DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

DETERMINATION OF OIL PUMP COVERS' THICKNESS USING THE FINITE ELEMENT METHOD

by

Fatma Özge IRMAK

October, 2019

İZMİR

DETERMINATION OF OIL PUMP COVERS' THICKNESS USING THE FINITE ELEMENT METHOD

A Thesis Submitted to the
Graduate School of Natural and Applied Sciences of Dokuz Eylül University
In Partial Fulfillment of the Requirements for the Master of Science of
Department of Mechanical Engineering, Mechanics Program

by

Fatma Özge IRMAK

October, 2019

İZMİR

M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **“DETERMINATION OF OIL PUMP COVERS’ THICKNESS USING THE FINITE ELEMENT METHOD”** completed by **FATMA ÖZGE IRMAK** under supervision of **PROF. DR. EMİNE ÇINAR YENİ** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

Prof. Dr. Emine Çınar YENİ

Supervisor

Prof.Dr.Buket Okutan Baba

(Jury Member)

Prof.Dr. Binnur GÖREN KIRAL

(Jury Member)

Prof.Dr. Kadriye ERTEKİN

Director
Graduate School of Natural and Applied Sciences

ACKNOWLEDGEMENTS

Foremost; I would like to express my sincere gratitude to my supervisor Prof. Dr. Emine Çınar YENİ for the continous support of my Master of Science study and research, for her patience, motivation and immense knowledge. I have been extremely lucky to a have a supervisor who cared so much about my study and who responded to my questions and queries so promptly.

My sincere thanks also goes to Mr. Faruk Mehmet ELBEK, Engineering Manager at Stackpole International Fluid Power Solutions, İzmir and Mr. Levent KANTAR, Operations Manager at Stackpole International Fluid Power Solutions, İzmir for giving this research subject, their extensive personal and professional guiadance.

In particular, I must thank Mr. Abdalhameed Zubeidat, at Stackpole International, Fluid Power Solutions, Aachen supporting me about analysis and his great lead and infinite patience to my questions.

Besides, I also acknowledge Stackpole International Company for giving me the opportunity to do this study and also the rest of my managers, colleagues and friends. I am grateful and have a pleasure of being a member of this family.

Finally; I would like to thank my family, especially my parents, my ultimate role models, for their love, endless support, encouragement throughout my life wholeheartedly. I feel selfconfident and strong, based on their values giving me since my birth.

Fatma Özge IRMAK

DETERMINATION OF OIL PUMP COVERS' THICKNESS USING THE FINITE ELEMENT METHOD

ABSTRACT

The lubrication system is one of the most important systems that allows the engine to perform its task properly. Providing the necessary oil from the crankcase to the engine is maintaned by the oil pump. Oil pumps can be different types, volumes and have different operating systems. A hydraulic pump is the main part of the lubrication system and can be defined as a mechanical device that converts mechanical power into hydraulic energy such as flow, pressure. The two main parts that affect the weight of the oil pump are the housing and the cover. This study includes the thickness determination of the oil pumps' covers using the finite element method.

The aim of the cover thickness determination is to reduce the fuel consumption with lighter component in any auto, to contribute the environmental regulations and to increase the competitiveness power of any automotive company by reducing the raw material costs.

Keywords: Oil pumps, variable vane oil pumps, finite element analysis, computer aided design and computer aided analysis

YAĞ POMPASI KAPAKLARININ KALINLIĞININ SONLU ELEMANLAR YÖNTEMİ İLE BELİRLENMESİ

ÖZ

Yağlama sistemi bir motorun görevini düzgün bir şekilde yerine getirmesini sağlayan en önemli sistemlerden biridir. Motordan kartere süzülen yağın tekrar motora sağlanması yağ pompası aracılığı ile olur. Yağ pompaları farklı modellere, hacimlere ve çalışma sistemine sahiptirler. Hidrolik yağ pompası, mekanik enerjiyi, basınç ve akış gibi hidrolik enerjiye dönüştüren, yağlama sisteminin ana bileşenidir. Yağ pompasının ağırlığına etki eden iki temel parçası gövde ve kapaktır. Bu çalışma bir yağ pompası modelinde kullanılan kapağın sonlu eleman analizi yöntemi ile kalınlığının belirlenmesini içermektedir.

Kapak kalınlığı belirlenmesi çalışmasının amacı, herhangi bir araçta daha hafif komponentler ile yakıt tüketimini azaltmak, çevresel düzenlemelere katkıda bulunmak ve ham madde maliyetlerinnde azalma sağlayarak, endüstride rekabetçi yapıyı arttırmaktır.

Anahtar sözcükler: Yağ pompaları, değişken hacimli yağ pompası, sonlu eleman analizi, bilgisayar destekli tasarım ve bilgisayar destekli analiz

CONTENTS

LIST OF FIGURES

Page

LIST OF TABLES

CHAPTER ONE

INTRODUCTION

1.1 World History of Automotive Industry

The development belonging to the automotive industry took place over a period of more than 200 years. In the 1770’s, prototype vehicles were produced with steam engines and these vehicles were followed by electrically driven vehicles in the 1980’s. However, the vehicles operating with these technologies were quite heavy, they moved slowly and their costs were very high; therefore the steam and industrial vehicles were replaced by the vehicles that were working with gasoline in a short time. Steam technology had been used in locomotives, whereas electric technology had been used in trams. The first patent for gasoline-powered cars were taken in 1886 by Karl Benz (Teker & Felekoğlu, 2007).

It was assumed that automotive industry was established in the early 1900’s after the first patent was taken. (Görener & Görener, 2008) In this industry, serial production in the US by Ford since 1910 has continued its effects on the US and foreign markets and has developed in an environment where there is a wide market, cheap and unskilled labor. In 1960’s the US had 54% and Europe had 41% of the sector whereas in 1990’s the US had 22%, Europe had 39%, Japan and Korea combined had 28% of the sector (Teker & Felekoğlu, 2007).

The crisis experienced in the 1970’s has played a vital role to affect the US and European producers’ competition power. In these years, Japan has reached at the top of the international competition by providing mass production, cost and quality improvements. The understanding of lean manufacturing combined the understanding of productivity of the scale economy that prevailed all over the world until the 1970’s, and the flexibility understanding of workshop type production. Lean manufacturing launched by Toyota in Japan in the 1970’s has been an important revolution in the automotive industry. Thus, this understanding has reduced the manufacturing costs while increased the creativity and employee commitment to the business with the applications such as "kaizen", "just-in -time manufacturing" and "six sigma". This success of Japan has caused a big shock in US and European producers (Teker & Felekoğlu, 2007).

In the 1980's "CIM", "CAM" and "CAD" systems have taken a place in the automotive manufacturing sector. The world automotive sector entered into a restructuring process with a certain saturation of the large markets in Japan, North America and Europe, and the fact that the Japanese gained more production advantage as a result of their development in design and production techniques and developed new technologies (Teker & Felekoğlu, 2007).

In the 1990's, Japan's automotive industry product exports were decreased. As a reason of this, the factors such as completion of Japan's investments in other countries and commencement of production, reduction in export markets, appreciation of Japanese yen, increasing labor cost after 1980's in Japan, breaking of Japanese' competitive power with developed lean manufacturing philosophy in Europe and United states can be shown. (Görener & Görener, 2008) In 1995 production rates changed as 21% of the US, 36% of Europe, %26 of Japan and Korea (Teker & Felekoğlu, 2007).

In 2000's, the production rate of England decreased, however the production rate of Spain, Mexico and Middle East European countries have increased (Yayar & Yılmaz, 2016).

In 2013, 85 billion euro investment expenditure has been done in the sector in terms of production and research-development (R&D), and tax income has been provided in the invested countries over 433 billion euros. It approximately possesses a 2 trillion euros financial turnover and has direct jobs over 8 million. This figure is estimated to be over 50 million with indirect employment. There are approximately 50 motor vehicle producer for the automotive industry around the world (Yayar & Yılmaz, 2016).

