ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

MULTI-OBJECTIVE OPTIMIZATION OF DYNAMIC BEHAVIOUR OF AUTOMOTIVE CLUTCH SYSTEM AND POWER TRANSMISSION

Ph.D. THESIS

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Department of Mechanical Engineering

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JULY 2015

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

OTOMOTİV DEBRİYAJ SİSTEMİNİN VE GÜÇ İLETİMİNİN DİNAMİK DAVRANIŞININ ÇOK AMAÇ FONKSİYONLU OPTİMİZASYONU

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Onur Ozansoy, a **Ph.D.** student of ITU **Graduate School of Science Engineering and Technology** student ID 503052007, successfully defended the **thesis/dissertation** entitled "**MULTI-OBJECTIVE OPTIMIZATION OF DYNAMIC BEHAVIOUR OF AUTOMOTIVE CLUTCH SYSTEM AND POWER TRANSMISSION**", which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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To my family,

FOREWORD

I would like to dedicate this thesis to my lovely wife Ceren Daver OZANSOY, my parents and my whole family who never stopped believing me to complete this long study and I would like to express my special thanks to my advisors Prof. Dr. Talat TEVRUZ and Prof. Dr. Ata MUGAN that they have great contribution on this project.

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July 2015 Onur OZANSOY

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ABBREVIATIONS

DMF	**:** Dual Mass Flywheel
Hz	**:** Hertz
RPM	**:** Revolution per Minute
GHz	**:** Giga Hertz
SMF	**:** Solid Mass Flywheel

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MULTI-OBJECTIVE OPTIMIZATION OF DYNAMIC BEHAVIOUR OF AUTOMOTIVE CLUTCH SYSTEM AND POWER TRANSMISSION

SUMMARY

Optimum design of a clutch and dual mass flywheel system is investigated. The clutch systems are generally studied by considering rotational vibrations of engines and by assuming that they are usually exposed to axial vibrations. They generally do not have any specialized equipment to damp axial and rotational vibrations due to the following two reasons: first, the package area of the system is very limited in clutch systems and there is no available space to put some springs and dampers to reduce such vibrations. Second, these axial vibrations are normally very insignificant to be considered in the analyses and most of such vibrations are damped by the diaphragm spring and cushion springs. Nonetheless, this may sometimes lead to some unexpected results in the powertrain system such as the rattle noise on clutch itself or transmission to solve which global optimization techniques are employed in this study. Genetic algorithm methods have been choosen as the global optimization algorithm. The components in the clutch system are modeled analytically. Then, by considering the pedal characteristics and vibrations of pressure plate as objective functions, a multiobjective (Pareto) optimization problem is solved. It is shown that analytical models agree well with the experimental measurements and vibrations in the clutch system can be reduced significantly by optimization.

In the study, subjective and objective measurements have been completed and transfer path has been found firstly. Mechanics of the vibration has been understood and optimization parameters have been identified. Once the parameters have been decided, other systems, which are affected by these parameters, have identified. It is understood that the clutch pedal efforts are affected significantly which has the major importance of the drive environment and driver comfort.

Once these initial studies were completed, mathematical models for clutch pedal efforts and the vibration phenomenon have been built and simulation tools have been developed. Measurement and simulation results have been compared and it has been observed that they are very close to each other.

During this period, another test has been performed to identify axial resonance frequencies of the clutch and flywheel system, and it was observed that the frequency of the rattle noise coming from transmission and resonance frequency of the clutch and flywheel system are very close to each other. This also proves that first assupmtions which is the clutch system amplifies the engine vibrations on the $0,5^{th}$ engine order especially on the resonance frequency area (2500 to 3000 RPM) on the pressure plate. The excessive axial vibrations on the clutch pressure plate causes torque truncation on the transmission which causes this rattle noise phenomenon during the clutch engagement.

Once root cause of the rattle noise have been identified and the simulation models have been built, performed and comparable results handled, optimization algorithm has been built and system optimized considering genetic algorithm rules by using Matlab Simulink Gnetic algorithm toolbox.

After the optimization process has been completed, a prototype has been built with the optimized parameters (considering the tolerances and producability) and noise measurements repeated with the new parts. It has been observed that the system with optimized clutch system showed much better results (very close to simulation results) in objective and subjective measurements.

OTOMOTİV DEBRİYAJ SİSTEMİNİN VE GÜÇ İLETİMİNİN DİNAMİK DAVRANIŞININ ÇOK AMAÇ FONKSİYONLU OPTİMİZASYONU

ÖZET

Bu tez çalışması kapsamında, dinamik sistemlerin optimizasyonu incelenmiş, optimizasyon işlemi için en doğru yöntem seçilmiş ve bu yöntem dinamik bir problemi çözmek maksadıyla kuru sürtünmeli bir debriyaj sistemi üzerinde çalışılmış, gerekli modeller kurulup, sonuçlar fiziksel ölçümlerle karşılaştırılmıştır.

Yukarıda bahsedildiği gibi dinamik sistem olarak kuru sürtünmeli, bir otomobil debriyaj sistemi üzerinde çalışılmış ve çift kütleli volan sistemi de çalışmaya dahil edilmiştir. Debriyaj ve çift kütleli volan sistemleri özellikle yuvarlanma titreşimlerini sönümlemek için dizayn edildiği halde, şimdiye kadar fazla incelenme şansı bulmamış eksenel tireşimleri üzerinde durulmuştur.

Çalışmanın ilk böülümünde yapısal optimizasyon terimi üzerinde durulmuş, optimizasyon (en iyileme) kelimesinin tanımı yapılmış ve bir örnekle anlatılmıştır. Yine bu bölümde amaç denklemleri, dizayn değişkenleri ve durum değişkenleri gibi optimizasyon ile ilgili terimlerden bahsedilmiş ve optimizasyon problem üzerindeki yerleri ve önemleri açıklanmıştır. Takip eden bölümde kısıtlı ve kısıtsız optimizasyon problemlerinden bahsedilmiş ve aralarındaki farklar matematiksel olarak anlatılmıştır. Daha sonra dinamik sistemlerin optimizasyonunun tanımı yapılmış ve dinamik ve optimizasyon arasındaki ilişki anlatılmıştır. Bu bölüm içerisinde optimizasyon probleminin dinamik cevabı, simülasyon modelinin önemi ve olası çözüm yöntemleri matematiksel denklemleri ile açıklanmıştır.

Takip eden bölümde, çok amaçlı optimizasyonun tanımı yapılmış ve matematiksel denklemleri kurulmuştur. Müteakiben, bu tür problemleri çözmek için kullanılabilecek metodlara değinilmiş ve bunlar arasından pareto optimum setin oluşturulmasının önemi anlatılmıştır. Pareto optimum setin kurulmasında en başarılı yöntemler global optimizasyon teknikleridir. Bu nedenle, bu bölümden sonra global optimizasyon teknikleri üzerinde durulmuş ve bu teknikler belirli bir aile ağacının üzerinde gösterilmiştir. Bu ağaç üzerinde bu tekniklerin en önemlilerinden bahsedilmiş ve takip eden iki bölümde tavlama simülasyonu (simulated annealing) ve genetik optimizasyon teknikleri matematiksel denklemleri ile ayrıntılı olarak açıklanmıştır.

En önemli ve sürekli gelişmekte olan genetik algoritmalar üzerinde özellikle durulmuş, bu yöntemlerin tiplerinden bahsedilmiş, temelleri anlatılmış, algoritmaların daha iyi anlaşılması için çeşitli tanımlamalar yapılmıştır. Ayrıca genetik algoritmalar içerisinde genlerin (genomes) önemi anlatılmış ve tanımları yapılmıştır.

Takip eden bölümde literatür taraması yapılmış, bunun için çeşitli optimizasyon yöntemlerinin kullanıldığı, özellikle dinamik sistemlerin optimizasyonunun

araştırıldığı ve tezin konusu olan debriyaj sistemlerinin ve bu sistemlerin genel araç ses ve titreşimine etkisinin araştırıldığı bilimsel makaleler incelenmiştir.

Literatür araştırmasından sonra, tezin ana konusu olan debriyaj sisteminin öncelikle tarihsel gelişimi incelenmiş ve bu tarihsel gelişim içerisinde kullanılmış değişik debriyaj tiplerinden bahsedilmiştir. Bununla beraber debriyaj sistemleri, kullanım şekilleri, titreşim sönümleme özellikleri hakkında yeni bir literatür araştırılması yapılıp paylaşılmıştır.

Bu girişten sonra debriyaj sisteminin araç üzerindeki fonksiyonu anlatılmış ve debriyaj sistemindeki alt, üst merkez, debriyaj baskı ve balatası, volan vb. sistemi oluşturan parçalar incelenmiş ve görevleri açıklanmıştır. Sistemin açıklanmasına müteakip, projenin amacı, kapsamı ve metodolojisi açıklanmış ve arkasından çalışma planı ve problem tanımı yapılmıştır.

Bu bölümden sonra ise tezin amacı doğrultusunda kullaılmış test rig'leri, test ekipmanları ve ölçümler için kullanılmış olan sensörler ve özellikleri açıklanmıştır. Tez kapsamında, problemin kök sebebini anlayabilmek için öncelikle araç üstünde çeşitli enstümantasyonlar yapılmış ve ses problemine neden olan kök sebebin araştırılması amaçlanmıştır.Bu ölçümlerden çıkan sonuçlara göre bulunan titreşim frekansının incelenmesi amacı ile debriyaj ve volan montajı bir titreşim test ekipmanına monte edilmiş ve sistemin modal frekansları tespit edilmeye çalışılmıştır. Buradan çıkan sonuçlar göstermiştir ki debriyaj sisteminin eksenel yönde sahip olduğu modal frekanslar araç üstünden alınan ölçümlerde tespit edilen titreşim frekansı ile örtüşmektedir. Bu da sorunun kaynağının motor titreşimleri olsa da bu titreşimlerin özellikle debriyaj sisteminin doğal frekansları ile çakıştığı bölgelerde probleme yol açtığı ispatlamıştır.

Bununla berbaer, eksenel titreşimlerin ortadan kaybolması halinde sorunun giderilip giderilemeyeceği yönünde de araştırma yapılmış ve bu amaçla debriyaj baskı plakasının üzerine 3 adet sürtünme klibi eklenmiştir. Her ne kadar bu klipler seri imalat koşullarında çalışamasa da (sürtünme nedeniyle ömürleri çok düşük olmaktadır) ilk denemeler esnasında görevlerini yerine getirmiş ve debriyaj üzerindeki titreşimleri yok etmeyi başarmışlardır. Eksenel titreşimlerin yok olması ile beraber ses proble mi de ortadan kalkmıştır. Bu da probleme temelde debriyaj sistemindeki eksenel titreşimlerin yol açtığını ispatlayan diğer bir deneme olarak okuyucu ile paylaşılmıştır.

Bu kısma kadar problemi oluşturan sistem ve problemi çözmek için kullanılabilecek metodlar, test ekipman ve enstrümantasyonları açıklanmıştır. Bu bölümden sonra ise problemin tanımına geçilmiştir. İlk paragraflarda belirtildiği gibi, debriyaj sistemleri dönme (yuvarlanma) titreşimlerini sönümlemek için dizayn edilir, fakat motordan gelen eksenel titreşimlere de maruz kalırlar. Fakat bu titrşimleri sönümlemek için herhangi bir sönümleme elemaı ile donatılmazlar. Bunun belli başlı iki sebebi vardır. Öncelikle debriyaj sistemlerinin paket alanı çok dardır ve eksenel bir titrşimi sönümlemek için kullanılabilecek yay ve damper sistemlerini bu alana sığdırmak için gerekli olan dizayn çözümleri mevcut değildir. İkinci olarak ise motor üzerinden gelen eksenel titreşimler, dönme titreşimleri ile kıyaslandığında oldukça düşük mertebede kalırlar ve çoğu zaman dizayn sırasında göz önünde bulundurulmaları gerekmez.

Herşeye rağmen, bu durum, araç güç iletim sistemleri üzerinde çeşitli ses ve titreşim problemleri gibi beklenmeyen problemlere sebep olabilir. Bu çalışmada da böyle bir durum söz konusu olmuş ve problemin çözümü için global optimizasyon tekniklerinden yararlanılmıştır.

Bahsi geçen optimizasyon yöntemlerini kullanabilmek için öncelikle titreşime yol açan fiziksel sistemin matematiksel modeli oluşturulmuş, bu modele ait titreşim denklemleri kurulmuş ve bu denklemleri çözebilecek simülasyon kodları MATLAB ve Simulink yazılımları vasıtası ile gerçekleştirilmiştir.

Bu modeller oluşturulurken simülasyona dizayn parametreleri değişken olarak giren parçaların etkilediği başka bir önemli debriyaj parametresi olan debriyaj eforlarının etkilendiği gözlemlenmiş ve optimizasyon çalışması, bu parametreyi de içine alacak şekilde çok amaçlı optimizasyon olarak genişletilmiştir.

Bahsi geçen pedal efor ölçümleri hem araç üstünde hem de tez içerisinde anlatılan test rig'inde gerçekleitirilmiş olup, bu parametre için yazılmış olan simülasyon ile de sağlaması yapılmış ve optimizasyon algoritması içerisinde kullanılabileceği ispatlanmıştır. Pedal eforu ile beraber simülasyon programı hazırlanan titreşim denklemlerinin de doğru kurulup kurulmadığı, araç üzerinden alınan ölçüm sonuçları ile karşılaştırılmış ve doğruluğu ispatlanmıştır.

Simülasyonların oluşturulması ve çalıştıklarının ispatından sonra "Çok Amaçlı Genetik Optimizasyon" yöntemi kullanılarak sistem optimize edilmiş ve yeni dizayn parametreleri elde edilmiştir. Bu parametreler seri imalat koşulları ile üretilebilecek büyüklükler le modifiye edilerek, önce simülasyonlar üzerinde greçekten sorunun iyileşip iyileşmediği kontrol edilmiş, iyileştiği görülünce de parça fiziksel olarak üretilip bir araç üzerinde denenmiş ve problemin büyük ölçüde iyileştiği ispatlanmıştır.

1. INTRODUCTION

Engineering science covers many areas of activity such as analysis, design manufacturing, sale research and development. In the world developing fast, it is not satisfactory anymore to develop a system that only works. The important one is to develop the optimum system. We can give several meanings to the concept "optimum" such as lightest, covering the least space; cost the minimum, the most productive the most stable etc… Such a system design can be formulated as an "optimization problem" and can be solved. With its simplest definition, an optimization problem is taking one of the meaning mentioned above as the objective expressing it as a mathematical function-namely constructing a mathematical model, and finding the minimum (sometimes the maximum)values of the function within the constraints of the problem [1].

Before going any deeper, let's try to define what a structure is and what structural optimization is. In mechanics, the structure is defined as "any assemblage of materials which is intended to sustain loads" by J.E. Gordon [2].With the basic and simplest explanation, Optimization means making things the best. Thus, structural optimization is the subject of making an assemblage of materials sustains loads in the best way.

1.1 Purpose of Thesis

Vibrations in a power transmission system may lead to detrimental rattle noise phenomena and structural failures. On the other hand, solving the optimization of a dynamic system is considerably difficult. Axial transient behavior of clutch system is commonly ignored in literature. Motivated by this fact, this study is undertaken to optimize the dynamic behavior of a clutch system in terms of axial vibrations and avoid the rattle noise phenomena in power transmission systems. The purpose of thesis is to model an automotive dry clutch system as a dynamic structural model and optimize the dynamic model by defining an optimization problem using optimization objective functions and optimization limitations by considering the transient response of the power transmission through the clutch system. By this way, it is proven that the issues arising in practice dynamic models can also be solved as an optimization problem. In particular, optimization of components in a power transmission is achieved and rattle noise phenomena is examined. It is shown that the rattle noise can be avoided by solving an appropriate optimization problem. Numerical and experimental results are presented to show the advantages of optimized system over the conventional one.

Furthermore, clutch pedal efforts are also examined since the same parameters with transmission axial vibrations are affected. Model for clutch pedal efforts is developed and proved with the physical test results. Once the model is proven, this model is also added to multi-objective problem and solved with vibration problem together simultaneously.

Phisical tests are also repeated with the results of multi-objective problem. Tests showed that outcomes are inline with the simulation results and rattle phenomena is improved significantly.

1.2 Structural Optimization

In order to fix the ideas, think of a situation where a load is to be transmitted from a region in space to a fixed support as in Figure 1.1, we want to find the structure that performs this task in the best possible way. However, to make any sense out of that objective we need to specify the term "best." The first such specification that comes to mind may be to make the structure as light as possible, i.e., to minimize weight. Another idea of "best" could be to make the structure as stiff as possible, and yet another one could be to make it as insensitive to buckling or instability as possible. Clearly such maximizations or minimizations cannot be performed without any constraints. For instance, if there is no limitation on the amount of material that can be used, the structure can be made stiff without limit and we have an optimization problem without a well-defined solution. Quantities that are usually constrained in structural optimization problems are stresses, displacements and/or the geometry. Note that most quantities that one can think of as constraints could also be used as measures of "best," i.e., as objective functions. Thus, one can put down a number of measures on structural performance—weight, stiffness, critical load, stress, displacement and geometry—and a structural optimization problem is formulated by picking one of these as an objective function that should be maximized or minimized and using some of the other measures as constraints [3].

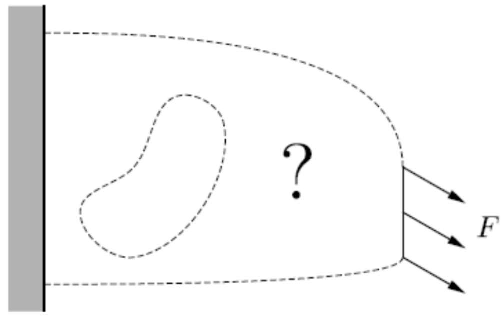

Figure 1.1 : Structural Optimization Problem. Find the structure which best transmits the load F to the support [3].

It's the best to give some basic terms and their explanations that will be used frequently in upcoming parts such as "objective function, design variable, state variable" before going one step more. These function and variables are always present in a structural optimization problem [3].

Objective (Cost) function (f): A function used to classify designs. For every possible design, f returns a number which indicates the goodness of the design. Usually we choose f such that a small value is better than a large one (a minimization problem). Frequently f measures weight, displacement in a given direction, effective stress or even cost of production.

Design variable (x): A function or vector that describes the design, and which can be changed during optimization. It may represent geometry or choice of material. When it describes geometry, it may relate to a sophisticated interpolation of shape or it may simply be the area of a bar, or the thickness of a sheet.

State variable (y): For a given structure, i.e., for a given design x, y is a function or vector that represents the response of the structure. For a mechanical structure, response means displacement, stress, strain or force.

So, we can define the *structural optimization problem* by using the information above as; "Minimizing the $f(x,y)$ with respect to x and y and subject to behavioral constraints on y, design constraints on x and equilibrium constraints.

1.2.1 Constrained and unconstrained optimization

Let's define an optimization problem as:

Minimize $f(X)$

Subject to $X \in \Omega$

The function f is the cost function that we wish to minimize as explained above. That function $f : \mathbb{R}^n \to \mathbb{R}$ is a real valued function. The vector X is an n vector of independent variables, that is; $X = [X_1, X_2, \dots, X_n]^T \in \mathbb{R}^n$. The set Ω is a subset of $\mathbb{R}^n$ called *constrained set* or *feasible set.*

The optimization problem can be seen as a decision problem that involves finding the "best" vector X of the variables over all possible vectors in Ω "best" vector means that one that results in the smallest value of the objective function. This vector is called the minimizer of f over Ω It is possible that there may be many minimizers. In this case, finding any of the minimizers will be sufficient.

There are also optimization problems that require maximization of the objective function. These problems, however, can be represented in the above form because maximizing f is equivalent to minimizing$-f$. Therefore we can focus our attention to minimization problems without loss of generality.

The above problem is a general form of a *constrained* optimization problem, because the variables are constrained to be in the constraint set Ω. If $\Omega = \mathbb{R}^n$, then we refer to problem as an *unconstrained* optimization problem [4].

1.2.2 Optimization of structural dynamic systems

Many structures in aerospace, aircraft, automotive, mechanical and civil engineering applications are subjected to transient dynamic loads. Therefore it is important to account for such loads in the design process. It is also important to optimize the design of structures so as to minimize their total cost while accounting for all the safety and performance requirements dictated by the design codes. [5].

Before going into optimization of dynamic systems, it's better to define the "dynamics" and its relations to optimization.

Dynamics and optimization

At its most basic level, mechanics is the study of forces and their effects. Elementary mechanics is divided into statics, the study of objects into equilibrium and dynamics, the study of objects in motion [6]. In other explanation, Dynamics is the subsection mechanics that deals with the structures which are in motion under the effect of forces [7]. So the dynamic system is a system which is in motion under the effects of various forces such as external forces, gravity, inertia etc…

Dynamic response optimization problem

Formulation of an optimal design problem requires identification of design variables that describe the system, a cost function that needs to be minimized and all performance and safety constraints for the system. It can be seen that this formulation of the problem depends on the type of application and objectives to be achieved. Those might be shape, sizing or topology design problem [5].

Firstly, let's define a general mathematical model for design optimization of structural systems subjected to dynamic loads. This model also encompasses optimal design and control of nonlinear structures.

Simulation model:

Mathematical model is very important for the simulation of the system in any design optimization problem formulation. Let X be an n dimensional vector to represent the "design variables" for the problem. Design variables might be shape, sizing and topological variables, and these also might be the parameters which are related to the control variables for the system. And let $Z(t)$ be a κ dimensional vector that represents the "state variables" for the problem. These usually represent the generalized displacements for the finite element model of the structure. The equation of motion for the system to determine the state variables can be written in the general form as

$$P_I + P_D + P_M = P \tag{1.1}$$

P_1 is the inertial force due to the mass of the structure or external masses. P_D is the damping force related to the velocity of the material points or external dampers at the generalized coordinates. P_M is the internal force due to the relative displacement of the material points of the system. P is the force due to the external disturbance or contact with other bodies. Equation (1.1) is usually 2nd order differential equation in terms of the generalized coordinates. It may also be a nonlinear equation due to large deformation, large displacement or nonlinear constitutive models for the material behavior (for example nonlinear spring or damper characteristics, etc…). Each terms are depends on the state and design variables. Therefore, the solution of the equation (for example response) depends on the design of the system. However, in general, it is not possible to obtain an explicit expression for $Z(t)$ in terms of X. That means, $Z(t)$ is an implicit function of the design variables X. But once X is specified, $Z(t)$ can be calculated by integrated Eq. (1.1). In many structural applications, the assumptions of small displacements and deformations and elastic material behavior are reasonable. In such cases the state variable vector $Z(t)$ is governed by the following nonhomogeneous linear system of 2nd order differential equations [8]:

$$\boldsymbol{M}(\boldsymbol{X})\ddot{\boldsymbol{Z}}(\boldsymbol{t}) + \boldsymbol{C}(\boldsymbol{X})\dot{\boldsymbol{Z}}(\boldsymbol{t}) + \boldsymbol{K}(\boldsymbol{X})\boldsymbol{Z}(\boldsymbol{t}) = \boldsymbol{P}(\boldsymbol{X}, \boldsymbol{t}) \qquad \textbf{(1.2)}$$

Where the coefficients$M(X)$, $C(X)$ and $K(X)$ are matrixes with the dimensions κ x κ which are also independent of time. And $P(X,t)$ is the generalized force vector of dimension κ x 1 which is independent of X for many applications (for example when the body forces are neglected). We can also write the initial conditions as $\dot{Z}(0) = \dot{Z}_0$ and $Z(0) = Z_0$ [8-9].

Cost function

A cost functional for the problem that needs to be minimized is expressed in general, as a functional of the state and design variables.

$$\boldsymbol{f} = \boldsymbol{c_0}(\boldsymbol{X}, \boldsymbol{t}) + \int_{\boldsymbol{0}}^{\boldsymbol{T}} \boldsymbol{k_0}\big(\boldsymbol{X}, \boldsymbol{Z}(\boldsymbol{t}), \dot{\boldsymbol{Z}}(\boldsymbol{t}), \boldsymbol{t}\big)\boldsymbol{dt} \qquad \textbf{(1.3)}$$

This functional can be used to represent a variety of cost functions for the problem. For example the function c_0 can be used to represent mass (weight) of the structure and time of control (minimum time control problems). The integral part can be used to represent displacement at a point, stress at a point, average energy, or any other function involving state and control variables.

Constraints

The constraints for the problem can be divided into two types. These are integral type constraints and time dependent constraints. The integral type constraints sometimes also called functional constraints. A general representation of those type constraints can be seen below (Eq. 1.4).

$$\boldsymbol{g_i = c_i(X,t) + \int_0^T h_i(X, Z(t), \dot{Z}(t), t)dt} \begin{cases} = \mathbf{0}, & \boldsymbol{for\ i = 1, \dots, p'} \\ \leq \mathbf{0}, \boldsymbol{for\ i = (p'+1), \dots, m'} \end{cases} \quad \mathbf{(1.4)}$$

The constraints, that need to be imposed at each point of entire time interval $t \in [0, T]$, are called dynamic constraints. Those are written as (also called as point-wise constraints):

$$\boldsymbol{g_i = h_i(X, Z(t), \dot{Z}(t), t)} \begin{cases} = \mathbf{0}, \boldsymbol{for\ i = (m'+1), \dots, p} \\ \leq \mathbf{0}, \boldsymbol{for\ i = (p+1), \dots, m} \end{cases} \quad \mathbf{(1.5)}$$

Constraints in eq. (1.4) includes all performance or failure constraints that are independent of time such as explicit on the design variables, time average quantities, or state dependent constraints at a time point (converted to an integral form by use of Dirac delta function).

Eq. (1.5) includes dynamic response point-wise constraints, such as stress and displacement constraints. The functions c_i and h_i in eqs. (1.4) and (1.5) have appropriate forms depending on the constraint type.

