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Fatigue and design optimization of single plate clutch

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ABSTRACT

This project concerning for the design for Fatigue life of single plate clutch as well as optimization of same to fulfil the criteria of weight lightness. Clutch will undergo in dynamic pressure loads as well as dynamic thermal loads. To simulate the dynamic physics we need to consider the variable pressure loads and temperature loads. Dynamic structural analysis would be solved using MBD solver. Then after the result of structural utilize to obtain the fatigue life of clutch under cyclic loading condition. For the topology optimization of clutch, same structural results will be utilized to generate the lightweight clutch with most optimum performance. Optimized clutch further utilize to check the fatigue life and result of both actual and optimized design will be compared.

I. INTRODUCTION

All During the course of over 100 years of automotive history, almost all components have undergone enormous technological development Reliability, production costs and ease of maintenance, as well as environmental compatibility have been and continue to be the criteria demanding new and better solutions from automotive engineers. Clutch is a mechanism which enables the rotary motion of one shaft to be transmitted, when desired, to a second shaft the axis of which is coincident with that of the first. The clutch is a mechanical device, which is used to connect or disconnect the source of power from the remaining part of power transmission system at the will of the operator. The main primary function of the clutch is to transmit the torque from engine to drive shaft & engage and disengage the transmission system. The secondary function is related to vibration & damping. When the friction clutch begins to engage, slipping occurs between the contact surfaces such as pressure plate, friction plate and flywheel and due to this slipping, heat energy will be generated on friction plate surfaces. A popularly known application of clutch is in automotive vehicles where it is used to connect the engine and the gear box. Clutches are also used extensively in production machinery of all types. In friction clutches, the connection of the engine shaft to the gear box shaft is affected by friction between two or more rotating concentric surfaces. The surfaces can be pressed firmly against one another when engaged and the clutch tends to rotate as a single unit.

Clutch closed

In the engaged state, the force of the diaphragm spring acts on the pressure plate. This pushes the axially movable clutch disc against the flywheel. A friction lock-up connection is created. This allows the engine torque to be directed via the flywheel and the pressure plate to the transmission input shaft.

Clutch open

When the clutch pedal is pressed, the release bearing is moved against the diaphragm spring load in the direction of the engine. At the same time, the diaphragm springs are deflected over the support rings, and the force on the pressure plate is reduced. This force is now so low that the tangential leaf springs are able to move the pressure plate against the diaphragm spring load. This creates play between the friction surfaces, allowing the clutch disc to move freely between the flywheel and the pressure plate. As a result, the power flow between the engine and transmission is interrupted.

In single plate clutch, a friction plate is held between flywheel and pressure plate. There are springs depending upon design arranged circumferentially, which provide axial force to keep the clutch in engaged position. The friction plate is mounted on a hub which is splined from inside and thus free to slide over the gear box shaft. Friction facing is attached to the friction plate on both sides to provide two annular friction surfaces for the transmission of power. A pedal is provided to pull the pressure plate against the spring force whenever it is required to be disengaged. Ordinarily it remains in engaged position. When the clutch pedal is pressed, the pressure plate is moved to the right against the force of the springs. This is achieved by means of a suitable linkage and

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a thrust bearing. With this movement of the pressure plate, the friction plate is released and clutch is disengaged.

In actual practice the construction of clutch is differs. The pressure plate, the springs, the release lever and the cover forms the sub-assembly, called cover assembly which can be mounted directly to the engine block, placing the clutch plate in between the flywheel and pressure plate with the clutch shaft inserted in. Rajesh Purohit et al (2014) presented the design and finite element analysis of an automotive clutch assembly. Theassembly consists of a clutch plate, a pressure plate and a diaphragm spring. The material selected for the three parts are structural steel, cast iron GS-70-02 and spring steel. Structural static structural analysis of each part was done. The plots for equivalent stress, total deformation and factor of safety were obtained and analysed. The finite element analysis was carried out in pre-processing, solving and post processing. Uniform wear theory was used for analysis. The Solid Works Office Premium software and ANSYS software has been used for designing and analysis purpose. The results show that designed friction clutch assembly is safe.

A Clutch is a machine member used to connect the driving shaft to a driven shaft, so that the driven shaft may be started or stopped at will, without stopping the driving shaft. A clutch thus provides an interruptible connection between two rotating shafts. Clutches allow a high inertia load to be stated with a small power. Clutches are used whenever the ability to limit the transmission of power or motion needs to be controlled either in amount or over time (e.g. electric screwdrivers limit how much torque is transmitted through use of a clutch; clutches control whether automobiles transmit engine power to the wheels). In the simplest application clutches are employed in devices which have two rotating shafts. In these devices one shaft is typically attached to a motor or other power unit (the driving member) while the other shaft (the driven member) provides output power for work to be done. In a drill for instance, one shaft is driven by a motor and the other drives a drill chuck. The clutch connects the two shafts so that they may be locked together and spin at the same speed (engaged), locked together but spinning at different speeds (slipping), or unlocked and spinning at different speeds (disengaged). A popularly known application of clutch is in automotive vehicles where it is used to connect the engine and the gear box. Here the clutch enables to crank and start the engine disengaging the transmission Disengage the transmission and change the gear to alter the torque on the wheels.

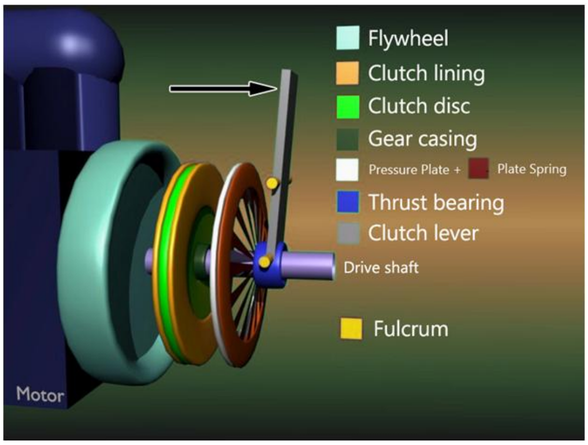

Figure I-1Friction clutch plate assembly

Clutches are also used extensively in production machinery of all types.When your foot is off the pedal, the springs push the pressure plate against the clutch disc, which in turn presses against the flywheel. This locks the engine to the transmission input shaft, causing them to spin at the same speed. Clutch for a drive shaft: The clutch disc (center) spins with the flywheel (left). To disengage, the lever is pulled (black arrow), causing a white pressure plate (right) to disengage the green clutch disc from turning the drive shaft, which turns within the thrust-bearing ring of the lever. Never will all 3 rings connect, with no gaps.

The field of engineering design (ED) is large and applicable to almost all aspects of engineering practice. Engineers use engineering design as an important tool to develop, research, troubleshoot, investigate, analyse and maintain products and services that meet the user-prescribed objectives. ED commonly involves an iterative process where the designer's decision-making skills and disciplinary knowledge are coherently applied. Moreover, by properly following the engineering design process, one can synthesize, build, test and evaluate products to achieve results.

The design process consists of several steps leading to the finalization of the design, followed by fabrication, as illustrated in Fig. 1.2. Dym and Little (2000) gave an adequate overview of the design process with elements from Dieter (2000), which entailed problem definition, concept generation, concept evaluation, detailed design and design communication. The phases of conceptual design were given due attention by Pahl et al. (2007) within their formulation known as systematic design; this proved to be an efficient methodology for concept generation and evaluation. The different stages involved in

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conceptual design are shown in Fig. 1.3, as a flowchart that helps one arrive at a suitable concept.

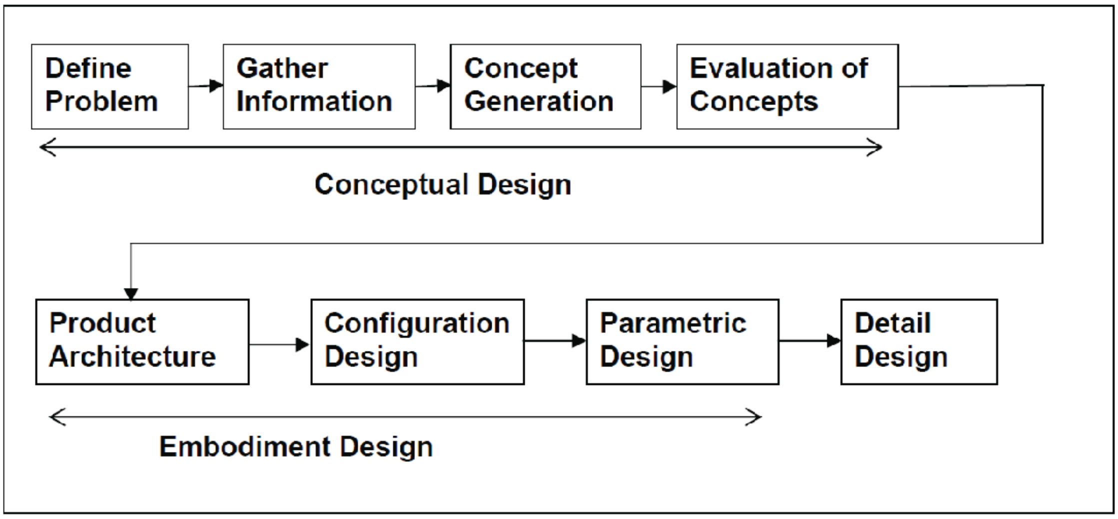

Figure I-2The design process overview by Dieter (2000)

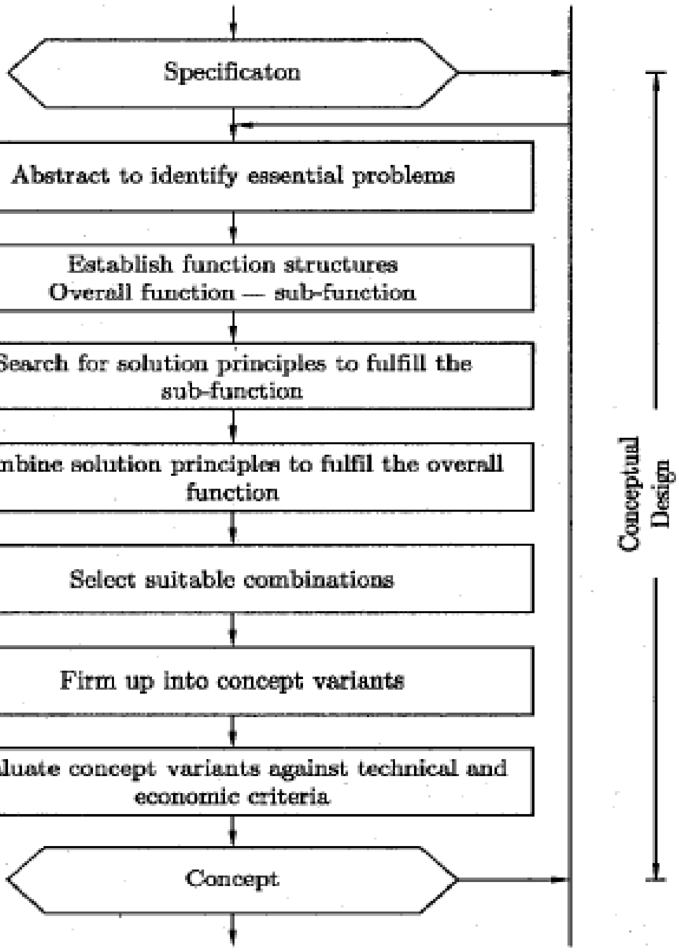

Figure I-3A owchart of steps involving conceptual design (Pahl et al., 2007)

In engineering design, individuals have divergent opinions on: how to design, what distinguishes a good design from a bad design, whether design is more art than science, and so forth. So much has been written about engineering design that no attempt is made here to cover such a wide range of opinions on the subject. It seems the harder the design problem, the less likely that two individuals will agree on its solution. This probably is an asset, as it leads to a diversity of solutions, more thorough deliberation, and usually better designs. Although a designer's pride may get in the way, peer-review is an important element to a successful design. Self-review of the design should occur all along the process. All aspects of a design should have a reason for being while one should be able to defend the logic or agree when it is arbitrary. Artifiicial constraints in the creative thinking process tend to obscure the simple and elegant design solutions that most strive to achieve. Peer- and self-reviews tend to uncover the constraints that complicate the design or lead to unnecessary aspects.

How does one go about solving a new design problem? It all depends on the problem, one's knowledge, and available resources, besides the physical, legal and financial constraints. Sometimes the design solution is found as a result of an analysis task, for instance, optimizing the profile of a cam or synthesizing the geometry of an aircraft wing. Experience and practice lead to a variety of problems and solutions, as one becomes adept at solving new design problems. Creativity is an exceptional ability of the human mind, which is intriguing and poorly understood, yet fascinating as a philosophical subject.

II. LITERATURE REVIEW

O.I. Abdullah et al (2013) investigated stresses and deformations of a dry friction clutch system. To study the stresses and deformations for clutch the finite element method is used. There are five algorithms used for surfaceto-surface contact type are Penalty method, Augmented Lagrange, Lagrange multiplier on contact normal and penalty on tangent, Pure Lagrange multiplier on contact normal and tangent, Internal multipoint constraint method. To obtain pressure distribution between contact surfaces Penalty and Augmented Lagrangian algorithms are used. ANSYS 13 software has been used to perform the numerical calculations. The effect of contact stiffness factor FKN on pressure distribution between contact surfaces, stresses and deformations was studied.

O.I. Abdullah et al (2013) explained optimization of shape and design parameters of the rigid clutch disc using FEM. The numerical solution of computing the stresses and vibration characteristics of rigid clutch disc is presented. Two types of rigid clutch discs have been investigated the reference and new suggested model. The response of the new suggested models have been compared with the reference model. For numerical solution ANSYS and Solid Works have been used. The results show that by adjusting the design parameters stresses and vibration characteristics can be controlled and suggested models improve the response of friction clutch.[3] Ali Belhocine et al (2016) explained a numerical parametric study of mechanical behavior of dry contact slipping on disc pads interface. The determination and visualization of structural deformations due to contact of slipping between the disc and pads is presented. The variations of stresses in rotating disc and ring bodies are predicted by meshed models. A convergence test is intended to evaluate the influence of the mesh on the accuracy of the numerical solution. Using the developed model, the influence of design parameters on result was examined using Finite Element Method. Influence of fine mesh, pad material, young's modulus of pad material, friction coefficient, rotational speed of the disc on the computation results was studied.[4]

V.J. Deshbhratar et al (2013) analyzed design and Structural Analysis of single plate friction clutch. In designof friction clutches knowledge of thermo-elasticity is very important. The stress analysis of single plate clutch of an automobile is presented. The stresses and forces in the clutch disc are tried to reduce with the help of software. For modeling and analysis pro-e and ANSYS software are used. Based on the results it is clear that value of equivalent stresses for given loading conditions are less than allowable stresses for particular condition. Hence the design is safe and the efficient and reliable design of clutch is find out.[5]

Animesh Agrawal et al (2014) presented Optimization of Multi plate friction clutch for maximum torque transmitting capacity using uniform wear theory. Uniform wear theory is used to solve the optimized results for multi plate clutch. Operation research is the branch of mathematics which deals with application of scientific methods and technique to design problems and establishing optimal solutions. The two methods used for optimization are stochastic programming and geometric programming. In actual practice, due to tolerances design variable becomes probabilistic.

This gives proper considerations at the time of design. [6]The objective function was calculated by variation in value of number of friction surface, in value of friction coefficient, in value of intensity of pressure. The result charts can be used for design purpose so as to improve the design.

Clutch Assembly By Rajesh Purohit [7] The Static Design And Finite Element Analysis Of An Automotive Structural Analysis Was Done Using Ansys Software Of The Assembly Of The Clutch Plate, The Pressure Plate And A Diaphragm Spring. The Plots For Equivalent Stress, Total Deformation And Factor Of Safety Were Obtained And The Design Was Continuously Optimized Till A Safe Design Was Obtained. Uniform Wear Theory Was Used For The Analysis. They Said That It Is Possible To Predict Clutch Wear. The Front Surface Temperature Of A Clutch Pressure Plate Is Studied For Clutch Wear Prediction. A Combined Deterministic Plus Stochastic Modeling Approach Is Used To Fit The Front Surface Temperature Data. The Material Assignment Is As Follows: Clutch Plate- Structural Steel, Pressure Plate- Cast Iron Gs-70-02 And Diaphragm SpringSpring Steel. The Friction Material Assumed Is Molded Asbestos Opposing Cast Iron/ Steel Surface. So Finally They Conclude From The Finite Element Analysis Was Carried And Find Equivalent VonMises Stress, Total Deformation And Stress Tool (Factor Of Safety) Were Calculated And Analyzed. The Finite Element Analysis Showed That The Designed Friction Clutch Assembly Is Safe.

