ (3.9) \]

\[ M_c \dddot{Z}_D - r_2 \ddot{Z}_A = r_2 \ddot{Z}_D - K_2Z_A + K_2Z_D = 0 \]  
\[ (3.10) \]

In a typical modern train, with moderate damping rates there are two distinct modes of vibration, a chassis mode around 1 Hz and a bogie mode around 8 Hz. A fortuitous side effect of these ratios is that the single passive damper imparts similar damping ratios to each of the modes.

### 3.3 Design of Control Strategy

In this section, we deal with the problem of designing a control law for semi active suspension systems. A semi active suspension also consists of a spring and a damper, but unlike a passive suspension, its characteristics can be varied, governed by a suitable control strategy designed to obtain the desired performance.

The most common control strategy used in industry for improving ride quality is the conventional Skyhook control strategy, which has been described in detail in Chapter 1. For our study, we used Skyhook control strategy as a reference to compare and evaluate the other 4 control logics designed namely: Frequency based control on body acceleration, Frequency based control on chassis velocity, Force Amplitude control loop and Force control loop with fixed reference. In the next section, we discuss each of these control strategies in detail.

### 3.4 Control Strategies studied

#### 3.4.1 Frequency based control on Chassis Acceleration

This is a feedback loop control strategy. The aim of this strategy is to implement a frequency based control on the most critical parameter affecting ride
comfort i.e. the acceleration of the chassis. The intention is to determine the frequency of acceleration on the chassis and calculate suitable opening angle of the external valve, such that we can obtain maximum damping rate at the first mode of resonance i.e 1 Hz, we also want to reduce the transmitted vibrations to the chassis at higher frequencies to ultimately reduce the rms acceleration on the chassis.

Therefore, we measure chassis acceleration as an input, which is then filtered to compute Fourier transform to obtain a spectrum. This spectrum is multiplied by a window as shown in Fig. 3.4, which gives high weighting to frequencies in the range 0.5 - 3 Hz. The purpose of this window is to avoid opening the external valve at frequencies close to first resonance mode i.e 1 Hz. This is required in order to have high value of equivalent damping at these frequencies, as opening of external valve leads to reduction in equivalent damping, which results in higher vibration amplitude at resonance and hence reducing the ride comfort. For the same purpose, a condition is added in the target law to keep the valve closed if the measured frequency of input excitation is less than 2hz.

![Figure 3.4: Frequency weighting window](image)

The weighted spectrum obtained after multiplication with this window is then required to be further processed to give a unique value of frequency. For this purpose, we pick all the peaks of the spectrum above a threshold value, which has to be suitably assigned. If the threshold is set too high, we might miss important components of frequency, and on the other hand, if it is too low, the resulting data will contain considerable noise and other non-significant components. Therefore, for our study we used a threshold value of $0.4A_{max}$, where $A_{max}$ is the peak amplitude of the spectrum.
The frequency peaks, whose amplitudes are higher than threshold are then average weighted on their amplitudes to obtain a unique value of frequency, which can also be termed as equivalent frequency.

\[ f_{eq} = \sum_{i=1}^{n} \frac{A_i f_i}{A_i} \]  

(3.11)

where,

\[ n = \text{number of peaks with amplitude above the threshold} \]

Then this equivalent frequency is used to find out the suitable valve opening angle, through the target correlation curve between measured frequency of input and valve opening angle (explained in details in next section). The complete schematic of the control strategy is shown in fig 3.5.

