ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

# DYNAMIC MODELING OF HORIZONTAL AXIS WASHING MACHINE AND OPTIMIZATION OF VIBRATION CHARACTERISTICS USING GENETIC ALGORITHMS

**M.Sc. THESIS**

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**Department of Mechatronic Engineering**

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

# YATAY EKSENLİ ÇAMAŞIR MAKİNALARININ DİNAMİK MODELLENMESİ VE TİTREŞİM DAVRANIŞININ GENETİK ALGORİTMA İLE OPTİMİZASYONU

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**OCAK 2012**

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**Date of Submission : 12 December 2011**
**Date of Defense : 03 January 2012**

***To my father,***

## FOREWORD

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January 2012

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## ABBREVIATIONS

| | |
|---|---|
| **FIR** | **:** Finite Impulse Response |
| **IIR** | **:** Infinite Impulse Response |
| **GA** | **:** Genetic Algorithm |

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**DYNAMIC MODELING OF HORIZONTAL AXIS WASHING MACHINE AND OPTIMIZATION OF VIBRATION CHARACTERISTICS USING GENETIC ALGORITHMS**

## SUMMARY


Washing machines generate vibrations in the full washing cycle but during the spinning cycle, the most excessive vibrations occur. These excessive vibrations can cause high wear and fatique failure of parts. Main reason for vibrations during spinning cycle is that laundry in drum sticks to inner wall of drum and uneven distribution of laundry causes unbalanced mass. With the new developments of higher spin speeds in washing machines, centrifugal forces increases. As a result of this, greater vibration amplitudes occur. Design parameters of washing machine, such as stiffness of springs, damping coefficient of shock absorbers and joining positions of these elements have to be chosen carefully. The system has to be stiff enough to keep oscillations small so that oscillating parts do not hit the cabinet and has to be not so stiff to prevent transmission of dynamic forces to cabinet. If the stiffness is less than the optimum condition, oscillation parts may hit the cabinet. Furthermore, if the stiffness is higher than optimum values, the so-called stepping condition may occur.

In this thesis, all the parts contributing vibration behavior are explained and a simple mathematical model is developed to predict dynamic behavior of a horizontal axis washing machine. The mathematical model is implemented in a computer simulation, using Simulink environment and model is solved numerically. Measurements are conducted on real washing machine ramping-up the full spin cycle, in order to validate the mathematical model. Some variables are derived from the acceleration data measured which required the application of appropriate signal processing tools such as filters and transforms. Therefore, the details of the signal processing are also mentioned in the measurement section.

For minimization of vibration amplitudes, a design vector is defined comprising the parameters of the washing machine. Optimization algorithms, for selecting optimum design vector, are reviewed and genetic algorithm is selected amongst them for optimization implementation for this study. Fundamentals of genetic algoritm are explained and a suitable algorithm for this study is developed. Different fitness functions are generated focusing on different characteristics of vibration behavior including transient and/or steady state. First, a fitness function considering maximum displacement of oscillating parts from origin is derived to minimize wobbling motion. Second, a fitness function focusing on steady state vibration amplitudes is defined. Finally, a fitness function, which is considering both transient and steady state vibrations is employed. For each fitness function, genetic algorithm is run and results are compared including the three optimization results and the simulation results with parameters before optimization, which is developed earlier in the study.

In this thesis, the main objective is to prove a concept that a simple mathematical model and genetic algorithm implementation can be involved to washing machine design process easily and effectively.

## YATAY EKSENLİ ÇAMAŞIR MAKİNALARININ DİNAMİK MODELLENMESİ VE TİTREŞİM DAVRANIŞININ GENETİK ALGORİTMA İLE OPTİMİZASYONU

### ÖZET


Çamaşır makineleri genel olarak yatay ve dikey eksenli olarak iki grupta toplanabilir. Amerika ve uzak doğuda dikey eksenli çamaşır makineleri popülerken, Avrupa ülkelerinde yatay eksenli çamaşır makineleri daha popülerdir. Yatay eksenli makinlerin dikey eksenlilere göre enerji tüketimi anlamında daha verimli olmaları bu grubu öne çıkartmıştır. Yatay eksenli çamaşır makinelerinin pazar payının artmasıyla bu makinelerin geliştirilmesinde harcanan çaba da artmıştır. Günümüzde çamaşır makinelerinin daha küçük hacimlerde olması istenirken sıkma devirleri de artmaktadır. Çamaşır makineleri çalışmaları esnasında titreşim üretirler. Özellikle sıkma devirlerinde bu titreşimler büyük genlik değerlerine ulaşırlar. Bu titreşimler parçalarda yüksek aşınma ve yorulma kırılmalarına sebebiyet verebilir. Bu titreşimlerin ana sebebi, sıkma devrinde, tamburdaki çamaşırların iç yüzeylere yapışarak dengesiz dağılımı sonucunda, dengesiz kütle oluşturmasıdır. Yeni geliştirilen çamaşır makinelerinde uygulanan yüksek sıkma devirleri nedeniyle oluşan merkez kaç kuvvetleri daha yüksek seviyelere ulaşmaktadır. Bu nedenle oluşan titreşimleri en aza indirgemek için çamaşır makinesi tasarımındaki; yay sertliği, sönüm katsayısı, yay ve şok emicilerin bağlantı pozisyonları gibi parametreler çok dikkatli seçilmelidir. Tasarlanan sistem, aşırı titreşim genliklerine müsaade etmeyecek kadar sert ve oluşan dinamik kuvvetleri gövdeye çok fazla iletmeyecek kadar da yumuşak olmalıdır. Aksi takdirde, hareketli parçalar gövdeye çarpabilir ya da yürüme olarak adlandırılan durum gerçekleşebilir.

Literatürdeki konuyla ilgili çalışmalara bakıldığında çok fazla çalışma olmadığı ya da olan çalışmalarında ticari sebeplerle gizlendiği görülmüştür. Literadürde mevcut çalışmalarda ise farklı amaçlar için matematiksel modeller oluşturuşmuştur. Literatürde yapılan optimizasyon çalışmalarında ise güncel olmayan optimizasyon metodları ile yalnızca yay ve sönüm özelliklerinde optimizasyon yapılmaya çalışılmıştır. Bu çalışmada ise basit bir matematiksel model ve genetik algoritmalı optimizasyon kullanılmıştır. Optimizasyon parametresi olarak yalnızca yay ve sönüm özellikleri değil, yay ve şok emicilerin bağlantı pozisyonları da kullanılmıştır.

Çamaşır makinesinin çalışması sırasında, makinenin kontrol ünitesinden elektrik motoruna istenen devir özelliklerine göre uygun miktarda elektrik enerjisi iletilir. İletilen elektrik enerjisi, motorda mekanik enerjiye dönüşür. Motorda oluşan açısal hız ve moment, kayış yardımıyla kasnağa iletilir. Kasnak, ortasında sabitlenen mile momenti iletir. Bu mil, rijit rulmanlarla kazana sabitlenmiştir ve ekseni etrafında dönebilmektedir. Milin diğer ucu da tambur yıldızı ile tambura bağlıdır. Böylece motorda oluşan moment ve açısal hız, tambura iletilmiş olur. Tamburun içerisinde bulunan yıkama sıvısı ve çamaşırlar bu hareket ile yıkama işlemine geçerler. 80-90 devir/dakikanın altındaki hızlarda çamaşırlar tamburun en alt kısmında kalır ve dönerler. Ancak bu hızlardan daha yukarı çıkıldığında, merkezkaç kuvvetlerinin

etkisiyle çamaşırlar tamburun iç yüzeyine yapışırlar. Bu yapışma sırasında oluşan eşit olmayan dağılım sonucunda dengesiz kütle oluşur ve titreşimlere sebebiyet verir. Kazan grubu, üst kısmından yaylar ile asılıdır. Bu yaylar da askı sacı denilen parçaya yani dolayısıyla çamaşır makinesi kabinine bağlıdırlar. Kazan grubunun alt tarafında ise yaylar ile aynı düzlemde bulunan şok emiciler mevcuttur. Şok emiciler de diğer uçlarından gövdeye bağlıdırar. Bu yay ve şok emiciler ile titreşimler belirli sınırlar içerisinde tutulurken, gövdeye iletilen kuvvetler de düşük tutularak makinenin sıkma devirlerine geçerken zıplaması ya da kayması engellenmektedir. Buradaki yay sertlikleri şok emicilerin sönümleme katsayıları ve bu elemanların bağlantı açılarının seçiminde optimizasyon süreci devreye girmektedir.
Optimizasyonu kolaylıkla gerçekleştirebilmek için de çamaşır makinesinin dinamik yapısının parametrik matematiksel modeline ihtiyaç duyulmaktadır. Bu çalışmada, belirli kabuller altında basit bir matematiksel model oluşturulmuştur. Yapılan kabullere göre, çamaşır makinesi osilasyon grubunu yalnızca x ve z eksenlerinde hareket etmektedir. Y ekseninde oluşan genlikler gerçek sistemde düşük olduğundan bu eksendeki hareket matematiksel modelde ihmal edilmiştir. Tambur ve kazan arasında herhangi bir göreceli hareket olmadığı kabulu yapılmıştır. Yani kazan ve tambur rijit olarak kabul edilmiştir. Çamaşır makinesi gövdesi yere sabitlenmiş ve rijit olarak kabul edilmiştir. Çamaşırların oluşturduğu dengesiz kütleyi temsil etmek için tambur iç yüzeyinde orta düzlemde 1 kg'lık noktasal yük olduğu kabulü gerçekleştirilmiştir. Osilasyon grubu bir silindir olarak modellenmiş ve bu silindirin kütle merkezinin orta eksenden geçitiği varsayılmıştır. Yine bu orta eksende yay ve şok emiciler bağlanmıştır. Osilasyon grubuna bağlı olan esnek elemanların sertlik ve sönüm karakteristikleri ihmal edilmiştir. Tambur yapılan modelde, 0 devir/dakikadan başlayıp 900 devir/dakika hıza kadar yükseltilmiştir.
Bu yapılan kabullerin ardından, çamaşır makinesindeki yay ve şok emici gibi elemanların karakteristiklerini öğrenmek için gerçekleştirilen ölçüm sonuçları değerlendirilmiş ve uygun olan lineerleştirmeler gerçekleştirilmiştir. Daha sonra sistemde oluşan merkez kaç kuvvetleri hesaplanmıştır ve bu kuvvet girdisi altında oluşan yerdeğiştirmelerden yararlanılarak yay ve şok emicilerin dinamik kuvvetleri hesaplanmıştır. Oluşan kuvvetlerin ifadeleri, Newton'un 2. Kanuna göre düzenlenmiş ve hareket serbestliği olan eksenlere indirgenmiştir. Ortaya çıkartılan matematiksel model, Matlab Simulink ortamında sayısal olarak çözülmüştür ve titreşim genliklerine ulaşılmıştır.
Çalışmanın bir sonraki adımında ise, oluşturulan matematiksel modeli doğrulamak maksadı ile deneysel ölçüm düzeneği oluşturulmuştur. Matematiksel modelde olduğu gibi çamaşır makinesi tamburunun iç yüzeyine 1 kg'lık dengesiz kütle yerleştirilmiştir. Çamaşır makinesinin elektrik motorunun ve dolayısıyla tamburun dönüş devir hızını kontrol edebilmek için varyak olarak bilinen voltaj ayarlayıcı cihaz ile elektrik motorunun giriş voltajı manuel olarak kontrol edilmiştir. İstenen devir hızları bu ayarlama sayesinde yapılmıştır. Oluşan genlikleri kaydedebilmek için ise kazanın üst kısmında osilasyon grubunun orta düzlemine ivmeölçerler yerleştirilmiştir. Bu ivme ölçerlerden elde edilen analog sinyal, veri toplama kartı kullanılarak bilgisayara sayısal olarak aktarılmıştır. Bilgisayara aktarılan titreşim verisi üzerinde, yüksek frekans gürültüsünü ve kapasitif ivme ölçerlerden dolayı oluşan DC terimi yok etmek için uygun filtreleme işlemleri uygulanmıştır. Filtreleme işlemlerinde IIR filtre sınıfından 5. derece Butterwort filtreler kullanılmıştır. Bu filtrenin seçim sebebi ise hem geçiş bandında hem de durdurma bandında düz bir karakter sağlaması ve düşük derecelerde keskin kesme frekansları sağlayabilmeleridir. Bu avantajlarının yanı sıra sinyalde faz kaymasına sebep

vermelerinden ötürü de uygun işlemler uygulanmış ve tam lineer faz elde edilmiştir. Filtreleme işlemlerinden sonra ise ivme verisi iki kez arka arkaya integre edilmiş ve önce hız daha sonra da yer değiştirme verisi elde edilmiştir. Bu kısımda, yalnızca yerdeğiştirme verisi bulunmamış aynı zamanda da tambur miline enkoder bağlanmasındaki güçlüğü aşmak için de anlık frekans kestirim yöntemleri kullanılarak tambur devrinin ne olduğu da hesaplanmıştır.

Bir sonraki bölüm olan doğrulama bölümünde ise ölçüm evresinde bulunan titreşim genlikleri ile çalışmanın başında gerçekleştirilen matematiksel modelden elde edilen titreşim genlikleri ve karakteristikleri karşılaştırlmıştır. Yapılan değerlendirme sonunda ölçüm genlikleri ve matematiksel modelden elde edilen genlikler arasında hatanın az olduğu görülmüştür. Hem gerçek sistemin hem de oluşturulan simülasyonun doğal frekanslarını karşılaştırmak için giriş devri cinsinden genlik verileri çizdirilmiş ve karşılaştırılmıştır. Bu karşılaştırma sonunda gerçek ve model arasında yapılan kabullere bağlı küçük kaymalar gözlemlenmiştir. Oluşturulan modelin optimizasyon için yeterli olduğu kararı alındıktan sonra da optimizasyon kısmına geçilmiştir.

Optimizasyon bölümünde, optimizasyon yöntemleri genel olarak sınıflandırlmış ve bu sınıfların üstünlükleri kıyaslanarak sayısal yöntemler tercih edilmiştir. Sayısal optimizasyon metotları da gözden geçirildikten sonra genetik algoritmalı optimizasyon metodu seçilmiştir. Genetik algoritmalı optimizasyonda doğadan esinlenilmiş ve Darwin'in en iyilerin seçimi teorisi uygulanmıştır. Bu yöntem ilk olarak 60'larda optimizasyon problemlerinde kullanılmış ve günümüze kadar popülerliği artarak her alanda kullanılmışlardır. Optimizasyon için öncelikle genetik algoritmanın temelleri anlatılmış ve problem parametreleri genetik terimler cinsinden tanımlanmıştır. Optimizasyonda kullanmak üzere oluşturulan tasarım vektörüne, yay katsayısı, sönüm katsayısı, yay bağlantı noktalarını x ve z eksenlerindeki mesafeleri, şok emicilerin x ve z eksenlerindeki bağlantı noktalarının mesafeleri olmak üzere 6 adet değişken kullanılmıştır. Bu tasarım parametrelerini kullanarak bir arama uzayı oluşturulmuştur. Uzayın seçiminde ise şu anki çamaşır makinesinin mevcut parametreleri göz önünde bulundurulmuş ve mevcut tasarımın etrafında daha iyi bir çözüm olup olmadığı araştırılmıştır. Tasarım parametreleri ikilik sisteme göre kodlanmış ve genetik algoritmaya tanıtılmıştır. Bu parametreler kodlandıktan sonra rastgele 20 bireylik bir popülasyon oluşturulmuştur. Bir sonraki adımda en iyilerin seçilebilmesi ve çoğalabilmesi için seçim operatörü tanımlanmış ve uygunluk orantılı seçim yöntemi kullanılmıştır. Seçim operatörü tarafından seçilen bireyler önce çaprazlamaya tabi tutulmuşlar ve yeni bireyler üretilmiştir. Üretilen yeni bireyler de mutasyon işlemine tabi tutulmuşlardır. Bu çaprazlama ve mutasyon işlemlerinin oluşma olasılığı da genetik algortimanın parametrelerinden olup kontrol altında tutulmuştur. Yeni bireylerde farklılık olabilmesi için çaprazlama olasılığı yüksek tutulmuştur. Mutasyon olma olasılığı ise oldukça düşük tutulmuştur. Bu operatör çözümün yerel minimumlarda takılmasını önleyen bir operatördür ve eğer olasılığı yüksek tutulursa arama uzayında gereksiz sıçramalara yol açabilmektedir. Bir sonraki işlem olarak genetik algoritmada elde edilen bireylerin uygunluğunun değerlendirilmesi gerekmektedir. Bu amaçla üç farklı uygunluk fonksiyonu yazılmıştır. Bu fonksiyonlardan ilki çamaşır makinesinin hareketli parçalarının ilk konumlarından ne kadar uzağa gittiği ile ilgilenmiş ve bunu minimize etmeye çalışmıştır. İkinci uygunluk fonksiyonu ise kararlı rejimde oluşan titreşimler ile ilgilenmiş ve bu bölgedeki genlikleri minimize etmeye çalışmıştır. Üçüncü ve son uygunluk fonksiyonu ise hem geçici rejimdeki büyük genlikler hem de kalıcı rejimdeki genlikler ile aynı anda ilgilenmiş ve toplamda bir minimizasyon yapmıştır.

Bu uygunluk fonksiyonuna göre uygunluk değerleri atanan bireylerin uygunluk değerleri bir sonraki seçimde kullanılmıştır. Genetik algoritma bu bahsedilen, seçim, çoğalma, uygunluk atanması, seçilme şeklinde giden çevrimi istenilen durma kriterine kadar gerçekleştirmektedir.

Durma kriteri olarak bu çalışmada iki aşamalı bir kriter kullanılmıştır. Birinci aşamada oluşturulan jeneresyonlardaki iyileşme miktarı gözlemlenmiş ve kayda değer gelişme olmadığı görüldüğünde algoritma sonlanmıştır. Bu kriter genetik algoritmanın olasılıksal doğası gereği her zaman sağlanamamaktadır. Bu sebeple ikinci aşamaya ihtiyaç duyulmaktadır. Bu aşamada ise azami bir iterasyon sayısı koyulmuştur. Birinci aşamada durmadığı takdirde genetik algoritma ikinci aşamada durdurulmuştur. Elde edilen optimizasyon sonuçları ilk olarak çalışmanın başında geliştirilen matematiksel modelin sonuçları ile karşılaştırılmış daha sonra da optimizasyon sonuçları kendi içinde karşılaştırılmıştır.

Bu çalışmada, basit bir matematiksel model ve bu model üzerinden genetik algoritmalı optimizasyonun tasarım sürecine uyarlanabilirliğinin konsepti kanıtlanmaya çalışılmıştır.

# 1. INTRODUCTION

In modern world, we use various types of washing machines with different purposes and in different applications areas such as industrial and domestic. In the class of domestic washing machines there are two main types; (i), front-load horizontal axis washing machine and (ii) top-load vertical axis washing machine. In USA and Far East countries, vertical axis washing machines are very popular because of ergonomic loading of machine. In contrast to this, among the European countries horizontal axis washing machines are highly popular. In addition to ergonomy, another aspect in design of washing machines is energy efficiency. With growing common sense in energy efficiency, energy consumption of house appliances become a very important issue. Related to this topic, research shows that horizontal axis washing machines are more efficient than vertical axis washing machines [1]. Therefore, horizontal axis washing machines are increasing their share in market.

With increasing market share of horizontal axis washing machines, development efforts on this topic became larger. In the development of new types of washing machines, the requirements of modern life are needed to be taken into account. The modern life requirements and standards impose shorter washing cycle times, less vibration and less noise and little space for washing machines at homes. Considering all these constraints, some design problems and conflicts arise. For low noise, low vibration and little spaces, spring and shock absorber and joining positions of these elements in washing machine has to be selected carefully. On the other hand, motion of washing machine cabinet relative to ground, so called stepping, is a very crucial performance criterion. In each of all the cases with any loading condition and for any spinning speed of drum, during ramp-up and ramp-down, thus, in normal operation boundaries there must not be any stepping. With shorter washing cycles, spinning cycles have to happen in higher rotational speeds which causes more centrifugal force, hence more vibration and noise.

For all types of washing machines common cause of vibration and noise is centrifugal forces generated by uneven distribution of laundry mass in the drum and higher spin speeds. In ramp-up and ramp-down of spinning cycles, drum sweeps all the frequency range including the natural frequency of washing machines oscillation group, therefore high amplitude vibration may occur at natural frequency. These high amplitudes may cause hitting of moving parts to stationary ones or high wear of spring and shock absorbing elements. To reduce high amplitude vibration, one can easily use stiffer springs and shock absorbers and this may solve the problem but in this case, very high forces will be transmitted to washing machine cabinet and this situation may lead to so called stepping condition. Designing the washing machine to avoid stepping condition, which is a critical performance criterion, and to minimize vibrations and noise of washing machine lead us to an optimization problem.

The purpose of this thesis is to develop a simple mathematical model that represents real dynamics of a horizontal axis washing machine with certain assumptions and write a genetic algorithm optimization in order to optimize spring and damping characteristics while not violating stepping condition. The geometric location of spring and shock absorber joining points to cabinet is also in scope of optimization.

### 1.1 Background and Literature Review

Although washing machines are very common and well known systems, there are a few of studies in modeling and optimizing of vibration characteristics of washing machines. Probably the main reason for this situation is that studies funded by washing machine manufacturers are kept as a commercial secret. Some of the leading studies about dynamic behavior of washing machines are surveyed here to assess the state of art in modeling and optimization in this field.

İ.Tahsin Sümer developed a theoretical simulation model, which can give good predictions of an automatic horizontal axis washing machine in his MSc thesis under supervision of O.Türkay. İ.T Sümer used his model to understand dynamic behavior of horizontal axis washing machine and tuned system mass, stiffness and damping characteristics in order to obtain acceptable vibrations of oscillation group meanwhile satisfying the condition of no-stepping [2].

Secondly, again under supervision of O.Türkay, B.Kıray, used the mathematical model which was developed by İ.T.Sümer mentioned above and generated an easy and simple optimization software to make a parametric optimization with three design variables. They used grid method and sequential quadratic programming technique and they discussed implementation of these two methods to general optimal suspension design problem to horizontal axis washing machine [3].

Hee-Tae Lim et. al conducted dynamic analysis of horizontal axis washing machine using simplified dynamic model considering gyroscopic effects. They were mainly interested in relative motion between drum and tub during spinning. They had three significant out comes from the research (i) the model in their study was consistent (ii) design cycle could be shortened with the model and (iii) it was found that the stiffer drum was more efficient than stiffer tub [4].

M.Ersin Öztürk, in his MSc thesis, under supervision of H.Erol, used a commercial software(MSC ADAMS) to dynamically model the whole washing machine considering all parts. In this study, they were interested in both cabinet vibrations and oscillation group vibrations. They validated their model and declared that developed model can be used for design and development of new washing machines [5].

F.Bayraktar and H.T. Belek in their study developed a methodology to optimize vibration behavior of a washing machine. Within 0-400Hz frequency range, they modeled the vibration of side panels on cabinet using finite element method, in addition to application of experimental modal analysis and component mode synthesis methods. They recommended a new structural design to minimize cabinet vibrations [6].

S.Bae et.al developed a mathematical model for vertical axis washing machine and designed a hydraulic balancer to reduce the vibrations during spinning cycle and they showed parameters are affecting the vibration of a washing machine most [7].

E.papadopoulos and I. Papadimitriou studied a portable washing machine. In their study, they formed a simple model with a single degree of freedom to predict walking behavior of horizontal axis washing machine. In addition to this, they designed an active vibration control system taking a contrary approach to S.Bae et.al's passive balancing [8].

C.D.Carroll, in his PhD thesis, studied both horizontal and vertical axis washing machines. He built simple models to examine dynamic behavior. In his study, he drew the attention of design engineers to dynamic constraints and he also designed a new active balancer [9].

The literature aforementioned has some common ground and is generally about modeling vibration characteristics of oscillation group, cabinet vibrations, relative motion of tub and drum, active balancers, hydraulic balancers and spring- damping characteristics optimization separately. In this thesis, vibration of cabinet or side panels, active or hydraulic balancers and relative motion between tub and drum are not considered. Instead, a simple model, which matches measured data well, is developed in order to optimize spring and damping characteristics as well as geometry of joining positions of spring and shock absorbers to cabinet. In this study, we used one of the evolutionary algorithms, genetic algorithm optimization method whereas the literature mentions the use of grid method and sequential programming.

## 1.2 Method of Study

The study consists of the following major phases, which correspond to several chapters in this thesis.

i. **Dynamic Modeling Of Washing Machine** (Chapter 2)

In this phase, definition of engineering problem is carefully stated. Actual washing machine, its components and interactions of components are described. Assumptions that used in mathematical model are identified. Region of interest in both output and input is clarified. Modeling of each component is explained separately. Mathematical model and its representative in Simulink environment are explained in detail. Finally, results of mathematical model are shown.

ii. **Vibration Measurement of Washing Machine** (Chapter 3)

This phase is concerned with data acquisition of acceleration data from real washing machine. Data acquisition system is described briefly and signal processing applications in order to condition the signal are explained. Process of obtaining the displacement information from acceleration data is described in detail such as filters and integration methods. Drum rotational spin speed is also extracted from acceleration data. In this process, instantaneous frequency estimation and smoothing applications are described and used.

**iii. Model Validation** (Chapter 4)

In order to verify the accuracy of mathematical model, comparison with measured data is performed in this phase. Simulation results obtained in Chapter 2 and measurements performed in Chapter 3 are compared in Chapter 4.

**iv. Parameter Optimization** (Chapter 5)

Optimization methods are reviewed first. As an evolutionary algorithm, genetic algorithm (GA) is selected. Fundamentals and flow chart of genetic algorithm are shown and all operators used in genetic algorithm are described. Next, goals of optimization, objective functions and constraints on the objective functions are defined. And finally, results of optimization are presented with comments on the further studies.

# 2. DYNAMIC MODELING OF WASHING MACHINE

In this chapter, dynamic modeling phase is discussed. First of all, real system is investigated and complexity level of the model is determined. Hereafter, a computer simulation is introduced in Simulink environment. The term *computer simulation* refers to a technique in which mathematical models are developed to capture some important characteristics of a system under study. These mathematical models, often in form of ordinary differential equations, are then solved numerically in a computer simulation [9]. Here, mathematical model is developed in order to represent system dynamics in real washing state and that model is used in order to optimize the system. Here, a critical point arises, how complex the model will be and which properties to be neglected. This is a very important question to be answered before modeling phase starts.

The real physical system, its components and effects of these components on each other should be known. Next, the area of interest in dynamics should be clarified before the study and then decision can be made for deciding on which properties to take into account or not. By using this path, a lean and simple model is developed in order to represent interested area in dynamics.

## 2.1 Actual Washing Machine System Definition

A commercially available washing machine is shown in figure 2.1. This washing machine is a classic example of front load horizontal axis washing machine, which is the case of study in this thesis.

**Figure 2.1 :** Front load, horizontal axis washing machine.

In figure 2.1 only control panel, detergent dispenser and window can be seen. These are the visual parts and they have nothing to do with dynamic characteristics of washing machine. In this aspect, interior parts of washing machine is shown separately in figure 2.2. In figure 2.3 and figure 2.4 parts are marked and explained sequently.