Design And Analysis Of Clutch Using Sintered Iron As A Friction Material This Paper By Mamta G. Pawar [8] The Modeling Of Clutch Is Done In Detailed Using Modeling Software. After That The Fem Analysis Is Done For Sintered Iron Friction Material. The Stresses & Deformation Obtained For This Friction Material Is Then Compared To Analysis Software Result. At High Sliding Velocity, Excessive Frictional Heat Is Generated Which Lead To High Temperature Rise At The Clutch Disc Surface, And This Causes Thermo-Mechanical Problems Such As Thermal Deformations And Thermo-Elastic Instability Which Can Lead To Thermal Cracking, Wear And Other Mode Of Failure Of The Clutch Disc Component.By Analysis They Concluded The Stresses Using Kevlar As A Friction Material & Sintered Iron Is Near About Same. Torque Transmission Capacity Of SinteredIron Friction Material Is 350 To 400n Which Is More Than Kevlar. Total Deformation In Kevlar Material Is Less Than Sintered-Iron Friction Material. Sintered-Iron Material Can Sustain Higher Temperature.

The past literature that May Thin Gyan, Hla Min Htun, and HtayHtay Win [9] are proposed different materials for a single plate clutch and structural analysis of a single clutch plates.

B.Sreevani, and M.Murali Mohan [10] has focused on comparision of different materials for single plate clutch and static and dynamic analysis of single plate clutch.

Vishal J. Deshbhratar, and Nagnath U. Kakde [11] has performed design and Structural Analysis of Single PlateFriction Clutchthe values of Equivalent stresses for material loading conditions it is clearly seen that these are less than the allowable stresses for that particular material under applied conditions the part not going to yield and hence the design is safe.

G.Kannan, K.Krishnamoorthy, and K.Loheswaran [12] has performed Review on Different Materials Utilized in Clutch Plate.

Anil Jadhav, GauriSalvi, Santosh Ukamnal, and Prof.P.Baskar[13] were demonstrated Static Structural Analysis of Multiplate Clutch with Different FrictionMaterials the result of stress distribution has been carried out.

AbhijitDevaraj [14] performed optimize the design of the clutch plate, so as to deliver maximum performance and last longer.

A.Krishna Reddy, SeshaTalpa Sai, and Mangeelal [15]has performed stresses as well as deformationclear the idea about what parameter should have been taken into account while defining the single plate friction clutch.

SagarOlekar, Kiran Chaudhary, Anil Jadhav, and P. Baskar [16] were demonstrated the total deformation of clutch plate for different materials to find the better lining material and structural analysis of multi plate clutch using ANSYS.

Ganesh Raut, Anil Manjare, and P.Bhaskar [17] were demonstrated the static analysis on Friction clutch by using Finite element analysis the results of stress distribution, maximum shear stress and total deformation has been carried out.

Shaik Mohammad Ali and N.Amaranageswara [18] has studied about different materials for friction clutch plate and find the stress values for structural analysis and temperature values for thermal analysis of positive multiple Friction plate using FEA.

Clutches are mechanical subsystems of the power-train. For intermittent periods only, the transmission of rotary motion from one shaft to another is achieved by a clutch. A clutch function

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is to produce a smooth, i.e. \jerk free", and gradual increase in the angular velocity of the driven shaft, until full coupling between the two shafts is achieved. Then, this coupling must be maintained for transmitting the entire mechanical power from the driving shaft to the driven shaft without subsequent slip (Bezzazi et al., 2006).

In vehicles equipped with automatic or hybrid transmissions, several types of clutches exist that allow transmission of torque, switching of gears and prevention of rotation of certain elements, as needed (Ingram et al., 2010). Moreover, these transmissions remove the need for the user to operate a manual clutch. The torque converter, featured in some automatic transmissions (GM Powertrain, 2007), is a special fluid coupling that includes a stator component. The converter usually forms the primary component for the transmittal of power from the engine wheel to the transmission input shaft. The torque converter allows the engine to run when the vehicle is stationary. In some HEV transmissions such as the GM two-mode powertrain (Hendrickson, Holmes and Freiman, 2009), a torque converter is not required, since there is an operating mode where the engine can be rotating while the vehicle is at a standstill. In that case, the torque converter is typically not used in order to remove the losses associated with torque converter slip and the design compensates for the loss of the torque converter's torque multiplication factor. However, this varies for different manufacturers. Mechanical multi-plate disk clutches can be input, range, or brake clutches, all being of the \wet" type, i.e. their friction surfaces are wetted by the working fluid. These clutches are located at several places within the gear transmission assembly.

The input clutch assembly is located inside the input shaft and housing assembly. When fully applied, the input clutch provides the power to the gear sets. Correspondingly, range clutches allow selective components of the respective gear range in operation to rotate. Brake clutches, on the other hand, mechanically hold their transmission element at rest with respect to the transmission case.

A typical multi-plate disk clutch pack consists of a number of alternating friction and separator plates as displayed in the below figure, a clutch hub, pistons, return springs, snap rings, a retainer-ball assembly, bearings, a backing plate and a clutch drum. With the engine running, the line pressure from the oil pump assembly is fed through drilled holes in the valve body, through the case cover assembly to the clutch housing.

Automatic transmission fluid flows in between these plates and soaks the friction material. Therefore, the fluid wets the surfaces of the plates and forms an oil film between them. This fluid typically enters through the clutch hub, flows radially between the plates by centrifugal force, and is discharged through the oil orifices of the clutch drum (Yang and Lam, 1998).

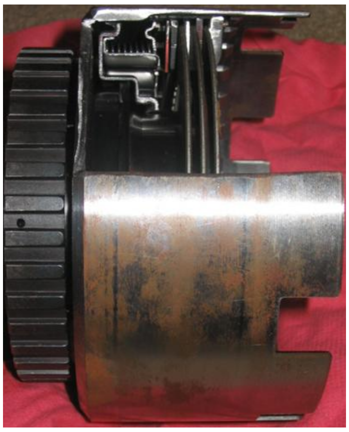

Figure II-1Sectional view of a typical multi-plate disk clutch assembly

Clutch performance is highly dependent on the high friction interface between the disks. The friction disk consists of a steel ring coated by a friction material (usually a paper-based material) on both sides; the separator disk consists of a steel counter-surface, which leads to a combination of high friction coefficient with low wear. Shown in the below figure is the combination of friction and separator disks in a clutch pack. This friction material must be capable of operating at very high temperatures, and must provide a high coefficient of friction throughout the life of a transmission (Ingram et al., 2010). An extensive literature is available on the tribology of wet clutches, explaining the different phenomena involved. Important tribological observations are outlined below.

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Figure II-2Clutch disks

Lubrication in All Regimes. A wet clutch pack is immersed in a lubricant oil commonly called automatic transmission fluid (ATF). The main functions of the ATF are to perform lubrication, to maintain a suitable temperature within the transmission, and to aid in transporting chemical additives to the disk surfaces(Ingram et al., 2010). The working of a wet clutch demands lubrication in all regimes (Larsson, 2009), such as:

- Hydrodynamic lubrication, or full _lm lubrication, which exists when the clutch is disengaged (Yuan et al., 2007; Rivire and Myhra, 2009). A thick ATF film typical thickness of 100 _m (Yuan et al., 2007)|is maintained between the contacting disk surfaces. The lubrication here is mostly affected by the viscosity of the ATF and the relative speed between the disks (Rivire and Myhra, 2009). Viscous shearing of the ATF between the plates in this regime causes a drag torque (Yuan et al., 2007; Kitabayashi, Li and Hiraki, 2003).
- Elastohydrodynamic lubrication, which occurs at the very beginning of clutchengagement. As the contacting disks approach each other, a squeezing effectby the hydrodynamic film pressure reduces the film thickness (Ting,1975; Gao and Barber, 2002). ATF begins to escape through tiny poresand grooves on the friction disc (Ingram et al., 2010). A low _lm thickness(typically from 0.01 _m to 10 _m) and increase in contact pressures result in local elastic deformation on the surfaces without any asperity contact(Rivire and Myhra, 2009).
- Mixed lubrication begins as the fluid film undergoes a squashing effect andthe surface asperities begin to come into contact. With the engagementof the clutch close to completion, deformations begin on the contactingsurfaces and torque is transmitted by both asperity contact and viscousfriction (Ting, 1975). A mixture of elastohydrodynamic lubrication andboundary lubrication (i.e. the next lubrication regime) is noticed with anATF _lm thickness between 0.01 _m and 1 _m. Moreover, the uid behavesmore like a solid than a liquid (Rivire and Myhra, 2009).
- Boundary lubrication exists when the clutch is completely engaged (Ingramet al., 2010; Gao and Barber, 2002). This complex phenomenon occurswhen the two disks and the thin oil _lm rotate as a rigid body. Filmthickness typically ranges between 1 nm and 100 nm, with the ATF usuallybeing adsorbed in the solid surface via its pores and asperities. The ATF reduces wear and lowers the coefficient of friction because of the lower shearstress of the fluid, as compared to that of the solid disk (Larsson, 2009;Rivire and Myhra, 2009). The real surface contact area mainly consists ofdeformed asperities and, hence, is only a fraction of the nominal surface area(Ingram et al., 2010; Eguchi and Yamamoto, 2005). Friction-produced heatoccurs in this regime; it is a major cause of material and ATF degradation(Gao, Barber and Chu, 2002; Yang and Lam, 1998).

Friction Material. Several types of friction material combinations are used in automatic transmissions. These include paper-based friction-steel disks, sinteredbronze-steel disks, steel-steel disks, carbon fibre-steel disks and Kevlar disks(Ingram et al., 2010; Lam, Chavdar and Newcomb, 2006; Marklund and Larsson, 2007). The most commonly used paper-based friction material is manufactured in the same fashion as regular paper. A stock is prepared by mixing fibrous material, fillers and chemical additives together, then soaking the mixture in water. This is dried on a moving wire line, and then pressed into rolls of paper. After saturating and curing this paper with resin, it is bonded adhesively onto a steel core plate, thereby forming the friction disc. The separator counter-face disk is made of steel (Ingram et al., 2010). Fibres include materials such as asbestos, cellulose, cotton linter, aramid, Kevlar, carbon fibre, lapinus and basalt, while fillers can be diatomaceous earth, clay, silicone particles, cashew dust, barites and calcium carbonate (Lam, Chavdar and Newcomb, 2006). Chemical additives usually include alumina, chromium oxide and silicone. Several resins can be used such as phenolic resin, modified phenolic resin and cresylic phenolic resin (Ingram et al., 2010). The frictional properties of the material can be altered by varying the components used, or by changing the steps involved inthe manufacturing process (Lam, Chavdar and Newcomb, 2006).

The porosity is one important property that can be modified during the friction material design process. The permeability of the friction material is a measure of the ATF ability to flow, or penetrate the friction material via its pores (Marklund and Larsson, 2007). The permeability varies with mechanical wear and thermal degradation of the friction material, and consequently changes over the life of the clutch (Yang and Lam, 1998). A new wet friction material possesses good permeability; however, a glazed friction material shows poor permeability (Gao and Barber, 2002; Marklund and Larsson, 2007). This effect, influencing the frictional behaviour andengagement time of the clutch, will be briefly discussed.

Friction Characteristics. Two main types of friction exist in wet clutches,Coulomb friction and Eyring viscosity friction (Eguchi and Yamamoto, 2005). The coefficient of friction, μ, in wet clutches is dependent on several parameters such as material, type of ATF used and contact surface nature. Its value is strongly influenced by the sliding velocity, v, of the clutch disks, the operating temperature, and the applied normal loads (Gao and Barber, 2002; Gao, Barber and Chu, 2002; Lam, Chavdar and Newcomb, 2006). Friction characteristics are well described by μ-v ratios for a given set of temperature, pressure, geometry and ATF. These μ-v curves have a strong dependence on the unstable shudder effect. Stick-slip-induced friction causes undesirable vibrations that are responsible for the shudder effect, and are also a major cause of discomfort to the users during clutch engagement (Gao and Barber, 2002). Their detrimental effects can hamper the performance of the clutch, and cause fatigue failure, surface damage, noise and severe wear. Positive slopes of μ-v curves generally show no shudder effects, these curves commonly appearing among new friction material and new oils. Negatively sloped curves are common in worn-out friction material and degraded oils. Shudder is mostly noticed in the negatively sloped μ-v curves (Gao, Barber and Chu, 2002; Lam, Chavdar and Newcomb, 2006).

Contact Properties of the Friction Material. The real area of contact is only a small percentage of the nominal surface area (Ingram et al., 2010; Gao andBarber, 2002). This contact area can be calculated by capturing grey-scale images, and then converting them into binary images, where the perimeters of the contact area can be traced. This real contact area increases with the contact pressure, as more fibres deform and wear. It is also observed that the shear strength of the boundary film has a strong dependency on the contact area (Ingram et al., 2010; Eguchi and Yamamoto, 2005).

Automatic Transmission Fluid. The automatic transmission fluid (ATF) is an important part of the clutch system. It must provide the desired friction characteristics, and maintain its performance throughout the working life of the transmission. Moreover, it must lubricate all elements within the automatic transmission housing. Current ATFs manufactured have been standardized (Arakawa, Yauchibara and Murakami, 2003), such as DEXRON-VI, to meet stringent requirements. The formulation of ATF involves several base oils and additives, with surface-active additives enhancing the frictional performance. Base oils are usually mineral oils and synthetic fluids like paraffinic oil (Kugimiya et al., 1997). The additives usually include properties such as anti-oxidant, dispersant, detergent, anti-wear, viscosity-index improver, anti-rust, corrosion inhibitor, anti-forming and friction modifier, among others (Shirahama, 1994).

Clutch Engagement: Torque Transmission. The ATF _lm thickness betweenclutch disks varies with time during engagement. As the clutch plates approach each other this film thickness rapidly decreases. Numerical simulations were carried out by Gao and Barber (2002) using the Runge-Kutta method. These simulations were conducted for a wet clutch functioning as a brake, i.e., with the relative angular velocity between the friction and separator plates decreasing to zero and one side of the clutch at zero angular velocity. The instant this velocity reaches zero is denoted as the end-of-engagement. Film thickness is also observed to decrease rapidly untilit nearly plateaus when the clutch plates are completely engaged (Gao and Barber, 2002).

Hot Spot Generation during Short-Term Engagement. A common problem observed in clutches is the formation of hot-spots caused by thermo-elastic instabilities. Local areas of surface failure caused by substantially high temperatures and high pressures are macroscopically seen on the steel separator surface. Hot spots cause high thermal stresses, which lead to plastic deformation, and transformation of the ferrous material into martensite. This results in cracks and, ultimately, in clutch failure. Severe hot spotting occurs during high initial sliding speeds, small geometric imperfections being capable of triggering large hot spot formations. Lowering the Young's modulus of the friction material can effectively lower the hot spots growth area (Zagrodzki and Truncone, 2003).

Clutch Disengagement: Drag Torque. The generation of undesirable hydrodynamic torque is inherent to a disengaged clutch pack. It is deemed an energy sink that lowers the efficiency of all transmissions that employ them. The viscous effect of the ATF causes a drag on the rotating member of the disengaged clutch. A number of factors can influence drag torque, most notably, the distance between the clutch disks, ATF flow rate, groove patterns, disk facing area and the waviness of the disk plates (Kitabayashi, Li and Hiraki, 2003). A hydrodynamic model was developed for the drag torque and shear stress on the rotating plate (Yuan et al., 2007). This model, which incorporated the surface-tension effects of the oil film, showed better agreement with test data for a non-grooved clutch pack. According to this model, for an open clutch pack operating at steady state in an incompressible ATF film, rotating in a turbulent regime, the shear stress and parasitic drag torque vary with the radius where full oil _lm breaks due to centrifugal effects, the inner radius of the rotating disc, the ATF viscosity, the angular velocity of the rotating disc, the clearance between disk and the Reynolds number based on the clutch clearance (Yuan et al., 2007).

Fish (1991) evaluated drag losses in wet clutches using the SAE #2 machine and concluded that drag torque can be minimized by

_ increasing the gap between clutch plates, i.e. pack clearance;
_ lowering the ATF level and flow;
_ reducing the viscosity of the ATF;
_ lowering the clutch speed;
_ altering the grooves on the friction plates;
_ altering the geometry of the clutch plates, e.g. introducing waves.

Hydraulic Actuation of Wet Clutches. Transmission pumps deliver fluid from a low pressure reservoir to a high pressure line in order to actuate several elements, such as clutch packs, regulating valves, torque convertor clutch, and band clutches. They also aid in circulating the fluid to regulate the transmission temperature and in fluid filtration. However, they have several losses which correspond to parasitic losses in the transmission.

Typical range clutches of automatic transmission are actuated hydraulically by feeding in ATF through the driven sprocket support and into the input shaft-housing assembly. A feed hole in the input shaft allows fluid to enter between the piston and the input shaft-housing assembly. Fluid pressure seats a ball check valve assembly that allows the movement of the piston to compress a spring-retainer assembly.

The piston continues to move until it contacts the apply plate, where it compresses the neighbouring plate to cushion the apply plate, and holds the alternating plates against the backing plate and snap ring. When fully closed, the range clutch provides the power to the corresponding transmission component through the friction and separator plates. To release the range clutch assembly, fluid pressure exhausts through the apply passage in the input shaft-housing assembly and driven sprocket support.