Optimization model

The problem of optimal design of structural systems subjected to dynamic loads is now defined as:

Determine the design variable vector X that minimizes the cost functional of eq. (1.3) subject to constraints of equations. (1.4) and (1.5)

It is assumed that the design variables X are continuous and all the functions c_i and h_i for the problem are twice continuously differentiable with respect to their arguments.

Optimization procedures

Most of the computational algorithms for continuous variable design optimization problems involve the following basic steps:

1. An initial estimate of the design variables.
2. Calculation of all the problem functions.

3. Evaluation of the gradient of the cost function and the constraint functions (some algorithms require gradients of only the critical constraints while the others require gradients of all the constraints).
4. Definition of a sub-problem to determine the design improvement (search direction)
5. Calculation of the design improvement direction (search direction) by solving the sub-problem and determination of step size along the search direction.
6. Stopping criteria checks. Either stop or calculate a design change, update design variables and go to step 2

The computational steps can be summarized in the following general iterative prescription:

$$\boldsymbol{X^{(k+1)} = X^{(k)} + \alpha_k d^{(k)}; \quad k = 0, 1, 2, ...} \tag{1.6}$$

Where the superscript *k* represents the iteration number, $X^{(k)}$ is the crrent estimate of the optimum point, $\alpha_k d^{(k)}$ is a change in design, α_k is a step size, $d^{(k)}$ is a search direction calculated using function values and their gradients. And $X^{(0)}$ is an initial design estimate given by the designer.

1.2.3 Multi objective optimization

Basic important concepts of multi objective optimization was originated from Edgeworth and Pareto [52, 54] from the late of nineteenth century to the beginning of twentieth century, while a mathematical development was made by Cantor at almost the same ages. Today, we usually refer a solution of multi objective optimization to as a Pareto solution.

Mathematical foundations

Multi objective programming problems are formulated as follows:

$$\text{Minimize} \qquad \boldsymbol{f(x) = (f_1(x), \ldots, f_r(x))^T} \tag{1.7}$$

$$\text{Subject to} \qquad \boldsymbol{x \in X \subset \mathbb{R}^n} \tag{1.8}$$

The constraint set *X* may be given by:

$$\boldsymbol{g_j(x) \leqq 0, \quad j = 1, \ldots, m} \tag{1.9}$$

We can define the pareto solution as:

- If there is no $x \in X$ such that $f(x) \leq f(\hat{x})$, $\hat{x}$ is referred as Pareto optimal or simply to as a Pareto solution [10].

In this study, multi objective optimization will be especially introduced with particular reference to multi-objective programming

Methods to solve multi-objective programming (MOP) problems

The solution of multi-objective programming (MOP) problems can be classified according to how the designer (i.e. the decision maker, DM) manages preference information, that is how the designer (searches for and) chooses the optimal solutions.

There are three general approaches to solve multi-objective programming

(MOP) problems. They are:

- Pareto-optimal set generation methods
- Preference-based methods
- Interactive methods

In Pareto-optimal set generation methods, the decision maker chooses one of the alternative optimal solutions within the Pareto-optimal set after the Pareto-optimal set has been generated. In the preference-based methods the preferences of the decision maker are taken into consideration before the optimization process. In the interactive methods the preferences are considered as the optimization process goes on [11].

Details of these methods will be explained during the study when necessary.

1.2.4 Global optimization techniques

One of the most fundamental principles in our world is the search for an optimal state. It begins in the microcosm where atoms in physics try to form bonds1 in order to minimize the energy of their electrons [12]. When molecules form solid bodies during the process of freezing, they try to assume energy-optimal crystal structures. These processes, of course, are not driven by any higher intention but purely result from the laws of physics.

The same goes for the biological principle of survival of the fittest which, together with the biological evolution, leads to better adaptation of the species to their environment. Here, a local optimum is a well-adapted species that dominates all other animals in its surroundings. Homo sapiens have reached this level, sharing it with ants, bacteria, flies, cockroaches, and all sorts of other creepy creatures.

If something is important, general, and abstract enough, there is always a mathematical discipline dealing with it. Global optimization is the branch of applied mathematics and numerical analysis that focuses on, well, optimization. The goal of global optimization is to find the best possible elements "x" from a set "X" according to a set of criteria F = {f1, f2, .., fn}. These criteria are expressed as mathematical functions (objective functions) [13].

A classification of optimization algorithms

This classification can be handled according to many different ways. Figure 1.2 is one of the most robust classifications as it shows all important deterministic and non-deterministic procedures. This figure also outlines the situation of natural techniques among other well-known search procedures.

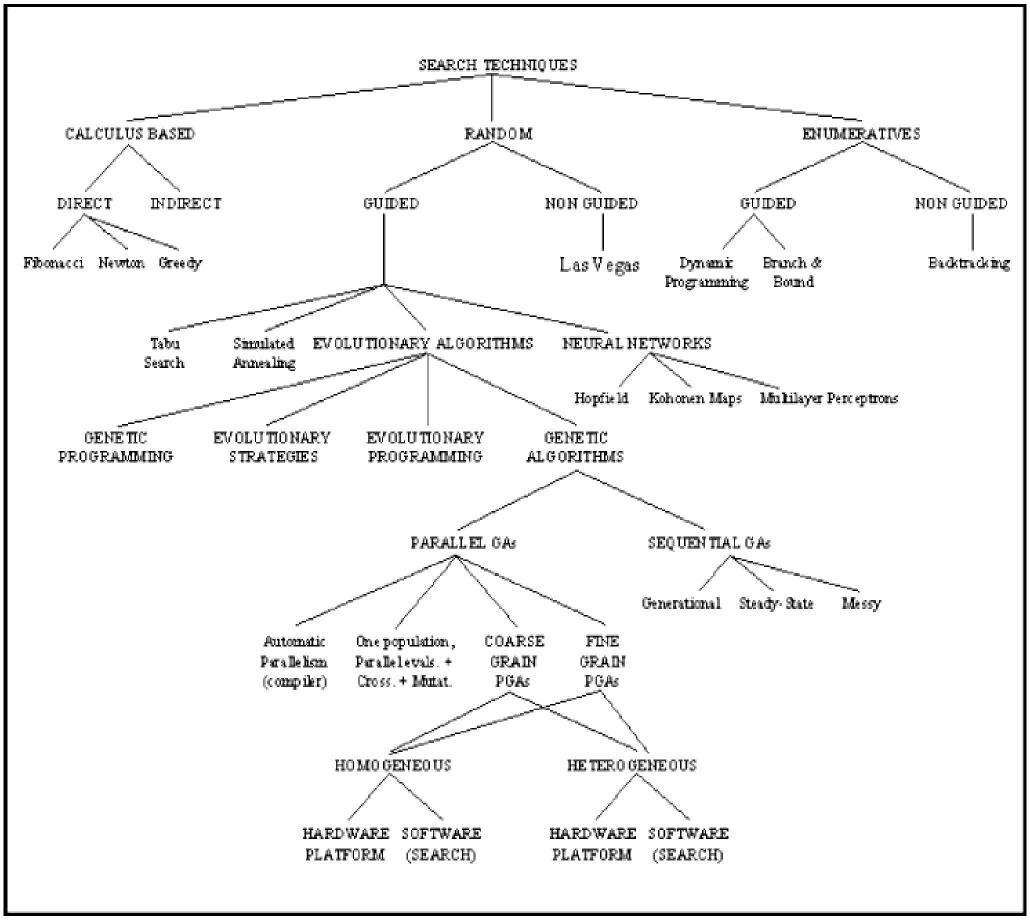

Figure 1.2 : Well-known search procedures [14].

In the scope of this study, a detailed explanation will be given for *Simulated Annealing (SA)* since it is one of the most popular global optimization algorithm and can be used for continuous problems, and *Genetic Algorithm (GA)* will also be introduced and explained deeply as it will be the algorithm that we will use for discrete optimization problems when necessary.

The names of “Simulated Annealing” and “Genetic Algorithms” are not accidental since the ideas since the ideas behind them are derived from natural process. SA is based on cooling of metals to obtain defined crystalline structures based on minimum potential energy. GA is based on the combination and recombination of the genes in the biological system leading to improved DNA sequences or species selection. These methods can be used can be used for continuous and discrete categories of the optimization problem.

They share an important characteristic with other optimization techniques in that they are primarily search techniques. They identify the optimum by searching the design space for the solution. Unlike the gradient-based methods, these techniques are largely heuristic, generally involve large amounts of computation, and involve some statistical reasoning. Global optimization is being pursued seriously today attracting a significant amount of new research to the area of applied optimization**.** A significant drawback to many of these techniques is that they require empirical tuning based on the class of problems being investigated. There is also no easy way to determine technique/problem sensitive parameters to implement an automatic optimization technique.

The global optimum cannot really be characterized as being different from the local optimum. The latter is referenced when only a part of the design space (as opposed to the complete space) is viewed in the neighborhood of a particular design point. The standard optimum techniques will identify the local optimum only. This is also more emphatic in numerical investigations of continuous problems where the algorithms are expected to move toward the optimum values close to where they are started (initial guess). Although it makes sense to look for the global optimum, there is no method to identify if such an optimum exists for all classes of problems. At the present only continuous convex problems are guaranteed to have a global solution. It is not difficult to argue that such a class represents only small amount of useful problems and most real optimization problems are not necessarily convex. Therefore in the literature, both of these methods are expected to produce solutions to close global optimum solution, rather the local optimum solution themselves. This is not the suggestion that the investigation of the global solution is a waste of time if possibility of one can not be established with certainty. Enough justification can be advanced by competitive and financial gains to warrant an investigation, even without the guarantee of useful results. Global optimization techniques are simple to implement while requiring exhaustive calculations. This can potentially soak up all of the idle time on the personal desktop computing resource, lending it immediate attractiveness [15].

Simulated annealing

In metallurgy and material science, annealing is a heat treatment of material with the goal of altering its properties such as hardness. Metal crystals have small defects, dislocations of ions which weaken the overall structure. By heating the metal, the energy of the ions and, thus, their diffusion rate is increased. Then, the dislocations can be destroyed and the structure of the crystal is reformed as the material cools down and approaches its equilibrium state. When annealing metal, the initial temperature must not be too low and the cooling must be done sufficiently slowly so as to avoid the system getting stuck in a meta-stable, non-crystalline, state representing a local minimum of energy [13].

In physics, each set of positions of all atoms of a system pos is weighted by its Boltzmann probability factor "$e^{-\frac{E(pos)}{k_B T}}$" where E(pos) is the energy of the configuration "*pos*", "*T*" is the temperature measured in Kelvin, and k_B is the Boltzmann's constant, (eq. 1.10)

$$\boldsymbol{k_B = 1.380\ 650\ 524 \cdot 10^{-23}\ \mathrm{J/K}}. \qquad \textbf{(1.10)}$$

The Metropolis procedure was an exact copy of this physical process which could be used to simulate a collection of atoms in thermodynamic equilibrium at a given temperature. A new nearby geometry pos_{i+1} was generated as a random displacement from the current geometry pos_i of an atom in each iteration. The energy of the resulting new geometry is computed and ΔE the energetic difference between the current and the new geometry was determined. The probability that this new geometry is accepted $P(\Delta E)$ is defined in eq. 1.13

***posi+1*) −*E*(*posi*)** **(1.12(1.11)**

$$\boldsymbol{\Delta \mathrm{E} = E(pos_{i+1}) - E(pos_i)} \qquad \textbf{(1.12)}$$

$$\boldsymbol{P(\Delta E) = \begin{cases} e^{-\frac{E(pos)}{k_B T}} if\ \Delta E>0 \\ 1\ otherwise \end{cases}} \qquad \textbf{(1.13)}$$

Thus, if the new nearby geometry has a lower energy level, the transition is accepted. Otherwise, a uniformly distributed random number $r = random_u() \in [0,1)$ is drawn and the step will only be accepted in the simulation if that "r" is less or equal the Boltzmann probability factor, (for example $r \leq P(\Delta E)$ At high temperatures T, this factor is very close to 1, leading to the acceptance of many uphill steps. As the temperature falls, the proportion of steps accepted which would increase the energy level decreases. Now the system will not escape local regions anymore and (hopefully) comes to a rest in the global minimum at temperature T=0K.

The abstraction of this method in order to allow arbitrary problem spaces is straightforward that is, the energy computation $E(pos_i)$ is replaced by an objective function f or even by the result v of a fitness assignment process. *Algorithm 1.1* illustrates the basic course of Simulated Annealing.

Algorithm 1.1 : $\boldsymbol{x^*} = \boldsymbol{simulatedAnnealing(f)}$ [13].

Input ➔f: the objective function to be minimized
Data ➔p_{new}: the newly generated individual
Data ➔p_{cur}: the point currently investigated in problem space
Data ➔p^*: the best individual found so far
Data ➔T: the temperature of the system which is decreased over time
Data ➔t: the current time index
Data ➔ΔE: the enery difference of the $x_{new}$and x_{cur}
Output ➔x^*: the best element found
begin
    create $p_{new}.g$
    //Implicitly: $p_{new}.x$=$gpm(p_{new}.g)$
    $p_{cur} = p_{new}$
    $p^* = p_{new}$
    $t = 0$
    while terminationCriterion() do
        $\Delta E = f(p_{new}.x) - f(p_{cur}.x)$
        if $\Delta E \leq 0$ then
            $p_{cur} = p_{new}$
            if $f(p_{cur}.x) < f(p^*.x)$ then $p^* = p_{cur}$
        else
            T=getTemperature(t)
            if $random_u() < e^{-\frac{E(pos)}{k_B T}}$ then $p^* = p_{cur}$
        $p_{new}.g = mutate(p_{cur}.g)$
    // Implicitly: $p_{new}.x$=$gpm(p_{new}.g)$
        t=t+1
    return $p^*.x$
end

It has been shown that Simulated Annealing algorithms with appropriate cooling strategies will asymptotically converge to the global optimum. The temperature schedule defines how the temperature in Simulated Annealing is decreased. As already mentioned, this has major influence on whether the Simulated Annealing algorithm will succeeded, on whether how long it will take to find the global optimum, and on whether or not it will degenerate to simulated quenching. There are number of ways to decrease the temperature during simulated annealing [13].

Genetic algorithms

Despite the scope of this study is "Genetic Algorithms" for discrete optimization problems, it is worth to give the introduction for all "Evolutionary Algorithms" in order to understand all the "Evolutionary Computation" techniques. As it can be seen from Figure 1.2, evolutionary algorithms can be investigated under 4 different headlines. These are:

- Genetic Programming
- Evolutionary Strategies
- Evolutionary Programming
- Genetic Algorithms

Charles Darwinian evolution in 1859 is intrinsically a so bust search and optimization mechanism. Darwin's principle "Survival of the fittest" captured the popular imagination. This principle can be used as a starting point in introducing evolutionary computation. Evolved biota demonstrates optimized complex behavior at each level; the cell, the organ, the individual and the population. Biological species have solved the problems of chaos, chance, nonlinear interactivities and temporality. These problems proved to be in equivalence with the classic methods of optimization. The evolutionary concept can be applied to problems where heuristic solutions are not present or which leads to unsatisfactory results. As a result, evolutionary algorithms are of recent interest, particularly for practical problems solving.

Over time, the entire population of the ecosystem is said to evolve to contain organisms that, on average, are more fit than those of previous generations of the population because they exhibit more of those characteristics that tend to promote survival.

Evolutionary computation (EC) techniques abstract these evolutionary principles into algorithms that may be used to search for optimal solutions to a problem. In a search algorithm, a number of possible solutions to a problem are available and the task is to find the best solution possible in a fixed amount of time. For a search space with only a small number of possible solutions, all the solutions can be examined in a reasonable amount of time and the optimal one found. This exhaustive search, however, quickly becomes impractical as the search space grows in size. Traditional search algorithms randomly sample (e.g., random walk) or heuristically sample (e.g., gradient descent) the search space one solution at a time in the hopes 1 of finding the optimal solution. The key aspect distinguishing an evolutionary search algorithm from such traditional algorithms is that it is population-based. Through the adaptation of successive generations of a large number of individuals, an evolutionary algorithm performs an efficient directed search. Evolutionary search is generally better than random search and is not susceptible to the hill-climbing behaviors of gradient-based search [14].

The major difference between evolution strategies and genetic algorithms lies in the representation of the genotype and in the way the operators are used (which are mutation, selection, and eventually recombination). In contrast to GAs, where the main role of the mutation operator is simply to avoid stagnation, mutation is the primary operator of evolution strategies.

Genetic programming (GP), an extension of the genetic algorithm, is a domain-independent, biologically inspired method that is able to create computer programs from a high-level problem statement. In fact, virtually all problems in artificial intelligence, machine learning, adaptive systems, and automated learning can be recast as a search for a computer program; genetic programming provides a way to search for a computer program in the space of computer programs. Similar to GAs, GP works by imitating aspects of natural evolution, but whereas GAs are intended to find arrays of characters or numbers, the goal of a GP process is to search for computer programs (or, for example, formulas) solving the optimization problem at hand. As in every evolutionary process, new individuals (in GP's case, new programs) are created. They are tested, and the fitter ones in the population succeed in creating children of their own whereas unfit ones tend to disappear from the population [16].

Evolutionary programming took the idea of representing individuals' phenotypic ally as finite state machines capable of responding to environmental stimuli and developing operators for effecting structural and behavioral change over time. This idea was applied to a wide range of problems including prediction problems, optimization and machine learning [14].

There is a rough classification of using these evolutionary algorithms as; Genetic Programming – which develops programs that simulate evolutionary process; Genetic Algorithms – optimization of combinatorial/discrete problems; Evolutionary Programming – optimizing continuous functions without recombination;

Evolutionary Strategies – optimizing the continuous functions with recombination [17]. However, this classification is quite loose. For example, GAs have been used and continue to be used effectively to solve the continuous optimization problems.

The basics of genetic algorithms

An algorithm is a series of steps for solving a problem. A genetic algorithm is a problem solving method that uses genetics as its model of problem solving. It's a search technique to find approximate solutions to optimization and search problems.

GA handles a population of possible solutions. Each solution is represented through a chromosome, which is just an abstract representation. Coding all the possible solutions into a chromosome is the first part. Nevertheless, a set of reproduction operators has to be determined, too. Reproduction operators are applied directly on the chromosomes, and are used to perform mutations and recombinations over solutions of the problem. Appropriate representation and reproduction operators are really something determinant, as the behavior of the GA is extremely dependant on it. Frequently, it can be extremely difficult to find a representation, which respects the structure of the search space and reproduction operators, which are coherent and relevant according to the properties of the problems.

Selection is supposed to be able to compare each individual in the population. Selection is done by using a fitness function. Each chromosome has an associated value corresponding to the fitness of the solution it represents. The fitness should correspond to an evaluation of how good the candidate solution is. The optimal solution is the one, which maximizes the fitness function. Genetic Algorithms deal with the problems that maximize the fitness function. However, if the problem consists in minimizing a cost function, the adaptation is quite easy. Either the cost function can be transformed into a fitness function, for example by inverting it; or the selection can be adapted in such way that they consider individuals with low evaluation functions as better.

Once the reproduction and the fitness function have been properly defined, a Genetic Algorithm is evolved according to the same basic structure. It starts by generating an initial population of chromosomes. This first population must offer a wide diversity of genetic materials. The gene pool should be as large as possible so that any solution of the search space can be engendered. Generally, the initial population is generated randomly.

Then, the genetic algorithm loops over an iteration process to make the population evolve (Figure 1.3).

Each iteration consists of the following steps [14]:

- SELECTION: The first step consists in selecting individuals for reproduction. This selection is done randomly with a probability depending on the relative fitness of the individuals so that best ones are often chosen for reproduction than poor ones.
- REPRODUCTION: In the second step, offspring are bred by the selected individuals. For generating new chromosomes, the algorithm can use both recombination and mutation.
- EVALUATION: Then the fitness of the new chromosomes is evaluated.
- FITNESS ASSIGNMENT: use the objective values to determine fitness values

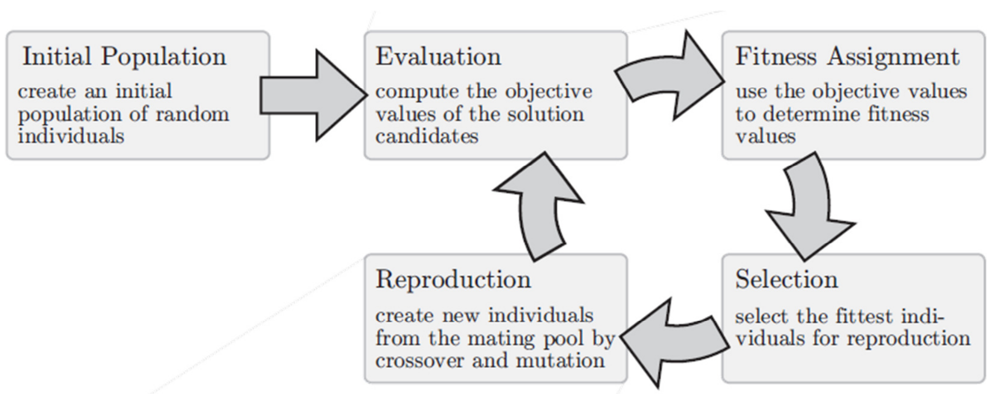

Figure 1.3 : Basic Cycle of Genetic Algorithm [13].

Genomes in genetic algorithms

Most of the terminology which we have defined in before and used throughout this study stems from the GA sector. The search spaces G of genetic algorithms, for instance, are referred to genome and its elements are called genotypes. Genotypes in nature encompass the whole hereditary information of an organism encoded in the DNA. The DNA is a string of base pairs that encodes the phenotypical characteristics of the creature it belongs to. Like their natural prototypes, the genomes in genetic algorithms are strings, linear sequences of certain data types [83-85]. Because of the linear structure, these genotypes are also often called chromosomes. In genetic algorithms, we most often use chromosomes which are strings of one and the same data type, for example bits or real numbers [13].

A string chromosome can be either a fixed-length tuple $G = \{\forall\ (g[1], g[2], .., g[n]) : g[i] \in Gi\ \forall i \in 1..n\}$ (1.14) or a variable-length list $G = \{\forall \text{lists } g : g[i] \in GT\ \forall 0 \leq i < len(g)\}$**(1.15)**.

In the first case, the loci i of the genes Gi are constant and, hence, the tuples may containelements of different types Gi.

$$G = \{\forall\ (g[1], g[2], .., g[n]) : g[i] \in Gi\ \forall i \in 1..n\} \quad \textbf{(1.14)}$$

This is not given in variable-length string genomes. Here, the positions of the genes may shift when the reproduction operations are applied. Thus, all elements of such genotypesmust have the same type GT .

$$G = \{\forall \text{lists } g : g[i] \in GT\ \forall 0 \leq i < len(g)\} \quad \textbf{(1.15)}$$

String chromosomes are normally bit strings, vectors of integer numbers, or vectors of real numbers. Genetic algorithms with numeric vector genomes in their natural representation, i. e., where $G = X \subseteq Rn$ are called real-encoded [86]. Today, more sophisticated methods for evolving good strings (vectors) of (real) numbers exist (such as Evolution Strategies,Differential Evolution, or Particle Swarm Optimization) than processing them like binary strings with the standard reproduction operations of GAs.

Bit string genomes are sometimes complemented with the application of gray coding during the genotype-phenotype mapping. This is done in an effort to preserve locality and ensure that small changes in the genotype will also lead to small changes in the phenotypes [87]. Collins and Eaton [88] studied different encodings for GAs and found that their E-code outperform both gray and direct binary coding in function optimization. Messy genomes where introduced to improve locality by linkage learning. Genetic algorithms are the original prototype of evolutionary algorithms. They provide search operators which closely copy sexual and asexual reproduction schemes from nature. In such “sexual” search. operations, the genotypes of the two parents genotypes will recombine. In asexual reproduction, mutations are the only changes that occur.

It is very common to apply both principlesin conjunction, i. e., to first recombine two elements from the search space and subsequently make them subject to mutation. In nature, life begins with a single cell which divides6 time and again until a mature individual is formed7 after the genetic information has been reproduced. The emergence of a phenotype from its genotypic representation is called embryogenesis in biology and its counterparts in evolutionary search are the genotype-phenotype mapping and artificial embryogeny [13].

1.3 Literature Overview

Literature on the optimization methods is very extensive. Lots of books and review articles are available. Most generally, the methods can be classified into two categories as "discrete variable optimization and "continuous variable optimization". Discrete variable optimization methods are very useful for practical applications. The design variables in these applications must be selected from a specified set of discrete variables. In the continuous variable optimization problems , the design variables are assumed to be continuous and all the problem functions to be differentiable. These methods can be classified into two broad categories as "primal methods" and "transformation methods" [18-19]. In primal methods, each constraint of the problem is treated separately in computations. This category includes modern methods such as "sequential quadratic programming (SQP)" methods [20-24]. And older methods such as "sequential linear programming (SLP)", "feasible directions method" [25-26], "gradient Projection method" [27], "generalized reduced gradient method" [28-29], "numerical optimality Criteria and convex linearization" [30]. Methods for multi criteria optimization have also been developed by Osyczka [31]. Transformation methods convert the original constrained optimization problem to a series of unconstrained problems. The unconstrained problems involve certain parameters that are adjusted after each unconstrained minimization. Gill et al [32] and Luenberger [33] showed that the solution of the unconstrained problems can be shown to coverage to the solution of the original constrained problems. Belengundu and Aorta [34] presented that an advantage of these methods is that they do not require gradients of individual constraints because gradient of the transformed function can be calculated directly. This is desirable for dynamic response optimal design and control problems [35-36].This class of methods includes the barrier or interior penalty methods, exterior penalty methods, and the multiplier or augmented Lagrangien Methods [32-33, 37-38].