In last years, India and China had significant improvements also. According to the McKinsey Reports, in Europe, the automotive industry is in charge of roundly 12 million jobs; in the U.S. over 8 million and also in Japan over 5 millions (Mckinsey, 2017).

At the present time automotive production spreads from countries that have this industry to other developing countries. The main purpose of this expansion is to obtain a higher market share by producing in more competitive conditions and closer to the market (Yayar & Yılmaz, 2016). Figure 1.1 obtained from Organisation Internationale des Constructeurs d'Automobiles (OICA) shows the vehicles sales between the years 2015-2017. When the number of sales is examined, it is observed that there is a steady growth since 2010.

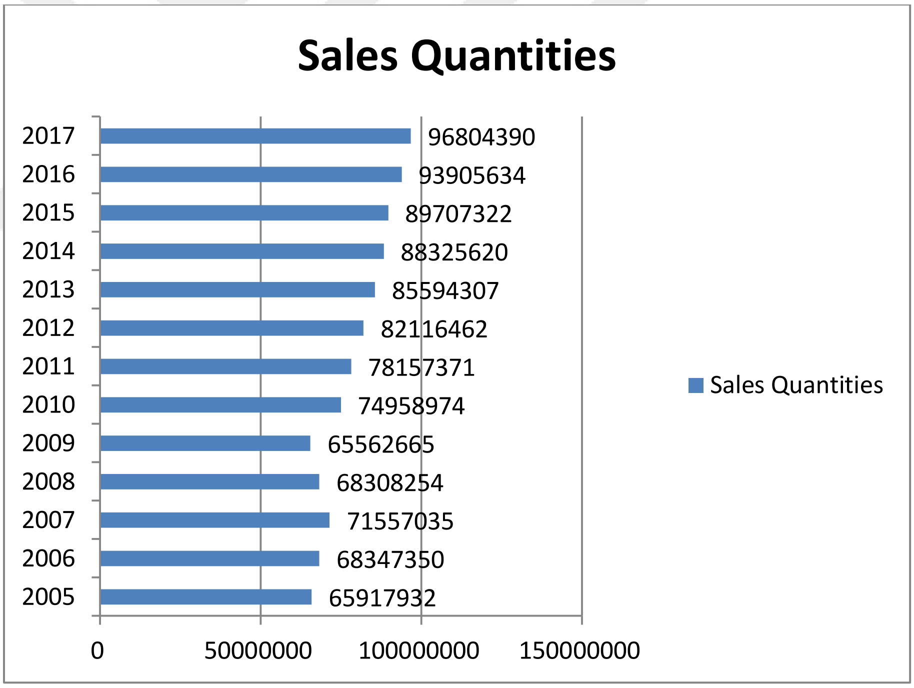

Figure 1.1 Sales quantities (The International Organization of Motor Vehicle Manufactures [OICA], 2018)

The distribution of world vehicle production by regions is analyzed and presented in Figure 1.2

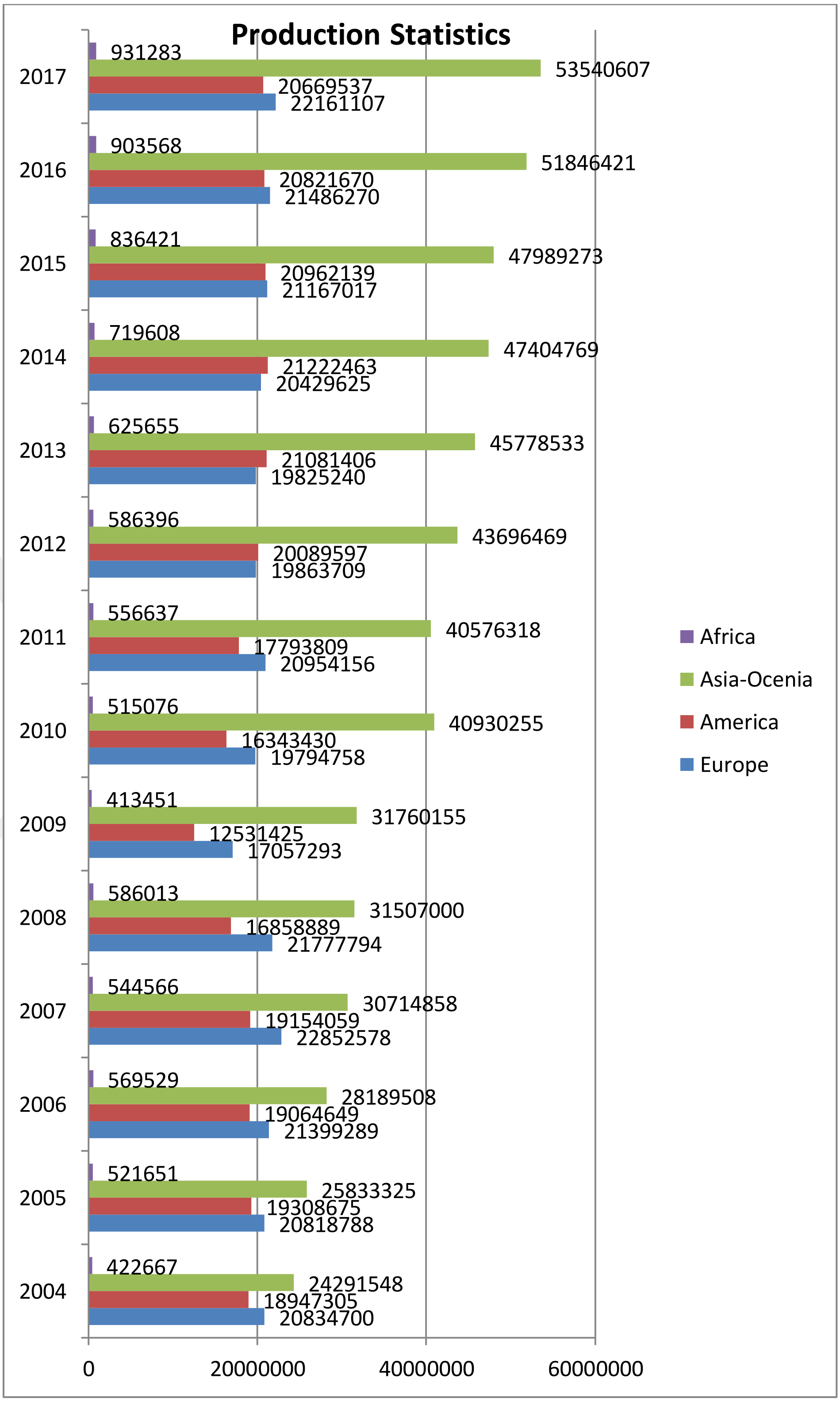

Figure 1.2 Motor vehicle production statistics in the world (OICA, 2018)

It is seen that Europe and America show an almost stable situation; whereas production volumes in Asia-Ocenia and Africa have increased significantly. This table also proves that the sector investments are shifted to regions that have cheap labor cost. According to the OICA reports; Asia-Ocenia is the strongest region having almost half of export & import activities of the world.

Today, intense interest, increasing demand in the sector and close relationship with technological development has made the automotive sector one of the most invested sectors in the world.

Figure 1.3 gives the 2017 production rates based on countries and shows the distribution in detail throughout the world.

