VIBRATION CONTROL IMPLEMENTATION IN A BRAKING SYSTEM

Thesis submitted in
partial fulfillment of the requirements for the award of degree of

Master of Engineering
in
Electronic Instrumentation and Control

Submitted by

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DECLARATION

I hereby certify that the work which is being presented in the thesis entitled " **Vibration Control Implementation in a Braking System"** in partial fulfillment of award of degree of **Master of Engineering** in **Electronic Instrumentation and Control** submitted in Electrical and Instrumentation Engineering department, Thapar University, Patiala is an authentic record of my own work carried under the supervision of **Dr. Yaduvir Singh**, Associate Professor, Department of Electrical and Instrumentation Engineering, Thapar University, Patiala, Punjab.

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I certify that the above statement made by the student is correct to the best of my knowledge and belief.

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ABSTRACT

The spring mass damper can be built or represented on the computer instead of going to the workshop to fabricate such system and its performance under various conditions can also be observed without having to subject the real system to these conditions hence, you save materials and money, since the system can be used countless times. Energy is also saved because such system is more easily built on a computer than physically. Moreover, it may be very difficult to measure some outputs of some systems such as displacement but such values can be measured with ease through simulation.

In this thesis, a novel approach to reduce the effect of negative damping that causes brake noise is proposed by applying an Active Force Control (AFC) based strategy to a single degree-of-freedom as well as two degree-of-freedom model of a disk brake system. At first, the disc brake model is simulated and analyzed using a closed loop pure PID controller. Later, it is integrated with AFC and simulated under similar operating environment. After it is integrated with Fuzzy +AFC and simulate under the similar operating environment. After running several tests with different sets of operating and loading conditions, the results both in time and frequency domains show that the PID controller with AFC is much more effective in reducing the vibration and noise, compared to the pure PID controller alone and Fuzzy +AFC is more effective in reducing the vibration and noise as compared to PID controller with AFC.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS

2-D	Two-dimensional
3-D	Three-dimensional
DOF	Degree-of- freedom
MATLAB	Matrix laboratory
PID	Proportional-integral-derivative
AFC	Active force control
FLS	Fuzzy logic system
MF	Membership function
COA	Centre of area
ANN	Artificial Neural Network
GA	Genetic algorithms
HAVS	Hand-arm vibration syndrome
VWF	Vibration white finger
EU	European Union
LQG	Linear Quadratic Gaussian
FIV	Friction induced vibration

CHAPTER 1

INTRODUCTION

1.1: Overview

The increasing demands of high productivity and economical design led to higher operation speeds of machinery and efficient use of materials through light weight structures. Theses makes the trend of resonance conditions more frequent the periodic measurement of vibrations characteristic of machinery and structures become essential to ensure adequate safety margins. Any observed shift in the natural frequencies or other vibration characteristics will indicate either failure or a need for maintenance of the machine. The measurement of the natural frequencies of the structure or machine is useful in selecting the operational speed of nearby machinery to avoid resonant conditions. The theoretically computed vibration characteristics of a machine or structure may be different from the actual values due to the assumptions made in the analysis. In many applications survivability of a structure or machine in a specified vibration environment is to be determined. If the structure or machine can perform the expected task even after completion of testing under the specified vibration environment, it is expected to survive the specified conditions.

Continuous systems are often approximated as multidegree of freedom systems for simplicity .If the measured natural frequencies and mode shapes of a continuous system are comparable to the computed natural frequencies and mode shapes of the multidegree of freedom model, then the approximation will be proved to be a valid one. The measurement of the input and the resulting output vibration of a system help in identifying the system in terms of its mass stiffness and damp. The information about ground vibration due to earthquakes, fluctuating wind velocities on structures, random variation of ocean waves and road surface roughness are in the design of structures, machines oil platforms and vehicle suspensions systems.

1.2: Problem Statement

A physical system is to be replaced by a mathematical model in order to predict its vibration behavior. The accuracy of the predicted behavior depends on the level of difficulty associated with the mathematical model.

The model must account for the four basic phenomena associated with the physical system, namely, the elasticity, inertia, excitation or input energy, and damping or dissipation of energy.

The mathematical model should not be too complex and overly sophisticated to include more details of the system than are necessary

1.3: Objective of Thesis

To investigate the performance of a spring mass damper system, under various conditions, through modeling, without having to subject the real system to these conditions.

The present work aims at developing a Fuzzy logic controller for one of the types of brake suspension system and focus on comparing the obtained results with that of general passive suspension system in order to reduce the spring mass displacement for ride comfort, by considering two different road profiles.

1.4: Organization of the Thesis

The structure of this thesis is set out into five sections.

Chapter 1

In this chapter the brief introduction to this thesis and a brief description of the different areas that make up the thesis.

Chapter 2

This chapter deals with detail description of PID controller and basic PID controller is discussed.

Chapter 3

This chapter deals with the brief introduction of various artificial controller theories like Fuzzy logic controller, Neural network and Genetic algorithm.

Chapter 4

This chapter deals with literature survey related to the thesis.

Chapter 5

This chapter deals the description of vibration and its effect.

Chapter 6

This chapter deals the Modeling and simulation of case study of the single degree of freedom system and its result and applied various control strategy to reduce the vibration in brake system.

Chapter 7

This chapter deals the modeling and description of case study of the two degree of freedom system and applied various control strategy to reduce the vibration in brake system especially in Hoffmann model.

1.5: Flow Chart of Thesis

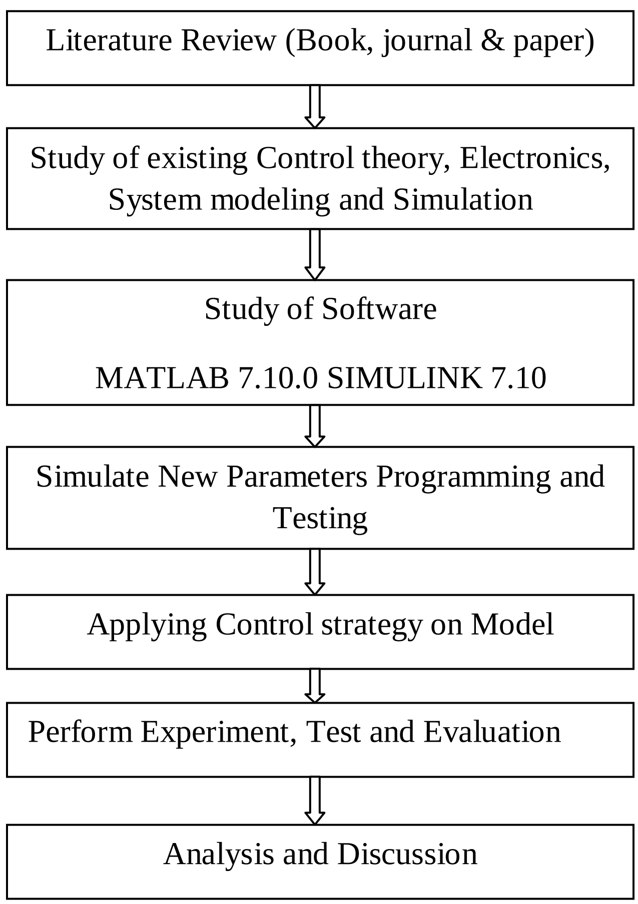


Figure 1.1: Thesis flowchart

CHAPTER 2

CONTROLLER

A proportional–integral–derivative controller (PID controller) is a generic control loop feedback mechanism widely used in industrial control systems – a PID is the most commonly used feedback controller. A PID controller calculates an "error" value as the difference between a measured process variable and a desired setpoint. The controller attempts to minimize the error by adjusting the process control inputs. In the absence of knowledge of the underlying process, PID controllers are the best controllers. However, for best performance, the PID parameters used in the calculation must be tuned according to the nature of the system – while the design is generic, the parameters depend on the specific system.

2.1 PID Control

PID control consists of three types of control, Proportional, Integral and Derivative control.

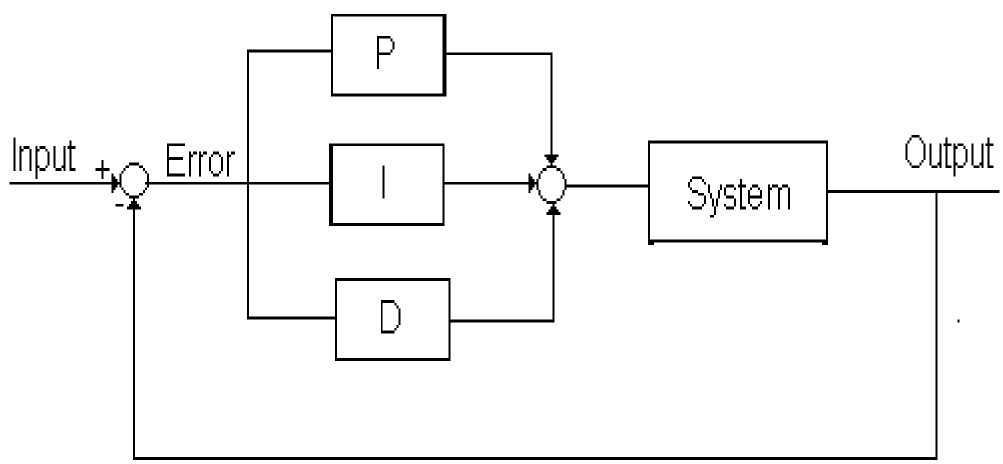


Figure 2.1 Schematic of PID Controller.

2.1.1: Proportional Control

The proportional term in the controller generally helps in establishing system stability and improving the transient response. The proportional controller output uses a 'proportion' of the system error to control the system. However, this introduces an offset error into the system.

$$P = K * Error$$

2.1.2: Integral Control

Integrator term that introduces a pole at s = 0 in the forward loop of the process. This makes the compensated1 open loop system (i.e. original system plus PID controller) a type 1 system at least; our knowledge of steady state errors tells us that such systems are required for perfect steady state set point tracking. The integral controller output is proportional to the amount of time there is an error present in the system. The integral action removes the offset introduced by the proportional control but introduces a phase lag into the system.

$$I_{term} = K \times \int Error\, dt \qquad (1)$$

2.1.3 Derivative Control

Derivative term is often used when it is necessary to improve the closed loop response speed even further. Conceptually the effect of the derivative term is to feed information on the rate of change of the measured variable into the controller action. The derivative controller output is proportional to the rate of change of the error. Derivative control is used to reduce/eliminate overshoot and introduces a phase lead action that removes the phase lag introduced by the integral action.

$$D_{term} = K_D \times \frac{d(Error)}{dt} \qquad (2)$$

2.1.4 Continuous PID Controller

The three types of control are combined together to form a PID controller with the transfer function:

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

$$= K_p \left(1 + \frac{1}{T_i S} + T_d S\right) \qquad (3)$$

Where: Kp: = Proportional Gain

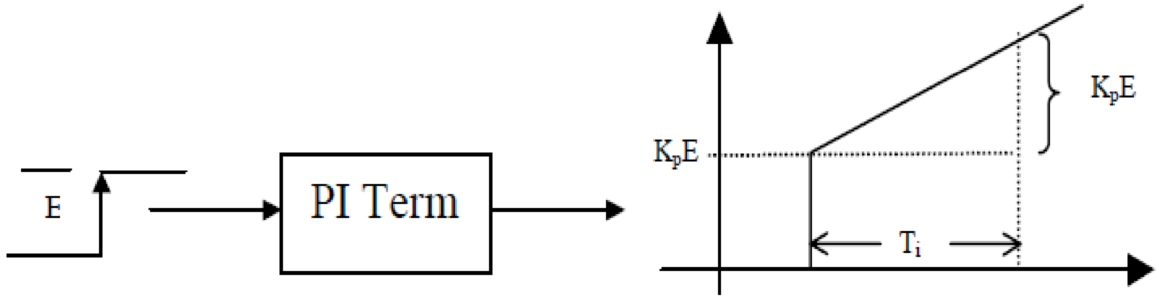
K_i: = Integral Gain

T_i: = Reset Time =Kp/Ki

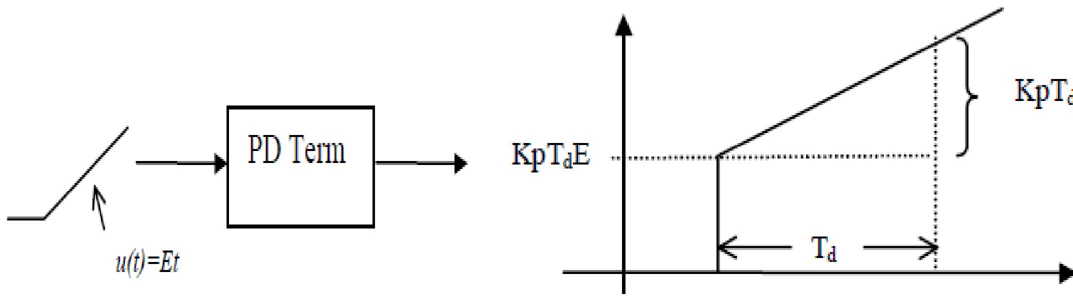
K_d: =Derivative gain

T_d: = Rate time or derivative time=$K_p K_d$

The **reset time** is the time taken for the integrator term output to equal the proportional term output in response to a step change in input applied to a PI controller.



The **rate time** is the time taken for the proportional term output to equal the derivative term output in response to a ramp change input applied to a PD controller.



In addition, the Proportional gain, Kp, is often expressed as a proportional band (%):

$$PB=1/K_p$$

PB is actually the fractional error change, relative to the error range, required to produce a 100% (full range) change in the proportional term output.

Mathematically, we have:

$$K_p = \frac{\Delta o}{\Delta e} = \frac{\text{Output change}}{\text{Input change}}$$

$$= \frac{\Delta o/\Delta omax}{\Delta e/\Delta emax}$$

$$= \frac{1}{PB} \times \frac{\Delta omax}{\Delta emax} \leftarrow \text{This last term is usually 1.}$$

$$= \frac{1}{PB}$$

In practice PB is expressed as a percentage so

$$PB\% = \frac{100}{Kp}$$

Thus a PB of 5% ⇔ Kp =20. We also note that:

- In addition, most controllers operate on relative (percentage or per unit) units. In this regard, quantities are scaled relative to their maximum range. This makes it easy to translate from one unit basis to another. For example, if our PV is a temperature in the range $0^0 - 100^0$ mapped to a current range of 4-20mA, then a PV 20% translates to an absolute temperature of 20^0 and an equivalent current signal reading of 4+ 16*.2=7.2mA. NB: If analysis is performed on any system the units used will determine the gains in the various transfer boxes.

Table 2.1: Common PID Controller Variations

Controller Type	**K_p**	**K_i**	**K_d**	**C(s)**
P (Proportional)	≠0	Zero	Zero	Kp
I (Integral)	Zero	≠0	Zero	$\frac{K_i}{S}$
PI (Proportional + Integral)	≠0	≠0	Zero	$\frac{K_p S + K_i}{S}$
PD(Proportional + Derivative)	Zero	≠0	Zero	$K_d S + K_P$
PI (Proportional + Integral + Derivative)	≠0	≠0	≠0	$\frac{K_d S^2 + K_p S + K_i}{S}$

The actual variation used depends on the specifications to be met. Purely derivative or integral plus derivative variations are almost never used. In all cases except proportional control, the PID compensator introduces one pole and at least one zero.

2.1.5 Discrete PID Controller

To facilitate the real time aspect of this, a discrete PID controller must be used. The PID controller will be discretised using the Trapezoidal Difference method.

2.1.5.1 Trapezoidal Difference Method

The trapezoidal difference method is the most popular method for discretizing a PID controller. The trapezoidal difference method maps a stable continuous controller to a stable discrete controller. The substitution $S=\frac{2Z-1}{TZ+1}$ is used to produce a mapping as shown in Figure 2.2.

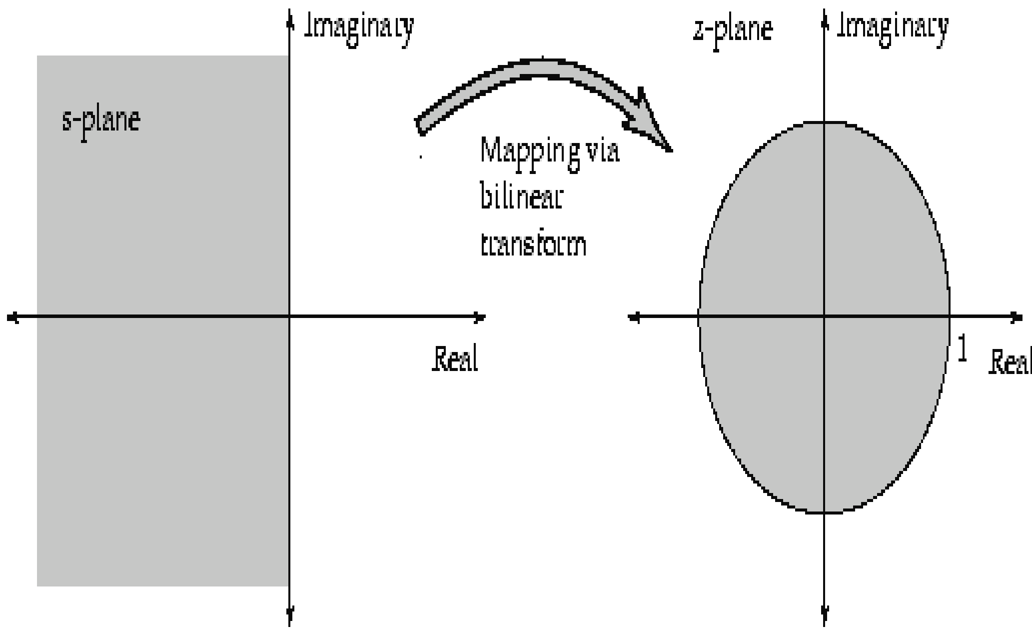


Figure 2.2 Illustration of Mapping using Trapezoidal Difference Method

The trapezoidal method is implemented in Matlab using the 'tustin' operator in conjunction with the C2D (Continuous to Discrete) command:

```
Discrete_PID = c2d (Continuous_PID, 0.01,'tustin');
```

2.2: Tuning Methods

There are several methods for tuning a PID loop. The most effective methods generally involve the development of some form of process model, then choosing P, I, and D based on the dynamic model parameters. Manual tuning methods can be relatively inefficient, particularly if the loops have response times on the order of minutes or longer. The choice of method will depend largely on whether or not the loop can be taken "offline" for tuning, and the response time of the system. If the system can be taken offline, the best tuning method often involves subjecting the system to a step change in input, measuring the output as a function of time, and using this response to determine the control parameters.

2.2.1: Manual Tuning

If the system must remain online, one tuning method is to first set Ki and Kd values to zero. Increase the K_p until the output of the loop oscillates, then the K_p should be set to approximately half of that value for a "quarter amplitude decay" type response. Then increase K_i until any offset is correct in sufficient time for the process. However, too much K_i will cause instability. Finally, increase Kd, if required, until the loop is acceptably quick to reach its reference after a load disturbance. However, too much K_d will cause excessive response and overshoot. A fast PID loop tuning usually overshoots slightly to reach the setpoint more quickly; however, some systems cannot accept overshoot, in which case an over-damped closed-loop system is required, which will require a K_p setting significantly less than half that of the K_p setting causing oscillation.

Table 2.2: Effects of increasing a parameter independently

Parameter	Rise Time	Overshoot	Settling Time	Steady-State Error	Stability
K_P	Decrease	Increase	Decrease	Decrease	Degrade
K_I	Decrease	Increase	Decrease	Decrease Significantly	Degrade
K_D	Minor Decrease	Minor Decrease	Minor Decrease	No effect in theory	Improve if K_D small

2.2.2: Ziegler-Nichols Tuning

In 1942 Ziegler and Nichols, both employees of Taylor Instruments, described simple mathematical procedures, the first and second methods respectively, for tuning PID controllers. These procedures are now accepted as standard in control systems practice. Both techniques make a priori assumptions on the system model, but do not require that these models be specifically known. Ziegler-Nichols formulae for specifying the controllers are based on plant step responses.

2.2.2.1 The First Method

The first method is applied to plants with step responses of the form displayed in Figure 2.3. This type of response is typical of a first order system with transportation delay, such as that induced by fluid flow from a tank along a pipe line. It is also typical of a plant made up of a series of first order systems. The response is characterized by two parameters, L the delay time and T the time constant. These are found by drawing a tangent to the step response at its point of inflection and noting its intersections with the time axis and the steady state value. The plant model is therefore

$$G(s) = \frac{Ke^{-sL}}{Ts+1} \qquad (4)$$

Ziegler and Nichols derived the following control parameters based on this model:

Table 2.3: Ziegler-Nichols – First Method

PID Type	**K_p**	**$T_i = K_p/K_i$**	**$T_d = K_d/K_p$**
P	$\frac{T}{L}$	∞	0
PI	$0.9\frac{T}{L}$	$\frac{L}{0.3}$	0
PID	$1.2\frac{T}{L}$	2L	0.5L

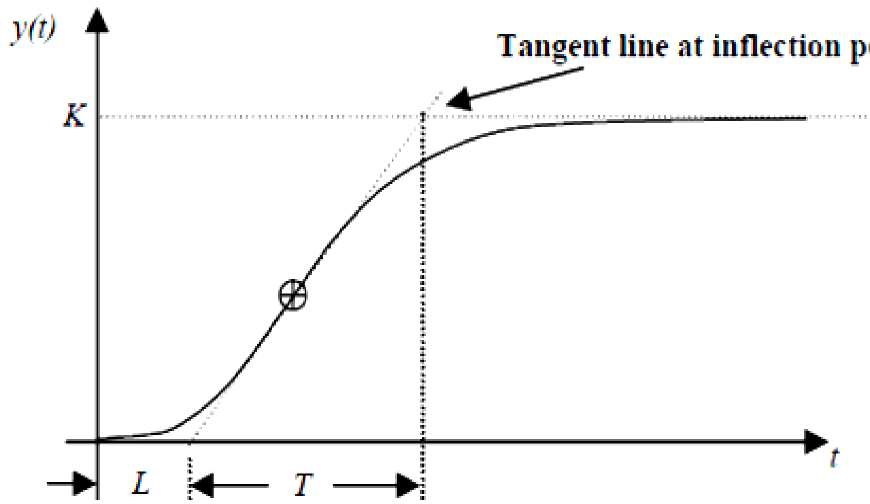


Figure 2.3: Response Curve for Ziegler-Nichols First Method

It should be noted that the response curve of Figure 2.3 is also typical of over damped second order systems.

2.2.2.2: Second Method

The second method targets plants that can be rendered unstable under proportional control. The technique I s designed to result in a closed loop system with 25% overshoot. This is rarely achieved as Ziegler and Nichols determined the adjustments based on a specific plant model.

The steps for tuning a PID controller via the 2nd method are as follows. Using only proportional feedback control:

- Reduce the integrator and derivative gains to 0.
- Increase Kp from 0 to some critical value Kp=Kcr at which sustained oscillations occur. If it does not occur then another method has to be applied.
- Note the value Kcr and the corresponding period of sustained oscillation Pcr.

The controller gains are now specified as follows

Table 2.4: Ziegler Nichols – Second Method

PID Type	**Kp**	**Ti**	**Td**
P	$0.5K_{cr}$	∞	0
PI	$0.45K_{cr}$	$\frac{P_{cr}}{1.2}$	0
PID	$0.6K_{cr}$	$\frac{P_{cr}}{2}$	$\frac{P_{cr}}{8}$

CHAPTER 3

ARTIFICIAL INTELLIGENCE

Fuzzy Inference Systems (FIS) are popular computing frameworks based on the concepts of fuzzy set theory, which have been applied with success in many fields like control, decision support, system identification, etc. Their success is mainly due to their closeness to human perception and reasoning, as well as their intuitive handling and simplicity, which are important factors for acceptance and usability of the systems

3.1 Fuzzy Logic Systems: Principles of Operation

Fuzzy logic systems are one of the main developments and successes of fuzzy sets and fuzzy logic. A FLS is a rule-base system that implements a nonlinear mapping between its inputs and outputs. A FLS is characterized by four modules

- Fuzzifier
- Defuzzifier
- Inference engine
- Rule base

A schematic representation of a FLS is presented in Fig. 3.1.