![Control loop for control on Chassis Acceleration](image)

**Figure 3.5: Control loop for control on Chassis Acceleration**

### 3.4.2 Frequency based control on Chassis Velocity

The intention behind this control strategy is to exercise a control on the velocity of the chassis, like in the case of a skyhook, which is the most widely used control strategy for improving ride comfort. The difference being that skyhook applies a damping force proportional to the amplitude of velocity at the chassis, whereas in our strategy we design a frequency based control. A skyhook control is very effective in reducing vibrations close to first mode of resonance, as the velocity of vibration is high at resonance, therefore skyhook...
tries to offer maximum damping, but at high frequency excitations, even at moderate amplitudes, the measured velocity of vibration remains high and therefore skyhook tries to offer high damping by having lower opening angle of external valve, which is exactly opposite to our requirement as we have discussed earlier that because of the changes in damper characteristics at high frequencies, we need to open the valve to reduce transmissibility. We try to counter this problem by having a frequency based control on velocity, so we can have higher damping near resonance and can reduce transmissibility at high frequency.

The control algorithm is the same like the previous case, only difference is that instead of measuring acceleration, here we measure velocity of the chassis as an input. Which is then used to calculate the frequency and hence the valve opening angle. Fig. 3.6

![Figure 3.6: Control loop for control on Chassis Velocity](image)

### 3.4.3 Force amplitude control loop

This control strategy is designed to have a control on the force transmitted to the chassis as a result of input excitations. Here also we have a frequency based control, where the component of frequency at which chassis force is higher is given more weightage while calculating equivalent frequency, therefore we can obtain higher damping at frequencies close to first mode of resonance and can reduce the transmissibility at higher frequencies by opening external valve to alter damper characteristics.
The control algorithm takes Force on the chassis as an input, which in turn is used to compute frequency by the method used in previous case and hence we obtain the required value of valve opening angle through the target curve. Fig. 3.7

\[ F_{rms} = \sqrt{\int_{0}^{f_m} P(f) dF} \]  

(3.12)

where,

- \( F_{rms} \) is the RMS Force
- \( P(f) \) is the power spectral density function
3.5. DETERMINATION OF TARGET LAW

$f_m$ is the frequency band of interest.

dF is the width of each spectral bin.

This RMS value is then compared with a fixed reference value, in our case we took it as zero. Then based on the difference between the reference value and the RMS value of the Force, the suitable value of valve opening angle is decided again by using a target curve, properly tuned for giving optimum performance.

![Control loop for control on Body Force with a fixed reference](image)

Figure 3.8: Control loop for control on Body Force with a fixed reference

These 4 control strategies have to be evaluated on the basis of their performance, ease of installation and robustness. In the next section, we use the numerical model of a quarter-train to evaluate the different control strategies formulated and compare it with a conventional skyhook to shortlist the best possible solution for our semi-active suspension system.

3.5 Determination of Target law

We need to establish a relation between the measured frequency of the signal and the corresponding value of the required opening angle of the external valve. For this purpose, we need to know the effect of opening angle of external valve on the equivalent stiffness and damping of the hydraulic damper at
different frequencies in our band of interest i.e. 1Hz - 10Hz.

Experiment

![Stiffness and damping curves for hydraulic damper for sweep input](image)

Figure 3.9: Stiffness and damping curves for hydraulic damper for sweep input at 5mm amplitude and different valve opening angles

We perform a set of experimental tests on the physical damper by giving a sweep input at an amplitude of 5mm and frequency 1Hz to 10Hz at different external valve opening angles adjusted manually. The value 5mm is chosen for target law determination as it is approximately the mean amplitude of vibration of the chassis. We start from full closed valve and recorded the data for the complete sweep input and then repeated the test for different valve opening angles. We organize all this data on a single graph as shown in Fig 3.9

Analysis

From this data, we observe that as the valve opening angle increases at a particular frequency, the equivalent stiffness decreases which is the required behaviour, however, it also reduces the equivalent damping at the same time which has adverse effect on the system performance in
3.5. DETERMINATION OF TARGET LAW

the frequency range close to the natural frequency of the system, as we need higher damping to control the vibration amplitude at resonance.