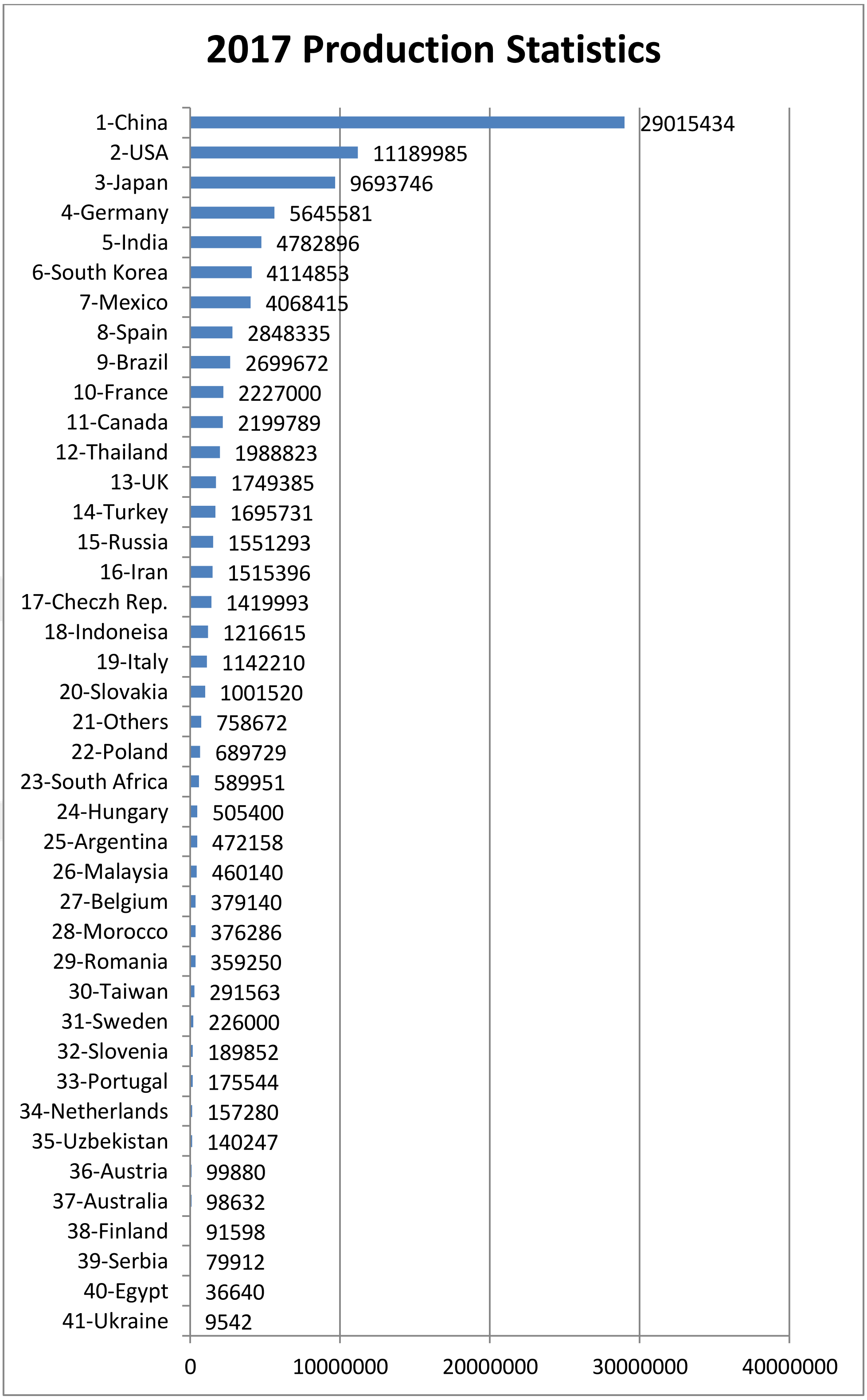

Figure 1.3 2017 Production rates based on countries (OICA, 2018)

1.2 Turkey History of Automotive Industry

Turkey's first acquaintance with the car and the beginning of car production took place in the early 20th century after the First World War, when Istanbul was under occupation. Ford and Chevrolet brand cars and trucks of the USA, have entered in Turkish market by American Foreign Trade company. In the same years, Fiat brand automabiles of Italy has entered to Turkish market with a special office in Istanbul that works as a branch with its center in Torino (Yayar & Yılmaz, 2016).

In 1929, Ford Motor Company started its first assembly line trials in İstanbul. It has been foreseen that some of the cars, trucks and tractors that would be produced in the first assembly plant were to be exported to the Soviet Union. This first assembly plant produced 48 cars and trucks per day and employed 450 employees with technological infrastructure. But the targeted export could not be realized due to negative effects of the World Economic Crisis in the 1930's. However due to other reasons, in 1934 the factory stopped production and thus, the first assembly production attempt has failed (Yayar &Yılmaz, 2016).

In the mid 1940's, the need for transportation increased and the development of the automotive sector was further triggered with the acceleration of urbanization. Untill 1950's, while government incentives for investment were at the forefront, private sector investments started after the 1950's (Yayar & Yılmaz, 2016).

The automotive industry had a close relationship with the EU automotive industry, since 1960. In the 1970's, production in the Turkish automotive industry was started after gaining license "technical cooperation". In the mid-1980's, this cooperation was transformed into "economic cooperation" with increasing foreign capital participation (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

At the Turkey-EU Partnership Council meeting on 6 March 1995, Turkey signed the treaty on the accession to the European Union. After this, Turkey entered the Customs Union on January 1, 1996 (Görener & Görener, 2008). With the launch of the Custom Union Process, the automotive sector has undergone radical changes for the transition to a fully competitive environment and has gained a competitive "export-oriented" status with the encouragement of investments towards new and current model vehicle production. The cooperation with global firms that already gone in for production in Turkey; and the realization of intensive integration between partners in Turkey have started this process and developed (Tübitak, 2017). As a result of this; foreign investors/shareholders has incorporated the facilities in Turkey to their own global strategic development projects. The facilities in Turkey have reached the stage in terms of quality and cost/efficiency that can make a production to global markets and this is called as “full integration”. The technology and research & development potential of global companies has been used by the Turkish automotive industry also. In order to ensure the sustainable competitiveness in this industry, this production capability should continue by strengthening; in parallel, the studies in the fields of research and development are required. The goal of this competency is; to have proprietary technology to develop products with the intellectual property rights in Turkey (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

Automotive industry in the last 10 years has become one of the leading sectors of both Turkish and global economy due to extensive new model investments and export projects. Turkish automotive sector is progressing rapidly to become one of the production centers of Europe. The sector is associated at the rate of eighty-five percent in production with automotive industry of Europen Union. Considering this data, it is obviously clear that the sector has many potential advantages and high integration with global markets; especially in Europe. The main strategy can be defined as, becoming higher value added, competitive, open for sustainable improvement and having advanced research & development (R&D) investment with qualified human resource. This strategy covers all processes from production based understanding to center of excellence understanding (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

Automotive industry has had significant improvements since the 1960's both in Turkey and all over the world. Todays; global integration has been almost accomplished in production and marketing sides. Automotive industry has proved the qualification and quality management and efficiency with the exports that have been made to the global markets. Some of the companies operating in the automobile industry all over the world; participated as a co-designer at vehicle manufacturing phases in Turkey. It is foreseen that Turkish firms will become licence sellers in the near future with many years of experience and know-how. Because of the the geoeconomic and geopolitical situations of Turkey, the automotive supply industry is seen as a significant potential investment area by the global automotive firms (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

1.3. General Aspect of Auto Industry

The auto industry is one of the biggest and of the leading sectors of the economy in all developed states. The reason for the driving force in the economy is due to its close proximity to other sectors of the economy. This industry is the main buyer and developer of other major industries such as rubber and petrochemical products. All kinds of vehicles that are used in the sectors such as tourism, infrastructure, construction, transportation etc. are supplied by this industry. Therefore, this sector plays a vital role and changes in this sector directly affect the whole economy (Görener, & Görener, 2008). Competitiveness is the key element of a healthy economy. The competitiveness of the economy depends on increasing productivity and innovation power. Innovation and productivity is a result of technological advances. The automotive suppliers industry possesses the characteristic of enhancing the competitiveness of the national economy with its continuous investments, technical employment creation and the ability to spread technical culture to other sectors and society (Tübitak, 2017).