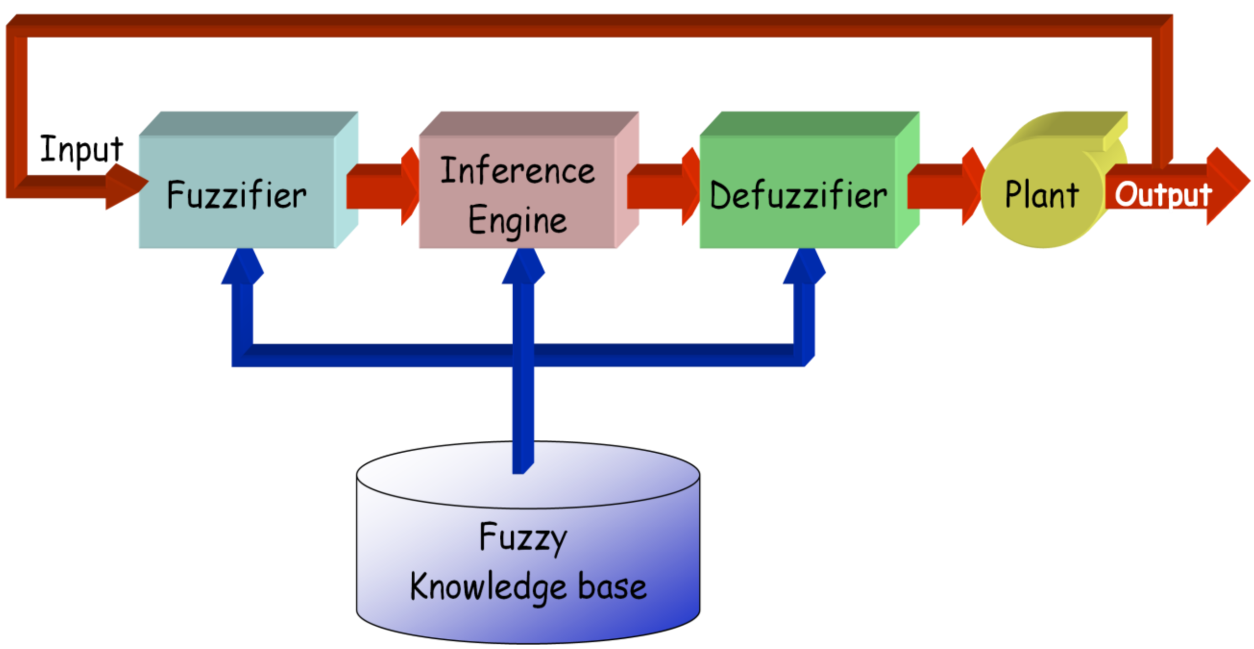


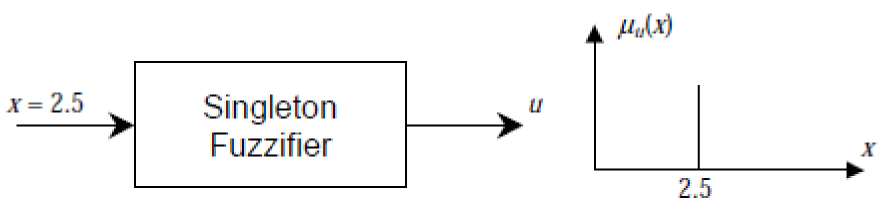
Figure 3.1: Structure of fuzzy logic system

A fuzzy inference system (FIS) consists of four functional blocks as shown in Figure 3.1

- **Fuzzification:** transforms the crisp inputs into degrees of match with linguistic values.
- **Knowledge base:** consists of a rule base and a database. A rule base contains a number of fuzzy if-then rules. A database defines the MFs of the fuzzy sets used in the fuzzy rules.
- **Fuzzy inference engine:** performs the inference operations on the rules.
- **Defuzzification:** transforms the fuzzy results of the inference into a crisp output.

3.1.1 Fuzzifier

Fuzzification can be defined as the operation that maps a crisp object to a fuzzy set, i.e., to a membership function. Fuzzifiers are generally divided in singleton and non-singleton ones. A singleton fuzzifier maps an object to the singleton fuzzy set centered at the object itself (i.e., with support and core being the set containing only the given object). A non singleton fuzzifier, maps an object to a fuzzy set generally centered at the object itself (i.e., the core of the fuzzy set contains the object) and with support containing the object but being a set bigger then only the object itself. A non-singleton fuzzifier maps an object into a non-singleton fuzzy set generally centered at the object itself. Typically, the use of a singleton fuzzifier is very common. Non-singleton fuzzifiers are also used, especially in the presence of e.g., noisy measurements. Indeed, in this case the input crisp value is affected by some uncertainty, thus, the corresponding input fuzzy set can reflect this uncertainty by allowing non-zero membership values around the (noisy) measurement. Therefore, when a non-singleton fuzzifier is used, the width of the corresponding fuzzy set is generally proportional to the amount of noise affecting the measurement. Figure 3.2 shows an example of singleton and non-singleton fuzzification.



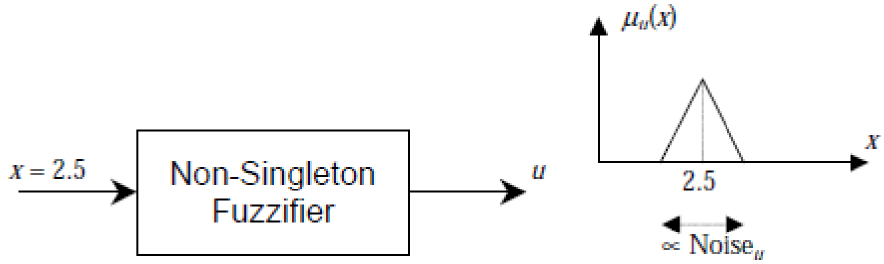


Figure 3.2: Example of singleton and non-singleton fuzzifier.

Singleton and non-singleton fuzzifiers can be defined in more precise mathematical terms. Let ϑ be the set of all possible continuous membership functions over continuous sets. This can be loosely defined as

$$\vartheta = \{\mu \mid \mu: X \to [0, 1], \mu \in C^0(X)\} \tag{5}$$

Where X is any continuous set (e.g., the set of real numbers$\Re$), and $C^0(X)$ denotes the set of all continuous functions in X. A singleton fuzzifier is thus a mapping

$$sg : X \to \vartheta \quad \ni \quad \forall\, x \in X \to \mu(\bullet) = sg[x] : \mu_x(z) = f(x) = \begin{cases} 1, & z = x \\ 0, & z \neq x \end{cases} \tag{6}$$

Analogously a non-singleton fuzzifier is a mapping

$$nsg : X \to \vartheta \quad \ni \quad \forall x \varepsilon\, X \to \mu_x(\bullet) = nsg[x] : \mu_x(x) = 1 \wedge \text{Support}\,[\mu_x(\bullet)] \supset \{x\}$$

3.1.2 Defuzzifier

At the output of the fuzzy inference there will always be a fuzzy set $\mu_y(y)$ that is obtained by the composition of the fuzzy sets output by each of the rules. In order to be used in the real world, the fuzzy output needs to be interfaced to the crisp domain by the defuzzifier.

The output fuzzy set indicates what the output is in fuzzy terms. This fuzzy output will be a membership function that provides the degree of membership of several possible crisp outputs. Thus, the point corresponding to the highest degree of membership in the fuzzy output has to be sought. This operation would correspond to a type of defuzzification, called max defuzzification. Unfortunately, in most practical cases the situation is not so simple, since there might be many points having the same maximum degree of membership in the fuzzy output, and an indecision on which one of these points to choose arises. Moreover, choosing the maximum point of the membership function is an operation that discards most of the information contained in the membership function itself.

There is the need for a technique that summarizes the information contained in the membership function. The crisp output corresponding to a certain fuzzy output set should be a number that takes into account all the points in the support of this fuzzy output, weighing the points with high membership degree more than the ones with small or no membership degree. This corresponds to a center of gravity operation, and it is illustrated through an example in Fig. 3.3

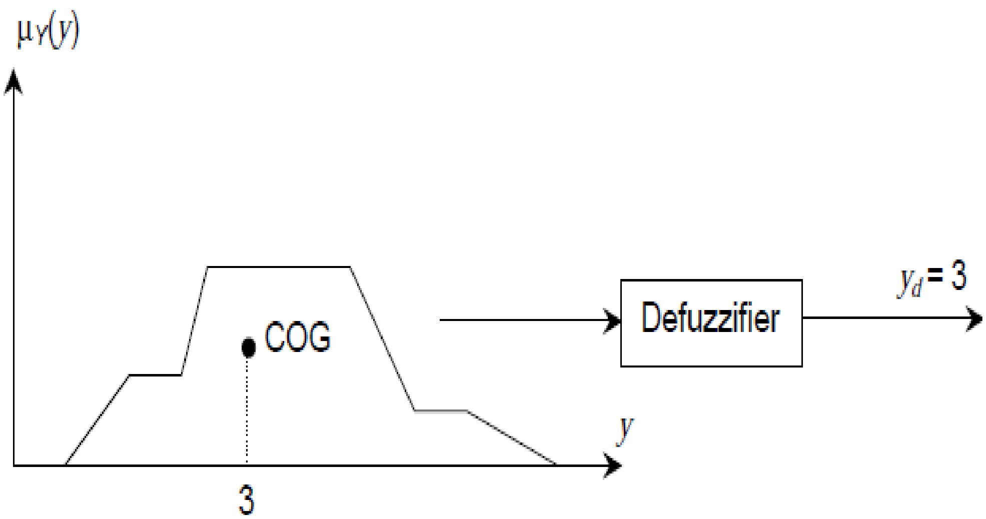


Figure 3.3: Example of defuzzification by centroid defuzzifier

Thus, one widely used defuzzifier is the centroid defuzzifier that transforms a fuzzy output set into a number that is the x-coordinate of the set's center of gravity. The output of this defuzzifier is a number y_d given by

$$y_d = \frac{\int_s y \mu_y(Y) dy}{\int_s \mu_y(Y)} \tag{7}$$

Where S is the support of $\mu_y(y)$ one drawback of this kind of defuzzification is the complexity involved with finding the center of gravity (i.e., integration). For this and other reasons for a more detailed account of defuzzification approaches), easier defuzzification schemes are generally employed for reduced computational burden.

One of the most popular defuzzifiers is the center of area (COA) defuzzifier (also called height defuzzifier). In this approach the overall center of gravity is approximated by the center of gravity of "point-masses" located at the center of gravity of each individual rule's output fuzzy set, with "mass" equal to the membership degree at that point. This is not an approximation if the supports of the fuzzy sets corresponding to the output of each rule do not overlap and the consequent membership functions are isosceles triangles with equal bases. In the general case of overlap this might be an approximation to the actual center of gravity depending also on the rule connective that is used. Calling δ_l the center of gravity of fuzzy set B^{l*} output of the l-th rule, the output of the COA defuzzifier is given by

$$\frac{\sum_{l=1}^{R} \delta_l \mu_{B^{l*}}(\delta_l)}{\sum_{l=1}^{R} \mu_{B^{l*}}(\delta_l)} \tag{8}$$

This equation is very easy to use since the centers of gravity of commonly used membership functions are known ahead of time. Regardless of the t-norm used (minimum or product) the center of gravity for commonly used symmetric membership functions (triangular, Gaussian, trapezoidal, bell shaped) does not change after inference. In other words for commonly used symmetric consequent membership functions, the center of gravity of B^l and B^{l*} is the same.

3.1.3 Inference Engine and Rule Base

Mamdani and Takagi-Sugeno fuzzy systems are the examples of fuzzy inference systems. Mamdani fuzzy inference system was first used to control a steam engine and boiler combination by a set of linguistic rules obtained from human operators. Figure 3.4 illustrates how a two rule Mamdani fuzzy inference system derives the overall output z when subjected to two numeric inputs x and y.

Takagi- Sugeno fuzzy inference system was first introduced by Takagi and Sugeno. The difference of Takagi-Sugeno model is that each rule has a crisp output, and the overall output is determined as weighted average of single rules output. This type of fuzzy inference system is shown in Figure 3.5

Sugeno output membership functions (z) are either linear or constant. A typical rule in a Sugeno fuzzy model is:

If Input 1 = x and Input 2 = y, then Output is z = ax + by + c

For a zero-order Sugeno model, the output level z is a constant (a=b =0).

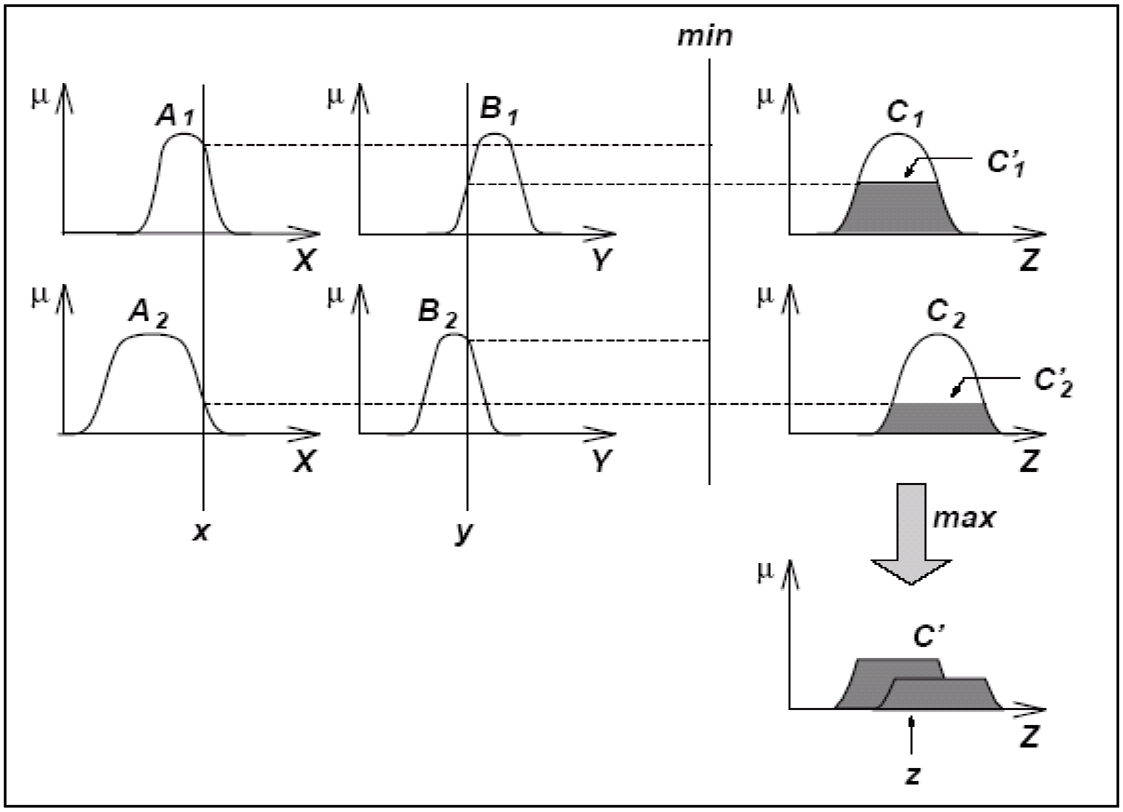


Figure 3.4: Mamdani Fuzzy Inference System

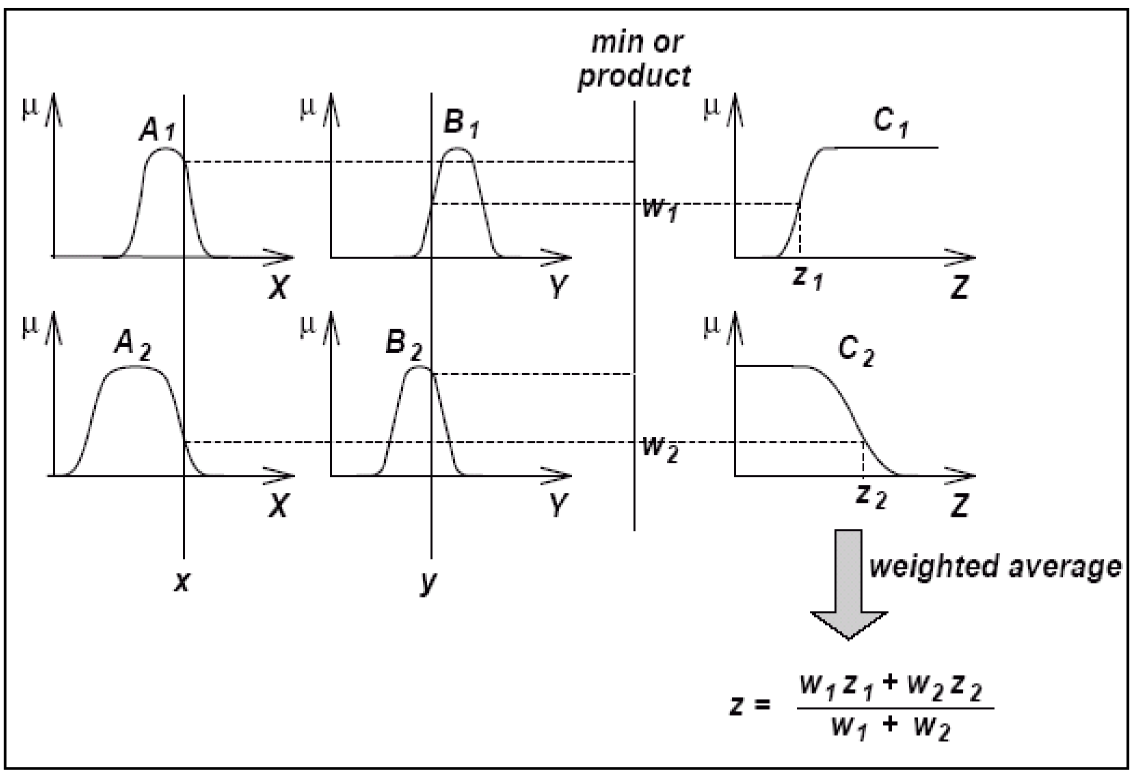


Figure 3.5: Takagi-Sugeno Fuzzy Inference System

3.1.4 Design Optimization of Fuzzy Logic Systems

Fuzzy logic was used to convert heuristic control rules, stated by a human operator, into an automatic control strategy that proved to be far better than expected. After this first pioneering application of FLSs many followed, first in Europe and later in Japan. Finally, things also caught up in the USA, where everything had started. A common denominator of all the applications of fuzzy logic was the existence of known practical approaches to the control problem. Indeed, in most of these problems there was an expert that was successful in controlling the plant and that was used as the basis in implementing a strategy in an automatic fashion. Fuzzy logic was the perfect tool for acquiring the knowledge of an expert and embedding it in a systematic and sound mathematical framework. Moreover, when an expert was not available, some easy and intuitive control rules could be stated by an understanding of the first principles at the basis of the system's functioning. The general design approach for a FLS was based on understanding the (human) expert approach to solving a control problem, implementing the strategy by direct translation of linguistic control rules, and testing the developed FLS. This process will lead to a satisfactory FLS design, but in general a sub-optimal one. The parameters of such a heuristic FLS design can be further adjusted, such that some opportunely defined performance measure is maximized. The general approach to this successive tuning has been by trial-and-error, a very common engineering practice that always yields some good practical results, but that offers no guarantees of optimality and no automatization of its evolution (i.e., man-time is "wasted" in a process that could be generally made automatic).

In this sense the structural learning or design of a FLS is characterized by the choice of fuzzification, t-norm, inference, defuzzification, number of inputs, type and number of membership functions used for each input and output, rule base (i.e., number of rules and rules). The parametric learning (or design) problem is a parametric problem, whereby the numeric value of the parameters defining a particular FLS design needs to be fixed in order to maximize a given performance metric.

The approaches to design optimization of a FLS tried to tackle the structure and parameter learning in two ways:

- Iteratively performing structural and parametric learning in an alternating fashion, until convergence is reached.

- Performing structural and parametric learning at the same time in order to reach the true optimum for the problem (the first approach might not converge to such an optimum).

While the structural learning approaches are generally heuristic, the parametric learning approaches tend to be cast in the form of optimization problems in general solved by gradient descent or variations thereof. In the following we will concentrate on the parameter-learning problem. The structure learning problem is generally solved by heuristics; it seems more intuitive to approach it on a problem-by-problem basis where, either through the help of experts or by our own understanding, we can define the necessary number of membership functions etc. Moreover, since design simplicity is always desired, we can always start with a simple structure and try to further understand where the trade-off between performance and simplicity can be set. Different t-norms and inferences can be tried, and an understanding of the best one for the problem at hand can be achieved. The parameter-learning problem is probably the most time consuming, thus some trial-and-error can be left with the structure learning process. Furthermore, techniques developed for parametric learning could be integrated with the existing structural learning methods. Finally, the parametric learning problem seems to be a good start for the problem of optimizing a given design.

3.2 Neural Network

Artificial neural networks, commonly referred to as neural networks are systems that are constructed to make use of some organizational principles similar to those of the human brain. They represent a promising new generation of information processing systems. Neural networks have a large number of highly interconnected processing elements (nodes or units). A neural network is a large and heavily parallel distributed processor inspired by the real biological neuron in the brain; therefore, it has the ability to learn, recall, and generalize as a consequence of training patterns or data. There are two types of neural network, first is biological neural network and other is artificial neural network. Basic details of these types of the neural network are given below

3.2.1 Biological Neural Network

According to the theory of modern brain science, the brain is made up of different processing elements called neurons. There are approximately 10^{11} neurons of different shapes in human brain. The neurons are like a polarized cell. The neurons have three main regions to its structure. First is cell body that is heart of the cell and contain the nucleus and maintain protein synthesis. Second is dendrites which branch out is in a treelike structure that receive signal from other neuron. Third is axon by which a neuron sends information to other neurons. The end of an axon splits into strands. Each strand terminates in a small bulblike shape called a synapse (There are approximately 10^4 synapses per neuron in a human brain), where the neuron introduces its signal to the neighboring neurons. The signal in the form of an electric impulse is then received by dendrites. This type of signal transmission involves a complex chemical process in which specific transmitter substances are released from the sending side. This raises or lowers the electric potential inside the cell body called soma of the receiving neuron. The receiving neuron fires if its electric potential reaches a certain level called threshold, and a pulse or action potential of fixed strength and duration is sent out through the axon to synaptic junctions to other neurons. After firing, a neuron has to wait for a period of time called the refractory period before it can fire again. Synapses are excitatory if they let passing impulses cause the firing of the receiving neuron, or inhibitory if they let passing impulses hinder the firing of the neuron.

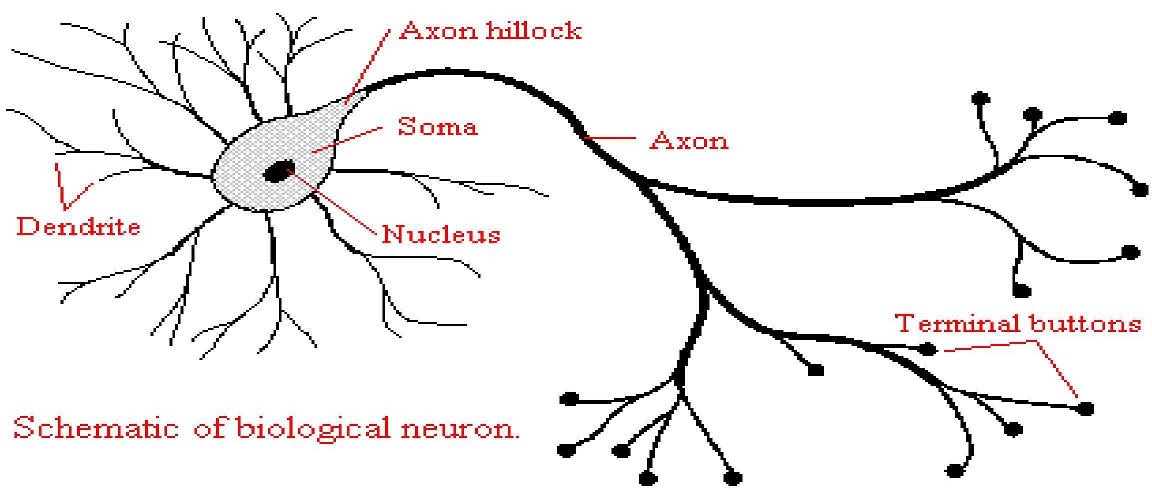


Figure 3.6: Biological Neuron

3.2.2 Artificial Neural Networks

Artificial neural network research is developed to make mathematical models of its biological counterpart in order to mimic the capabilities of biological neural structures

Warren McCulloch and Walter Pitts introduced the first mathematical model of neuron. Figure 3.7 shows an example of this model. It is known as the McCulloch- Pitts model, it does not possess any learning or adaptation capability. Many of the later neural network models use this model as the basic building block. This model consists of a single neuron, which receives a set of inputs (x_1, x_2, x_3 ……..x_n). This set of inputs is multiplied by a set of weights (w_1, w_2 ……w_n). Here, weights are referred to as strengths of the synapses. These weighted values are then summed and the output is passed through an activation (transfer) function. The activation function is also referred to as a squashing function in that it squashes (limits) the permissible range of the output signal to some finite value.

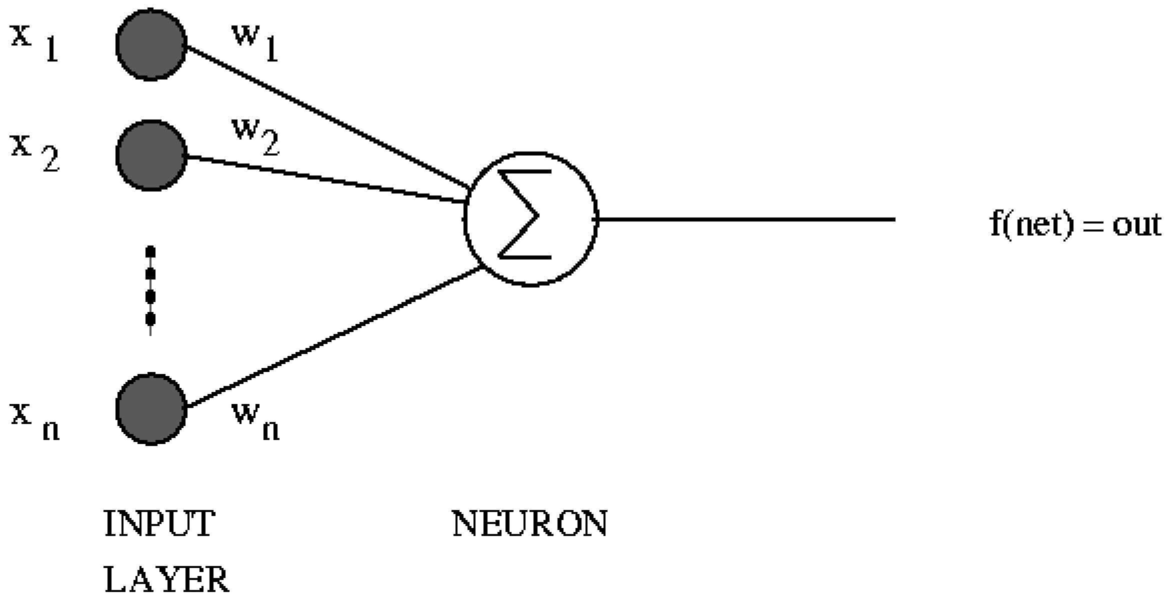


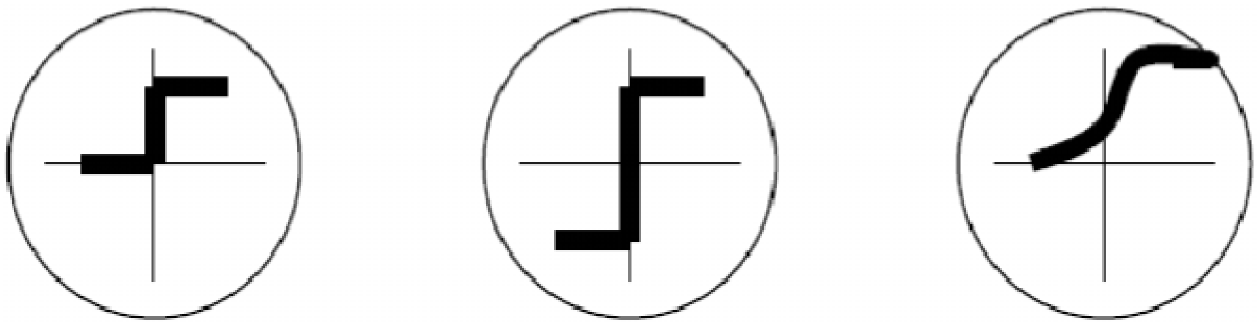
Figure 3.7: Artificial Neuron

3.2.3 Transfer Function

The behavior of an ANN (Artificial Neural Network) depends on both the weights and the input-output function (transfer function) that is specified for the units. This function typically falls into one of three categories:

- **Linear (or ramp):** the output activity is proportional to the total weighted output.
- **Threshold:** the output is set at one of two levels, depending on whether the total input is greater than or less than some threshold value.