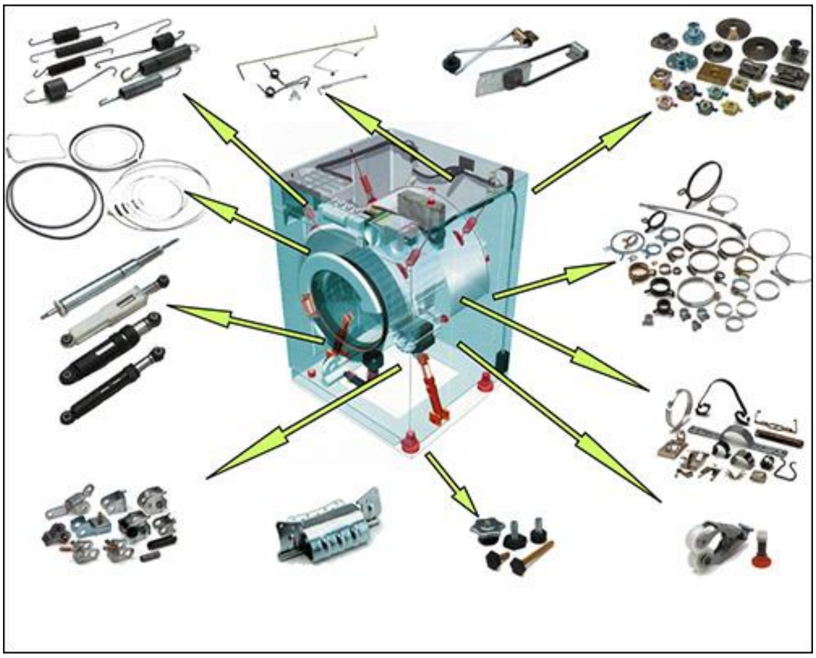

**Figure 2.2 :** Insight of washing machine.


**Figure 2.3 :** Front view of open washing machine.

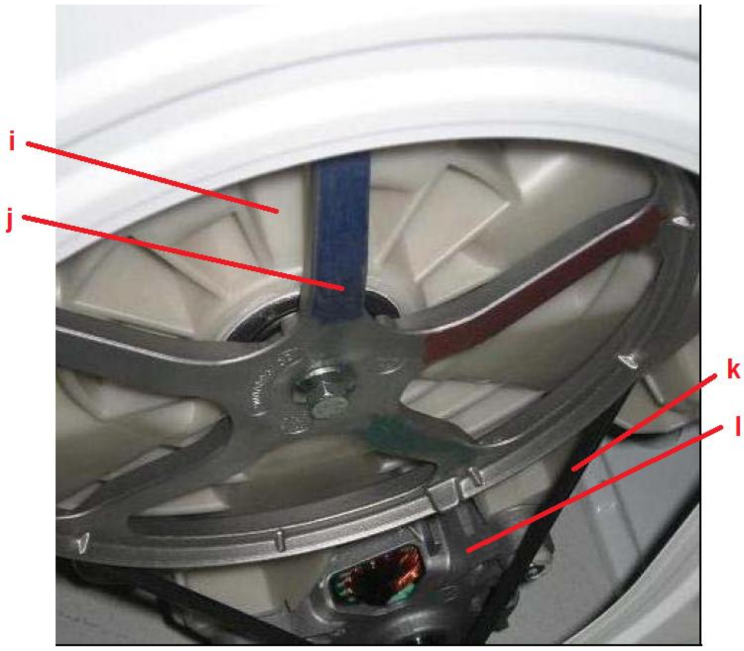


**Figure 2.4 :** Back view of open washing machine.


**Figure 2.5 :** Bottom front view of open washing machine.

a. Top counter weight

b. Spring strut

c. Spring

d. Tub

e. Drum

f. Cabinet

g. Front counter weight

h. Unbalance load

i. Back of the tub

j. Pulley

k. Belt

l. Electric motor

m. Shock absorber

n. Gasket

As in figure2.2, there are several components in the part list of a washing machine. The important point here is to select whether the certain parts are going to be taken into account or not. The relevant interior parts of washing machine are listed above. Brief information of interior parts is as follows. Drum is the inner part of tub, holds the laundry inside, and rotates around horizontal axis. After certain spin speeds, laundry sticks to inner surface of drum and causes uneven distributed load and consequently centrifugal forces. In figure 2.3, window and gasket are removed so that unbalanced load can be seen. Drum is connected to the tub with bearings rigidly. The connecting shaft is coupled with a pulley as can be seen in figure 2.4. Pulley and drum is driven by the belt which is actuated by an electric motor. The electric motor is fixed under the tub.

Tub is the main part of washing machine that gathers all parts on it. The tub and all other parts attached to tub are together called as oscillation group. The name is defining the phenomena clearly, that all of the parts are hanged to cabinet with spring and shock absorbers are vibrating all together as a rigid body. Main duty of tub is to hold the washing liquid in it and not to leak it out. Springs and shock absorbers are attached to tub. With the effect of unbalanced load spinning, the tub and all other parts on it vibrates. In order to reduce these vibrations and to prevent tub and all the oscillation group to hit the cabinet, counter weights are attached to the tub. Figure 2.6 shows the sealing element named gasket.

Water inlet is the flexible part that connects the detergent dispenser to the tub. Gasket is the sealing element that prevents water leakage during wash cycle between the tub and the window.


**Figure 2.6 :** Oscillation group with gasket.

Spring is attached to tub at one end and the other end is attached to spring strut on the cabinet. Shock absorbers can be seen in figure 2.5 are also attached to cabinet at the bottom. Cabinet is a housing part to all other parts such as oscillation group, control panel, detergent dispenser and water inlet.

To clarify the operation flow of washing machine and defining relations of the parts between each other, the working principle is detailed. Electric motor attached to tub is driven by a control signal and starts to produce torque and rotational speed. That torque and speed is transmitted to drum with belt and pulley mechanism. Drum is fixed to tub with rigid bearings. The laundry in drum sticks to the inner surface of drum after rotational speed about 80-100 rpm [11]. Sticked laundry creates an unbalanced load and with the rotation of drum, it generates centrifugal forces. Centrifugal forces, causes oscillation group to move in the cabinet. Springs and shock absorbers are connecting oscillation group to cabinet and they prevent oscillation group to hit the cabinet while they absorb some of the energy and prevent excessive force transmitting to cabinet, which may cause stepping condition if in excess amounts.

## 2.2 Model Assumptions

Not all the parts described before in the washing machine are going to be involved in the mathematical model. There have to be some simplifications done in order to reduce the complexity and to analyze the model easily. Figure 2.7 shows the positive directions of selected axes. Vibrations occur during the ramp-up period of washing machine. Drum sweeps all the frequencies, which include natural frequencies of oscillation group, up to steady state spin speed. When drum reaches the natural frequency zone, vibrations of oscillation group reaches the maximum amplitudes. These vibrations occur in all axes; x,y and z. There are also rotational vibrations around x, y and z axes. In brief, washing machine has 6 degrees of freedom to describe its motion as it vibrates.

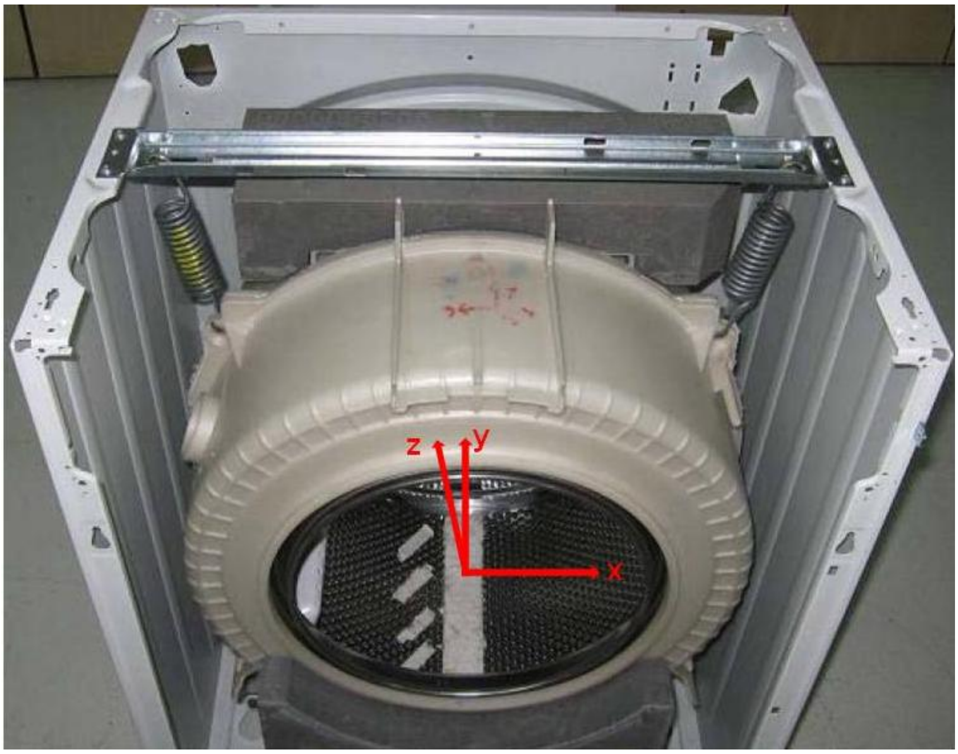


**Figure 2.7 :** Global axes.

The schematic front view of drum and the unbalanced mass is shown in figure 2.8. $m_u$ here is the unbalanced mass, $\beta_d$ is the rotation angle of drum. If plane showed in figure 2.8 is in the middle plane of drum, then all the centrifugal forces will be distributed to x and z axes with respect to rotation angle $\beta_d$ and there will be no force on y axis. If the unbalanced mass is not in the center of drum, in other words if the plane showed in figure 2.8 is not in the middle, then there will be some centrifugal forces on y axis caused by rotations around x and z axes. Hence displacements on y axis will occur, which means oscillation group moves forward and backwards beside moving on x and z axes.

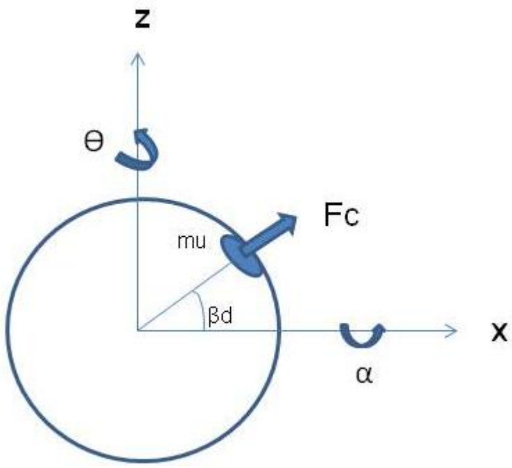


**Figure 2.8 :** Simplified unbalanced mass distribution in the middle of y axis.

According to vibration measurements, done in 3-axis, the vibration on y axis was not significant as the vibration occurs in x and z axes. The reason for this phenomena is obvious that the components of the centrifugal force are aligned on x and z axes when decomposed in its components, thus centrifugal force does not have a y-axis component.

Hence, our mathematical model will not have any degree of freedom on y axis. Also the rotational vibrations around x, y and z axis, respectively $\alpha, \beta$ and $\theta$ are not taken into account. After these assumptions our model becomes 3 degrees of freedom, translations on x and z axis and rotation of drum in x-z plane.

As mentioned before, the location of the laundry in the drum rotates with drum on washing cycle. However, in the spin-drying cycle, drum rotates up to 900 rpm and laundry sticks to inner wall of drum. Sticking event happens around 80-100 rpm spin speeds. In the light of this information, centrifugal forces during spin speeds less than 100 rpm will not be significant. Hence, our area of interest in vibrations is narrowed down to 100 rpm to steady state spin speeds, which in our case will be 100-900 rpm spin speed. In mathematical model, the input of the system will be centrifugal forces and these forces depend on spin speed and the unbalanced mass values. In our case, unbalanced mass is a constant value hence the main variable input of our model is spin speed.

Gasket and the water inlet are flexible parts and they have their own damping and spring characteristics. They obviously have some effects on vibrating dynamics of oscillation group but according to İ.T.Sümer's study [2], the effects of these elements are not very significant compared to main oscillation group elements. Hence, in this study they are not taken into account. The main terms of our model will be springs, shock absorbers, unbalanced mass, rotational speed of drum and the mass of oscillation group.

To conclude the model forming section, following assumptions and simplifications are employed in the model.

- The oscillation group can move on x and z axis only. Translation on y and all rotations are restricted.
- Drum can rotate in the tub but there are no gyroscopic effects on drum.
- Washing machine cabinet is rigid and fixed to ground
- Unbalanced mass is constant and fixed to inner wall of drum (sticking condition around 100 rpm).
- Unbalanced mass is in the middle plane of oscillation group. (The plane that also springs and shock absorbers are attached)
- Oscillation group is modeled as a cylinder and the mass center is on the cylinder center.
- Spin speed of drum sweeps spin speed from 0 to 900 rpm and rises as in experiments.
- Stiffness effects of flexible parts such as gasket and water inlet to tub are neglected.

### 2.3 Mathematical Model

In this model, Newton's law of motion is employed to generate dynamic equilibrium equations of washing machine oscillation group. Generation of external forces caused by unbalanced mass is explained and then the reaction, forces of external force by springs and shock absorbers are discussed. After employing mentioned assumptions, model is shown in figure 2.9 and figure 2.10.

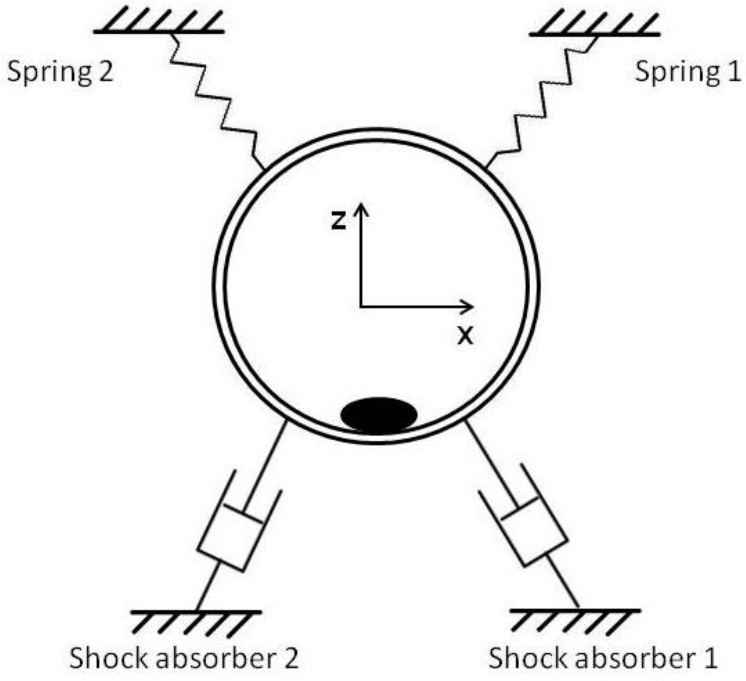


**Figure 2.9 :** Washing machine at equilibrium state.

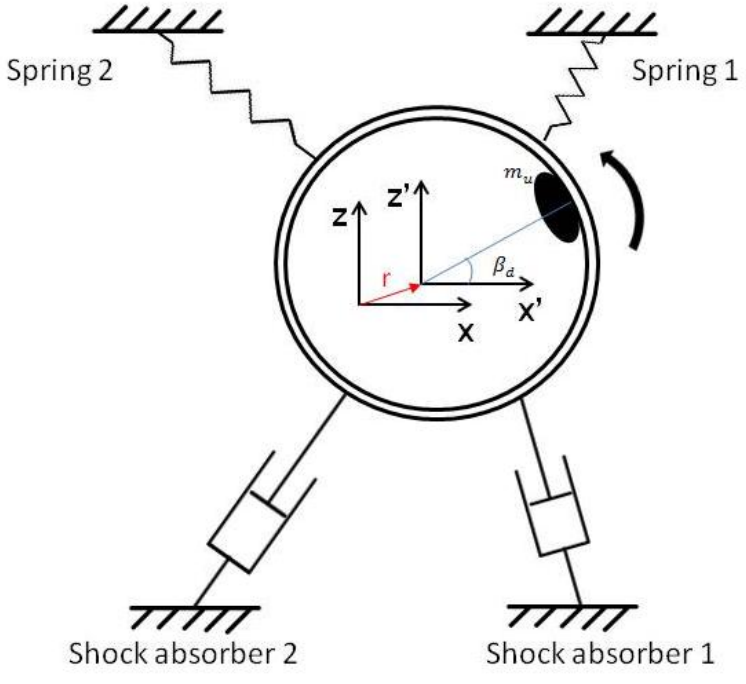


**Figure 2.10 :** Washing machine moved on positive x and z direction.

Figure 2.9 shows washing machine at initial position at rest, while Fig 2.10 shows the drum spinning and unbalanced mass sticked on the inner wall with the generated external forces. In Fig 2.10, center of oscillation group movement is shown with the positive displacement vector $\vec{r}$ .

### 2.3.1 Centrifugal forces

With the rotation of drum and the existence of unbalanced mass, there will be centrifugal forces generated and with respect to drum rotation angle, we can reduce these forces on the x and z axes. Spin speed reaches to 900 rpm in the experiments; hence, in simulation as an input we have to generate a similar data as real input spin speed. In order to obtain a ramp-up data, following equation is derived.

$$\beta_d{}' = N(1 - e^{-1/1,8t}) \tag{2.1}$$

*N*, here is the final revolution speed of drum in rpm and *t* is the time. The exponential expression is used to obtain characteristic speed curve having a ramp-up and a settlement to a final value. With this equation we can obtain spin speed of drum as an input of simulation. After integration, we can obtain angle of unbalanced mass $\beta_d$ and with a derivation we can obtain angular acceleration of drum. Figure 2.11 shows$\beta_d{}'$, the spin speed of drum.

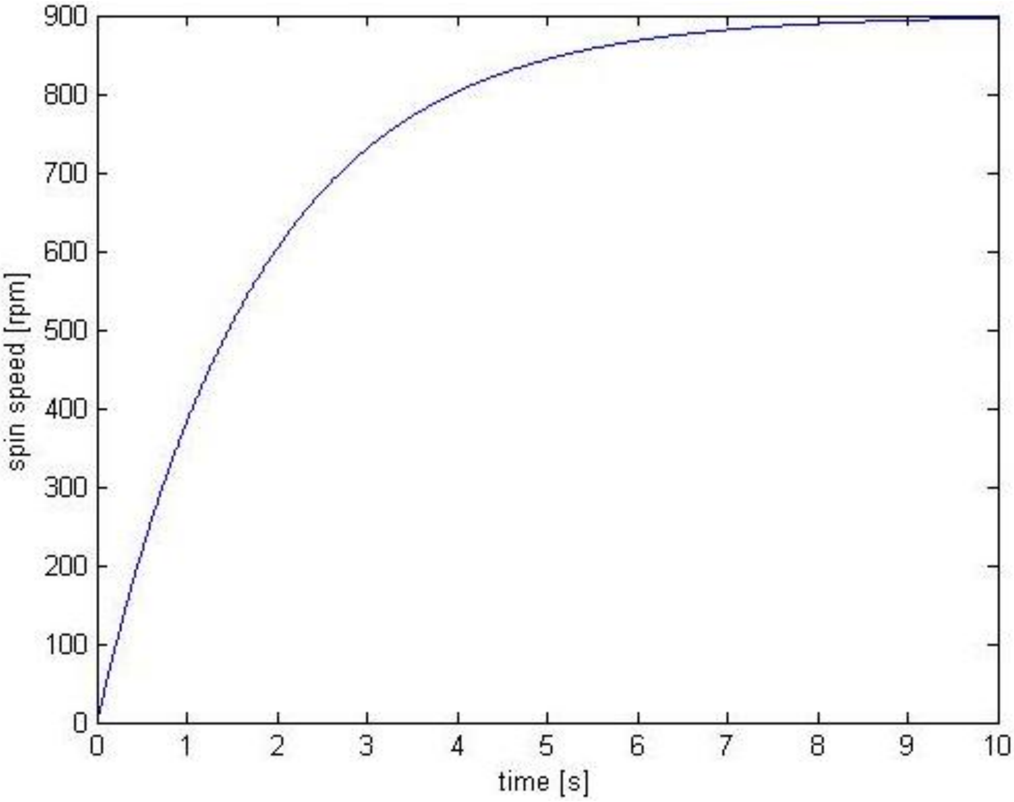


**Figure 2.11 :** Spin speed of drum in simulation.

The figure 2.12 shows the rotation direction of drum and the forces acting on unbalanced mass, decomposed in tangential and normal directions.

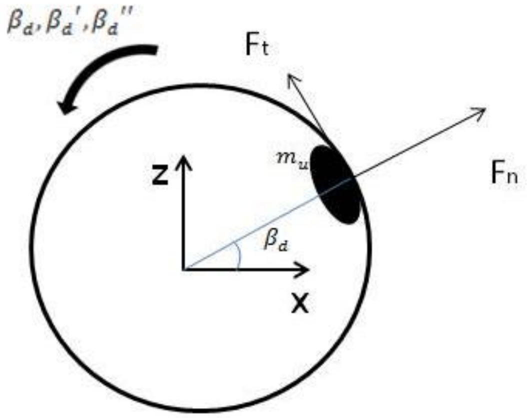


**Figure 2.12 :** Normal and tangential centrifuge forces.

According to Newton's second law of motion, the forces acting on unbalanced mass is as fallows,

$$F_n = m_u r (\beta_d{}')^2 \quad \textbf{(2.2)}$$

$$\mathrm{F_t = m_u r \beta_d{}''} \quad \textbf{(2.3)}$$

Reducing these forces on to x and z axis, we use simple geometric relations shown in figure 2.12

$$F_{x\,ext} = F_n \cos(\beta_d) - F_t \sin(\beta_d) \quad \textbf{(2.4)}$$

$$F_{z\,ext} = F_n \sin(\beta_d) + F_t \cos(\beta_d) \quad \textbf{(2.5)}$$

The simulation of these $F_{x\,ext}$ and $F_{z\,ext}$ forces yielded the results shown in figure 2.13 .

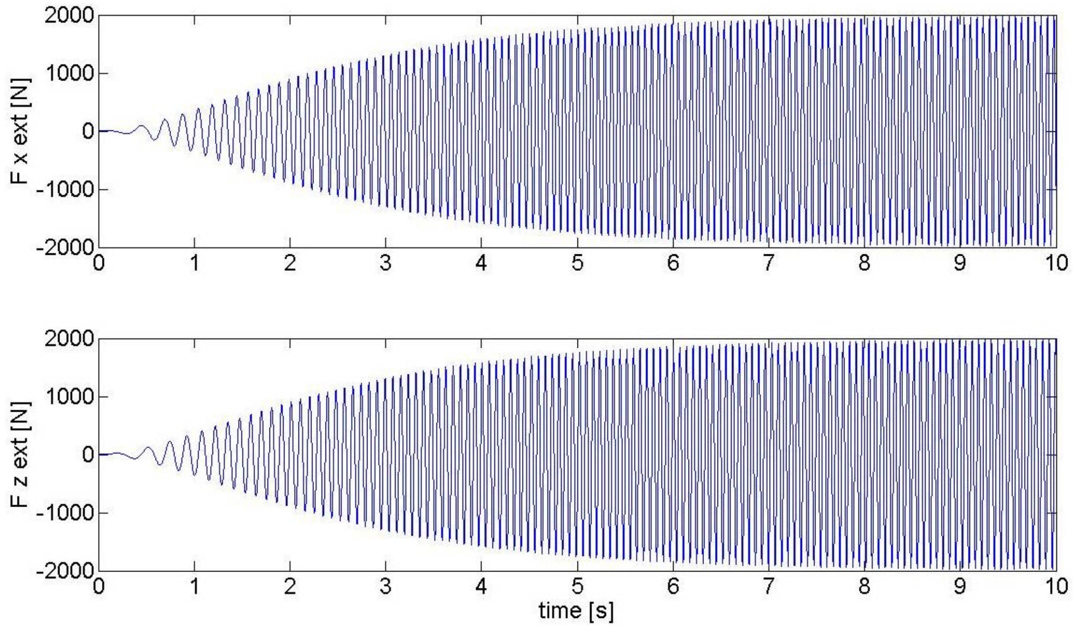


**Figure 2.13 :** Simulation results of external forces.

### 2.3.2 Spring model

Springs used in washing machine are simple tension springs and their characteristics are linear. Plot of experimental measurements of springs is shown in Fig 2.14.

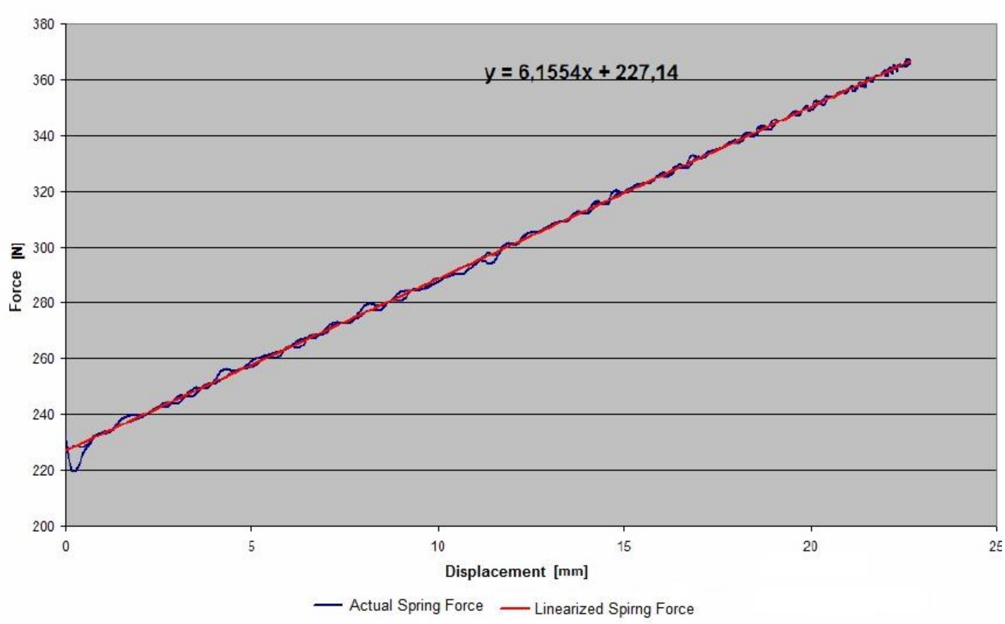


**Figure 2.14 :** Measurement of spring characteristics.

According to measurements performed on the spring, the stiffness coefficient can be taken as 6155 N/m. Although the spring is linear, assembly position of spring makes the spring reaction forces nonlinear. The following example shown in figure 2.15 demonstrates the nonlinearity of the reaction forces of the spring.