Fox and Kapor [39] made the one of the earliest work on optimization of structures under dynamic conditions. They minimized the mass of framed structures subjected to base motion and constraints on deflection and stresses. The equations of motion were uncoupled by using the vibration modes and solved for maximum value of modal response by using a shock spectrum approach. A simplification was introduced by summing maximum modal responses; thus, Sevin and Pilkey [40], Wilmert and Fox [41], Afimiwala and Mayne [42] used another formulation to solve dynamic response optimization problems. In that approach, the state variables were also discretized and used as design variables. The optimization problem became very large as a result of this approach. So only small scale problems could be solved. Later, Cassis and Schmit [43] used approximation concepts to treat dynamic response constraints. They replaced the original problem by an approximate explicit problem that could be linear or nonlinear. Feng et al [44] used the adjoint variable method of sensitivity analysis for optimal design of framed structures subjected to dynamic loads. The method was later extended to more general dynamic response optimization problem by Haug and Arora [37]. Hsieh and Arora [45] showed that numerical difficulties in convergence to the optimal solution can be encountered with the equivalent functional constraint formulation which was used in these publications. Several ways of treating the point-wise state variable constraints with the first and second order forms of the equations of motion were developed to overcome these difficulties. Worst case design formulation was used where the point-wise constraint was imposed at all the local maximum points. There was a disadvantage of this approach. That was the local maxima can be large in some problems whose response functions are jagged. With respect to that, Grandhi, et al [46] [47] proposed several methods for obtaining few important constraints at discrete points in time. In 1989, Tseng and Arora [48] proposed five possible approaches for treating time dependent constraints. However, the adjoint variable method can be used to directly evaluate the gradient of the of the functional for the augmented Lagrangien method, and thus, gradients of individual constraints are not needed [34-36, 49].

There are very few papers regarding structural optimization with nonlinearities in the dynamic response. One of the rare can be counted in is the work of Ray, et al [50] where direct differentiation and adjoint variable methods of sensitivity analysis were used for nonlinear structures (material properties having hysteresis loss) using a first order form of the state equations. Geometric nonlinearity and large deformation were not treated. The system of differential equations obtained for sensitivity was transcribed into second order form for numerical implementation. Jao and Arora [51] have also treated shape and non shape optimization of nonlinear structures.

Scalarization techniques for multiple objective functions were already presented by Edgeworth more than one hundred years ago [52]. Kuhn and Tucker [53] developed several mathematical properties with respect to mathematical programming in the middle of the 20th century. As we discussed before, we usually refer a solution of multi-objective optimization to as a Pareto solution. Pareto presented the idea Pareto solution in 1906 [54]. The same ideas can be referred, in particular, from a viewpoint of utility theory in Stigler [55]. Pareto Solution also called as "efficient solution" by Koopmans [56], "nondominated solution" by Neumann and Morgenstern [57] and "noninferior solution" by Zadeh [58].

Kirkpatrick et al [59] presented the first application using simulated annealing for optimization. The examples covered problems from electronic systems, in particular, component placement driven by the need for minimum wiring connection. Cerny [60] employed a similar approach to the traveling salesman problem. Metropolis et al [61] demonstrated that simulated annealing itself was based on the use of statistical mechanics to establish thermal equilibrium in a collection of atoms. The term itself is more representative of the process of cooling materials, particularly metals, after raising the temperature to achieve a definite crystalline state. The extension to optimization is mainly heuristic. The simple idea behind simulated annealing can be seen through the modified basic algorithm of SA due Bochavesky et al [62]. Nolte and Schrader [63] and van Laarhoven and Aarts [64] showed that simulated annealing will converge to a global optimum if $t \to \infty$ iterations are performed with a list of the most important works. Anily and Federgruen, [65] Gidas [66] and Mitra et al [67] have solved several problems providing deterministic, non-infinite boundaries for asymptotic convergence. Nolte and Schrader [63] gave also a list of these problems. In the same paper, they introduce a significantly lower bound, which, however, still states that Simulated Annealing is probably in an optimal configuration after the number of iterations exceeds the cardinality of the problem space which is slow. In other words, it would be faster to enumerate all possible solution candidates in order to find the global optimum with absolute certainty than applying Simulated Annealing. This does not mean that Simulated Annealing is always slow. It only needs that much time if we persist on the optimality. Speeding up the cooling process will result in a faster search, but voids the guaranteed convergence on the other hand. Such speeded-up algorithms are called Simulated Quenching (SQ) [68-69].

The roots of genetic algorithms go back to the mid-1950s, where biologists like Barricelli [70-72] and the computer scientist Fraser [73] began to apply computer-aided simulations in order to gain more insight into genetic processes and the natural evolution and selection. Bregermann [74], used evolutionary approaches based on binary string genomes for solving inequalities, for function optimization, and for determining the weights in neural networks in the early 1960s. At the end of that decade, important research on such search spaces was contributed by Bagley [75] (who introduced the term genetic algorithm). Holland [76-78] developed a new approach to the genetic algorithms for problem solving could be formalized finally became widely recognized and popular. Today, there are many applications in science, economy, and research and development that can be handled with genetic algorithms. Therefore, various forms of genetic algorithms have been developed to. Some genetic algorithms like the human-based genetic algorithms (HBGA), for instance, even require human beings for evaluating or selecting the solution candidates [79-81].

2. AUTOMOTIVE DRY CLUTCH AND FLYWHEEL AS A DYNAMIC STRUCTURAL SYSTEM

2.1 A Brief History of Clutch System

In the course of over 100 years of automotive history, nearly all components have undergone enormous technological developments. Reliability, production costs and service-friendliness as well as, more recently, environmental safety, have been and continue to be the criteria demanding new and better solutions from automotive engineers. The basic designs are usually known early on, but only the availabilityof new materials and processing procedures makes their realisation feasible.

The operating principles of the first clutches originated in the mechanised factories of early modern industry. By analogy with the transmission belts used there, flat leather belts were now introduced into motor cars. When tensioned by a roller, the belt transmitted the drive output of the engine's belt pulley to the drive gears, and when loosened, it slipped through i. e. disengaged. As this procedure caused the leather belts to wear out fast, a new tactic was adopted of installing an idler pulley of the same size beside the drive belt pulley. By moving a lever, the transmission belt could be guided from the idler pulley on to the drive pulley. The motor car patented by Benz in 1886, which Bertha Benz used to make the first long-distance journey in the history of motor vehicles – from Mannheim to Pforzheim – already operated according to this clutch concept.

The disadvantages of a belt drive, such as low efficiency, high susceptibility to wear and inadequate running characteristics especially. This pushed the engineers to seek better alternatives to automotive clutches.

The results were a wide variety of clutch types. It is not necessary to mention all the alternatives. But some of these new alternative were the forerunners of the present-day clutch. all based on the principle of the friction clutch. Here the disc is located on the end of the crankshaft and is joined by a second, stationary disc. When the two make contact, friction is produced and the secondary disc is set in motion. As the clamping load is increased, the driving disc carries along the driven disc with increasing speed until power transmission is reached, and both discs have the same rotational speed. In the period up to full engagement, the main driving energy is converted into heat as the discs slide across one another. This arrangement meets the two chief demands – on the one hand gradual and gentle engagement, so that, when driving off, the engine is not cut off and does not jerk with the drive train, and on the other hand loss-free power transmission with the clutch engaged. The clutch is actuated via the foot pedal, which pulls back the cone carrier via a release lever against the spring force and thus disengages the clutch.

While clutch operation for a long time required strong legs, as pedal loads had to be transmitted via the linkage and shafts, comfort was improved in the 1930s with the use of control cables, and in the 1950s with the use of hydraulic actuation.

Since the 1960s dry clutch designs are based on a single friction disk compressed by a washer spring.5 The single-disk design offers a great compactness, very important for transversal engine architectures where the engine, the clutch, the gearbox and the differential have to fit in between the wheels [89].

Easy operation was also promoted by various attempts to automate the clutch process: in 1918 Wolseley had the first idea of an electromagnetic clutch. In the early 1930s the French firm Cotal built a pre-selector gearbox with an electromagnetic clutch, which was used in luxury cars. Best known were the centrifugal clutch, which regulated its clamping load by the centrifugal force, and automatic clutches such as Saxomat (Fichtel & Sachs), LuKomat (LuK), Manumatik (Borg & Beck) and Ferlec (Ferodo).

None of these was able to prevail; the competition from manual and automatic transmissions with torque converters was too great [91].

2.2 Literature Overview for Clutch Vibrations

Since engineering structures are mostly subjected to dynamic loads, it is important to design these structures in consideration of their dynamic behavior whose optimization is difficult since the associated mathematical models are complicated and computational costs are expensive. In this study, dynamic behavior of a clutch and dual mass flywheel (DMF) system is optimized.

The clutch and dual mass flywheel system existing in manual transmissions of internal combustion engines are important components in a vehicle that affect the ride comfort of the vehicle and may cause noise, fatigue of components and longitudinal vibrations during the cruise of the vehicle. It is noteworthy that the loading on the clutch and dual mass flywheel system is transient and a dynamic model has to be employed in optimization studies. The clutch systems are essentially designed for damping out rotational vibrations of the engine; thus, it helps increase the ride comfort; nonetheless, they are exposed to axial vibrations as well. They generally do not have any component to annihilate axial vibrations due the following reasons: first of all, packaging area of the clutch system is limited between engine and transmission and there is no available space to add some springs and dampers to annihilate axial vibrations. Second, axial vibrations are normally insignificant to be considered in design phase that are supposed to be damped by the diaphragm spring and cushion springs. But, axial vibrations may lead to some unexpected results in the power train such as the rattle noise that is examined in this study by using global optimization techniques.

Many studies on the clutches of different types exist in literature. For instance, wet clutches and dual clutches, which are mostly used on automated manual transmissions, engagement characteristics and their effects on engine characteristics, shift characteristics and their effects on shaft vibrations, modal frequencies of the clutch system and clutch control mechanisms are some of the investigation topics [92-98]. There are also studies in literature on modeling of one way clutches [99-100]. Some researchers [101-109, 132-133] examined manual transmission dry clutch systems and their rotational behavior. For example, Crowther and Zhang studied the torsional behavior of the clutch as an elastic system in the driveline [102]. Duan and Singh formulated the clutch engagement and disengagement as a nonlinear dry friction problem with harmonically varying normal load and investigated the stick-slip motions to understand the clutch response under various torque inputs [103-104]. Awrejcewicz and Grzelczyk studied the wear of clutch friction surfaces [109]. Being one of the most important components of a clutch system, the diaphragm spring (which determines clutch pedal load) is studied by some researchers and optimized for the best release load and durability [110-116]. Some researchers studied the Almen-Laszlo formula (which is used to calculate disc spring characteristics) [117] to improve its accuracy [118-121]. Information about main components of a clutch system can be found in [122-123].

One of the rare investigations on longitudinal vibrations in a clutch system is presented in [124], where Esfahani *et al* studied in-cycle vibrations on the clutch pedal which is excited by the crank shaft's axial vibration; at this point, the authors remark about the same challenge as we faced in this study which is the lack of an extensive axial clutch vibration studies in literature whilst the problem observed are connected with axial clutch vibrations. They focused on the clutch pedal vibrations around 257Hz, which is originated from the clutch axial vibrations, and their simulation showed the same vibration level around 230-240Hz which is a good correlation with actual condition, e.g., see [124]. But they did not investigate why their system showed the whoop frequency, or in cycle vibration, around 250Hz and did not present a discussion or proposal how this could be reduced. They only suggested using such kind of simulation toolboxes to foresee any possible problems before handling the physical parts.

Reitz *et al.* worked on a special test bench to investigate NVH problems in a clutch system [137]. They reported that torsional and axial flywheel vibrations excite the clutch system. Torsional flywheel vibrations are caused by the 2nd engine order and axial flywheel vibrations are caused by the crankshaft bending vibrations which are excited by the 4th cylinder firing. This means that axial vibrations are caused by the 0.5th engine order which is the same outcome of our investigations as explained in upcoming sections in detail. They also reported the test results of three different clutches but did not share the clutches' specifications due to confidentiality requirements. They presented their test bench that enables to determine the clutch systems NVH behavior during engagement and disengagement in detail and allow them to predict the behavior of the clutch system in a real vehicle.

Kelly and Rahnejat investigated the pedal vibrations and influence of individual components on the transmission of the subsequent vibration and noise [138]. They investigated the vibrations in a vehicle and multi-body model of clutch and observed that the pedal vibration and noise occurred due to the 0.5th engine order excitation from the 4th cylinder firing resulting in bending of the crankshaft. This is actually the same root cause of the clutch axial vibrations investigated in this study and although the source is the crankshaft bending vibrations, these vibrations are transmitted by the flywheel and clutch system. They also showed that cable properties and mass fixes can influence the vibrations.

Hasabe and Seiki studied the clutch pedal vibrations which are excited by the clutch axial vibrations [139]. They conducted an experimental study based on an on-vehicle observation of the vibrations and a test bench. They tried to develop a method for a test bench evaluation to predict the levels of vibration and noise in a vehicle having a 4 cylinder engine and observed that the pedal vibrations exist at 4400RPM engine speed when the clutch is half engaged and in the 1st, 3rd and 6th harmonic of the engine vibration. They observed the same behavior on the clutch side which proves that the vibrations are transmitted by the clutch system. They also stated that the clutch system has the major contribution on pedal vibrations. Their measurements showed that their sample clutch has the critical resonance frequency between 200-300Hz. Unlike other studies in literature, since their investigation covered the whole range of release bearing travel, they observed a relatively larger range for critical frequency and they proposed two different clutch behaviors at engaged and disengaged positions which lead to this larger range. They built a correlation between the vibration level and frequency range of two different natural frequencies of engaged and disengaged positions as well.

If the clutch model and its optimization are put aside, there are many studies related to gear rattle phenomenon in literature [105-106, 125-129]. Some of these studies focused on neutral gear rattle [125-126], some of them focused on backlash effects [106, 127-128]. Barthod *et al.* studied gear rattle phenomenon to understand the rattle threshold and rattle noise evolution in relation with excitations and mechanical gearbox parameters [129]. On the other hand, ergonomic issues of the clutch pedal characteristic for different driver profiles are studied in [130]. There are also some studies to identify the heat energy generation in friction clutch systems. To this end, Shabibi investigated the transient solution of heat conduction problem in solids with insulated boundaries [131].

Padmanabhan and Singh [105] employed a simplified rotational clutch model and showed that if clutch parameters are selected appropriately, it can help reduce the transmission rattle noise; although they employed a simplified clutch model, it is a good example which shows that clutch parameters are very important to reduce transmission rattle noise problem. They proposed a dry friction three-stage clutch with moderate first stage spring stiffness. Although we are using a dry clutch, the clutch examined in this study has one-stage spring set on it and this spring set is selected carefully to accommodate corresponding engine torque. It is noteworthy that three-stage clutches are not normally used with DMF systems; instead, they are used with single mass flywheel systems due to their relatively higher rotational vibration frequencies. Furthermore, three-stage clutches are more expensive than single stage clutches due to 2 more spring sets and friction surfaces. Nonetheless, since our target should be getting the best results with a competitive price, we avoid bringing add on cost to our system.

On the other hand, analyses of longitudinal dynamic behavior of a clutch system and seeking the solution of corresponding optimization problem have not been pursued in literature. Motivated by these facts, this study is initiated in which dynamic models of components of a clutch system are derived and constructed in Matlab-Simulink, mathematical models are verified by measurements, a multi-objective optimization problem is formulated having two objective functions (i.e., pressure plate vibrations and pedal characteristics) and corresponding Pareto optimal solution is obtained and discussed. A prototype clutch is manufactured by using the design variables obtained by optimization runs and then it is tested. It is shown that analytical models agree well with the experimental measurements and vibrations in the clutch system can be reduced significantly by choosing the design parameters with optimization tools.

2.3 Clutch Function

The clutch and dual mass flywheel system existing in manual transmissions of internal combustion engines are important components in a vehicle that affect the ride comfort of the vehicle and may cause noise, fatigue of components and longitudinal vibrations during the cruise of the vehicle. It is noteworthy that the clutch and dual mass flywheel system are subjected to dynamic loads, which is selected as the scope of this study. A clutch system is essentially designed for damping out rotational vibrations of the engine; thus, the driver and passenger comfort increases. Many studies exist in literature that examine rotational vibrations of this system. Lots of improvements and enhancements are developed in decades and research studies have still been undertaken to increase the damping capacity of these systems. Figure 3.2 shows a modern clutch and dual mass flywheel system and damping of rotational vibrations due to these systems. On the other hand, Figure 2.7 shows different working regimes and spring characteristics of a clutch system.

Even though we mentioned above that clutch systems mostly deal with rotational vibrations of an engine, they are usually exposed to axial vibrations as well. But they generally do not have any specialized component to annihilate axial vibrations due the following reasons: the first reason is that packaging area (assembly area) of the clutch system is very limited and there is no available space to add some springs and dampers to annihilate axial vibrations. The second reason is that the axial vibrations are normally insignificant to be considered in design phase and most of the axial vibrations is damped on the diaphragm spring and cushion springs. But, this may sometimes lead to some unexpected results in the powertrain systems as the rattle noise that we will be discussed later and is examined to be resolved in this study by using global optimization techniques.

Before going deep into multi-objective optimization of automotive dry clutch system, let's try to give some brief information about the automotive driveline system, clutch system as a subsystem in driveline, clutch usage and working principles.

The driving comfort is highly important for the commercial success of a vehicle. This must be considered with dynamic performances or its fuel efficiency. But, though, this element is much more difficult to measure. During the design phase and the final design improvements of a new car so much efforts are made to insure a correct level of the different comfort performances. These performances depend on most of the components of the vehicle especially of the driveline.

The clutch is a key element for the powertrain driving comfort during standing-start and gear-shifting maneuvers in manual transmission cars, which constitute the vast majority of the European automotive park. Furthermore, driven by a strong pressure for fuel efficiency, automated manual transmission and dual-clutch transmission systems are introduced in historically automatic transmission markets such as North and South Americas and far eastern countries like Korea and Japan. These systems, essentially improved robot controlled manual transmission gearboxes, make use of one or more clutches during standing-start and gear-shifting maneuvers and, therefore, share most driving comfort issues with their manual counterparts [89].

After this, we can look into how a clutch and actuation system are designed and affected by the neighborhood parts and constraints.

First and one of the most important design parameter is the max torque output of engine. Due to consistently increasing performance demand, torque output of engines has been continuously increasing over the past few years. New technologies are also on the way and it seems that increase on torque demand will continue in the future.

When the clutch is fully engaged, no slippage can be allowed. It must ensure whole engine torque is transferred to the transmission. While engine torque is increasing, clutch torque transfer capacity must be increased in parallel. There are two ways to achieve this:

- Increasing the diameter of the friction discs
- Increasing the normal force (clamp load) on the pressure plate

Because of the packaging constraints, it is not possible to increase the diameter of the friction discs anymore. These are mostly at design limit. Furthermore, increasing the disc diameter also makes disc inertia is bigger and this is not also allowed due to engine constraints. The other solution is to increase the clamp load on the disc by modifying the pressure plate stiffness. But, this has a direct relationship with the required force on the clutch pedal that driver needs to apply to disengage the clutch. Current vehicles are almost at the limit and increasing the clutch pedal effort more degrades the driver comfort. Of course this is not a good reputation, especially while the comfort of the current vehicles is becoming more and more important. The other important negative effect is the sharper engagement curve. Sharper engagement curve gives less slippage time during clutch engagement and this makes the clutch engagement harder to achieve for a normal driver. Fast engagement leads to torque shocks and engine stalls.

In order to overcome these problems, the only solution is longer clutch pedal travel, but this is also limited due to ergonomics constraints.

After this brief explanation of the dry friction clutch, we may give some more detailed definitions and descriptions and explain the subcomponents and of the clutch system.

Clutch system is basically used to allow transferring and cutting the engine torque to the driveline through the transmission. The system is composed of a connection element (clutch itself) and its actuation system. Under this study, we will discuss only dry single disc clutches with an hydraulic actuator.

Clutch's main functions can be seen below:

- Decoupling of the Engine and the Driveline This decoupling can be either of short duration, like, for example, while performing a standing-start ora gear shift, or much longer in order to provide a neutral position in the case of a AMT vehicle. The residual torque in the decoupled position is the main performance indicator for this function.

- Allowing standing-starts Since an internal combustion engine cannot operate at a revolution speed lower than a certain minimum, called the idle speed, at which the available engine torque is equal to the internal friction and pumping losses,4 a clutching mechanism is needed to smoothly launch the vehicle to this minimal speed. The performance indicator for this function is the clutch's dosability, meaning the ease with which the driver can control the clutch torque.

- Easing Gear Shifting While gear-shifting the clutch eases the synchronization of the crankshaft and primary gearbox shaft speeds. The engagement is quite short but the high torque levels reached can lead to uncomfortable driveline oscillations.

- Engine Acyclicity Filtering The engine acyclicity causes torsional vibrations of the crankshaft that, if not filtered out, are transmitted through the driveline to the vehicle body. In order to prevent this a system of damping springs is mounted on the clutch disk. Due to the increasing need of acyclicity filtering, the more powerful engines are equipped with a DMFW that assures a better filtering action. In this latter case the clutch disk presents no damping springs. [89].

2.4 Clutch Cable - Hydraulic Actuator

The clutch cable transmits the pedal force which is provided by the driver over the pedal ratio to the release bearing. The biggest disadvantage is the decreasing efficiency of the clutch cable over life time caused by the increasing friction of the cable. That bring about that the pedal force increases which causes customer dissatisfaction. An other important point is the bending radius of the clutch cable. Additionally the clutch cable routing is relevant for the function of the clutch system.

Hydraulic actuation systems are basically working with two cylinders which have different piston surfaces. One cylinder, the master cylinder, is fitted to the clutch pedal and the other cylinder, the slave cylinder, is fitted in or out of the clutch housing. In order to this the pedal force will be transmitted by the pedal ratio which includes the master cylinder and by the hydraulic ratio. For semi hydraulic systems which are described below there is one additional ratio between slave cylinder and release bearing. Figure 2.1 shows the pictures of the cable and hyraulic actuation systems.

Hydraulic actuation systems can be split in two groups. Full hydraulic and semi hydraulic systems. At full hydraulic systems the slave cylinder is fitted concentrically to the transmission input shaft. Semi hydraulic system mostly have an slave cylinder which is located out of the clutch housing. So an additional lever is necessary to transmit the force to the release bearing which has the advantage of one added ratio in the actuation system.

The increasing wear of the clutch disc over life time leads to an movement of the release bearing travel. Because of this it is necessary to compensate the movement of the release bearing travel. To compensate this movement a self adjustment mechanism or a manual adjustment must be planned. Cable actuated systems are adjustable by several mechanism at the pedal or at the cable. Hydraulic systems are normally adjust by regulation of the oil volume.

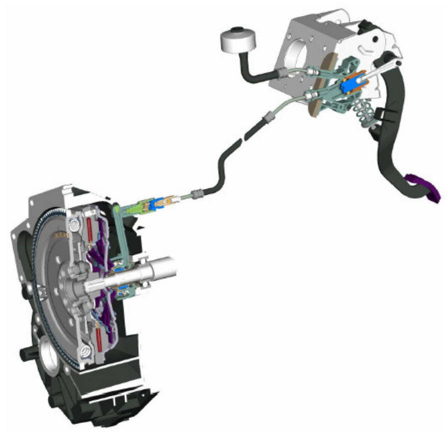

Figure 2.1 : Cable and hydraulic actuator system example [91].

Modern vehicles are using hyraulic systems since the cable systems cause more clutch pedal efforts and customer dissatisfaction. The system we are working on is also using fully hadraulic actuation system.

2.5 Clutch Cover and Pressure Plate

A composition of friction clutch which used on manual transmission are clutch cover, clutch disc and release bearing. A diaphragm spring of clutch cover is connected with cover plate and pressure plate. A diaphragm spring load push pressure plate and clutch disc by clutch pedal.The rotation power of engine is transferred from flywheel, which is connected with crankshaft to transmission throw clutch disc, located between flywheel and pressure plate. When push clutch pedal by driver, release bearing push diaphragm spring and then it is lifted the pressure plate toward vertical-axle direction. When design clutch, it is decided the characteristic of clamp load and release load firstly. It is calculated from the shape of diaphragm spring, pressure plate and cover plate. Also, it is decided the size of clutch disc and cover using its material property. It is very difficult to design the diaphragm spring due to decide its shape, dimension, material, method of machining, heat treatment, durability characteristic [90].

The flywheel is fixed on the crankshaft that revolves at the engine speed. The clutch external structure, the washer spring and the pressure plate are screwed to the flywheel. The clutch torque is generated by the friction of the friction material pads on each side of the clutch disk against the flywheel and the pressure plate. The clutch disk is fixed at the end of the gearbox primary shaft and transmits the generated torque to the driveline. The disk itself presents spring dampers for filtering the engine acyclicity and a flat spring between the friction material pads. The non-linear stiffness of this flat spring has a paramount role in the ease of use of the clutch [89].

The non-linear stiffness characteristic of the washer spring has a dip in the middle; a clever choice of the shape of this characteristic, set by the dimensions of the washer spring, matched by a corresponding flat spring allows to strongly reduce the force needed to fully open the clutch. Figure 2.2 shows the subcomponent of clutch system [89].