The competition ability of the automotive companies in both local and international markets depend on the sale of renewed vehicles at attractive prices. Productivity and labor cost are the main input for the short term activities despite the fact that R&D investments and innovative competition are the main input for the long term strategies. Parallel to the increase of innovative activities in automotive main and subsidiary industries; the demand for knowledgeable and educated workforce has also increased (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

The value and importance of knowledge is increasing. And also creating difference and innovation have been notable elements in competition. In the forthcoming period developments in science and technology and production based on knowledge will keep going to be key determinants of growth. Therefore; some technological investments and research and development (R&D) activities are being developed not only by the free market mechanism, but also by the directive regulatory and supportive approaches of the public. However, many R&D studies are of the international quality and mostly carried out by large global companies. Developed economies maintain their leadership in most sectors with high added value as the center of R&D and innovation, while losing their competitive edge relative to labor cost sensitive sectors. Developing countries, especially China, India and Brasil are also in the process of knowledge-based production. In the coming period, technological developments are expected to shape economic, social and military developments by focusing on certain areas (Strateji ve Bütçe Başkanlığı, 2013).

The automotive industry is the second complex and multi-disciplinary technology that includes specific and key engineering studies after the aerospace industry (Strateji ve Bütçe Başkanlığı, 2013). As of 2016, the automotive industry has the 4th largest economy in the world (4 trillion dollars) (Otomotiv Sanayi Derneği [OSD], 2019). The quantities of global vehicle production and sales proves this clearly as given in Figure 1.4.

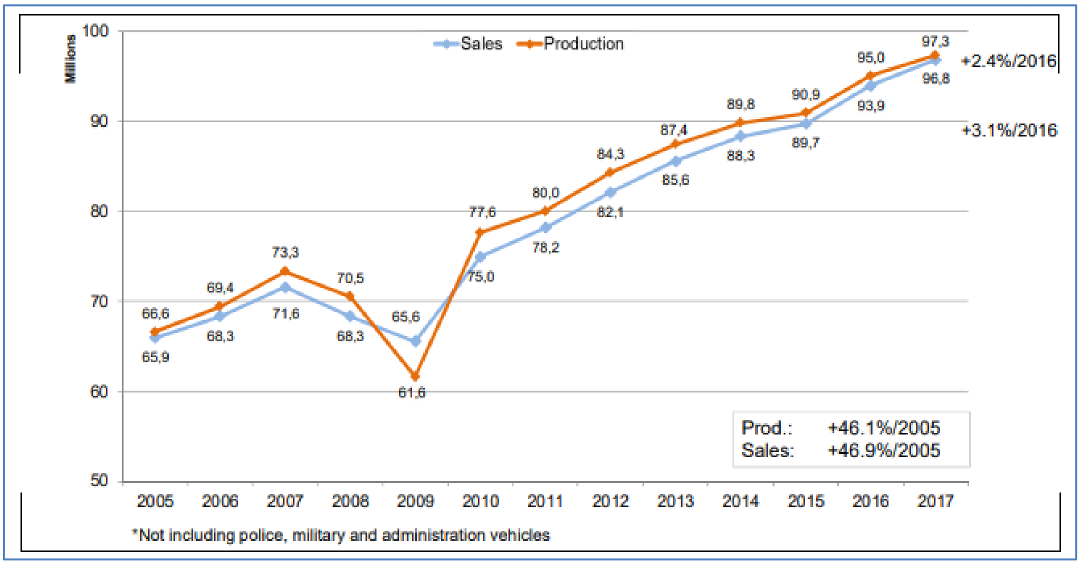

Figure1.4 Global vehicle production and sales (OICA, 2018)

A motor vehicle is produced with combining approximately 5000 pieces with each piece having different material, process, technology and production locations with the common quality management and efficiency approach (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

Each company has to produce motor vehicles in compliance with approximately 50 global technical regulations on safety, traffic and the environment and each one has to be documented. There are also about 100 other international legislations that can be applied on a voluntary basis. These legislations are constantly being renewed on the basis of the developments of the technology and especially the preparations that are related to the environment are putting great pressure on the sector (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

Besides this, The Kyoto Protocol puts certain boundary to reduce the carbon levels of atmosphere that were measured in the 1990's (T.C. Dışişleri Bakanlığı, 2018). The industry is committed to contributing to reducing carbon emissions in order to create a cleaner environment together with other stakeholders. Automobile manufacturers constantly develop engines and drivetrains with more efficiency to make better fuel consumption; and cleaner cars are available on the roads thanks to a wide range of complex emission control technologies. More vehicles with modern exhaust emission performance can provide clear improvements in air quality. This improvement will continue with the replacement of older and more polluting vehicles (OICA, 2018). Safety, comfort, communication, quality, consumption and driving pleasure are important issues in the automotive market today and in the future. The priority issues that will be important in achieving success in the competitive environment for Turkish automotive manufacturers have a strong position in the domestic market and to maintain it, they have to establish good relations with suppliers, adopt concurrent engineering, reduce the cost reduction measures in all processes from the design of the product to the customer and to improve the brand image (Teker & Felekoğlu, 2007).

In the automotive industry, parts and components such as distributors, brakes, steering wheels, tires, rims, seat covers, windshield wipers, exhaust, airbag, trip computer are manufactured by a number of independent enterprises called “automotive supply industries.” Companies that supply goods to the main industry directly, which is called Tier 1, will work with Tier 2 that supplies good to Tier 1, suppliers by considering product quality and financing responsibilities with the development of the supply chain (İstanbul Sanayi Odası Sektör Raporu 2007, 2017).

The individual requirements of customers are a triggering factor. On issues such as comfort, prestige, safety, communication and individuality, functionality leads to a wide variety of innovations in vehicle production. (Tübitak, 2017) Customer satisfaction can be achieved through technological development with the intense competition in the market. For this reason, intensive research and development is essential in the sector (Istanbul Sanayi Odası Sektör Raporu 2007, 2017). Especially, today the changing customer demands lead the direction of technological developments and the topics focused on and the idea of transforming the automotive interior into a “living space” has been adopted all over the world. The new and developing technologies such as virtual screen technologies, driver warning systems, vehicle safety systems, haptic sensors, ambitious innovations for entertainment, increased comfort on the interior surfaces of the cars, 3D laminated glass are accepted by the traditional suppliers. So this situation has reshaped the automotive industry structures and relationships and the balance between OEM's (Original Equipment Manufacturer) and Tier 1 & Tier 2 suppliers has changed (2017 Automotive Industry Trends, 2017).

The rising cost is also a concern for the automotive industry. Given the pressure on costs, there is no easy way to improve capital returns for OEMs or suppliers. For this reason, the companies in the automotive sector aim to survive with various strategies and maintain their profitability. Auto makers should take into account the strategies such as redesigning distribution models, manufacturing and sharing platforms, offloading more development work for technology suppliers (2017 Automotive Industry Trends, 2017).

1.3.1 Redesigning Distribution Models

There is a general consensus that 15% of the cost of an automobile costs are logistical costs, although it may vary by country and region. For this reason; OEM's particularly in USA and Europe in order to reach automobile sellers by using efficient ways, started looking for ways to lower the costs and lobbying. With these changes in the distribution system it should be aimed to minimize retail point quantity and cost, and to reduce costs using technology for better inventory control (2017 Automotive Industry Trends, 2017).

1.3.2 Sharing Platforms and Manufacturing

The main objective is to improve the efficiency of capital expenditures. Chassis and powertrain investments can be convenient areas for this. Since each company makes a design of its own engines, transmissions and related equipment and manufactures them, this situation multiplies the processes and creates a wasteful capacity. Because end users focus on features such as reliability, quality and style etc. rather than buying cars for the platform. Most OEMs built platforms called as "repurpose" to reduce the costs. For example, Nissan uses "*Modular Front Architecture*" platform for some models. Ford and GM designed a new transmission.