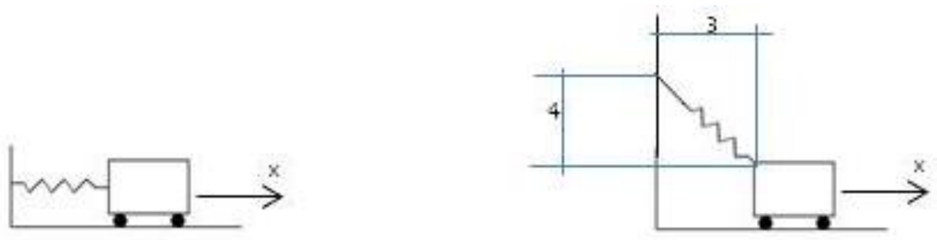


**Figure 2.15 :** Linear and angular attachment cases of springs.

Let's consider a spring attached to a slider, that slides frictionless and linear. In first situation, spring is attached in movement direction and in second situation spring is attached similar to our case. In this example consider spring stiffness a constant value: 5, and displacement x is from 0, up to 15 in order to show nonlinearity easily.

Forces on the x axis can be expressed as follows.

$$f_1 = kx \tag{2.6}$$

$$f_2 = [\sqrt{3^2 + 4^2} - \sqrt{4^2 + (3 + x)^2}]k \quad \textbf{(2.7)}$$

$$f_{2x} = f_2 \frac{3 + x}{\sqrt{4^2 + (3 + x)^2}} \quad \textbf{(2.8)}$$

The comparison of forces $f_1$ and $f_{2x}$ is shown in figure 2.16.

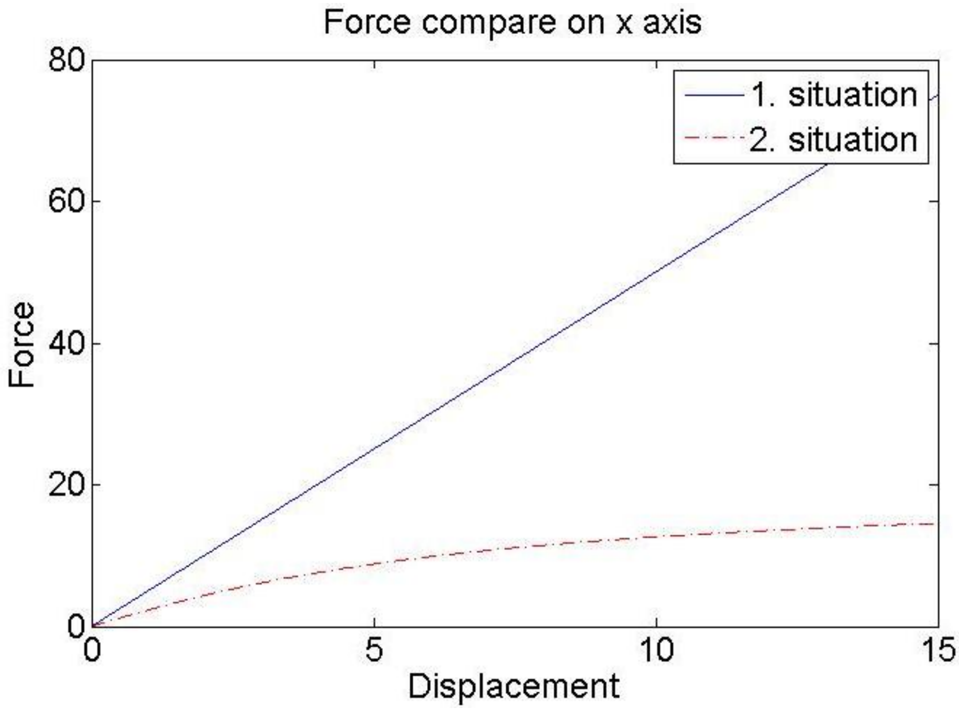


**Figure 2.16 :** Comparison of spring attachment cases.

This figure shows clearly that a linear spring with angular attachment to a slider has a nonlinear reaction force on a linear axis. As a conclusion to this demonstration, springs and shock absorber forces on x and z axes behave nonlinearly in our model.

### 2.3.2.1 Spring 1 model

Forces generated by spring 1 is simply calculated by taking dynamic length change of spring and multiplying it with spring constant. Following figures are for demonstrating the dimensions of spring 1 and calculation process. $xs1$ and $zs1$ are the distances between spring strut joint and the spring tub joint on x and z axis respectively. $x$ and $z$ are the displacements of the oscillation group, caused by centrifugal forces, $r$ is the vector showing displacement of oscillation group. Black lines show the initial position and blue lines and blue dotted lines shows dynamic position of elements in figure 2.17 and figure 2.18. We calculate dynamic distances and calculate dynamic length of spring, hence we obtain the dynamic force.

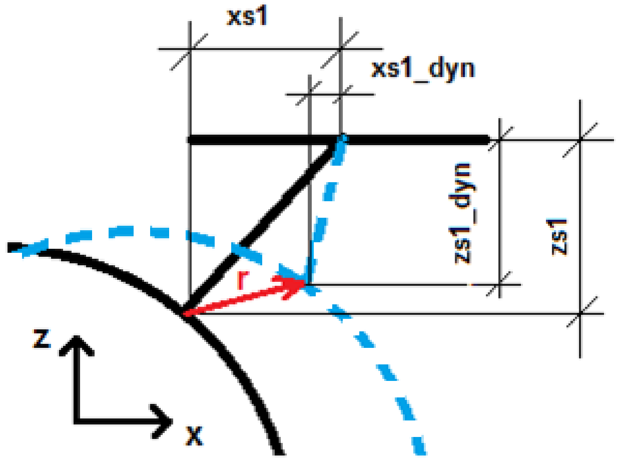


**Figure 2.17 :** Initial and dynamic positions of spring 1.

$$xs1_{dyn}(t) = xs1 - x(t) \tag{2.9}$$

$$zs1_{dyn}(t) = zs1 - z(\mathrm{t}) \tag{2.10}$$

$$l_{s1} = \sqrt{xs1^2 + zs1^2} \tag{2.11}$$

$$l_{s1\ dynamic}(t) = \sqrt{xs1\_dyn(t)^2 + zs1\_dyn(t)^2} \tag{2.12}$$

$$F_{s1}(t) = [l_{s1} - l_{s1\ dynamic}(t)]k \tag{2.13}$$

Equation (above) shows the calculation of force of spring 1 $F_{s1}$. Decomposing this force on x and z axes is in figure 2.18. Red arrows are reaction forces.

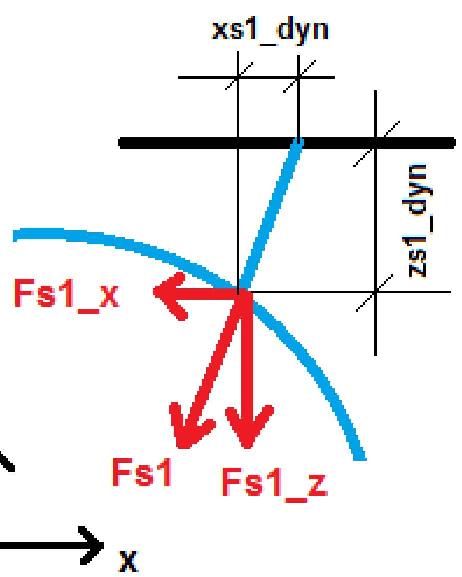


**Figure 2.18 :** Decomposition of dynamic spring 1 force onto x and z axes.

$$F_{s1_x}(t) = F_{s1}(t)\frac{\text{xs1\_dyn(t)}}{l_{s1\ dynamic}(t)} \tag{2.14}$$

$$F_{s1_z}(t) = F_{s1}(t)\frac{\text{zs1\_dyn(t)}}{l_{s1\ dynamic}(t)} \tag{2.15}$$

#### 2.3.2.2 Spring 2 model

Calculating the forces of spring 2 is similar to spring 1 and equations are as follows. Dynamic and static positions are shown in figure 2.19 and figure 2.20.

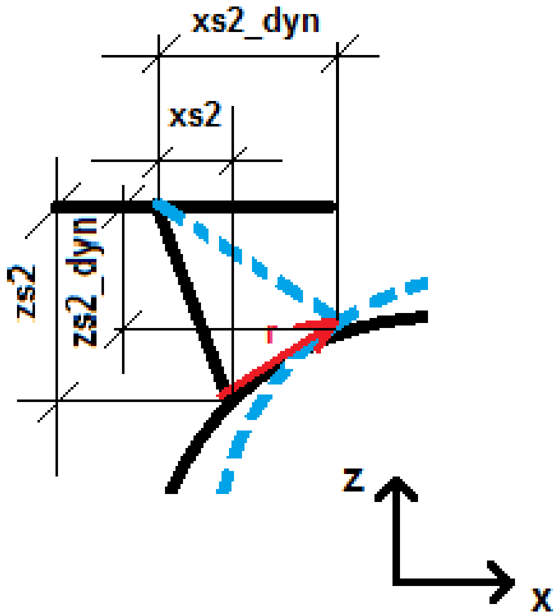


**Figure 2.19 :** Initial and dynamic positions of spring2.

$$xs2_{dyn}(t) = xs2 + x(t) \tag{2.16}$$

$$zs2_{dyn}(t) = zs2 - z(\text{t}) \tag{2.17}$$

$$l_{s2} = \sqrt{xs2^2 + zs2^2} \tag{2.18}$$

$$l_{s2\ dynamic}(t) = \sqrt{xs2\_dyn(t)^2 + zs2\_dyn(t)^2} \tag{2.19}$$

$$F_{s2}(t) = [l_{s2} - l_{s2\ dynamic}(t)]k \tag{2.20}$$

$$F_{s2_x}(t) = F_{s2}(t)\frac{\text{xs2\_dyn(t)}}{l_{s2\ dynamic}(t)} \tag{2.21}$$

$$F_{s2_z}(t) = F_{s2}(t)\frac{\text{zs2\_dyn(t)}}{l_{s2\ dynamic}(t)} \tag{2.22}$$

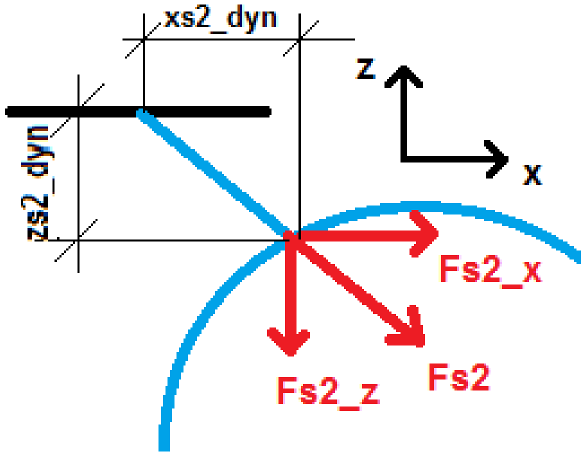


**Figure 2.20 :** Decomposition of spring 2 forces onto x and z axes.

### 2.3.3 Shock absorber model

Shock absorbers' damping force vs velocity plot can be seen in figure 2.21 that is obtained from experimental measurements in [5]. If we simply divide force to velocity we can find damping coefficient, which in this case is obviously not linear. The nonlinear damping coefficient is shown on figure 2.22 and force-displacement plot is shown on figure 2.21.

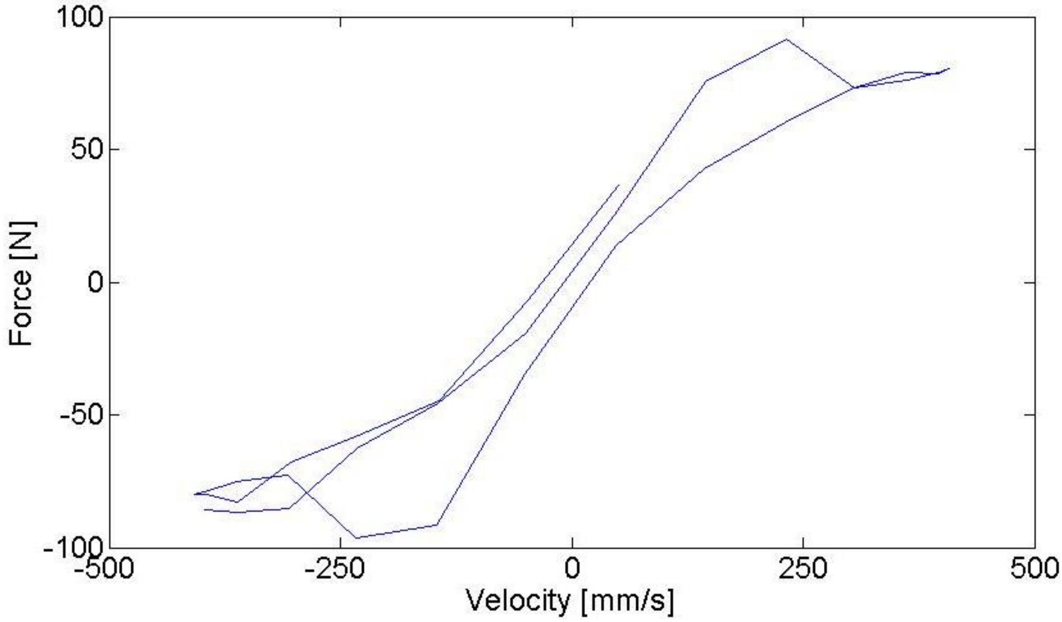


**Figure 2.21 :** Damping characteristics of shock absorbers.

We assumed that the constant damping coefficient of shock absorbers is mean value of the graphic shown in figure 2.22. This assumption yields with a damping coefficient 308,6 Ns/m.

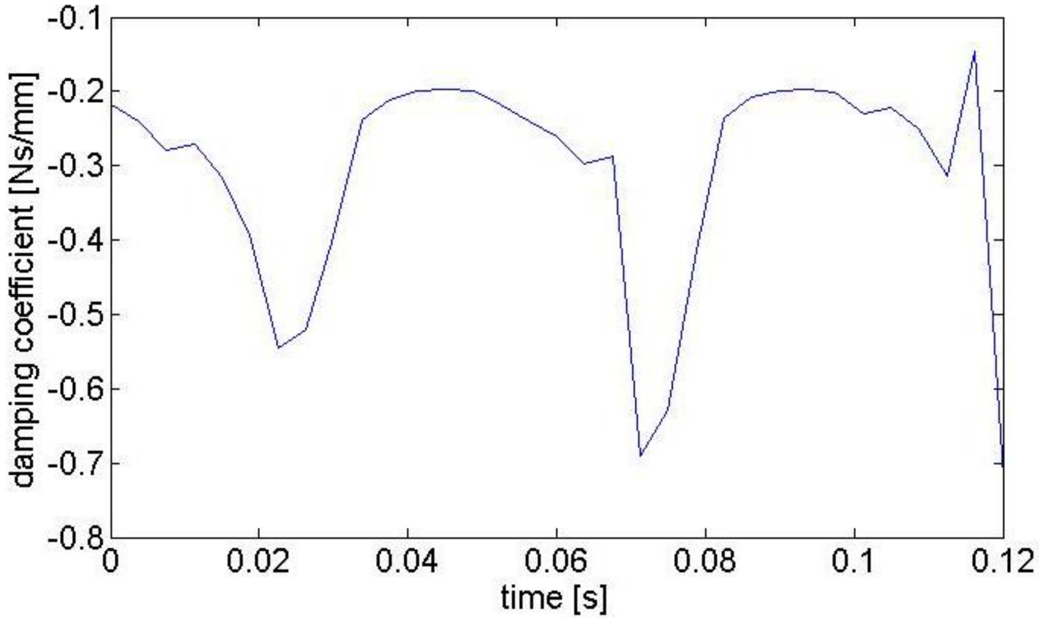


**Figure 2.22 :** Damping coefficient over time.

#### 2.3.3.1 Shock absorber 1 model

We use the same technique as spring force calculation. We calculate first the dynamic length of shock absorber then we find the derivative of dynamic length which gives us the velocity at the same direction of shock absorber motion. Next, multiplying velocity with damping coefficient, we obtain shock absorber forces. Dynamic and static positions of shock absorber 1 is shown in figure 2.23 and forces are shown in figure 2.24.

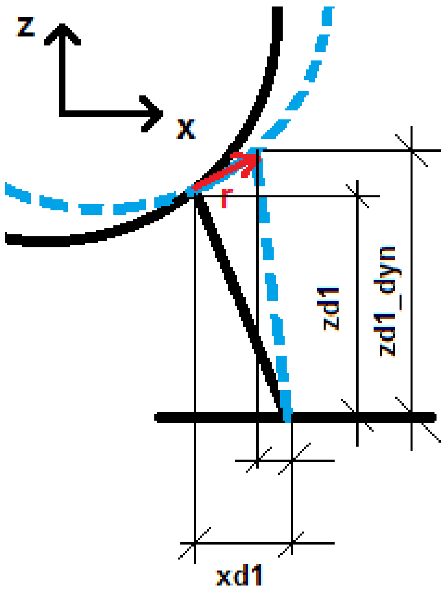


**Figure 2.23 :** Initial and dynamic position of shock absorber 1.

$$xd1_{dyn}(t) = xd1 - x(t) \tag{2.23}$$

$$zd1_{dyn}(t) = zd1 + z(\mathrm{t}) \tag{2.24}$$

$$l_{d1} = \sqrt{xd1^2 + zd1^2} \tag{2.25}$$

$$l_{d1\ dynamic}(t) = \sqrt{xd1\_dyn(t)^2 + zd1\_dyn(t)^2} \tag{2.26}$$

$$v_1 = \dot{l}_{d1\ dynamic}(t) \tag{2.27}$$

$$F_{d1}(t) = v_1 c \tag{2.28}$$

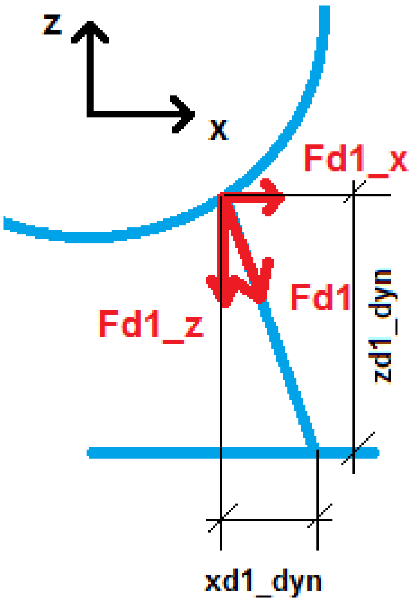


**Figure 2.24 :** Decomposition of shock absorber 1 forces onto x and z axes.

$$F_{d1_x}(t) = F_{d1}(t)\frac{\mathrm{xd1\_dyn(t)}}{l_{d1\ dynamic}(t)} \tag{2.29}$$

$$F_{d1_z}(t) = F_{d1}(t)\frac{\mathrm{zd1\_dyn(t)}}{l_{d1\ dynamic}(t)} \tag{2.30}$$

**2.3.3.2 Shock absorber 2 model**

Same technique is used to calculate shock absorber 2 forces as shock absorber 1. Equations are as follows. Dynamic and static positions and forces are shown in figures 2.25 and 2.26.

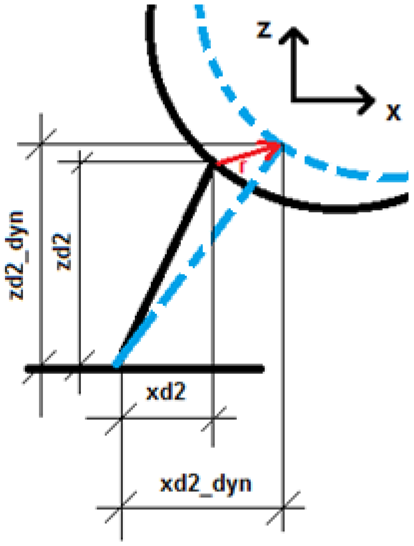


**Figure 2.25 :** Initial and dynamic positions of shock absorber 2.

$$xd2_{dyn}(t) = xd2 + x(t) \tag{2.31}$$

$$zd2_{dyn}(t) = zd2 + z(\mathrm{t}) \tag{2.32}$$

$$l_{d2} = \sqrt{xd2^2 + zd2^2} \tag{2.33}$$

$$l_{d2\ dynamic}(t) = \sqrt{xd2\_dyn(t)^2 + zd2\_dyn(t)^2} \tag{2.34}$$

$$v_2 = l_{d2\ dyna\dot{m}ic}(t) \tag{2.35}$$

$$F_{d2}(t) = v_2 c \tag{2.36}$$

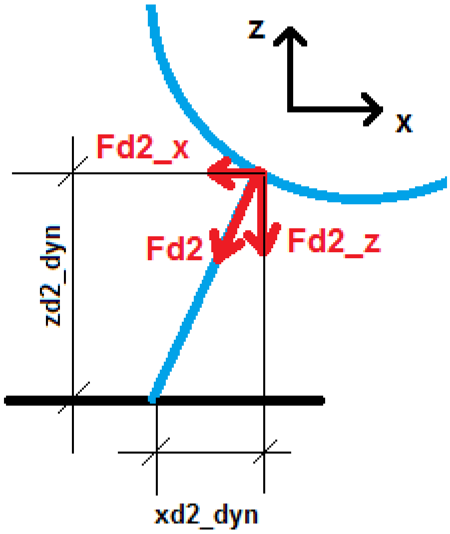


**Figure 2.26 :** Decomposition of shock absorber 2 forces onto x and z axes.

$$F_{d2_x}(t) = F_{d2}(t)\frac{\text{xd2\_dyn(t)}}{l_{d2\ dynamic}(t)} \quad \textbf{(2.37)}$$

$$F_{d2_z}(t) = F_{d2}(t)\frac{\text{zd2\_dyn(t)}}{l_{d2\ dynamic}(t)} \quad \textbf{(2.38)}$$

#### 2.3.3.3 Complete model

We derived equations belonging to all elements which affect the motion such as centrifugal forces of unbalanced mass, spring forces and shock absorbers damping forces. According to Newton's II. Law, we summed up all the forces and obtained force balance on each axis using D'Alembert Principle.

$$m\ddot{x} + F_{s1_x} - F_{s2_x} - F_{d1_x} + F_{d2_x} - F_{x\ ext} = 0 \quad \textbf{(2.39)}$$

$$m\ddot{z} + F_{s1_z} + F_{s2_z} + F_{d1_z} + F_{d2_z} - F_{z\ ext} = 0 \quad \textbf{(2.40)}$$

One can notice that the sign of $F_{s2_x}$ and $F_{d1_x}$ is negative whereas all the rest of the forces has a positive sign. The reason for this situation can be seen on figure 2.10 and 2.24. The directions of these forces are represented in local coordinates and they are opposite to our chosen coordinate system. Hence, they had to be corrected. If we re-arrange these equations and write them in matrix format and leave the acceleration alone for simulation, we have following matrices.

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{z} \end{bmatrix} = \begin{bmatrix} F_{x\ ext} - F_{s1_x} + F_{s2_x} + F_{d1_x} - F_{d2_x} \\ F_{z\ ext} - F_{s1_z} - F_{s2_z} - F_{d1_z} - F_{d2_z} \end{bmatrix} \quad \textbf{(2.41)}$$

$$\begin{bmatrix} \ddot{x} \\ \ddot{z} \end{bmatrix} = {}^{1}/_{m}\begin{bmatrix} F_{x\ ext} - F_{s1_x} + F_{s2_x} + F_{d1_x} - F_{d2_x} \\ F_{z\ ext} - F_{s1_z} - F_{s2_z} - F_{d1_z} - F_{d2_z} \end{bmatrix} \quad \textbf{(2.42)}$$

#### 2.3.3.4 Simulink representation of the model

In order to solve this mathematical model, stated earlier, model is numerically implemented in the Matlab Simulink Environment. In Simulink diagram, main Matlab function (function block in the middle, see figure 2.27) is the function contains the equations of motion mentioned earlier on this chapter. Constant matrix is containing design variables, which are used in optimization. The output of the Matlab function is accelerations of x and z axes. With one integration, the velocity and with double integration the translations are obtained. Translations obtained are fed back in order to be used in calculation in main Matlab function. Also, lengths of shock absorbers, horizontal and vertical forces are outputs of main function. Lengths of shock absorbers are then sent to derivative blocks to obtain velocity of shock absorbers and they are fed back to main function to calculate shock absorber forces. Horizontal and vertical forces generated in the washing machine, which are the output of main Matlab function, are used in optimization as a restriction, which will be mentioned later on.

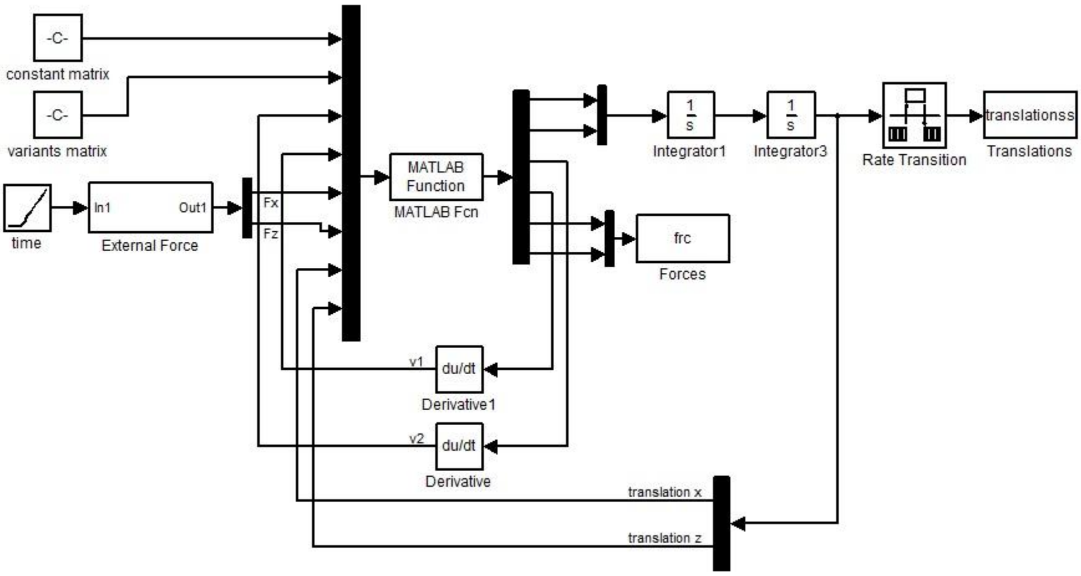


**Figure 2.27 :** Simulink diagram of mathematical model.

In main diagram a subsystem called "external force" is used to generate external forces caused by unbalanced mass. Following figure shows subsystems details. Output of ramp-up function is the angular velocity of drum, in *ext_force* function we need all acceleration and rotation of drum as well as velocity. Hence, an integrator and derivative block is employed to obtain necessary variables. "ext_force" function's output is external forces on x and z axes. External force sub-system is shown in figure 2.28.