- **Sigmoid:** the output varies continuously but not linearly as the input changes. Sigmoid units bear a greater resemblance to real neurons than do linear or threshold units, but all three must be considered rough approximations.



Step function

Step(X) =1, If

X>= threshold

X< threshold

Sign function

Sign(X) = +1

If X >= 0

Sign (X) = -1

If X < 0

Sigmoid Function

Sigmoid (X) = $\frac{1}{(1+e^{x})}$

Figure3.8: Transfer Function

3.2.4 Type of Neurons

Neuron can be classified in two types namely, simple neuron and complicated neuron. Following is the basic information of these two given in detail:

- **Simple neuron** An artificial neuron is a device with many inputs and one output. The neuron has two modes of operation; the training mode and the using mode. In the training mode, the neuron can be trained to fire (or not), for particular input patterns. In the using mode, when a taught input pattern is detected at the input, its associated output becomes the current output. If the input pattern does not belong in the taught list of input patterns, the firing rule is used to determine whether to fire or not. Figure 3.9 show an example of simple neuron.

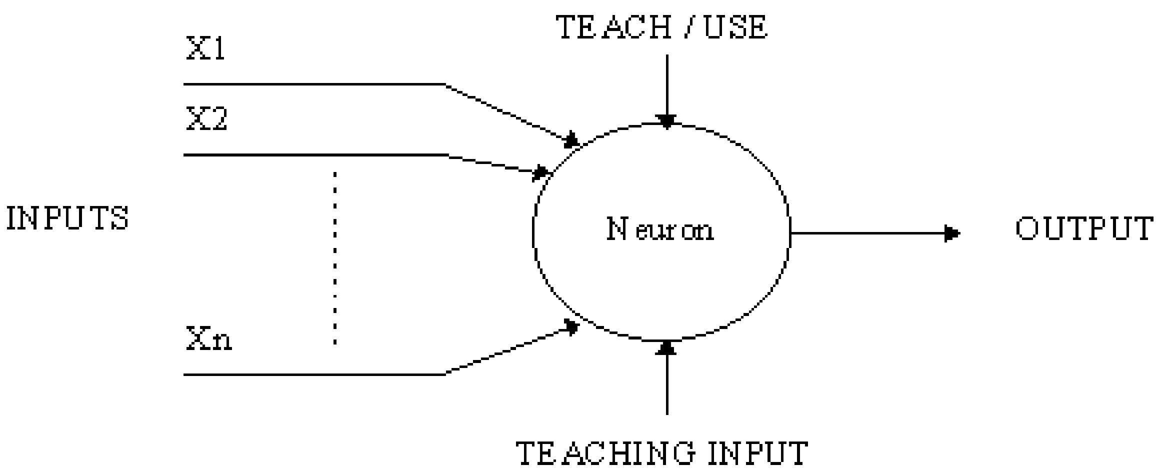


Figure 3.9: Simple Neuron

- **Complicated neuron**: The simple neuron doesn't do anything that conventional computers don't do already. Figure 3.10 is the example of complicated neuron. The difference from the previous model is that the inputs are ‘weighted’; the effect that each input has at decision making is dependent on the weight of the particular input. The weight of an input is a number which when multiplied with the input gives the weighted input. These weighted inputs are then added together and if they exceed a pre-set threshold value, the neuron fires. In any other case the neuron does not fire.

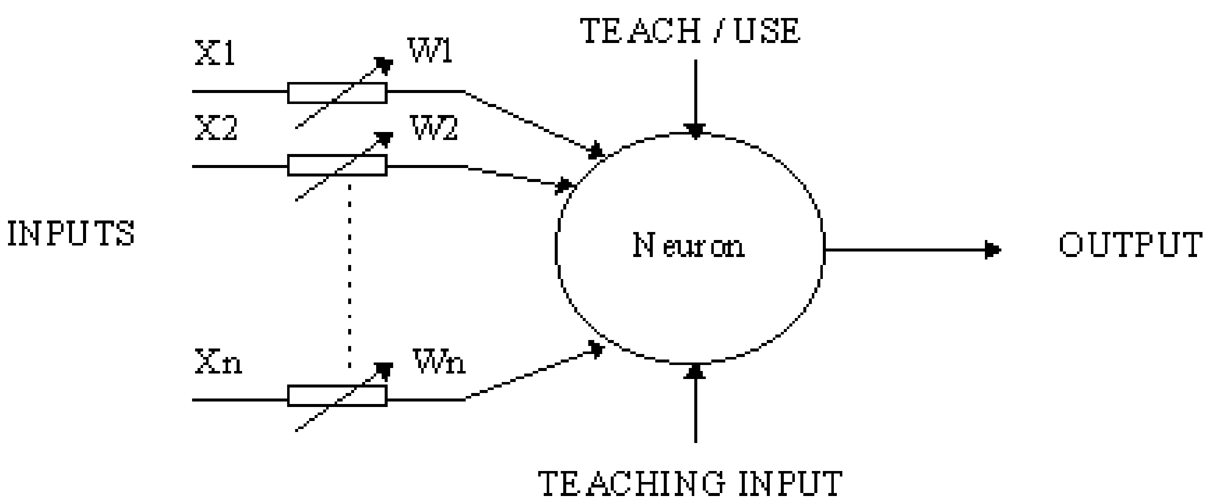


Figure 3.10: A Complicated Neuron

In mathematical terms, the neuron fires if and only if;

$$X_1 W_1 + X_2 W_2 + X_3 W_3 + \ldots > T \qquad (9)$$

The addition of input weights and of the threshold makes this neuron a very flexible and powerful one. The complicated neuron has the ability to adapt to a particular situation by changing its weights and/or threshold. Various algorithms exist that cause the neuron to 'adapt'; the most used ones are the Delta rule and the back error propagation.

3.3 Genetic Algorithms

Genetic Algorithms (GA's) are a stochastic global search method that mimics the process of natural evolution. The genetic algorithm starts with no knowledge of the correct solution and depends entirely on responses from its environment and evolution operators (i.e. reproduction, crossover and mutation) to arrive at the best solution. By starting at several independent points and searching in parallel, the algorithm avoids local minima and converging to sub optimal solutions.

In this way, GAs have been shown to be capable of locating high performance areas in complex domains without experiencing the difficulties associated with high dimensionality, as may occur with gradient decent techniques or methods that rely on derivative information .

A genetic algorithm is typically initialized with a random population consisting of between 20-100 individuals. This population (mating pool) is usually represented by a real-valued number or a binary string called a chromosome. For illustrative purposes, the rest of this section represents each chromosome as a binary string. How well an individual performs a task is measured is assessed by the objective function. The objective function assigns each individual a corresponding number called its fitness. The fitness of each chromosome is assessed and a survival of the fittest strategy is applied. In this project, the magnitude of the error will be used to assess the fitness of each chromosome.

There are three main stages of a genetic algorithm; these are known as reproduction, crossover and mutation.

3.3.1: Reproduction

During the reproduction phase the fitness value of each chromosome is assessed. This value is used in the selection process to provide bias towards fitter individuals. Just like in natural evolution, a fit chromosome has a higher probability of being selected for reproduction. An example of a common selection technique is the ‘Roulette Wheel’ selection method, Figure 3.11. Each individual in the population is allocated a section of a roulette wheel; the size of the section is proportional to the fitness of the individual. A pointer is spun and the individual to whom it points is selected. This continues until the selection criterion has been met. The probability of an individual being selected is thus related to its fitness, ensuring that fitter individuals are more likely to leave offspring.

Multiple copies of the same string may be selected for reproduction and the fitter strings should begin to dominate. However, for the situation illustrated in Figure 3.11, it is not implausible for the weakest string (01001) to dominate the selection process.

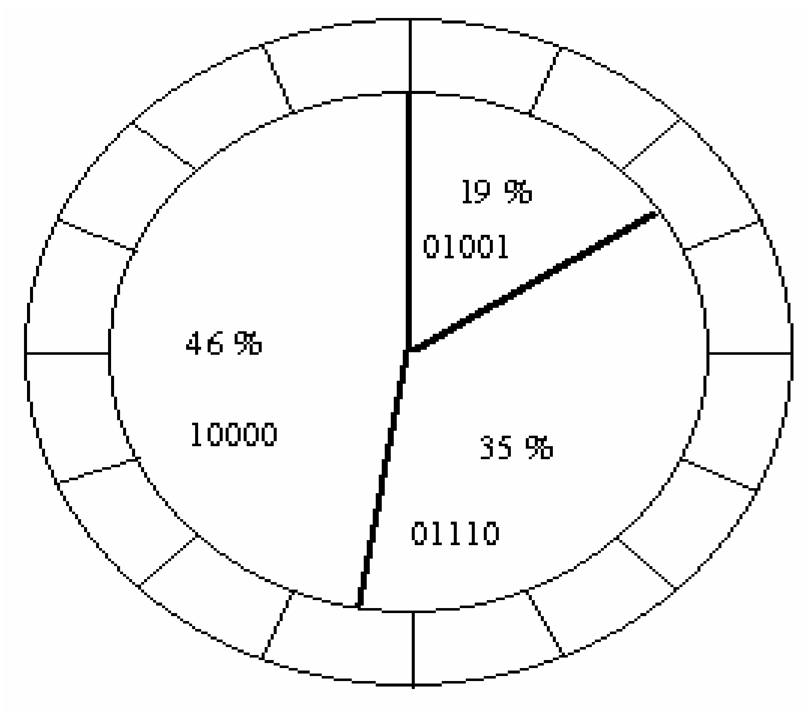


Figure 3.11: Depiction of roulette wheel selection.

There are a number of other selection methods available and it is up to the user to select the appropriate one for each process. All selection methods are based on the same principal i.e. giving fitter chromosomes a larger probability of selection. Four common methods for selection are:

- Roulette Wheel selection
- Stochastic Universal sampling
- Normalized geometric selection
- Tournament selection.

3.3.2 Crossover

Once the selection process is complete, the crossover algorithm is initiated. The crossover operations swap certain parts of the two selected strings in a bid to capture the good parts of old chromosomes and create better new ones. Genetic operators manipulate the characters of a chromosome directly, using the assumption that certain individual's gene codes, on average, produce fitter individuals. The crossover probability indicates how often crossover is performed. A probability of 0% means that the 'offspring' will be exact replicas of their 'parents' and a probability of 100% means that each generation will be composed of entirely new offspring. The simplest crossover technique is the Single Point Crossover. There are two stages involved in single point crossover:

- Members of the newly reproduced strings in the mating pool are 'mated' (paired) at random.
- Each pair of strings undergoes a crossover as follows: An integer k is randomly selected between one and the length of the string less one, [1,L-1]. Swapping all the characters between positions k+1 and L inclusively creates two new strings.

More complex crossover techniques exist in the form of Multi-point and Uniform Crossover Algorithms. Multi-point crossover is an extension of the single point crossover algorithm and operates on the principle that the parts of a chromosome that contribute most to its fitness might not be adjacent. There are three main stages involved in a Multi-point crossover.

- Members of the newly reproduced strings in the mating pool are 'mated' (paired) at random.
- Multiple positions are selected randomly with no duplicates and sorted into ascending order.
- The bits between successive crossover points are exchanged to produce new offspring.

3.3.3 Mutation

Using selection and crossover on their own will generate a large amount of different strings. However there are two main problems with this:

- Depending on the initial population chosen, there may not be enough diversity in the initial strings to ensure the GA searches the entire problem space.
- The GA may converge on sub-optimum strings due to a bad choice of initial population.

These problems may be overcome by the introduction of a mutation operator into the GA. Mutation is the occasional random alteration of a value of a string position. It is considered a background operator in the genetic algorithm

The probability of mutation is normally low because a high mutation rate would destroy fit strings and degenerate the genetic algorithm into a random search. Mutation probability values of around 0.1% or 0.01% are common, these values represent the probability that a certain string will be selected for mutation i.e. for a probability of 0.1%; one string in one thousand will be selected for mutation.

CHAPTER 4

LITERATURE SURVEY

Undesired vibration can disturb our comfort, damage to structures, reduction of tools performance and machinery noise level but somehow it can be channeled for producing or extracting energy rather than suppress it . However, the command practice that being implemented is to control and suppress the undesirable vibration because it is creating unwanted sound (noise), the secondary major problem. One of the effects of undesired vibration is the Hand-arm Vibration Syndrome (HAVS). This syndrome involves circulatory disorders (e.g. vibration white finger), sensory and motor disorders, and musculoskeletal disorders, which may occur in workers who use vibrating powered portable tools.

A few studies related to hand-transmitted vibration has been revealed and show a very serious phenomenon. The effects on the operator exposed to a high vibration level over a long period of time are usually permanent in character, and are therefore considered to be an occupational disease leading to invalidity [1].

Goglia in his research reported that under the given measuring condition, the whole-body vibration transmitted to a frame saw operator during an ordinary working day, is exposed to the higher vibration level than the guidelines given in the new ISO 2631-1-1997 [2].

A survey in 1997/98 gave a national prevalence estimate of 301,000 sufferers from vibration white finger (VWF) a vascular disorder caused by hand and arm exposure to vibration [3].

In 2002, European Commission reveals that 17% of the European workers were reportedly being exposed to vibration from handheld tools or machinery for at least half of their working time [4]. The Spanish scenario reveals that 22.8% of the workers that use portable electric and pneumatic tools were reportedly being exposed to vibration. In order to protect the workers from HAVS, different countries have proposed and developed their own criteria. For the example, the European Union (EU)

Sweden National Institute for Working Life [5], and OPERC database [6] has formed Human Vibration Directive and Hand-Arm Vibration Database respectively. Their main objective is to measure a vibration level of handheld powered tools, reveals the effect of HAVS to human body, etc. Different frequencies of vibration interfere with different body parts and systems where whole body vibration occurs at frequencies from l-30 Hz, while segmental vibration, which interferes with the hand-arm system, is between 30-100 Hz [7]. Above 100 Hz threshold, the hand in particular is affected [8].

Yildrim [9] has presented a neural network scheme for controlling a bus suspension system and comparing it to PID, PI and PD control schemes. The finding shows that neural network scheme gives better robust performance compared to other control schemes. An AVC scheme called Linear Quadratic Gaussian (LQG) control has been implemented in woodcutting machine.

By Chen to reduce saw blade vibration and sawdust [10]. He found that vibration of the saw blade is the key reason for poor wood recovery. This technique involves the use of magnetic actuator which produces a counter force to suppress the vibration of the saw blade.

Friction-induced vibration due to a sliding contact between two systems is a phenomenon that arises across a diverse range of scales and contexts, including musical instruments, machine tool vibration, railway wheel noise, earthquakes and vehicle brake squeal. The literature on brake squeal has moved from very simple lumped parameter models to ever more sophisticated finite element models, but testing theory against measurements has always proved difficult.

This research has been summarized by North [11], Ibrahim and Rivin [12], Kinkaid et al. [13] and Ouyang et al. [14]. A recurring theme is the difficulty in obtaining repeatable experimental results that correlate with theoretical models: to date, there is no validated predictive model of friction-induced vibration. This difficulty in validating theoretical models suggests that friction-coupled systems are highly sensitive to parameter changes beyond an experimenter’s control.

The study of sensitivity in such systems has now begun to attract some attention (e.g. Guan et al. [15] and Huang et al. [16]. This research follows and extends a theoretical model developed by Duffour and Woodhouse [17, 18] which allows for direct testing against experimental data. The model, summarized in the next section, is based on a linear transfer function analysis of two subsystems coupled by a sliding point contact. The predicted system stability depends on the values of the complex zeros of two characteristic equations. Although a simplified analysis, it is general enough for the theory to be directly applicable to results from a test rig. Obtaining repeatable results is still difficult, so this research focuses on exploring the reasons for the sporadic nature of squeal. Initial results show that even measurement uncertainty is sometimes enough to significantly affect predictions, and that the sensitivity of predictions is highly dependent on the system parameters (see Butlin and Woodhouse [19]).

The brake system of an automobile typically consists of the contact metallic solids rubbing against each other, which frequently generates undesirable noise and vibrations. Thus, noise generation and suppression have become an important factor to be considered in the design and manufacture of brake components.

By Abendroth and Wernitz [20], a large number of manufacturers of brake pad materials spend up to 50% of their engineering costs on issues related to noise, vibration and harshness.

By N.M. Kinkaid et all [21] brake noise vibration phenomena are described by a number of terminologies that are sometimes interchangeably used such as squeal, groom, chatter, judder, moan and hum. There is no precise definition of brake squeal that has gained complete acceptance. It is also worth mentioning that since in a vehicle with disc brakes installed at the front wheels while drum brakes at the back wheels, around 70% of the braking action occurs at the front wheels. Thus, it is expected that most of the noise and squeal is coming from the front disc brake system.

By Yi Dai, Teik C. Lim [22] has been carried out for reducing brake noise using finite element (FE) method. They developed a dynamic FE model of the brake system, and based on their analysis, the pad design changes can be made in the FE model to determine the potential improvements in the dynamic stability of the system and also in noise reduction.

By Wagner et al. [23] proposed a new mathematical rotor based model of a brake system that is suitable for noise analysis. Brief descriptions of the previous mathematical models that have been developed by other researchers were explicitly outlined in their study.

Besides, there is also an active control method known as dither control which makes use of high frequency disturbance signal for the suppression of the automotive disc brake squeal. Through this scheme, the dither signal stabilizes friction induced self-oscillations in the disc brake using a harmonic vibration, with a frequency higher than the squeal frequency generated from a stack of piezoelectric elements placed in the caliper piston of the brake system. The resulting control vibration was not heard from the brake system if an ultrasonic control signal was activated. This system assumes an open loop control mode in which there is no requirement to detect the presence of squeal and is much simpler in design than the feedback control [24].

By N. Hoffmann, N. Wagner, and L. Gaul [25] presents a closed loop control employing system Active Force Control (AFC) with PID element applied to a brake model. In order to suppress the brake noise and squeal the main advantage of the AFC technique is its ability to reject disturbances that are applied to the system through appropriate manipulation of the selected parameters.

AFC as first proposed by Hewitt and Burdess [26] is very robust and effective in controlling a robot arm.

By Mailah [27] has successfully demonstrated the application of the technique to include many other dynamical systems with the incorporation of artificial intelligence (AI) methods.

By Chatterjee and Mahata (2009) [28] employed an active absorber based on the time-delayed displacement difference feedback in controlling friction-driven vibrations. The local stability analysis shows that the static equilibrium can be locally stabilized by suitably selecting the control gain and the time-delay. The regions of stability are delineated in the plane of the control parameters. Numerical simulations of the system show that proper selections of the control parameters can also achieve the global stability of the system.

By S.M. Hashemi-Dehkordi et al. (2008) [29] a novel approach to suppress vibration that causes brake noise is proposed employing a closed-loop feedback control method using an Active Force Control (AFC) based strategy. It is used in conjunction with the classic proportional-integral-derivative (PID) scheme that is typically incorporated in the outermost positional control loop. The idea is to introduce an active element that dynamically compensates the disturbances through a control mechanism that takes into account the direct measurements and estimation of parameters in the AFC section. A disc brake model is considered and simulated taking into account a number of operating and loading conditions. Results clearly show the superiority of the proposed AFC-based scheme compared to the pure PID counterpart in suppressing the vibration and hence the brake noise.

By Sayed-Mahdi Hashemi-Dehkordi et al. (2009) [30] a novel approach to reduce the effect of negative damping that causes brake noise is proposed by applying an Active Force Control (AFC) based strategy to a two degree-of-freedom model of a disk brake system. At first, the disc brake model is simulated and analyzed using a closed loop pure PID controller. Later, it is integrated with AFC and simulated under similar operating environment. After running several tests with different sets of operating and loading conditions, the results both in time and frequency domains show that the PID controller with AFC is much more effective in reducing the vibration and noise, compared to the pure PID controller alone.

By S.M. Hashemi-Dehkordi et al. (2010) [31] this paper serves to highlight a potential and effective active force control (AFC) based scheme to suppress friction induced vibration that is caused by the mechanisms of negative damping and mode-coupling. Two mathematical models that are based on Shin and Hoffmann schemes are first simulated and analyzed using a conventional closed loop proportional-integral-derivative (PID) controller. Later, the models were seamlessly integrated with AFC elements to develop into a two degree-of-freedom (DOF) controller that is designed to effectively reject the disturbances and consequently reduce the vibrations in both models. It is found that the integrated PID-AFC scheme is very effective in suppressing vibration compared to the pure PID controller alone as clearly demonstrated through the results presented both in time and frequency domains.

CHAPTER 5

VIBRATION AND ITS EFFECTS

Vibration is time dependent displacements of a particle or a system of particles w.r.t an equilibrium position. If these displacements are repetitive and their repetitions are executed at equal interval of time w.r.t equilibrium position the resulting motion is said to be periodic.

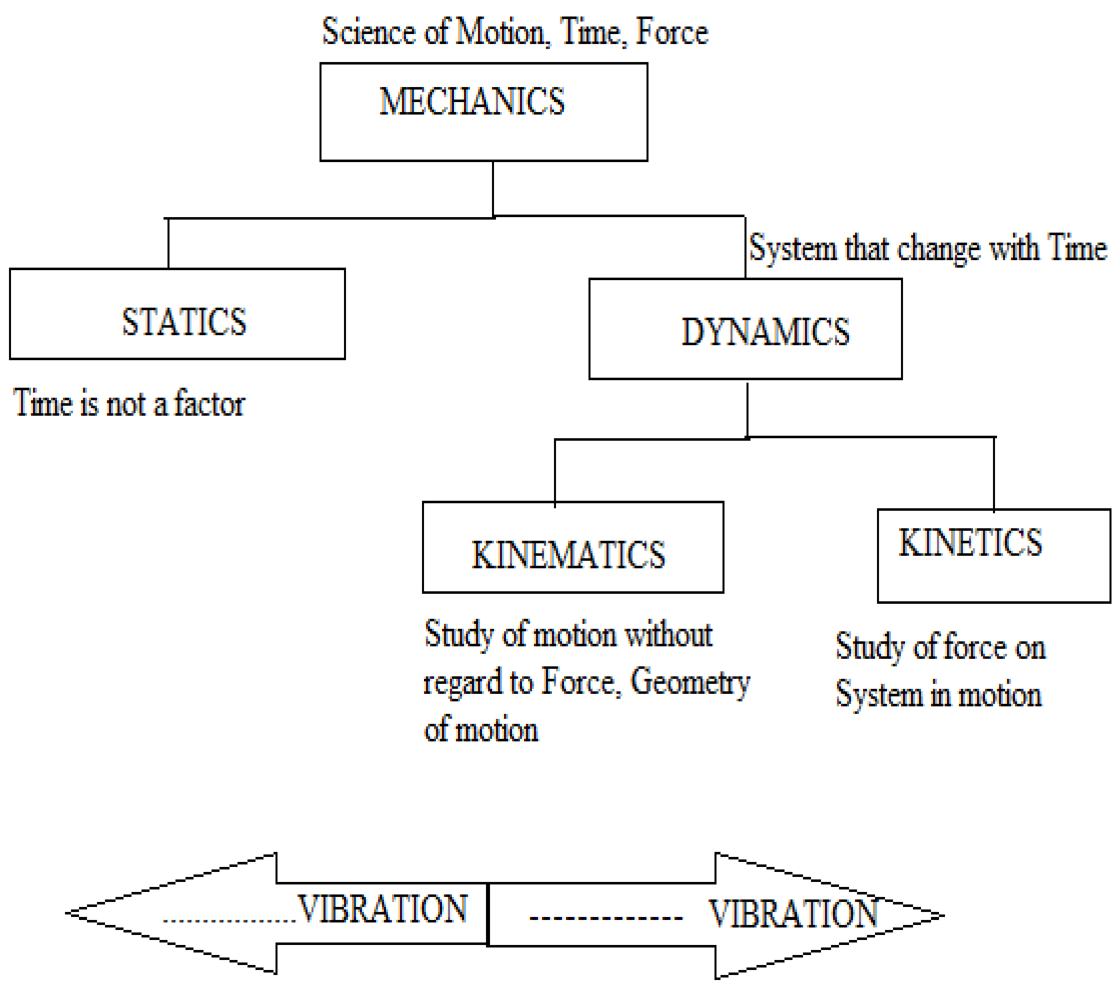


Figure 5.1: Vibration Mechanics

5.1: Classification of Vibration

Vibration can be classified in several ways. Some of the important classification is as follows:

5.1.1: Free and forced Vibration

If a system, after an internal disturbance, is left to vibrate on its own, the ensuing vibration is known as free vibration. No external force acts on the system. The oscillation of the simple pendulum is an example of free vibration.