Calculation of Target law

Figure 3.10: Selection of valve opening angles at different frequency based on required stiffness and damping characteristics

Considering these two opposing factors, we need to find a trade off and design the best suited curve on this graph for stiffness and damping with different frequencies. This is done by choosing a valve opening angle for each frequency on the graph below, such that we get a low value of stiffness, but with reasonable amount of damping. We decide to keep the valve close till 2 Hz frequency as we need maximum damping close to the first natural frequency of the system i.e. 1Hz. The best tuned curve thus selected is shown in Fig 3.10.
Table 3.1: Selected valve opening angles for different frequencies

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Valve angle (degrees)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>54</td>
</tr>
<tr>
<td>4</td>
<td>80</td>
</tr>
<tr>
<td>5</td>
<td>216</td>
</tr>
<tr>
<td>7</td>
<td>270</td>
</tr>
<tr>
<td>8</td>
<td>324</td>
</tr>
<tr>
<td>9.5</td>
<td>360</td>
</tr>
</tbody>
</table>

Figure 3.11: Selection of valve opening angles at different frequency based on required stiffness and damping characteristics

Now we interpolate the selected points to find a curve which very closely gives the required value of equivalent stiffness and damping at each frequency. This curve we will use as Target law in our control strategies.

The target law derived by this method is as follows:-

\[ \theta = 0.366(freq)^4 - 8.783(freq)^3 + 70.12(freq)^2 - 159.82(freq) + 100 \]  

(3.13) where,

\( \theta \) is the opening angle of the external valve
freq is the measured frequency

3.6 Numerical Test

3.6.1 Input Excitation Methods

There are many input excitation methods which can be used to study the effect of semi-active control on vibration characteristics, for example, sinusoidal, frequency sweep, step input and random profile. But the most precise method is to use the track irregularity data as the input.

![Figure 3.12: Track irregularity input](image)

The random irregularities of a railway track can be represented by a spatial power spectrum as shown in Fig. 3.12, which can be then transformed into vertical displacement vector using inverse Fourier transform. For secondary-suspension assessment, the higher-order terms do not have any significant effect above 10 Hz or so, and can therefore be neglected. [14]

But this input has useful frequency content confined in a very narrow range, therefore, it does not provide a thorough comparison. For this purpose, we use 2 more input signals for our comparative analysis:-
1. Track irregularity with an additional peak at 3Hz to simulate the condition of a vibration input just above the natural frequency region.

![Figure 3.13: Track irregularity input with an added peak at 3Hz](image)

2. Track irregularity with an additional peak at 5Hz to simulate the condition of a high frequency vibration input.

![Figure 3.14: Track irregularity input with an added peak at 5Hz](image)
3.6. NUMERICAL TEST

3.6.2 Results

In order to compare the performance of the control strategies with respect to conventional Skyhook control and to shortlist the optimum one, we performed numerical simulation on all control strategies and also on skyhook and passive system. The simulations are done for 3 input excitation types described above. The results are recorded and compared for the following parameters:

1. RMS Acceleration

2. RMS Velocity

3. RMS Displacement

4. Peak Acceleration

5. Peak Velocity

6. Peak Displacement

All the recorded values are then normalized by the values for passive damper so that we can notice the improvement or deterioration with respect to the passive condition. We also simulated the control strategies with and without the frequency weighting window described in the previous section to observe the effect of the window on the performance.

The results are summarized in the following tables and figures:
### Table 3.2: RMS and Peak values for Track irregularity Input

<table>
<thead>
<tr>
<th>CONTROL STRATEGY</th>
<th>RMS VALUES</th>
<th>PEAK VALUES</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>RMS Acc</td>
<td>RMS Vel</td>
</tr>
<tr>
<td>Passive closed</td>
<td>0.196</td>
<td>31.810</td>
</tr>
<tr>
<td>Passive semi-open</td>
<td>0.229</td>
<td>37.700</td>
</tr>
<tr>
<td>Logic with Acceleration</td>
<td>0.196</td>
<td>31.810</td>
</tr>
<tr>
<td>Logic with Velocity</td>
<td>0.196</td>
<td>31.810</td>
</tr>
<tr>
<td>Skyhook</td>
<td>0.196</td>
<td>31.880</td>
</tr>
<tr>
<td>Logic with Force on chassis</td>
<td>0.196</td>
<td>31.810</td>
</tr>
<tr>
<td>Force control with zero reference</td>
<td>0.224</td>
<td>36.880</td>
</tr>
<tr>
<td>Logic with Acceleration (window)</td>
<td>0.196</td>
<td>31.810</td>
</tr>
<tr>
<td>Logic with Velocity (window)</td>
<td>0.196</td>
<td>31.810</td>
</tr>
</tbody>
</table>