In the absence of fluid pressure, the input spring-retainer assembly moves the piston assembly and releases the apply plate, the cushion plate, and other stacked plates from the backing plate and snap ring. During the release of the fluid, the ball check valve assembly, located in the clutch housing, unseats. Centrifugal force, resulting from the rotation of the input shaft-housing assembly, drives residual clutch fluid to the outer perimeter of the piston housing, and through the unseated retainer-ball assembly. If the fluid does not completely exhaust, there can be partial engagement (GM Powertrain, 2007).

According to Kluger et al. (1996), hydraulic pumping systems commonly used can account for up to 20% of the total parasitic losses in a typical automotive automatic transmission during the Environmental Protection Agency (EPA) city cycle.

Moreover, Kluger et al. (1996) reported that the pumping pressure, fluid temperature and component clearances had the largest influences on pump leakage. In their paper, Kluger et al. evaluated several pumping systems on the basis of their mechanical efficiency, volumetric efficiency, pumping torque, discharge flow and overall efficiency. Hysteresis pressure losses have been commonly found in brake actuation systems and have been identified as another energy sink in hydraulic systems (Tretsiak et al., 1975). Besides these losses, unavoidable head losses inherent in fluid flow in line channels are a consistent drawback of hydraulic actuation.

2.1 Summary

There is always a scope for design improvement. To optimize the design of the clutch, finite element analysis can be used to assist the engineer in which shapes lead to a better design. Based on these sensitivity calculations the shapes can be improved. This method is used in the thesis to optimize the clutch shapes.Kevlar Aramid Fiber 49 material is used for the clutch lining material as a better choice. We can do optimization for all design parameters to check structural integrity but we will use three parameters as a optimization scope due to time constrains and computer system constrains.

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III. RESEARCH STATEMENT AND OBJECTIVES

3.1 Research Statement

From the literature review of a clutch, there is a scope of improvement of a clutch design. In the literature, parameters affecting on a clutch performance and research work are carried out. However, the optimization of the design of a single plate clutch by using simulation technique is the scope of the research.

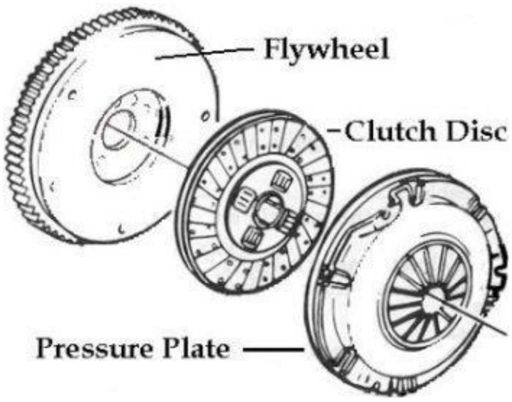

Figure III-1Clutch Assembly

Gradual engagement clutches like the friction clutches are widely used in automotive application for the transmission of torque from the flywheel to the transmission. The three major components of a clutch system are the clutch disc, the flywheel and the pressure plate. Flywheel is directly connected to the engine's crankshaft and hence rotates at the engine rpm. Bolted to the clutch flywheel is the second major component: the clutch pressure plate. The spring-loaded pressure plate has two jobs: to hold the clutch assembly together and to release tension that allows the assembly to rotate freely. Between the flywheel and the pressure plate is the clutch disc. The clutch disc has friction surfaces similar to a brake pad on both sides that make or break contact with the metal flywheel and pressure plate surfaces, allowing for smooth engagement and disengagement.

When the clutch begins to engage, the contact pressure between the contact surfaces will increase to a maximum value at the end of the slipping period and will continue to stay steady during the full engagement period. During the slipping period, large amount of heat energy is generated at the contact surfaces, which gets converted to thermal energy by first law of thermodynamics. The heat generated is dissipated by conduction between the clutch components and convection to the environment. Another loading condition is the pressure contact between the contacts surfaces that occurs due to the axial force applied the diaphragm spring. In addition to the above output responses, this work also considers the Vibrational characteristics of the clutch plate during the full engagement period. The engine and the transmission components experience dynamically varying loads during normal operation. This will cause vibrations and hence, one must design the clutch system so as to avoid resonance with the transmission and engine components.

3.2 Objectives

The primary aim of this work is to design a rigid drive clutch system that meets multiple objectives such as Vibrational rigidity, Structural and Thermal strength. Also, to demonstrate a systematic approach to solving multi-objective problems by ANSYS software.

IV. DESIGN OF A CLUTCH

For the optimization of a single plate clutch, following **General Nomenclatures are considered:**

P1 – Friction pad inner diameter
P2 – Friction pad thickness
P3 – Friction facing thickness
Ri – Inner radius of clutch disc in meters
Ro – Outer radius of the clutch disc in meters
N – Speed of engine in rpm = 3750 rpm
ωr – angular velocity in rad/s
Pmax – clamping pressure in MPa

The material considered for the friction pad is Kevlar 49 Aramid. Uniform Wear Theory is considered for calculations, and accordingly, the intensity of the pressure is inversely proportional to the radius of friction plate.

4.1 Design parameters

$R=Ri+Ro\ /\ 2=0.1m$

In general, the frictional torque acting on the clutch plate is given by,

$T=N\times\mu\times W\times R$

In general, the frictional torque acting on the clutch plate is given by,

$W\ =\ 3000NP\times r=C\ (constant)$

Axial force on the clutch pad,

$W=2\pi\times C\times(Ro-Ri)$

$C\ =\ 0.0119Nm$

The maximum pressure occurs at the inner radius and the minimum pressure at the outer radius.

In general, the frictional torque acting on the clutch plate is given by,

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$Pmin = CRo = 0.0994$ $Mpa Pmax = CRi = 0.1492\ Mpa$

Here, we consider the maximum pressure value obtained in the Finite Element Analysis of the clutch plate.

4.2 Thermal Calculation

T – Temperature of the disc in Celsius

T_l – Limiting temperature of the material in Celsius = 150ºC

μ - Coefficient of friction of the material = 0.4

k – Thermal conductivity of the material in Watts per meter Kelvin

h – Heat transfer coefficient of the material.in Watts per sq. meters per Kelvin.

q – Heat energy generated in watts

q_f – heat flux in W/m2

t – Slip time in seconds = 0.5s

A – Area of a friction pad = $0.000931 m^2$

$\omega_r = 2\times\pi\times N / 60 = 392.6\ rad/s$

$q = \mu\times P_{max}\times\omega_r = 23.4375\ W$

$q_f = q / A = 25155\ W/m^2$

4.3 Frequency Calculation

F_e – Engine frequency

n – Order of frequency (1st order & 2nd order)

Ne – Engine rpm range (1000 rpm-4750 rpm)

$F_e = Ne \text{ x } n / 60$

4.4 Model Analysis and Sub- System optimization Study

4.4.1 Design Variables

Three design variables are chosen that will serve as constraints for all the three sub-systems. In a preliminary simulation study, it was observed that these parameters have a greater impact on the characteristics under study. Also, the rationale behind choosing wide range of values is to avoid excluding good designs.

$g1{:}140 \le P1 \le 160$

$g2{:}\ 2 \le P2 \le 10$

$g3{:}\ 0.5 \le P3 \le 3.5$

4.4.2 State Variables

The response quantities that are dependent on the above design variables are

1. Temperature
2. Vibration frequency
3. Equivalent Stress

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4.5 Sub-system Optimization Process

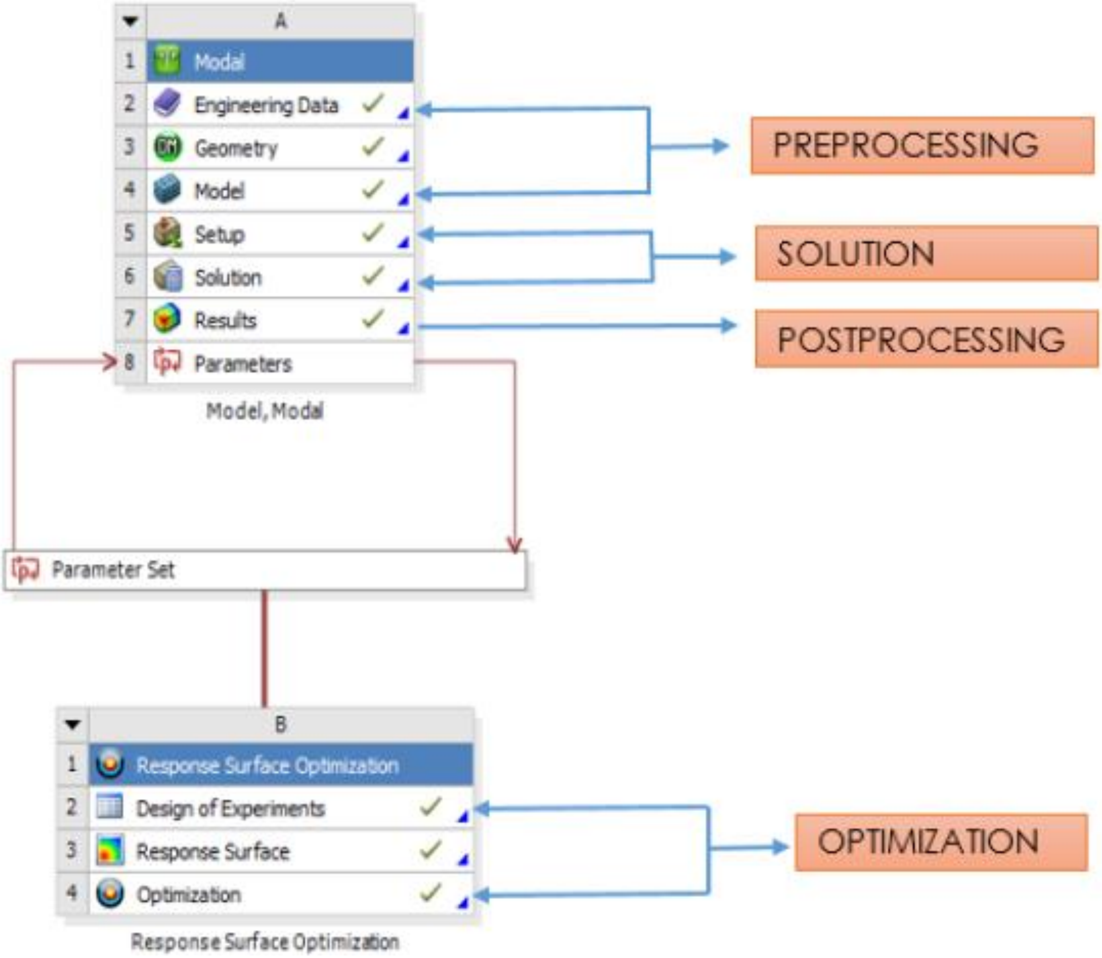

Figure IV-1Schematic Diagram of subsystem optimization in ANSYS

The desired output response for each sub-system is obtained through Finite Element Analysis using ANSYS by selecting a parametric model with the design variables as input geometric parameters.

The following steps were sequentially carried out for each sub-system analysis and further for optimization.

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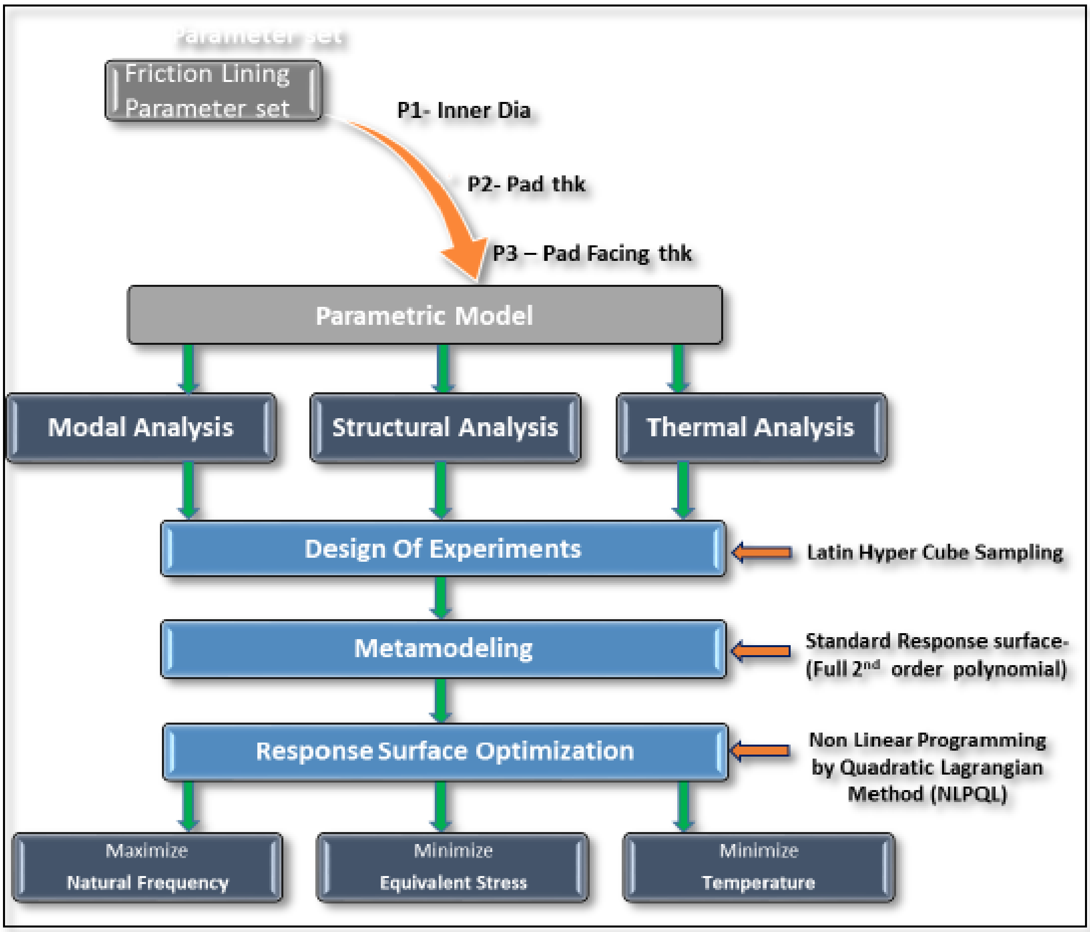


Figure IV-2Subsystem Optimization Methodology

4.6 Creation of Parametric model

a. Create the CAD model for simulation using ANSYS Design Modeler.
b. Define the input parameters (Design Variables) to be investigated. The output parameters (State Variables) are chosen from the simulation results.
c. Define the design space by giving lower and upper bounds for the parameters and based on this the Design of Experiments (DOE) part will sample the design space.
d. Obtain an approximate response of the system by creating a Response Surface for each output parameter.
e. Repair errors to obtain an accurate response surface approximation.
f. Choose a suitable optimization technique after setting the constraints to finally identify suitable design candidates from the Response Surface.