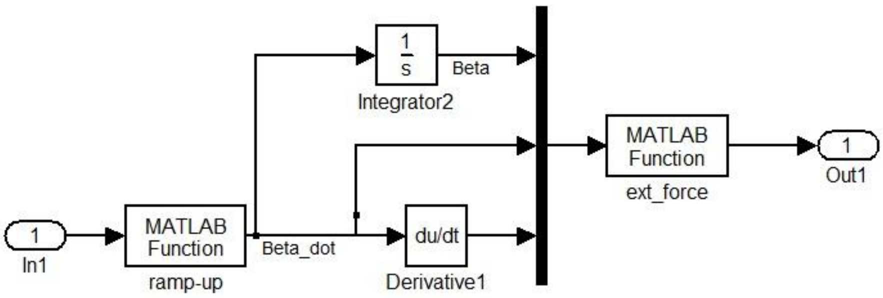


**Figure 2.28 :** External force sub-system diagram.

# 3. VIBRATION MEASUREMENT OF WASHING MACHINE

In order to validate the dynamic simulation generated, output of the real systems should be known. Vibration amplitudes and characteristic of a horizontal load washing machine is measured for validation purposes and this chapter is explaining the measurement setup and measurement methodology.

## 3.1 Measurement Setup

For exciting the washing machine, sensing the vibration and recording the vibration data require some hardware, which are individual professional tools in the related area. These measurement equipments include variable autotransformers, accelerometers and a data acquisition card, which are summarized in the following section.

**Variable Autotransformers:** First of all, for excitation of washing machine, input signal of electric motor has to be controlled. For this purpose a variable autotransformer is used. Such devices are often called as *variac* after name of a trademark. This device is made by exposing part of the winding coils and making the secondary connection through a sliding brush which enables us to have a variable turns ratio $N$ (figure 3.1). Output of variable autotransformers is connected directly to input of the electric motor. With the ability of autotransformer input voltage is adjusted between 0-250V. Hence, spin speed of washing machine is adjusted manually by this way.

**Figure 3.1 :** Variable autotransformer.

**Accelerometer:** For the measurements of washing machine vibration, single axis capacitive accelerometers are used. Capacitive accelerometers are typically made from a silicon micro-machined sensing element which is basically two plates placed face to face and with the acceleration, capacitance between them changes. Hence, acceleration is measured by measuring the capacitance. This type of accelerometers performance is superior in the low frequency and even static acceleration [12]. Sensor we used has a frequency range between 0-350 Hz [13]. Sensors used in measurements can only measure the acceleration on a single axis. Hence, a simple part is used to attach 3 accelerometers with 90 degrees to each other, which is shown in figure 3.2.

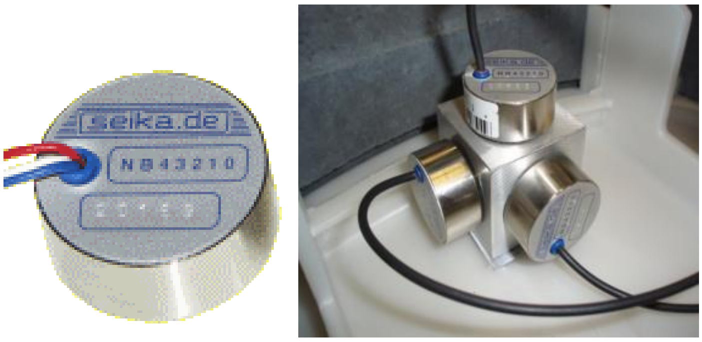


**Figure 3.2 :** Accelerometers used in measurements.

**Data Acquisition Card**: In order to digitize the analog signal coming from the sensors and process it in the computer we need a data acquisition card. During the measurement, a multi channel logger is used, named as a trademark “spider8”. This logger enables 8 channels of measurements simultaneously. Data acquisition card used in experiments showed in figure 3.3.


**Figure 3.3 :** Data acquisition card.

Measurement setup built for experiments can be explained better with the figure 3.4. Voltage coming from variable autotransformer excites the electric motor and the motor accelerates the drum. To simulate the uneven distribution of laundry, an unbalanced mass is fixed on the drum. This way the vibration occurs on the oscillation group just like the real case of sticking laundry on the inner surface of the tub. The accelerometer fixed on the oscillation group gives analog output to data acquisition card. Through the data acquisition card, acceleration signal reaches to computer in terms of $m/s^2$. With appropriate signal processing, acceleration data is converted to displacement data. The technique used for this conversion will be described later.


**Figure 3.4 :** Measurement setup flow.

### 3.2 Signal Processing Flow

For the purpose of obtaining displacement data from acceleration data, operation flow in figure 3.5 is used. First, a low pass filter is implemented then a high pass filter and double integration with a high pass filter between integrations are done. Finally, displacement data is obtained. Selection and properties of filters and method of integration will be explained in following sections.

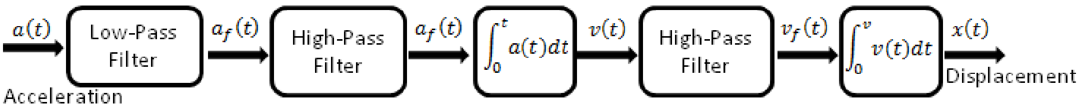


**Figure 3.5 :** Signal processing flow.

Although the hardware has capacity to measure up to 350 Hz frequency, vibrations generated in the system is around 15 Hz. According to Nyquist's sampling theorem; if the highest frequency component in a signal is $f_{max}$, than the signal must be sampled at the rate of at least $2f_{max}$ to describe the signal completely.

$$F_s \geq 2f_{max} \tag{3.1}$$

During the measurements, we stick to this rule and selected sampling frequency as 300 Hz which is high enough to prevent aliasing and to describe the signal properly.

### 3.3 Filter Selection

A filter is essentially a system or a network that selectively changes the wave shape, amplitude-frequency and/or phase-frequency characteristics of a signal in a desired manner. Common filtering objectives are to improve the quality of a signal, reducing or eliminating the noise or to separate two signals combined with each other previously.

A digital filter is a mathematical algorithm implemented in a hardware or software that operates on a digital input to produce digital output signal for achieving filter objectives. Digital filters operate on digitized analog signals or digital numbers stored on computers. The output of the filtering operation is digital as well [14]. Digital filters has some advantages and disadvantages over the analog filters, however, in this thesis all the signals and consequently all the filters will be digital hence comparison of digital and analog filter will not be mentioned.

Filters are broadly divided into two classes, namely infinite impulse response (IIR) and finite impulse response (FIR) filters. Either type of filter, in its basic form, can be expressed by its impulse response sequence, h(k) as shown in figure 3.6.

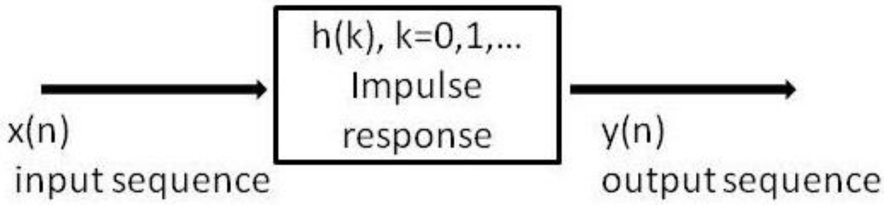


**Figure 3.6 :** Schematic digital filter.

The input and the output are related by the convolution sum, which is given in equations 3.2 for IIR and 3.3 for FIR filters.

$$y(n) = \sum_{k=0}^{\infty} h(k)x(n-k) \qquad \textbf{(3.2)}$$

$$y(n) = \sum_{k=0}^{N-1} h(k)x(n-k) \qquad \textbf{(3.3)}$$

It is evident from these equations that, for IIR filters, impulse response is of infinite duration whereas for FIR its duration is finite. In practice, it is not feasible to compute the output of IIR filter because length of impulse response is infinite. Instead of this, we can write IIR filter equation in a recursive form.

$$y(n) = \sum_{k=0}^{\infty} h(k)x(n-k) = \sum_{k=0}^{N} a_k x(n-k) - \sum_{k=1}^{M} b_k y(n-k) \qquad \textbf{(3.4)}$$

Here $a_k$ and $b_k$ are filter coefficients. Equation 3.3 and 3.4 are the difference equations for FIR and IIR respectively.  Its an important task to find the coefficients and $h(k)$ in filter design problem [14]. In the following equations transfer functions of FIR and IIR filters are shown respectively. These equations are very useful in evaluating their frequency responses.

$$H(z) = \sum_{k=0}^{N-1} h(k)z^{-k} \qquad \textbf{(3.5)}$$

$$H(z) = \frac{\sum_{k=0}^{N} a_k z^{-k}}{(1 + \sum_{k=1}^{M} b_k z^{-k})} \qquad \textbf{(3.6)}$$

The choice between FIR and IIR filters depends highly on application. Main advantage of these filters is, for FIR filter has an exact linear phase response. There is no phase distortion of the output signal compared to input signal. This property is very important in many engineering applications such as transmission, biomedicine, audio and imaging. Phase response of IIR filters is nonlinear, especially at the band edges. Another important criterion for selecting filter type is sharpness of cutoff frequency. If for an application a sharp cut off frequency is needed, selection of IIR filter will be a wise choice. The reason is that a FIR filter needs higher orders for sharp cutoff frequency, whereas IIR filters can perform better with very low orders compared to FIR filters. For real time applications, IIR filters are selected because of the lower orders, hence shorter processing times.

In our application, all signals are digital and filtering will be performed off line. Sharp cut off frequencies are required in this application, therefore, IIR filters are selected. There is one more problem remains to deal with. IIR filters have the advantage of sharp cut-off frequency at lower order , however they introduce phase distortion in output signals. If an acceleration data with a phase distortion integrated, results will be inaccurate, therefore it needs to be taken care of. In order to eliminate the phase distortion, after filtering the data, time signal is reversed and filtered again. On the second time, filtering will correct the phase distortion and we will have an exact linear phase response. However, now the magnitude of the filter has changed and order of filter is effectively doubled [15]. For impelenting this technique, in MATLAB there is a command "filtfilt" that performs this filtering operation effectively [24].

### 3.3.1 IIR filters

The equations of IIR filters are given in equation 3.2. For a standart frequency selective filtering task, filter can be derived from the classical Butterworth, Chebyshev, or elliptic functions. Butterworth filter is characterized by the following magnitude squared frequency response.

$$|H(w')|^2 = \frac{1}{1 + (w'/w_p')^{2N}} \quad \textbf{(3.7)}$$

Where N is the order of the filter and $w_p'$ is the 3dB cutoff frequency. Low pass butterworth filter is monotonic in both the passband and the stopband.

There are two types of Chebyshev filters. Type I, with equal ripple in the passband and monotonic in the stopband. Type II, with equal ripple in the stopband and monotonic in the passband. Elliptic filter exhibits *equiripple* behavior in both the passband and the stop band. The figure 3.7 shows a comparison of gains of Butterworth, Chebyshev, and elliptic filters with fifth order. The Butterworth filter is a type of signal processing filter designed to have as flat a frequency response as possible in the passband so that it is also termed a maximally flat magnitude filter. Compared with a Chebyshev Type I/Type II filter or an elliptic filter, the Butterworth filter has a slower roll-off, and thus will require a higher order to implement a particular stopband specification, but Butterworth filters have a more linear phase response in the pass-band than Chebyshev Type I/Type II and elliptic filters can achieve. For the reasons and advantages mentioned, Butterworth filter is selected for our application.

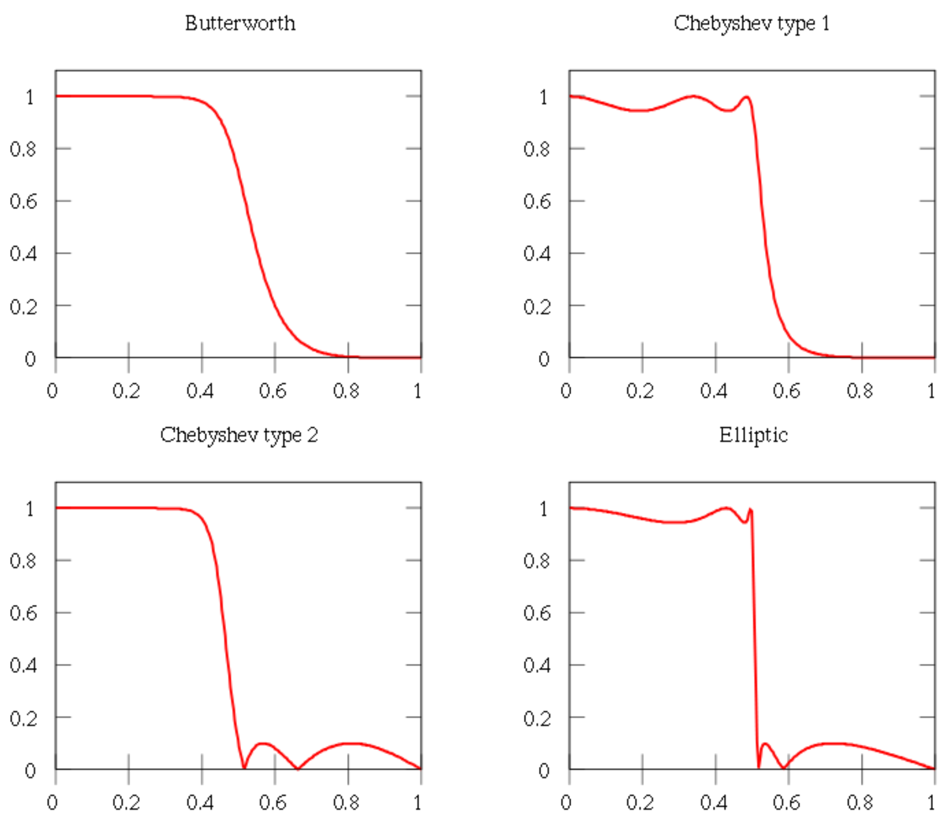


**Figure 3.7 :** Comparison of filters [16].

### 3.4 Integration Method

From a position $x(t)$, the velocity $v(t)$ can be obtained by taking the first derivative. Next, the acceleration $a(t)$ can be obtained by taking derivative of velocity.

$$a(t) = \frac{d^2x}{dt^2} \tag{3.8}$$

However, we are interested in the reverse of this process for our application. From measurements, we have acceleration data and we need to obtain position data. Therefore, a double integration has to be performed.

$$v(t) = v(t_0) + \int_{t_0}^{t} a(t)dt \tag{3.9}$$

$$x(t) = x(t_0) + \int_{t_0}^{t} v(t)dt \qquad \textbf{(3.10)}$$

The acceleration data is in digital form hence we need to implement one of numerical integration methods. Basic numerical integration methods are rectangular method and trapezoidal method. Integration is simply performed by computing the area under the curve. The simplest approximation of the area under the curve is the sum of rectangles shown in Figure 3.8 In the rectangular approximation, function is evaluated at the beginning of the interval, and the approximate Fn of the integral is given by,

$$F_n = \sum_{i=0}^{n-1} f(x_i)\Delta x \qquad \textbf{(3.11)}$$

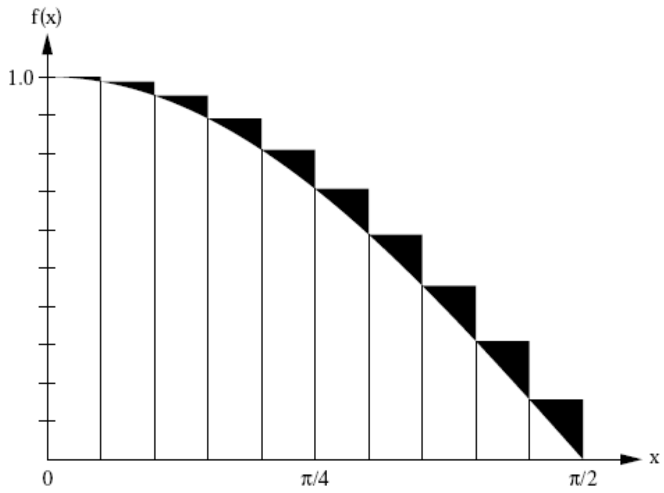


**Figure 3.8 :** Rectangular integration.

Function here is $f(x) = cosx$ for $0 < x < \pi/2$ and the error in rectangular approximation is shaded in the figure. Also in this figure, it can be seen that as interval of rectangles decreases the approximation error decreases, too.

On the other hand, according to trapezoidal approximation rule the integral is approximated by computing the area under a trapezoid with one side equal to function value at the beginning of the interval and the other side is equal to function value at the end of the interval, given by formula in equation **(3.12)** [17].

$$F_n = \left[\frac{1}{2}f(x_0) + \sum_{i=1}^{n-1} f(x_i) + \frac{1}{2}f(x_n)\right]\Delta x \qquad \textbf{(3.12)}$$

Trapezoidal integral acts like a first order hold on the system, whereas rectangular integration acts like a zero order hold [15]. A comparison of trapezoidal and rectangular is shown on a 1Hz sinusoidal signal in the figure 3.9.

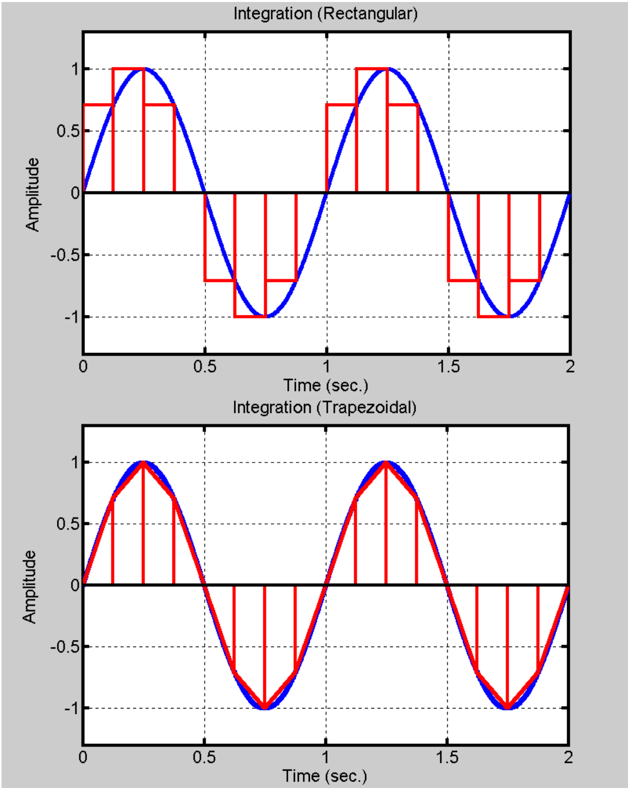


**Figure 3.9 :** Rectangular and trapezoidal integration.

There are several other methods for numeric integral but integration will be done in MATLAB environment. MATLAB has its own commands for integration and it uses trapezoidal rule. The accuracy of trapezoidal rule is sufficient for our signals and sampling interval is considerably small to minimize errors. As a conclusion, trapezoidal method is used for numerical integrations.

### 3.5 Implementation of Filters and Integration

At the beginning of signal processing, raw acceleration signal is first filtered with a low pass filter in order to get rid of the high frequency noise in the signal. As mentioned before, sampling frequency of this acceleration signal is 300 Hz and any frequency component above 20 Hz is irrelevant and needed to be filtered, hence first filter is a fifth order butterworth with 20 Hz cutoff frequency.

Accelerometers used in measurements are capacitive type and they have a DC component even if they are not exposed to any acceleration. In figure 3.10, raw acceleration data, is shown. It's obvious that acceleration can not be oscillating around any other term than zero and DC component has to be removed. In order to remove DC component a highpass filter with a very low cutoff frequency is implemented secondly on the signal. For second filter a fifth order highpass Butterworth filter with a cutoff frequency 1,5 Hz is implemented. Harmonics with 1,5 Hz in the signal will be removed with this operation but the frequency range of spinning of drum is not in the region of interest here. As explained earlier, laundry in drum does not stick to inner walls under that frequency which is corresponding to 90 rpm spin speed of drum.

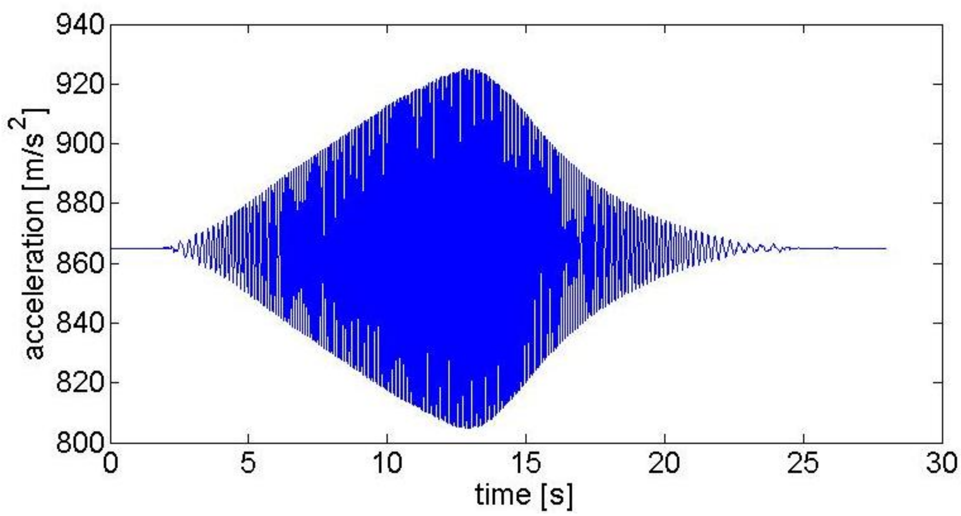


**Figure 3.10 :** Raw acceleration data.

Following figure shows the signal after filtering out the DC component. As can be seen clearly from Fig 3.11, mean value of the signal is settled to 0 and DC term is eliminated. Next, as in figure 3.5, integration process starts here. After integration, a DC component will arise in velocity data. If the acceleration signal had integer number of periods , thenthis DC component may not occur but we cannot be sure and DC component may occur. Hence, a high pass filter with a very low frequency cutoff is implemented again.

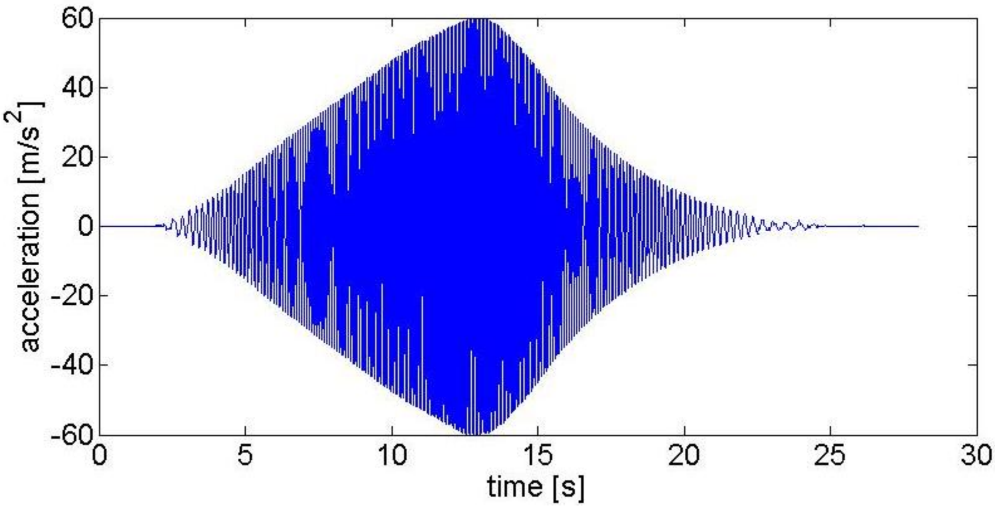


**Figure 3.11 :** Filtered acceleration data.

If the filtering here is not performed, then there will be an initial value problem introduced in the integration hence position errors will occur at the output of integral. In order to visualize the mentioned problem, velocity data is integrated without filtering and with filtering and results are shown respectively in Fig 3.12 and Fig. 3.13. Figures clearly show integration errors without filtering.

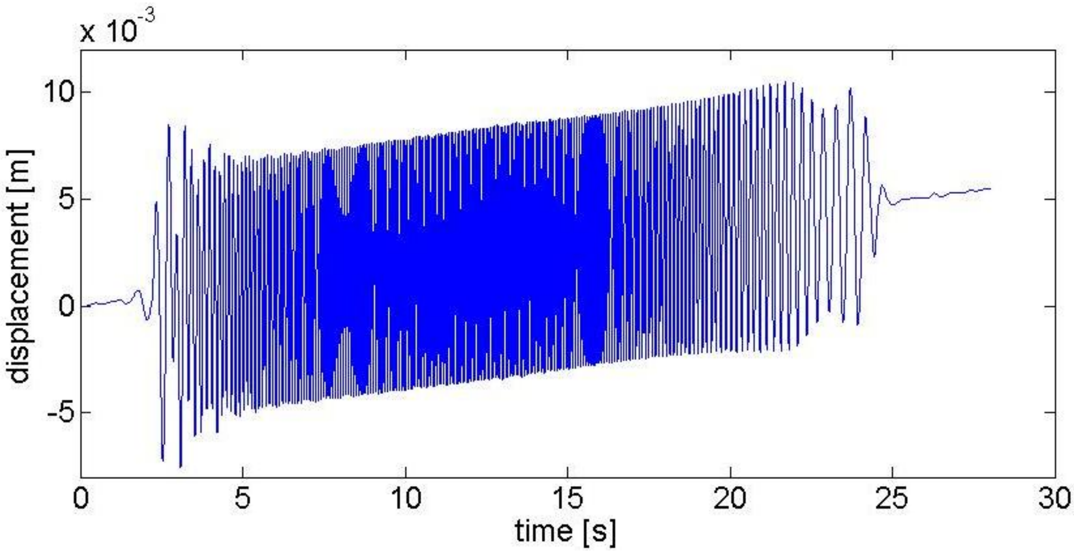


**Figure 3.12 :** Displacement calculated without high pass filtering.

In figure 3.12 effect of DC components is obvious and it causes a rise in the displacement data. Integration performed after filtering has normal results in position data. As expected, position oscillates around zero. Filtering application is not applied after the second integration because there is no operation after obtaining position data.

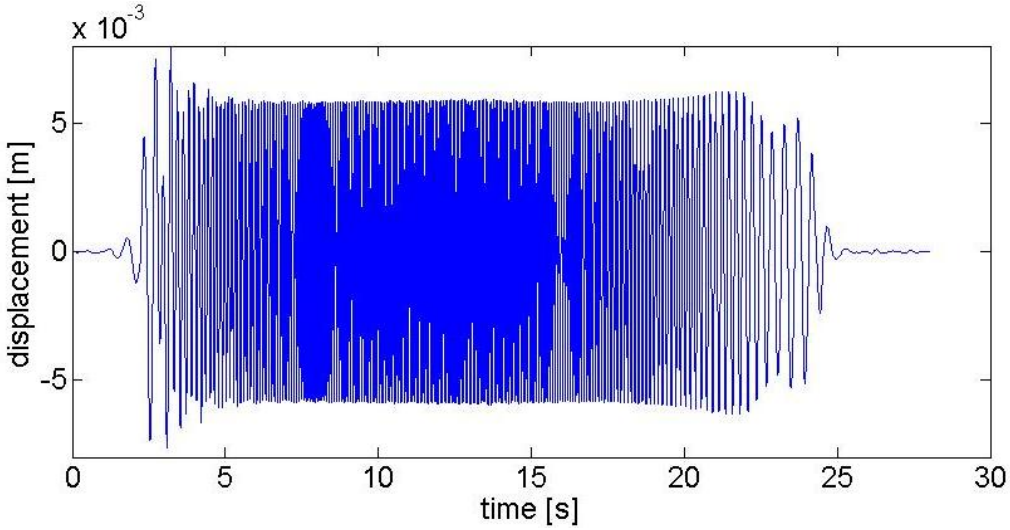


**Figure 3.13 :** Displacement calculated with high pass filtering.

### 3.6 Instantaneous Frequency Estimation

As mentioned before, input signal to electric motor is supplied from a variable autotransformer in order to adjust the spin speed of drum manually. Spin speed of drum is not always possible to measure directly with an encoder because of shaft accessibility problems. Hence, there is no spin speed data of drum. To obtain such data, an instantaneous frequency estimation is performed using the acceleration data.There are various methods for estimating the instantaneous frequency, and a review of them is given in the following section.