### Table 3.3: Normalized values for Track irregularity Input

<table>
<thead>
<tr>
<th>CONTROL STRATEGY</th>
<th>NORMALIZED VALUES</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Normalized</td>
</tr>
<tr>
<td></td>
<td>RMS Acc</td>
</tr>
<tr>
<td>Passive closed</td>
<td>1.000</td>
</tr>
<tr>
<td>Passive semi-open</td>
<td>1.169</td>
</tr>
<tr>
<td>Logic with Acceleration</td>
<td>1.000</td>
</tr>
<tr>
<td>Logic with Velocity</td>
<td>1.000</td>
</tr>
<tr>
<td>Skyhook</td>
<td>0.999</td>
</tr>
<tr>
<td>Logic with Force on chassis</td>
<td>1.000</td>
</tr>
<tr>
<td>Force control with zero reference</td>
<td>1.144</td>
</tr>
<tr>
<td>Logic with Acceleration (window)</td>
<td>1.000</td>
</tr>
<tr>
<td>Logic with Velocity (window)</td>
<td>1.000</td>
</tr>
</tbody>
</table>

### Figure 3.15: Comparative results for Track irregularity Input
3.6. NUMERICAL TEST

Observations

1. Since the frequency content of the input excitation is concentrated in the narrow frequency band less than 2Hz so the external valve remains closed for our control strategies based on frequency measurement i.e. logic on acceleration, velocity and force, so the results are similar to passive condition.

2. For passive semi-open condition, the external valve is always half open, therefore it offers lesser damping and as the signal is concentrated in narrow frequency band close to first eigen mode of natural frequency, therefore, resonance is more prominent due to less damping which is an adverse effect.

3. Skyhook offers some improvement in peak acceleration as it is based on velocity of chassis, which is higher in frequency range close to resonance because of higher amplitude of vibration, therefore it offers maximum damping.

4. For the control strategy on Chassis Force with zero reference, the external valve opens as it senses a considerable amount of force at resonance, again reducing the effective damping which is not as per the requirements.

<table>
<thead>
<tr>
<th>CONTROL STRATEGY</th>
<th>RMS VALUES</th>
<th>PEAK VALUES</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>RMS Acc</td>
<td>RMS Vel</td>
<td>RMS Disp</td>
<td>Peak Acc</td>
<td>Peak Vel</td>
<td>Peak Disp</td>
</tr>
<tr>
<td>Passive closed</td>
<td>0.257</td>
<td>30.400</td>
<td>7.035</td>
<td>0.723</td>
<td>72.240</td>
<td>19.860</td>
</tr>
<tr>
<td>Passive semi-open</td>
<td>0.272</td>
<td>36.450</td>
<td>7.947</td>
<td>0.760</td>
<td>80.950</td>
<td>23.680</td>
</tr>
<tr>
<td>Logic with Acceleration</td>
<td>0.255</td>
<td>30.440</td>
<td>7.045</td>
<td>0.699</td>
<td>72.240</td>
<td>19.860</td>
</tr>
<tr>
<td>Logic with Velocity</td>
<td>0.257</td>
<td>30.390</td>
<td>7.990</td>
<td>0.753</td>
<td>72.240</td>
<td>19.860</td>
</tr>
<tr>
<td>Skyhook</td>
<td>0.248</td>
<td>29.960</td>
<td>7.027</td>
<td>0.659</td>
<td>74.130</td>
<td>20.250</td>
</tr>
<tr>
<td>Logic with Force on chassis</td>
<td>0.254</td>
<td>30.400</td>
<td>7.041</td>
<td>0.698</td>
<td>72.240</td>
<td>19.860</td>
</tr>
<tr>
<td>Force control with zero reference</td>
<td>0.273</td>
<td>36.760</td>
<td>7.896</td>
<td>0.718</td>
<td>89.980</td>
<td>23.780</td>
</tr>
<tr>
<td>Logic with Acceleration (window)</td>
<td>0.255</td>
<td>30.480</td>
<td>7.054</td>
<td>0.705</td>
<td>72.240</td>
<td>19.860</td>
</tr>
<tr>
<td>Logic with Velocity (window)</td>
<td>0.257</td>
<td>30.390</td>
<td>7.029</td>
<td>0.734</td>
<td>72.240</td>
<td>19.860</td>
</tr>
</tbody>
</table>