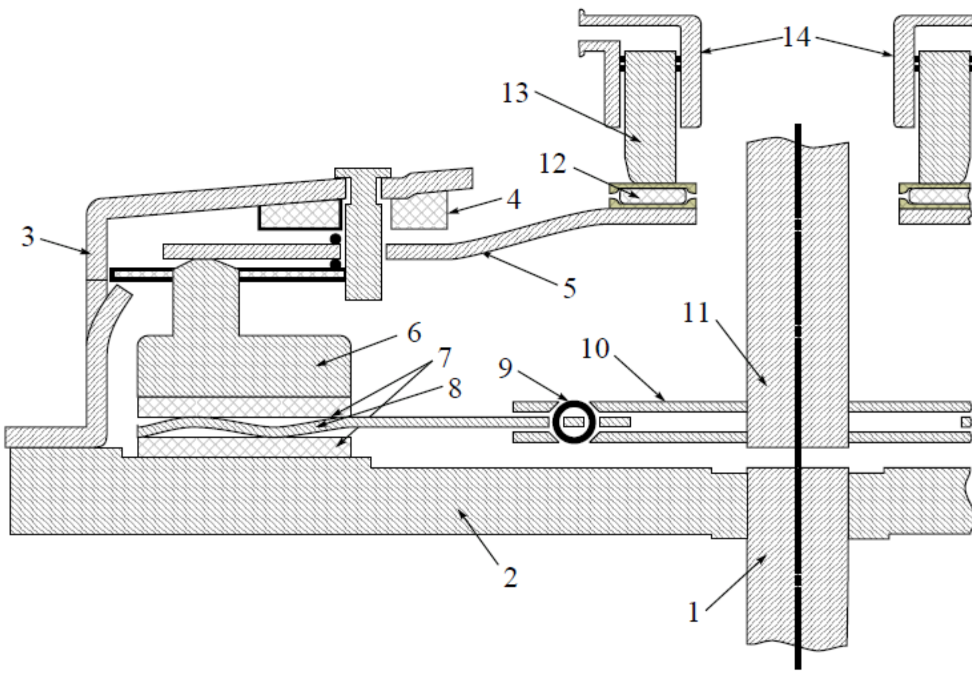

Figure 2.2 : Clutch structure, axial cut. 1 crankshaft 2 flywheel 3 clutch external structure 4 wear-compensation system 5 washer spring 6 pressure plate 7 friction pads 8 flat spring 9 spring damper 10 clutch disk 11 gearbox primary shaft 12 needle roller bearing 13 concentric slave cylinder (CSC) piston 14 concentric slave cylinder (CSC) [89].

The diaphragm spring, which is yoked between cover and pressure plate, is the central part of the pressure plate, and requires the necessary clamp load to transmit the engine torque. For modern clutch the diaphragm spring is often bent very extremely which is caused by the limited package situation. In this case it is important to guarantee the returning of the pressure late after engaging the clutch at high engine speed. If the bending of the diaphragm spring is unfavourable, the returning of the pressure plate could be avoided by the centrifugal force as shown in Figure 2.3 [145].

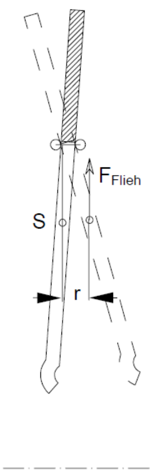

Figure 2.3 : Unfavorable bent diaphragm spring [145].

To avoid this situation the diaphragm spring become specially formed to move the center of gravity to the transmission side. This special contour causes a decrease of the centrifugal force. The shape of such like this diaphragm spring is shown Figure 2.4 [145].

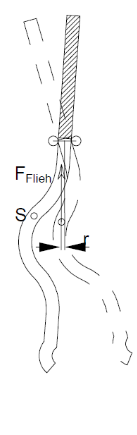

Figure 2.4 : Special shaped diaphragm spring [145].

The second important part is the pressure plate. During every clutch engagement a lot of heat originate at the friction surfaces. So it is important to avoid a distortion of the pressure plate.

This distortion comes up in a conic friction surface which decreases the under this influence. As a result the average friction diameter decreases and the transmittable torque as well. To keep the conic distortion in defined limits the thickness of the pressure plate has to be chosen very carefully to ensure a sufficient heat flow. The pressure plate is a casting part and therefore the burst speed resistance has to be checked. To compensate for the conic distortion the friction surface should be manufactured slightly conic.

The pressure plate cover is an other important part which is the interface to the flywheel. It also serves to provide the counterforce for the diaphragm spring. Furthermore the pressure plate cover is connected to the pressure plate via leaf springs. The leaf springs transmitting the torque and ensuring the clearance between pressure plate and clutch disc during disengagement.

2.6 Clutch Discs:

The clutch disc can be described as the connection between engine and transmission. It transmits the provided engine torque via pressure plate and flywheel to the transmission input shaft. The torque flow will be achieved by clamping the clutch disc between pressure plate and flywheel. Mainly one distinguished two different types of clutch discs, clutch discs with and clutch discs without torsional dampers. Both types are described in the following sections.

2.6.1 Rigid clutch discs

Nowadays rigid clutch discs are only used in combination with dual mass flywheels, which are responsible for damping the torsional vibrations of the engine. These torsional vibrations occur through the periodical combustion of the engine. The damper causes that the torsional vibrations from the engine will be transmitted to the transmission input shaft in a damped characteristic.

A further task of the clutch disc is the compensation of misalignment between crankshaft and transmission input shaft. This can be happens in case of an unfavorable tolerance siuation. This misalignment could brings out idle noise and increased wear at the clutch hub profile. To avoid this phenomenon the most clutch discs are equipped with a radial and axial misalignment compensation which is normally realized at the clutch hub.Figure 2.5 shows two examples of rigid clutch discs with and without misalignment compensation.

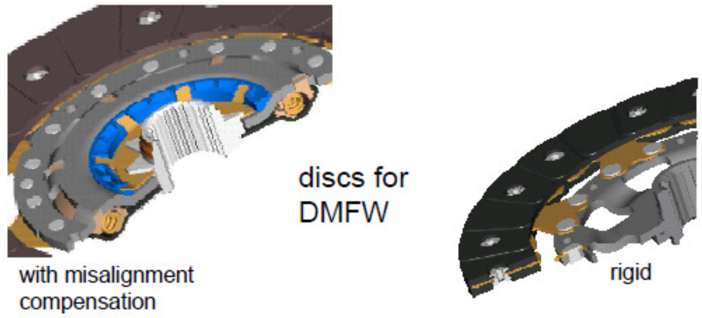

Figure 2.5 : Rigid clutch discs with and without misalignment feature [145].

2.6.2 Clutch discs with torsion dampers

Clutch discs with torsion dampers are only used in combination with stiff flywheels. The upcoming torsional vibrations will be damped by the inertia of the flywheel respectively pressure plate and the torsion dampers of the clutch disc. The damping result of the clutch disc is essentially worse than the damping result of the of a dual mass flywheel. To cover the most important working points it is necessary to apply two or more damper stages for very rattle sensitive transmissions for example. So it is possible to fit in a special tuned idle damper to avoid idle rattle. Figure 2.6 shows a clutch disc with torsion damper.

Figure 2.6 : Conventional damped Disc [91].

The diameter of main damper has a big influence on the axial package situation. Because of the limited space of the drive plate the spring travel is restricted, which leads to an non-damped resonance range within the work area. To reduce these amplitude peaks friction elements with a defined hysterisis are fit in the clutch disc. This kind of damper characteristic is shown in Figure 2.7.

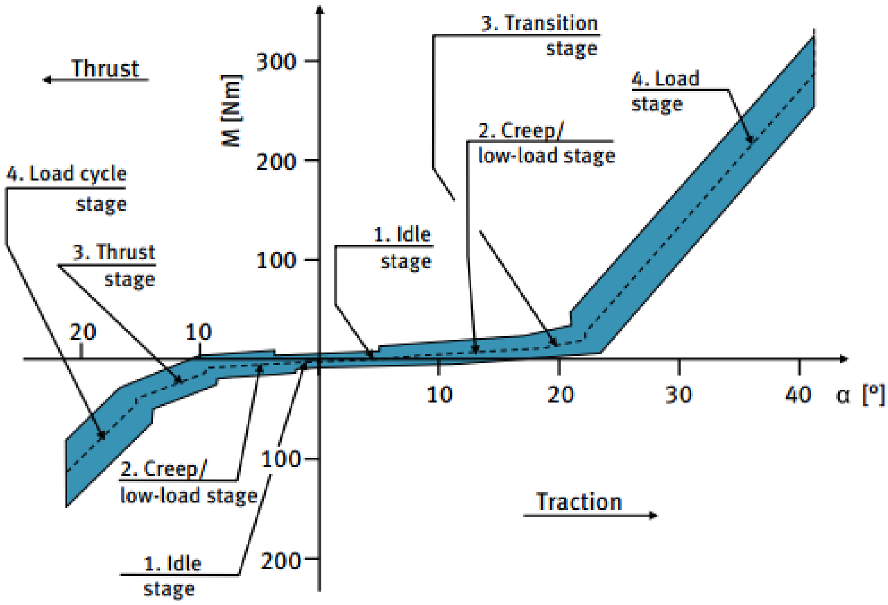

Figure 2.7 : Torsion damper characteristic [91].

The torsion damper characteristic can be modified by variation of the spring geometry and by variation of the number of springs. An other tuning method is movement of the touch point to the damper in the drive plate at two opposite sides. So it is possible to adjust the engagement rotation angle of the damper.

2.6.3 Friction linings and cushion spring

The cushion spring is a very important part of the clutch disc which is used in stiff clutch discs as well as in clutch discs with torsion damper. Usually the cushion spring consist in some wavy spring segments where the friction lining is fit to by rivets. The cushion spring helps to reach a smooth torque increase at vehicle launch. Besides it partly compensates the conic distortion of the pressure plate at high temperatures. Additionally the cushion spring provides a ventilation of the linings which are under high thermic effort. As a result of this properties the safety factor of the clutch gets higher.

The clutch lining can be produced with several procedures. Normally the linings will be winded and subsequently pressed. An other method is to press the linings. The material base for this method is a granulate. Clutch linings need to fulfill some requirements. The most important safety requirement is the burst speed. The wear reserve of linings is round about 0,8-1,0 mm for one lining. That means 1,6-2,0 mm for each clutch disc. The friction coefficient is an other very important value. It depends very strong on the temperature The grooves in the linings are responsible to avoid the sucking effect between clutch disc and pressure plate which can occur through the wavy linings and results in shiftability problems.

2.7 Flywheels

In the following two chapters the most important types of flywheels will be explained. Mainly there are stiff and dual mass flywheels.

2.7.1 Stiff flywheels – Single Mass Flywheels (SMF):

Stiff flywheels are directly fit to crankshaft to decrease the vibrations of the engine. The second task is to provide the friction surface for the clutch disc. Furthermore the flywheel should guarantee the necessary heat flow as well as the starter ring gear assembly and clutch disc. This type of flywheels are mostly used on gasoline engines due to lower krank-shaft vibrations.

2.7.2 Dual Mass Flywheels (DMF)

The dual mass flywheel consists of two masses which are connected via arc springs. The mass which is connected to the engine is the primary mass and the other one which is the counterpart to the pressure plate is the secondary mass. Primary and secondary mass are distortionable connected to each other. Due to the subdivision of the masses the inertia on the transmission side increases without any negative influence to the shiftability. The movement of the resonance range below engine idle speed is an other advantage of the mass splitting to avoid uncomfortable resonances during driving. The arc springs which are connecting the two masses are responsible for the damping of the torsional vibrations. Compared to a conventional clutch disc with torsion dampers the dual mass flywheel compensates the vibrations in a much better way than the clutch disc. This causes by the essentially longer travel of the arc springs. Because of the less inertia of a stiff clutch disc which is used in combination with dual mass flywheels, the synchronizers are less loaded which results in a better shiftability. The less inertia is caused by the missing damper unit. Figure 2.8 shows one example for a dual mass flywheel.

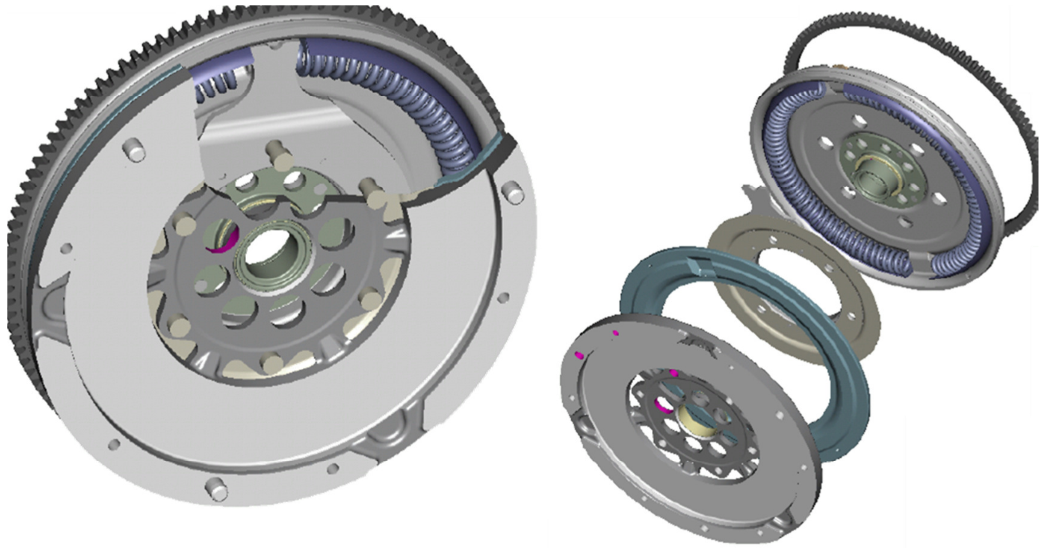

Figure 2.8 : Dual Mass Flywheel [91].

3. PURPOSE, SCOPE AND METHODOLOGY

As we discussed previously, aircraft, automotive, mechanical and civil engineering applications are subjected to dynamic loads. Therefore, it is important to design dynamic systems, considering such loads. Amplitude, frequency and anything related with load are important and need to be considered before starting the design. Even though the optimization itself has a big progress and improvement since long years, optimization of dynamic systems could not make that jump due to big expense of computational needs. With the huge development of computational technologies during the last decade, studies and works have started and going on. Furthermore, these studies still needs huge amount of research.

As it can be seen from latest papers, there are many different studies about optimization of dynamic systems. However, it can also be seen that studies are very specific to certain systems, and there are no generalized methodologies for the optimization techniques. Well, the reason is obvious that is, some methods are more suitable for some systems, and others not. Therefore, it is always necessary to think an optimization technique for certain problems. While looking for this technique, it must be decided, if we are looking for a local optimum (which might be a cheaper but not the best solution) or if are looking for the global optimum (best solution). For this purpose, “Global optimization techniques” will be used and especially “Genetic Algorithms” will be tried to be modified and make suitable for the dynamic problems. While doing this, no precondition will be asked such as differentiability, convexity, constrained or unconstrained condition etc…

It can be seen that some of the studies and papers are used in Genetic algorithms for the purpose of optimization of dynamic systems. However, they are very rare and very specific for some certain systems. As a scope of this study, we will also consider multi-criteria optimization (optimization considering more than one objective function) which is also rare topic.

As the scope of this study, we will deal with mostly spring mass and damper systems. Even if the problem is more complicated than this, it will be transformed into a lumped system formed of spring, mass and damper systems. However, this system could have material damping (spring hysteresis) or friction surfaces. If we face such complications, we will also add them into the problem. We will also add any nonlinearity (such as nonlinear spring or damper characteristics or material properties) into the problem. Figure 3.1 shows some spring mass and damper systems, which can be optimized for frequencies, amplitude, or both.

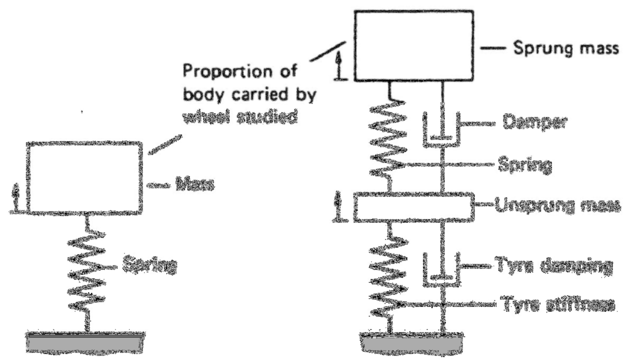

Figure 3.1 : Some spring, mass and damper systems [146].

As it can easily be seen, simple mass and spring system can only be optimized under certain conditions (bounds on mass, limitations on spring property or bounds on displacement). Otherwise, results can be infinitely stiff or soft spring, which obviously cannot be possible. Multi-objective optimization can be applied for the Figure 3.1. While we can keep the mass, springs and dampers as variables, we can select optimization criteria as min natural frequency or min amplitude. For that purpose, we firstly need to write equation of motion $\boldsymbol{M(X)\ddot{Z}(t) + C(X)\dot{Z}(t) + K(X)Z(t) = P(X,t)}$ (3.1) in matrix form using the variables and solve the differential equation for the objective functions subjected to any necessary constraints.

$$\boldsymbol{M(X)\ddot{Z}(t) + C(X)\dot{Z}(t) + K(X)Z(t) = P(X,t)} \qquad \textbf{(3.1)}$$

Once we solve the differential equation, we have eigenvalues, eigenvectors, modal matrix, equations which give displacement vs. time and natural frequency terms, etc… After finding all necessary equations, objective functions can be selected and solving them with respect to variables subject to constraints. Optimization algorithm will be chosen among the global optimization techniques (genetic algorithm or simulated annealing) suitable for the optimization problem and will be modified it to use the algorithm for the practical problems if necessary.

With all the findings we obtain in these works and studies, coupled mechanical systems and their dynamic responses will be optimized and effect of this optimization on a powertrain system (transmission, clutch etc…) will be investigated by the help of FEM or similar models. However, the big powertrain system suppliers have those models amd they are confidetial. It is also very hard to find a study or paper for the optimization of these dynamic and complex systems for an academic approach. Figure 3.2 can give an idea for the system, which this optimization algorithm might be applied.

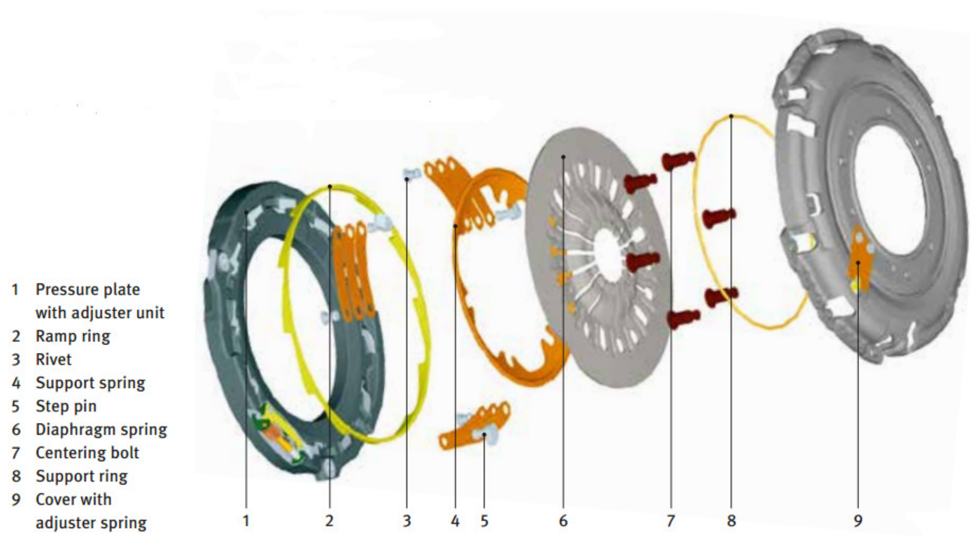

Figure 3.2 : An example of a clutch system [91].

3.1 Problem Description

As discussed before, for whole vehicle variants, current trend is to increase engine torque. It is also very important to improve driver ergonomics and comfort while increasing engine torque. However, in order to transfer engine torque without allowing any slippage on the clutch system, clamp load on the pressure plate should also be increased. But this means also increasing the clutch pedal load which is causing degradation on the driver comfort very much. To be able to keep the clutch pedal load bigger and bigger clutch systems need to be designed which has also a limit. Inertia of the clutch system, package of the clutch area is limiting this.

Regardless of these limitations, bigger clutch has a positive effect on the clutch system lifetime.

Considering all these boundary conditions, it is very important to design an optimum clutch system. Once the initial design is built, it is very common to build some prototypes and try them on prototype vehicles, and perform different test (e.g. durability, performance, noise, vibration, etc…).

One of these tests is NVH (Noise, Vibration, and Harshness) test. It is observed that, transmission system has a rattle noise during clutch engagement (while the driver pulling his foot from clutch pedal). To understand the problem, a number of measurements have been conducted and some results given in the previous report and some of them will be shared in the next pages.

As we mentioned above that the clutch systems mostly deal with rotational vibrations of engine, it is very likely these systems are exposed to axial vibrations from engine crankshaft as well. It is seen in the measurements and noise records; this rattle noise is caused by these axial vibrations. But the clutches generally don't have any specialized equipment to damp such vibrations. There are couple of reasons for this. The first but not the last is the package area of the system. Package area (assembly area) is very limited in clutch system and there is no available space to put some springs and dampers to reduce such vibrations. The second reason is, these axial vibrations are normally very insignificant to consider in the calculations and most of the vibration is damped on the diaphragm spring and cushion springs. But, this may sometimes lead some unexpected results in the powertrain system as the rattle noise that we will discuss and try to resolve in this study by using global optimization techniques. These techniques and results will be used on mathematical models and will be tried to prove by physical measurements as discussed above

So, scope of these studies can be split into 2 parts. We can say that the first part is to build of mathematical model of the system and the second is the physical measurements with design intent and other possible parts. Surely, these two parts will be handled together and results from both sides will feed each other.

The mathematical model will also cover two separate items. One of them, of course, is going to be related with the rattle issue which is caused by the axial vibrations of pressure plate. So these axial vibrations must be handled. As geometrical and physical variables which caused these vibrations is much related with clutch system release load characteristics (which are also related with the clutch pedal load characteristics) second mathematical model must calculate the release load characteristics and pedal load characteristics.

4. OPTIMIZATION of AUTOMOTIVE DRY CLUTCH SYSTEMS

A multi-objective pareto optimal solution of a clutch system is sought by using the pedal characteristics (namely, release load characteristics) and vibration level of pressure plate as objective functions. The first objective function is the curve of pedal characteristics which is affected by a number of variables such as the geometry of diaphragm spring, material properties, release bearing stroke, dimensions of the pedal mechanism etc. All these variables are chosen as design variables in the related optimization problem. Figure 4.1 is an example for a clutch pedal load curve.

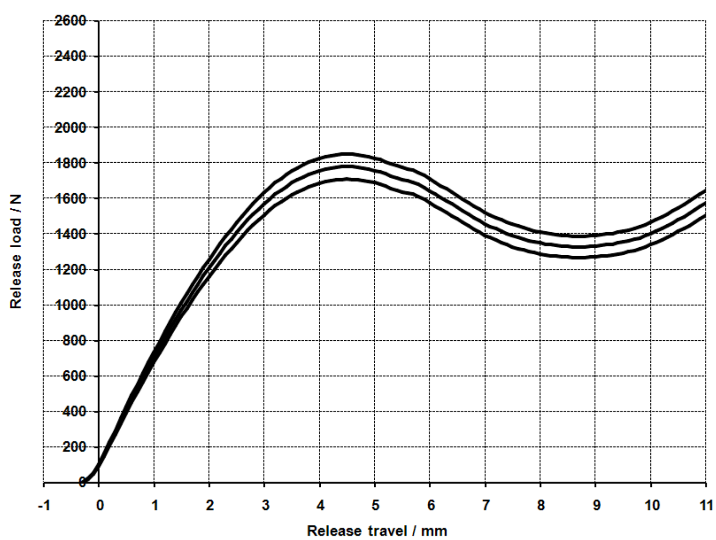

Figure 4.1 : Release load curve for a clutch system.

As mentioned above, the other objective function is to minimize axial vibrations of the clutch pressure plate while the input vibration level remains unchanged. Figure 4.2 shows the measurements taken from the engine crankshaft, flywheel, clutch cover and pressure plate. The aim is to reduce pressure plate vibrations as much as we can while keeping the pedal characteristics given in the specifications which are directly related to the release load curve. If design parameters are chosen without considering the axial vibrations of the pressure plate, excessive vibrations may emerge and may cause interruptions in torque transmission during engagement and disengagement of the clutch. Such an interruption in torque transmission is the main cause of the well known transmission gear rattle problem.

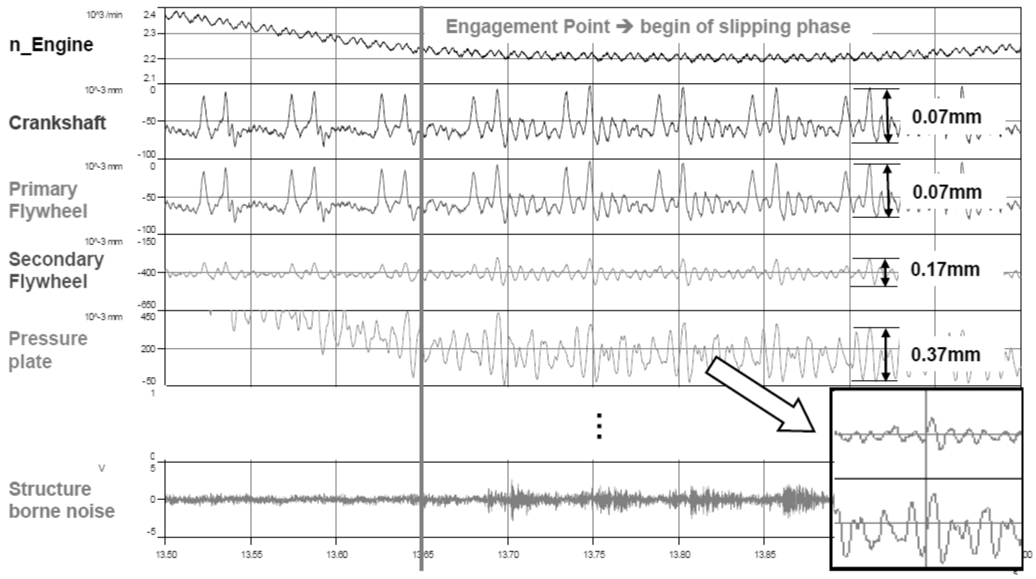

Figure 4.2 : Axial vibration measurements on crankshaft, primary and secondary flywheel and pressure plate. And its relation between rattle noises measured on the transmission.