Y.Christos et al studied harmonic signal decomposition to estimate instantaneous frequency of rotating machineries. They used vibrations signals of a pump in the ramp-up and ramp-down periods and estimated revolution speeds, which differs considerably. Details of their study can be found in [18] A.Akan et al proposed an iterative method to estimate instantaneous frequency of a signal based on discrete evolutionary transform [19]. L.Angrisani and M.D'Arco used Chirplet transform for estimating frequency [20]. T.Matsuoka et al proposed a new method for estimating fundamental frequency of speech signals with a low pass filter and Hilbert transform [21]. The method they proposed is quiet simple considering the others and easy to implement in MATLAB. Hence, it is used to estimate the spin speed of washing machine drum. This method simply uses the Hilbert transform and converts signal to analytic form, finds phase angle and taking derivative of phase angle yields the instantaneous frequency.

### 3.6.1 Hilbert transform

In 1743 famous Swiss mathematician Leonard Euler derived the following equation.

$$\mathrm{e}^{\mathrm{j}\mathrm{x}} = \cos(\mathrm{x}) + \mathrm{j}\sin(\mathrm{x}) \tag{3.13}$$

150 year later, Arthur E. KEnnelly and Charles P. Steinmetz used this equation to introduce the complex notation of harmonic waveforms in electrical engineering, that is,

$$\mathrm{e}^{\mathrm{jwt}} = \cos(\mathrm{wt}) + \mathrm{j}\sin(\mathrm{wt}) \tag{3.14}$$

Later on German scientist David Hilbert showed that $\sin(\mathrm{wt})$ is the Hilbert transform of $\cos(\mathrm{wt})$. This gives us the $\pm\pi/2$ phase shift operator, which is the basic property of Hilbert transform [22].

$H[g(t)]$ the Hilbert transform of $g(t)$ is defined as,

$$H[g(t)] = g(t) * \frac{1}{\pi t} = \frac{1}{\pi}\int_{-\infty}^{\infty} \frac{g(\tau)}{t-\tau} d\tau = \frac{1}{\pi}\int_{-\infty}^{\infty} \frac{g(t-\tau)}{\tau} d\tau \tag{3.15}$$

One can easily notice that integral in equation 3.12 is improper but with using Cauchy principle, it becomes;

$$H[g(t)] = \frac{1}{\pi}\lim_{\epsilon\to 0+} (\int_{t-1/\epsilon}^{t-\epsilon} \frac{g(\tau)}{t-\tau} d\tau + \int_{t+\epsilon}^{t+1/\epsilon} \frac{g(\tau)}{t-\tau} d\tau) \tag{3.16}$$

One of the basic properties of the Hilbert transform is the output of the transform is time domain signal as well as the input. If $g(t)$ is real valued then so is its Hilbert transform $\hat{g}(t)$ [23].

Hilbert transform converts a real signal sequence to its complex sequence with same length. The analytic signal is useful in calculating instantaneous attributes of a time series, at any point in time. The instantaneous amplitude of input sequence is the amplitude of the analytic signal. The instantaneous phase angle of the input sequence is the angle of analytic signal and the instantaneous frequency is the time rate change of the instantaneous phase angle [24].

### 3.6.2 Implementation of instantaneous frequency estimation method

We get the raw acceleration signal and filter it with a high pass filter with a very low cutoff frequency in order to eliminate the DC component in the signal first just like the process in ontaining the displacement from acceleration. Next, we take Hilbert transform of the signal and obtain analytical form of it and find the phase angle of the signal. The phase angle plot is shown in the figure 3.14.

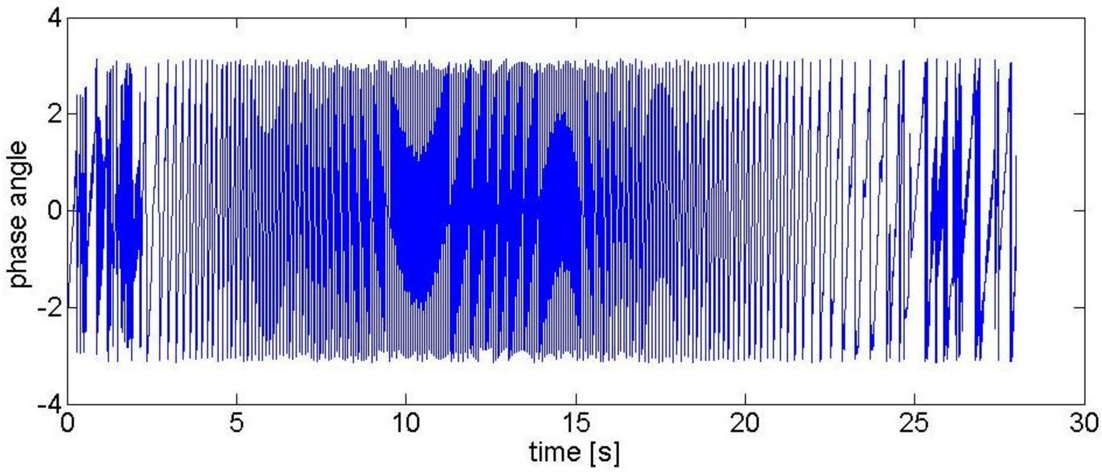


**Figure 3.14 :** Phase angle of the signal.

The phase angle is oscillating between -3.14 and 3.14 meaning that $[-\pi, +\pi]$. If we unroll the phase angle and get it out from $[-\pi, +\pi]$ interval we get the following plot in fig 3.15. If we take the derivative of this angle with respect to time, we obtain the instantaneous frequency of the signal, shown in figure 3.16. The instantaneous frequency corresponds to spin speed of the drum in our case. In the measurement scenario to record this data, spin speed starts with 0 rpm, make a ramp-up and goes to 0 rpm at the end. Although we get the instantaneous frequency, we have some problems under the frequencies about 1.5 Hz and we have noise in the signal. For low frequencies, as mentioned earlier they are not in the scope of interest and we can just eliminate them, fit a linear rise, and fall to start and end of the signal respectively.

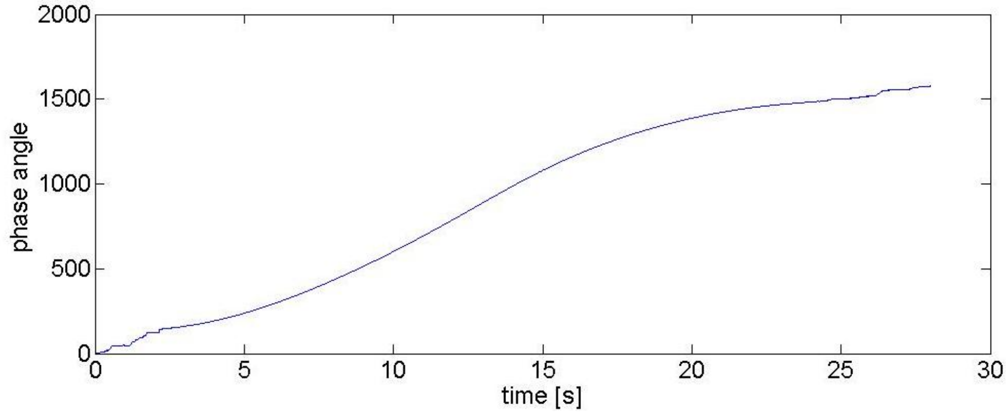


**Figure 3.15 :** Unwrapped phase angle of the signal.

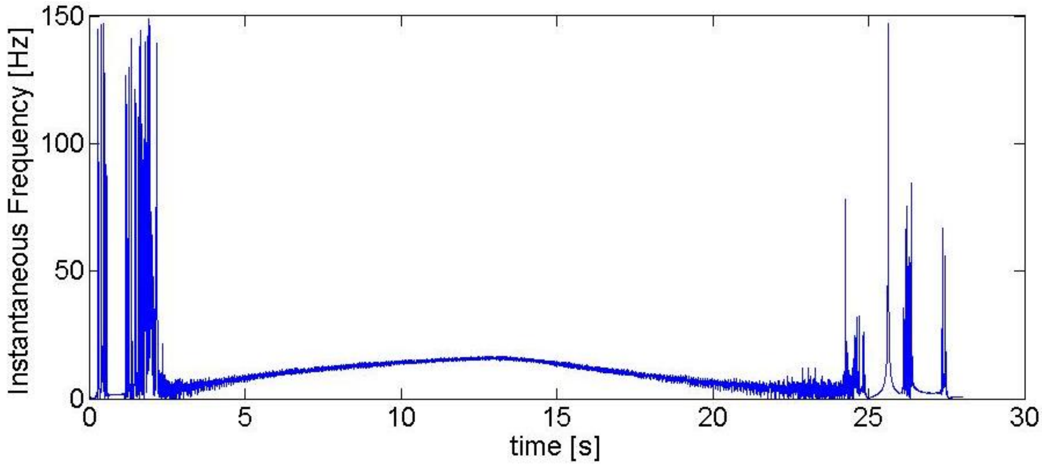


**Figure 3.16 :** Calculated instantaneous frequency.

Moreover, for the rest we need to perform some smoothing operation. After some smoothing, instantaneous frequency yields the following figure 3.17.

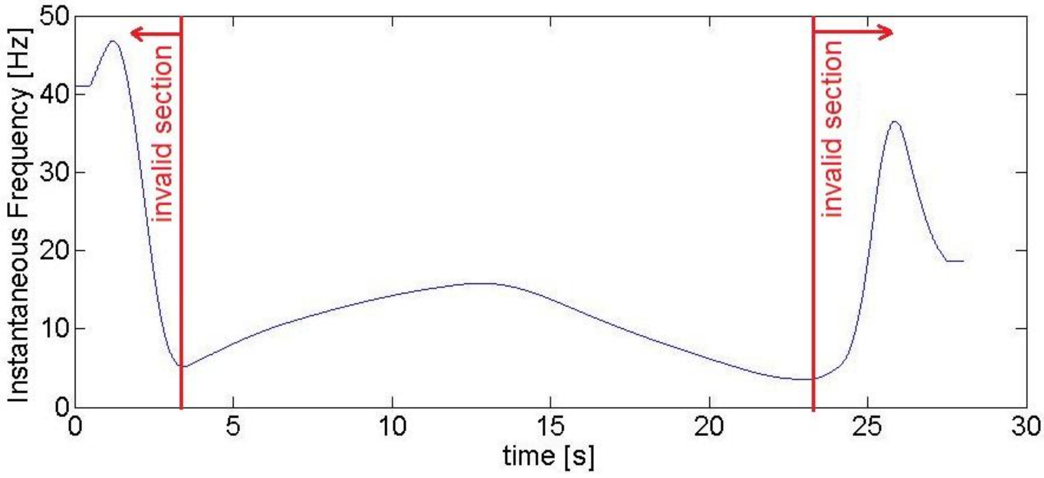


**Figure 3.17 :** Smoothed instantaneous frequency.

Figure 3.17 shows the instantaneous frequency of drum. If we just skip the invalid sections shown in figure 3.17 (start and end parts), we can see that frequency rises up to 15 Hz and falls again. This situation corresponds to 900 rpm spin speed of drum and falling to 0. Figure 3.18 shows the ramp up spin speed of drum in rpm. In addition, incorrect start up data is replaced with a linear rise and ramp down data is removed.

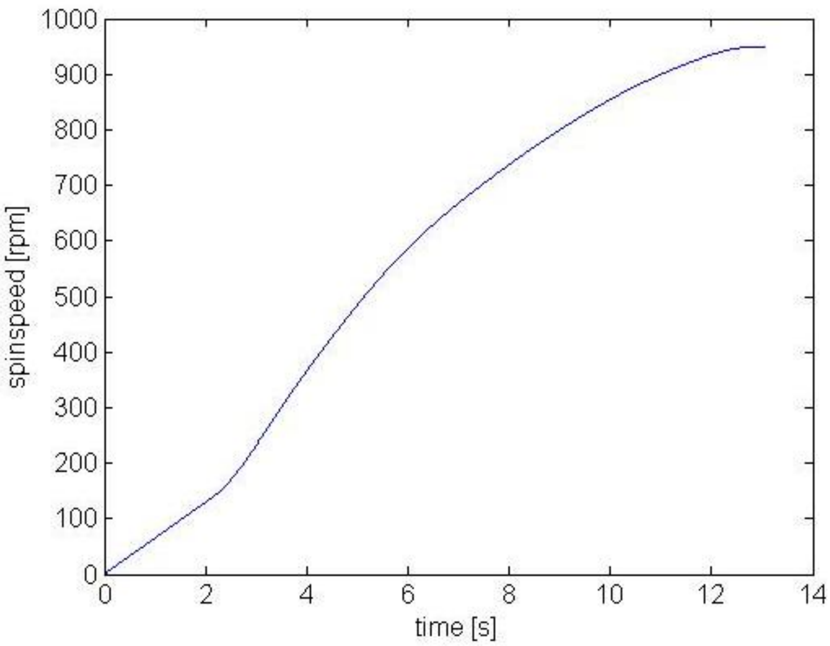


**Figure 3.18 :** Ramp up of spin speed.

Although the method is simple, we have to validate it with the measured spin speed data. For this validation, a measurement is done with both the encoder and the accelerometer attached on washing machine and the machine is run twice form 0 rpm to 600 rpm one after the other. Encoder data of the measurement is shown in figure 3.19. The figure 3.20 shows the spin speed that is estimated with the same technique mentioned above. If we plot the measurement data from the encoder and data from the instantaneous frequency estimation we get a very good match. At lower frequencies estimation method has some weakness and can not estimate lower frequencies correctly but the rest of the frequency range fits perfectly well with the encoder data and that validates the technique. Comparison of measurement and estimation is shown in figure 3.21. The frequency jumps on the beginnings and ends are not in the area of interest, hence this technique can be used to estimate spin speed of drum in the frequency range we are interested in [100-600 rpm].

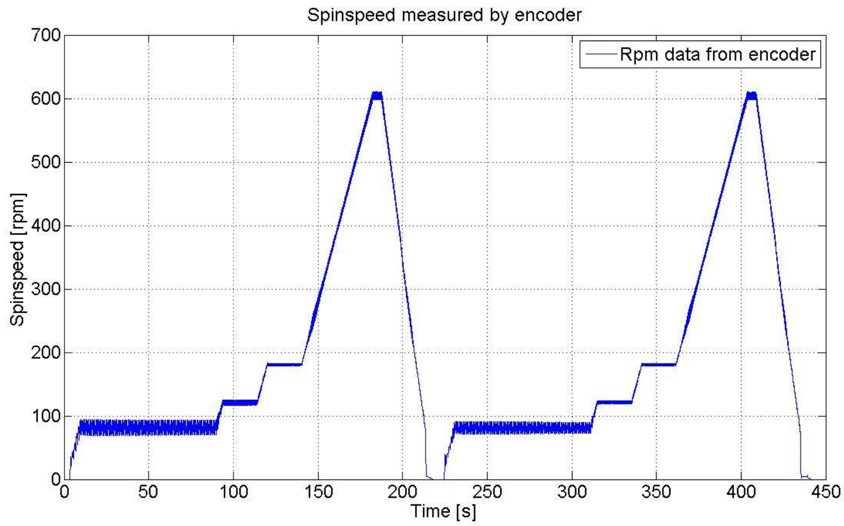


**Figure 3.19 :** Measured spin speed.

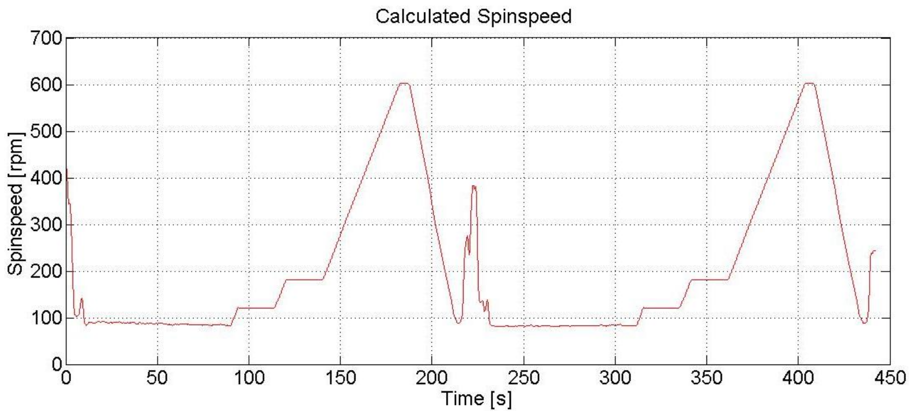


**Figure 3.20 :** Estimated spin speed.

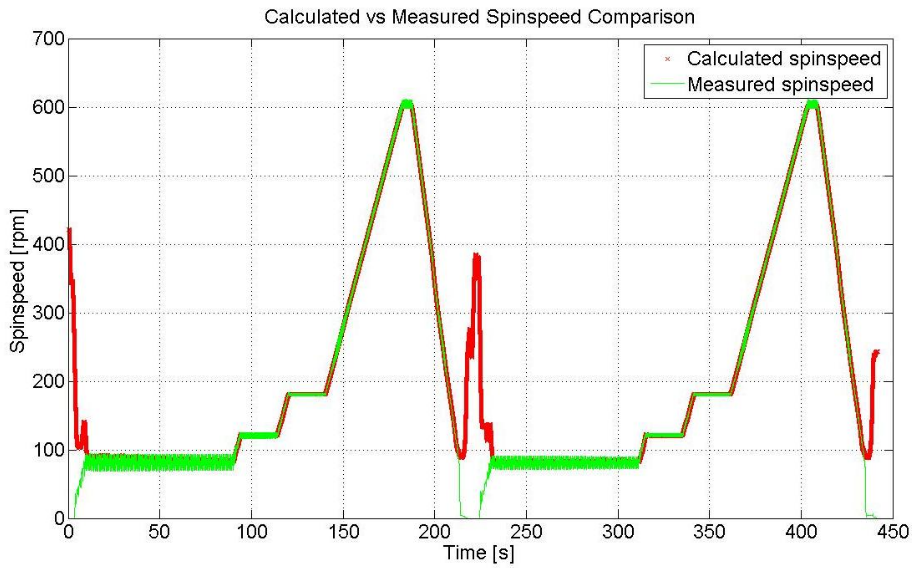


**Figure 3.21 :** Comparison of measurement and estimation.

### 3.6.3 Smoothing and Savitsy-Golay filtering

In order to reduce the noise in calculated instantaneous frequency, a smoothing algorithm called Savitsky-Golay filtering is used. The method of Savitsky-Golay filtering of digital smoothing polynomial filtering is a widely used technique in absorption spectroscopy that is applicable to the filtering and differentiation of reflectance spectra. In short, Savitsky-Golay filtering performs noise reduction while preserving higher order moments of the original spectrum. The preservation of these moments corresponds to less distortion, particular of spectral features such as absorption band heights and widths. What makes Savitsky-Golay filtering even more attractive is straightforward and efficient manner in which the filtering is accomplished. Only a linear convolution with a set of filter coefficients is required. The resulting spectrum contains less noise than the original and exhibits less distortion compared to moving averaging filtering techniques of the same order. Other filtering techniques such as moving averaging filters may be able to remove more noise but as the filter order increases more distortion is introduced. By utilizing the Savitsky-Golay technique, the philosophy assumed is to place more priority on preservation of spectral characteristics as opposed to noise removal [25]. Savitsky-Golay filters are often preferred because when they are appropriately designed to match the waveform of an oversampled signal corrupted by noise, they tend to preserve the width and heights of the peaks in the signal waveform. While such performance features are often explained in terms of the implicit polynomial fitting process (where it is assumed that the fitted polynomial slopes are matched to those of the signal) the reason for this behavior is also obvious from the frequency domain properties of the filters. Specifically, they have extremely flat pass bands with modest attenuation in their stop-bands. Furthermore, the symmetric Savitsky-Golay filters have zero phase so that features of the signal is not shifted [26].

Savitsky-Golay filtering has its roots in least squares polynomial smoothing. The objective of least squares smoothing is to find a smoothed value for each point in the spectrum based on a least squares polynomial fit of a subset of data within a window. The window contains the point to be smoothed in the center position as well as several of its neighbors to either side of spectrum. Figure 3.22 shows a representation of such windows.

All the data within the window is used to perform the least square fit, but only central point is smoothed for each window position. The other points are smoothed by moving the window across the spectrum point by point, performing a least squares approximation to the windowed data at each location. A low order polynomial is typically used to perform the approximation, so the low frequency characteristics of the spectrum are approximated best by the polynomial, while the high frequency noise is lost in the approximation error [25].

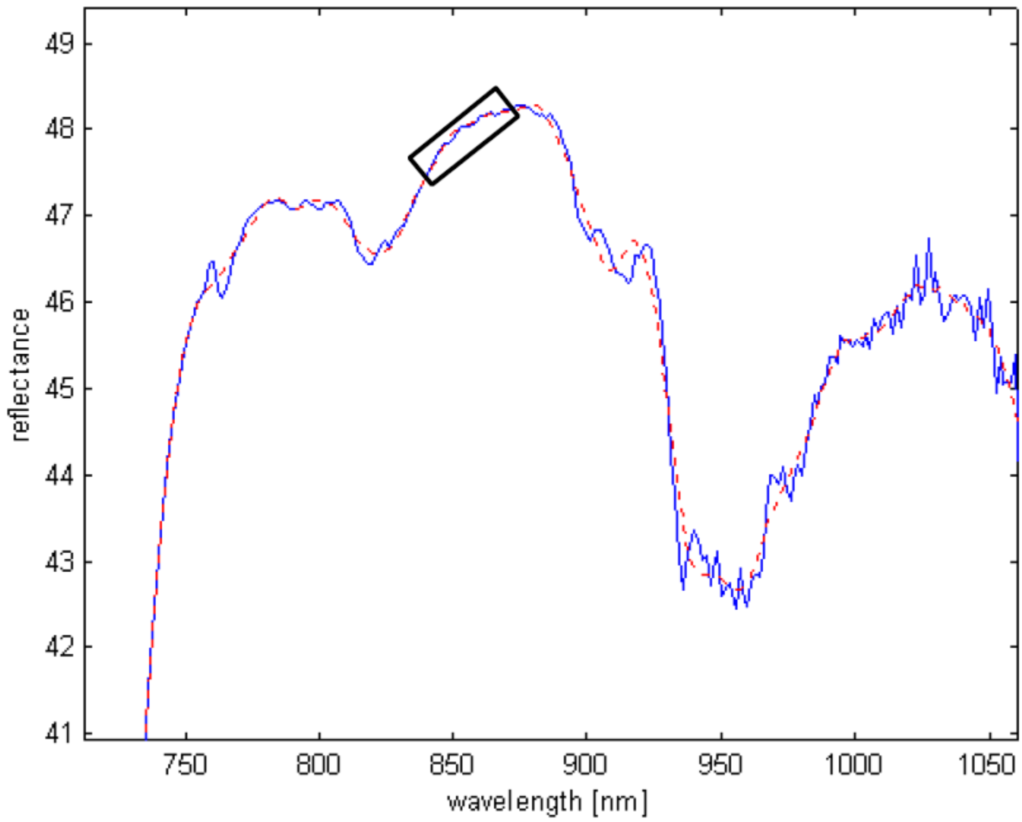


**Figure 3.22 :** Data window used in Savitsky-Golay filters [25].

# 4. VALIDATION OF MODEL

The results of the computer simulations are compared with the measurements conducted. In comparison phase, two points are important and they will be compared in order to validate model (i) Maximum amplitudes of the transient vibrations, (ii) Maximum amplitudes of steady-state vibrations. As mentioned before x and z axis translations are important and only they are compared. Simulation results and experimental results are shown in figure 4.1and figure 4.2.

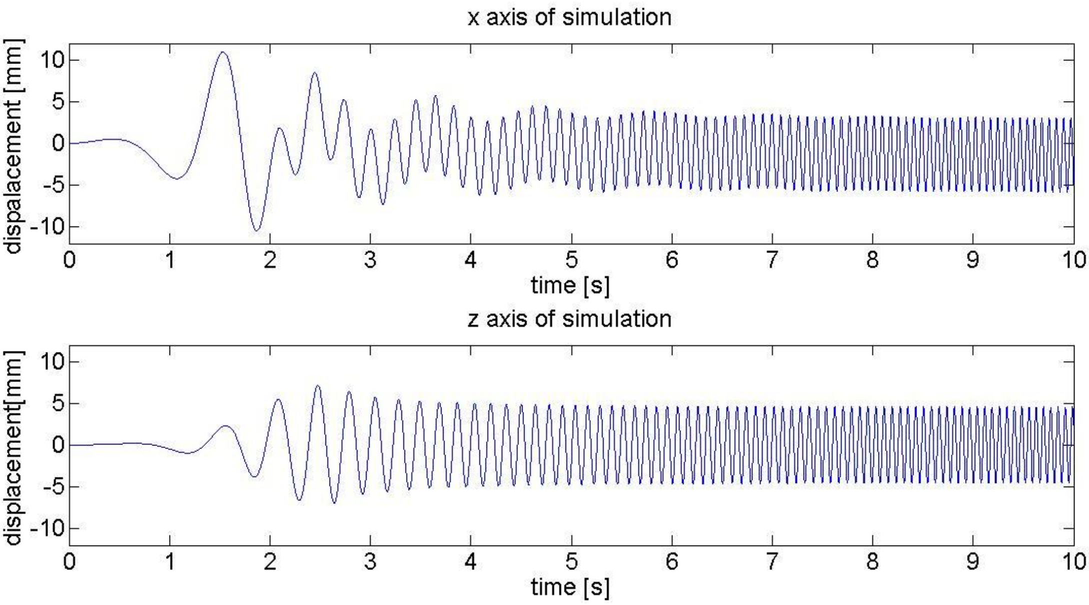


**Figure 4.1 :** Simulation results with the same parameters in experimental setup

The input spin speed of the washing machine drum and simulation input spin speed estimated with technique mentioned in chapter 3 is shown in figure 4.3. Sequent, with using this data, a curve fitted spin speed data is obtained and showed in figure 4.4.