In these two examples; the cost reduction was aimed especially in research & development and material supply (2017 Automotive Industry Trends, 2017).

1.3.3 Offloading More Development Work to Technology Suppliers

Almost all automakers engage in design of components such as digital components, electrification, autonomous features, software and the new technologies demanded by the customers. Giving more responsibilities to the suppliers and the shareholders of any project contribute to create their own know-how and add value to products that they produce. This increases the exchange of the information and makes the producers experts in their profession. Improving processes reduce the costs, rate of scraps and other cost items Tübitak, 2017).

All over the world, various companies have different strategies. For example, hinge manufacturer Edscha works to produce standard parts and achieve market share through cost reduction strategy. Brake manufacturer Brembo works with specific customers. This company has become profitable by focusing on customer-specific solutions. The company Gentex, which is a pioneer in the production of automatic anti-glare built-in mirrors, and Luk that manufactures the chain ring for the continuous-speed technology gears, follow the same strategy. Trico, a windshield wiper company, produces pioneering technological products with continuous innovation management. The manufacturer of exhaust systems, Zeuna-Staerker, integrates parts into customer solutions in a module or system and optimally stabilizes the functionality of its systems with its own module. An approach that predicts the probable change in vehicle design with a proactive perspective brings the success. Continental which operates in the chassis sector, has completed its product portfolio with purchasing companies (Tübitak, 2017).

1.4 Todays Research & Development Issues (Technological Trends)

The development of technological infrastructure and product development has become a necessity in terms of maintaining the lives of sub-industry firms. The success of automotive sub-industry firms will be largely determined by the ability to design jointly and make a difference in production. For this purpose, the main and side potential forces must be combined in R & D studies and to develop special products by specialization of sub-industry.

The companies that manufacture products to automotive industry focus on three main subjects on their research and development studies:

- Performance and safety
- New production technologies
- Lightened vehicles

While fuel prices are increasing worldwide; efforts to reduce carbon emissions have also accelerated. Therefore, to lighten the vehicles has become a common target of the automotive sector. In a 100 kg lighter vehicle, fuel consumption is about 0.8 liters less at 100 km. For this reason, new material concepts such as reinforced steel, metal foam, magnesium, ceramic products, aluminum etc. are needed. For example, VW uses laser welding instead of traditional spot welding operations. With a rational approach, more stability is achieved by using shaped bodies and laser welding, using less material than point welding (Tübitak, 2017).

This study includes the thickness optimization of the oil pumps' covers using the FEM. Therefore, the aim is the thickness optimization to reduce the fuel consumption with lightened component in any auto, to contribute the environmental regulations and to increase the competitiveness power of the company by reducing the raw material costs.

CHAPTER TWO

LITERATURE REVIEW

Reddy and Nagaraju (2017) studied to reduce the weight of drive shafts and compared the performance of the types of composite and steel drive shaft in their article. It was noted that the weight reduction of the drive shaft can play a certain role in the overall weight reduction of the vehicle and was highly desirable if it could be achieved without an increase in cost and without a decrease in quality and reliability. This study included the performance of the composite drive shaft for vans, small trucks and passenger cars, optimum stacking order, material selection and design. It was also aimed to propose the suitability of composites in the automotive industry. Dimensional buckling, free vibration and Static analysis were performed and found that the steel drive shaft had a 97% chance of weight reduction. Boron / Epoxy, Carbon / Epoxy and E-Glass / Epoxy are used in the design. Analysis and modeling were performed using Ansys. When the results were compared based on dimensional buckling analysis, it was found that among these three composite materials, Boron Epoxy performed well showing that buckling torques were higher than operating torques (3500 NM) even at minimum wall thickness. Results compared to weight savings; Among these three composite materials, Carbon / Epoxy was found to be less dense than the other two. As a result, it has been found that the savings achieved when the drive shaft is produced with Boron / Epoxy are greater than 97%. When the drive shaft is produced with Carbon / Epoxy, it is found that the weight saving is greater than 97% compared to steel. When the drive shaft is manufactured with E-Glass / Epoxy, the weight saving is more than 95%.

Yanhong and Feng (2011) studied the fatigue crack occurred in the Dump Truck’s (YJ3128-type) Sub-frames worked in unsuitable condition. The truck’s load features and the operating conditions were researched. And also in order to analyze the stress of the sub-frame, Ansys software has been used. The manufacturer found that within 3-5 months of the frame being used, cracking occurred on the right side of the square beams and at the turning point of the left beam. The safety factor for poor working environments was taken as 1.2 and the material stress that could be used under such conditions was found to be 287.5 MPa. Due to the fact that YJ3128-Type dump truck was used in the field operation places, this situation caused the sub frame's fatigue failure. During the operation; each part of the sub-frame is subjected to different loads. All these loads were specified in the article with a table. After analysis in the Ansys; it has been seen that three relatively large stresses have occurred. One of them was in the final end of frame and exceeded the allowable stress of the material. The other two stresses were less; the allowable stress and these stresses located near rear wheel and at the turning point of the stringer because of fatigue failure.

Figure 2.1The fatigue crack of the sub-frame (Yanhong, & Feng, 2011)

The cracks were occurred due to lack of torsional stiffness of the sub-frame. Therefore, a beam was added to the font of the subframe and then the finite element analysis was repeated. As a result, maximum Von-Mises stress was achieved under the permissible stress of the material and the other two stresses were 50% of the permissible stress. Large turning transformations and excessive stress concentrations were not observed. It has assumed that the study made reasonable recommendations.

Silori, Shaikh, Kumar and Tandon (2015) studied mechanical behavior of traction/ spur gear assembly for different materials, estimation of the fatigue and the stress behavior at the root radii of a gear in their article. If fatigue life is estimated; fatigue failure can be prevented without part damage, components or system and components can be exchanged with the new ones. Therefore, the authors stressed that designers should have good empirical and analytical knowledge to prevent failure. The aim of this study was to estimate contact fatigue cracking caused by high stresses or strains during the meshing process. For this study; AISI 4140 Alloy Steel and Ti6242S materials are used to see the mechanical behavior and performance of spur gears.

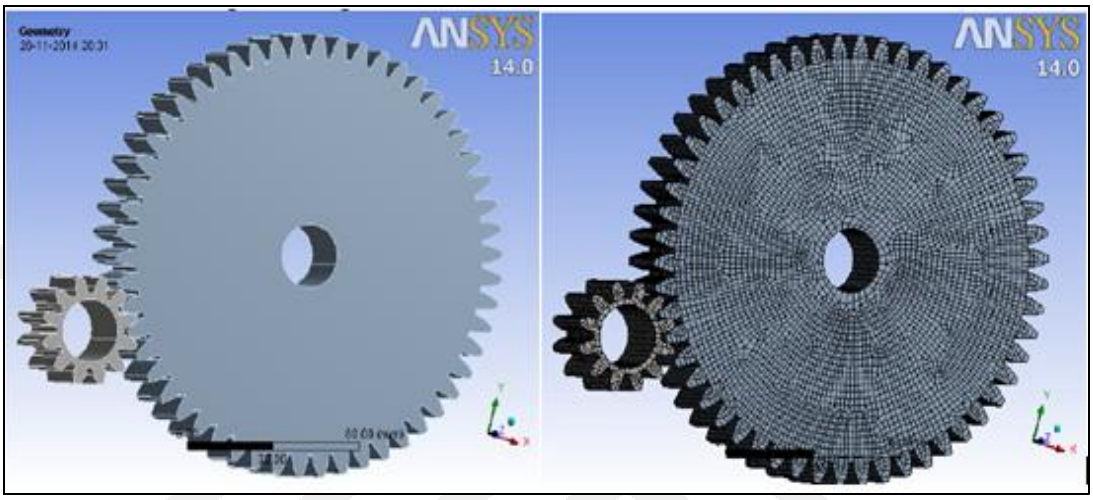

Figure 2.2 Spur/traction gear assembly and meshed model (Silori, Shaikh, KC & Tandon, 2015)

All boundary conditions and necessary loads were applied and assumptions were made for the analysis. According to the results, AISI 4041 Alloy Steel with root radius of 0.8 mm and 1.4 mm was the best for the design. Ti6242S alloy with root radius of 1.0 mm was best for fatigue life, von-Mises stresses and total deformation. Comparing with the AISI 4140 and Ti6242S materials, AISI 4041 Alloy Steel showed good bending resistance on the other hand, Ti6242S alloys were more advantageous for fatigue life and fail safe design.