Table 3.4: RMS and Peak values for Track irregularity input with an added peak at 3Hz
CHAPTER 3. DEVELOPMENT OF THE SEMI-ACTIVE DAMPER

Table 3.5: Normalized values for Track irregularity input with an added peak at 3hz

<table>
<thead>
<tr>
<th>CONTROL STRATEGY</th>
<th>NORMALIZED RMS Acc</th>
<th>NORMALIZED RMS Vel</th>
<th>NORMALIZED RMS Disp</th>
<th>NORMALIZED Peak Acc</th>
<th>NORMALIZED Peak Vel</th>
<th>NORMALIZED Peak Disp</th>
</tr>
</thead>
<tbody>
<tr>
<td>Passive closed</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Passive semi-open</td>
<td>1.057</td>
<td>1.199</td>
<td>1.115</td>
<td>0.955</td>
<td>1.245</td>
<td>1.192</td>
</tr>
<tr>
<td>Logic with Acceleration</td>
<td>0.990</td>
<td>1.001</td>
<td>1.001</td>
<td>0.954</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Logic with Velocity</td>
<td>0.999</td>
<td>1.000</td>
<td>0.999</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Skyhook</td>
<td>0.963</td>
<td>0.888</td>
<td>0.999</td>
<td>0.890</td>
<td>1.020</td>
<td>1.020</td>
</tr>
<tr>
<td>Logic with Force on chassis</td>
<td>0.968</td>
<td>1.000</td>
<td>1.002</td>
<td>0.952</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Force control with zero reference</td>
<td>1.059</td>
<td>1.209</td>
<td>1.122</td>
<td>0.578</td>
<td>1.246</td>
<td>1.125</td>
</tr>
<tr>
<td>Logic with Acceleration (window)</td>
<td>0.992</td>
<td>1.003</td>
<td>1.003</td>
<td>0.961</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Logic with Velocity (window)</td>
<td>0.999</td>
<td>1.000</td>
<td>0.999</td>
<td>1.001</td>
<td>1.000</td>
<td>1.000</td>
</tr>
</tbody>
</table>

Observations

1. Passive semi-open and control strategy on Chassis force with zero reference offers poor results because of the same reason discussed before.

2. Control logic with acceleration shows some improvement in RMS Acceleration and peak acceleration but not by a considerable amount as the weighted averaged frequency of input signal measured is low so external valve opens by a small angle.

3. Here we can observe that because of the effect of addition of window, control strategy on Acceleration gives worse results as higher weightage
is given to lower frequencies therefore valve opening angle reduces, in this case it is always closed when we use the window.

4. Logic with Velocity offers no improvement as velocity spectrum gives higher weightage to lower frequencies, therefore average weighted frequency measured keeps the external valve closed, therefore results are equivalent to passive.

5. Skyhook offers best performance for this input excitation as in excitations close to natural frequency, velocity is high due to high amplitude, hence skyhook offers maximum damping force.