4.7 Engineering Data

Engineering data involves defining clutch material and properties:

Table IV-1Kevlar Aramid Fiber 49 properties

Type	Clutch base plate	Clutch plate lining
Material	Structural Steel	Kevlar Aramide Fiber 49
Properties		
Density	7850	1440
Young's Modulus	2×10^{11} Pa	1.12×10^{11} Pa
Poisson's Ratio	0.3	0.36
Bulk Modulus	1.667×10^{11} Pa	1.333×10^{11} Pa
Shear Modulus	7.6923×10^{10} Pa	4.1176×10^{10} Pa
Specific heat Capacity	434 J kg^-1 C^-1	1420 J kg^-1 C^-1
Isotropic Thermal Conductivity	60.5 W m^-1 C^-1	0.04 W m^-1 C^-1

4.8 Geometry

The base dimensions for the model are:

Table IV-2 Initial Input parameter values

Parameter	Value
P1	160 mm
P2	2.7 mm
P3	0.8 mm

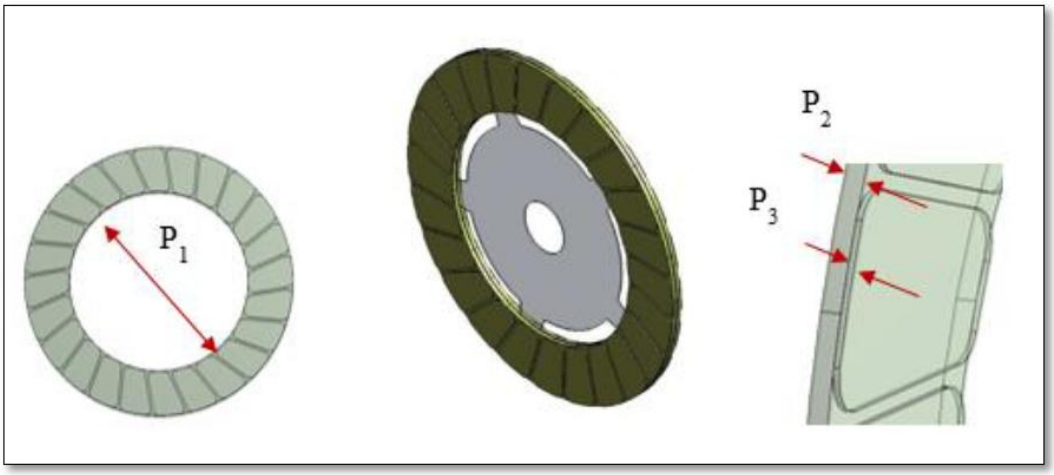
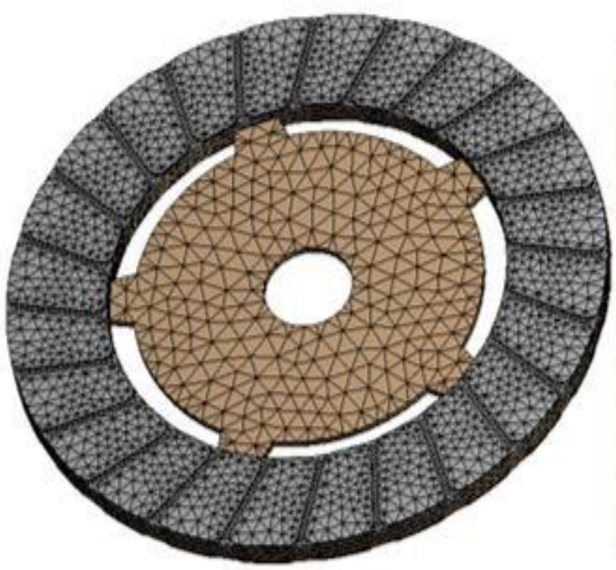


Figure IV-3 Parametric CAD model

System optimization is basically the process of enhancing the capabilities of a system by integrating the subsystems of which the former is made to the extent that all of them operate above the user expectations. In the project of optimizing the clutch design, the optimized subsystems viz. Modal Analysis, Structural Analysis and Thermal Analysis will be integrated to create an optimal Pareto surface. A Pareto surface is the surface containing optimal points corresponding to the optimal solution of a particular trade-off among the conflicting objectives of the subsystem. In other words, selecting one point from the Pareto surface will always sacrifice the quality for at least one objective, while improving the other objective. Here optimization will be done by two different methods to create a Pareto optimal surface namely Multi-Objective Genetic Algorithm (MOGA) and Non-Linear Programming by Quadratic Lagrangian (NLPQL).

4.9 FEM model

Meshing is an integral part of the FEA process. The mesh influences the accuracy, convergence and speed of the solution. A free mesh is applied with fine element size for the CAD model. A free mesh has no specific element shape or pattern associated with it.



Model (A4) > Mesh

Object Name	Mesh
State	Solved
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Size Function	Off
Relevance Center	Fine
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	High
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	6.1316e-004 m

Figure IV-4ANSYS FEM model

4.10 Objective functions

4.10.1 Modal Analysis

Maximize the 1st order frequency to avoid resonance with Engine and Transmission vibrations.

4.10.1.1 Boundary conditions & Loads

Since this is free vibration analysis, no external forces or loads were applied onto the FEM model. Practically, while measuring the clutch plate natural frequency, it is mounted on its base plate hole. In ANSYS simulation, the clutch plate given was fixed support constraint at its base plate hole diameter.

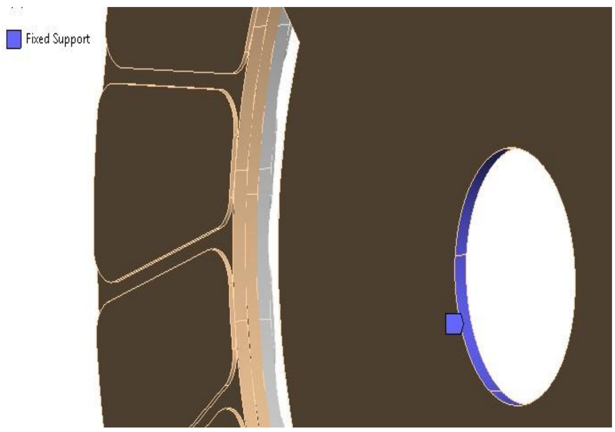


Figure IV-5Boundary conditions, fixed support

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4.10.2 Structural Analysis

Minimize the Max. Equivalent stress acting on the friction pad.

4.10.2.1 Boundary conditions & loads

1) The relative rotational velocity between the clutch plate & flywheel for t = 0.5 s slip is applied on the clutch plate. Here the clutch plate is made to rotate with flywheel kept stationary.

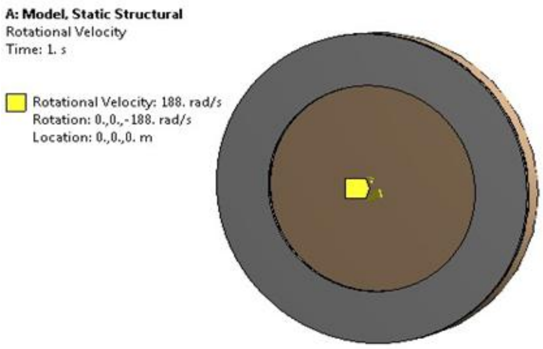

Figure IV-6Rotational Velocity in Z direction

2) The contact pressure on clutch plate exerted by the pressure plate is applied on the clutch plate surface as per calculated value.

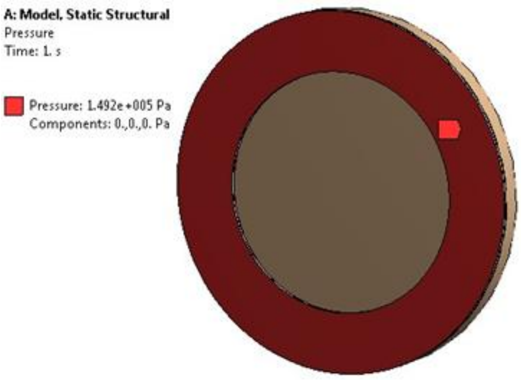

Figure IV-7Clamping pressure load applied on the friction lining

3) The clutch plate is constrained to rotate only about Z direction.

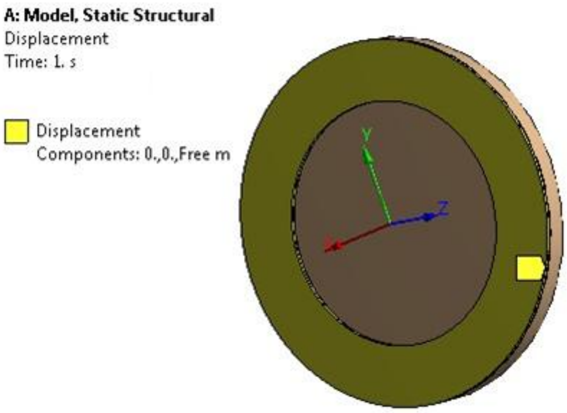

Figure IV-8Clutch lining constrained to rotate in Z direction only

4) The flywheel is given fixed support constraint and is constrained for no rotation in all directions.

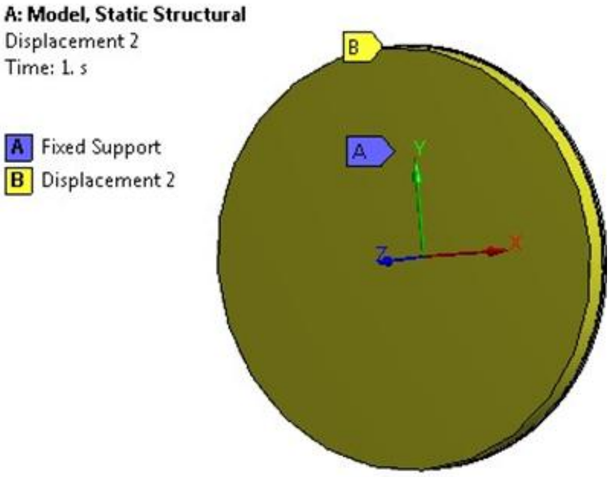

Figure IV-9Constraints applied on flywheel

4.10.3 Thermal Analysis

Minimize the Max. Temperature due to the heat generated on friction pad.

4.10.3.1 Boundary conditions & loads

1) During engagement of the clutches, the friction surface is in contact with the flywheel. The maximum heat flux of 25155 W/m^2 is applied onto the friction pads. The initial temperature for the analysis is set at ambient temperature (35 degree Celsius).

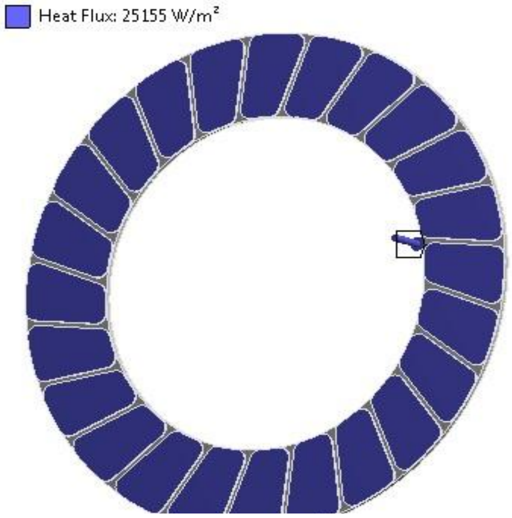

Figure IV-10Heat flux applied on the friction pads

2) The heat generated by the clutch engagement is dissipated through convection and radiation. A convective heat transfer coefficient of 40 W/m^2 (of free air) is applied. For radiation 35 degree Celsius was applied

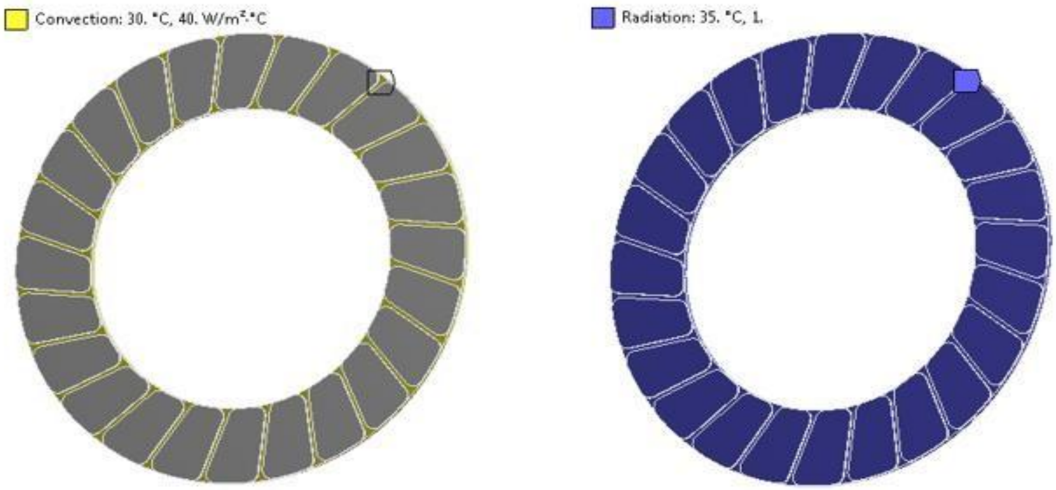

Figure IV-11Loads applied for heat dissipation

4.11 Design of Experiments

The next step is to establish a relationship between the design variables and the output response. Since all the design variables are continuous within their bounds, one needs to sample the design space to determine how many and which parameters should be chosen for creating a response surface. Here, the widely used stochastic sampling technique, Latin Hypercube Sampling (LHS) is adopted. A set of *l* samples are randomly generated regardless of the number of design variables. The advantage of LHS is that the samples do not share the same values on any variable and hence offers a good distribution in the design space. Also, unlike other deterministic methods like Central Composite Design (CCD) and Full factorial methods, the number of simulations required for LHS remain constant (after converging to a maximum) even with an increasing number of parameters. At the end of this stage, output parameters corresponding to the design points as defined by the DOE is obtained.

4.12 Metamodeling

Further, using metamodeling (regression analysis) techniques, a response surface that provides a functional relationship between the output response and the input parameters is generated. A full second order polynomial is the preferred model and uses the method of least squares to determine the value of the unknown coefficients A, B and C in the second order polynomial, $Ax^2 + Bx + C$.The advantage of nonlinear least squares regression like the second order polynomial over many other techniques is the broad range of functions that can be fit.

4.13 Response Surface Optimization

Now that the meta-model of the problem has been developed, gradient-based methods can be applied to search for the optimal point in the response surface. When the design variables are continuous and the optimization is single objective, NLPQL (NonLinear Programming with Quadratic Lagrangian) is a very efficient algorithm. NLPQL uses quasi-Newton methods to converge to the solution. It generates a sequence of QP subproblems which is obtained by quadratic approximation of the Lagrangian function and linearization of constraints. And finally, to stabilize and ensure global convergence, an Armijo line-search is performed.

Once the response surface optimization is completed, a manual refinement of the response surface is done by inputting the optimal point as a design point and re-solving the optimization problem until a good approximation of the actual response is obtained.

4.14 Parametric Study

The effect of input parameters on output parameter is studied from design sample space and the variation of output parameter against each input parameter is plotted

4.14.1 Modal Analysis

Below graphs show the response in frequency with the change in input parameters.

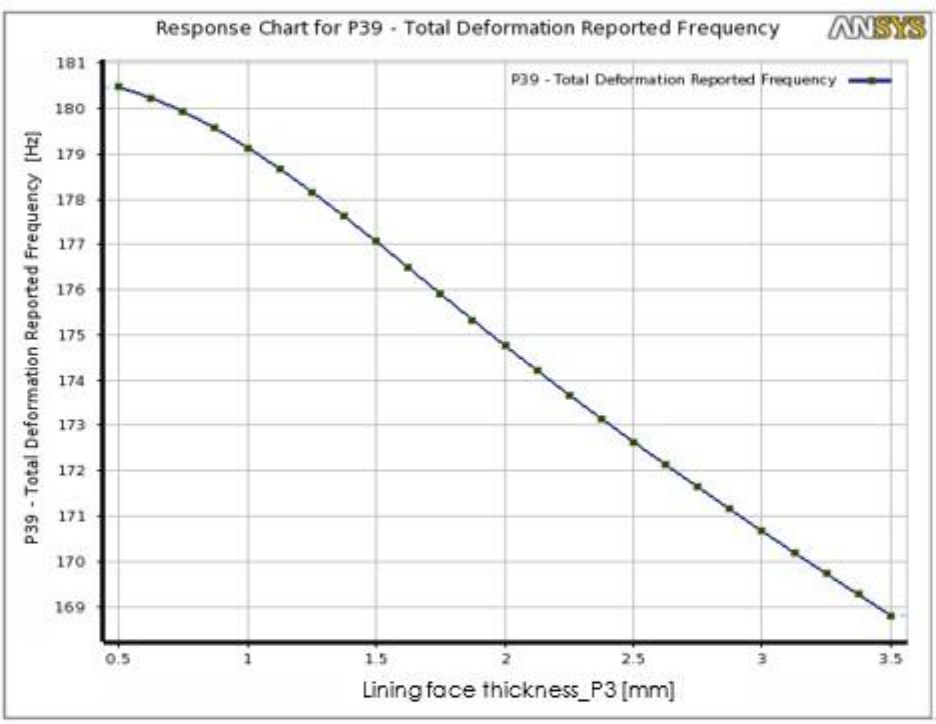

Figure IV-12Frequency vs Lining Face Thickness

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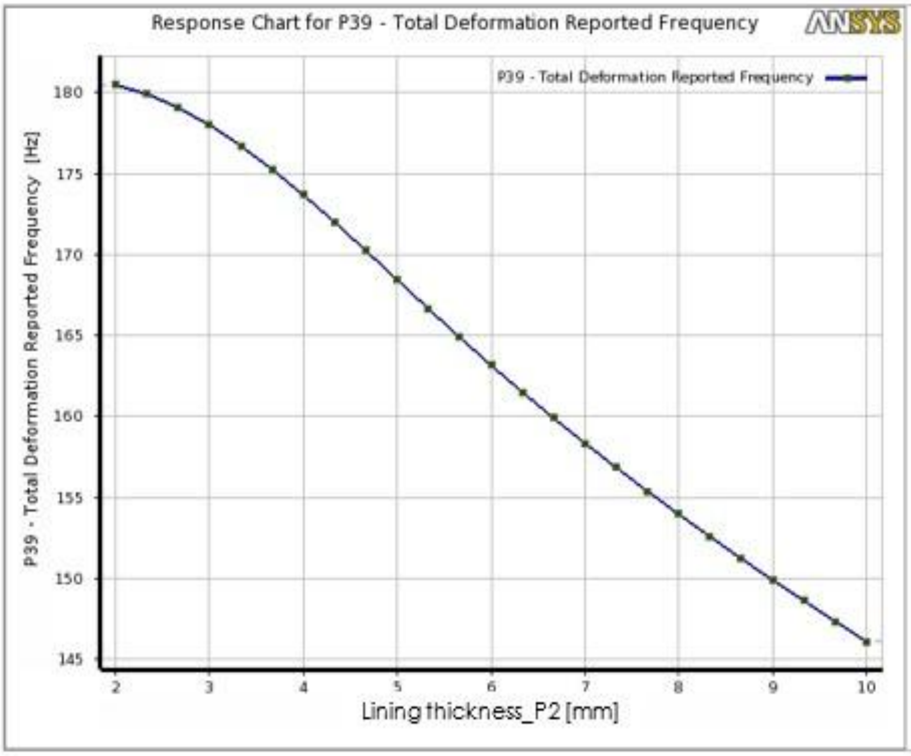