If a system is subjected to an external force (often, a repeatating type of force), the resulting vibration is known as forced vibration. The oscillation that arises in machineries such as diesel engines is an example of forced vibration.

If the frequency of the external force coincides with one of the natural frequencies of the system, a condition known as resonance occurs, and the system undergoes dangerously large oscillations. Failures of such structures as buildings, bridges, turbines and airplane have been associated with the occurrence of resonance.

5.1.2: Undamped and damped Vibration

If no energy is lost or dissipated in friction or other resistance during oscillation, the vibration is known as undamped vibration. If any energy lost in this way, however, it is called damped vibration. In many physical systems, the amount of damping is so small that it can be disregrated for most engineering purposes. However, consideration of damping becomes extermely important in analyzing vibratory system near resonance.

5.1.3: Linear and nonlinear Vibration

If all the basic components of vibratory system—the spring, the mass and the damper—behave linearly, the resulting vibration is known as linear vibration. if ,however, any if the basic components behave non linearly, the vibration is called non linear vibration.

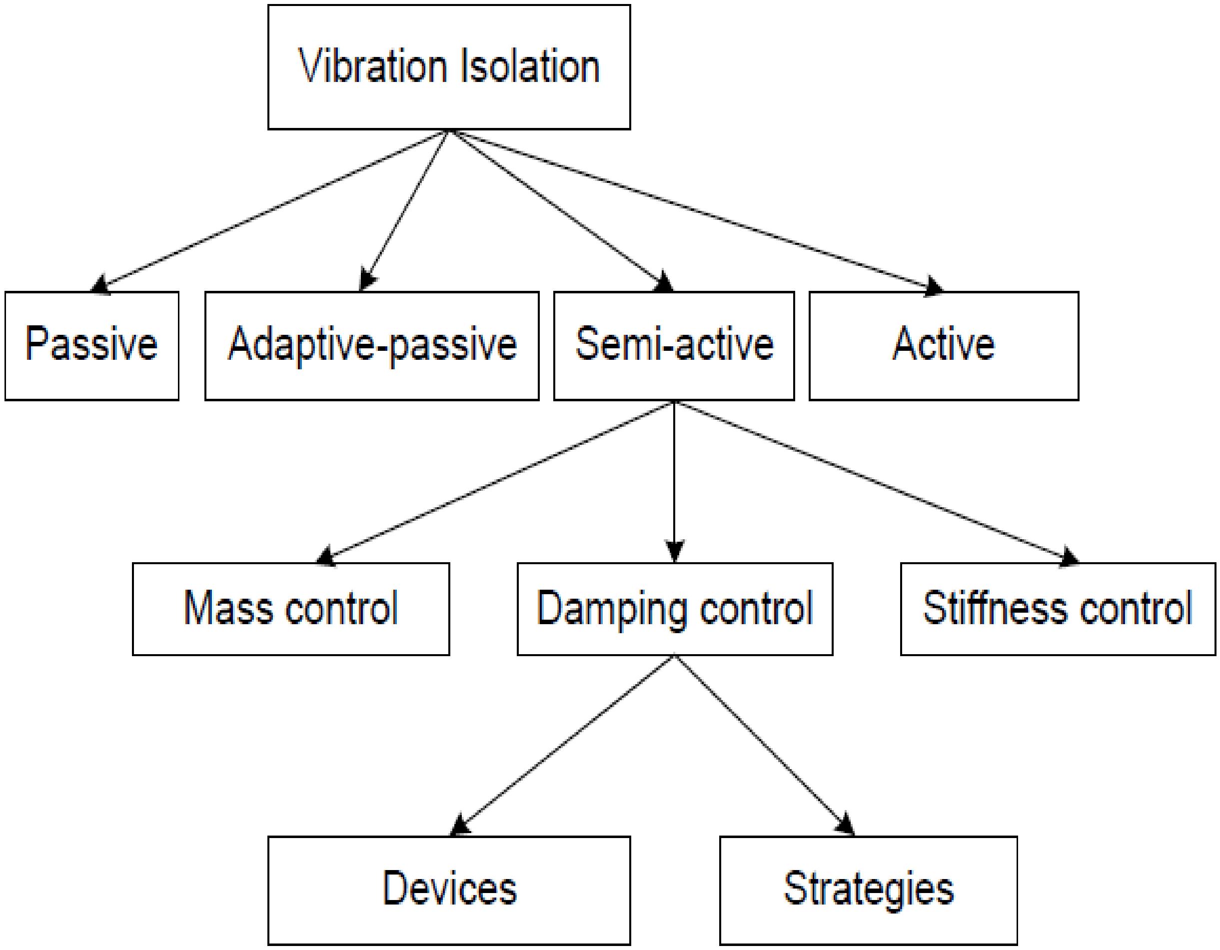


Figure 5.2: Deterministic Signal

5.2: Vibration Control

Vibration can be pleasant and useful as in massaging and therapeutic application but at the same time it can be unpleasant and harmful too because it can lead to catastrophic failures of components and materials. Unwanted vibration can interfere with our comfort, damage to structures, and reduction of equipment performance and machinery noise level. For examples, vibration of car or any other automobile can lead to driver discomfort and eventually fatigue. Electronic component used in automobile or planes, machines, structures and so on may also fail because of vibration. Furthermore, a vibration environment can cause malfunction or failure of mechanical systems and may cause injury to human beings. Therefore, it is envisaged that a form of vibration control or suppressor is needed to compensate the undesirable vibration effect so that it will not damage the systems or cause injury/discomfort to human beings.

Vibration absorber and vibration isolator are two important concepts of vibration control. Vibration isolation is a common method used to protect receivers (device, human, system, machine or structure) from the vibration source. Car suspension system is one example of a system that employs an isolation concept in an effort to reduce the car body displacement due to the road profile. In this context, vibration isolation or vibration 'absorption' is primarily realized by passive methods which deal directly with the physical properties of a mechanical structure itself that is related to stiffness, damping and mass. The performances of the passive systems are highly system dependent as they are unable to adapt or re-tune to changing disturbances or structural characteristics over time

5.3: Modeling of physical (dynamic) systems

A mathematical model of a dynamic system is defined as a set of equations that represents the dynamics of the system accurately. A mathematical model is not unique to a given system. A system may be represented in many different ways and, therefore, may have many mathematical models, depending on one's perspective.

The dynamics of many systems, whether they are mechanical, electrical, thermal, economic, biological, and so on, may be described in terms of differential equations. Such differential equations may be obtained by using physical laws governing a particular system, for example, Newton's laws for mechanical systems and Kirchhoff's laws for electrical systems. It must be kept in mind that deriving reasonable mathematical models is the most important part of the entire analysis of control systems.

Mathematical models may assume many different forms. Depending on the particular system and the particular circumstances, one mathematical model may be better suited than other models. For example, in optimal control problems, it is advantageous to use state-space representations. On the other hand, for the transient-response or frequency-response analysis of single-input-single-output, linear, time-invariant systems, the transfer function representation may be more convenient than any other. Once a mathematical model of as system is obtained, various analytical and computer tools (e.g. MATLAB) can be used for analysis and synthesis purposes.

5.3.1: Modeling a spring mass damper system

Based on the nature of the mathematical model used, the system may be called a discrete (or lumped) system or a continuous (or distributed) system. In the discrete model, the physical system is assumed to consist of several rigid bodies (usually considered as point masses) connected by springs and dampers. The springs denote restoring forces that tend to return the masses to their respective undisturbed (or equilibrium) states. The dampers provide resistance to velocity and dissipate the energy of the system.

In the continuous model, the mass, elasticity, and damping are assumed to be distributed throughout the system. The equations of motion of a discrete system are in the form of a system of n coupled second order ordinary differential equations, where n denotes the number of masses (discrete masses or rigid bodies). The number of independent coordinates needed to describe the configuration of a system at any time during vibration defines the degrees of freedom of the system. For example, Figures. 5.3, 5.4 and 5.5 denote typical one-, two-, and three-degree-of-freedom systems, respectively.

A point mass can have three translational degrees of freedom while a rigid body can have three translational and three rotational degrees of freedom. Many mechanical and structural components and systems such as bars, beams, plates, and shells have distributed mass, elasticity, and damping. The equation of motion of a continuous system is in the form of a partial differential equation. A continuous system can be modeled either as a discrete- or lumped parameter system with varying number of degrees of freedom or as a continuous system with infinite number of degrees of freedom.

The oscillatory motion of a body may be harmonic, periodic, or non periodic in nature. If the time variation of the displacement of the mass is sinusoidal, the motion will be harmonic. The number of cycles of motion per unit time defines the frequency, and the maximum magnitude of motion is called the amplitude of vibration. If the periodic variation of motion is not harmonics, the motion will be periodic. In this case, the periodic motion can be expressed as a sum of harmonic motions of different frequencies. If the time variation of the displacement of the mass is arbitrary (non periodic), the motion is said to be non periodic.

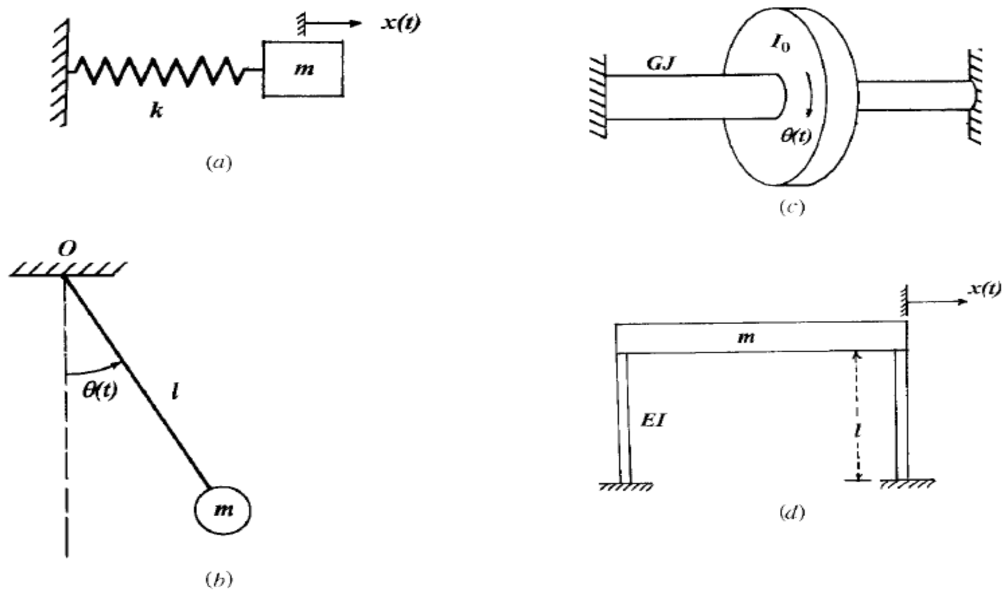


Figure 5.3 Typical One-degree-of freedom systems

If the non periodic motion can be described either by an equation or by a set of tabulated values, the motion is considered to be deterministic. On the other hand, if the motion cannot be described by any equation or tabulated values, it is said to be random or probabilistic. When an external force or excitation is applied to a mechanical or structural system, the amplitude of the resulting vibration can become very large when a frequency component of the applied force or excitation approaches one of the natural frequencies of the system, particularly the fundamental one. Such a condition, known as resonance, and the attendant stresses and strains might cause a failure of the system. Because of this, designers should have a means of determining the natural frequencies of mechanical and structural systems using analytical or experimental approaches.

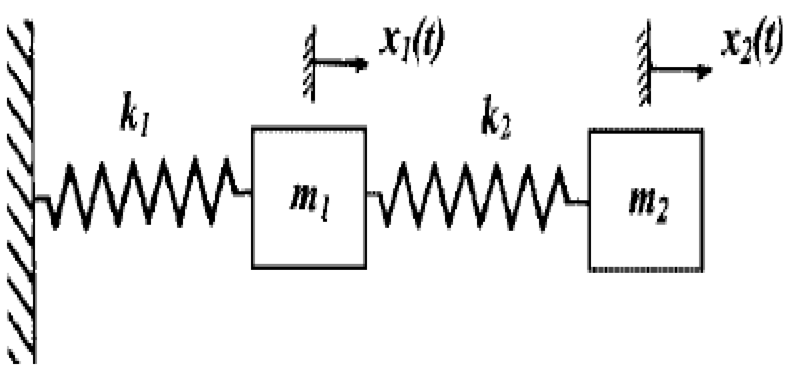


Figure 5.4 Two-degree-of-freedom systems

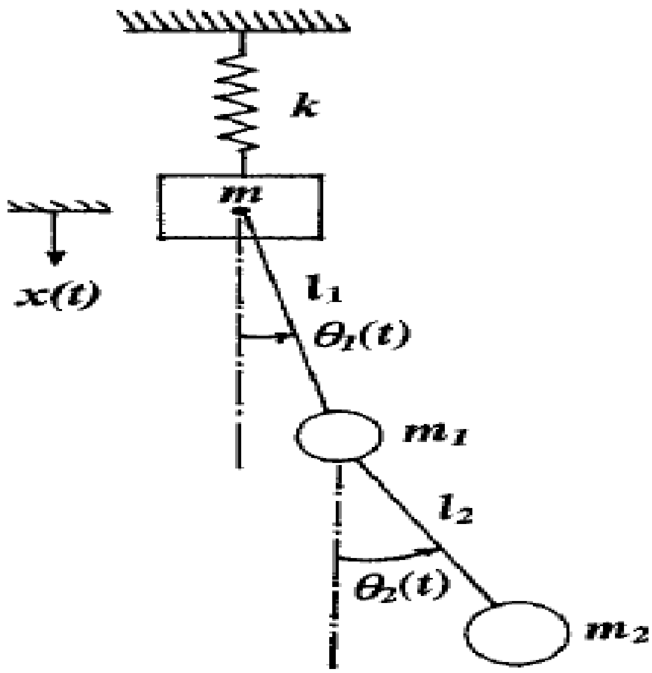


Figure 5.5 Three-degree-of-freedom systems

5.3.2: Single-degree-of-freedom system

A degree of freedom is a single coordinate of relative motion between two bodies. Such a coordinate is free only if it can respond without constraint or imposed motion to externally applied forces or torques. For translational motion, a DOF is a linear coordinate along a single direction. For rotational motion, a DOF is an angular coordinate about a single, fixed axis.

A prismati*c* joint primitive represents a single translational DOF. A revolute joint primitive represents a single rotational DOF. A spherical joint primitive represents three rotational DOFs in angle-axis form. A weld joint primitive represents zero DOFs.

A study of the vibration characteristics of a single-degree-of-freedom-system is extremely important in the study of vibration and shock because the approximate or qualitative response of most systems can be determined by using a single-degree-of-freedom model for the system.

A general single-degree-of-freedom system consists of a mass m, a spring of stiffness k, and a viscous damper with a damping constant c, as shown in Fig. 5.3a, the significance of the quantities m, c and k for different types of systems is given in Table 5.1

The equation of motion is given by:

$$mx + cx + kx = F(t) \quad (10)$$

Where $\dot{x}$ denote first and $\ddot{x}$ second derivatives respectively.

Table 5.1 Significance of m, c, and k in different systems

S. No.	Vibrating System	M	C	K	Variable x
1	Translatory spring-mass-damper system Fig. 5.3a	Mass (kg)	Viscous damping constant (N.s/m)	Spring stiffness (N/m)	Linear displacement (m)
2	Rotational spring mass-damper system Fig.5.3c	Mass moment of inertia($kg.m^2$)	Torsional damping constant(m.N.s/rad)	Torsional spring stiffness (m.N/rad)	Angular displacement (rad)
3	Swinging pendulum, Fig. 5.3b	Moment of inertia of bob($kg.m^2$)	Damping constant of surrounding medium(m.N.s/rad)	Angular stiffness constant due to Gravity(N.m/rad)	Angular displacement (rad)
4	Transversely vibrating cantilever beam,Fig5.3d	Mass at end of beam (kg)	Damping constant due to surrounding medium(N.s/m)	Flexural stiffness of beam (N/m)	Transverse displacement of mass at end of cantilever(m)

5.3.3: Multi-degree-of-freedom system

Most mechanical and structural systems have distributed mass, elasticity, and damping. These systems are modeled as multi- (n-) degree-of-freedom systems to facilitate analysis of their vibration behavior. Several methods are available to construct an n degree-of-freedom model from a continuous system. These include the physical lumping or modeling method, finite element method, finite difference method, modal analysis method, Rayleigh–Ritz method, Galerkin method, and many others.

In most cases, the number of degrees of freedom (n) to be used in the model depends on the frequency range. If the system is expected to undergo significant deformations at higher frequencies, the model should include enough number of degrees of freedom to cover all the important frequencies. Most vibration characteristics of a n-degree-of-freedom system are similar to those of a single-degree-of-freedom system. An n-degree-of-freedom system will have n natural frequencies, its free vibrations denote exponentially decaying motions, its forced vibrations exhibit resonance behavior, etc. However, there are some vibration characteristics that are unique to an n-degree-of-freedom system which are absent in single-degree-of-freedom systems.

5.4: Common practical examples of mass spring damper system

These include:

Automobile suspension system

Quarter car model

Tuned mass damper

Muscles and tendons in the human body

5.4.1: Automobile suspension system

The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system and/or a control bandwidth. A passive suspension system is a conventional suspension system consists of a non-controlled spring and shock-absorbing damper as shown in figure 5.6. The semi-active suspension as shown in figure 5.5 has the same elements but the damper has two or more selectable damping rate. An active suspension is one in which the passive components are augmented by actuators that supply additional force. Besides these three types of suspension systems, a skyhook type damper has been considered in the early design of the active suspension system. In the skyhook damper suspension system, an imaginary damper is placed between the sprung mass and the sky. The imaginary damper provides a force on the vehicle body proportional to the sprung mass absolute velocity.

5.4.1.1: Passive Suspension System

The commercial vehicles today use passive suspension system to control the dynamics of a vehicle's vertical motion as well as pitch and roll. Passive indicates that the suspension elements cannot supply energy to the suspension system. The passive suspension system controls the motion of the body and wheel by limiting their relative velocities to a rate that gives the desired ride characteristics. This is achieved by using some type of damping element placed between the body and the wheels of the vehicle, such as hydraulic shock absorber.

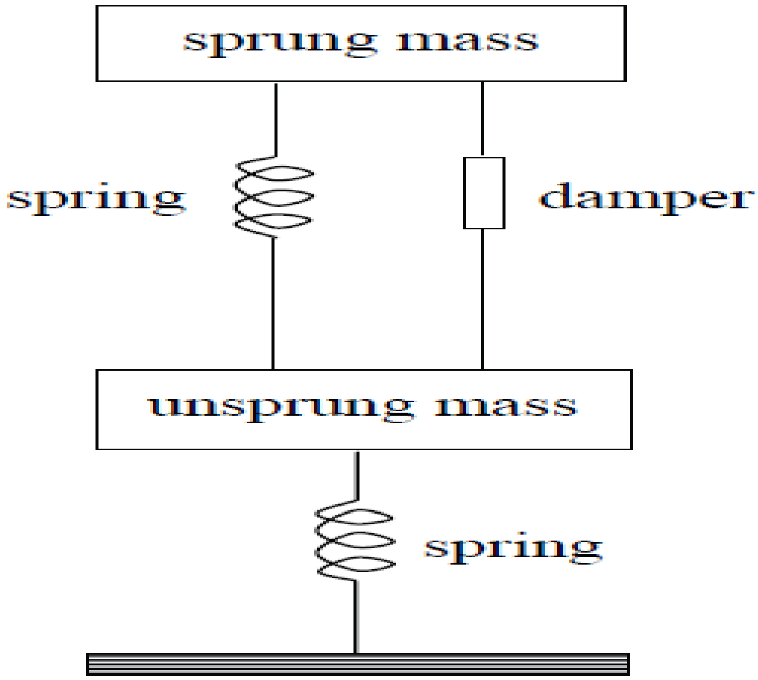


Figure 5.6 Passive suspension systems

5.4.1.2: Semi-Active Suspension System

In early semi-active suspension system, the regulating of the damping force can be achieved by utilizing the controlled dampers under closed loop control, and such is only capable of dissipating energy. Two types of dampers are used in the semi- active suspension namely the two state dampers and the continuous variable dampers. The two state dampers switched rapidly between states under closed-loop control. The disadvantage of this system is that while it controls the body frequencies effectively, the rapid switching, particularly when there are high velocities across the dampers, generates high-frequency harmonics which makes the suspension feel harsh, and leads to the generation of unacceptable noise.

The continuous variable dampers have a characteristic that can be rapidly varied over a wide range. When the body velocity and damper velocity are in the same direction, the damper force is controlled to emulate the skyhook damper. When they are in the opposite directions, the damper is switched to its lower rate, this being the closest it can get to the ideal skyhook force. The disadvantage of the continuous variable damper is that it is difficult to find devices that are capable in generating a high force at low velocities and a low force at high velocities, and be able to move rapidly between the two.

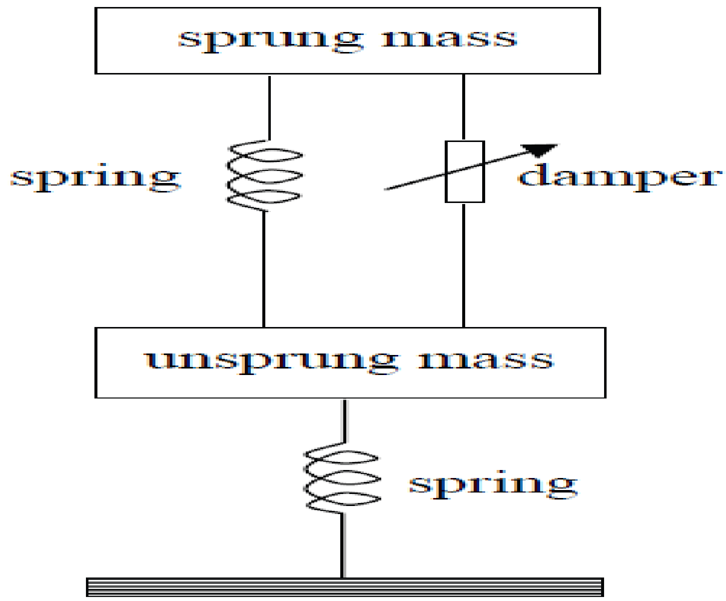


Figure 5.7 Semi-active suspension systems

5.4.1.3: Active Suspension System / Active Vibration Control

Active suspensions differ from the conventional passive suspensions in their ability to inject energy into the system, as well as store and dissipate it.

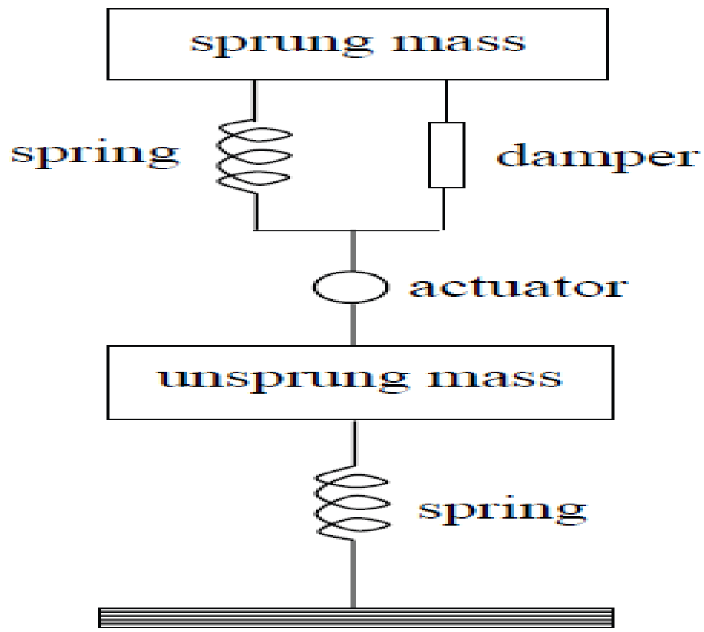


Figure5.8: A low bandwidth or soft active suspension system

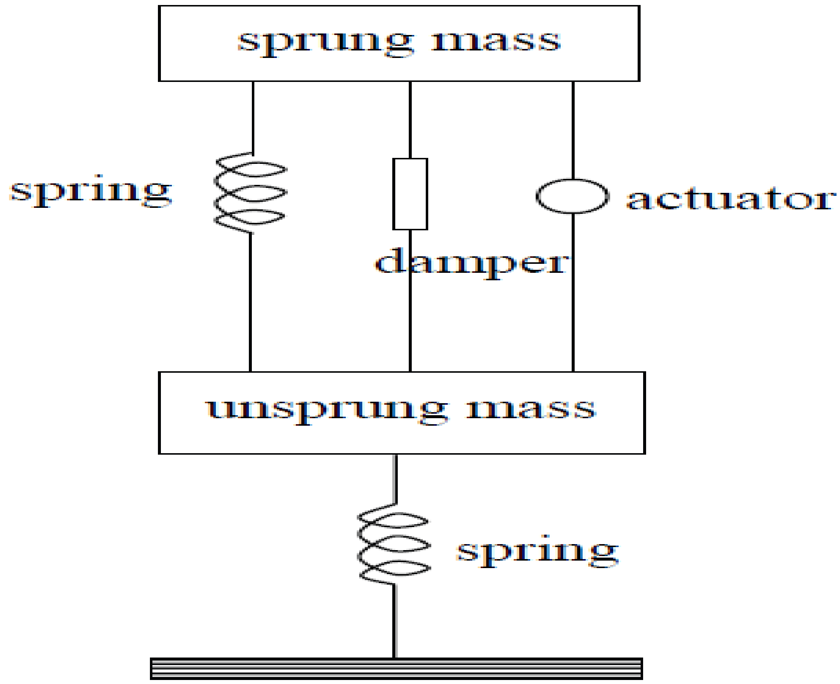


Figure 5.9: A high bandwidth or stiff active suspension system

An active vibration control is a method that relies on the use of an external power source called actuator (e.g. a hydraulic piston, a piezoelectric device or an electric motor). The actuator will provide a force or displacement to the system based on the measurement of the response of the system using feedback control system.