Ansys Workbench was used as a finite element simulation software for static structure and modal analysis. Face sheets and core were put together to obtain a very efficient load bearing structure. The model has been created according to a three-point test methods Then; 8 noded brick element has been used for all composite structures. This type of definition is called Solid 185 in Ansys. Density, Poisson’s ratio, shear modulus and Young’s modulus were defined for both face sheets and core. Polyurethane (PUR) is a material of foam and aluminum /steel used in face sheets. As a next step; boundary conditions and loads were applied to the static model and in modal analysis. For the analytical validation, matlab codes were created using standart procedures in literature to calculate the deflection and stresses. At the next phase of this study; parameter correlation study was succeedded. Latin Hyper Cube Sampling Technique has been used for random selection and to be able to see the different results of any two inputs. The purpose was to obtain a better correlation (The Spearman's Rank Correlation) matrix with higher accuracy. Using 100 samples for this study, change in skin thickness and parameters such as natural frequencies, mass, stiffness, total deformation, stress were obtained. The same procedure has been used to set a correlation matrix between parameters and the effect of change in foam thickness. Then design of experiment procedure has been set taken at design points. Static analysis has been applied to the samples and the graphics have been drawn to show the relationship between the effect of change in thickness of core, the effect of equivalent stress, shear stress and stifness for both aluminum and steel face sheets. The same graphics have been drawn to show the effect of change in thickness in face sheets. The same procedure has been followed for the dynamic analysis. When the results were examined, it was found that skin thickness and core thickness were inversely related to deformation, shear stress and equivalent maximum stress. Secondly; the stiffness of the sandwich structures was shown to be directly affected by both the thickness of the core and skin. Thirdly, the core thickness has no effect on the mass and elastic strain; but if the skin thickness increases, the mass will increase directly and decrease of elastic strain of the structure is observed and if the core shear failure exists the sandwich structure will fail. To finalize; the study shows that keeping the thickness of face sheets minimum as much as possible and optimizing the core thickness; give a better result considering stiffness and weight of the structure. Thus, it has been observed that the core plays a vital role.

Asl, Torabi and Nourbakhsh (2009) studied model generation, static structural analysis and geometrical modifications of real centrifugal-pumps. Using hydrostatic tests, the same geometrical modifications were done not to affect the results, by avoiding from excessive manufacturing costs in order to examine failure of the volute-casing. Reliable pump performance depended on two important points, namely, proper material and hydraulic design. The casing of centrifugal-pump consist of two main parts; one of them absorbs the fluid whereas the other removes it. The strength of casing depended on geometry, material and temperature of water. In this article; it was focused on the optimization of volute casing of centrifugal pumps. The corrosive fluid has high impact on the stress of the pump. Due to being a fragile structure of cast iron; if the stresses are larger than the strength of the material, this makes the pumps unreliable. To be able to produce reliable pumps, mechanical capacity of the pump casing should be predicted and the geometrical size of the pump should be optimized. This article represented the benchmarking of theoretical and experimental results. The centrifugal pump casing with axial suction consisted of volute casing and an impeller with six vanes. Running at 2900 rpm, the best efficiency point was shown at 85 liter/minute flow and 28 m head. The casing body was made of GG25 cast iron and casing cover was made of AISI 304 type steel. The cast iron body is fitted to the steel cover, then a suction pipe is fixed using threaded connection. After 2000 h service period, some cases of cracking in pump casing represent crack initiation near body of casing on suction side and the discharge nozzle at interface of volute-shaped segment. It is obviously clear that, if the pumping fluid is more corrosive, the strength of the component will be lower. For this reason, ensuring reliable working conditions; optimized dimensional design has been studied. Performing experimental validations, model analysis was performed to verify mesh validity. Accordingly, the hydrostatic pressure equation simulating the hydrostatic test conditions according to API 610 was used. The location of maximum failure-equivalent stress of the cast iron was easily identified. The stress analysis results accurately confirm the crack start position. According to the optimized model, 6 prototype models were produced to verify the mechanical capacity of the volute casing using a test equipment to measure the hydrostatic pressures of crack initiation. The hydrostatic test results were recorded and it was observed that the typical operating life in corrosive water services for initial and optimized centrifugal pumps with corrugated housing had a satisfactory effect on pump performance. The hydraulic test results showed that the optimized volute casings have high mechanical capability, but the overall performance and efficiency degradation is negligible due to changes in the hydraulic design of the volute shape segment. These changes can be easily implemented without excessive cost.

Dong, Bao and Zheng (2012) studied the stress analysis of elbows which have unequal wall thicknesses with two methods which were named as eccentric circle method and multi-spot spline curve method. Due to manufacturing defects, elbows are always the weakest part of these systems. Many numerical simulation methods have been used for the stress analysis of elbows under different loads, but most accept the thickness of the walls as constant. In this article, multi-spot spline curve and eccentric circle method is used for unequal elbow walls and stress analysis results were studied and compared. In this article, all design parameters and material properties such as design pressure, design temperature, inner diameter, material type, modulus of elasticity, yield strength, fluid density in pipe and wall thickness of flat pipe have been determined. The minimum and maximum elbow thickness was calculated using these inputs. Loads and structures are considered symmetrical; therefore, a ¼ value of the structure was created for finite element analysis and a multi-spot spline curve and eccentric circle method were applied respectively. The differences in the stress distribution of the two models are almost the same, and the maximum stress is seen at $\theta = 45^{o}$ in the inner wall. In this study, the parametric FEM of the structure was established using APDL and the stresses of different R & D elbows were examined. When the results were compared, it was found that the greatest stress and location were almost the same. However, the eccentric circle method is simpler than the other method and has no diameter limitations. Therefore, it is stated that eccentric circle method is more suitable for use in finite element modeling analysis.

In the article entitled thermal and mechanical welding simulation was realized using ANSYS FE procedure. Because of a lot of welding process parameters, in this study, there are some assumptions which reduce the modelling efforts and time. This study has been divided into three parts such as: thermal analysis, sensitivity analysis and mechanical analysis. In this study; materials AISI 316 and INCONEL 625 have been used and thermal material properties have been given as a table. During the welding process; the temperature has risen up to 1650 K and since these properties are not valid at this degree, they can be changed. For this reason after 1643 K; some properties were assumed as constant values. At laser welding conditions, molten materials have been described “live” due to absorbing all the heat; then at the beginning of the material analysis the same elements have been kept as “died” to be able to simulate the real division, so this is called birth and death technique. The same procedure is also valid for TIG welding. This technique has been applied to filling material also. The welding specimen‘s geometrical dimensions have been given as 315 mm x 300 mm and the lateral constraints have been defined with DOF nodes. The connection between regions have had different mesh densities. The heat affected zone had a finer mesh. At the sensitivity analysis part, experiment sets were developed. In the first set, welding speed was changed. It was shown that the slope of temperature raised directly with the welding speed. Secondly, keeping the welding speed constant, the mesh size was changed. So, it was understood that the thermal distribution were similar. Next, sensitivity analysis was conducted. Mesh size was a minor effect; but welding speed was one of the primary parameters. The time necessary for the thermal calculation has been found as 20h/ pass/mm. While performing the mechanical analysis, kinematic hardening option was selected ignoring the viscous effects.