Figure IV-13Frequency vs Lining Thickness

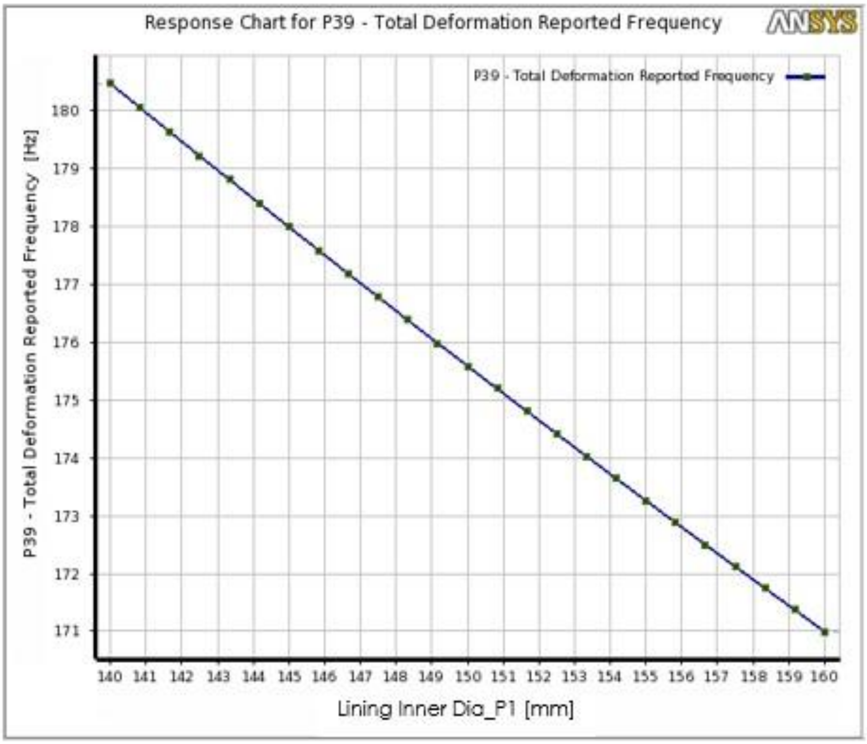

Figure IV-14Frequency vs Lining Inner Diameter

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4.14.2 Structural Analysis

Below graphs show the response in equivalent stress with the change in input parameters.

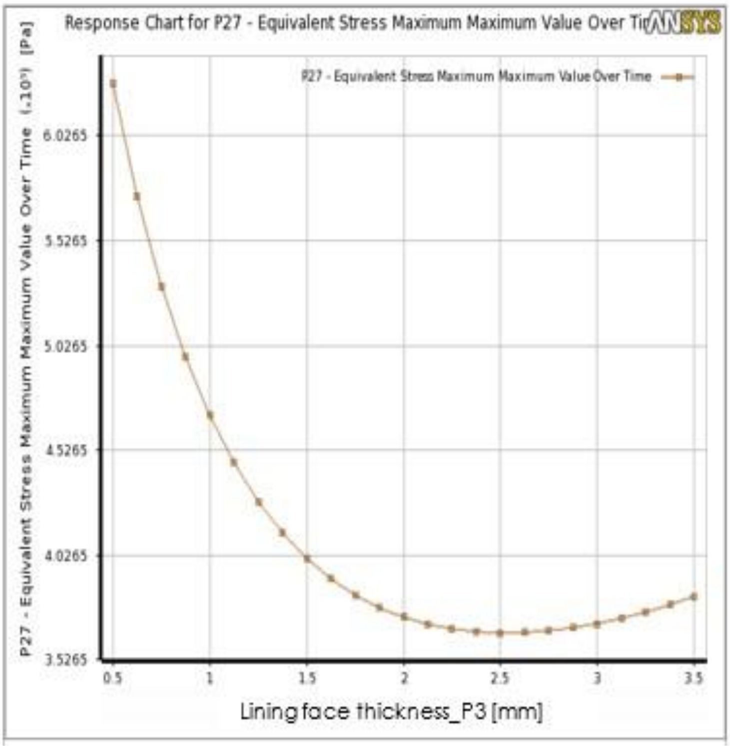

Figure IV-15Equivalent Stress vs lining face thickness

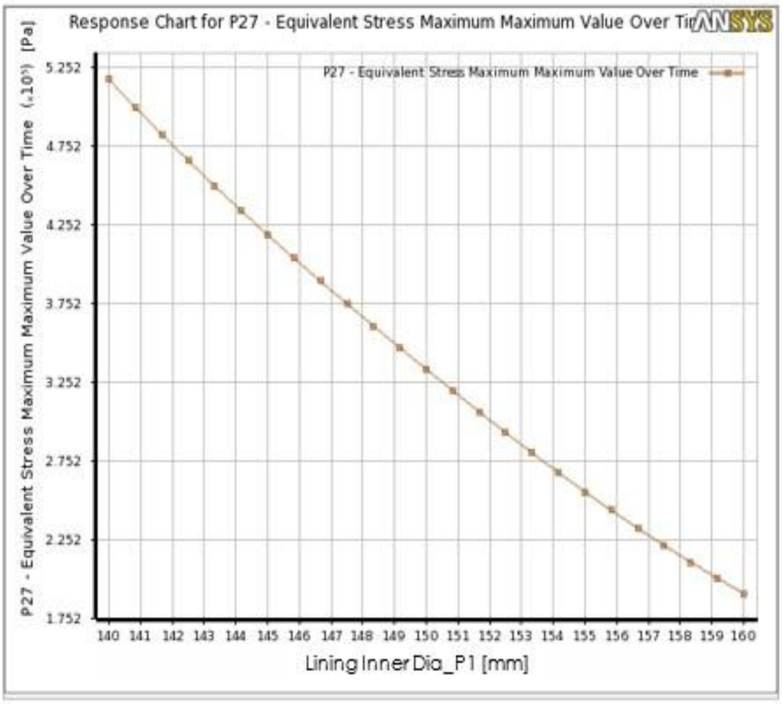

Figure IV-16Equivalent Stress vs Lining Inner Diameter

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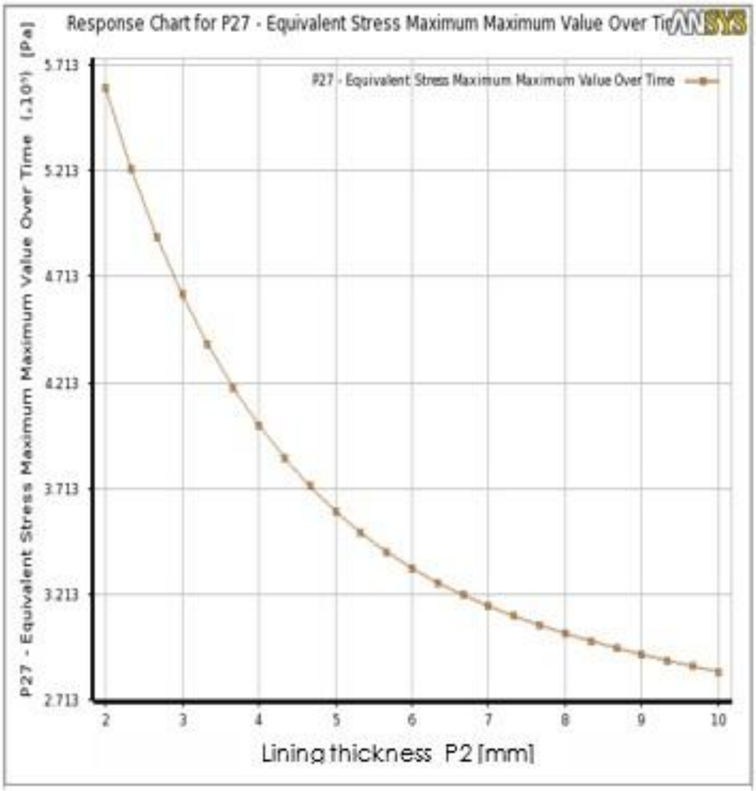

Figure IV-17Equivalent Stress vs Lining Thickness

4.14.3 Thermal Analysis

Below graphs show the response in maximum temperature with the change in input parameters.

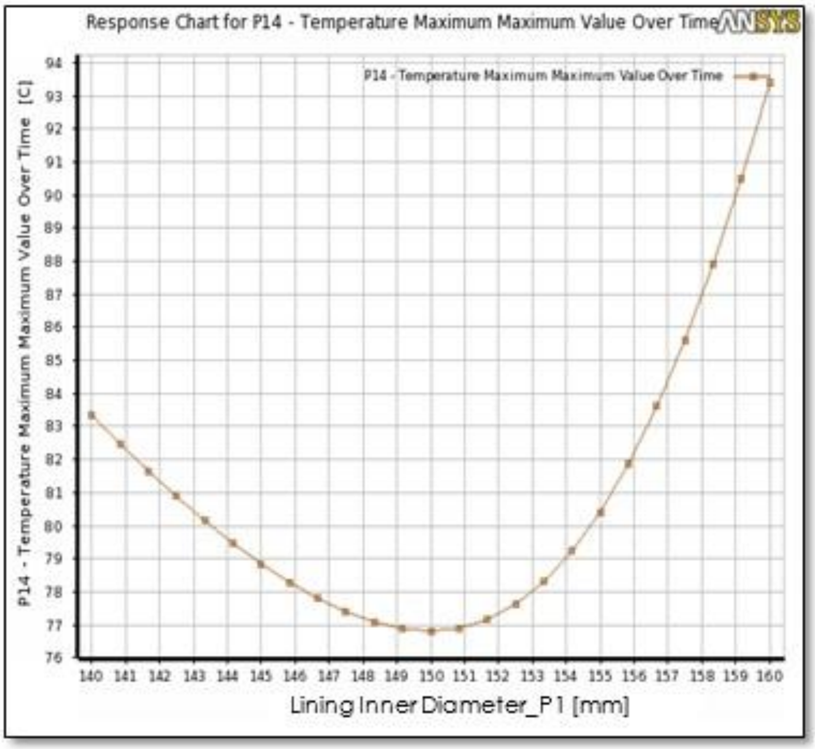

Figure IV-18Temperature vs Lining Inner Diameter

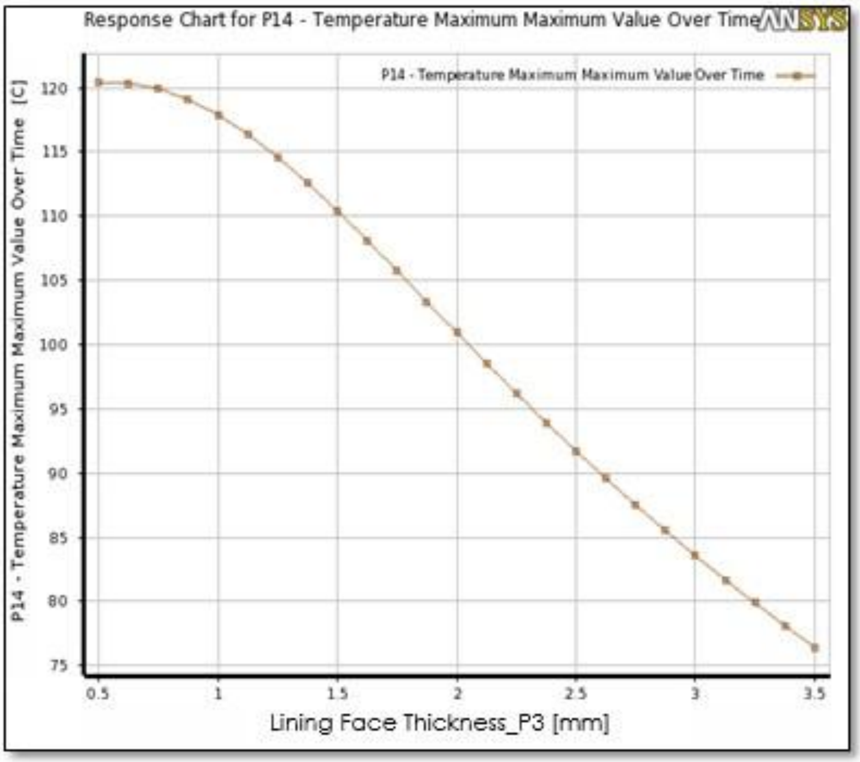

Figure IV-19Temperature vs Lining Thickness

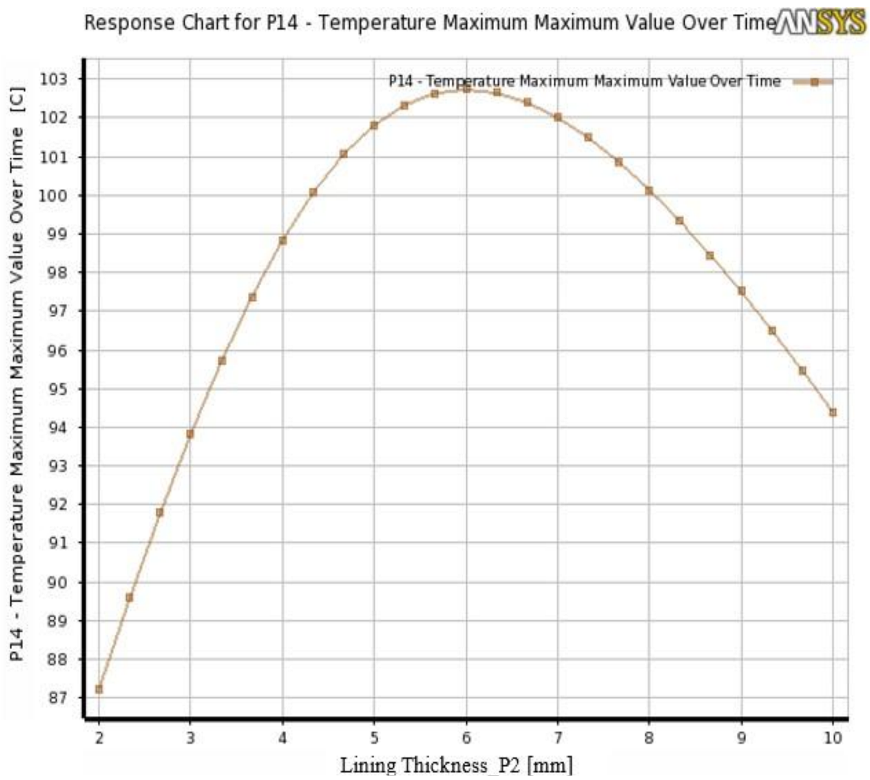

Figure IV-20Temperature vs Lining Thickness

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4.15 Discussion of Results

4.15.1 Modal Analysis (Vibrational Analysis)

The basic equation solved in a typical undamped modal analysis is the classical Eigen value problem

$$[\boldsymbol{K}]\{\emptyset_I\} = \boldsymbol{\omega_i}^2[\boldsymbol{M}]\{\emptyset_I\}$$

Where

$[\boldsymbol{K}]$ = Stiffness matrix

$\{\emptyset_I\}$= Mode shape vector (Eigen vector) of mode

$\boldsymbol{\omega_i}$ = Eigen value

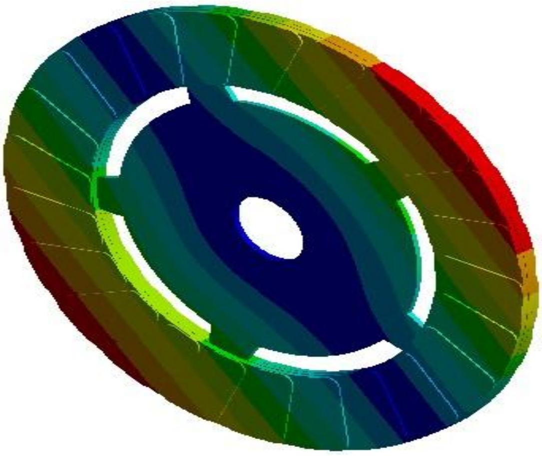
Ω =Natural circular frequency

By default, ANSYS Mechanical APDL uses Block Lanczos Mode Extraction Method to extract modes Following natural frequencies are reported after simulation

Table IV-3Frequency response

Mode	Frequency[Hz]
1	167.42
2	167.44
3	253.96
4	668.67

The mode shapes are



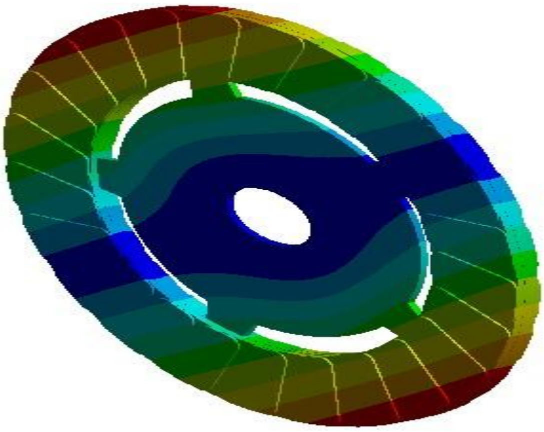
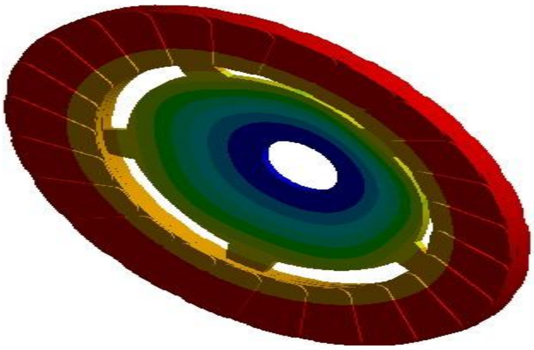


Figure IV-21i)167.44 Hz **ii)167.42 Hz**



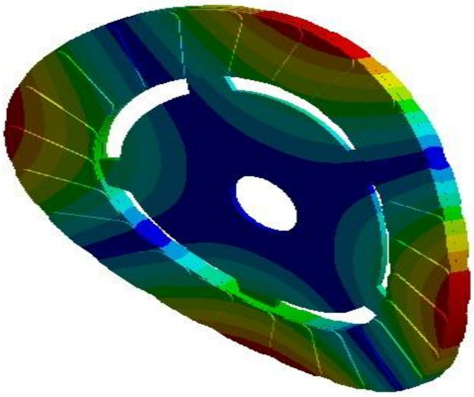


Figure IV-22 **i)253.96 Hz** **ii)668.67 Hz**

The engine rpm range is from 1000 rpm (idling speed) to 4750 rpm (engine fly-up rpm). Taking into standard operating tolerance of 10% on the frequency range, the corresponding 1st order frequency range is **18.326 Hz- 87.076 Hz** and 2nd order frequency ranges **36.663 Hz -174.163 Hz**. These are the frequency bands with which clutch plate frequency should be decoupled.