The input of simulation spin speed is curve fitted from real data. A 3$^{rd}$ degree curve is fitted and used in the simulation. Same input spin speed is tried to be implemented in simulation with the fitted curve.

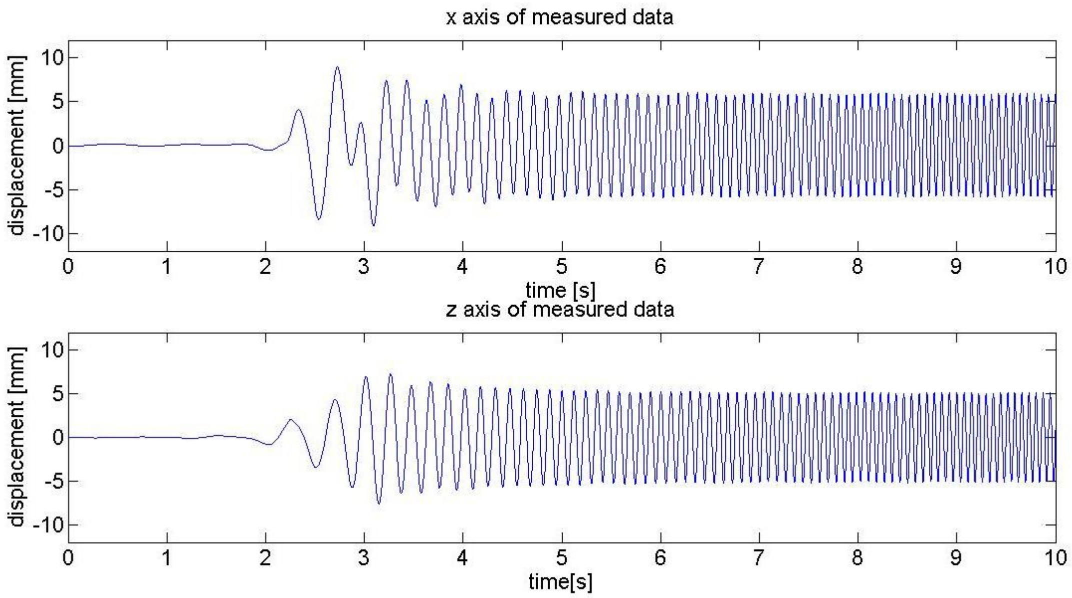


**Figure 4.2 :** Experimental results with the original design of the washing machine.

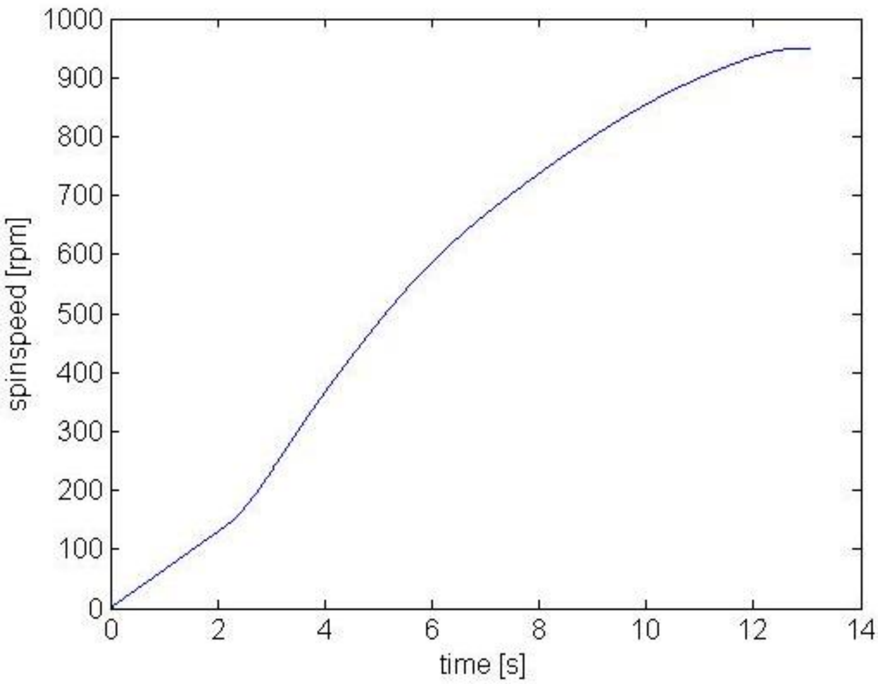


**Figure 4.3 :** Spin speed of experimental measurement.

In figure 4.5, vibration amplitudes versus input frequency are plotted in order to show frequency characteristics of simulation.

Main characteristics of vibrations are well matched. In simulation, x axis vibrations resonance range is 1.5Hz to 3Hz and in the measured x axis resonance range is 2.5 Hz to 4Hz. Also in the z axis, resonance range of simulation is 2Hz to 3.5Hz and resonance range in measurement is 3Hz to 4.5Hz.

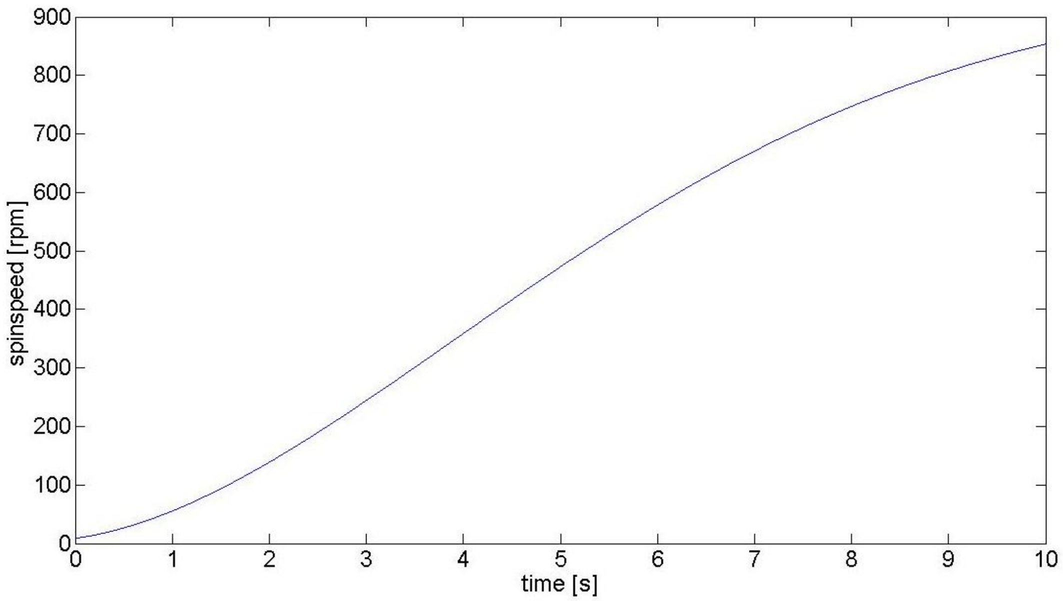


**Figure 4.4 :** Spin speed of simulation.

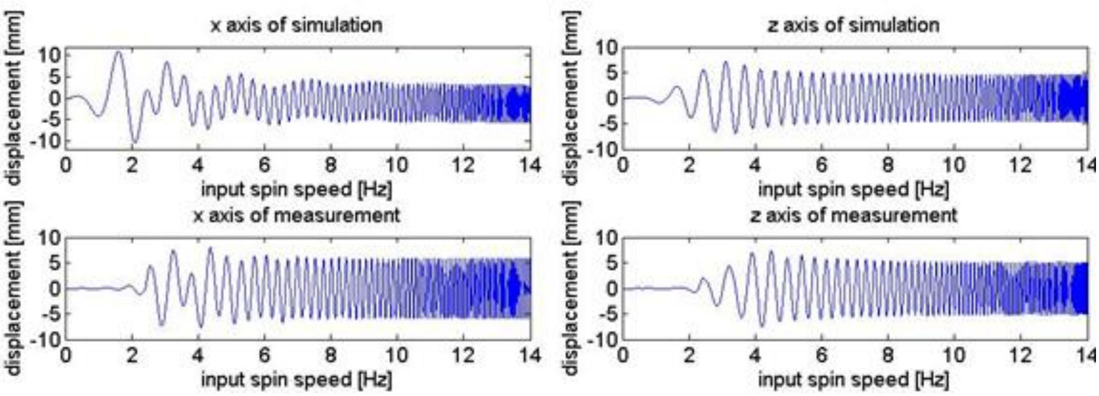


**Figure 4.5 :** Simulation results vs experimental results.

There is a shift in both axes resonance frequency range, comparing simulation to measurement. The reason of this shift is the parts that are neglected in the mathematical model, such as structural damping and stiffness, flexible parts' stiffness characteristics, nonlinear behavior of shock absorbers and so on. As far as the main interest of this study is not in frequency domain studies such as designing a frequency domain controller, mentioned shift can be acceptable. Main interest of this study is amplitudes of vibrations and they are well matched.

The comparison of critical points is shown in table 4.1 clearly and also model error is shown for each critic point.

**Table 4.1 :** Comparison of amplitudes for critical points.

| # | Simulation | Measurement | Absolute Error |
|---|---|---|---|
| **x transient** | | | |
| **max [mm]** | 10,93 | 8,9 | -2,03 |
| **min [mm]** | -10,46 | -9 | 1,46 |
| **z transient** | | | |
| **max [mm]** | 7,75 | 7,3 | -0,45 |
| **min [mm]** | -7,13 | -7,5 | -0,37 |
| **x steadt-state [mm]** | ±4,45 | ±5,8 | 1,35 |
| **z steady-state [mm]** | ±4,6 | ±5,1 | 0,5 |

The errors of approximation are considerably low and model is considered to be sufficient to represent the vibration dynamics of washing machine. Therefore, it is convenient to proceed to optimization phase.

# 5. PARAMETER OPTIMIZATION

Anyone, who intends to solve a problem, faces with an optimization process. Because every problem has more than one solution and everyone wants to select the best solution to his problem. This is the basic definition of optimization. For engineers optimization process is more important than it is to the others. Because in the solution of engineering problems there is a continuous improvement cycle which leads engineer to systematical optimization [27].

A typical optimization problem involves the following stages:

(i) The selection of a set of variables to describe the design alternatives,

(ii) The selection of an objective, expressed in terms of design variables which we seek to minimize or maximize,

(iii) The determination of a set of constraints, expressed in terms of the design variables, which must be satisfied by any acceptable design,

(iv) The determination of a set of values for the design variables which minimize or maximize the objective function while satisfying all the constraints.

In most engineering problems, the system is required to behave according to some law of physics such as Newton's law of motion. There can be an analytical expression that can be derived to express the system. There must be some set of variables to be identified to express the system. These variables are called design variables. The number of independent variables determines the design degrees of freedom and also number of design variables indicates the dimensions of optimization. We shall use the term "design variables" to indicate all unknowns of the optimization problem and represent them in the vector $\mathbf{x}$.

There can be many feasible designs for a system, and some are better than others. To compare different designs, we must have a criterion. The criterion must be a scalar function whose numerical value can be obtained once a design is specified, i.e., it must be a function of the design variable vector x. Such a criterion is usually called an objective function for the optimum design problem, which needs to be maximized or minimized depending on problem requirements.

All the restrictions of the design are called constraints. The constraints must depend on design variables. The constraints in design can be equality or inequality depending on the function.

The set of values for design variables satisfying all the constraints is called the feasible space or feasible domain. One of the design variables set must be selected that makes objective function maximum or minimum as the optimum solution. Any systematical procedure either analytical or numerical used for selecting the optimum design variable vector in order to maximize or minimize the objective function is called optimization method.

## 5.1 Optimization Methods

Optimization methods can be broadly divided in to two groups that have philosophically different points of views. (i) Analytical methods (optimality criteria) and (ii) numerical methods (search methods) (figure 5.1). Optimality criteria are the conditions a function must satisfy at its minimum point. On the other hand, search techniques starts with an estimate of design variables which usually does not satisfy optimality criteria, therefore it is improved iteratively until they satisfy optimality criteria [28].

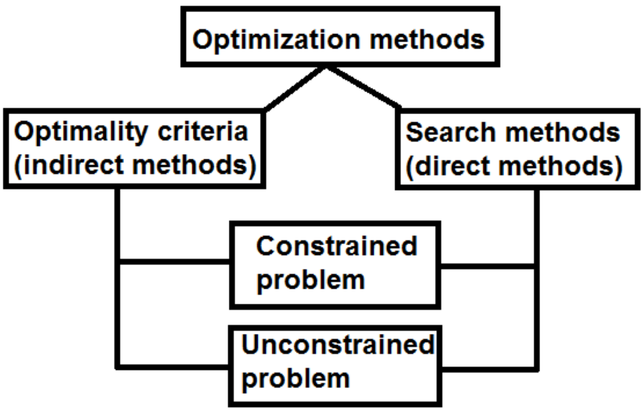


**Figure 5.1 :** General classification of optimization methods.

In most engineering problems optimization starts with a nonlinear multivariable objective functions, hence only nonlinear procedures are left in consideration. For solving these nonlinear problems, numerical methods are more powerful than analytical methods. One reason is that the analytical methods for solving some of the problems are too cumbersome to use. There are two basic reasons why the analytical methods are inappropriate for many engineering design problems:

(i) The numbers of design variables and constraints can be large. In that case, the necessary conditions give a large number of nonlinear equations, which can be difficult to solve. Numerical methods must be used to find solutions of such equations in any case. Therefore, it is appropriate to use the numerical methods directly to solve the optimization problems. Even if the problem is not large, these equations can be highly nonlinear and cannot be solved in a closed form.

(ii) In many engineering applications, objective functions and/or constraint functions are implicit functions of the design variables; that is explicit functional forms of the independent variables are not known. These functions cannot be treated easily in the analytical methods for solution of optimality conditions [28].

In the simplest of the search methods, the direction for minimization is determined solely from successive evaluations of the objective function. As a general rule, in solving unconstrainted nonlinear programming problems, gradient and second derivative methods converge faster than search methods. However, in practice, the derivative methods have two main disadvantages in their implementation. First disadvantage is that in problems with large number of design variables, it is laborious or impossible to provide analytical functions for derivatives needed in gradient or Hessian requiring algorithms. Second disadvantage is the optimization methods based on the evaluation of first and second derivatives, requires large amount of time for preparation of the user before he/she introduces the problem to algorithm. According to Arora the taxonomy of global optimization algorithms are shown in figure 5.2. Therefore, gradient and/or second derivative required techniques are not considered as a solution to washing machine vibration dynamics optimization problem.

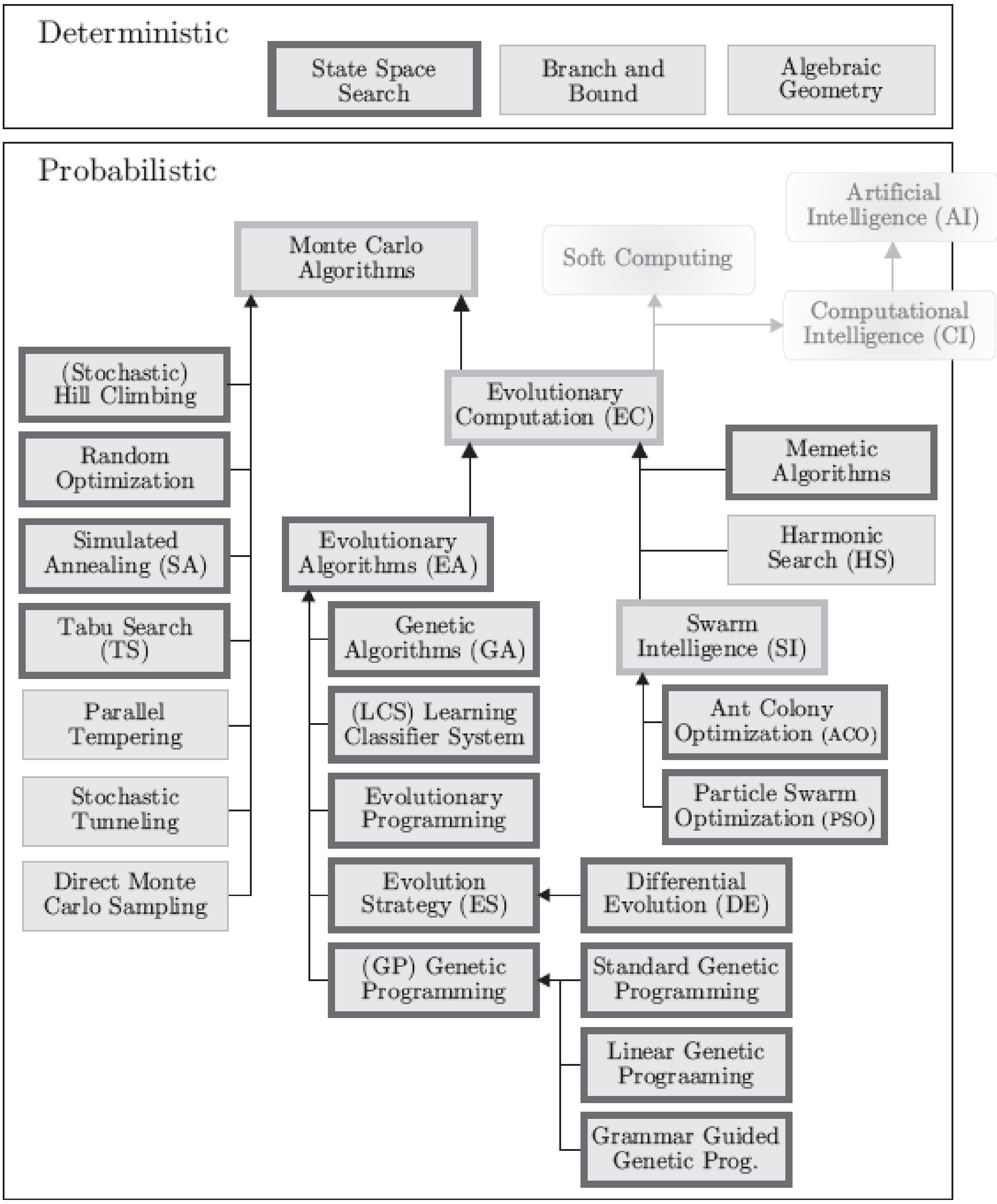


**Figure 5.2 :** The taxonomy of global optimization algorithms [28].

For the solution of optimization problem of vibration minimization of horizontal axis washing machine, genetic algorithm, which is one of evolutionary algorithms, is chosen. The reason for this selection is that the algorithms in this class is easy to implement and needs a little change in the algorithm to adopt to another problem or to changes in the optimization problem itself.

These algorithms do not guarantee the optimum solution. This does not mean that the results obtained using them are incorrect, they may just not be the global optima. On the other hand, a solution a little bit inferior to the best possible one is better than one which needs large amounts of time and cumbersome efforts [29]. This type of search for sub-optimal solutions in short design cycles is in well agreement with the concurrent engineering.

Some brief information about optimization algorithms in probabilistic algorithms class is stated below. Since, interest of this thesis is not explaining all algorithms, only some of them are mentioned briefly.

*Hill Climbing method* is a very old and simple search and optimization algorithm for single objective functions. In principle, hill climbing algorithms perform a loop in which new individual is better than its parent, the new offspring replaces the parent. Then the cycle starts all over again. In this aspect, it is similar to evolutionary algorithms with population size 1.

*Tabu search*, has been developed in the mid 1980s. The word “tabu” stems from Polynesia and describes a sacred place or object. Things that are tabu must be left alone and may not be visited or touched. Tabu Search extends Hill Climbing by this concept, it declares solution candidates which have already been visited as tabu. Hence they must not be visited again and optimization process is less likely to stuck on a local optimum. The simplest realization of this approach is to use a list of tabu which stores all solution candidates that have already been tested. If a newly created phenotype can be found on the list, it is not investigated but rejected right away.

*Differential Evolution* is a method for mathematical optimization of multidimensional functions that belongs to the group of evolution strategies. Developed by Storn and Price in 1974, the differential evolution has invented in order to solve Chebyshev polynomial fitting problem. It has proven to be very reliable optimization strategy for many different tasks where parameters that can be encoded in real vectors. The essential idea behind differential evolution is the way the recombination operator “deRecombination” is defined for creating new solution candidates. The difference, $x_1 - x_2$ of two vectors $x_1$ and $x_2$ in X is weighted with weight $w \in R$ and added to a third vector $x_3$ in the population.

$$x = deRecombination(x_1, x_2, x_3) \rightarrow x = x_3 + w(x_1 - x_2) \qquad \textbf{(5.1)}$$

Except for determining $w$, no additional probability has to be used and the differential evolution scheme is completely self organizing.

*Ant colony algorithm* inspired by the research done by Deneubourg et al. on real ants and probably by the simulation experiments by Stickland et al. in 1964. Dorigo et al. developed the ant colony optimization for problems that can be reduced to finding optimal paths in graphs in 1966. Ant colony optimization is based on the metaphor of ants seeking food. In order to do so, an ant will leave the anthill and begin to wander into a random direction. While the ant paces around, it lays a trail of pheromone. Thus after the ant has found some food it can track its way back. By doing so, it distributes another layer of pheromone on the path. Another ant that senses the pheromone will follow its trail with a certain probability. Each ant that finds the food will excrete some pheromone on the path. By time pheromone density of the path will increase and more and more ants will follow to food and back. The higher the pheromone density, the more likely will an ant stay on trail. However, pheromones vaporize after some time [29].

*Genetic algorithm* is the optimization algorithm selected for horizontal axis washing machine vibration minimization; hence, it is explained in detail in the next section.

## 5.2 Genetic Algorithm

Genetic algorithms use only the function values in the search process to make progress toward to a solution without regard to how the functions are evaluated. Continuity or differentiability of the problem functions is neither required nor used in calculations of the algorithms. Therefore, the algorithms are very general and can be applied to all kinds of problems discrete, continuous and non-differentiable. In addition, the methods determine global optimum solutions as opposed to local solutions determined by a continuous variable optimization algorithm. The methods are easy to use and program since they do not require use of gradients of cost or constraint functions [28].

### 5.2.1 Fundamentals of genetic algorithm

Inspiration of genetic algorithm is based on Charles Darwin's "Survival of the fittest" theory. The roots of genetic algorithms go back to the mid 1950s, where biologist like Barricelli and the computer scientist Fraser began to apply computer-aided simulations in order to gain more insight into genetic processes and the natural evolution and selection. Bremermann and Bledsoe used evolutionary approaches based on binary string genomes inequalities for function optimization and for determining the weights in neural networks in early 1960s. At the end of that decade, important research on such search spaces was contributed by John Holland at the University of Michigan. As a result of Holland's work, genetic algorithms as a new approach for problem solving could be formalized finally become widely recognized and popular. Today there are many applications of GA in science, economy and engineering [29].

### 5.2.2 Genetic algorithm definitions, operators and operation flow

In order to perform the genetic algorithm there are some fundamental operators and some biological definitions that refers to optimization terms. In this section, they are explained in detail.

**Population:** The set of design points at the current iteration is called population. It represents a group of designs as a potential solution. Number of designs in a population is $N_p$ ; also called population size.

**Generation:** An iteration of the genetic algorithm is called a generation. A generation has a population of size $N_p$ that is manipulated in a genetic algorithm.

**Chromosome:** This term is used to represent a design point. Thus, a chromosome represents a design of the system, whether feasible or infeasible. It contains values for all the design variables.

**Gene:** This term is used for a scalar component of the design vector. It represents the value of a particular design variable.

**Genotype:** This term refers to information of genes in chromosome. It represents the design parameter in the encoded form for algorithm.

**Phenotype:** This term refers to physical expression of genotype. It is the physical value of design parameter.

**Fitness:** Refers to measurement of the success of the organism in its life (survival). In other words, fitness shows how good or bad is the solution. The fitness function defines the relative importance of a design. A higher fitness value implies a better design.

Genetic Algorithm operates in the following way. First, a random population of design variables is generated. Then this population is tested and fitness values are assigned. In order to create new offspring, selection of parents hence the chromosomes is done. With the selected chromosomes, reproduction is performed with crossover and mutation operator, which is described later on. By doing so, according to "survival of the fittest" theory, the fitness of the new generations is improved. This cycle is repeated over and over until a predefined criterion is reached. Mentioned flow of genetic algorithm is shown in figure 5.3.

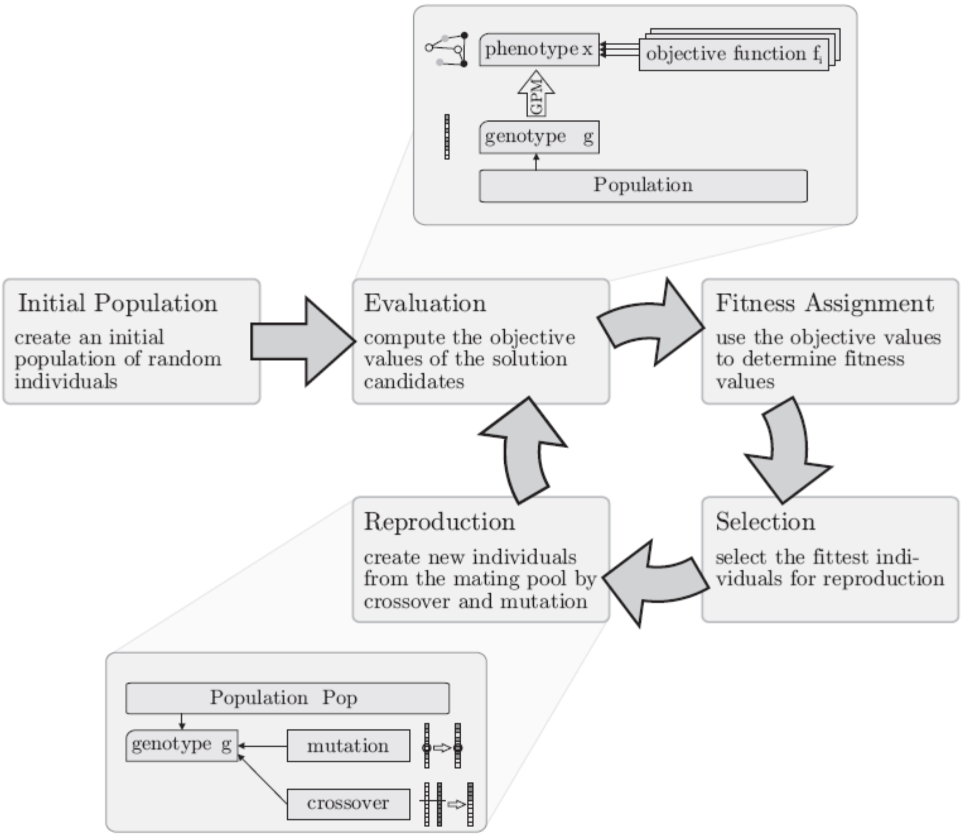


**Figure 5.3 :** Basic cycle of genetic algorithm [29].

### 5.2.2.1 Encoding operator

Likewise, the DNA in the nature, the characteristics of a design variable is encoded into genetic algorithm terms. Encoding is the process of including the possible solutions into the genetic algorithm in terms of coded strings. Encoding can be done in several ways.