As we have two basic objective functions, it will be a good idea to separate them from each other in the first place and try to build up their own mathematical models and find out how they are affected by the design variables. While trying to build up mathematical models, physical measurements will also be performed and compared with simulations.

As can be seen in Figure 4.2, the clutch system considered in this study works like an amplifier which results in transmission gear rattle during clutch engagement. Our objective is to decrease the amplitudes of vibrations of the pressure plate while keeping the clutch pedal force as low as possible.

Figure 4.3 shows the necessary inputs for the pedal characteristics simulation briefly in a chart as explained before. Beforehand, we will investigate three objective functions separately, try to understand how they are affected by the design variables and try to derive corresponding mathematical models. These mathematical models will also be verified by making comparisons with the experimental measurements and simulation results.

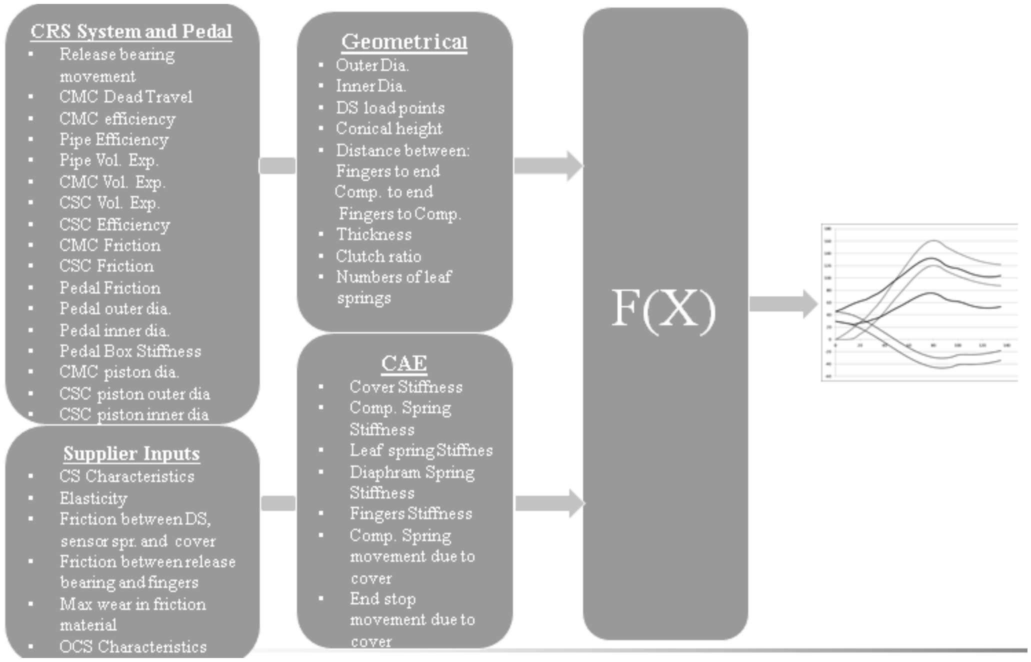

Figure 4.3 : Inputs required to simulate pedal characteristics.

4.1 Modeling of Clutch System for Release Load Characteristics

Parameters which affect axial vibrations of the clutch system will also affect clutch release load characteristics which will be discussed in the upcoming sections in detail. Since the clutch pedal load characteristics are dependent on release load characteristics, even if the vibration problem is well resolved, pedal load might be too high which may cause the clutch system becomes unusable. For that reason, clutch release load characteristics will be simulated first, and then optimization problem will be solved.

The clutch and DMF system that are the subject of this study is shown in Figure 4.4. Following, dynamic models of components in this clutch system are derived. Please note that primary side of the flywheel and some internal parts could not be shown in the figure not make it more intricate.

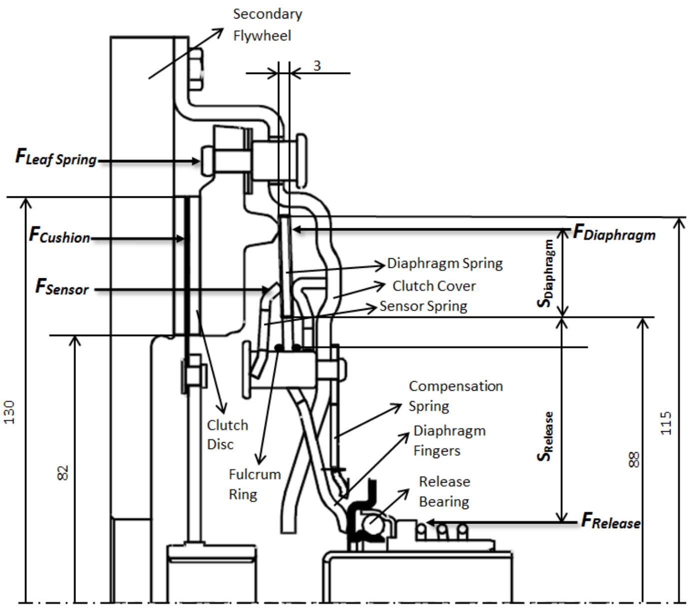

Figure 4.4 : Geometrical data of the simulated clutch and DMF system (at engaged position) [135].

4.2 Clutch Release Load Characteristic

Being one of the crucial elements determining the clutch dynamic behavior, clutch release load characteristic is derived from clutch release curve. A typical clutch release curve, pressure plate spring curve and lift off curve are shown in Figure 4.5. While clutch diaphragm spring is the most significant component influencing clutch release load characteristic, it is also affected by leaf springs on clutch cover (which helps pressure plate disengagement), cushion springs between friction surfaces of clutch disc and clutch cover stiffness. Following sections explain how these components are modeled.

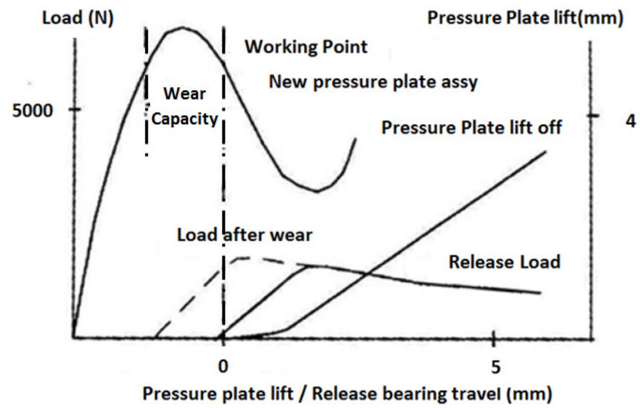

Figure 4.5 : An example of a clutch pedal curve and its important points [136].

As mentioned in the abstract, two objective functions such as pressure plate vibrations and clutch pedal curve characteristic are used in this study. Clutch pedal curve characteristic is calculated by dividing the release load curve to release system's ratio and release load curve gives us the load on the release bearing as the release bearing travel changes. Note that while calculating these curves, they are modeled parametrically by using cubic spline polynomials in the mathematical model of the clutch system and employed in the optimization runs.

4.3 Diaphragm Spring

Since the diaphragm spring has two sections having completely different characteristics, it is divided into two parts and their mathematical equations are derived separately, e.g., see Figure 4.6. While the disc spring is discussed below in Section 4.3.1, finger springs are studied in Section 4.5.1 along with the other components that are modeled by using finite element method (FEM).

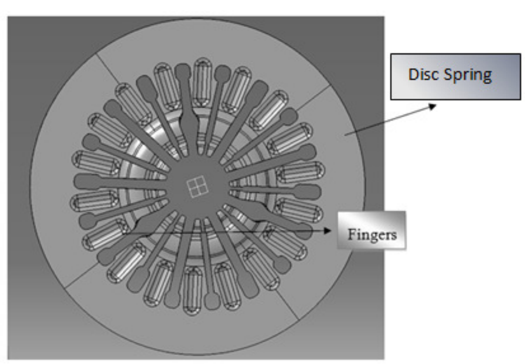

Figure 4.6 : Two sections of the diaphragm spring.

4.3.1 Modeling of disc springs

The Almen-Laszlo formula [117] is commonly used to model the disc springs in practice [110-113].

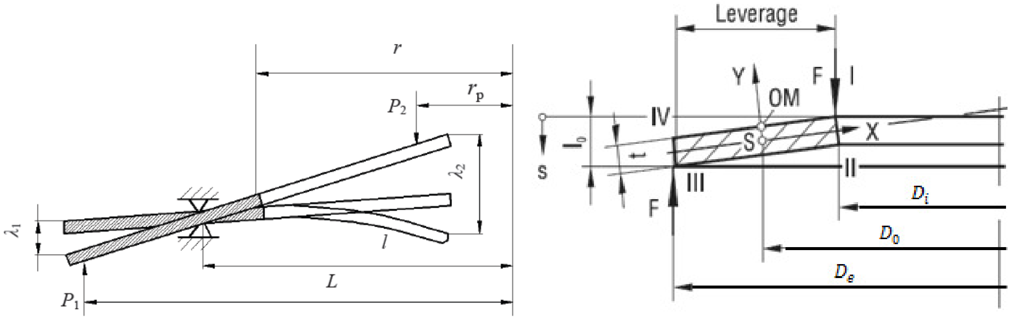

Figure 4.7 : Deformation of the diaphragm spring when its fingers are loaded [111] (on the left side) and its cross section at reference position [122] (on the right side).

The load P_1 acting on the outer radius of diaphragm spring shown in Figure 4.7 can be calculated as follows:

$$P_1 = \frac{\pi E t \lambda_1 . \ln(\frac{R}{r})}{6(1-\rho^2)(L-l)^2}\left[\left(h - \lambda_1 \frac{R-r}{L-l}\right)\left(h - \frac{\lambda_1(R-r)}{2(L-l)}\right) + t^2\right] \quad \textbf{(4.1)}$$

Where E is the elasticity modulus of spring material, t is the thickness of diaphragm spring, λ_1 is the displacement of outer diameter of disc spring, R is the radius of outer end of disc spring, r is the inner radius of disc spring, ρ is the Poisson's ratio of spring material, L is the position of load P_1, l is the diameter of wire ring and h is the internal cone height of an unloaded diaphragm spring [111].

The load P_2 acts on the inner radius of diaphragm fingers. This leads to a large deformation $\boldsymbol{\lambda_2}$ on diaphragm fingers and also $\boldsymbol{\lambda_1}$ on the outer radius of disc spring as shown in Figure 4.7. The release load P_2 is given by [111].

$$P_2 = \frac{\pi E t \lambda_1 . \ln(\frac{R}{r})}{6(1-\rho^2)(\mathrm{L}-\mathrm{l})(\mathrm{r}-\mathrm{r_p})}\left[\left(h - \lambda_1 \frac{R-r}{L-l}\right)\left(h - \frac{\lambda_1(R-r)}{2(L-l)}\right) + t^2\right] \quad \textbf{(4.2)}$$

Where r_p is the inner radius of diaphragm fingers where release bearing connection exists [111].

The cross section of disc spring part of diaphragm spring is presented in Figure 4.7 on the right without showing the diaphragm fingers, which is used to derive load equations of disc spring. The Almen-Laszlo formulas assume that the disc spring rotates around a center, which can be calculated by using the following formula [122-123].

$$D_0 = \frac{D_e - D_i}{ln(D_e/D_i)} \quad \textbf{(4.3)}$$

Then, the load on disc spring is given by

$$F = \frac{4E}{1-\rho^2} \cdot \frac{t^4}{K_1 \cdot D_0} \cdot \frac{s}{t} \cdot \left[\left(\frac{h}{t} - \frac{s}{t}\right) \cdot \left(\frac{h}{t} - \frac{s}{2t}\right) + 1\right] \quad \textbf{(4.4)}$$

$$K_1 = \frac{1}{\pi} \cdot \frac{\left(\frac{\delta - 1}{\delta}\right)^2}{\frac{\delta + 1}{\delta - 1} - \frac{2}{ln\delta}} \quad \textbf{(4.5)}$$

Where s is the deflection of disc spring, D_e is the outer diameter of disc spring, D_i is the inner diameter of disc spring and $\delta = D_e/D_i$ [122]. The disc spring characteristic, which can be found in literature [122-123], is shown in Figure 4.8 as h/t ratio changes.

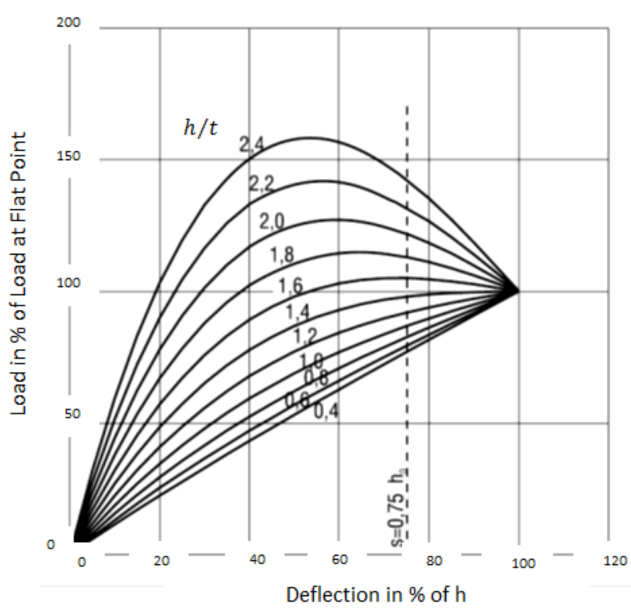

Figure 4.8 : Load-deflection characteristic for Belleville washers [122].

By using the formulations given above and geometrical properties of the clutch employed in this study, the clamp load curve is obtained as shown in Figure 4.9 where the maximum achievable load is calculated as approximately 18,000N.

Once the clamp load curve is built, operation point (λ_1) on the curve needs to be identified. In order to identify the value of λ_1, clamp load should be known which is the normal force on the clutch pressure plate at the engaged position and must be determined correctly to ensure the transmission of engine torque. A safety factor is also considered to guarantee no slippage on the clutch which is called the Slip Safety Factor (SSF) which is typically chosen as 1.2 [136]. The clamp load is calculated by using Eq. (4.7).

$$D_m = \frac{2}{3} \cdot \frac{D_e^3 - D_i^3}{D_e^2 - D_i^2} = 0.212m \qquad \textbf{(4.6)}$$

$$SSF(Slip\ Safety\ Factor) = \frac{F_{clamp} \cdot D_m \cdot \mu}{T_e} \geq 1.2 \qquad \textbf{(4.7)}$$

Where D_m is the diameter of gyration (centroid of friction facing surface diameter to which torque is applied to calculate the clamp load), F_{clamp} is the clamp load at the clutch operation point, μ is the friction coefficient of friction and T_e is the maximum engine torque.

By using Eq. (4.7), the minimum clamp load is calculated as 9,400N and nominal clamp load can be selected as approximately 10,000N. Considering this clamp load, nominal operation point is calculated as 8.1mm from the Figure 4.9.

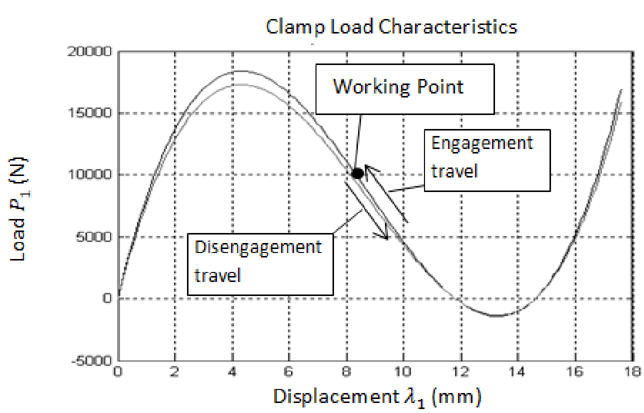

Figure 4.9 : Clutch clamp load characteristics. The curve at the top shows the spring compression, the curve at the bottom shows the decompression phase.

4.4 Modeling of Cushion Spring

The cushion spring significantly affects the release load characteristic. Its characteristic load curve used this study is shown in Figure 4.10 on the left [134] which is obtained from experimental measurements (e.g., see Section 5). It is parametrically modeled by using cubic spline polynomials in the mathematical model of the clutch system and employed in optimization runs.

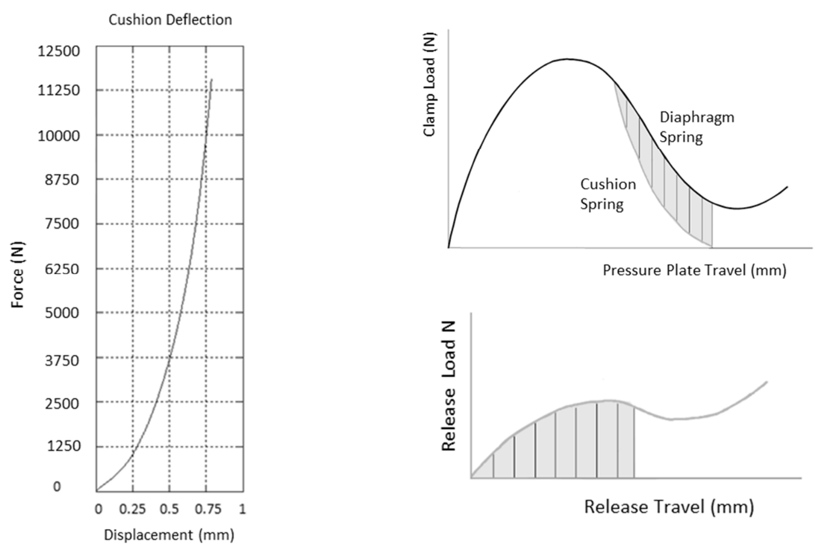

Figure 4.10 : The cushion spring and its characteristics for the new and worn springs [134].

The cushion spring segment of clutch disc provides a smooth release curve and prevents large loads on the pedal load curve. In order to handle release load curve, the difference between diaphragm spring and cushion spring is calculated and divided by clutch ratio. The shaded areas in Figure 4.10 on the right show this difference and related release load is on the left side.

4.5 Modeling of Diaphragm Spring Fingers, Clutch Cover, Compensation Spring and Leaf Springs

FEM is used to model diaphragm spring fingers, clutch cover, compensation spring and leaf springs. It is noteworthy that diaphragm spring stiffness is strongly related to the clutch release load characteristic.

4.5.1 Stiffness of diappragm spring fingers

In order to calculate the diaphragm spring stiffness under loading, it is important to accurately model the stiffness of fingers. To this end, the slave cylinder bearing is modeled as a solid cylinder on the fingers of diaphragm springs and a contact model is defined between the bearing and fingers. There are a number of studies reported in literature used shell elements [115] or tetrahedron solid mesh elements, e.g., [140], [142-143]. Quadrotic tetrahedron solid elements (with 10 nodes having three degrees of freedom (DOFs) at each node) are used in our study. In order to analyze only finger area of the diaphragm spring, all DOFs of the fulcrum ring connection are restricted as boundary conditions and distributed pressure is loaded on the release bearing. Finite element model and CAE analysis result of the diaphragm spring and bearing can be seen in Figure 4.11 and Figure 4.12.

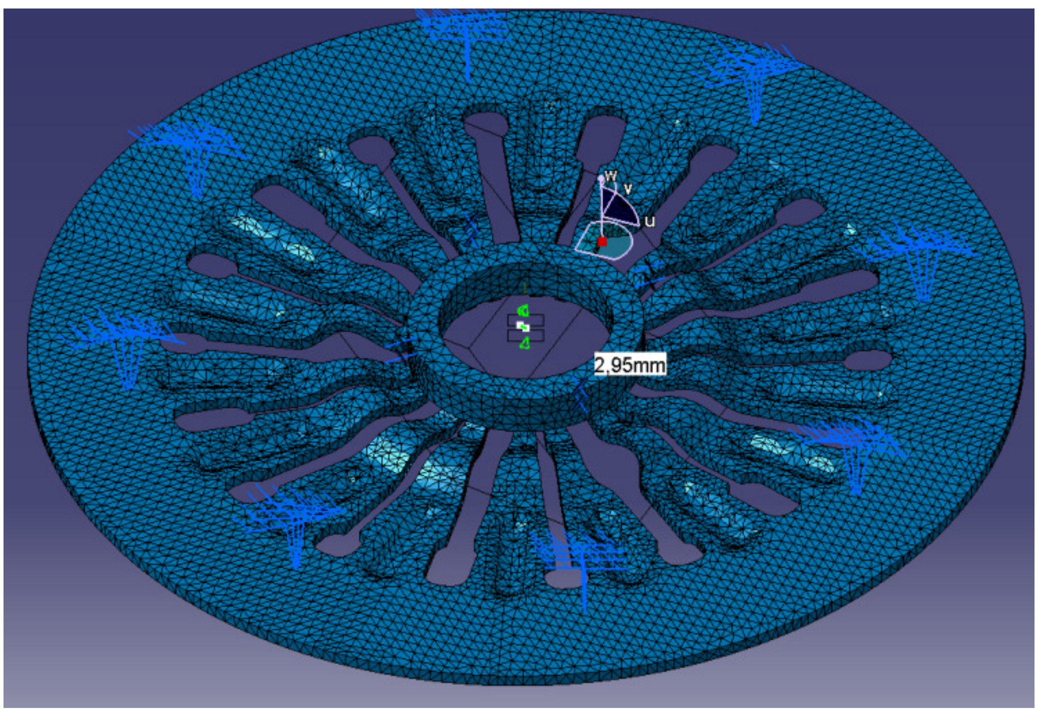

Figure 4.11 : Finite Element Model of Diaphram spring and bearing.

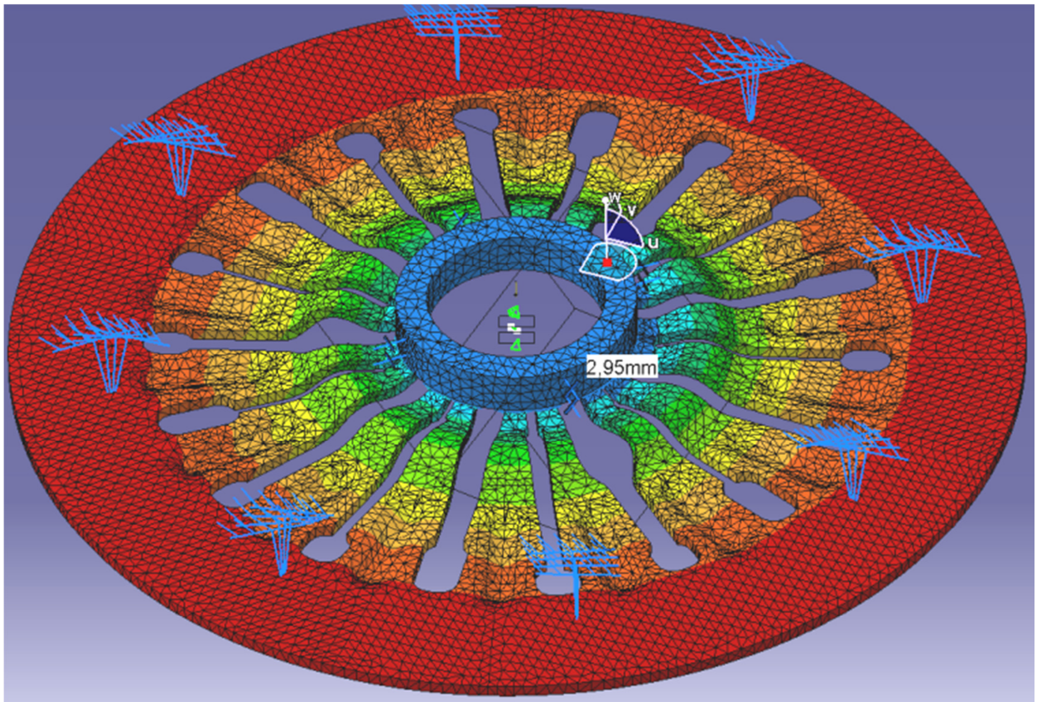

Figure 4.12 : Deflection on the fingers under 2000N of release load on slave cylinder bearing.

Based on FEM solutions, characteristics of fingers are found to be similar to a linear spring, which is also verified by experimental measurements, e.g., see Figure 4.18 (i.e., release load curve). Linear ramp up behavior of first part of the release load curve shows linear spring characteristics of diaphragm fingers. Many measurements and simulations are reported in literature on this issue, e.g., see [113, 115, 140-141].

4.5.2 Clutch cover stiffness

The clutch cover stiffness is also another important parameter affecting the release load characteristic. By using parabolic tetrahedron solid elements in FEM model, distributed load is applied to fulcrum ring area and all DOFs of the bolt holes around the cover where the cover is connected to flywheel are restricted. Note that the diaphragm spring position, compensation spring position and its end stop position are different between the loaded and unloaded positions of diaphragm spring. These positions change under the loading during the engagement and disengagement of clutch and significantly affect the release load curve characteristic. In order to identify these positions under different loading conditions, FEM solutions are used and relevant spring coefficients are calculated (Figure 4.13).