<table>
<thead>
<tr>
<th>CONTROL STRATEGY</th>
<th>RMS VALUES</th>
<th>PEAK VALUES</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>RMS Acc</td>
<td>RMS Vel</td>
</tr>
<tr>
<td>Passive closed</td>
<td>0.4644</td>
<td>28.11</td>
</tr>
<tr>
<td>Passive semi-open</td>
<td>0.3939</td>
<td>32.57</td>
</tr>
<tr>
<td>Logic with Acceleration</td>
<td>0.402</td>
<td>28.72</td>
</tr>
<tr>
<td>Logic with Velocity</td>
<td>0.434</td>
<td>37.58</td>
</tr>
<tr>
<td>Skyhook</td>
<td>0.4224</td>
<td>24.1</td>
</tr>
<tr>
<td>Logic with Force on chassis</td>
<td>0.4644</td>
<td>28.11</td>
</tr>
<tr>
<td>Force control with zero reference</td>
<td>0.3752</td>
<td>35.08</td>
</tr>
<tr>
<td>Logic with Acceleration (window)</td>
<td>0.4205</td>
<td>27.64</td>
</tr>
<tr>
<td>Logic with Velocity (window)</td>
<td>0.4833</td>
<td>28.03</td>
</tr>
</tbody>
</table>

Table 3.6: RMS and Peak values for Track irregularity input with an added peak at 5hz

<table>
<thead>
<tr>
<th>CONTROL STRATEGY</th>
<th>NORMALIZED VALUES</th>
<th></th>
<th>NORMALIZED VALUES</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>RMS Acc</td>
<td>RMS Vel</td>
<td>RMS Acc</td>
<td>RMS Vel</td>
</tr>
<tr>
<td>Passive closed</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Passive semi-open</td>
<td>0.838</td>
<td>1.152</td>
<td>1.101</td>
<td>0.869</td>
</tr>
<tr>
<td>Logic with Acceleration</td>
<td>0.866</td>
<td>1.022</td>
<td>1.023</td>
<td>0.973</td>
</tr>
<tr>
<td>Logic with Velocity</td>
<td>0.935</td>
<td>0.995</td>
<td>0.958</td>
<td>1.006</td>
</tr>
<tr>
<td>Skyhook</td>
<td>0.910</td>
<td>0.857</td>
<td>0.921</td>
<td>1.048</td>
</tr>
<tr>
<td>Logic with Force on chassis</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Force control with zero reference</td>
<td>0.808</td>
<td>1.248</td>
<td>1.159</td>
<td>0.842</td>
</tr>
<tr>
<td>Logic with Acceleration (window)</td>
<td>0.906</td>
<td>0.983</td>
<td>0.989</td>
<td>1.007</td>
</tr>
<tr>
<td>Logic with Velocity (window)</td>
<td>0.998</td>
<td>0.997</td>
<td>0.995</td>
<td>1.001</td>
</tr>
</tbody>
</table>

Table 3.7: Normalized values for Track irregularity input with an added peak at 5hz
CHAPTER 3. DEVELOPMENT OF THE SEMI-ACTIVE DAMPER

Figure 3.17: Comparative results for Track irregularity input with an added peak at 5hz

Observations

1. Control logic with Acceleration offers considerable reduction in RMS acceleration and also in peak acceleration because of presence of a high component of frequency.

2. Skyhook offers good performance on most parameters except Peak Acceleration.

3. Control logic with velocity offers some improvement, but lesser as compared to the control logic with Acceleration, because of the same reason, as it gives more weightage to lower frequencies, therefore, angle of external valve opening is less.

4. Control with force on chassis with zero reference offers good results for RMS Acceleration, but worst results for all other parameters.

5. For this excitation also, we can observe that the effect of window deteriorates the performance for the reason mentioned before.

In the next chapter we describe the physical prototype of the Semi-Active damper tested in the laboratory to validate the numerical model of the semi-active damper. After ensuring the conformance of the numerical model with the physical prototype, we will discuss these results in the final chapter.
Chapter 4

Prototype of the Semi-Active Damper

In this chapter we describe the prototype of the Semi-Active Hydraulic Damper we created and tested in laboratory. The main aim of the physical testing are as follows:-

1. Validation of the semi-active control action on the numerical model of the single damper with experimental results.

2. Determination of a correlation between measured frequency and corresponding value of required valve opening angle i.e. the target law.