5.5: Active Force Control (AFC)

Hewitt first introduced Active Force Control (AFC) strategy to control a dynamic system in the late 70s [8]. It has been shown that by using AFC, the system remains stable, robust and effective even in the presence of known/unknown disturbances, uncertainties and varied operating conditions.

The essence of the AFC strategy is to obtain the estimated disturbances force, F* via the measurement of mass acceleration, a and actuator force, Fa together with an appropriate estimation of the estimated mass, M* as described in the following equation.

$$F^* = Fa - M^*a \quad (11)$$

It is obvious that Equation (11) is very simple and is expected to be computationally light; this is in fact a very attractive option for real-time or on-line implementation.

Figure 5.10 illustrates a schematic of an AFC scheme applied to a dynamic system. The physical quantities, which need to be measured directly from the system, are the actuating force, Fa and the acceleration, at which can be obtained using some sensing elements. Then the estimated mass of the system, M* with the presence of the disturbances (e.g. vibration, friction and changes in the operating conditions) that partially contribute to the acceleration should be approximated appropriately. This can be achieved by using simple crude approximation (CA) method or artificial intelligent (AI) methods.

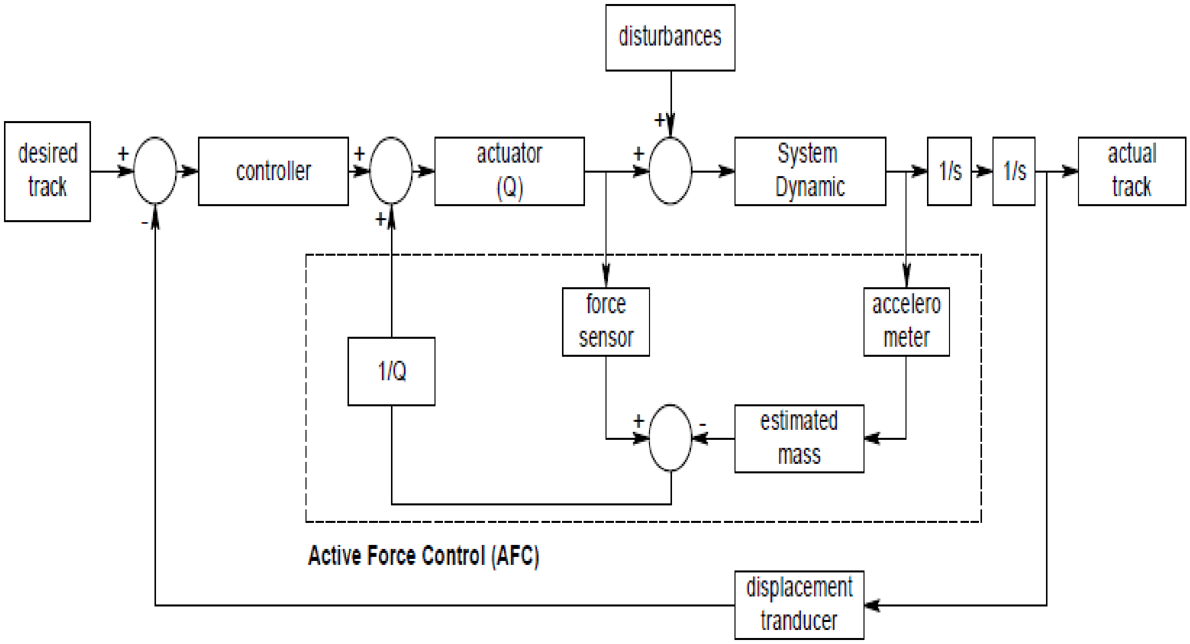


Fig. 5.10: Schematic Diagram of an AFC Strategy

5.6: Friction-induced Vibration

A historical review of structural and mechanical systems with friction is given by Feeny at all. Their paper starts from the first human experiences in fire-making and early inventions of the ancient cultures to the early works of Leonardo da Vinci, and expands to the modern-day scientific advances in friction utilization and prevention.

An essential part of any study on the behavior of a dynamical system with friction is to appropriately account for the friction effect using a sufficiently accurate friction model. There are numerous works found in the literature on the various friction models for simulation and analysis of dynamical systems

• Models that are based on the micro-mechanical interaction between rough surfaces and aim to explain the friction force.

• Models that incorporate various time or system dependent parameters to reproduce the effect of friction in a dynamical system.

Depending on the specific problem being investigated, appropriate friction model should be chosen to reflect the relevant features of the physical system. The simplest approximate friction model may be given by

$$F_f = \mu(v)N \, \mathrm{sgn}(v) \qquad (12)$$

Where F_f is the friction force, v is the relative sliding velocity, $\mu(v)$ is the velocity-dependent coefficient of friction, and N is the normal force pressing the two sliding surfaces together. This model is extensively used in the study of friction-induced vibration.

When some form of lubrication is present between the sliding bodies, the variations of friction with velocity is typically explained by the Stribeck curve. As shown in Figure 5.11, four different regimes are identified in this model.

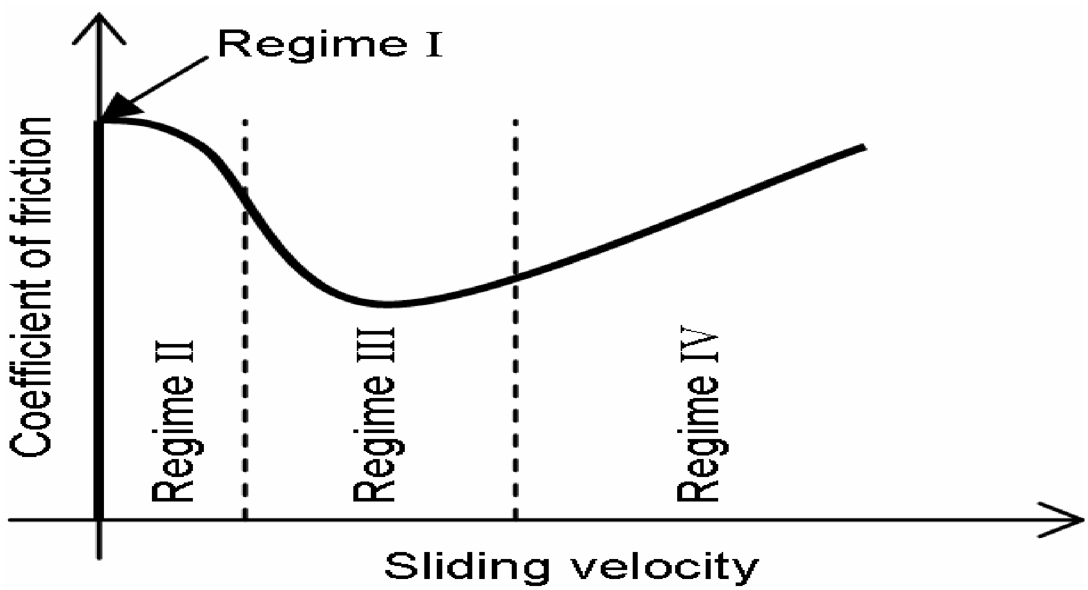


Figure 5.11: Stribeck curve

The first regime is the static friction where lubricant does not prevent the contact of the asperities of the two surfaces and friction acts similar to the no lubricant situation. In the second regime, the sliding velocity is not enough to build a fluid film between the surfaces and lubrication has insignificant effect. In the third regime with the increase of velocity, lubricant enters the load-bearing region, which results in partial lubrication. In this regime, increasing the sliding velocity decreases friction. Finally in the fourth regime, the solid-to-solid contact is eliminated and the load is fully supported by the fluid. In this regime, the friction force is the result of the shear resistance in the fluid and increases linearly with velocity.

Different models have been proposed to reproduce this type of velocity-dependent friction. The important feature of these models is the existence of a region of negative slope in the friction-velocity curve, which may lead to self-excited vibrations. This type of instability is discussed in below.

Wherever sliding motion exists in machines and mechanisms, friction-induced vibration may occur, and when it does, it can have severe effects on the function of the system. Excessive noise, diminished accuracy, and reduced life are some of the adverse consequences of friction induced vibration.

Numerous researchers have studied self-excited vibration phenomena in variety of frictional mechanisms. Possibly the closest mechanism to a lead screw driver in terms of dynamics and friction-induced instabilities, is a disk brake. Fortunately, there are innumerable publications found in the literature that are dedicated to various aspects of disk brake noise and vibrations. Major self-excited vibration mechanisms in the systems with friction relevant to the lead screw drivers can be categorized into three types.

- Decreasing friction force with relative velocity or negative damping
- Kinematic constraint instability
- Mode coupling

5.6.1: Negative Damping

The negative slope in the friction/sliding velocity curve or the difference between static and kinematic coefficients of friction can lead to the so-called stick-slip vibrations. In most instances, researchers adopted the well-known mass-on-a-conveyer model to study the stick-slip vibration. This simple model is shown in Figure 5.12. Here, for simplicity, the coefficient of friction is considered to decrease linearly with relative velocity.

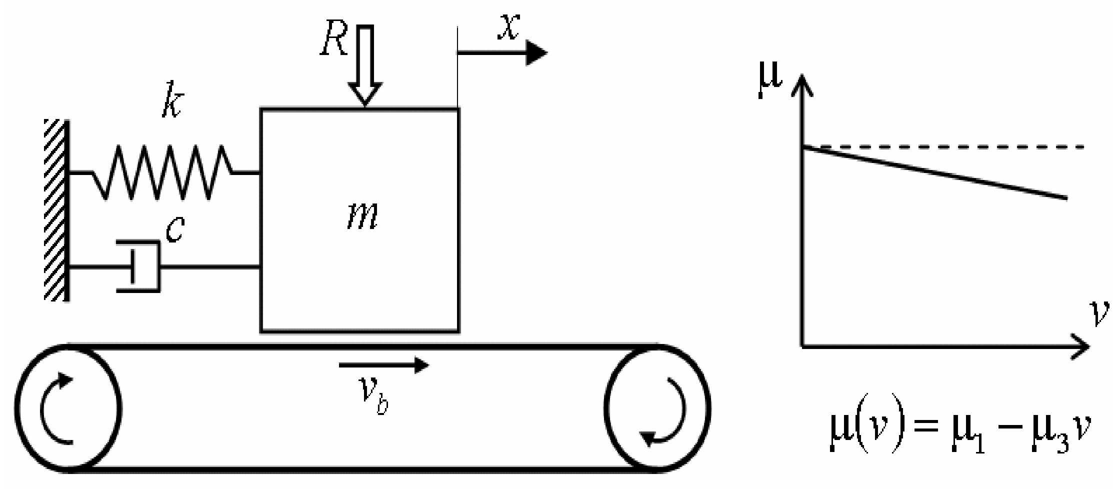


Figure 5.12: 1-DOF mass-on-a-conveyer model

The equation of motion for this model can be written as

$$m\ddot{x} + c\dot{x} + kx = N\mu(v_b - \dot{x})sgn(v_b - \dot{x}) \quad (13)$$

Where N > 0 is the normal force between the mass and the conveyer and V_b> 0 is the conveyer's constant velocity. Transferring the steady-sliding state to the origin gives

$$m\ddot{y} + c\dot{y} + ky = N[\mu(v_b - \dot{y})sgn(v_b - \dot{y}) - \mu(v_b)] \quad (14)$$

$$Where\ \ y = x - x_0\ \ and\ \ x_0 = \frac{N}{k}(\mu_1 - \mu_3 v_b)$$

Considering small perturbations around the steady-sliding fixed point where $(v_b - \dot{y}) > 0$

&, linearized equation of motion is found from (14) as

$$m\ddot{y} + (c - N\mu_3)\dot{y} + ky = 0 \quad (15)$$

It is obvious that when c < Nμ_3 the system (15) is unstable. In this situation, the vibration amplitude grows until it reaches the stick-slip boundary, i.e. $(v_b - \dot{y}) = 0$

Using an exponentially decreasing model for the coefficient of friction, Hetzler et al used the method of averaging to study the steady-state solutions of a system similar to the one shown in Figure 5.12. They showed that as damping is increased, the unstable steady-sliding fixed point goes through a subcritical Hopf-bifurcation, resulting in an unstable limit cycle that defines the region of attraction of the stable fixed point. Thomsen and Fidlin also used averaging techniques to derive approximate expressions for the amplitude of stick-slip and pure-slip (when no sticking occurs) vibrations in a model similar to Figure 5.12. They used a third-order polynomial to describe the velocity-dependent coefficient of friction. In cases where the coefficient of friction is a nonlinear function of sliding velocity the presence of one or more sections of negative slope in the friction-sliding velocity curve can lead to negative damping and self-excited vibration. In this type of friction instability, no sticking occurs between the two rubbing surfaces.

5.6.2: Kinematic constraint instability

When friction is present, the constraint equation used to model kinematic pairs in dynamical systems can lead to instabilities even when the coefficient of friction is assumed to be constant. In the context of constrained multi-body system dynamics with friction, the same mechanism is the cause of "jamming" or "wedging". In the study of disc brake systems, this type of instability is sometimes referred to as "sprag-slip" vibration. This type of instability is usually characterized by violation of the solution existence or uniqueness conditions of the system's equations of motion

The simplest example to demonstrate the kinematic instability is shown in Figure 5.13. In the model shown, a massless rigid rod pivoted at point O is contacting a rigid moving plane. A force L is pressing the free end of the rod against the moving plane. The normal and friction force applied to the rod are given by N and $F_f = N\mu_k$ where μ_k is the constant kinetic coefficient of friction. It can be shown that at equilibrium.

$$N = \frac{L}{1-\mu_k tan\theta} \tag{16}$$

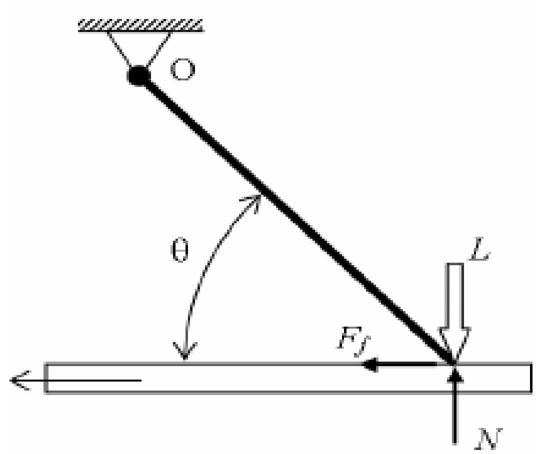


Figure 5.13: Simple model to demonstrate kinematic constraint instability

From equation (16) it is evident that If $\theta \to tan^{-1}\left(\frac{1}{\mu_k}\right)$, $then \quad N \to \infty$ and further motion becomes impossible. In a more realistic setting where some flexibility is assumed, the motion continues by deflection of the parts. After sufficient deformation of the contacting bodies, slippage occurs which allows the bodies to assume their original configuration and the cycle continues. This situation is known as the sprag-slip limit cycle.

5.6.3: Mode Coupling

In the context of the linear dynamical systems, the effects of non-conservative forces on stability are well understood. Consider the equations of motion of a second order undamped multi-DOF linear autonomous system as

$$m\ddot{q} + \left(k + s(\eta)\right)q = 0 \tag{17}$$

Where **q** is the vector of generalized coordinates, **m** is a positive-definite symmetric inertia matrix, **k** if the symmetric stiffness matrix, and $\boldsymbol{s(\eta)}$ is an asymmetric matrix originating from the non conservative forces, and h is a parameter of interest. The natural frequencies of this system are found from the solutions of the characteristic equation given by

$$\Delta(\omega^2, s\eta) = det(k + s(\eta) - \omega^2 m) \tag{18}$$

Assuming the initial system (h = h0) is stable, the stability may be lost by divergence or flutter as the parameter h is varied. The divergence instability boundary (d h = h) is found from

$$det(k + s(\eta) = 0 \tag{19}$$

At this critical value, one of the roots of equation (18) vanishes. The flutter instability boundary can be found by setting

$$\frac{\partial \Delta(\omega^2,\eta)}{\partial \omega^2} \tag{20}$$

Where D is given by (18) the flutter boundary (f h = h) is characterized by the coalescence of two of the system natural frequencies. By further variation of the parameter beyond its critical value, two roots become complex conjugate. In cases where **S**(h) is skew-symmetric, system equation (17) can only become unstable through flutter instability as divergence is not possible. The addition of velocity-dependent forces to this system yields

$$m\ddot{q} + (c + g)\dot{q} + (k + s)q = 0 \qquad (21)$$

Where **c** is positive semi-definite matrix and $g = -\, g^T$ defines the gyroscopic forces. It has been shown that the addition of damping can have a complex effect on the stability of the system and it may even destabilize the otherwise stable.

Chapter 6

Case Study 1: Vibration Control using 1- DOF

6.1: Dynamic System of a Mass-Spring-Damper System

Consider a spring-mass system having viscous damping, excited by a constant force function F_0 as shown in figure 6.1. At any instant, when the mass is displaced from the mean position through a distance x in the downward direction (positive direction of x), the external forces acting on the system are shown below figure:

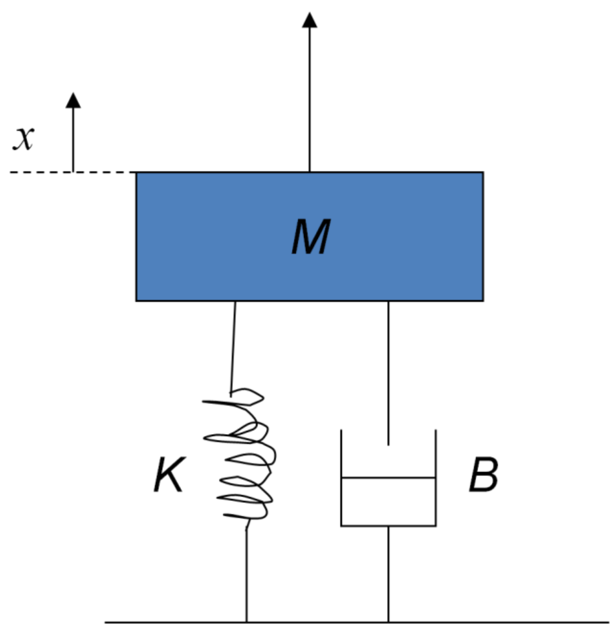


Figure 6.1: Forced vibration with constant excitation

These forces are

Kx in upward direction

$C\dot{x}$ in the upward direction

F_0 and F_0 sin(wt) in down word direction

The Mathematical model of the system is describe by

$$\ddot{x} = \frac{1}{M}(-B\dot{x} - Kx + F(t))$$

$$\ddot{x} = \frac{1}{2}(-2\dot{x} - 2x + F(t)) \tag{22}$$

Use the same mass-spring-damper system example and simulate the response using transfer function approach

$$\frac{X(S)}{F(S)} = \frac{1}{2S^2+2S+2} \tag{23}$$

The transfer function of the equation (assume all initial conditions =0)

6.2: Model parameters

- Body mass, m=2 kg
- Spring stiffness, k=2 N/m
- Damping coefficient C=2 N/m

Actuator

- Actuator gain, Q=0.5

Reference Value

- Reference input = 100N (i.e. vibration due to 100 Newton force)

6.3: Advantage of Subsystem

- Reduce the number of blocks display on the main window (i.e. simplify the model)
- Group related blocks together (i.e. More organized)
- Can create a hierarchical block diagram (i.e. you can create subsystems within a subsystem)
- Easy to check for mistakes and to explore different parameters

6.4: Passive Model for Brake System using Single Degree of Freedom

In this thesis work, the namely the step disturbance are deliberately introduced to the disc brake system to evaluate the robustness of the system. The Simulink diagram of passive disc brake system model is shown in figure 6.2. The schematic block diagram is constructed from figure 6.1.

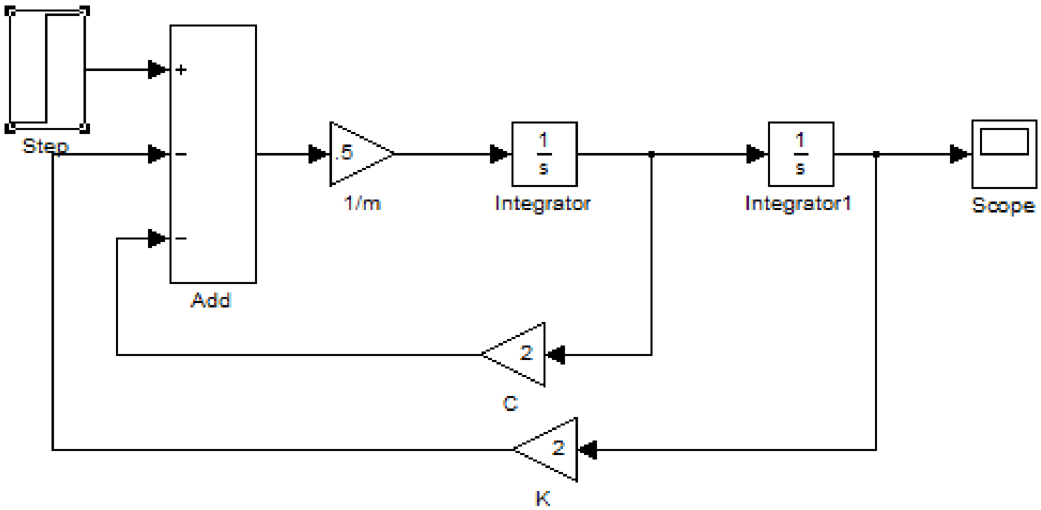


Figure 6.2: Simulink model for brake system using Single Degree of freedom.

6.4.1: Active Brake System with PID

In order to have an active disc brake system, an actuator force for compensating the disturbance force is required, and the actuator force is controlled by a PID controller which typically involves a negative feedback loop. Hence, there are two inputs to the dynamic disc brake system which is the step disturbance and actuator force input. Figure 6.3 shows the active disc brake system.

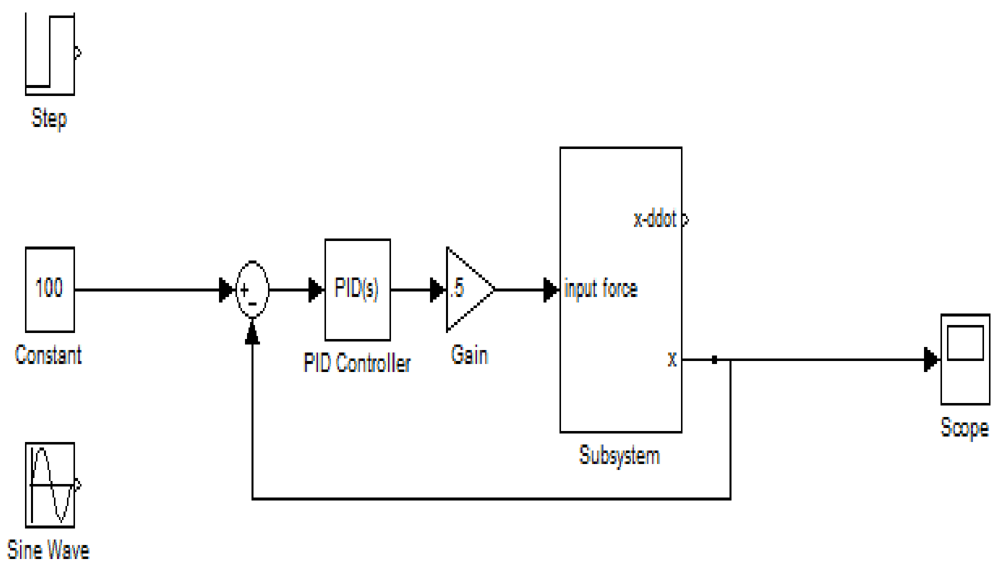


Figure 6.3: PID active brake control scheme

6.4.2: ACTIVE BRAKE MODEL WITH PID+AFC

To achieve better overall performance of the brake system, AFC is 'added' to PID controller. The AFC Simulink block includes the estimated mass parameter 1/Q and the percentage of AFC gain. The input to the AFC control is the mass acceleration and the output is summed with the PID controller output and then multiplied with the actuator gain which finally generated the actuator force. In order to get the effective results using this method. It is required to have a suitable mass estimation combined with the best tuning of the PID controller gains. Also a memory block is used to eliminate the algebraic loop and variable problem. Figure 6.4 shows an AFC scheme that has a step input as its disturbance.