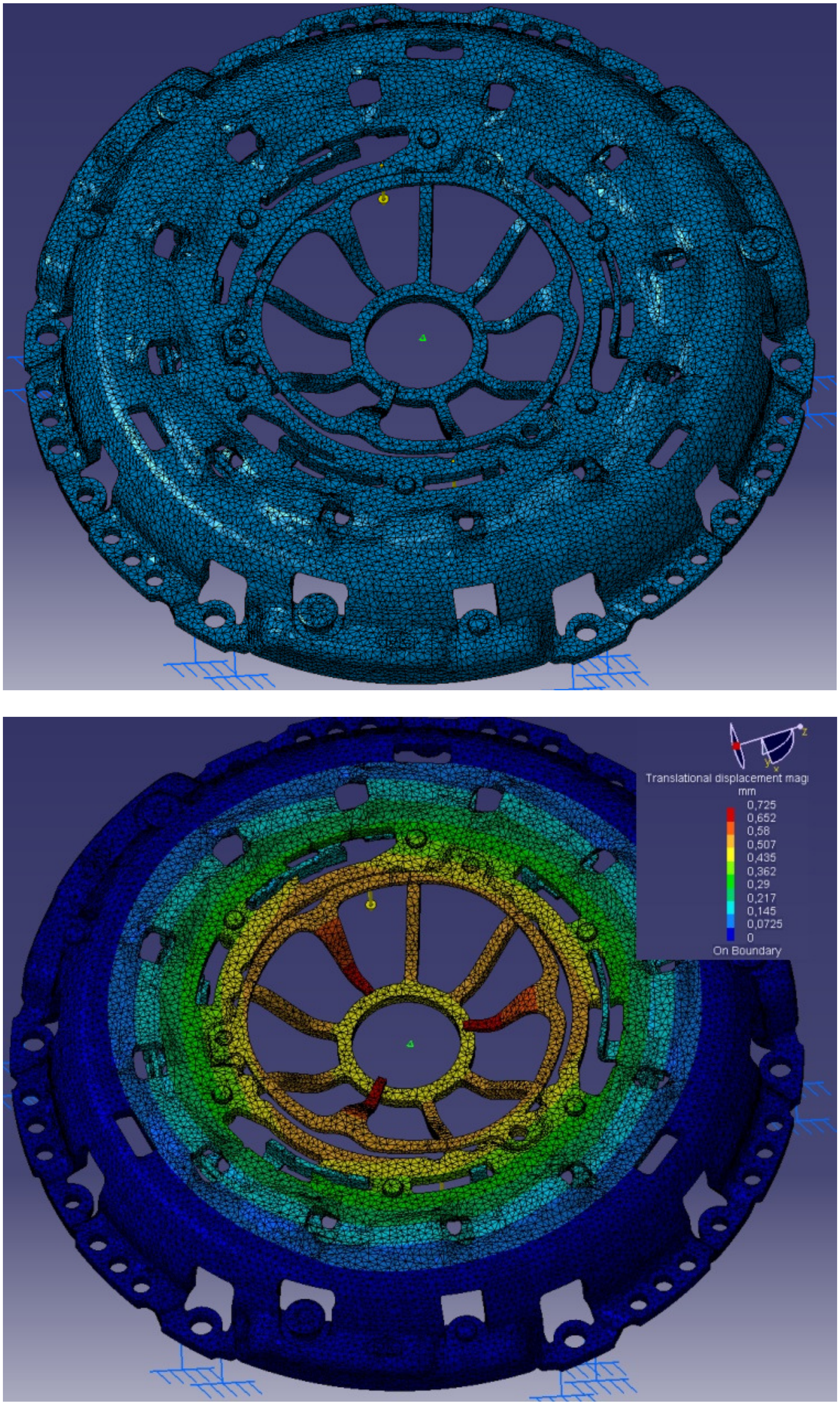

Figure 4.13 : Finite Element Model and Finite element analysis of clutch cover.

4.5.3 Compensation spring

The compensation spring is used to achieve a more flat release load characteristic which also leads to a more flat clutch pedal characteristic. The compensation spring is activated after the maximum loading is reached during its release. After this point, the release load begins to reduce dramatically. In order to prevent this, a linear spring is used to increase the pedal loads starting from the maximum release load to the end of release bearing travel; thus, more flat pedal characteristic can be achieved [135]. Figure 4.14 shows the working principle of compensation spring. Figure 4.15 also shows the FE analysis of compensation spring under 200N load. After the number of analysis, it is shown that the spring shows the linear properties.

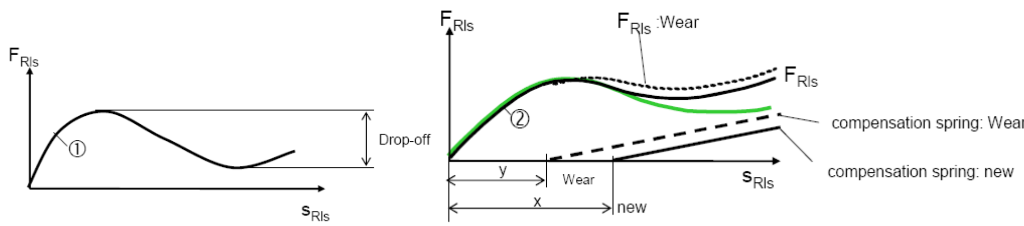

Figure 4.14 : Compensation spring working principle [135].

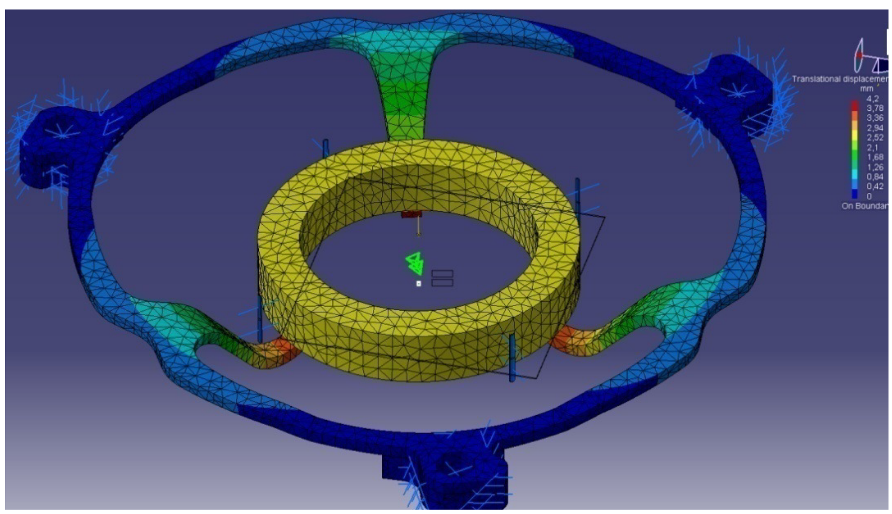

Figure 4.15 : Compensation spring deflection analysis under 500N load.

4.5.4 Leaf springs

Leaf springs are used to pull back the pressure plate from the clutch disc during the disengagement of clutch. These springs are thin planar plates located in front of the diaphragm spring. Due to the clamp load on the pressure plate, these plates are bent and have the function of a spring. They are only used to guarantee that the pressure plate does not touch to clutch disc after the disengagement. Leaf spring properties are also found out linear as expected. Figure 4.16 shows FE model and analysis result of a leaf spring.

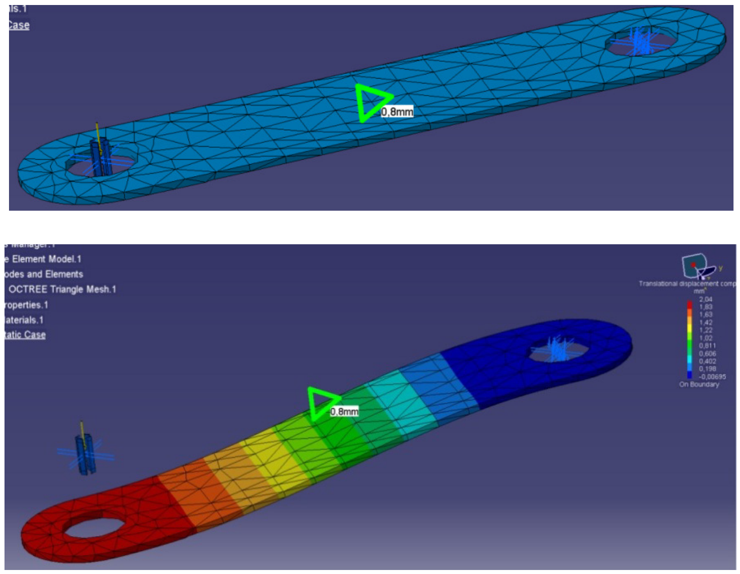

Figure 4.16 : Finite element model and analysis of a leaf spring.

In FEM models of compensation spring and leaf spring, parabolic tetrahedron solid elements with 10 nodes are used and all DOFs of rivet holes of these springs used for the connection to the cover are restricted from movement or rotation. The load between 0 to 400N is applied to leaf spring and distributed pressure is applied to the release bearing which contacts with compensation spring fingers at the release bearing travel where this contact condition is occurred.

FEM solutions of diaphragm spring, clutch cover, compensation spring and leaf spring are shown in Figure 4.17. Note that these results are used in the mathematical model of clutch system to derive their parametric models which is employed in the optimization runs.

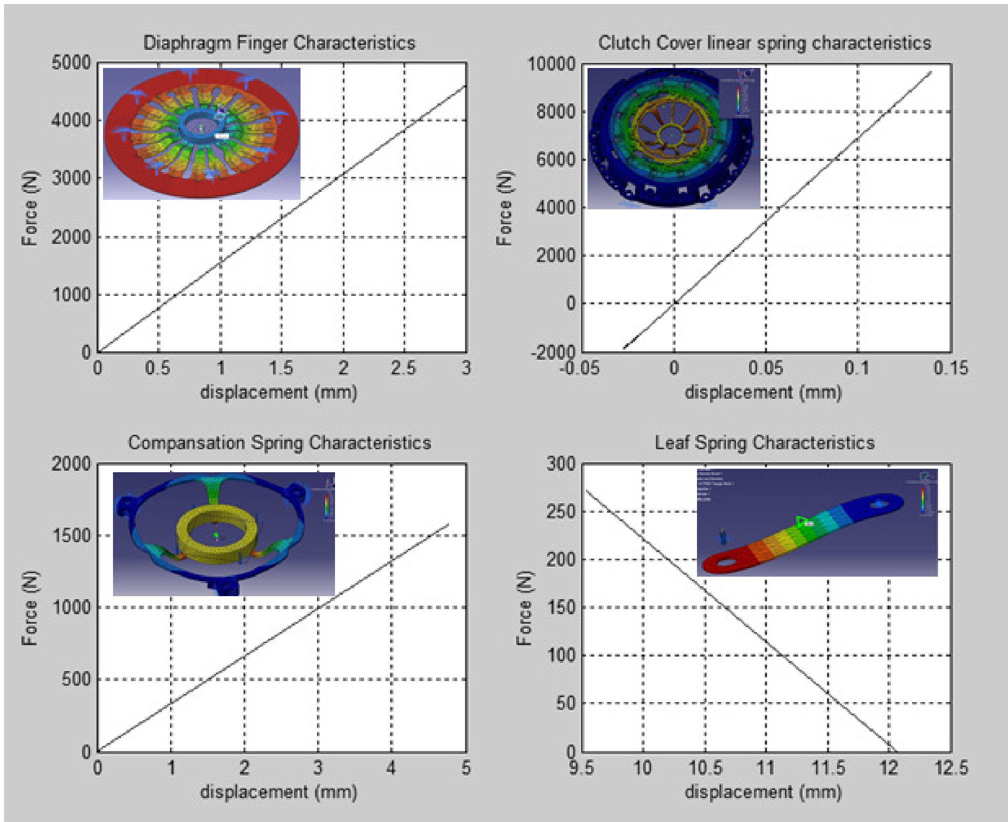

Figure 4.17 : FEM solutions of diaphragm spring fingers, clutch cover, compensation spring and leaf springs.

The usage of FEM was necessary since the spring characteristics of subcomponents are needed separately. Experimental results give us only total system behavior as shown in Figure 4.18 instead of handling the spring characteristics of all subcomponents separately. Furthermore, once these spring characteristics are identified, it is very easy to change the spring coefficients during optimization runs.

Once these numerical results are obtained and used in the calculations of clutch release load curve, the results are compared with experimental measurements (e.g., see Figure 4.18).

4.6 Derivation of Release Load Curve and Lift Off Curve

By the use of above results, we have all necessary inputs to calculate the release load characteristic and lift off curve. A simplified formulation to calculate the release load for pressure plate movement "*s*" is given by

$$F_R = \frac{F_D - F_{cushion} - F_{sensor} - F_{leaf}}{i} + F_{comp.} \quad \textbf{(4.8)}$$

where F_D is the diaphragm spring load, $F_{cushion}$ is the cushion spring load, F_{sensor} is the sensor spring load, F_{leaf} is the leaf spring load, $F_{comp.}$ is the compensation spring load which is zero at the beginning of disengagement movement of the release bearing and i is the clutch ratio. These forces are shown in Figure 4.4. Then, the clutch ratio is given by

$$i = S_{Release}/S_{Diaphragm} \quad \textbf{(4.9)}$$

where $S_{Release}$ and $S_{Diaphragm}$ are respectively distances between loads and connection rivets shown in Figure 4.4.

Sensor spring load in Eq. (4.8) is created by the sensor spring shown in Figure 4.4 and it increases as clutch friction faces get worn. After a certain level of load, clutch adjustment system sets in and sensor spring load decreases to the initial value. Detailed information about clutch adjustment system can be found in [135].

It can be seen in Eq. (4.8) that the clutch cover or diaphragm spring finger deflection is not considered in force equilibrium. However, while the load on these components changes during the engagement and disengagement process, their stiffness affects the positions of diaphragm fingers and subsequently the bearing position, which also determines the complete release curve characteristic. These issues are considered in the simulations to determine the mathematical model of the system and calculate the release characteristics of the clutch examined in this study.

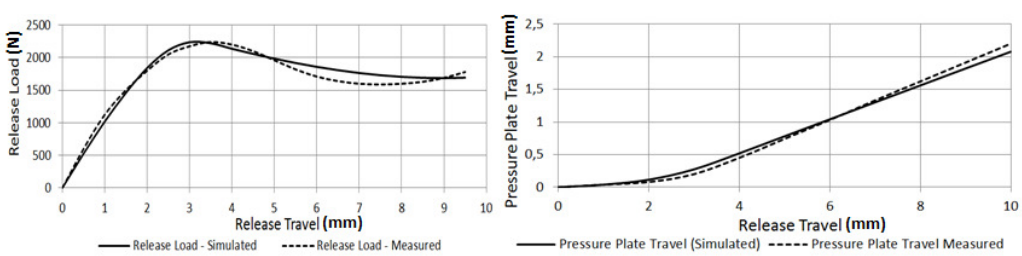

Figure 4.18: The release load and lift off curves.

The release load on clutch fingers and lift off curve (displacement of pressure plate and clutch finger travel) are simulated and compared with the measurements in

Figure 4.18. These curves are the basic clutch characteristics and can be used to characterize a clutch system [136].

As mentioned earlier, the release load on clutch fingers is directly related to the axial loading of clutch and load transferred to clutch pedal through clutch release system. It is observed in Figure 4.18 that the simulation results are very close to measurements and can be used for solving the optimization problem. Optimization results are obtained by using analytical formulas, spring characteristics of different sub-components which are obtained by FEM and parametric curve of cushion spring. Measurements are obtained by the clutch test bench which is explained in detail in section 5.1.

5. TEST BENCHES AND TEST PROCEDURES

5.1 Load and Travel Measurements

Figure 5.1 shows the clutch test bench and component layouts for release load, release travel, clutch pedal load and pressure plate travel measurements. Load sensors are PULS HT1 series compression load cell type of 50kN capacity. Travel sensors are Alfa Electronic TLM type magnetostrictive displacement sensor with +/- 30 μm precision.

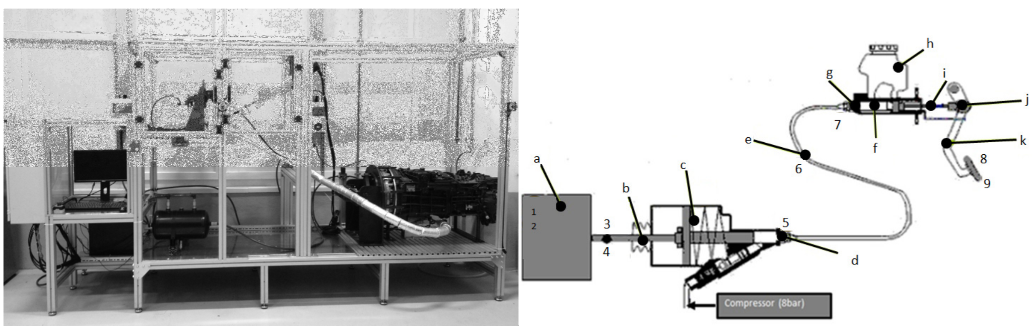

Figure 5.1 : Test bench for clutch pedal measurements.

System components of the test bench shown in Figure 5.1 are as follows:

a) Clutch pressure plate, disc, bearing, fork
b) Clutch slave cylinder push rod
c) Clutch slave cylinder
d) Clutch pipe end fitting
e) Clutch hydraulic pipe
f) Master cylinder
g) Clutch pipe end fitting
h) Reservoir
i) Clutch pedal push rod
j) Clutch pin point
k) Clutch pedal

Sensors on the test bench shown in Figure 5.1 are as follows:

1. Force measurement from pressure plate
2. Displacement measurement from pressure plate
3. Force measurement from clutch slave cylinder push rod
4. Displacement measurement from clutch slave cylinder
5. Pressure measurement from clutch slave cylinder inlet (50 bar clutch hyd. DOT4)
6. Thermocouple on clutch pipe
7. Pressure measurement from clutch master cylinder Outlet
8. Force measurement from clutch pedal
9. Displacement measurement from clutch pedal

5.2 Dynamic Vibration Measurements

In order to measure and understand the vibration phenomena in the clutch and flywheel system, a number of displacement and speed sensors and one microphone are equipped on a transmission and engine, and these instrumented parts are used on a vehicle to measure the rattle noise and vibrations on the clutch and flywheel system. Sensor numbers and positions are listed below and shown in Figure 5.2.

a) 3 displacement sensors for crankshaft vibrations at r = 58mm
b) 3 displacement sensors for primary flywheel vibrations at r = 140mm
c) 3 displacement sensors for secondary flywheel vibrations at r = 97mm
d) 3 displacement sensors for pressure plate vibrations at r = 118mm
e) 1 displacement sensor at the end of the transmission housing
f) 1 microphone at the end of the transmission housing (bandpass filter 1000-20000 Hz)
g) 1 speed sensor for the transmission input shaft speed
h) 1 speed sensor for the differential speed

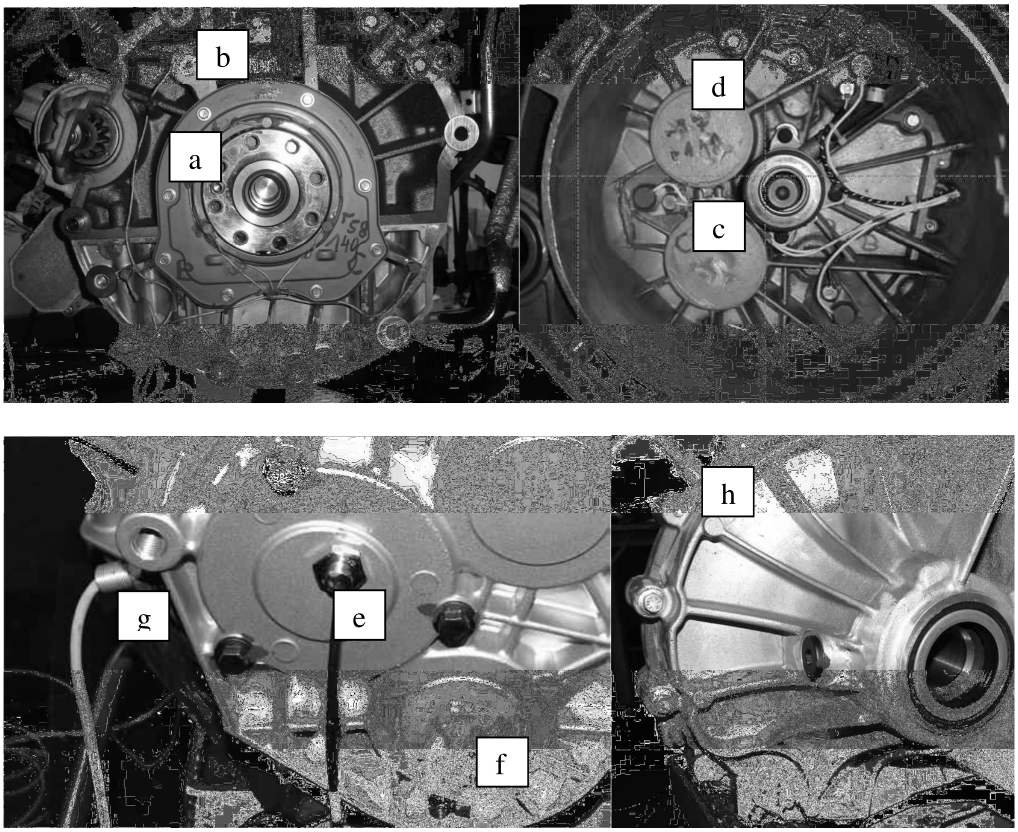

Figure 5.2 : Locations of displacement and speed sensors for vibration analysis.

Measurements are performed while driving the vehicle. Firstly, the engine is accelerated above 3500 RPM, then clutch pedal is depressed (i.e., clutch disengagement). Clutch pedal is released again in one second and the clutch is allowed to engage. Rattle noise and vibration signals of components in the clutch are recorded during the clutch engagement. Note that the rattle noise is irrespective of gear position.

The noise measurement data recorded with the microphone which is at the end of the transmission housing is shown in Figure 5.3 where it is observed that around the engine speed of 2500 RPM is the worst possible range for the emergence of rattle noise.

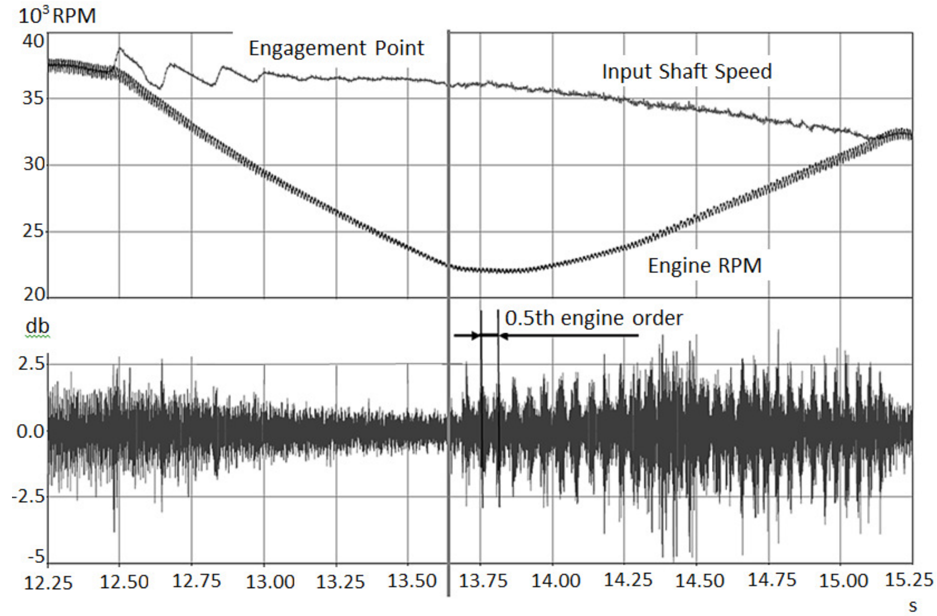

Figure 5.3 : Recorded noise data between 2000-3500RPM range. Rattle noise exists at the 0.5th engine order.

As mentioned before, references [124], [137], [138] reported that axial clutch vibrations are excited by the 4th cylinder of engine at the 0.5th engine order. Measurements tell us the rattle noise in our experimental setup occurred at the 0.5th engine order which is excited by the crankshaft bending vibrations which is verified by the references as shown in Figure 5.4.

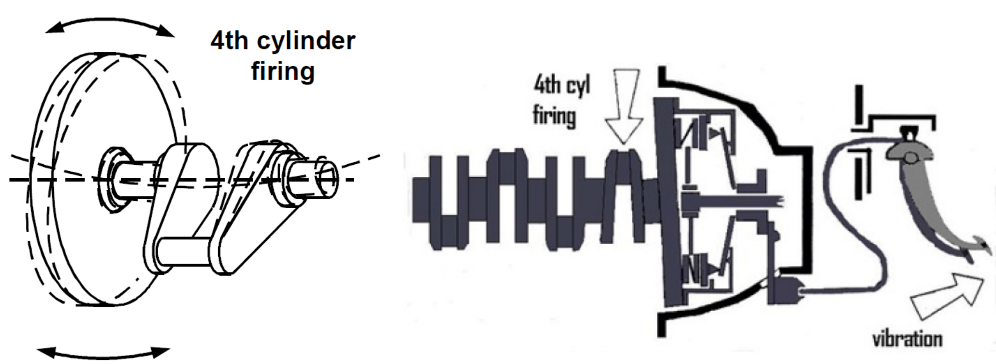

Figure 5.4 : Crankshaft bending excitation on clutch system resulting in axial clutch vibrations [124], [139].

Figure 5.5 shows the measurements taken from the engine crankshaft, flywheel, clutch cover and pressure plate. It is observed in this figure that the clutch system considered in this study works like an amplifier which results in transmission rattle noise during clutch engagement.