3. To evaluate the working of the hardware components like sensors, actuators, electronic control unit and other electronic components and connections required for the actual system.

4. To physically test and evaluate the proposed solution as a whole.

4.1 Semi-Active damper prototype

As we discussed in the previous chapter, for our study we decided to use a semi-active control system with an external valve to vary the characteristics of the hydraulic damper. For the actuator, we decide to use a servo motor.
fitted to the valve through a gearing arrangement, the reason for this choice is good positional accuracy of a servo motor, high speed and easy to integrate to our system. For the prototype of a Semi-Active damper we performed the following modifications in the original passive damper (Fig 4.1):

1. Connection between the 2 main chambers of the hydraulic damper externally through a control valve which is to be regulated through an actuator whose action is governed by the control strategy programmed in the electronic control unit. This control valve can be opened or closed by required amount to vary the damper characteristics.

2. Integration of the actuator i.e. the stepper motor with the external valve through the gearing arrangement designed to have low frictional resistance and minimum error due to backlash and tolerances.
3. Addition of an electronic control unit (Arduino) which is programmed as per the control strategy and connected to the actuator through electrical circuit.

4.2 Experimental Setup

The experimental setup created for the testing of the prototype damper is shown in the figure below:

![Experimental setup of the semi-active damper](image)

Figure 4.2: Experimental setup of the semi-active damper

The main components of the system are:

- A computer controlled hydraulic actuator.
- Prototype of Semi-Active damper
- Stepper motor with gearing arrangement
• Position measurement system with Lasers
• Accelerometer
• Load cell
• Low pass filter
• Microprocessor (Arduino)
• A breadboard and Resistors
• Connecting cables

4.2.1 Description

The overall schematic arrangement can be described by the block diagram as shown in Fig. 4.3.

![Figure 4.3: Block diagram of the experimental setup](image)

We acquire the signal from the accelerometer at 50 Hz so we need a low pass filter to remove the noise in the frequency range above the Nyquist frequency to avoid *Aliasing*. This signal is then fed to Arduino, where it is processed and then based on the control strategies discussed in the previous
4.3. EXPERIMENTAL RESULTS

chapter, a suitable value of opening angle for the external valve is calculated. An electrical signal is passed on to the stepper motor which through a gearing arrangement opens the external valve by the required angle.

The system is tested for different input signals based on track irregularity data (described in the previous chapter). Through the test we check the proper measurement of the data and functioning of all the electronic and mechanical components. For validation purpose, we compare the valve opening of the valve for the numerical and experimental tests.

4.2.2 Response time of sensors and actuators

An important step is improving the performance of the resulting suspension system taking into account the delay time $\Delta t$ that elapses between two updates of the damper coefficient. First of all, it is important to observe that the on-board implementation of the controller is typically done using a microprocessor with a scan time $\Delta t$ of the order of a few milliseconds. Every $\Delta t$ time units the controller should choose on the basis of the current state of the system, a new value for the damper coefficient, so as to minimize the difference between the semi active and the target law. Furthermore, it is also necessary to take into account the physical limits on the updating frequency of the damper coefficient. In fact, the actuators used to control the damper coefficient have a response time that cannot be neglected because it is usually larger than the scan time of the controller.