Here, the first mode frequency is considered as the output parameter for optimization as it is the fundamental natural frequency. The 1st natural frequency (167.42 Hz) needs to be optimized so that it doesn't fall into the engine frequency band thereby avoid resonance.

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4.15.1.1 Optimization Results

The NLPQL optimization is based on the response surface generated by regression analysis of design sample space as defined by below design points

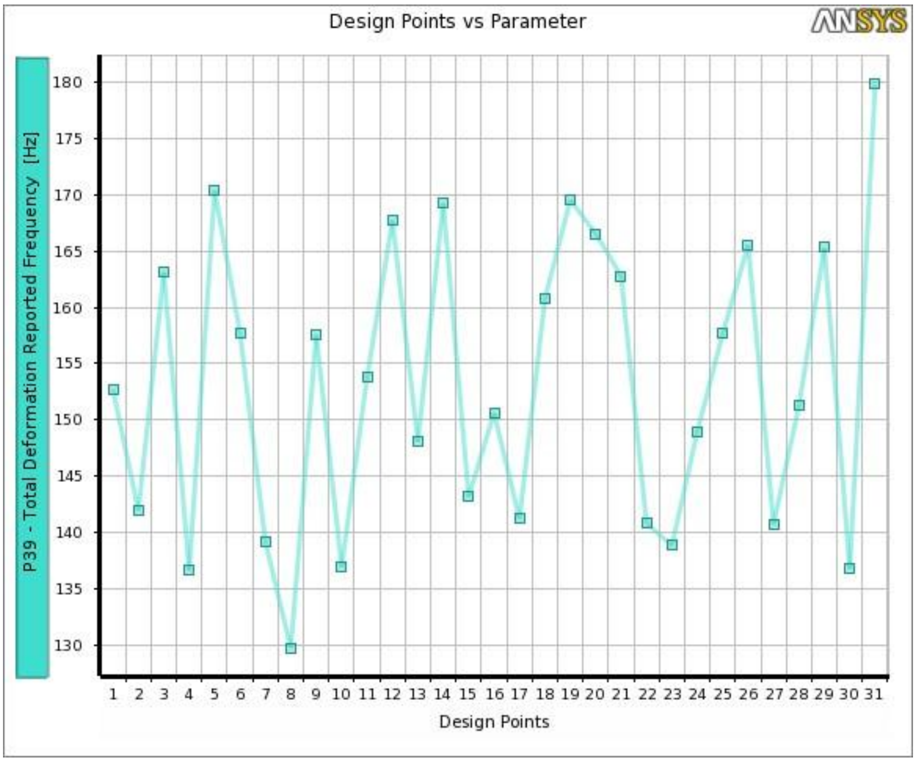

Figure IV-23DOE samples

With optimization, there is 7.42 % improvement in the output frequency which doesn't fall in engine frequency band.

Figure IV-24Optimized Design

Table IV-4Optimized Input Parameter Values

Parameter	Starting Point	Final Design
P1 (mm)	160	140 **(active)**

P2 (mm)	2.7	2 (**active)**	
P3 (mm)	0.8	0.5 **(active)**	
Output	Initial Value	Optimized Value	Simulated Value
Frequency (Hz)	167.42	180.47	179.85

The above table shows the predicted value from NLPQL and observed value from ANSYS Simulation are very close enough.

4.15.1.1.1 Robustness of Solution (Goodness of Fit)

Goodness of Fit shows that the output parameter has been very well approximated by the response surface. The coefficient of determination is 0.99876.

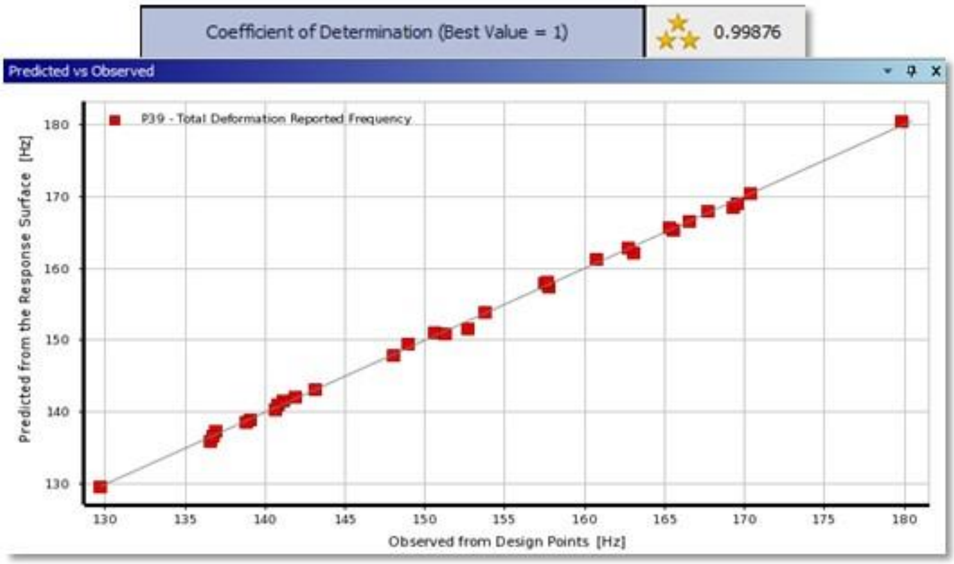

Figure IV-25Goodness of Fit

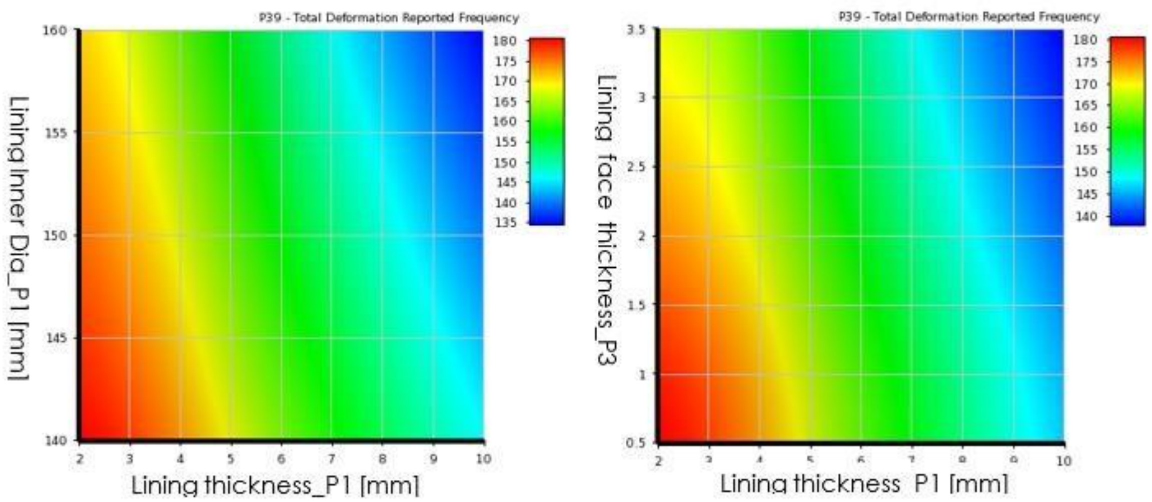

Figure IV-26Influence of parameters on output

From the above contour plots for output frequency vs Input Parameters, it can be seen that the at the lower bound of all the constraints g1, g2 &g3,the function is monotonically increasing and the constraints are active. The output frequency can be maximized further if the lower bound of constraints are relaxed.

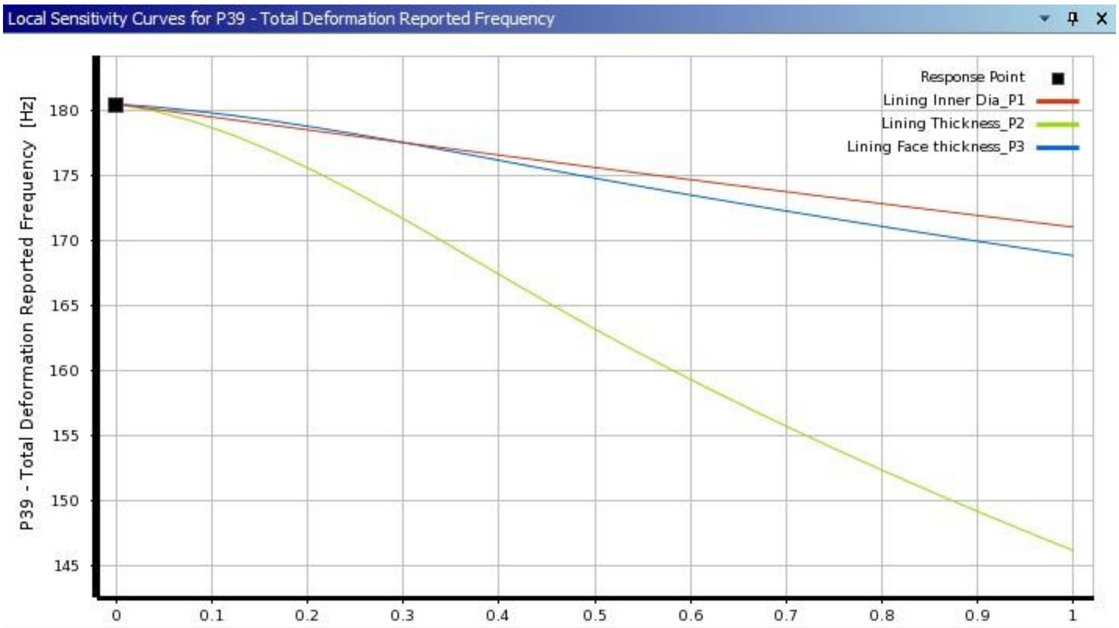

Figure IV-27Local Sensitivity Chart

Local Sensitivity Curve shows the impact of each input parameter on output.

Convergence Criteria shows the no. of Iterations required to achieve optimal solution

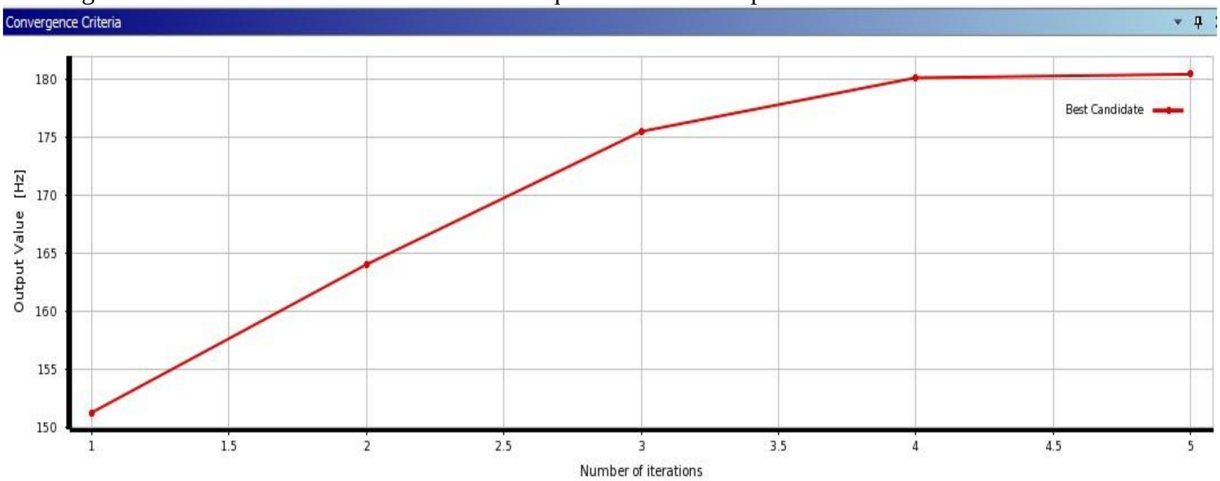

Figure IV-28Convergence Criteria

4.15.2 Thermal Analysis

The change in maximum temperature with respect to time on application of the heat flux and other loads is as shown below. The maximum temperature reached with the initial design at the end of 0.5 seconds, the slip time, is 109.9 degree Celsius

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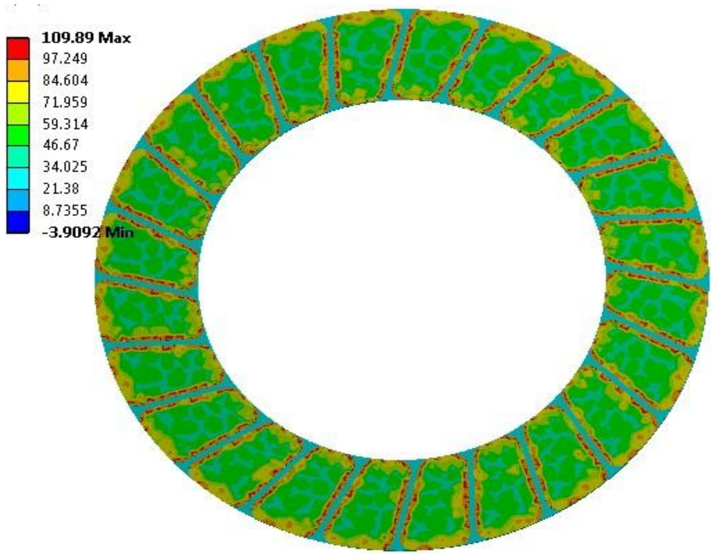

Figure IV-29Temperature distribution at the end of slip time

This output maximum temperature has to be minimized so as to avoid the failure of friction pad material due to repeated clutch engagements lowering its cooling period

4.15.2.1 Optimization Results

The NLPQL optimization is based on the response surface generated by regression analysis of design sample space as defined by below design points.

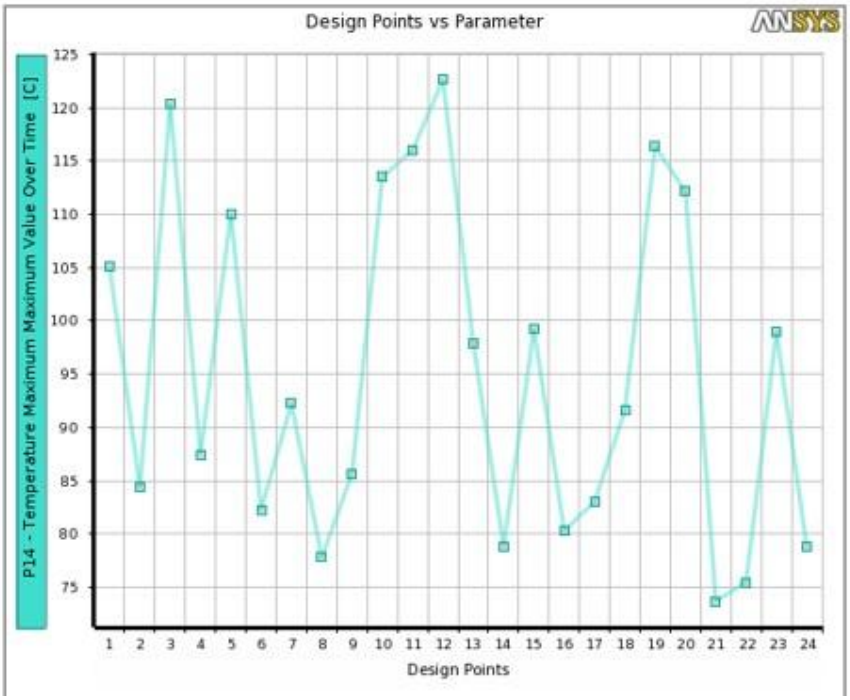

Figure IV-30 DOE Samples

With optimization, there is 31.3 % improvement in the output frequency which doesn't fall in engine frequency band.