**Binary Encoding:** This method is the most common used to represent information contained. In genetic algorithms, it was first used because of its relative simplicity. A chromosome consists of sequences of 0s and 1s. Every 1 and 0 represent some physical appearance dependent on the problem.

A gene looks like **(111010)**

A chromosome with 4 design variable looks like **(111010101001010111110001)**

**Value Encoding:** Value encoding can be used in problems where values such as real numbers are used. In value encoding, every chromosome is a sequence of some values and values can be anything connected to the problem such as;

Chromosome 1: (1.234 5.324 0.455 7.123)

Chromosome 2: (ABDFGHRTYOIULOP)

Choromosome 3: (back, right, forward, left)

For value encoding it is necessary to develop new types of crossover and mutation operators specific to each problem.

**Permutation Encoding:** This method is used in ordering problems such as traveling salesman problem or task ordering problem. In permutation encoding, every chromosome is a string of numbers that represent a position in a sequence.

Chromosome 1: (58945179)

Chromosome 2: (ABCDGHF)

#### 5.2.2.2 Decoding operator

This operator takes a role in the first fitness evaluation of individual population and after reproduction again for evaluation of fitness. Genotypes are needed to be converted in to problem parameters in order to evaluate them in fitness function. This is simply reverse process of encoding.

#### 5.2.2.3 Select operator

After initialization and fitness assignment to each chromosome, in order to reproduce and generate new offspring, two of them have to be selected for exchanging information included to generate new and different individuals, as parents. There are several ways for selecting parents. In selection process chromosomes are selected from the population to be parents to crossover. According to Darwin's evolution theory the best ones should survive and create new offspring. There are several methods to select the best chromosomes. Some popular of them are as follows.

**Roulette Wheel Selection:** This method is also known as "Fitness Proportionate Selection". Conceptually, this selection can be thought as a game of roulette. All chromosomes are distributed on a roulette wheel, proportional to their fitness. This means greater the fitness, bigger the share in the wheel. When the roulette wheel spins, every one of the chromosomes has chance to be selected but ones with greater fitness values are more likely to be selected. Concept is shown in figure 5.4.

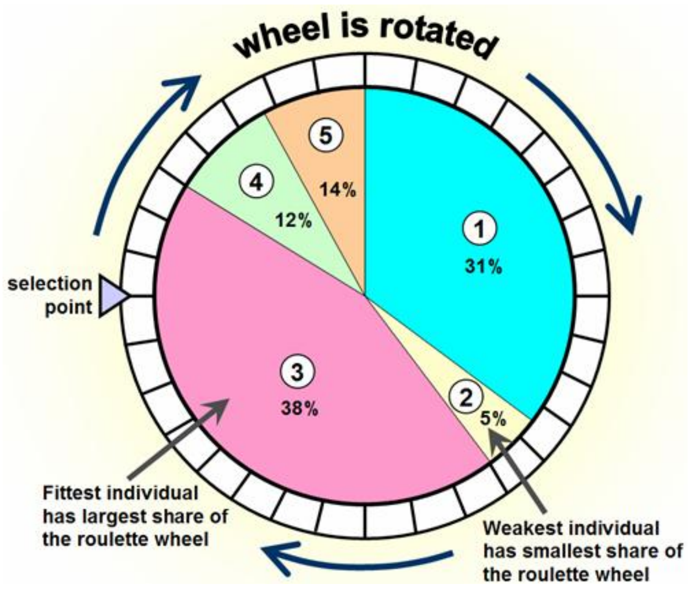


**Figure 5.4 :** Roulette wheel selection.

**Rank Selection:** In Roulette Wheel method, if one of the individuals has a great fitness such as 90%, this chromosome will be selected most of the time and other chromosomes have very few chances to be selected. If chromosome with a lower fitness value contains a good gene in itself it will probably be lost in roulette wheel selection. Rank selection on the other hand ranks all chromosomes in a way that best fitness value will have rank1 and worst one will have rank n. In figure 5.5, an illustration of this selection is shown comparing it to roulette wheel method. This selection converges slower than roulette wheel selection.

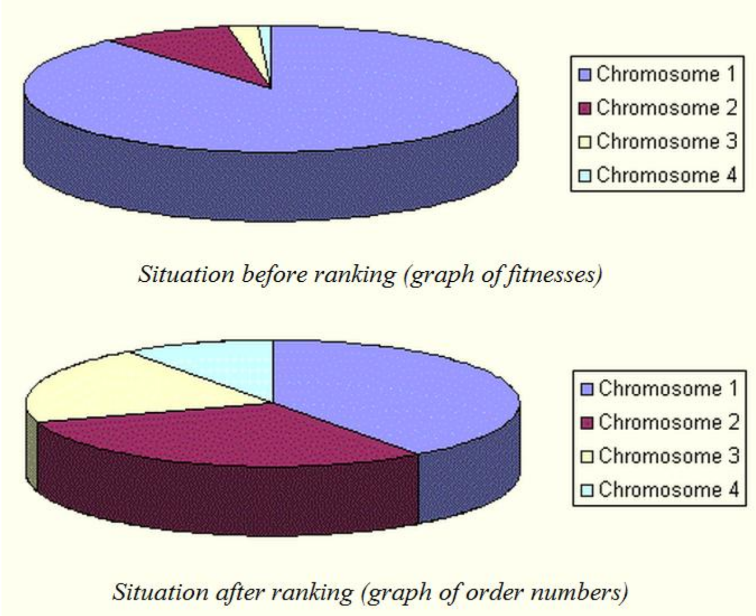


**Figure 5.5 :** Rank selection.

**Tournament Selection:** This method involves running several “tournaments” among few individuals chosen at random from the population. The winner of each tournament, which is the one with best fitness, is selected for crossover. Selection pressure is easily adjusted by changing the tournament size. If tournament size is larger, weak individuals have a smaller chance to be selected [33].

### 5.2.2.4 Cross over operator

Crossover is a genetic operator that combines two parent chromosomes to produce new chromosome (offspring). The idea behind the crossover is that the new chromosome may be better than both of the parents if it takes the best characteristics from each of the parents. Crossover occurs with a user-defined probability during the evolution. There are many types of crossover operators and the most commonly used ones are as follows. Performance of different types of crossover operators are almost equal [34].

**One Point Crossover:** One crossover point is selected on the chromosome, binary string from beginning of chromosome to the crossover point is copied from one parent and the rest is copied from second parent. The crossover point is selected randomly for each crossover operation. Figure 5.6 is showing clearly this phenomenon.

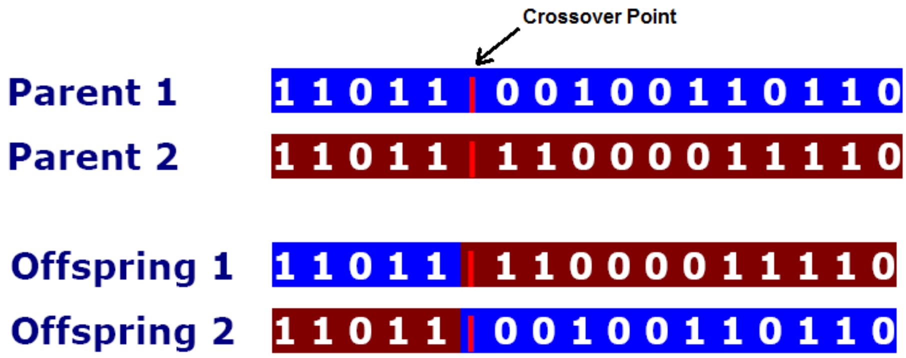


**Figure 5.6 :** One point cross over.

**Two Point Cross Over:** This method selects two crossover points and perform the same procedure as in one point crossover. First part of string is copied from first parent, second part is copied from second parent between crossover parts and rest is copied from first parent. Figure 5.7 shows two point cross over.

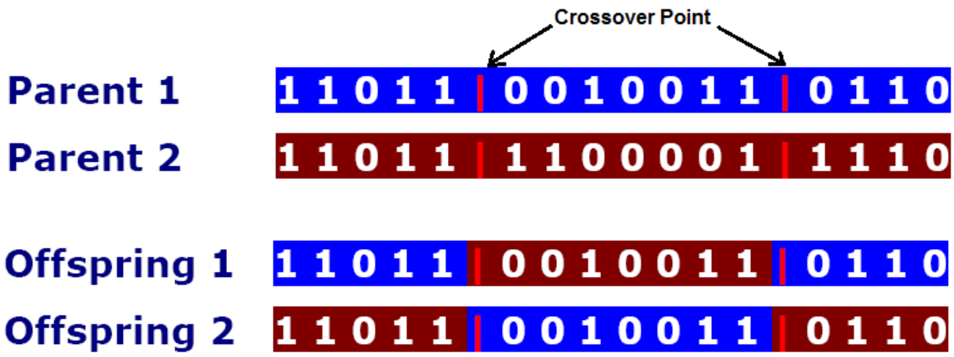


**Figure 5.7 :** Two points cross over.

**Uniform Crossover:** In this method, binary strings are randomly copied from each parent and new offspring is created. Figure 5.8 shows uniform crossover.

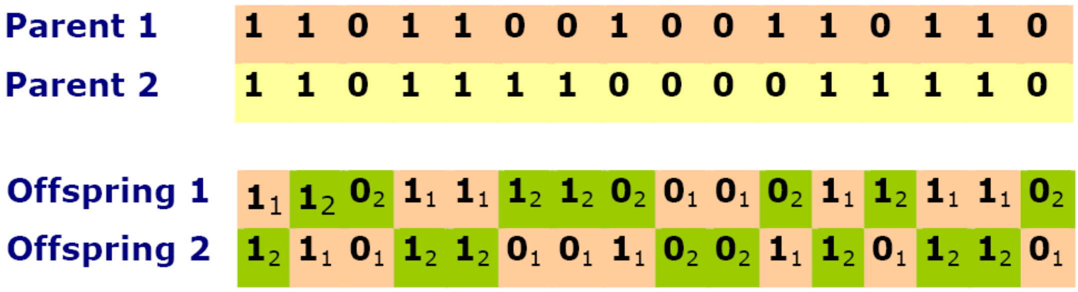


**Figure 5.8 :** Uniform cross over.

#### 5.2.2.5 Mutation operator

Mutation takes its place after the crossover operator. Mutation is a genetic operator used to maintain genetic diversity from one generation of a population of chromosomes to the next. Mutation alters one or more genes in the chromosome by toggling single or multiple bits. Mutation is an important part of the genetic search since it helps to prevent the population from stucking in local optima. Mutation probability is a user-defined parameter as well. However, probability of mutation has to be kept in lower values such as 1%. If mutation probability is high, then algorithm starts to jump all over the search space randomly. Hence, convergence problems occur. Figure 5.9 shows how it works. Mutation point or points are randomly selected.

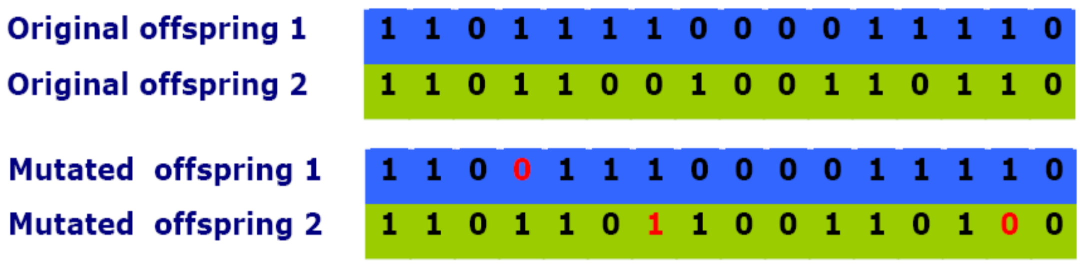


**Figure 5.9 :** Mutation mechanism.

### 5.3 Genetic Algorithm Implementation to Washing Machine Vibration Problem

Mathematical model of horizontal axis washing machine is derived and a Simulink representation of the model is generated on previous chapters. According to this model, vibration amplitudes are minimized using Genetic Algorithm Optimization method.

#### 5.3.1 Determination of optimization parameters

Horizontal axis washing machine vibration amplitudes minimization problem is introduced in this section in terms of Genetic Algorithm. As mentioned in chapter 2, vibrations on x and z axis are the main interests here and parameters that effects vibration amplitudes at these axes are also explained. The goal of this optimization study is to minimize vibrations on x and z axes. The following parameters of mathematical model are used as design variables in the optimization problem formulation.

Design vector consists of stiffness coefficient of springs, damping coefficient of shock absorbers, spring attachment distances between spring strut and tub in two axes, shock absorber attachment distances between tub and cabinet in two axes. These variables are mentioned in chapter 2 as $k$, $c$, $xs1$, $zs1$, $xd1$ and $zd1$ respectively. First spring and shock absorber attachment is symmetrical to second spring and shock absorber attachment. Hence, $xs2, zs2$, $xd2$ and $zd2$ are equal to , $xs1$, $zs1$, $xd1$ and $zd1$ respectively. There are 6 independent variables in design vector.

$$d = (k, c, xs, zs, xd, zd\ ) \qquad \textbf{(5.2)}$$

### 5.3.2 Fitness function assignment

In order to measure the performance of each individual in initial population and the ones generated by genetic algorithm some fitness function are assigned. For measuring the characteristics of horizontal axis washing machine vibration, three different fitness functions are generated.

(i) The wobbling motion around the origin. This can be illustrated by plotting x and z axes vibrations together. Figure 5.10 shows plot of wobbling movement of horizontal axis washing machine.

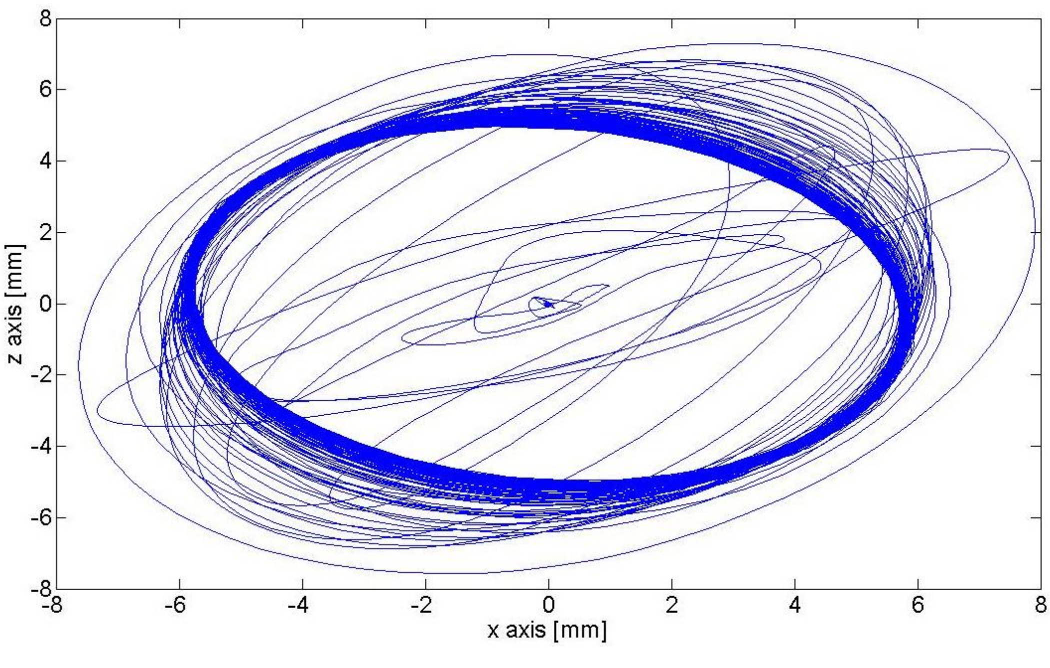


**Figure 5.10 :** Wobbling movement of oscillation group.

Maximum displacements occur during the transient vibrations. In order to find the maximum displacement from the origin and to minimize the vibrations, following fitness function is generated.

$$R = \sqrt{x^2 + z^2} \tag{5.3}$$

$$fitness1 = \frac{1}{R_{max}} \tag{5.4}$$

(ii) In the second case steady state vibrations are considered and following fitness function is generated.

$$fitness2 = \frac{1}{x_{st\,max} - x_{st\,min} + z_{st\,max} - z_{st\,min}} \tag{5.5}$$

$x_{st\,max}$ here is maximum steady state vibration amplitude and $x_{st\,min}$ is the minimum steady state vibration amplitude. They are subtracted because $x_{st\,min}$ has a negative value. Same thing is performed for z axis and both axis vibration amplitudes are added to each other in order to consider all of them at the same time. Figure 5.11 shows transient and steady state vibration amplitudes.

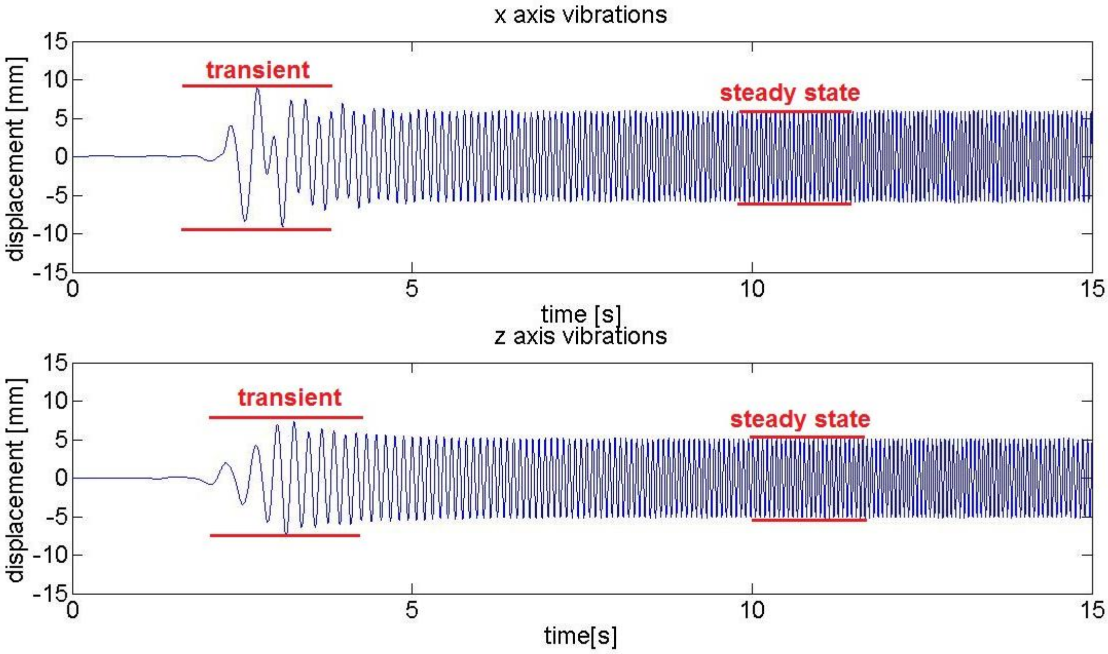


**Figure 5.11 :** Transient and steady state vibrations.

(iii) In the third case both transient and steady state vibrations are considered and a fitness function is generated in order to minimize both steady state and transient vibrations at the same time.

$$fitness3 = \frac{1}{x_{st\,max} + x_{tr\,max} - x_{st\,min} - x_{tr\,min} + z_{st\,max} + z_{tr\,max} - z_{st\,min} - z_{tr\,min}} \quad \textbf{(5.6)}$$

$x_{tr\,max}$ is here the maximum vibration amplitude in transient zone. Subscript “tr” denotes the transient zone vibrations. Results of three different fitness functions are illustrated in the section 5.3.9.

### 5.3.3 Optimization constraint

During the normal operation of a washing machine, no matter what happens, washing machine cabinet must not move relative to ground which is called stepping condition. Stepping condition can occur when total horizontal forces acting on washing machine exceeds the friction force between floor and washing machine. Hence, a constraint is obtained as given in equation **(5.7).**

$$(f_{ver} + Mg) * f_c - f_{hor} > 0 \quad \textbf{(5.7)}$$

$f_{ver}$ and $f_{hor}$ are vertical and horizontal forces acting on washing machine cabinet respectively. These forces are obtained from the simulation. $f_c$ is the friction coefficient and it is assumed to be 0.2 which is the coefficient of plastic foots and a smooth surface such as a bathroom surface. If any of individual causes stepping condition then no matter how good the vibration amplitudes and no matter how high fitness it has, its fitness value is replaced with zero. This means any of individuals violating inequality **(5.7)** considered as dead member with zero fitness value.

### 5.3.4 Creation of initial population

For each design variable, a range is specified. The range is chosen so that the actual values of design variables are in the middle. By this way, a search space is created and an initial population is generated. The values for each design variable and encoding of them is as follows.

**Stiffness Coefficient of Springs:** Design variable "k" is represented with 3 bits, hence there are eight values starting with 5000 and ending with 7500 N/m. A binary string is generated in order to represent design variable "k".

$$k = \begin{bmatrix} 000 \\ 001 \\ 010 \\ 011 \\ 100 \\ 101 \\ 110 \\ 111 \end{bmatrix} \quad \textbf{(5.8)}$$

Decimal equivalent of this string is differs between 0 and 7 hence for decoding some arrangements are made.

$$k_i = 5000 + k_{dec\_i}357.15 \quad \textbf{(5.9)}$$

After the arrangement, design variable becomes as following in terms of problem parameters.

$$k = \begin{bmatrix} 5000 \\ 537.15 \\ 5714.3 \\ 6071.45 \\ 6428.6 \\ 6785.75 \\ 7142.9 \\ 7500.05 \end{bmatrix} \quad \textbf{(5.10)}$$

**Damping Coefficient of Shock Absorbers:** Design variable "c" is also represented with 3 bits, hence there are eight values starting with 150 and ending with 400 Ns/m. Decimal equivalent of this string differs between 0 and 7 hence for decoding some arrangements is made.

$$c_i = 150 + c_{dec\_i}35.715 \quad \textbf{(5.11)}$$

After the arrangement design variable becomes as following in terms of problem parameters.

$$c = \begin{bmatrix} 150 \\ 185.715 \\ 221.43 \\ 257.145 \\ 292.86 \\ 328.575 \\ 364.29 \\ 400 \end{bmatrix} \tag{5.12}$$

**Attachment Distances of Springs and Shock Absorbers:** Design variable "xs" is represented with 3 bits, hence there are eight values starts with 0.03 and end with 0.13 m. Decimal equivalent of this string is differs between 0 and 7 hence for decoding some arrangements is made.

$$xs_i = 0.03 + xs_{dec\_i}0.0143 \tag{5.13}$$

After the arrangement, design variable becomes as following in terms of problem parameters.

$$xs = \begin{bmatrix} 0.03 \\ 0.0443 \\ 0.0586 \\ 0.0729 \\ 0.0871 \\ 0.1014 \\ 0.11157 \\ 0.13 \end{bmatrix} \tag{5.14}$$

Same way is followed to obtain $zs$, $xd$ and $zd$ decoding equations and value range is as follows respectively.

$$zs_i = 0.09 + xs_{dec\_i}0.0204 \tag{5.15}$$

$$xd_i = 0.03 + xd_{dec\_i}0.0143 \tag{5.16}$$

$$zd_i = 0.1 + zd_{dec\_i}0.01528 \tag{5.17}$$

$$zs = \begin{bmatrix} 0.09 \\ 0.1104 \\ 0.1309 \\ 0.1513 \\ 0.1717 \\ 0.1921 \\ 0.2126 \\ 0.233 \end{bmatrix}, \quad xd = \begin{bmatrix} 0.03 \\ 0.0443 \\ 0.0586 \\ 0.0729 \\ 0.0871 \\ 0.1014 \\ 0.11157 \\ 0.13 \end{bmatrix}, \quad zd = \begin{bmatrix} 0.1 \\ 0.115 \\ 0.1306 \\ 0.1459 \\ 0.1611 \\ 0.1764 \\ 0.1917 \\ 0.207 \end{bmatrix} \tag{5.18}$$

After encoding design parameters into binary numbers, every binary number generated placed in the population randomly. There are twenty chromosomes in initial population and number of members in the population is kept constant during the optimization process. Initial population is shown in table 5.1.

**Table 5.1 :** Initial population.

| # | k | | | c | | | xs | | | zs | | | xd | | | zd | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Chromosome 1** | 0 | 0 | 0 | 0 | 1 | 1 | 0 | 0 | 0 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 1 | 0 |
| **Chromosome 2** | 0 | 0 | 1 | 1 | 0 | 0 | 0 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 1 | 0 | 1 | 1 |
| **Chromosome 3** | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 0 |
| **Chromosome 4** | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 1 | 1 | 0 | 0 | 0 | 1 | 1 | 1 | 0 | 1 |
| **Chromosome 5** | 1 | 1 | 0 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 1 | 1 | 1 | 0 | 0 | 1 | 1 | 0 |
| **Chromosome 6** | 1 | 0 | 1 | 1 | 0 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 1 | 0 | 1 | 1 | 1 |
| **Chromosome 7** | 1 | 0 | 0 | 0 | 1 | 1 | 1 | 1 | 0 | 1 | 0 | 0 | 1 | 0 | 1 | 0 | 0 | 0 |
| **Chromosome 8** | 1 | 1 | 1 | 0 | 1 | 0 | 1 | 1 | 1 | 1 | 0 | 1 | 1 | 0 | 0 | 0 | 0 | 1 |
| **Chromosome 9** | 1 | 1 | 0 | 1 | 1 | 1 | 0 | 1 | 0 | 1 | 1 | 0 | 1 | 1 | 1 | 0 | 1 | 0 |
| **Chromosome 10** | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 1 |
| **Chromosome 11** | 1 | 1 | 0 | 1 | 0 | 1 | 1 | 0 | 0 | 0 | 0 | 0 | 1 | 0 | 1 | 0 | 0 | 0 |
| **Chromosome 12** | 1 | 1 | 1 | 1 | 0 | 0 | 1 | 0 | 1 | 0 | 0 | 1 | 1 | 0 | 0 | 1 | 0 | 0 |
| **Chromosome 13** | 0 | 0 | 0 | 0 | 0 | 1 | 0 | 0 | 0 | 0 | 1 | 0 | 0 | 1 | 1 | 1 | 1 | 0 |
| **Chromosome 14** | 0 | 0 | 1 | 0 | 0 | 0 | 1 | 0 | 0 | 0 | 1 | 1 | 0 | 1 | 0 | 0 | 0 | 0 |
| **Chromosome 15** | 0 | 1 | 0 | 0 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 1 | 1 | 1 | 1 | 0 | 1 |
| **Chromosome 16** | 0 | 1 | 1 | 1 | 0 | 1 | 0 | 0 | 1 | 1 | 0 | 1 | 1 | 0 | 1 | 0 | 1 | 1 |
| **Chromosome 17** | 1 | 0 | 0 | 1 | 1 | 1 | 0 | 1 | 0 | 1 | 1 | 0 | 0 | 1 | 1 | 1 | 0 | 1 |
| **Chromosome 18** | 1 | 0 | 1 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 1 | 1 | 0 | 0 | 1 | 1 | 1 | 1 |
| **Chromosome 19** | 1 | 1 | 0 | 0 | 0 | 1 | 1 | 0 | 0 | 1 | 0 | 0 | 0 | 1 | 0 | 1 | 0 | 0 |
| **Chromosome 20** | 1 | 1 | 1 | 0 | 0 | 0 | 1 | 0 | 1 | 0 | 1 | 0 | 1 | 1 | 0 | 0 | 0 | 0 |

### 5.3.5 Selection of parents

After initialization of first population, fitness values of each individual are calculated. To create better offspring, parents are selected among the individuals having better fitness values. For selection purposes, *fitness proportionate selection* also known as *roulette wheel selection* is used. The reason for this choice is faster convergence and higher chances of selection of better individuals. The method for this operation is shown in following steps.