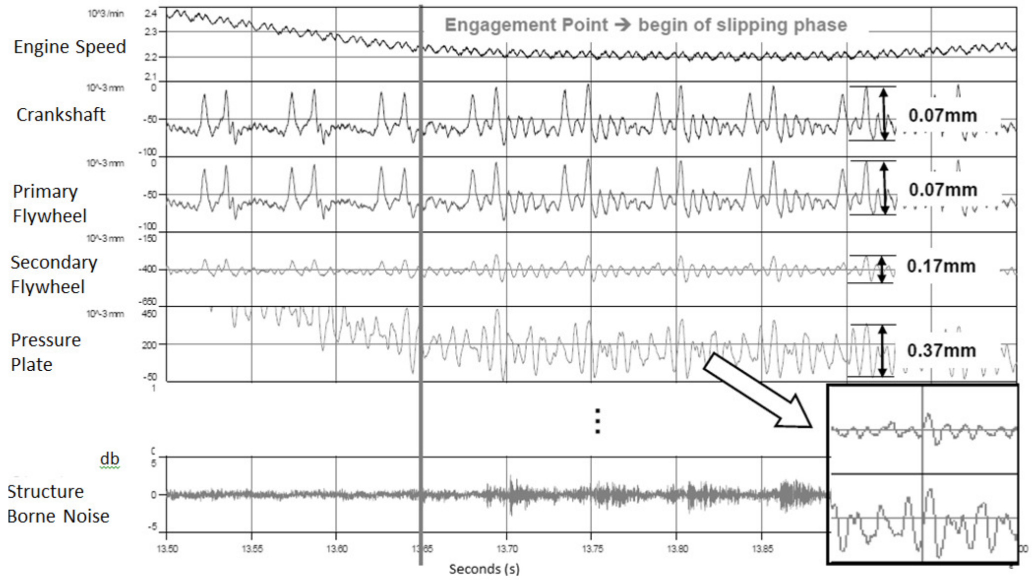

Figure 5.5 : Axial vibration measurements on the crankshaft, primary and secondary flywheels and pressure plate, and its relations with rattle noise measurements.

5.3 Analysis of Vibration Modes on the Clutch and Flywheel

In order to understand the axial vibration modes of clutch and dual mass flywheel system, a modal analysis test is performed on a shaker test bench and vibrations are measured in response to axial sinusoidal excitations having 5mm release bearing travel between 0-800Hz. The shaker, instrumented clutch and test rig configuration is shown in Figure 5.6. The parts shown in the test bench are as follows:

a) Shaker
b) Flywheel primary side
c) Flywheel secondary side
d) Pressure plate

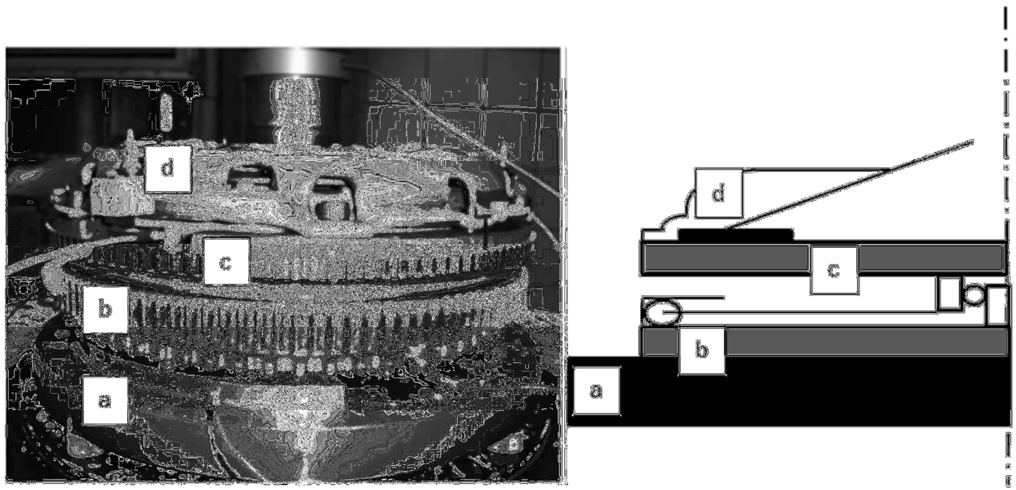

Figure 5.6 : Shaker test rig configiration and instrumented clutch.

Measurements obtained from this setup is presented in Figure 6.10 to verify the results of mathematical results where it is observed that the first natural frequency of pressure plate is around 165Hz with a phase difference between the flywheel and pressure plate. Measurements also show that there is a second dominant frequency around 150 Hz with no phase difference between the flywheel and pressure plate whilst pressure plate shows relatively larger amplitudes in compared with the whole frequency range.

6. MATHEMATICAL MODEL FOR PRESSURE PLATE VIBRATIONS

Note that we have flywheel vibration measurements at hand which are excited by the crankshaft bending vibrations (especially 4th cylinder firing is effective on this kind of vibration problem (Figure 5.4), e.g., see [124], [137]) and it is not possible to alter these vibrations unless the engine or cranking system is modified. Therefore, flywheel vibrations are used as an input to the vibration model of clutch system in this study.

On the other hand, in order to model the pressure plate axial vibrations, the system is modeled by the components such as the clutch disk, input shaft, pressure plate, diaphragm spring, clutch fingers and clutch cover. Engine vibration transmitted through this flywheel connection is the only source for clutch pressure plate vibration. Such a mathematical model for another clutch system can be found in [124] where a cable clutch system with an external slave cylinder and a lever system is modeled while the model in this study is a hydraulic system with a concentric slave cylinder.

The components of clutch system are shown in Figure 6.1. The mathematical model includes the geometrical and physical properties of these sub-components and stiffness properties of the springs. Since excessive vibrations of the pressure plate occur during the engagement (while releasing the clutch pedal), we will simulate the system starting from the disengaged position to the engaged position; that is, this model will be working at the time while the clutch pedal is being released.

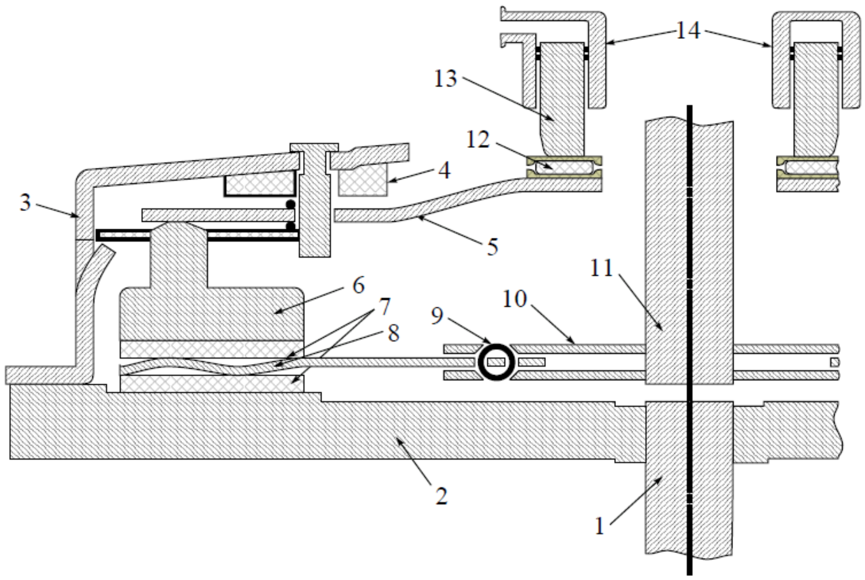

Figure 6.1 : Axial cut of the clutch structure: (1) crankshaft, (2) flywheel, (3) clutch cover, (4) wear-compensation system, (5) washer spring, (6) pressure plate, (7) friction pads, (8) flat spring, (9) spring damper, (10) clutch disc, (11) gearbox primary shaft, (12) needle roller bearing, (13) concentric slave cylinder piston, (14) concentric slave cylinder [89].

It must be underlined that this model is only built to study longitudinal vibrations of the clutch system and no rotational motion is considered. There are two main reasons for this assumption.

- If there are unexpected rotational vibrations in the engine, the noise phenomena would occur at idle or driving conditions as well. Instead, the noise is only heard during engagement phase in which pressure plate does not complete its full travel. Following, the excessive axial vibration causes pressure plate to press the clutch disc intermittently, then torque fluctuation is transmitted to the transmission that causes the rattle noise phenomena.
- Torque fluctuations can also be improved by modifying rotational dampers slightly. But this does not resolve the main source of vibration, only damps out the vibration transmitted to the transmission. This may also cause a negative effect on the dampers life time due to the vibrations on pressure plate.

The simplified model of the system is shown in Figure 6.2 in which diaphragm fingers are connected to the cover which can rotate around the fulcrum ring connection points. This is completely in line with the detailed representation of the clutch system shown in Figure 6.1; clutch disc is assumed to be consisting of two separate friction discs. The mass of these friction discs are equally added to the secondary side of flywheel and pressure plate. The cushion spring and its friction surface are shown in the model as a sub-component of the clutch disc.

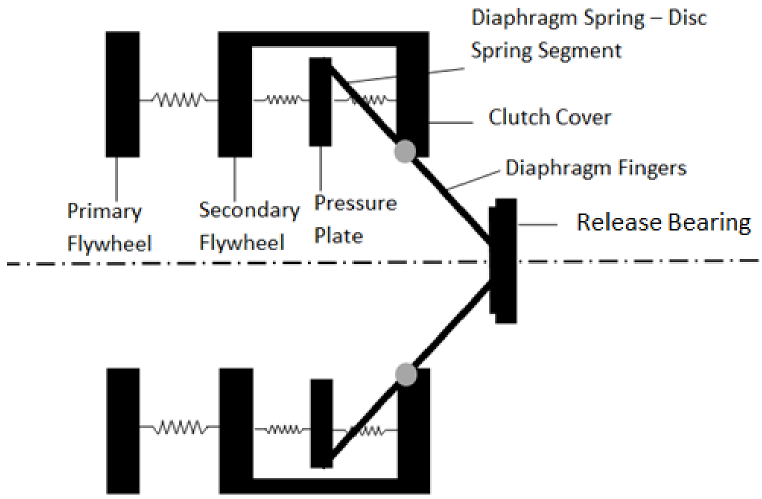

Figure 6.2 : Simplified representation of the clutch system by considering axial vibrations.

In Figure 6.3, experimental measurements taken from the engine is shown. As it can be seen from physical measurements, the crankshaft has two main peaks which are due to the 3rd and 4th cylinder firings which cause the crankshaft bending. These two peaks are repeated in every 720° of crankshaft rotation and used as inputs in the simulation model as shown in the same figure (below).

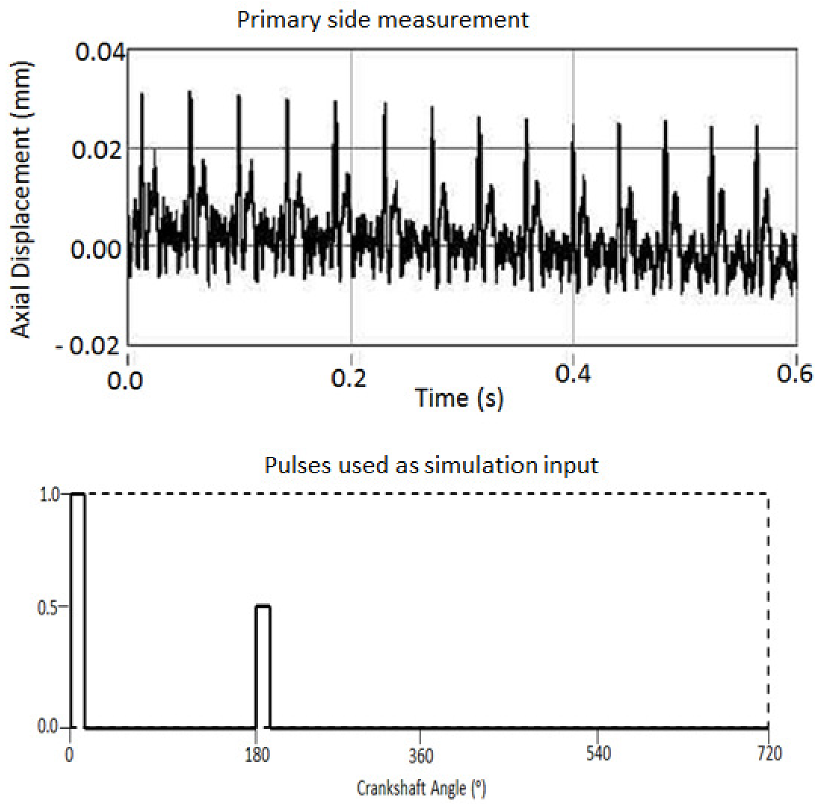

Figure 6.3 : Primary side axial vibration measurements and simulation input.

The mathematical model is obtained by using MATLAB software by using the simplified model shown in Figure 6.2. To this aim, the Simulink®, which is a MATLAB tool, is employed that helps simulate such vibration models. The high level Simulink model for the pressure plate and clutch cover is shown in Figure 6.4.

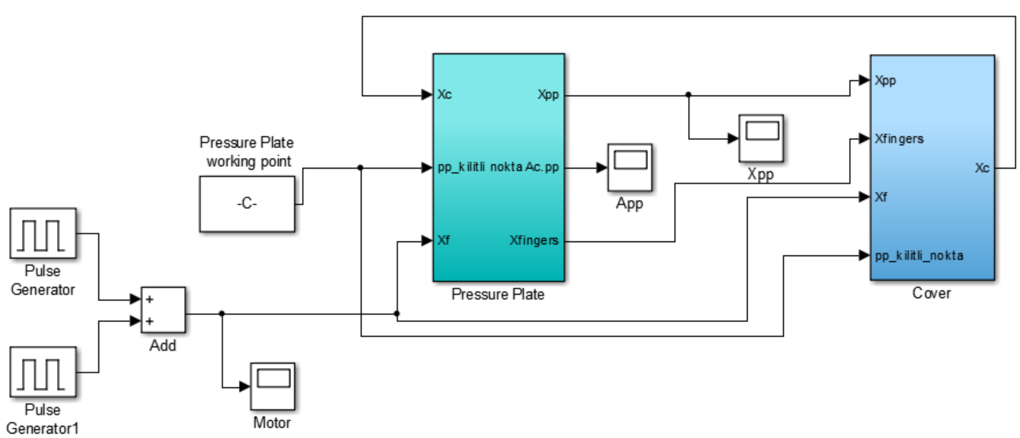

Figure 6.4 : High level Simulink model.

The model shown in Figure 6.4 has two pulse generators as explained in Figure 6.3, which simulate the engine excitation and is used as an input to the pressure plate and clutch cover. The main purpose is to find the pressure plate excitations. Following, simulation results will be compared with measurements.

Vibrations of the pressure plate are shown in Figure 6.5 where point A shows the engaged position of the pressure plate (operation point), point B represents the disengagement phase at pressure plate travel λ, and point C shows the cushion spring loading at pressure plate travel λ.

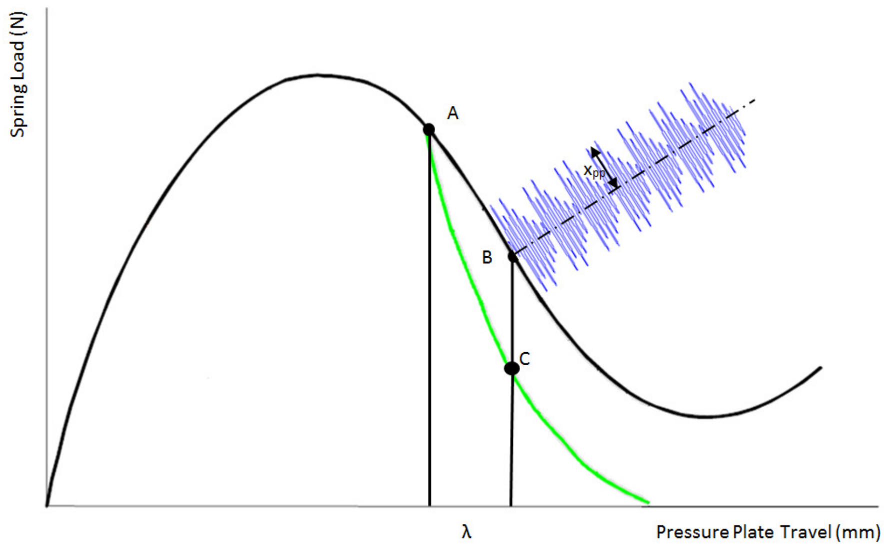

Figure 6.5 : Pressure plate vibrations during disengagement. Point A shows the engaged position of the pressure plate (operation point), point B represents the disengagement phase at pressure plate travel λ, and point C shows the cushion spring load at pressure plate travel λ.

In order to build sub-models, the mathematical equations should be obtained for vibrations. The following dynamic equations are of the system shown in Figure 6.2.

$$m_{pp}\ddot{x}_{pp} + F_{disc\_spring} - F_{cushion} - F_{fingers} = 0 \qquad \textbf{(6.1)}$$

$$m_c\ddot{x}_c - F_{cushion} + F_R + k_c \times (x_c - x_f) = 0 \qquad \textbf{(6.2)}$$

Where m_{pp} is the pressure plate mass, $\ddot{x}_{pp}$ is pressure plate acceleration, x_{pp} is the pressure plate travel around point B in Figure 6.5, λ represents pressure plate travel from the point where pressure plate vibrations are calculated, F_{disc_spring} is the load on the disc spring, $F_{cushion}$ is the load on the cushion spring, $F_{fingers}$ is the load on the pressure plate created by the release load, m_c is the cover mass, $\ddot{x}_c$ is the cover acceleration, x_c is the cover displacement, F_R is the release load on the diaphragm fingers, k_c is the cover stiffness given in Figure 4.17 and x_f is the secondary flywheel displacement. Now, we will write the terms in Eqs. (6.1) and (6.2) with known parameters as follows:

$$F_{disc\_spring} = F_{diaph}\left(\lambda + x_{pp} - x_c\right) \tag{6.3}$$

Where $F_{diaph}(.)$ is the clamp load at any x_{pp} (characteristic load curve for F_{diaph} is shown in Figure 4.9). Moreover,

$$F_{cushion} = F_{cush}\left(\lambda + x_{pp} - x_f\right) \tag{6.4}$$

Where $F_{cush}(.)$ is the cushion spring load at any x_{pp} which is given in Figure 4.10. In addition,

$$F_{fingers} = F_R \times i = F_{release}(x_{release} - x_c) \times i \tag{6.5}$$

Where $F_{release}(.)$ is the release load characteristic dependent on release travel given in Figure 4.18, $x_{release}$ is the release bearing travel at $\left(\lambda + x_{pp}\right)$ pressure plate travel and i is the clutch ratio. We can write this equation with x_{pp} by using lift off curve which is also given in Figure 4.18 as follows

$$x_{release} = k_{lift}\left(\lambda + x_{pp}\right) \tag{6.6}$$

That results in,

$$F_{fingers}\left(\lambda + x_{pp}\right) = F_{release}\left(k_{lift}\left(\lambda + x_{pp}\right) - x_c\right) \times i \tag{6.7}$$

Where $k_{lift}(.)$ is the lift off curve characteristic, which gives the pressure plate travel against release travel shown in Figure 4.18 on the right. We can also write release load F_R in Eq. (6.2) as follows;

$$F_R = F_{release}(x_{release} - x_c) = F_{release}\left(k_{lift}\left(\lambda + x_{pp}\right) - x_c\right) \tag{6.8}$$

Then, Eq. (6.1) and Eq. (6.2) become as follows;

$$\begin{aligned} m_{pp}\ddot{x}_{pp} + F_{diaph}\left(\lambda + x_{pp} - x_c\right) - F_{cush}\left(\lambda + x_{pp} - x_f\right) \\ - F_{release}\left(k_{lift}\left(\lambda + x_{pp}\right) - x_c\right) \times i = 0 \end{aligned} \tag{6.9}$$

$$m_c\ddot{x}_c - F_{cush}\left(\lambda + x_{pp} - x_f\right) + F_{release}\left(k_{lift}\left(\lambda + x_{pp}\right) - x_c\right) + k_c \times \left(x_c - x_f\right) = 0 \qquad \textbf{(6.10)}$$

After deriving the differential equations, we are ready to build the corresponding Simulink models. Since we have a number of non-linear springs and tables, it will be easier to solve the equations in time domain by using the Simulink blocks. Once these equations are solved in time domain, it is easier to examine the solutions in frequency domain.

The mathematical model has the input of 2500 RPM which is in the range such that the rattle noise is in the most noticeable range (i.e., Figure 5.3). The Simulink sub-models for the pressure plate and clutch cover are shown in Figure 6.6 and Figure 7,7 respectively. These Simulink models help us solve the nonlinear differential equations given by Eq.(6.9) and Eq.(6.10). Simulation results are presented and compared with measurements in Figure 6.8.

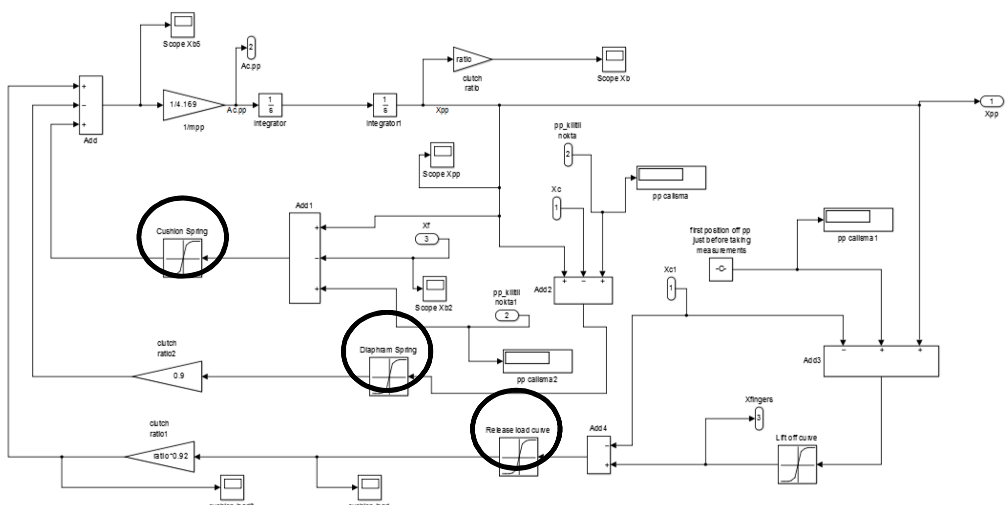

Figure 6.6 : Simulink model for the pressure plate.

Simulink model of the pressure plate is built by using the dynamic equation of the pressure plate. The model accepts flywheel vibrations (Figure 6.3) and pressure plate working point (Figure 4.9) as the inputs and then calculates the pressure plate vibrations x_{pp} and diaphragm finger vibrations. The model calculates every parameter in Eq. (6.9) by using cushion spring, diaphragm spring and release load characteristics which were explained in Section 4. These are highlighted with circles in Figure 6.6.

Following Simulink model of the clutch cover is constructed by using the dynamic equation of the cover. The model accepts pressure plate and diaphragm finger vibrations from the previous Simulink model, pressure plate operation point and flywheel vibrations as the inputs and then calculates the cover vibrations. The model calculates every parameter in Eq. (6.10) by using cushion spring and release load characteristics which were explained in Section 49. These are highlighted with circles in Figure 6.7.

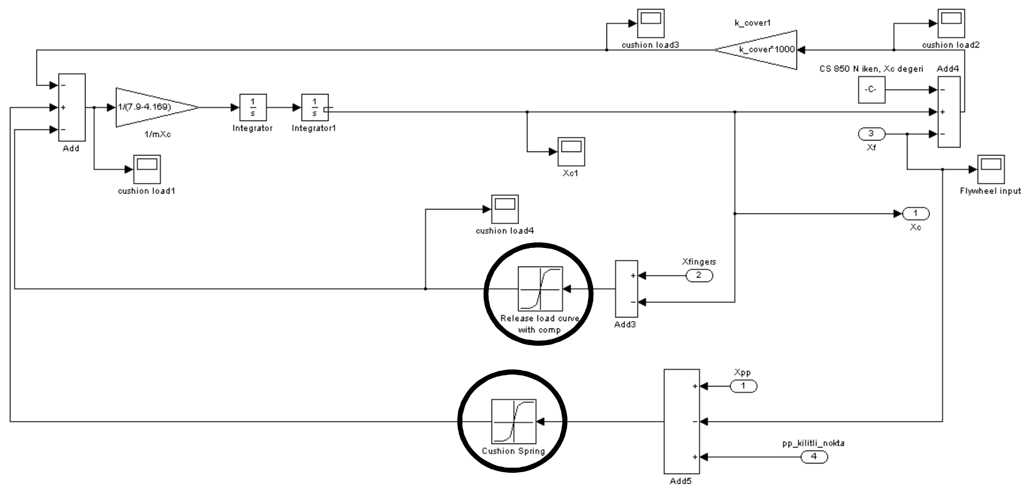

Figure 6.7 : Simulink model for the clutch cover.

Simulation results and experimental measurements are presented in Figure 6.8 where it is observed that the simulation results show a good correlation with experimental measurements that are obtained by using the test bench explained in Section 2.2. For instance, while the vibration amplitude in simulations is approximately 0.35 mm, it is 0.37 mm in experimental measurements at 2500 RPM. As mentioned before, measured flywheel vibrations are used as the excitation data in the simulation. All measurements are sampled during the engagement phase where the rattle noise occurs.

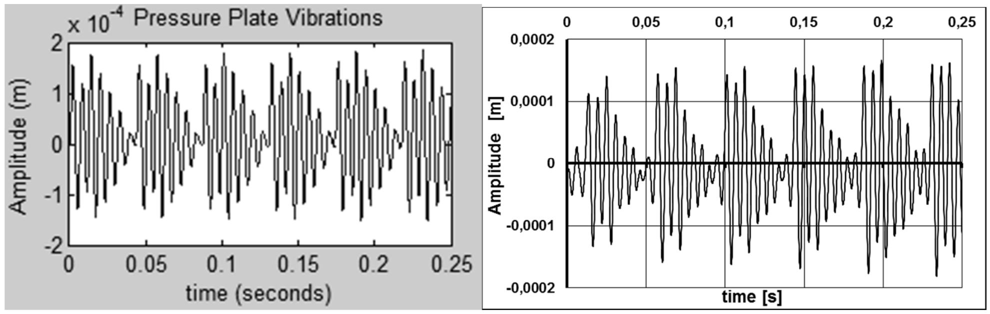

Figure 6.8 : Comparison between simulation results (on the left) and measured vibrations (on the right).