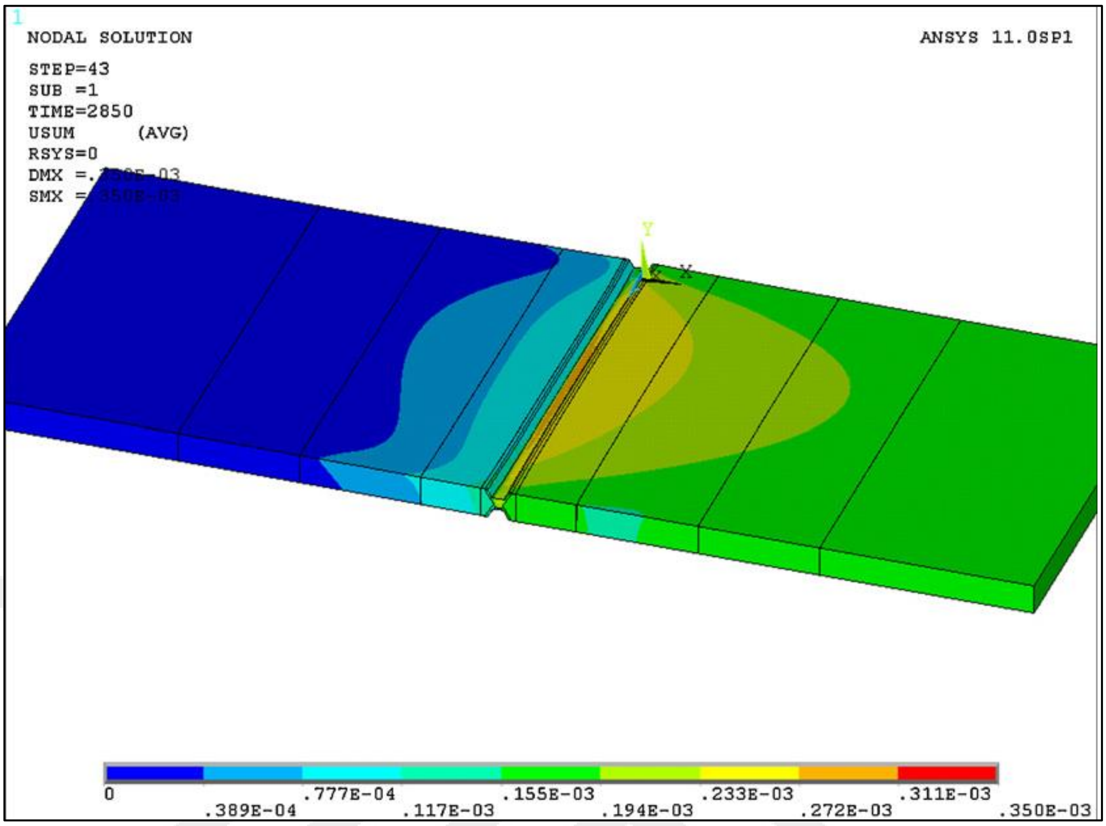

Figure 2.3 Contour plot of total displacement in a TIG welding (Capriccioli & Frosi, 2009)

Both the figures and experiments have shown that the specimens have remained almost plane. It has been understood that the calculation time has been dependent on model size but when it is compared to thermal analysis, the mechanical analysis has took almost 5 times longer. For this reason, a special procedure has been developed. So, the specimens have been restrained and the welding procedure has been applied then; the final linear and angular shrinkages have been calculated. Finally; CPU time have been reduced.

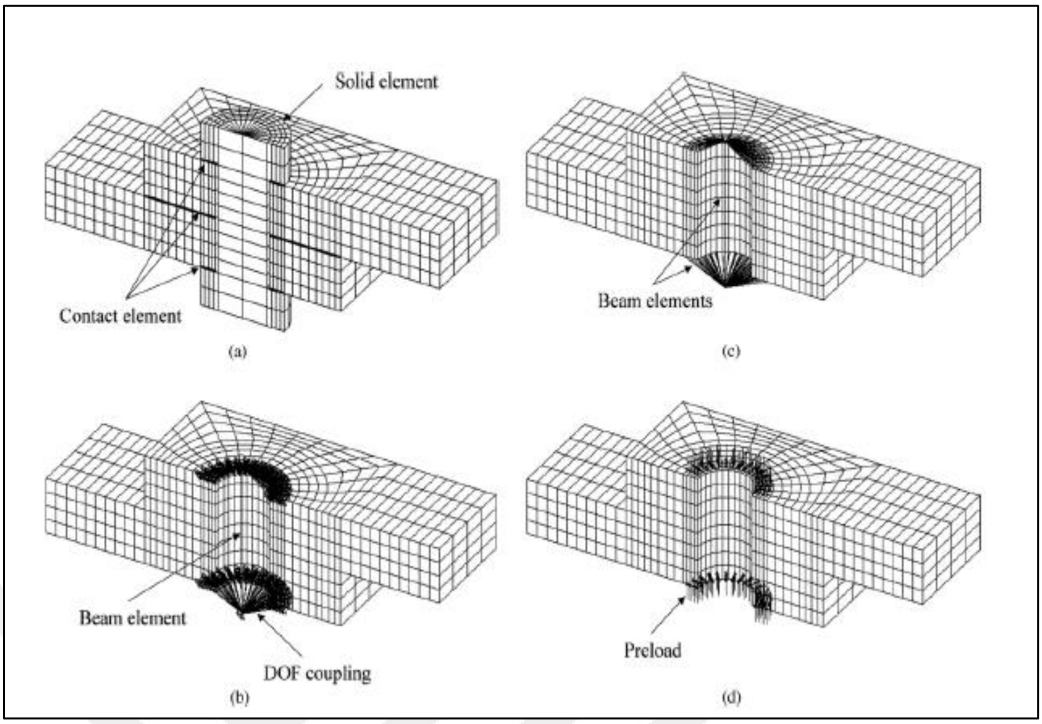

Figure 2.4 Finite element models: (a) Solid (b) Coupled (c) Spider (d) No bolt model (Kim, Yoon and Kang, 2007)

Çelik, Uçar and Cengiz (2007) studied the optimization of the body of hydraulic gear. In hydraulic pumps working under high axial and radial forces, pressure differences occur between inlet and outlet ports. In this study, various sizes of pump housings are modelled in Solidworks 3D design program using its own design table function. Each parameter such as, flow, revolution per minute, module, number of teeth, diameter of bolt holes, length of bolt holes, numbers of bolt holes, angles between bolt holes, wall thickness of the housing, definition of inlet/outlet holes with flange or without flange, flow rate of inlet and outlet section are defined. All these parameters create different types of pump bodies.

In this study, Ansys Workbench program is used for stress-strain analysis and optimization. Deformation on the pump body is simulated where operation is simulated as static linear and three dimensional considering some assumptions. It is assumed that there is a zero leakage on the pump, pressure increases from inlet to outlet linearly, gears are assembled to pump housing with bolts and these bolts hold the pump cover fit on the housing and finally bolts and pump cover are rigid. Under all these assumptions, a gear pump is modelled, whose number of teeth (Z) is 16. Considering rigid bolt and rigid cover assumption; the degree of freedom of inner surfaces of bolt holes on the pump housing is set as zero. In addition to this, pump cover contact to pump housing cannot move through the normal direction of housing contact surface and rotate. After defining all theseconstraints in the program, meshing operation is applied and the analysis is run.