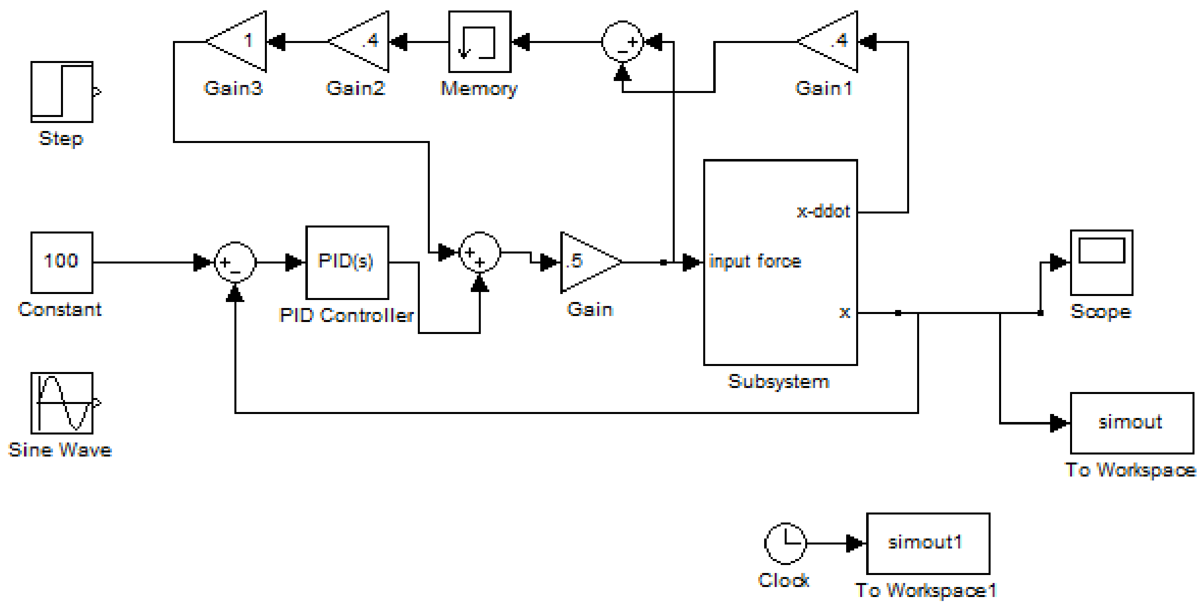


Figure 6.4: PID+AFC model for brake control

6.5: Results

In this chapter, there are three type of input source are used for the testing the stability analysis of braking system.

- With constant forced response (use constant of 100 N)
- Step forced response (use step value of 100 N [initial value of 0, final value of 100] and step time of 20 seconds)
- Sinusoidal forced response (use amplitude of 100 N, frequency of 3)

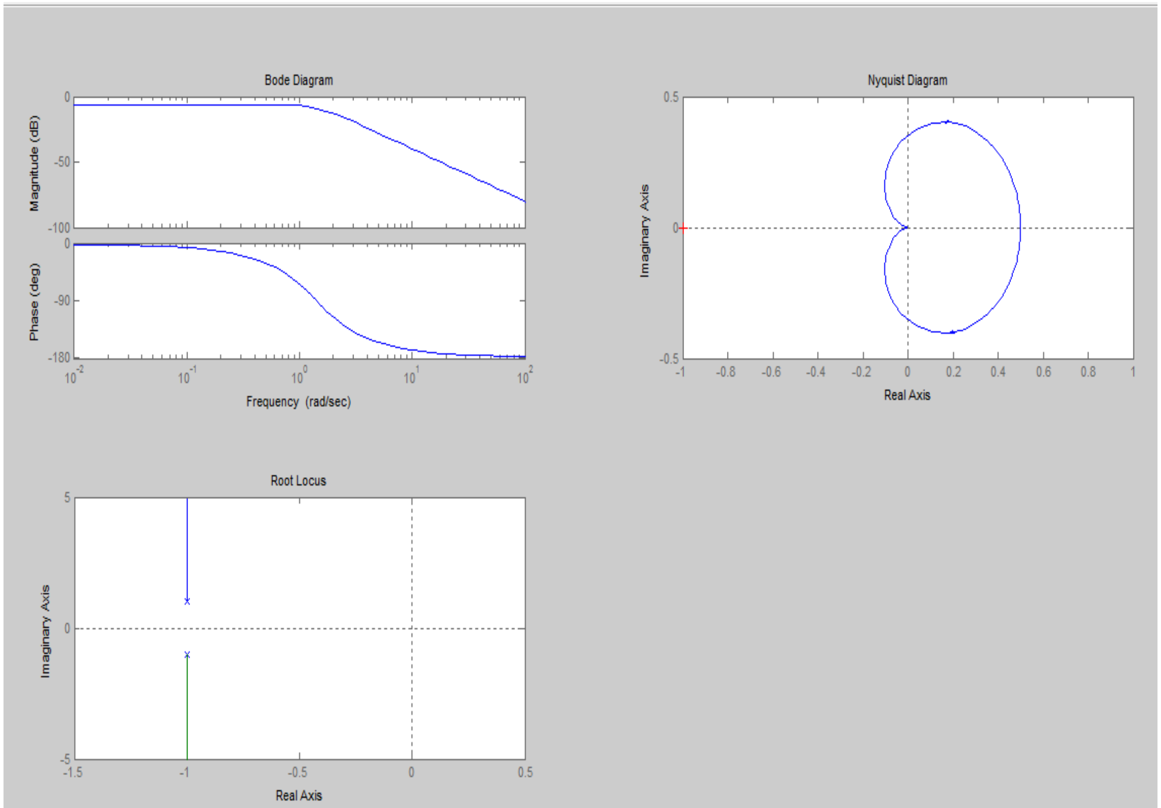


Figure 6.5: Stability Response for brake system without controller

The Bode plot method gives a graphical procedure for determining the stability of a control system based on sinusoidal frequency response.

The Bode plot give the variation of the gain (in db) and phase (in degree) of a transfer function, with change in frequency of input. These are log-log plots. Due log scale on x-axis, Bode plots cover a wide range of frequencies. Due to log scale on y-axis, Bode plots cover the wide range of gain. Thus, Bode plots present the behaviour over wide range.

Bode plot fascinate in the sense that gain and phase contribution of individual factor of transfer function are clearly indicated. But Bode plot are in generally use full for minimum phase transfer function. It also corresponds to any positive half of imaginary axis of s-plane and therefore gain margin and phase margin remains significant in this plot. Bode plot are also called as asymptotic plot / corner plot.

The Bode plot is plotted using PID controller .It has infinite gain margin and 60^0 phase margin. The PID controlled braking system is stable for closed loop.

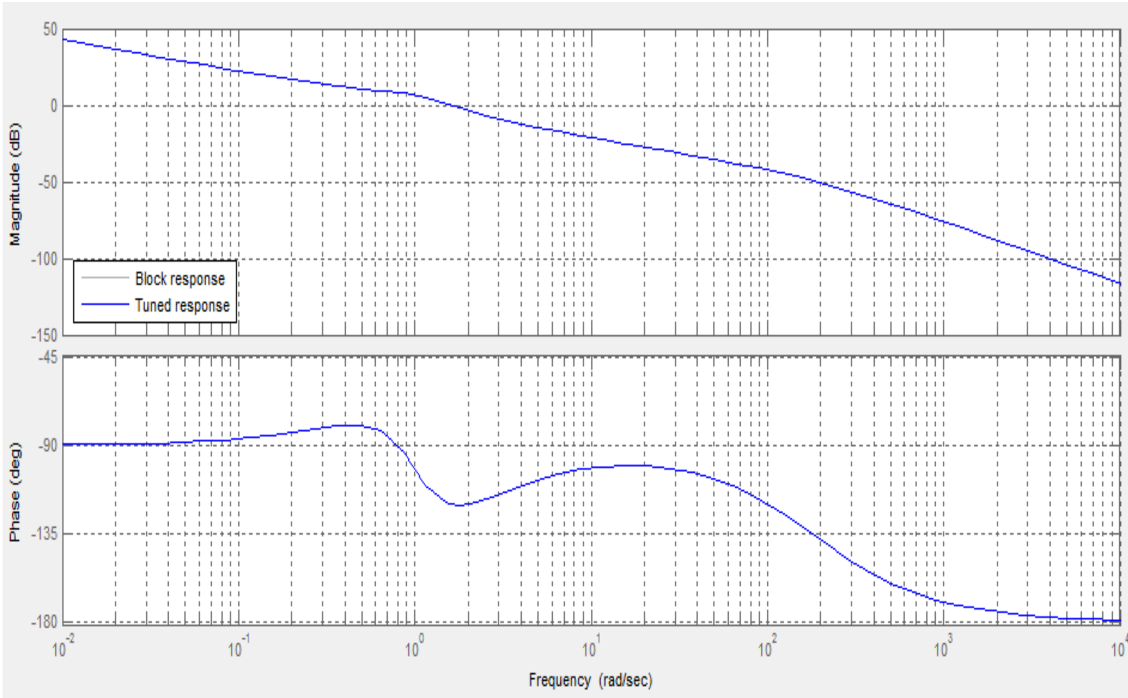


Figure 6.6: Bode plot using PID controller

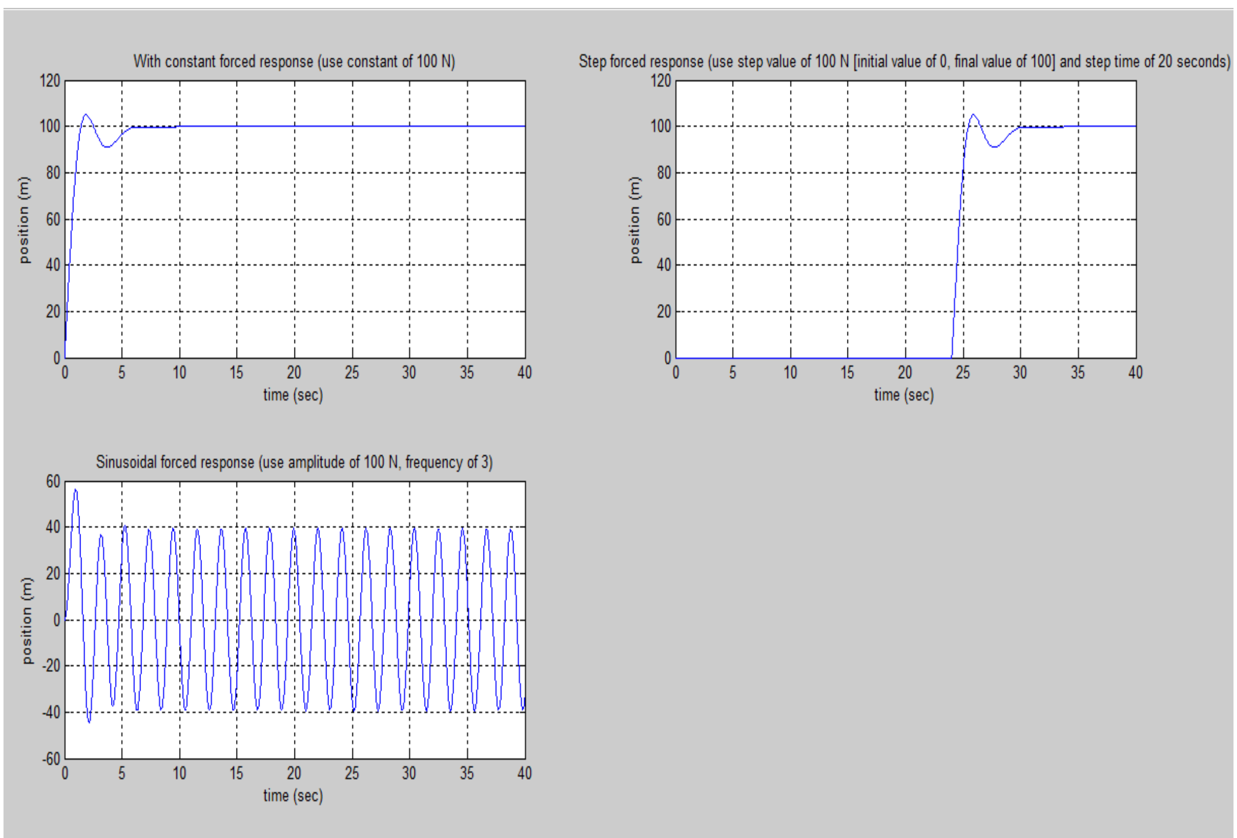


Figure 6.7: Response of brake system with PID controller

An alternative method for designing compensators is to use the Nichols plot, which combines gain and phase information in a single plot. The combination of the two is useful when you are designing to gain and phase margin specifications. It is superposing plot on Nichols chart to study close loop frequency response. Each and every time it repeats its value after interval of 360^0.

The intersection of open-loop gain phase plot with Nichols chart gives the gain and the phase of the closed-loop transfer function. At the point of intersection frequency ω is read from the gain phase plot.

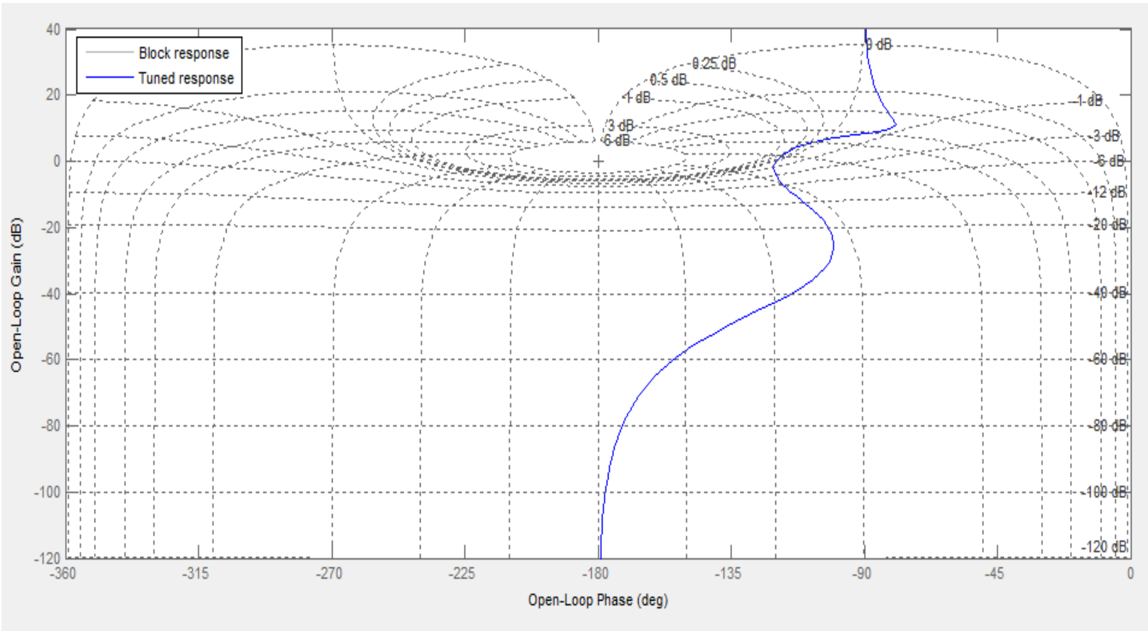


Figure 6.8: Open loop Nichols charts using PID Controller

The Bode plot is plotted using PID + AFC (Active force control) controller .It has infinite gain margin and 60^0 phase margin. The PID+ AFC controlled braking system is stable for closed loop.

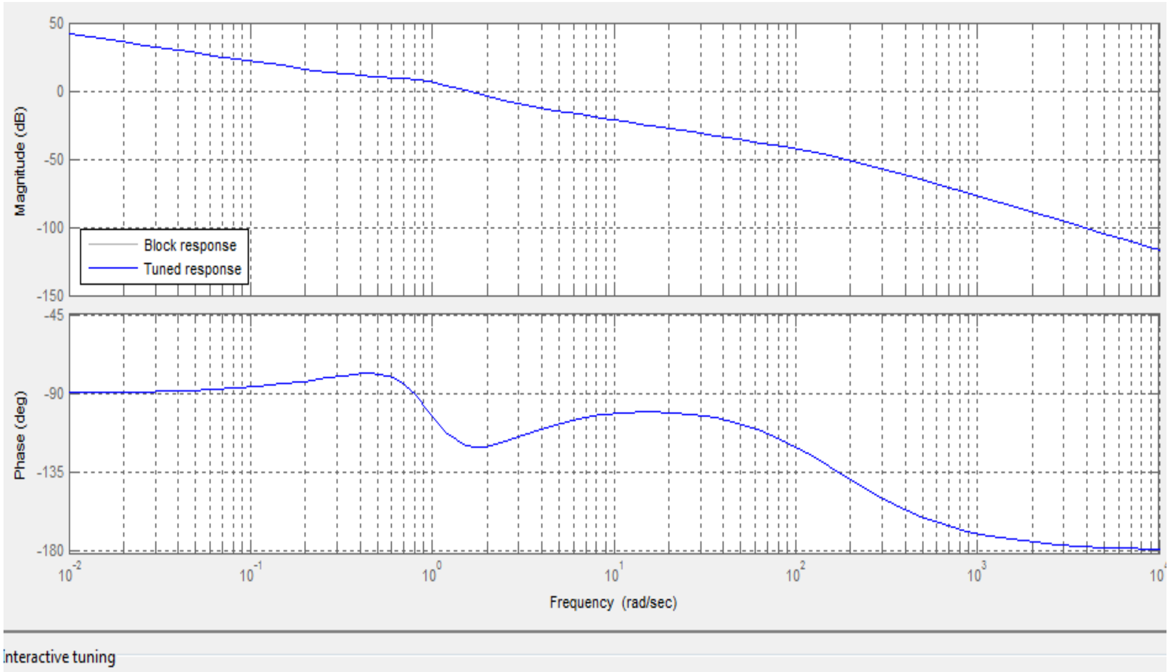


Figure 6.9: Bode plot using PID+AFC Controller

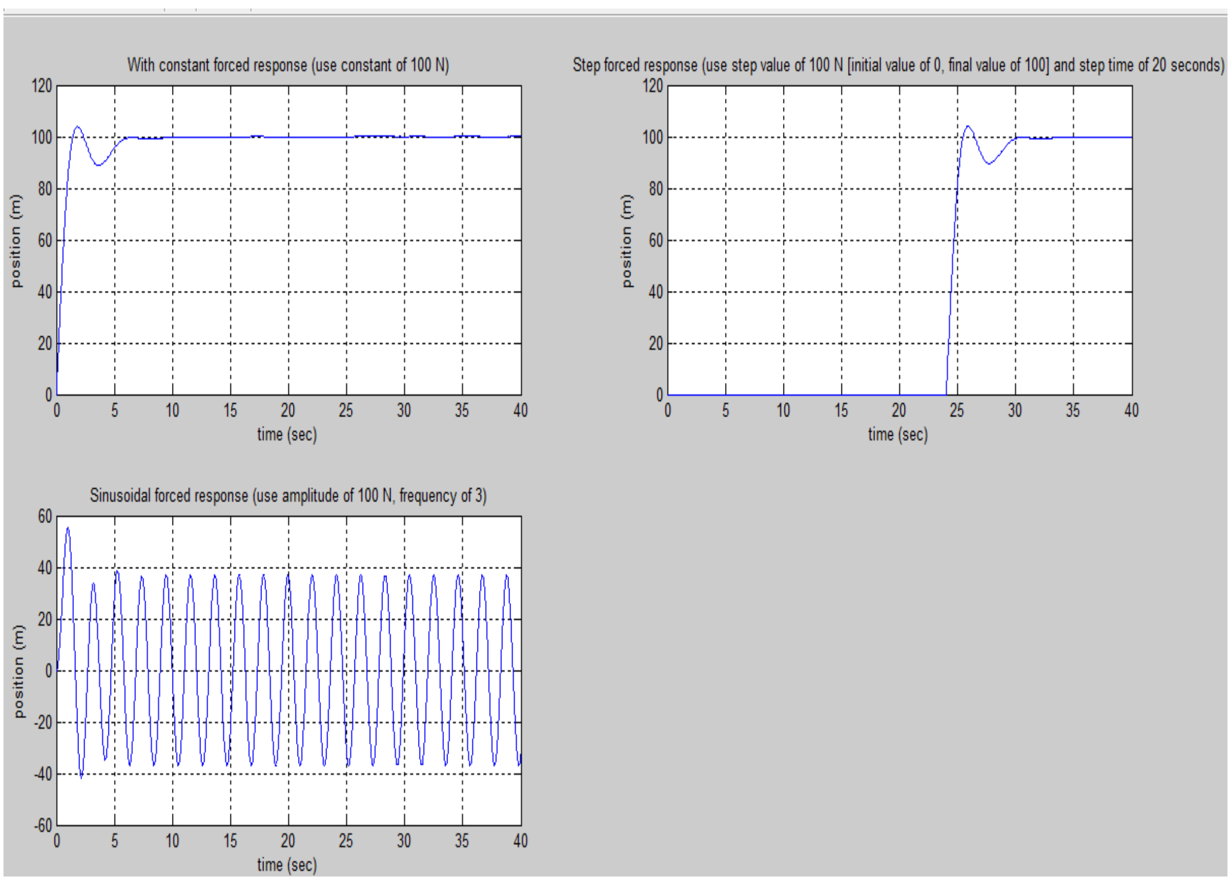


Figure 6.10: Response of brake system with PID+AFC controller

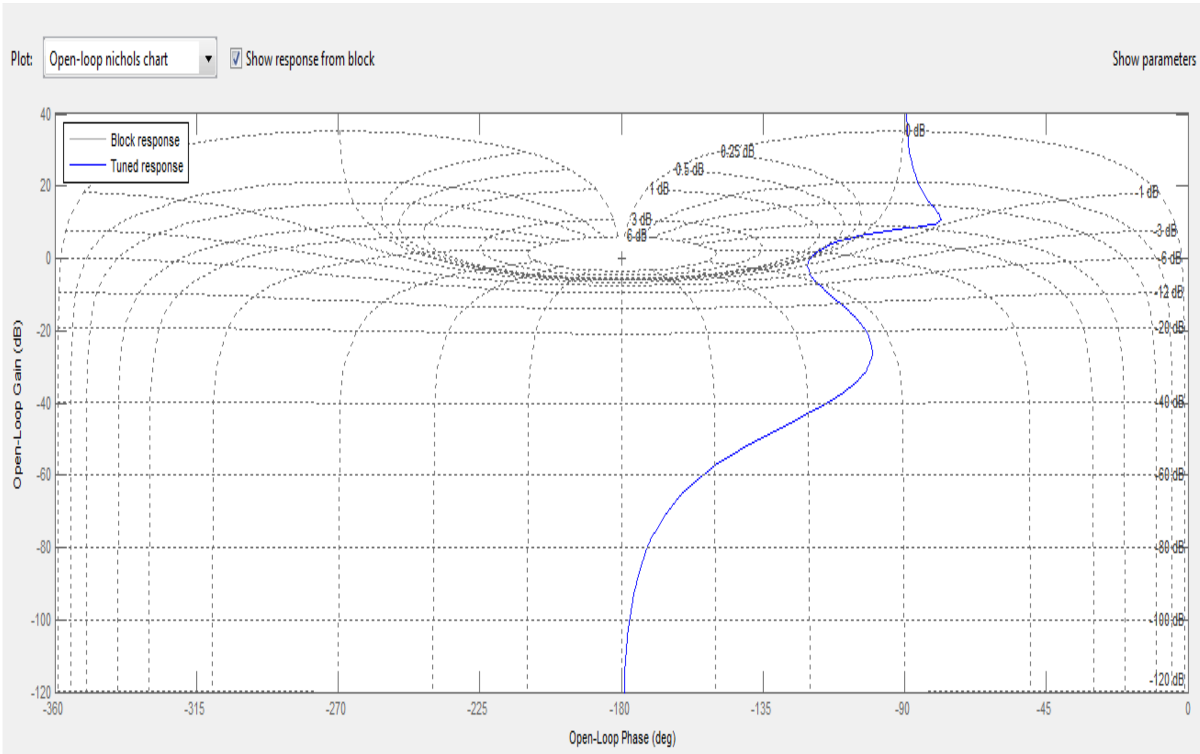


Figure 6.11: Open loop Nichols chart using PID+AFC

6.6: Discussion

Table 6.1: Comparison of different control scheme in single degree of freedom

PRAMETERS	PID	PID+AFC
Rise Time	1.04	1.09
Settling Time	5.37	5.59
Overshoot (%)	4.96	4.4
Peak Time	1.05	1.04
Gain Margin (db@rad/sec.)	INF@INF	INF@INF
Phase margin (deg@ rad/sec.)	60@1.59	60@1.54
Closed Loop Stability	STABLE	STABLE
P	8.549	6.6046
I	5.2682	3.9304
D	3.4207	2.7366

6.7: Conclusion

A novel AFC-based scheme has been proposed to suppress the vibration and noise (squeal) emanating from a disc brake system. Based on the simulation results, it is obvious that when a pure PID controller is applied to the brake system, vibration and noise are in fact reduced but still a noticeable amount of them remain. However, upon applying the AFC+PID the vibration and noise (squeal) must significantly reduce. If it is achieved then the proposed strategy is robust and effective in countering the undesirable effects.

Future works may include using fuzzy and neuro-fuzzy algorithm with AFC model and the study of the system behaviour considering other different operating and loading setting.

Chapter 7

Case Study 2: Two degree of freedom

The brake system of an automobile typically consists of the contact metallic solids rubbing against each other, which frequently generates undesirable noise and vibrations. Thus, noise generation and suppression have become an important factor to be considered in the design and manufacture of brake components.

7.1: The Brake Model

A disc brake system assuming a two degree of freedom model based on the Hoffman model. The model consists of a conveyor belt with constant velocity v_B that is pushed with a constant normal force F_N against a block modeled as a block of mass m. Fig. 7.1 shows that the model is just a single-point mass sliding over a conveyor belt and there are two linear springs k_1 and k_2 parallel and normal to the belt surface with the latter regarded as the physical contact stiffness between the objects in relative sliding motion. In addition, there is another linear spring k mounted at oblique angle of 45° constituting the off-diagonal terms in the model's stiffness matrix. For the friction component, a coulomb model is assumed such that $F_T = \mu F_N$, where μ is the coefficient of kinetic friction usually taken to be constant. F_N is a normal force and since the normal force at the friction interface is linearly related to the vertical displacement x_2 of the mass then the resulting friction will become $FF = \mu k_2 x_2$.

Assuming that the mass of the conveyor belt system is larger than the mass block, it implies that the vibration of the belt does not show any changes due to its inertia.