4.3 Experimental Results

We test the experimental setup for the physical prototype of hydraulic damper with 3 different inputs (as described in previous section):

1. Track irregularity Input
2. Track irregularity with an added peak at 3hz
3. Track irregularity with an added peak at 5hz
Main purpose of the testing is to check the proper functioning of all the components, to test the control strategy on the physical model and also to check the correlation with the numerical model. Therefore, we conduct the experimental test, to check the proper functioning of the system first, we record the data for operation of the external valve with 3 different excitation inputs described above. The results are shown in Fig below:-

Figure 4.4: Valve operation for Track irregularity Input

Figure 4.5: Valve operation for Track irregularity with added peak at 3hz
4.4 Comparsion with Numerical Results

The comparison of experimental results of operation of external valve for 3 different excitation inputs with the numerical model of the semi-active damper are shown in the figures below:

Figure 4.6: Valve operation for Track irregularity with added peak at 5hz

In the next section, we compare these experimental results with the results from the numerical simulation on the model of the semi-active hydraulic damper, in order to ascertain the accuracy of the numerical model.

Figure 4.7: Comparison of Numerical and Experimental results of valve operation for Track irregularity Input
From the comparison of the results of numerical and experimental models, we observe that the values are very similar but with some time delay for the experimental model which is expected because of the time required by actuator i.e. stepper motor to perform the valve opening operation. Therefore, we can conclude from these results that the numerical model and experimental models are correlated.
Chapter 5

Discussion and Conclusion

Now before concluding, here we summarize the major steps we followed in our study on introducing semi-active control in hydraulic dampers for improving ride comfort at high frequency excitations:

- First we defined the problem statement, understood the vibration characteristics of a vehicle with a passive damper in the frequency band of our interest and identified the goals and improvement which we want to achieve through our study.

- Then we studied the current status of the research in this field, existing and upcoming technologies and their implications, conventional control strategies used such as Skyhook.

- Then we shortlisted the solution which we wish to adopt for our study comparing relative pros and cons of available options.

- We created a numerical model of the passive damper, studied its characteristics at different frequencies and amplitudes of excitation to better understand its behaviour. Tested experimentally the passive damper to compare the results and validate the numerical model.

- Then we created numerical model of the semi-active damper as per the solution we adopted, designed various control strategies to obtain desired results, formulated a correlation between frequency of input and required external valve opening angle, then we performed numerical simulation
on a more relevant Quarter-train model using the control strategies designed by us and also for passive damper and skyhook control strategy for reference and comparison.

• Then finally we created a physical prototype of the semi-active damper to experimentally test the proposed solutions.

5.1 Conclusion

From our study and the comparison of different control strategies with Skyhook and Passive damper, on the basis of different parameters, i.e. RMS and peak values of Acceleration, Velocity and Displacement, we can derive following conclusions:

1. The conventional skyhook offers best performance at low frequencies close to the first mode of resonance and reasonably good performance at higher frequencies as well.

2. Control strategy with Acceleration offers improvement over passive damper over the complete bandwidth of frequency we studied and even exceeds the performance of a conventional skyhook at higher frequencies, therefore it can be a preferred choice for our application.

3. Control strategy with Velocity shows no improvement over passive for first 2 types of excitations, but shows marginal improvement at higher frequency but lesser as compared to Control strategy with Acceleration therefore it can be neglected.

4. Control logic on Chassis Force shows almost no improvement over passive, therefore it can be neglected.

5. Control logic on chassis Force with zero reference shows improvement in RMS acceleration only at high frequencies but very poor results with all parameters for all three types of input excitation, therefore it should also be neglected.
5.2 Further Scope

For carrying out further studies in this research field, we can pursue these possibilities, which may provide an improvement in the solution we propose.

1. The solution could be tried with a faster actuator such as a solenoid valve in order to reduce the time delay between actual change in the state of the system and completion of the control action.

2. The control strategy could be modified in order to take into account 2 inputs i.e. vibration amplitude and frequency instead of using only frequency as an input because vibration amplitude also has a significant effect on the damper characteristics at a constant frequency, this can be done by defining different target laws for different amplitude bands (each law giving a correlation for frequency of excitation and external valve opening angle for that particular amplitude band) and switching between them as per the amplitude of the excitation.

3. The target law followed for the control strategies may also be tuned to find the optimum curve for the actual excitation input which is encountered in the real application.
Bibliography