(i) Calculate all fitness values, and sum all of them and obtain total fitness F.

(ii) Generate a random number between zero and total fitness F, and call it S.

(iii) Start summing fitness values of individuals, starting from first one. This summed fitness is called partial sum P, when partial sum P is bigger than S, stop and return to the chromosome where P reached S. Declare, the chromosome as a parent.

By repeating this algorithm twice, two parents are obtained for creating new offspring. The roulette wheel is performed by generating a random number S and summing all fitness values starting from one to reach the S. Every time the random number S is different and higher fitness value is more close to be chosen.

### 5.3.6 Creating new offspring

In sections 5.2.2.4 and 5.2.2.5 crossover and mutation operators are explained. In order to create new offspring, crossover and mutation operators are used. Crossover is used as single point cross over and mutation is performed together with cross over operation. There are two probability values to be defined in this operation, crossover probability and mutation probability. These probabilities define the crossover and mutation occurring chance and affect the convergence speed of GA [35]. Creating new offspring is shown in following steps.

(i) Generate a random number between zero and one. Compare it with the crossover probability.

(ii) If crossover probability is smaller than the random number, there is no crossover and child is exactly the same with the parent.

(iii) If crossover probability is bigger than the random number, than divide chromosome into two parts and define the division point as random. Do this for both parents and obtain 4 allele genes.

(iv) Combine the allele genes in order to obtain a complete chromosome and a new individual.

(v) Generate a new random number during each combination and compare it with mutation probability each time.

(vi) If new random number is bigger than the mutation probability than do nothing and use the allele gen to create new offspring.

(vii) If new random number is smaller than the mutation probability than, toggle the binary numbers in the allele gene and than do the combination to create new offspring.

### 5.3.7 Elitism

After performing above mentioned operations and obtaining new offspring, new fitness values of each individual are calculated. For a faster convergence, better individuals are kept and the worst individuals are replaced with best individual. By doing so, solutions with low fitness value are eliminated and solutions with higher fitness value are kept in population. In the next generation, better offspring creation probability is increased because in the population better solutions are kept. By not eliminating the worst individual and replacing it with the best individual, the population size is kept constant.

### 5.3.8 Optimization termination criteria

There are several criteria used in GAs to stop the optimization. Most common approaches in stopping GA are as follows;

- Limiting the maximum number of iterations.
- A similar approach to iteration limit is the limited number of fitness function evaluation.

- Visiting all the search space is one criteria for terminating GA.
- Another approach for termination is to check a specific gene in chromosomes, if a specified gene reaches a pre defined percentage in chromosomes and one specific allele gen reaches to a pre defined percentage in population than GA ends. This method either can be performed by using phenotypes or genotypes.
- A criteria based on improvement in generations' fitness value is also used. If a number of generations' total fitness does not change significantly than GA stops [32].

In this study, a two stage termination criteria is used. (i) Total fitness value of generations are checked and compared to each other and if it does not change significantly, than GA terminated. Here, the last 5 generations' total fitness are compared and if the difference between maximum and minimum of the total fitness is less than 4, it is assumed that generations are not improving significantly and GA is terminated (ii) But in the probabilistic nature of GA, first condition might not happen all the time. Hence, an upper limit to iteration number is employed. The upper limit is chosen as 100 iterations.

If GA stops in first stage than last populations' best individual is declared as optimum solution. If GA stops in second stage, then the individual with the maximum fitness value in all populations is declared as optimum solution.

#### 5.3.9 Results of optimization

Fitness functions are mentioned in section 1.3.2 have different area of interest hence, results of optimization depends on the function used. Results are shown with respect to fitness functions. Here, one thing has to be kept in mind that genetic algorithms does not always converges to an optimal solution and does not always gives the global minimum but GA gives sub optimal points.

##### 5.3.9.1 Results for fitness function 1

In *fitness function 1*, GA did not terminate at the first stage of termination criteria and kept running for 100 generations. Hence, maximum fitness valued chromosome of all generations is declared as optimum solution according to *fitness function 1*. The change in total fitness of all generations and the change of maximum fitness of generations over one hundred generations are shown in figures 5.12 and 5.13.

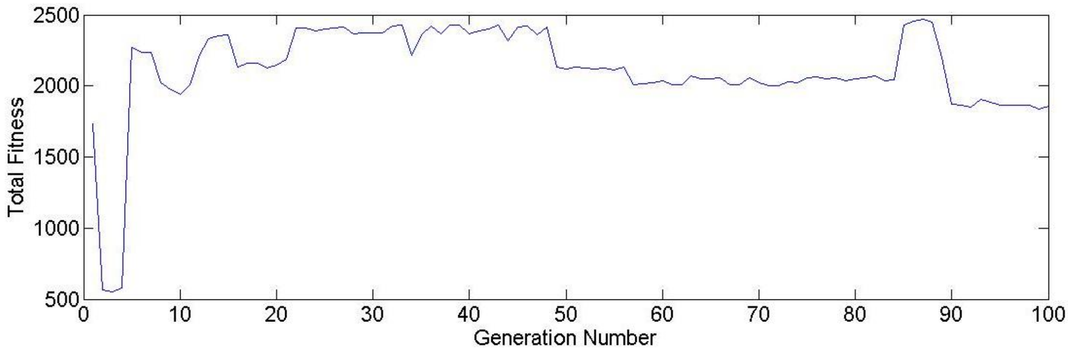


**Figure 5.12 :** Total fitness of *fitness function 1* over generations.

In figure 5.12 there are some flat regions in total fitness but it is not flat enough to terminate the GA. Hence, 100 generations are generated. Maximum fitness values of each generation over generation number are shown in figure 5.13.

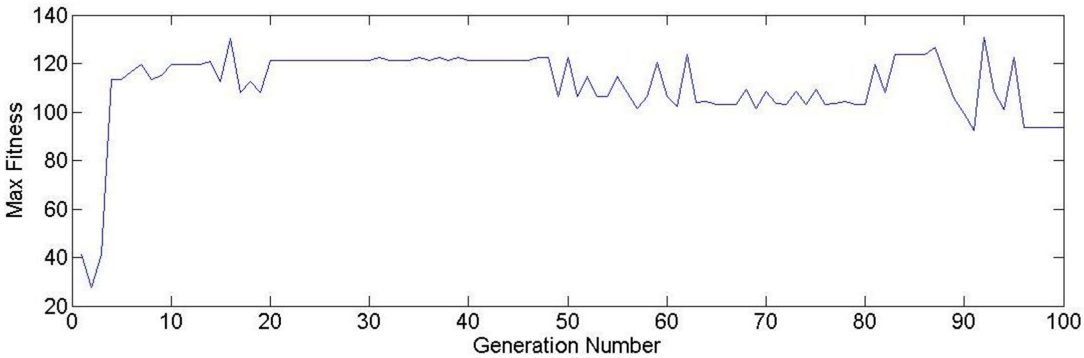


**Figure 5.13 :** Maximum fitness of *fitness function 1* over generations.

In figure 5.13 maximum fitness value of generations are converges around 120 and has maximum fitness value in 92th generation and fitness value is 130,5. Design parameters corresponding to this member are shown in terms of problem parameters;

$$k = 6071{,}4\ N/m \tag{5.19}$$

$$c = 400\ Ns/m \tag{5.20}$$

$$x_s = 0{,}0871\ m \tag{5.21}$$

$$z_s = 0{,}1513\ m \tag{5.22}$$

$$x_d = 0{,}0871\ m \tag{5.23}$$

$$z_d = 0{,}1153\ m \tag{5.24}$$

Figure 5.14 shows the result of optimization for fitness function1 and figure 5.15 shows results of optimization for fitness function1 and results of mathematical model at the same time.

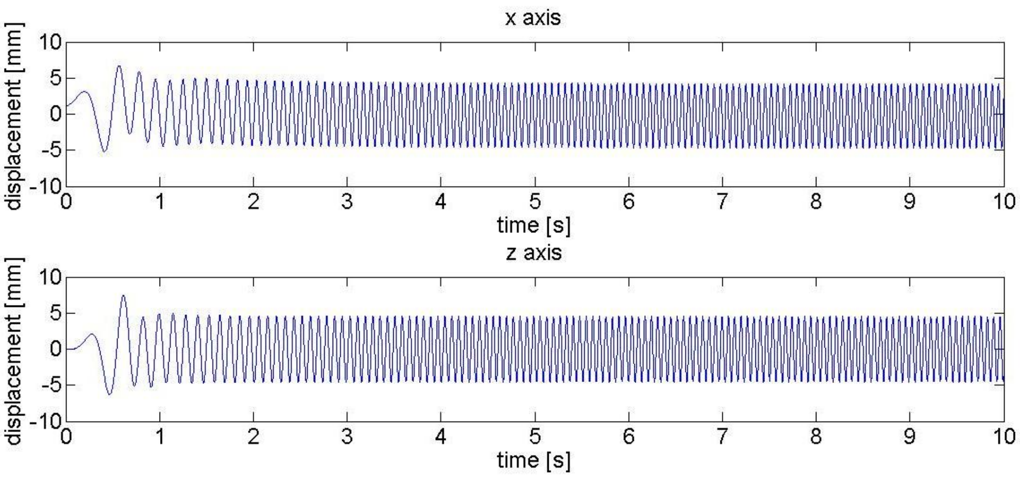


**Figure 5.14 :** Optimization results for *fitness function 1*.

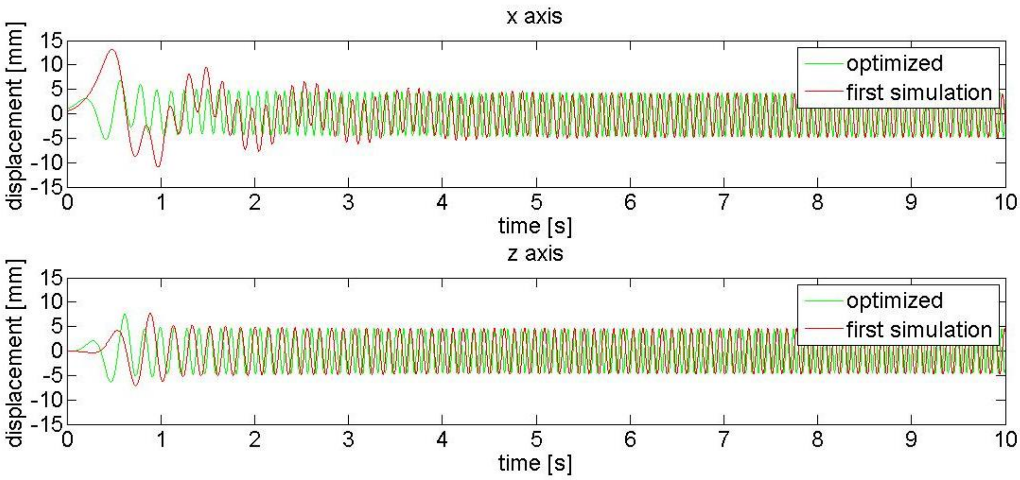


**Figure 5.15 :** First simulation and optimized results with *fitness function1*.

Figure 5.15 clearly illustrates that optimization process decreased amplitudes significantly by optimizing design variables. This result is obtained with *fitness* *function 1*, which is meant to minimize the radius of the wobbling motion around center. Figure 5.16 compares first simulation and optimization result by means of wobbling motion. It is clear that wobbling motion is decreased significantly. This optimization function generally improved the transient response of the system as well as decreasing the wobbling.

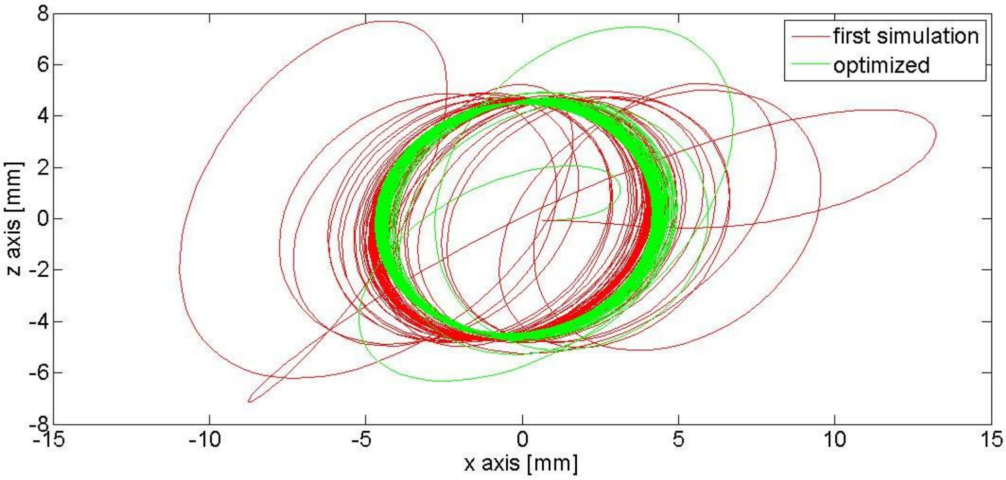


**Figure 5.16 :** Simulation and optimization results in terms of wobbling motion.

#### 5.3.9.2 Results for fitness function 2

*Fitness function 2* is dealing with steady state vibrations as mentioned in section 5.3.2. GA terminated on stage one termination criteria at 71th generation. Total fitness values over generations plot is shown in figure 5.17.

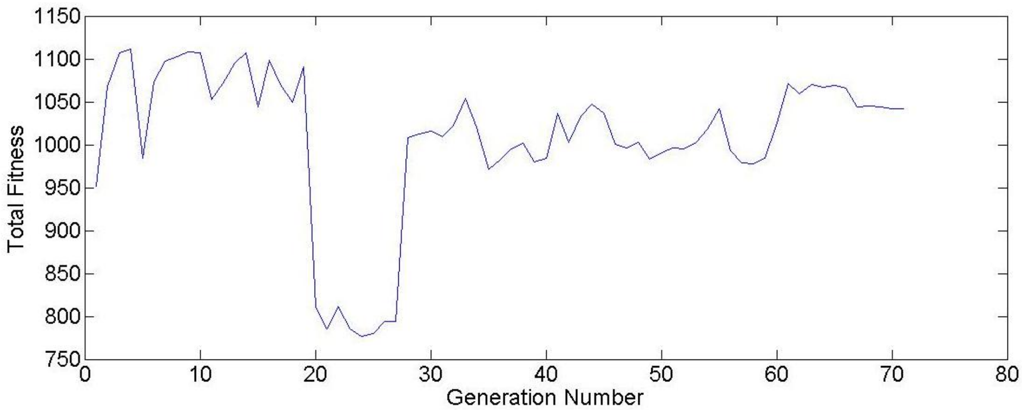


**Figure 5.17 :** Total fitness of *fitness function 2* over generations.

Total fitness of generations are converged at 71th generation. Maximum fitness of *fitness function 2* is 55,89 in the 67th generation. Design parameters are shown in terms of problem parameters;

$$k = 7142{,}9\ N/m \tag{5.25}$$

$$c = 150\ Ns/m \tag{5.26}$$

$$x_s = 0{,}0729\ m \tag{5.27}$$

$$z_s = 0{,}2126\ m \tag{5.28}$$

$$x_d = 0{,}0443\ m \tag{5.29}$$

$$z_d = 0{,}1m \tag{5.30}$$

It is clear from figure 5.18 that transient vibrations are not very good and transient vibrations of z axis are worse than first simulation. It is because *fitness function 2* is interested with the steady state region not with transient region.

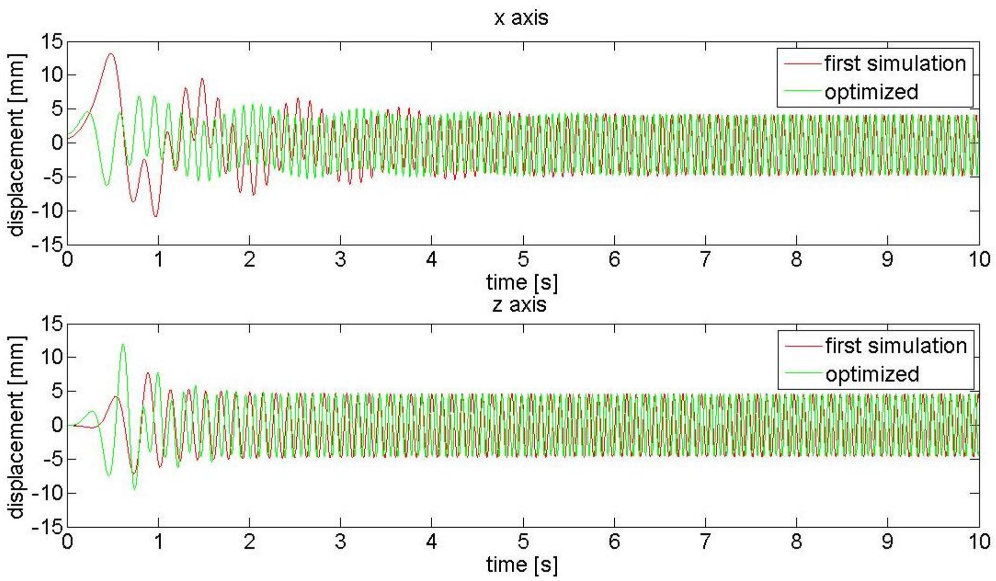


**Figure 5.18 :** First simulation and optimized results with *fitness function 2*

However, there are no significant improvements in steady state vibrations. Amplitudes of steady state has decreased 0,1 mm at max. The reason for this is, first simulations design parameters are same as the actual washing machine parameters and it already has some good steady statecharacteristics and not far from optimum design parameters. We conclude that there is no room left for the optimization in steady-state behavior with the components used in machine design.

#### 5.3.9.3 Results for fitness function 3

*Fitness function 3* is interested in both steady state and transient vibrations. GA terminated at the first stage of termination criteria and stopped at 14th generation. Maximum fitness value of *fitness function 3* in 14th generation is 22,14. Design parameters of optimization are shown in terms of problem parameters;

$$k = 5000\ N/m \tag{5.31}$$

$$c = 364{,}29\ Ns/m \tag{5.32}$$

$$x_s = 0{,}0729\ m \tag{5.33}$$

$$z_s = 0{,}1717\ m \tag{5.34}$$

$$x_d = 0{,}0871\ m \tag{5.35}$$

$$z_d = 0{,}1459\ m \tag{5.36}$$

In figure 5.19 it is obvious that vibration amplitudes decreased significantly on both axes.

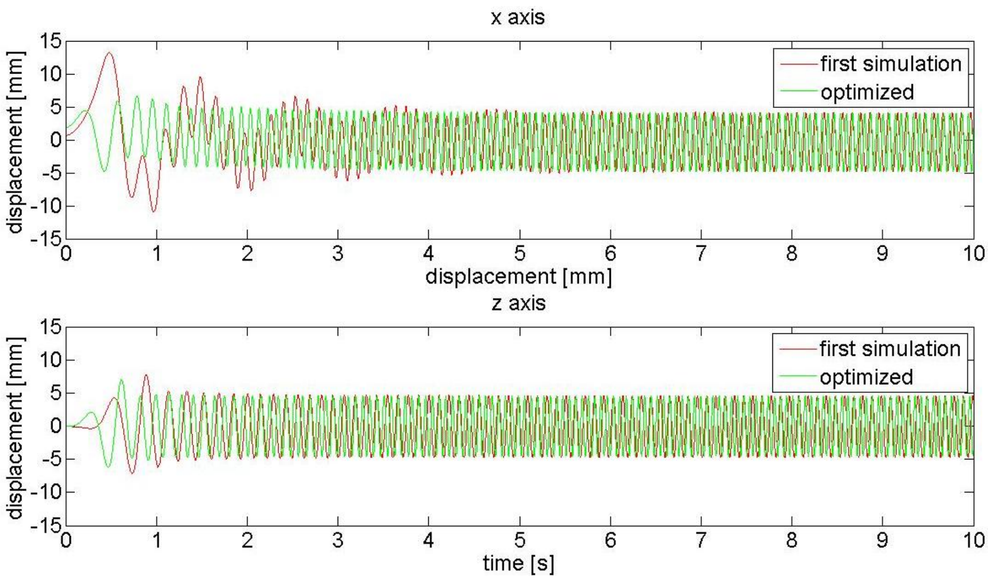


**Figure 5.19 :** First simulation and optimized results with *fitness function 3*.

All three fitness functions in GA performed well with specified equations. In order to see the differences between all results with three different fitness functions clearly, all results are plotted on each other in figure 5.20.

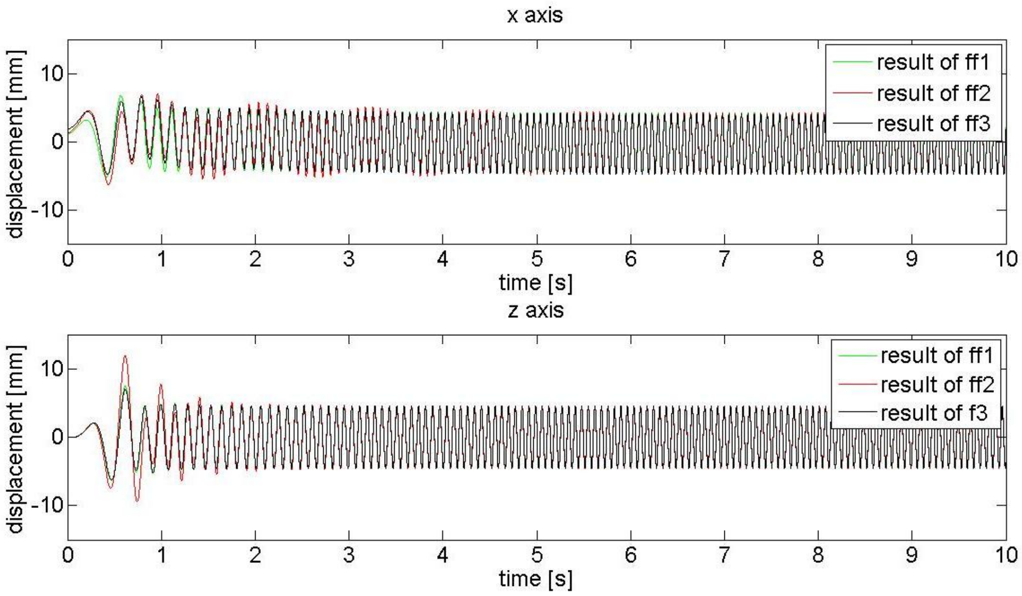


**Figure 5.20 :** Comparison of all optimized results.

"ff1", "ff2" and "ff3" denotes *fitness function1*, *fitness function2* and *fitness function3* respectively. Figure 5.20 shows that, GA optimization with different fitness functions gives closer results. *Fitness function1*'s and *fitness function 3*'s results are very close to each other because they both interested in similar conditions. Fitness function 2's result has bigger amplitudes in transient region comparing it to others, because fitness function 2 is interested only in steady state vibrations. Hence, damping coefficient obtained form second case is bit lower. Comparison of critical points displacements with first simulation is shown in table 5.2

**Table 5.2 :** Optimization results comparison.

| # | Simulation | Fitness Function 1 | Fitness Function 2 | Fitness Function 3 |
|---|---|---|---|---|
| **x transient** | | | | |
| **max [mm]** | 10,93 | 6,74 | 6,93 | 6,6 |
| **min [mm]** | -10,46 | -5,2 | -6,3 | -4,8 |
| **z transient** | | | | |
| **max [mm]** | 7,75 | 7,47 | 11,89 | 7,01 |
| **min [mm]** | -7,13 | -6,3 | -9,45 | -6,2 |
| **x steadt-state [mm]** | ±4,45 | ±4,43 | ±4,4 | ±4,43 |
| **z steady-state [mm]** | ±4,6 | ±4,5 | ±4,5 | ±4,5 |

In conclusion all three fitness functions performs minimization process and with a little difference they all give close results which is in the nature of GA. Every run in GA is unique because of probabilistic nature of it but at the end GA gives sub optimal points.

# 6. CONCLUSIONS

In this thesis work, first washing machine system, its components and interaction between components are presented. Then, some simplifications and assumptions are introduced and a simple mathematical model, for representing vibration characteristics of washing machines on horizontal and vertical axes, is derived. In order to validate the mathematical model, vibration measurements performed on washing machine. For obtaining displacement data from acceleration data and obtaining rotational speed, a data acquisition test-rig is built. During the data acquisition phase, signal processing techniques, such as filtering, integration and instantaneous frequency estimation are utilized. Results of mathematical model and the measurement performed are compared in validation section and it is concluded that, model is yielding sufficiently close data to real vibration characteristics to proceed for an optimization process. A general classification of optimization methods for parameter optimization is presented and *genetic algorithm optimization* is selected amongst them. In order to minimize the vibrations of washing machine, genetic algorithm is implemented on the previously derived mathematical model.

In application of GA to the vibration minimization problem, a design vector is defined consisting of spring stiffness, damping coefficients and joining positions of spring and shock absorbers and a search space is introduced. Three different fitness functions are derived for minimization purpose, which are interested in; transient region maximum displacement of washing machine tub from the origin, steady state vibrations and both transient and steady state. Also, to avoid stepping condition, a constraint is introduced to GA in order to prevent stepping condition in optimized solutions. It is concluded that *fitness function 1* minimized transient vibrations by using both axes' displacements for calculating the radius of wobbling motion but this may mislead because if one axis' vibration is so small and the other one can be high at the same time. Hence, considering single axis's vibrations, the amplitudes can remain high while other axis is small. Results of *fitness function 2*, revealed that steady state displacements are not significantly effected by design variables used in optimization and no significant improvements are observed. Hence, steady state vibration minimization can be excluded from optimization process. On the other hand, *fitness function 3* performed minimization by considering both axes' steady state and transient vibration amplitudes at the same time. Results of fitness function1 and fitness function3 have similar performance but *fitness function 3* is more successful in minimizing both axes' transient vibrations at the same time. It is shown that GA performs well in finding optimal/ sub-optimal solutions in search space. The design variables obtained from optimization process needs to be validated by experimental test. This validation of optimization results and implementing it to development phase of washing machine is left for development engineers working in the field of home appliance and it is an issue of technology realization. This thesis shows a concept proof that a simple mathematical model and an optimization with genetic algorithm implementation can be involved in design process easily and effectively.

In further studies mathematical model of washing machine can be expanded with more degrees of freedom, simplifications such as linear assumptions can be replaced with nonlinear ones and structural effects (i.e. structural damping) can be counted in. On the other hand, in order to minimize vibrations, active control strategies can be employed. For implementation of active control to actual washing machines, feasible actuators and active dampers can be studied.

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