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Figure IV-31Optimized Final Design

Table IV-5Optimized Input values

Parameter	Starting Point	Final Design	
P1 (mm)	160	150	
P2 (mm)	2.7	6	
P3 (mm)	0.8	3.5 (active)	
Output	Initial Value	Optimized Value	Simulated Value
Temperature(°C)	109.9	76.79	75.46

Robustness of Solution (Goodness of Fit)

Goodness of Fit shows that the output parameter has been very well approximated by the response surface .The coefficient of Determination is 0.92091

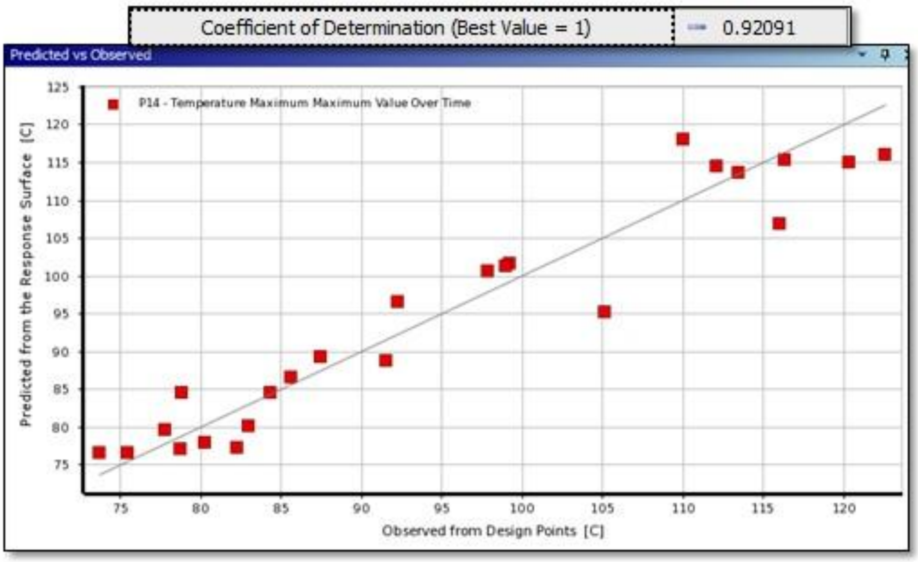

Figure IV-32Goodness of fit

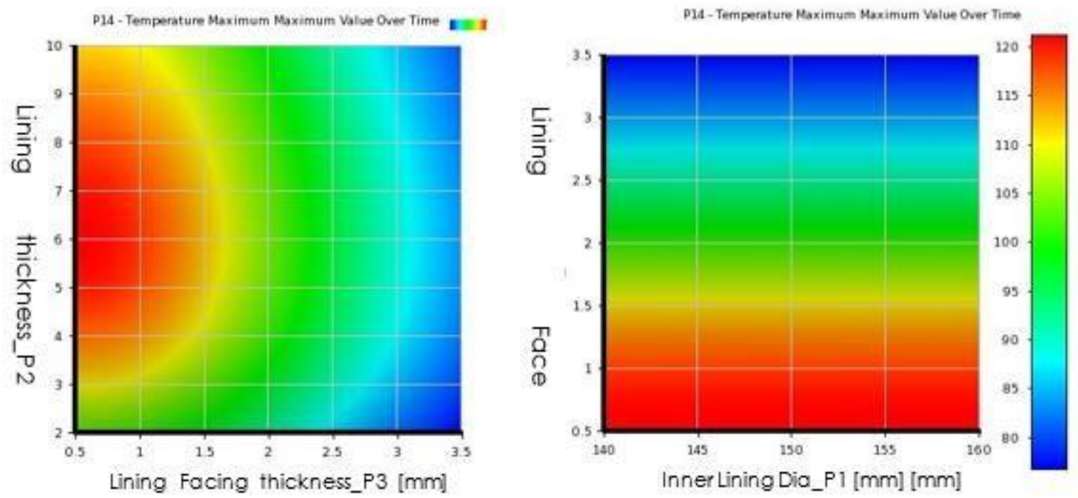

Figure IV-33Influence of Parameter values

From the above contour plots for Maximum temperature vs Input Parameters, it can be seen that at the upper bound of the constraints g3, the function is monotonically decreasing and the constraints is active from upper bound.

Local Sensitivity Curve shows the impact of each input parameter on output.

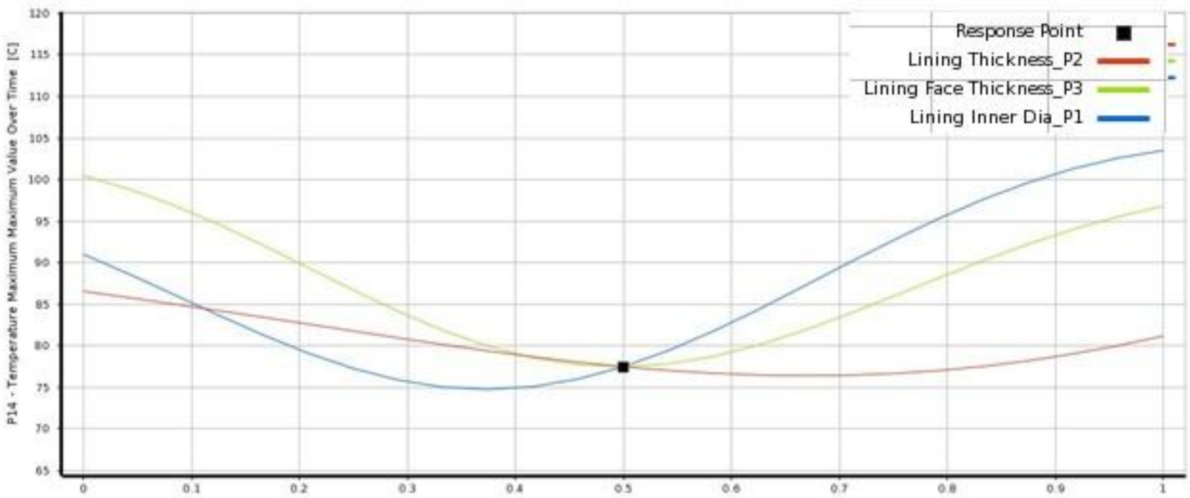

Figure IV-34 Sensitivity Chart

Convergence Criteria shows that two Iterations are required to achieve optimal temperature.

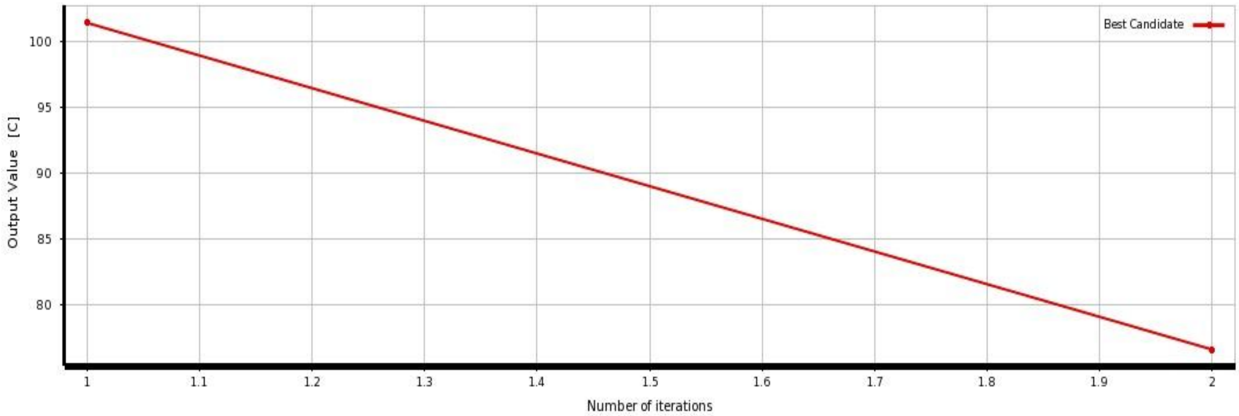

Figure IV-35Covergence Criteria

Structural Analysis

The Equivalent Stress reached with the initial design at the end of 0.5 seconds, the slip time, is 2.5452e5 Pa.

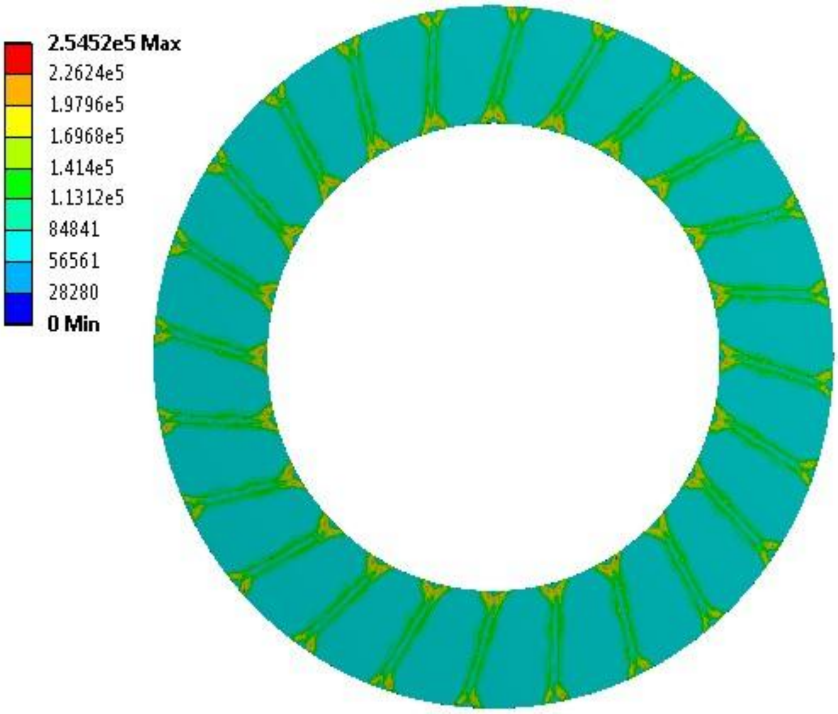

Figure IV-36Stress values at the end of slipping period

This equivalent stress has to be minimized so as to avoid clutch plate failure

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Optimization Results

The NLPQL optimization is based on the response surface generated by regression analysis of design sample space as defined by below 30 design points**.**

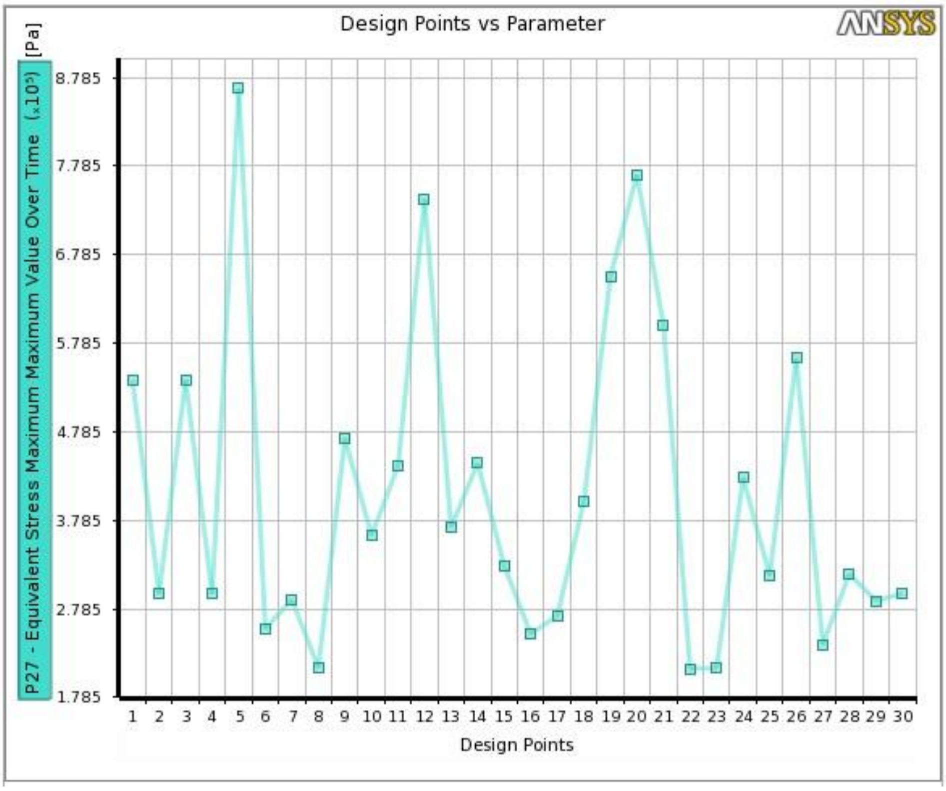

Figure IV-37 DOE Samples

With optimization, there is 27.61 % improvement in the output frequency which doesn’t fall in engine frequency band.

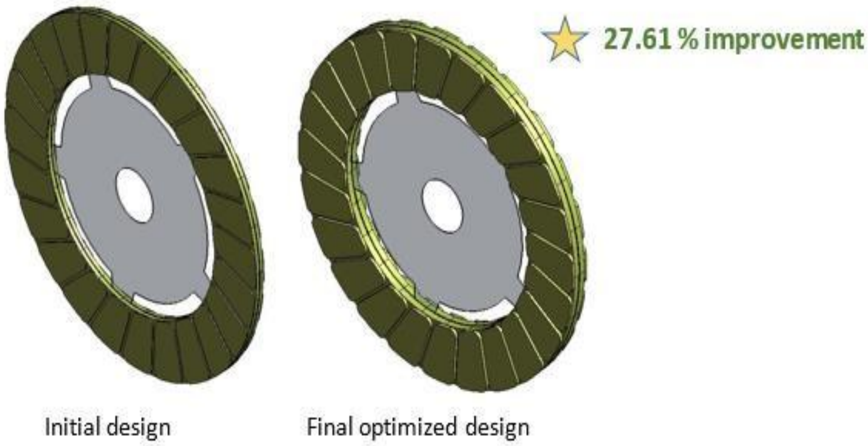

Figure IV-38Optimized Design

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Table IV-6Optimized Input parameter values

Parameter	Starting Point	Final Design	
P1 (mm)	160	160**(active)**	
P2 (mm)	2.7	10**(active)**	
P3 (mm)	0.8	2.52	
Output	**Initial Value**	**Optimized Value**	**Simulated Value**
Stress(MPa)	0.254	0.18	0.185

The above table shows the predicted value from NLPQL and observed value from ANSYS simulation are very close enough.