The matrix form of the equation of motion can be expressed as

$$\begin{pmatrix} m & 0 \\ 0 & m \end{pmatrix} + \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{pmatrix} + \begin{pmatrix} c_1 & 0 \\ 0 & c_2 \end{pmatrix} \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{pmatrix} + \begin{pmatrix} k_1 + k_1/2 & -k/2 \\ k/2 & k_2 + k_2/2 \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \end{pmatrix} = \begin{pmatrix} -\mu k_2 x_2 \\ N \end{pmatrix} \tag{24}$$

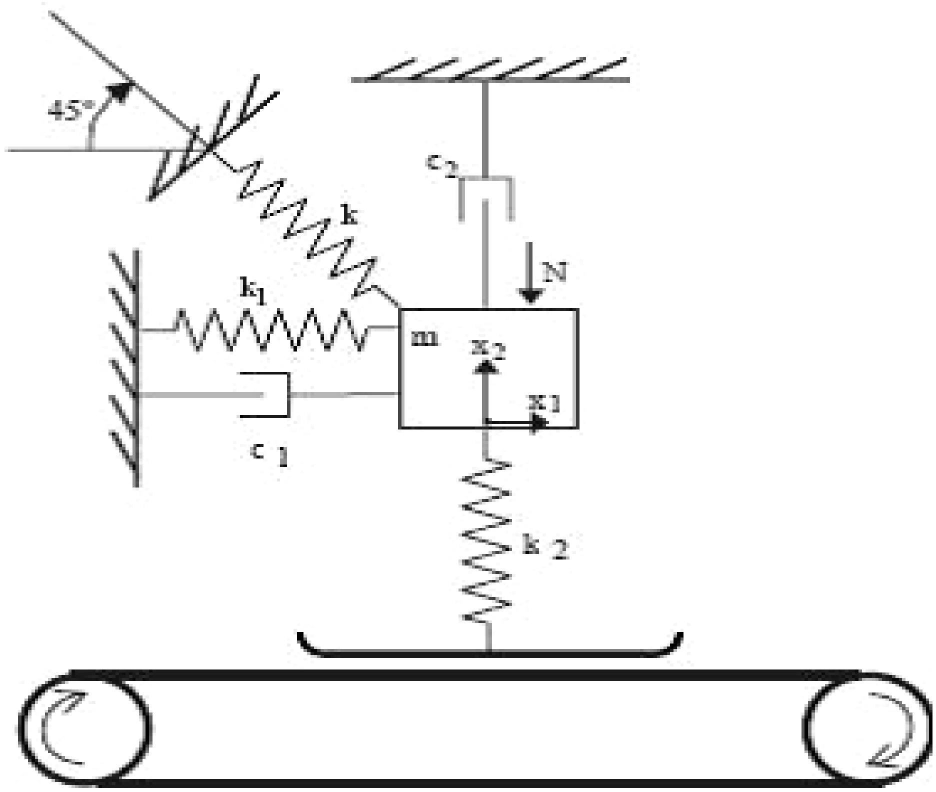


Figure 7.1: Two DOF model of a disc brake system

7.2: Control Strategy

After acquiring the model of the disc brake and its related equation of motion, it is required to control the vibration of the mass with respect to the vertical direction, x_2 considering an actuator that produces a force F in parallel with the preload force N. Thus, will be written as

$$\begin{pmatrix} m & 0 \\ 0 & m \end{pmatrix} + \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{pmatrix} + \begin{pmatrix} c_1 & 0 \\ 0 & c_2 \end{pmatrix} \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{pmatrix} + \begin{pmatrix} k_1 + k_1/2 & -k/2 \\ k/2 & k_2 + k_2/2 \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \end{pmatrix} = \begin{pmatrix} -\mu k_2 x_2 \\ N+F \end{pmatrix} \qquad (25)$$

A robust control strategy is proposed here employing an Active Force Control (AFC) based scheme that is used in conjunction with the conventional PID controller. The PID controller was first tuned with Ziegler-Nichol's method for good performance and later the AFC part was incorporated into the system to provide the compensation of the disturbances. Fig. 7.2 shows the AFC scheme applied to a dynamic translation system (disc brake). AFC scheme is shown to be very effective provided the actuated force and body acceleration are accurately measured and at the same time the estimated mass property approximated. The essential AFC equation can be related to the computation of the estimated disturbance F_d as follows

$$F_d = F - M'. a \tag{26}$$

Where F is the measured actuating force, M' is the estimated mass and a is the measured linear acceleration. This parameter is then fed back through a suitable inverse transfer function of the actuator to be summed up with the PID control signal. The theoretical analysis including stability of the proposed AFC method has been sufficiently described. A number of methods to estimate the mass have been proposed in previous studies such as through the use of artificial intelligence (AI) and crude approximation techniques [7-9]. In this study, the use of crude approximation method to approximate the estimated mass is deemed sufficient. The main challenge of the AFC method is to acquire appropriate estimation of the mass needed to compute the disturbance F_d in the feedback loop. A conventional PID that is used with the AFC scheme can be typically represented by the following equation

$$G_c(s) = K_p + K_i /S + K_d S \tag{27}$$

Where Kp, Ki and Kd are the proportional, integral and derivative gains respectively.

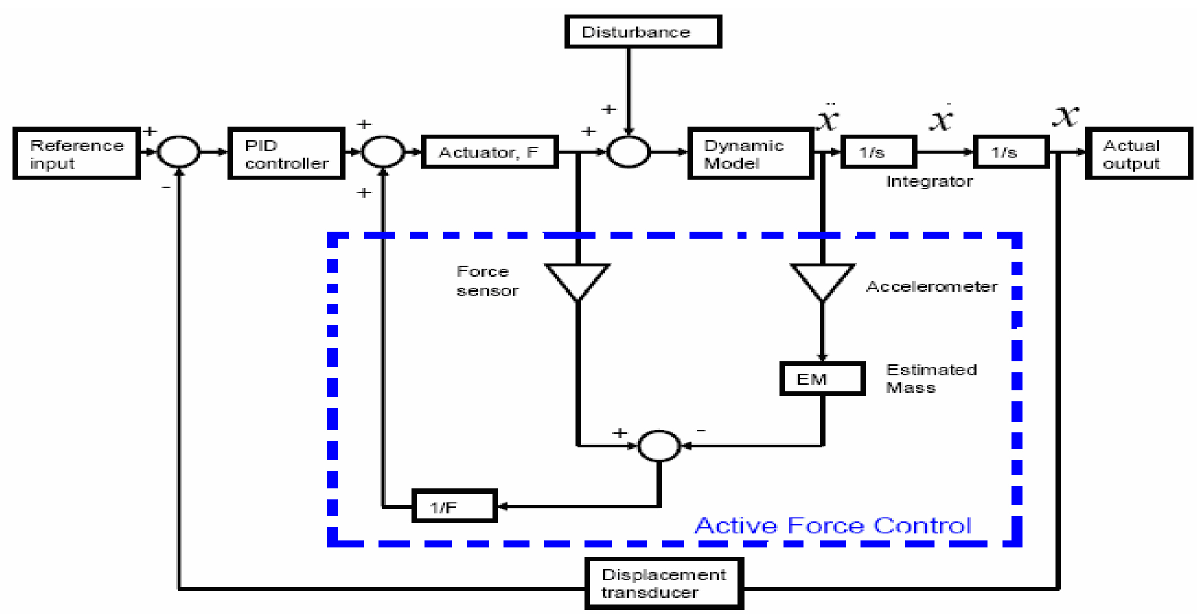


Figure 7.2 Schematic diagram of AFC strategy

7.3: Simulation

The actuator is assumed to be a linear type with a suitable constant gain. It provides the necessary external energy to suppress vibration in the model. The parameters used in this study were taken from the previous research. However, some of them need to be modified to suit the application in the simulation. The detailed parameters are as follows:

Minimal disc brake model parameters:

- Body mass, m = 0.7 kg,
- Spring stiffness, k = 10 N/m, k1 = 11 N/m, k2 = 20 N/m
- Damping coefficient, c1 = 0.4 Ns/m, c2 = 0.4 Ns/m
- Friction coefficient, μ = 0.3
- Normal preload, N = 5 N

Actuator

- Actuator gain, Q = 0.5

Reference value

- Reference input = 0.00 m (i.e. no vibration)

Disturbance

- Magnitude of step function = 1 N
- Frequency = 1.5 Hz

7.3.1: Model for Passive Brake System

In this thesis, namely the step disturbances are deliberately introduced to the disc brake system to evaluate the robustness of the system. The Simulink diagram of the passive disc brake system model is shown in Fig.7.3. The schematic block diagram was constructed from.

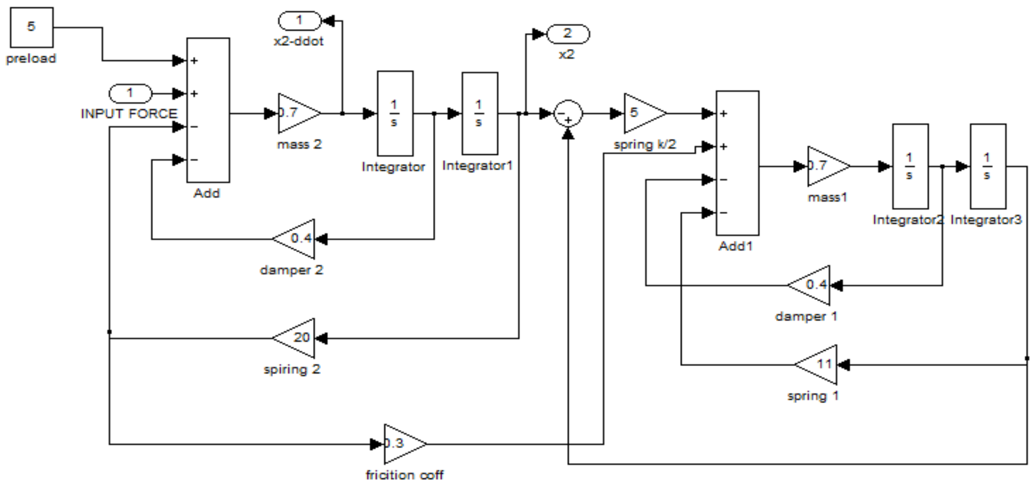


Figure 7.3: Passive brake system

7.3.2: Model for PID

In order to have an active disc brake system, an actuator force for compensating the disturbance force is required, and the actuator force is controlled by a PID controller which typically involves a negative feedback loop. Hence, there are two inputs to the dynamic disc brake system which is the step disturbance and the actuator force input. Figure 4 shows the active disc brake system.

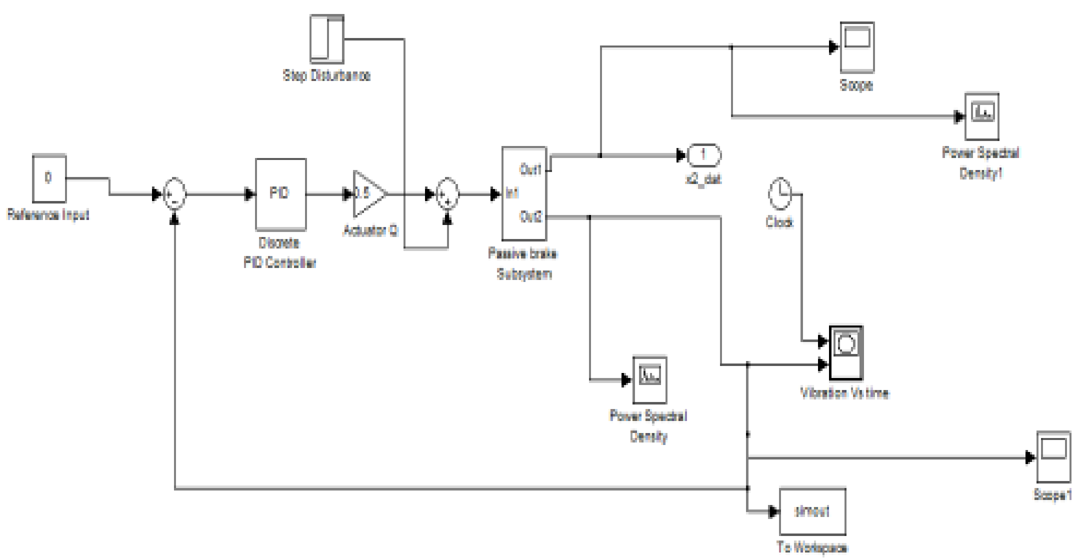


Figure 7.4: PID active brake control scheme

7.3.3: Model for PID+Fuzzy

To achieve better overall performance of the brake system, AFC is ‘added’ to the PID controller, The AFC Simulink blocks includes the estimated mass, parameter 1/Q and the percentage of AFC gain. The input to the AFC control is the mass acceleration and the output is summed with the PID controller output and then multiplied with the actuator gain which finally generated the actuator force. In order to get the effective results using this method, it is required to have a suitable mass estimation combined with the best tuning of the PID controller gains. Also a memory block is used to eliminate the algebraic loops and algebraic variable problems. Figure 5 shows an AFC scheme that has a step input as its disturbance.

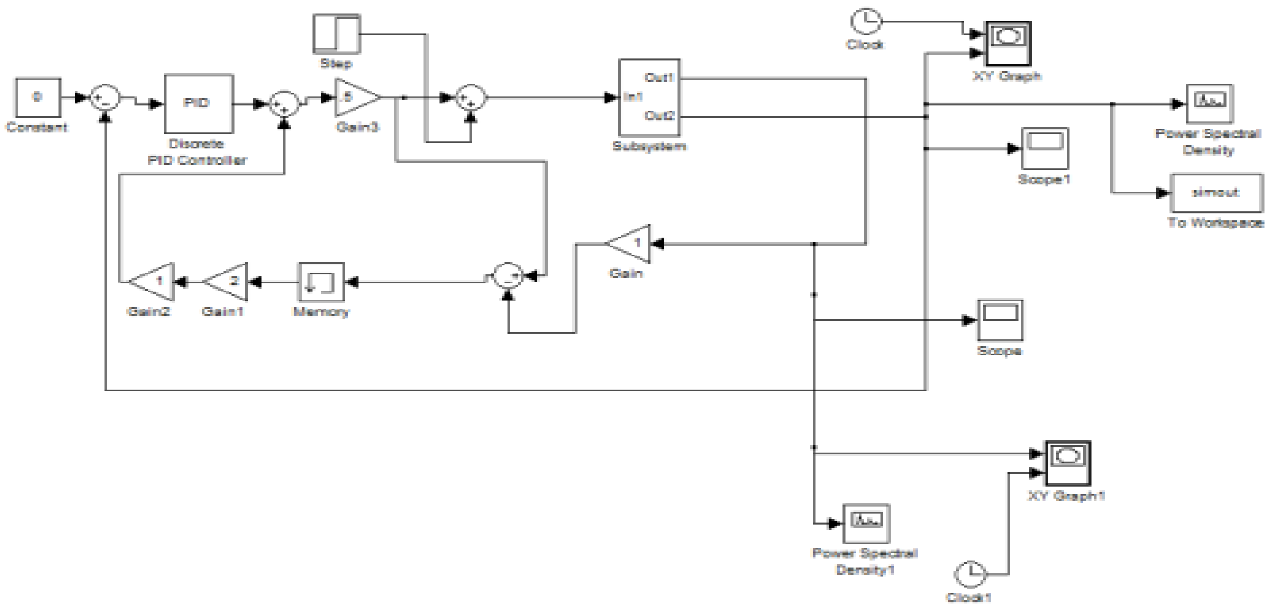


Figure 7.5: PID+ AFC model for brake control

To tune the PID controller, we use the Ziegler-Nichol’s method and the results are tabulated as shown in Table-1

Table 7.1: PID Parameters Tuned Using Ziegler-Nichol’s Method

PID	K_p	K_i	K_d	T_i	T_d
Gain	0.6	0.857	0.105	0.7	0.17

The estimated mass for the AFC loop was obtained by trial and error (crude approximation method) in which a suitable value was easily found to be 1.4 kg, and the percentage of AFC used is 100% implying that the AFC loop employs full AFC implementation.

7.3.4: Model for AFC+Fuzzy

FLC provides an algorithm which can convert the linguistic control strategy based on expert knowledge into an automatic control strategy. Experience shows that the FLC yields results superior to those obtained by conventional control algorithms. In particular, the methodology of the FLC appears very useful when the processes are too complex for analysis by conventional quantitative techniques or when the available sources of information are interpreted qualitatively, inexactly, or uncertainly. Thus fuzzy logic control may be viewed as a step toward a rapprochement between conventional precise mathematical control and human-like decision making.

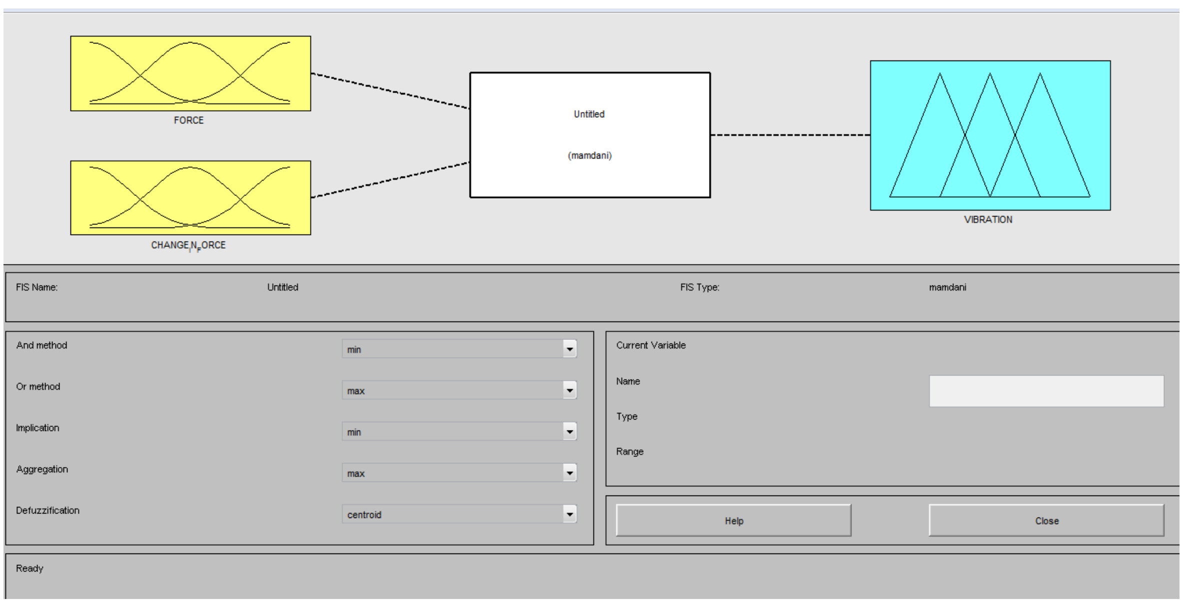


Figure 7.6: Fuzzy inference system

In this Thesis a fuzzy logic controller is developed. The inputs to the FLC are applied force and change in force. The output is change in vibration.

In the FIS system Mamdani type of rule-base model is used. This produces output in fuzzified form. Normal system need to produce precise output which uses a defuzzification process to convert the inferred possibility distribution of an output variable to a representative precise value. In the given fuzzy inference system this work is done using centroid defuzzification principle. In this min implication together with the max aggregation operator is used.