By examining Figure 6.8 in detail, we can observe that basically two different frequency contents exist in the vibrations. It is obvious that the signal content having relatively lower frequency is caused by the crankshaft axial vibrations which are exactly equal to the 0.5^{th} order of the engine speed. As a matter of fact, this is expected because this frequency content is one of the most critical frequencies of four cylinder engines.

If we study higher frequency vibrations in Figure 6.8, we observe that there is a second frequency content in the vibrations having approximately the frequency spectrum of 160Hz as shown in Figure 6.9 for both measured and simulated data. In sum, both measured and simulated data show a dominant frequency around 160Hz and similar sub-harmonics. The corresponding FFT analyses of measured data and simulation results show that there is another peak around 145Hz as well.

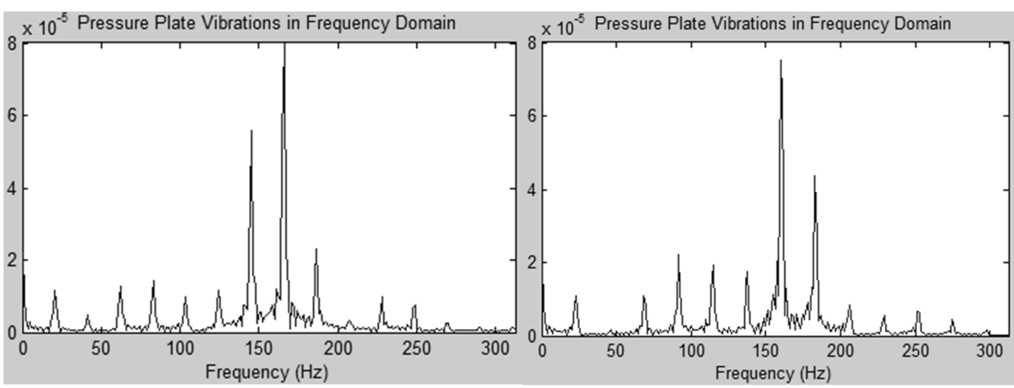

Figure 6.9 : Power spectrum of the pressure plate vibration measurements is on the left and that of simulation results is on the right.

Following, the shaker test is performed to find the natural frequencies of clutch system and reveal the other dominant frequencies as mentioned in Section 5.3. The frequency content of measurements during this test is given in Figure 6.10 in which it is observed that the first natural frequency of pressure plate is around 165Hz which also confirms our simulation results. These measurements also show that there is a second dominant frequency around 150 Hz which is also close to the observations in simulation results.

As mentioned before and shown in Figure 5.3, the worst range for the rattle noise problem is around 2500 RPM where the 4^{th} engine order around this RPM is 167Hz which is very close to modal analysis and simulation results of axial pressure plate vibrations. Esfahani *et al* also reported the same results that the $0{,}5^{th}$ engine order excitation caused the clutch pressure plate vibration around 4^{th} engine order frequency [124].

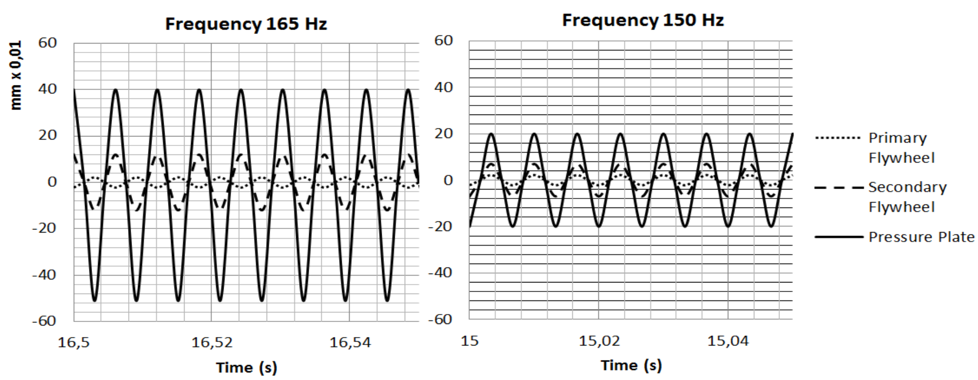

Figure 6.10 : Shaker test results shows that the natural frequency for clutch axial vibration is around 165 Hz and there is also a second axial dominant frequency at 150Hz.

To sum up, these measurements are very consistent with the simulation results. R. Esfahani et al. found that the clutch pedal vibration was around 250Hz [124]. They did not investigate the reason of this vibration. The reason was most likely that natural frequency of their clutch was around this frequency value. Since the weight of the clutch they studied was 1.9kg lighter than the one we studied, it is normal that they found the natural frequency higher than what we observed.

Hasebe et al. proposed a modified clutch assembly to change the natural frequency range of the axial clutch vibrations away from the frequency range of flywheel vibrations [139]. But, they did not clearly explain which parameters were modified. In the upcoming sections, we will also identify the most critical parameters of the clutch system and optimize the vibrations of clutch set by using multi-objective genetic optimization method while keeping the clutch release loads as low as possible.

7. MULTIOBJECTIVE PARETO OPTIMAL DESIGN OF THE CLUTCH SYSTEM

In this section, a multi-objective Pareto optimal solution of the clutch system is sought by using the pedal characteristic and vibrations of pressure plate as objective functions. The first objective function is the pedal characteristic curve which is affected by a number of variables such as the geometry of diaphragm spring, material properties, release bearing stroke and dimensions of the pedal mechanism. All these variables are chosen as design variables in the related optimization problem.

The other objective function is to reduce the axial vibrations of clutch pressure plate and keep the clutch pedal load as low as possible. If design parameters are selected without considering the axial vibrations of pressure plate, excessive vibrations may emerge and cause interruptions in torque transmission during the engagement and disengagement of clutch disc. Such an interruption in torque transmission is the main cause of the well-known transmission rattle noise problem.

Note that the pressure plate mass may also be considered as an additional design variable which affects both the amplitudes of pressure plate vibrations and heat energy absorbed by clutch friction surfaces. The pressure plate mass should be as big as possible to increase the absorbed heat energy by the clutch for a longer clutch life. Thus, the heat energy absorbed by clutch friction surfaces may be considered in the Pareto optimization problem. However, it was observed in optimization runs that consideration of heat energy absorbed by clutch friction surfaces as the third objective function did not have significant effect on Pareto optimum solutions. Therefore, the heat energy absorption is calculated separately after the optimization runs are completed. Beforehand, we will investigate design variables and objective functions.

7.1 Design Variables

There are number of variables affecting the amplitudes of pressure plate vibrations and pedal characteristic. In order to understand the effectiveness of design variables on pressure plate amplitudes, sensitivity analysis is performed and the results are listed in Table 7.1 where all possible design variables and their effects on simulation results are presented.

Table 7.1 : Sensitivities of variables on simulation results of pressure plate vibrations.

Variable	Increase Decrease	Effect	Magnitude of Effect
Ratio	↑ ↓	No significant effect	n/a
Pressure Plate Mass	↑	Improvement	Medium
Pressure Plate Mass	↓	Degradation	Medium
Leaf Spring Stiffness	↑	Improvement	Small
Leaf Spring Stiffness	↓	Degradation	Small
Clamp Load	↑	Degradation	Small
Cushion Deflection Stiffness	↓	Improvement	High
Cover Stiffness	↑	Improvement	Small
Cover Stiffness	↓	Degradation	Small
Diaphragm Spring Stiffness (h0)	↑	Degradation	Medium
Diaphragm Spring Stiffness (h0)	↓	Improvement	Medium

It is noteworthy that all modifications on the clutch design will bring tooling and test costs to the relevant subcomponents. Therefore, it does not make sense to modify the variables which does not create a significant effect on the vibrations. Considering the effects of design variables on the costs, the best three options are the cushion deflection stiffness, mass of pressure plate and diaphragm spring stiffness. The diaphragm spring stiffness depends on the inner and outer diameters, thickness and its conical height. However, in comparison with other design variables, modifying only the conical height is the optimum way to modify the diaphragm spring stiffness due to the ease of tool modification and relatively lower cost. The rest of design variables would result in relatively higher costs, require major modifications on the tools (or may even need a new tool) and very big lead-time.

7.2 Objective Functions and Constraints

When designing a clutch system, the first issue to be considered is to transfer the maximum engine torque to transmission system. To this end, effective clamp load should be selected carefully. Since the clamp load that is less than the required value may cause excessive slippage on the clutch system and degrades the clutch life significantly; on the other hand, the clamp load which is more than the required value may cause very high clutch pedal efforts and bad clutch feeling during the launch (e.g., very quick engagement and possible undesired stalls).

Subsequently, the first objective function should be the minimization of maximum pressure plate vibration amplitude which is the main source for rattle noise. Since the maximum vibration amplitudes are observed between the frequency range of 100 to 200 Hz (e.g., see Figure 6.9 and Figure 6.10), we will focus on this frequency interval for the first objective function. This frequency range also covers the 4th engine orders of the engine speed between 1500-3000 RPM range which is the critical engine order as discussed in Section 3. Moreover, as another constraint in optimization runs, the clamp load is chosen as 10,000N at the beginning of the design process which is given in the section on modeling of disc spring (i.e., Section 4.3.1).

The other basic function of the clutch system is to cut the torque transfer from the engine to transmission system when requested (to prevent stall or change transmission gear). While fulfilling this function, the maximum clutch pedal effort should be as low as possible. High pedal efforts would cause unsatisfactory drivers. Thus, the second objective function can be defined as the minimization of maximum clutch pedal effort which is directly connected with maximum release load. Considering these objective functions and constraints, the multi-objective optimization problem can be built as shown in Eqs. (6.1) and (6.2). That is,

$$minimize\ \ x_{pp},\ minimize\ F_R subject\ to \begin{cases} F_{clamp} >= 10000 \\ 100 < frequency < 200 \end{cases} \tag{7.1}$$

$$m_{pp}\ddot{x}_{pp} + F_{diaph}\left(\lambda + x_{pp} - x_c\right) - F_{cush}\left(\lambda + x_{pp} - x_f\right) - F_{release}\left(k_{lift}\left(\lambda + x_{pp}\right) - x_c\right) \times i = 0$$

$$m_c\ddot{x}_c - F_{cush}\left(\lambda + x_{pp} - x_f\right) + F_{release}\left(k_{lift}\left(\lambda + x_{pp}\right) - x_c\right) + k_c \times \left(x_c - x_f\right) = 0$$

$$F_R = \frac{F_D - F_{cushion} - F_{sensor} - F_{leaf}}{i} + F_{comp.} \quad \textbf{(7.2)}$$

One of the functions of clutch system is to transfer the heat energy generated by the friction on disc surfaces. To this end, the clutch heat energy generation calculations are completed after the optimization runs are completed by using the following [136].

$$Q = \frac{\mu F_{clamp} V_x t_x}{2} \quad \textbf{(7.3)}$$

Where Q is the dissipated heat energy, μ is the friction coefficient of clutch faces, F_{clamp} is the clamp force on the pressure plate, V_x is the vehicle speed and t_x is the time.

The ratio of $\frac{Q}{A}$ for the cross sectional area A is widely used to "normalize" heat generation as a sizing parameter. This normalized value is used to evaluate clutch heat effects. In addition, $\frac{Q}{m_{pp}}$ for the unit pressure plate mass m_{pp} is also used when considering "heat sink" capability of clutch pressure plate [136].

Since the clutch inner or outer diameters are not chosen as optimization parameters, $\frac{Q}{A}$ will not be changed. However, we need to check $\frac{Q}{m_{pp}}$ since pressure plate mass m_{pp} is an optimization variable and may be changed after the optimization runs. If the pressure plate mass m_{pp} is reduced, heat sink capability of the pressure plate will be reduced (more heat energy will need to be sank into unit mass).

7.3 Optimization of The Clutch System As a Dynamic System

Since we have more than one objective function, multi-objective genetic optimization algorithm in Matlab is used in the optimization runs. The genetic algorithm is a method for solving both constrained and unconstrained optimization problems based on natural selection that drives biological evolution. The genetic algorithm repeatedly modifies a population of individual solutions. At each step, it selects individuals randomly from the current population to be parents and uses them to produce the children for the next generation. At each iteration, the genetic algorithm performs a series of computations on the current population to produce a new population. Each successive population is called a new generation. An individual is any point to which you can apply the fitness functions whose value for an individual is its score. A population is an array of individuals that consist of optimization parameters. The fitness function is the name of the optimization functions for genetic algorithm [144]. Typically, the algorithm is more likely to select parents that have better fitness values.

The genetic algorithm begins by creating a random initial population by using optimization variables. The algorithm then creates a sequence of new populations. To create the new population, the algorithm performs the following steps:

1. Scores each member of the current population by computing its fitness value.
2. Scales the raw fitness scores to convert them into a more usable range of values.
3. Selects members, called parents, based on their fitness.
4. Some of the individuals in the current population that have lower fitness are chosen as elite. These elite individuals are passed to the next population.
5. Produces children from the parents. Children are produced either by making random changes to a single parent (mutation) or by combining the vector entries of a pair of parents (crossover).
6. Replaces the current population with the children to form the next generation.

At each step, the genetic algorithm uses the current population to create the children that make up the next generation. The algorithm selects a group of ndividuals in the current population, called *parents*, who contribute their *genes* (the entries of their vectors) to their children. The genetic algorithm creates three types of children for the next generation:

- **Elite children** are the individuals in the current generation with the best fitness values. These individuals automatically survive to the next generation.
- **Crossover children** are created by combining the vectors of a pair of parents.
- **Mutation children** are created by introducing random changes, or mutations, to a single parent. [144].

Figure 7.1 illustrates the three types of children.

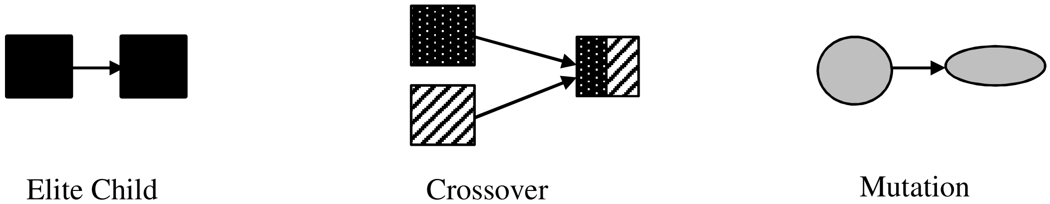

Figure 7.1 : Types of children [144].

As an outcome of the above described optimization problem, a Pareto chart is created. The curve in this chart is built with optimum points for both objective functions. It means that we do not have only one optimum solution, instead we have a list of optimum solutions and we need to select one of these solutions which can serve to our aim as the best result.

It must be noted at this point that all relevant optimization parameters are modified at each optimization iterations if the geometry of clutch, mass or stiffness are varied during optimization process. Thus, each iteration is actually an independent simulation which uses new design parameters and re-calculated curves.

Figure 7.2 shows the Pareto chart of the clutch system and optimum solutions of clutch system. The four solutions having relatively lower values for the objective functions 1 and 2 are encircled in Figure 7.2, which can be selected as the optimum solution by the designer. Note that the total solution time is 840 seconds on a computer having AMD Athlon 2.8 GHz dual core processor.

As it can be seen in the Pareto chart, global optimum points are calculated as;

- Cushion deflection under max clamp load = 0.75+0.188=0.938mm
- Pressure plate mass = 4.0387 kg
- Diaphragm spring conical height = 8.971mm

However, these values cannot be handled due to the tolerances of clutch manufacturing process. Reasonable values which can be manufactured by manufacturing processes can be found by rounding above design parameters as follows

- Cushion deflection under max clamp load = 0.75+0.2=0.95mm
- Diaphragm spring conical height = 9mm

Table 7.2 shows the comparison between the original parameters of the first physical clutch, optimized parameters and results of the objective functions.

Table 7.2 : Comparison of initial and optimized clutch parameters and their objective functions.

Clutch Type	Cushion Disp (mm)	Pressure Plate Mass	Diaphragm Spring Height (mm)	Max Release Load (N)	Pressure P. Amplitudes (mm)
Initial	0,75	3,9	9,2	2250	0,185
Optimized	0,95	4,04	9	2170	0,127

The simulation results which are shown in Figure 7.3 are calculated by using the variables given in Table 7.2. If the results are compared with the initial system (e.g., Figure 6.8 and Figure 6.9), it can be observed that the vibration amplitudes of pressure plate are reduced by 0.06mm that is significant. This makes 0,12mm improvement in total pressure plate displacement which makes approximately 35% improvement on vibration amplitudes. The FFT results also show the same improvement especially around the frequency range of 140-160Hz.

Following, the physical model is produced by using the optimum design variables given in Table 7.2 and then tested. Experimental measurements obtained from the optimized clutch system are shown in Figure 7.4 in which it is observed that the experimental results give close results to simulation results. Both simulated and measured data have the same dominant frequency around 160Hz and their FFT results are very close to each other. The maximum amplitudes are also very close to each other (around 0,125mm-0,130mm).

Moreover, it is observed in the experiments that the rattle noise is reduced approximately 40% based on microphone measurements that can be observed if Figure 7.5 is compared with Figure 5.3. The measurements are performed as explained in Section 5.2. In brief, the simulation results and experimental measurements are encouraging and axial vibrations in the drive line are reduced by optimization process.

Finally, if we check the heat sink capability of optimized mass and compare with that of the initial mass (i.e., 3.9kg), we can see that the heat sink capability is improved by 3.56% since the mass is increased; that is, $\left\{\frac{Q}{3,9}\Big/\frac{Q}{4,0387}\right\} \rightarrow 3,56\%$.

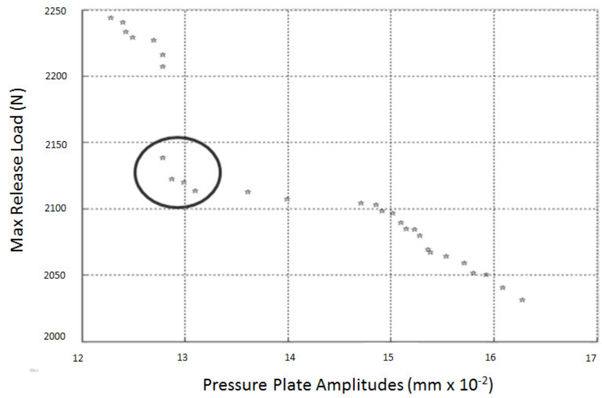

Figure 7.2 : Pareto chart of the optimization problem, where the objective #1 shows the pressure plate amplitude and the objective #2 shows the maximum pedal load.

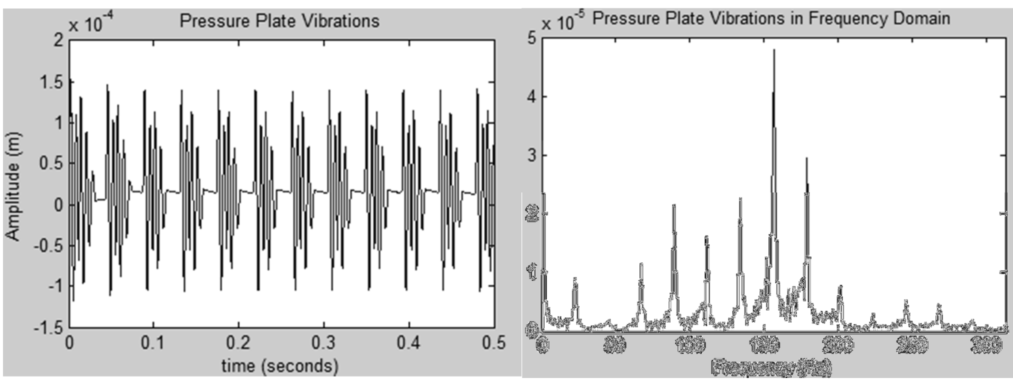

Figure 7.3 : Simulation results of pressure plate vibrations for the optimized clutch (in time domain on the left and in frequency domain on the right).

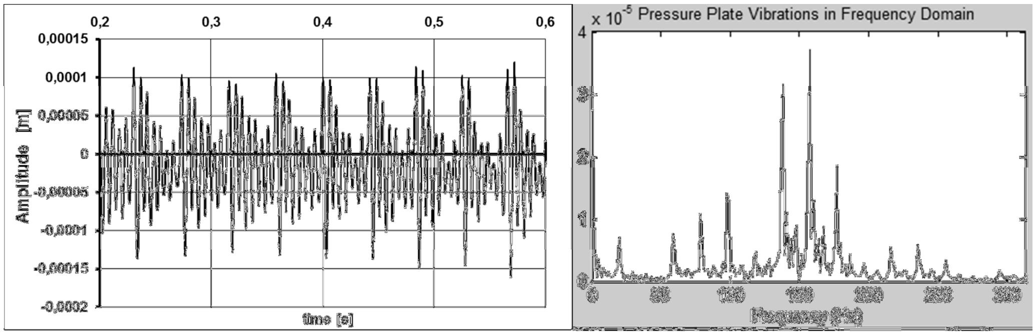

Figure 7.4 : Experimental measurements for pressure plate vibrations measured on the physical model produced by using optimized parameters (in time domain on the left and in frequency domain on the right).

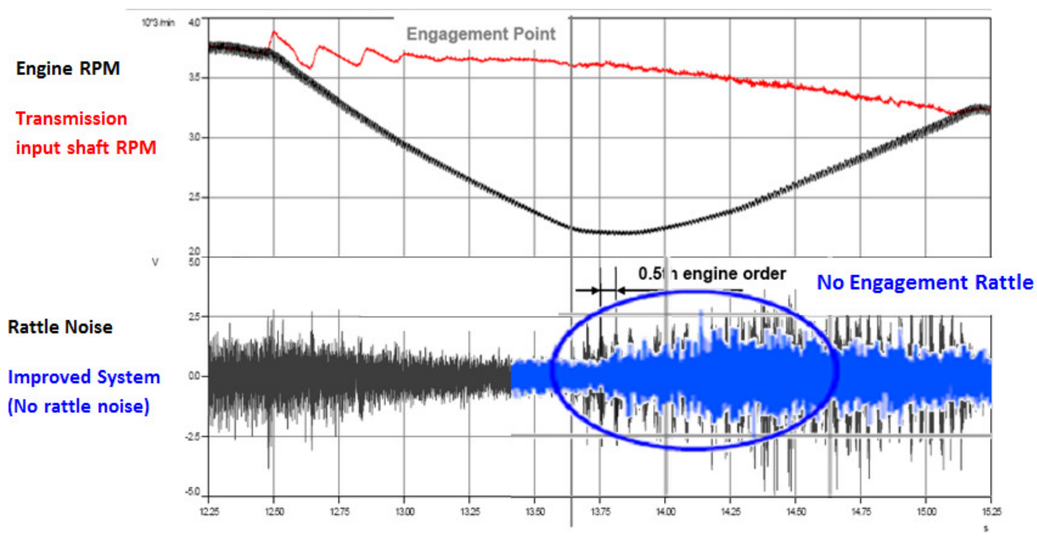

Figure 7.5 : Comparison of transmission rattle noise measurements between the original and optimized clutch systems.

8. CONCLUSIONS AND RECOMMENDATIONS

Axial vibrations in clutch systems are generally ignored in studies on dynamic behavior of clutch systems in literature, since the clutches are used to damp torsional vibrations instead of the axial ones. However, axial vibrations has to be examined for the driveline noise calculations, in particular in the existence of transmission rattle noise. On the other hand, axial vibrations and pedal efforts are actually highly related to each other, since they are both derived by the same variables such as stiffness values of springs in the clutch and clutch ratio. To this end, mathematical models for the clutch pedal effort and clutch pressure plate vibrations are derived and built by using the programs MATLAB and MATLAB-Simulink toolbox. Then, these models are verified by making comparisons with experimental measurements. By combining the simulation models together, a multi-objective optimization problem is formulated and solved such that axial vibrations and clutch pedal effort are minimized under some constraints. At the end of the study, a physical model is also produced by using the optimization results and physically tested.

It is observed that experimental measurements and simulation results are quite close to each other and axial vibrations in the drive line is reduced by optimization process. Under the light of findings of this study, it is possible to calculate and design the components of a clutch system in an optimum way, which will help decrease the test and prototype costs significantly. On the other hand, optimizing a clutch system considering axial vibrations only will not degrade the rotational vibrations and, in order to reduce the different level of rotational vibrations from engine to transmissions (e.g., idle vibrations, creep vibrations or drive vibrations, etc…), different types of rotational dampers are used on clutch discs. However, in our study we tried to reduce the axial vibrations of disengaged clutch, thus effects of these axial vibrations during half engaged clutch position (which can be called slipping phase or engagement phase). These axial vibrations have no influence on the system when the clutch is fully engaged since very high clamp load is already exerted on the pressure plate that prevents the axial vibrations. However, a further study may also be planned to optimize the rotational dampers so that they can be usable during engagement phase. Thus, the clutch can be optimized to minimize the axial vibrations and rotational dampers can damp the residual vibrations which cannot be fully damped by axial vibration optimization during engagement phase.

8.1 Practical Application of This Study

Clutch pedal efforts are always important specifications for a clutch systems. Noise and vibration phenomena casued by engine axial vbrations is associated with clutch pedal efforts with this study. Simulation algorithms for clutch pedal efforts and clutch axial vibrations are built and optimization algorithm for an optimum system is offered. It is also proved that, it is possible to calculate design parameters for an optimum clutch system is possible with simulation tools and proposed optimization algorithm. It is aimed that to decrease prototype costs and prototype phase problems by using the simulation tools and optimization method offered by this study.

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List of Publications and Patents:

- Kirca, M. & **Ozansoy, O.** (2006): Finite Element Modeling of Microcellular Carbon Foam. *5th International Conference on Advanced Engineering Design* Prague, Czech Republic : June 11-14

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