Robustness of Solution (Goodness of Fit)

Goodness of Fit shows that the output parameter has been very well approximated by the response surface.The coefficient of Determination is 0.98149

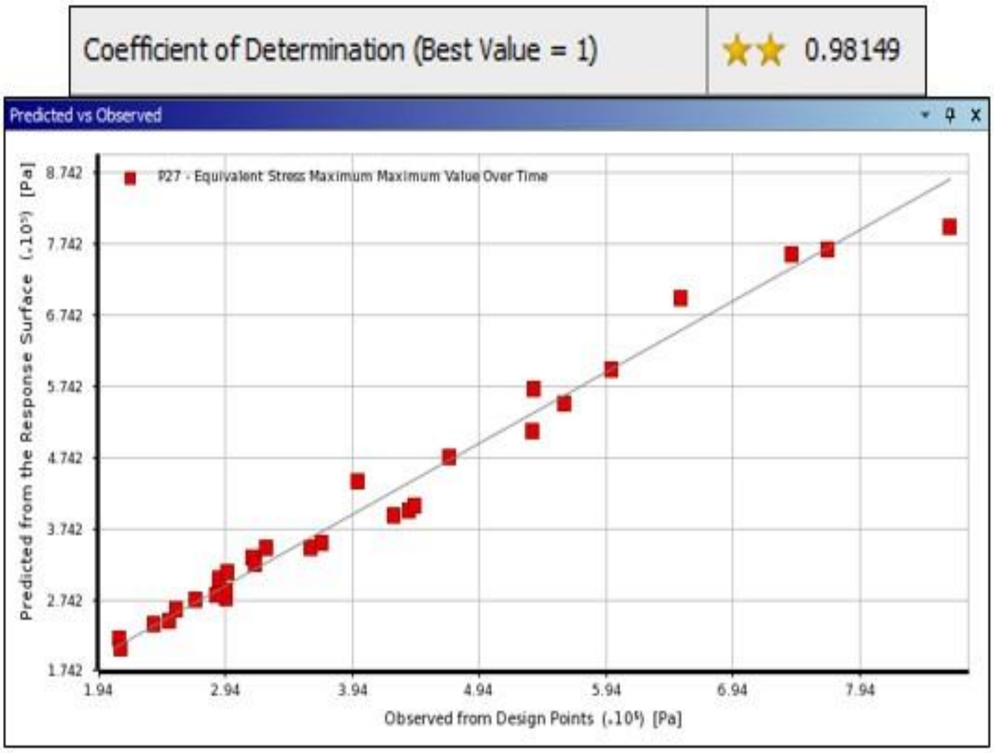

Figure IV-39 Goodness of Fit

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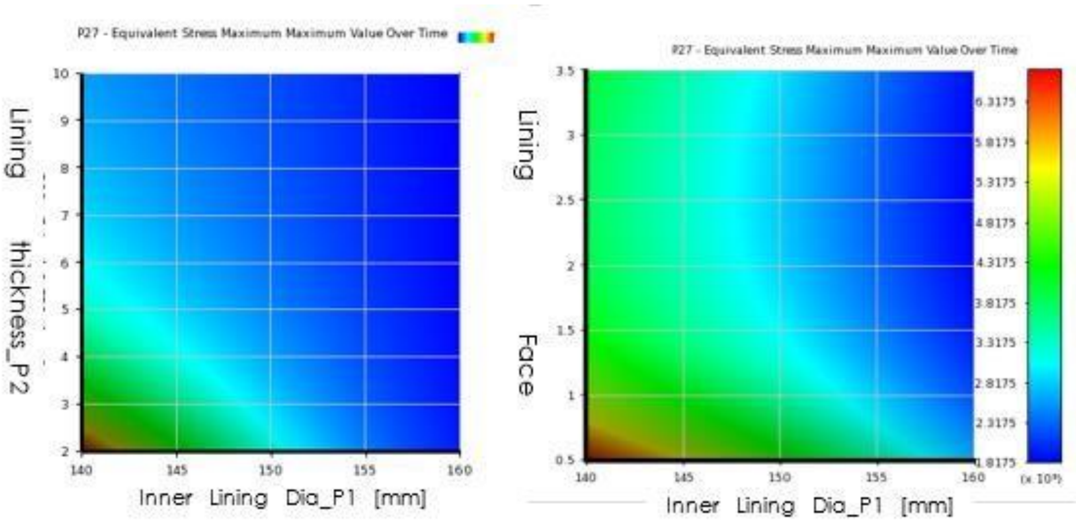

Figure IV-40Influence of Parameter values

From the above contour plots for Equivalent Stress vs Input Parameters, it can be seen that at the upper bound of the constraints g1 & g2, the function is monotonically decreasing and the constraints are active from upper bound.

Local Sensitivity Curve shows the impact of each input parameter on output

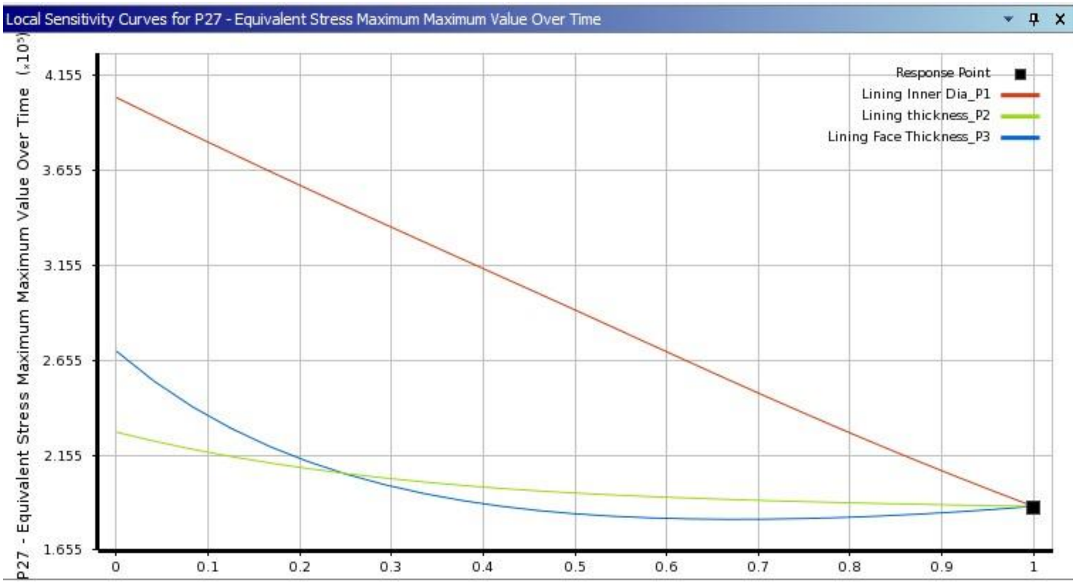

Figure IV-41 Sensitivity Chart

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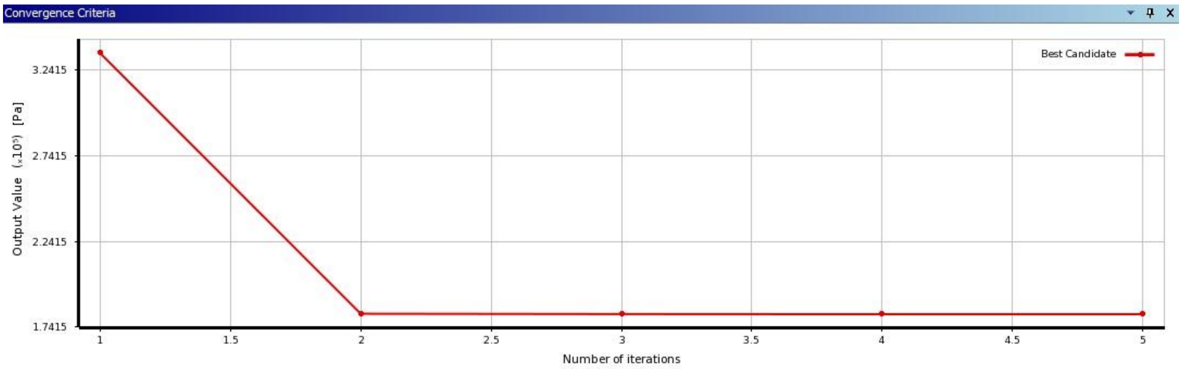


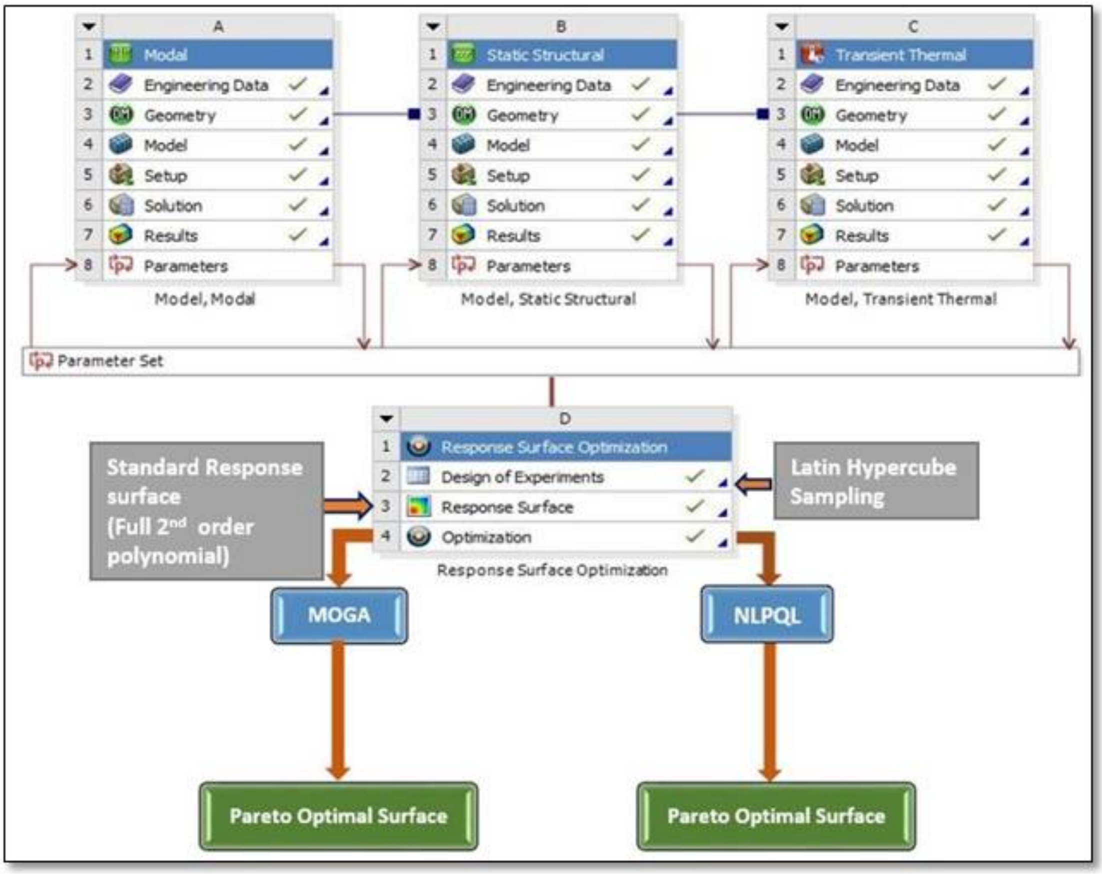
Figure IV-42 Convergence criteria

Convergence Criteria shows that five Iterations are required to achieve optimal temperature.

V. System Optimization

System optimization is basically the process of enhancing the capabilities of a system by integrating the subsystems of which the former is made to the extent that all of them operate above the user expectations. In the project of optimizing the clutch design, the optimized subsystems viz. Modal Analysis, Structural Analysis and Thermal Analysis are integrated to create an optimal Pareto surface. A Pareto surface is the surface containing optimal points corresponding to the optimal solution of a particular trade-off among the conflicting objectives of the subsystem. In other words, selecting one point from the Pareto surface will always sacrifice the quality for at least one objective, while improving the other objective. Here we have used two different methods to create a

Pareto optimal surface namely Multi-Objective Genetic Algorithm (MOGA) and NonLinear Programming by Quadratic Lagrangian (NLPQL).



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5.1 Multi-Objective Genetic Algorithm (MOGA)

MOGA is a hybrid variant of the popular NSGA-II (Non-dominated Sorted Genetic Algorithm). Only continuous problems can be solved using the same. The Algorithm goes through several iterations retaining the elite percentage of samples through each iteration and allowing the samples to evolve genetically until the best Pareto has been found. As mentioned above this method can handle multiple goals. Some of the other advantages are it helps identify the global and local minima of the function. It also provides several candidates in different regions giving accurate solutions.

5.1.1 Optimization Process

The following steps were taken to generate the Pareto optimal surface using MOGA.

1. Three subsystems were integrated with the common input parameters as design variables and the output as state variables.
2. Design of experiments was performed using Latin Hypercube Sampling (LHS)
3. A second order polynomial response surface was created from the DOE samples generated.
4. Optimal solutions were generated from the response surface using MOGA
5. The deviation in the predicted value from the MOGA optimization and the simulated value was corrected.
6. A Pareto optimal surface was generated using least squares regression analysis.

5.1.2 Results

The figure 47 shows the Pareto surface obtained from MOGA optimization. The nonpareto points are not considered for generating the surface.

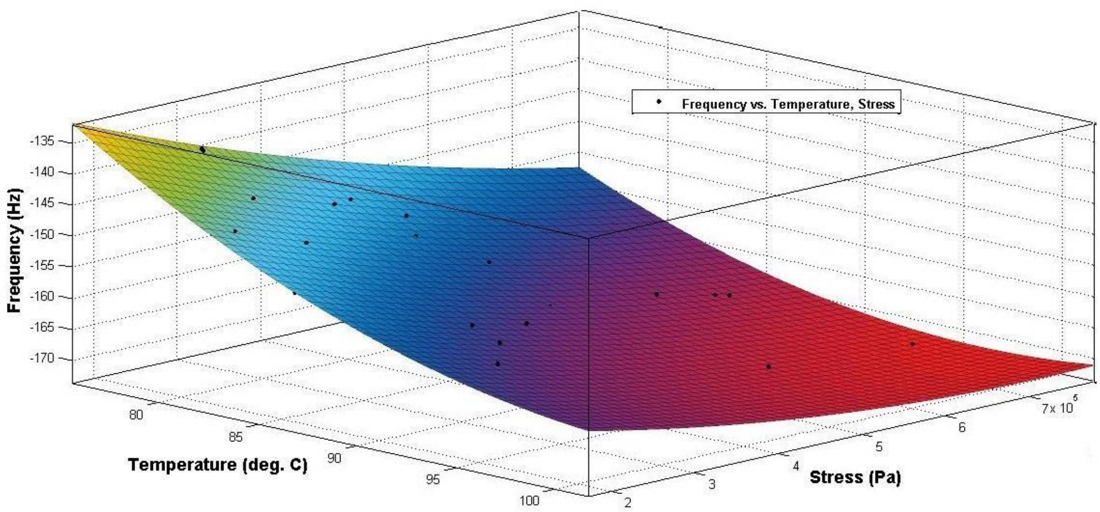

Figure V-1MOGA Pareto Surface

5.2 Non-Linear Programming by Quadratic Lagrangian (NLPQL)

NLPQL is a mathematical optimization Algorithm which solves non-linear programming problems. In this method the objective function and the constraints are assumed to be continuously differentiable. Here a sequence of QP sub problems are obtained by quadratic approximation of the Lagrangian function. The problem size cannot exceed 2000 input variables and has to be well scaled. Though the method is fast, the accuracy largely depends on the accuracy of the gradients. This is primarily used for single objective problems but can also be used for multi-objective problems by constraining the other output parameters.

5.2.1 Optimization Process

The following steps were taken to generate the Pareto optimal surface using NLPQL.

1. Three subsystems were integrated with the common input parameters as design variables and the output as state variables.
2. Design of experiments was performed using Latin Hypercube Sampling (LHS)
3. A second order polynomial response surface was created from the DOE samples generated.
4. Optimal solutions were generated by considering output frequency (maximize) from the modal analysis as objective and constraining the other two output parameters.

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5. The deviation in the predicted value from the NLPQL optimization and the simulated value was corrected.

6. A Pareto optimal surface was generated using least squares regression analysis.

5.2.2 Results

The figure 48 shows the Pareto surface obtained from NLPQL optimization. The nonpareto points are not considered for generating the surface.

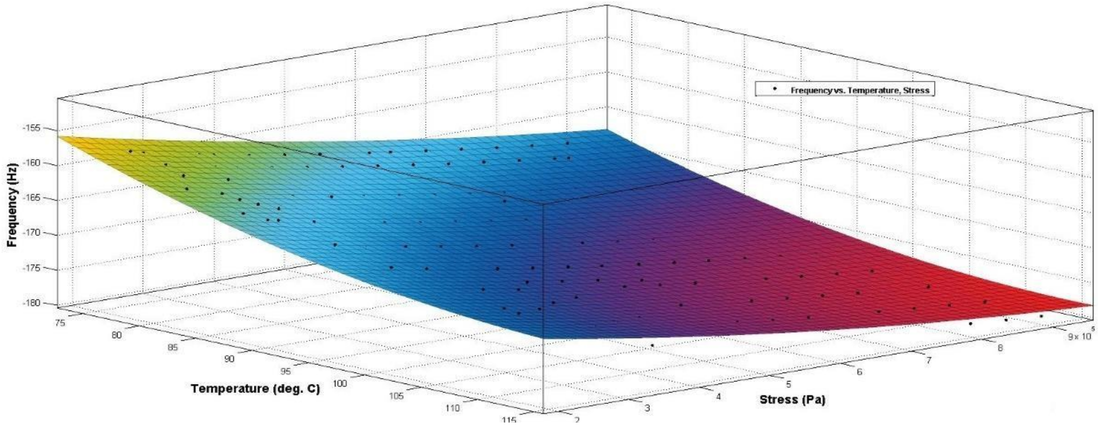

Figure V-2NLPQL Pareto Surface

5.3 Conclusion

The Pareto surfaces obtained from the optimization methods MOGA and NLPQL are very much comparable. There is no single point which serves as the best value for all objectives (Utopia point).To get an ideal point from the Pareto surface for a particular application the subsystem objectives need to be weighted. But in this project we have weighted all the objective equally.

VI. Work Plan

Sr. No.	Description	Duration	Month of working
1	Literature Survey and Problem Definition	Four Months	July to October
2	Design and ANSYS Tool	Two Months	November to December
3	Analysis, Conclusion and Future Scope	Four Months	January to April

Table VI-1 Time line

	July	August	September	October	November	December	January	February	March	April
Area Specification										
Research Material Collection										
Literature Survey										
Problem Definition										
Design										
Analysis										
ANSYS Tool										
Conclusion										
Future Scope										
Thesis Writing										

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