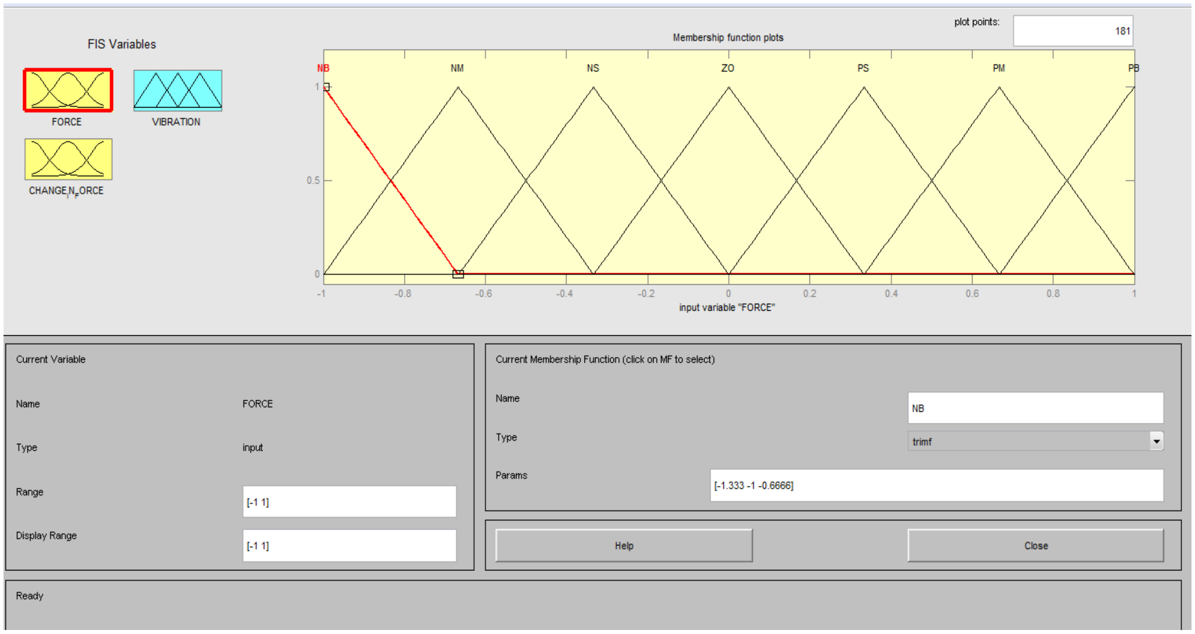


Figure 7.7: Membership function for force input

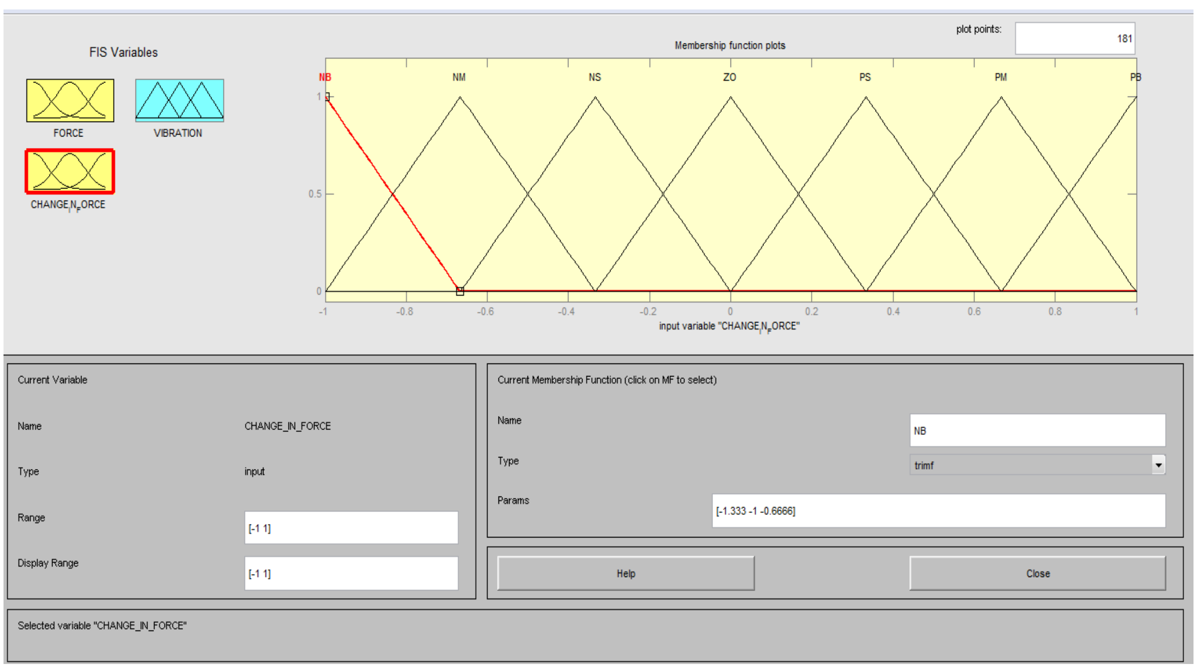


Figure 7.8: Membership function for change in force input

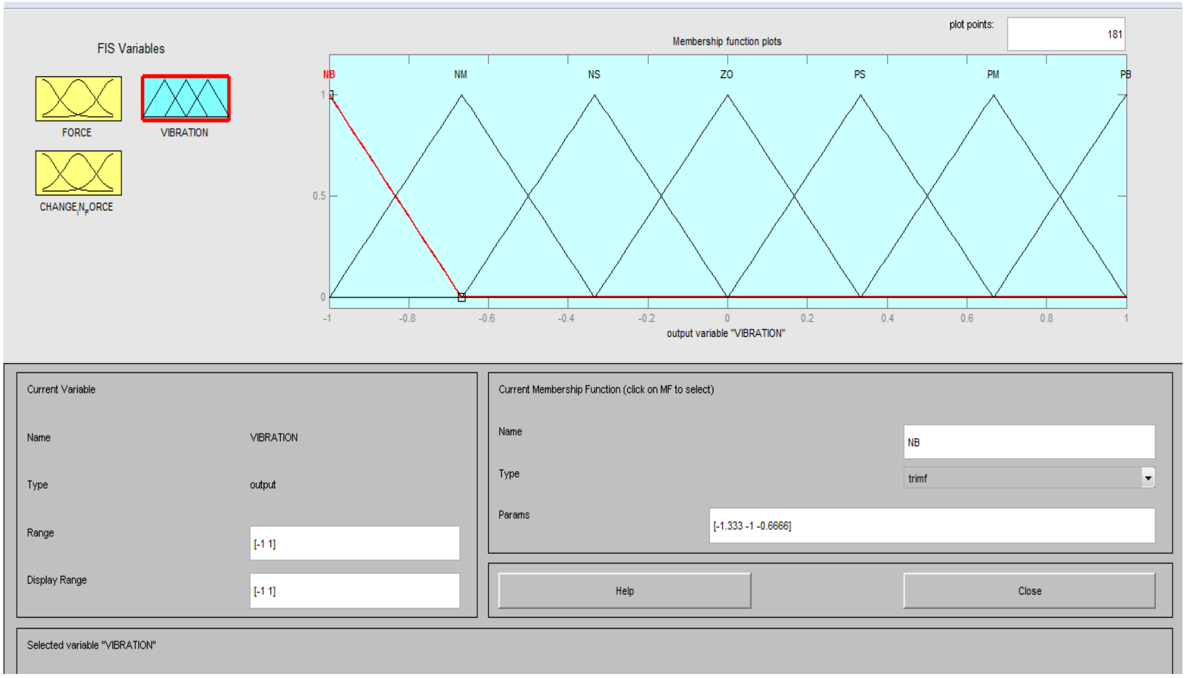


Figure 7.9: Membership function for vibration output

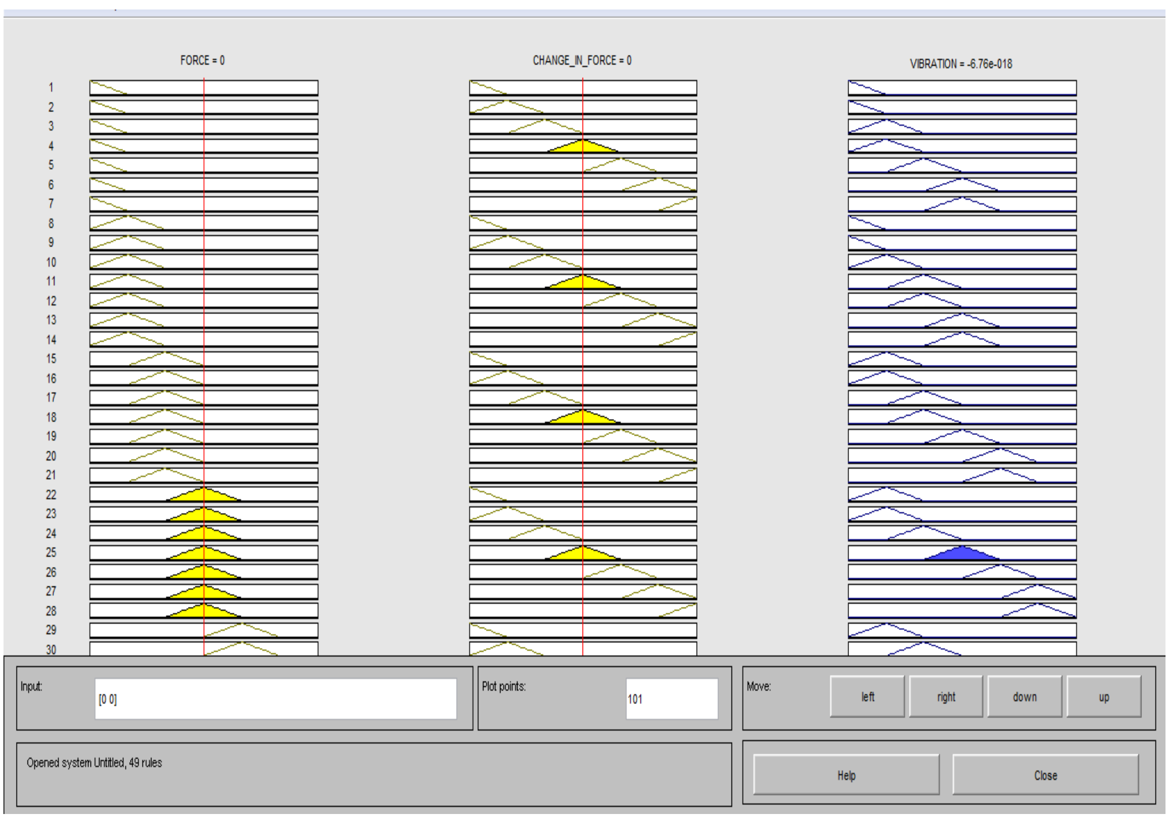


Figure 7.10: The rule viewer of fuzzy logic controllers

The Rule Viewer displays a roadmap of the whole fuzzy inference process. The first two columns of plots show the membership functions referenced by the antecedent, or the if-part of each rule. The third column of plots shows the membership functions referenced by the consequent, or the then-part of each rule. The yellow color (or shading) in first two plots represents the antecedent rules fired for a particular value and blue color (or shading) in third column represents the consequence of the antecedent on the output. Blue color line in the last block of third column represents the final precise value calculated using centroid defuzzification method.

The Rule Viewer shows one calculation at a time and in great detail. In this sense, it presents a sort of micro view of the fuzzy inference system. If the entire output surface of system is to be viewed, that is, the entire span of the output set based on the entire span of the input set, The Surface Viewer is required. Figure 7.11 shows the surface view of the system under consideration.

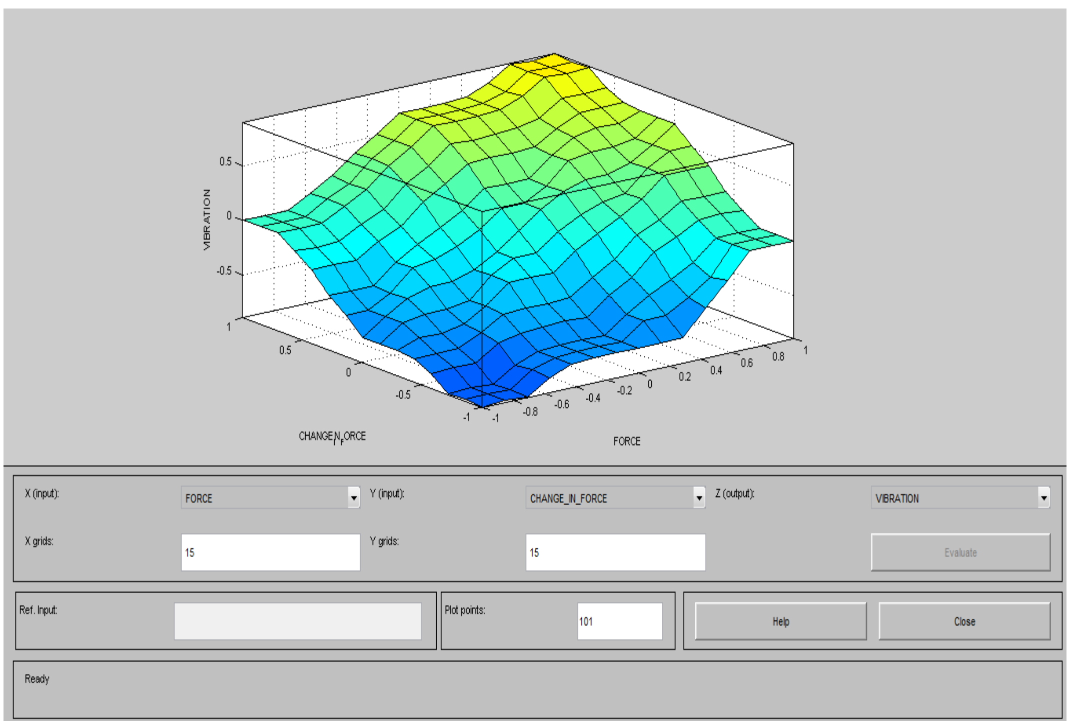


Figure 7.11: 3-D surface view of fuzzy logic controller

The Surface Viewer has a special capability that is very helpful in cases with two or more inputs and one output: we can actually grab the axes and reposition them to get a different three-dimensional view on the data.

Table 7.2: The fuzzy rules table

E \ Ec	NB	NM	NS	ZO	PS	PM	PB
NB	PB	PB	PM	PM	PS	ZO	ZO
NM	PB	PB	PM	PS	PS	ZO	NS
NS	PM	PM	PM	PS	ZO	NS	NS
ZO	PM	PM	PS	ZO	NS	NM	NM
PS	PS	PS	ZO	NS	NS	NM	NM
PM	PS	ZO	NS	NM	NM	NM	NB
PB	ZO	ZO	NM	NM	NM	NB	NB

The above table 7.2 shows the fuzzy rule base for tuning Fuzzy+AFC model for break control. There are total 49 rules in the above table.

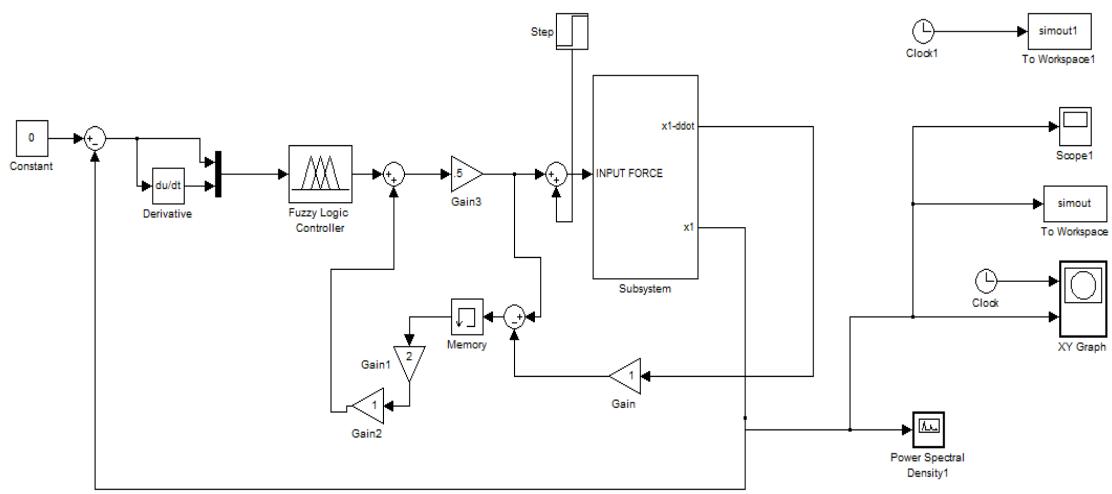


Figure 7.12: Fuzzy+AFC model for break control

7.4: Results and Discussion

After tuning the PID control system and obtaining suitable values for other relevant parameters, the simulation was executed for a period of 5 s since it is usual that the brake process does not take more than that duration. At first, the step disturbance is applied and then the simulation was performed without using AFC and only the pure PID controller was applied. The result of this process can be seen in Figure 11.13. It can be observed that the vibration that may result in producing noise is relatively high (a peak of more than 0.6 m in amplitude in some regions). Next the simulation was carried out again but this time considering 100% AFC mode plus the PID controller.

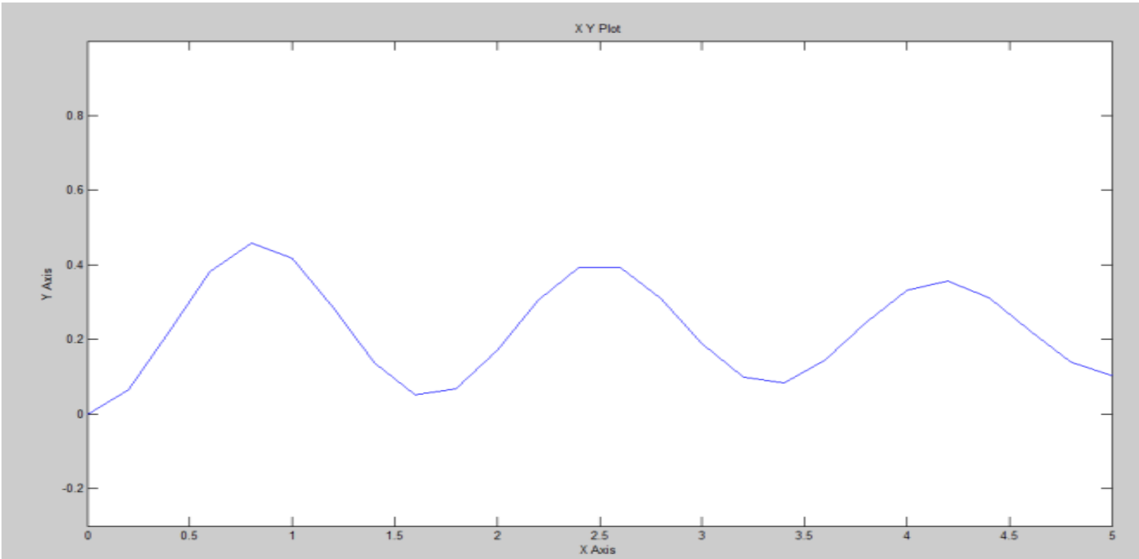


Figure 7.13: Response of the brake model with a step input disturbance for system with PID

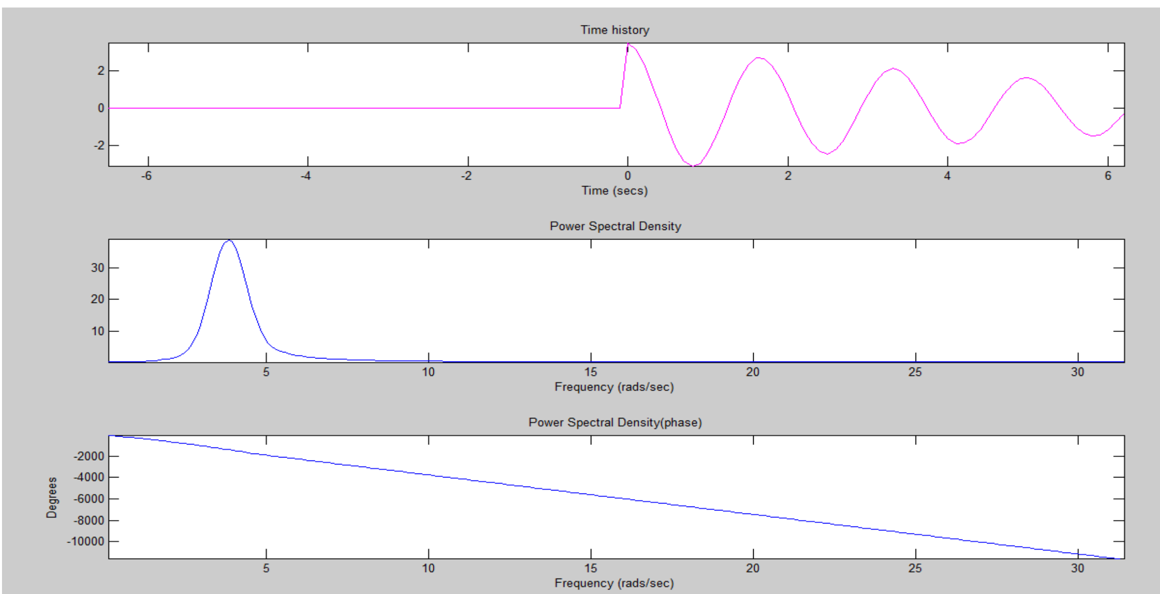


Figure 7.14: Response in frequency domain for PID

The result of the scheme is superimposed onto previous graph obtained as shown in Figure 7.13. It is evident that the vibration is virtually reduced to a very low value, implying that squealing can be almost if not totally avoided. A closed-up view of the graphical result of the PID+AFC scheme is depicted in Figure 7.15. It clearly shows the superiority of the scheme in rejecting the vibration of the system. The maximum amplitude of the vibration was less than 0.0004 m, which is very much lower compared to the brake system operated with a PID only controller.

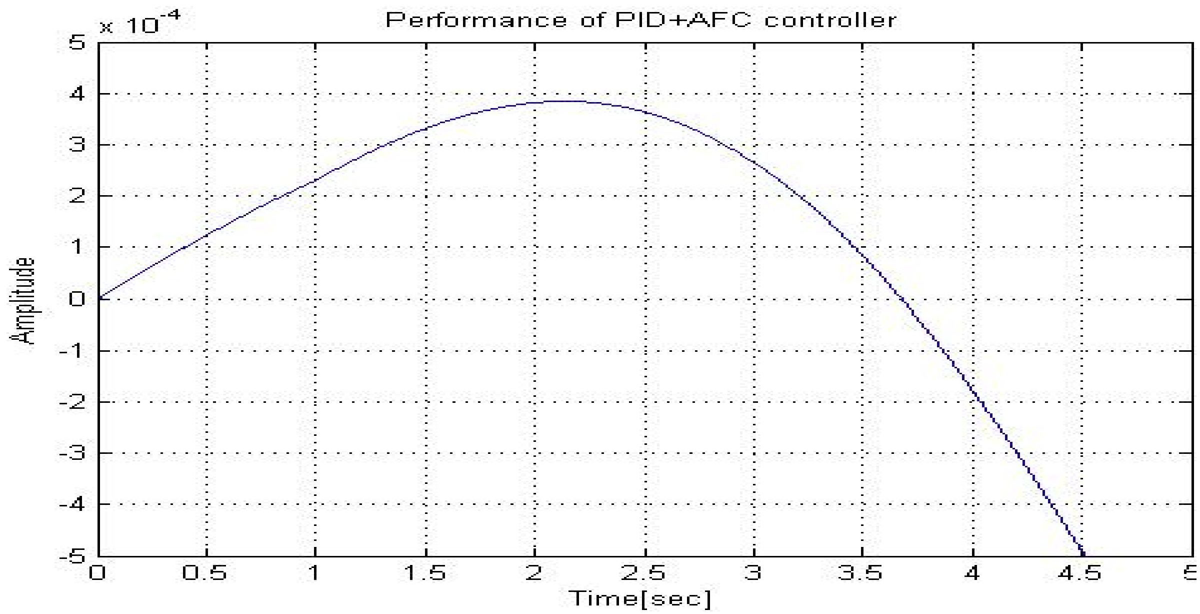


Figure 7.15 Performances of PID+AFC Controller

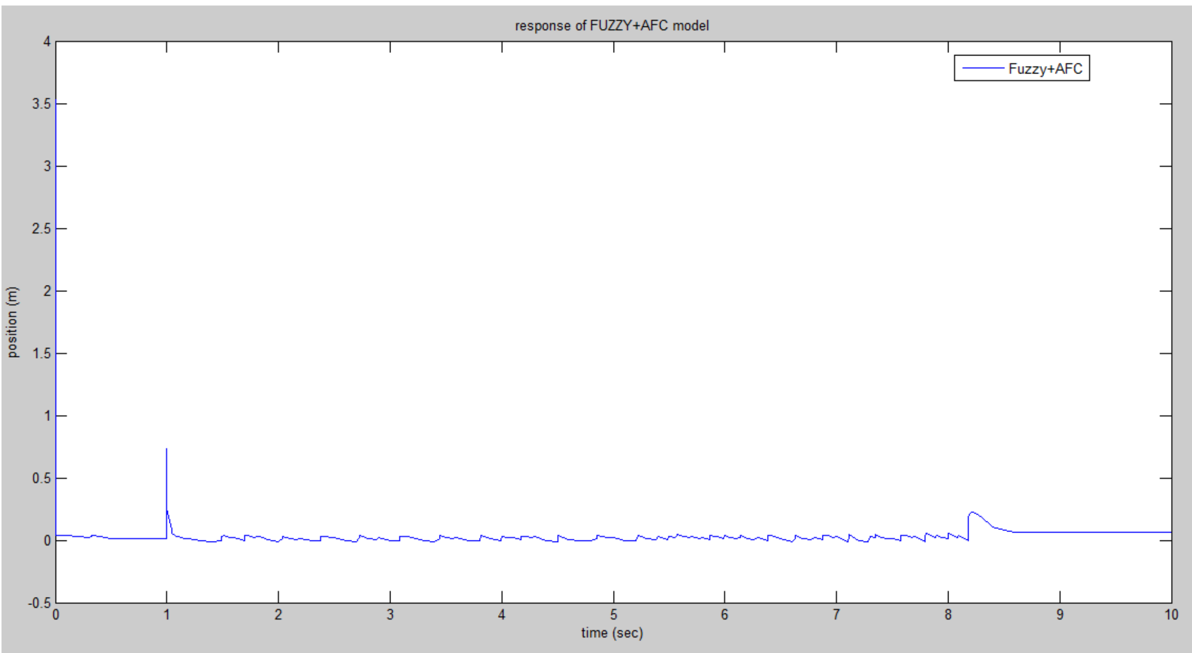


Figure 7.16: Amplitude vs. time plot using step disturbance in AFC+Fuzzy

Figure 7.16 clearly shows the superiority of the Fuzzy+AFC scheme in rejecting the vibration of the system. The maximum amplitude of the vibration was much lower compared PID+AFC model.

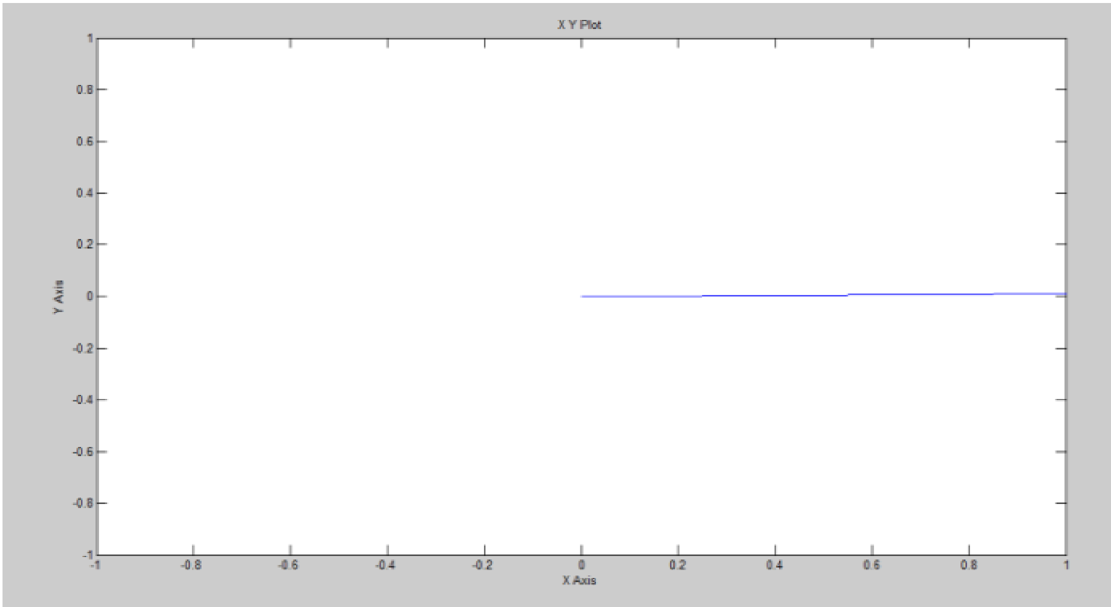


Figure 7.17: Response of the brake model with a step input disturbance for system with AFC+Fuzzy

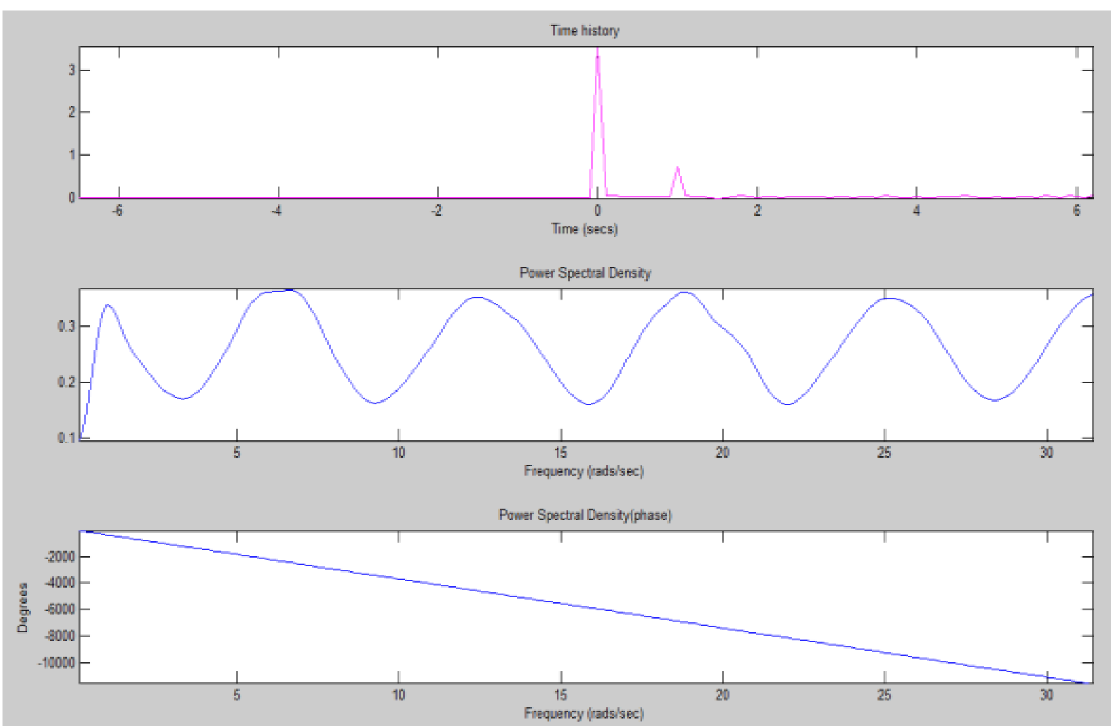


Figure 7.18: Response in frequency domain for AFC+Fuzzy

7.5: Conclusion and Future Scope

A novel AFC-based scheme has been proposed to suppress the vibration and noise (squeal) emanating from a disc brake system. Based on the simulation results, it is obvious that when a pure PID controller is applied to the brake system, vibration and noise are in fact reduced but still a noticeable amount of them remain. However, upon applying the AFC+PID the vibration and noise (squeal) are significantly reduced and approaching zero datum. Fuzzy+AFC strategy is robust and effective in countering the undesirable effects. Future works may include using neuro-fuzzy algorithm with AFC model.

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