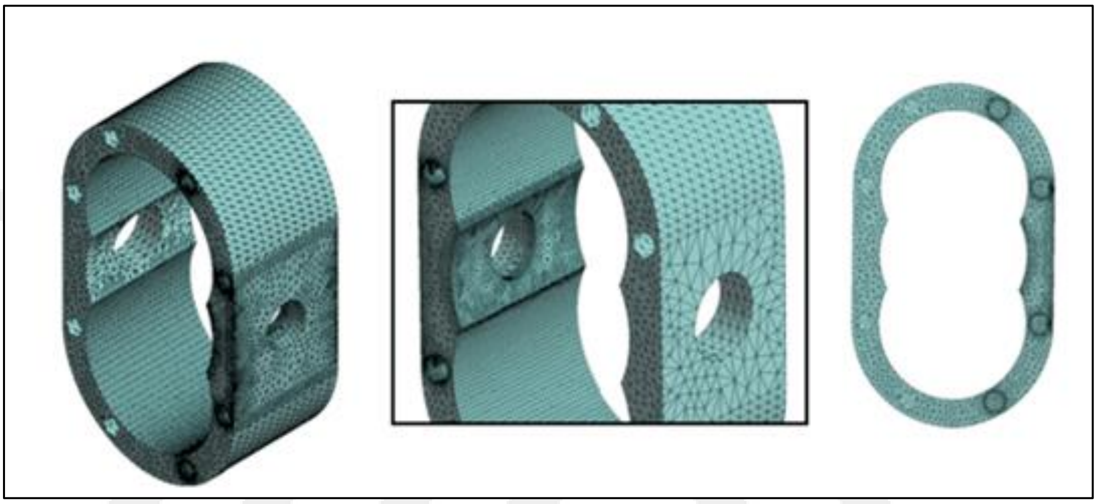

Figure 2.5 Finite element model of pump housing (Çelik, Uçar & Cengiz, 2007)

As expected, stress concentration occurs at outlet side due to outlet pressure and at the bolt holes. According to the analysis results, the maximum stresses that occurs on the pump housing are obtained under the yield strength of steel. The maximum displacement value is very small and has no effect on the operating conditions of the gears.

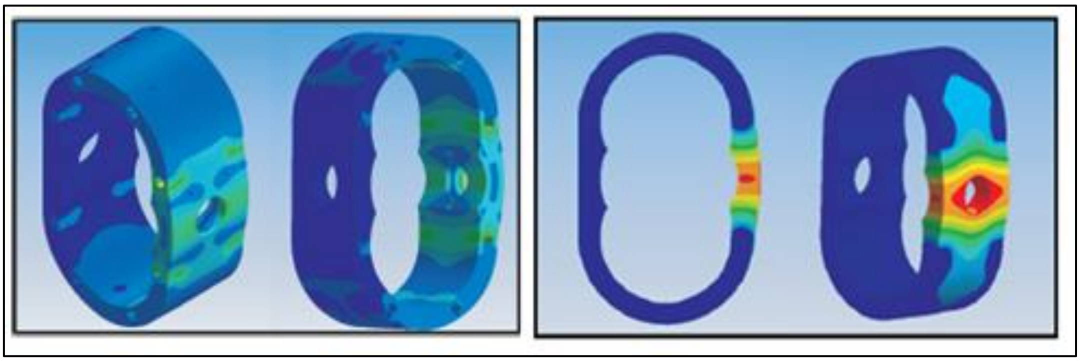

Figure 2.6 Equivalent stress and deformation on the pump housing (Çelik et al., 2007)

Although there are a lot of parameters that are used for modelling of pump housings, only some parameters are defined as design variables of analysis. Objective function is defined as minimum wall thickness using the "Goal Driven Optimization" approach in Design explorer module of Ansys. According to analysis results, wall thickness is decreased to 3 mm and weight of pump housing is decreased approximately to 1 kg. Stress-strain and deformation values are increased compared to initial values but these are still under boundary values. Displacement is increased to 1 µm near the outlet hole, but this value is too small to care about. So that it is seen that optimization integrated software represent the most effective design.

Gerdemeli, Kurt and Deliktaş (2010) studied the prevention of crane damage due to heavy loads with finite element method using the design software: Solidworks and Ansys. Variables such as the load type, crane location, geometric features, environmental conditions, motion type and design features affect the damage of the crane parts. In this study, a tower crane is modelled with some geometrical simplifications in the Solidworks program. Mesh operation is applied to each part of crane in Ansys program. The meshed components are assembled and the finite element model of a tower crane is obtained. Boundary conditions and loads are applied to the model. The material of the tower crane components is St-37. Internal-external loads, wind load, dynamic loads due to operating conditions and friction forces have an impact on the tower crane and all of them are taken into account in the analysis. In this system, some parts are under compression effect; whereas the others ones are under tension effect due to the load combination. In this study, considering these load combinations, finite element analysis is accomplished and maximum stress values are obtained.

Secondly; the stress values on the tower crane components are calculated analytically. It is observed that the deviation between these two methods are not much except the tower crane connection part. The reason of this deviation can be assumed to arise from the analytical calculation and the numerical method used in the finite element method and this seems unacceptable.

The results shows that, there are punctual stresses on the corners of the crane components. This situation can be evidence that finite element method analysis is insufficient in some regions, but in this study punctual loads are neglected. Based on the analysis results, some parts can be designed lighter, considering production cost and design duration.

Çelik, Topakçı, Yılmaz and Akıncı (2007) studied a simulation model of chisel that simulates working conditions. A chisel is an agricultural machine that has components with sharp edges and works without tipping the soil. So that, feet, body and the other connections support elements working under high force distribution. The components break or become unusable due to high deformation with the influence of these forces. Manufacturers use the materials with high safety coefficients or design machines by increasing the thickness of the components to be able to prevent the damages. So, this increases the weight and cost.

In this study, the chisel is modelled in Solidworks and simulated in Ansys Workbench program. In order to simulate real working conditions as much as possible, all components of the chisel is modelled and analyzed. The chisel is fixed from connection shafts and the forces are applied at only in vertical direction since the horizontal forces are too small to consider. After defining the material properties of each component in the system, mesh operation is applied. 10-Node tetrahedral structural solid (Solid 187), High-order Surface to surface contact (Conta174), 20-Node Hexahedral Structural Solid (Solid186) and Surface contact target (Targe170) element types are used. The results revealed that the maximum equivalent stress was obtained under yield strength for each component and deformations that occurred on the chisel can be defined as elastic.

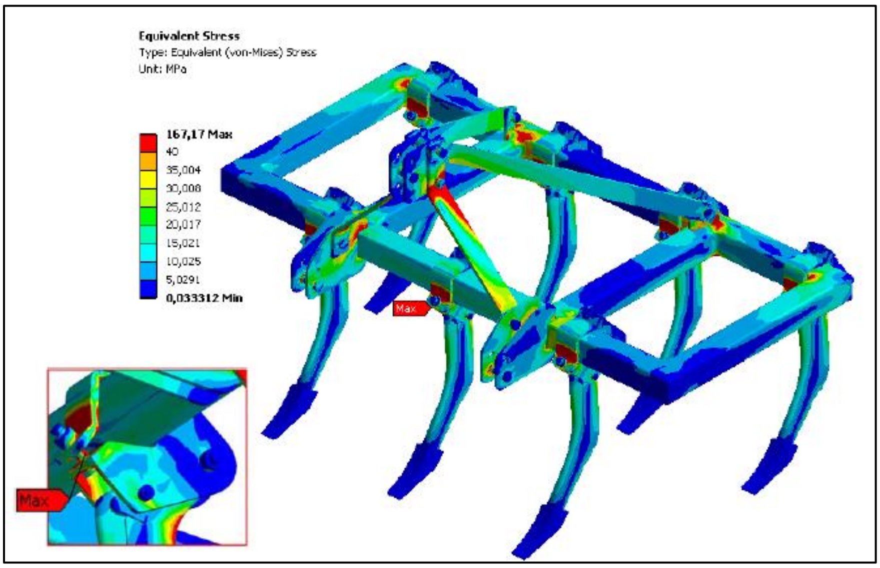

Figure 2.7 Maximum equivalent stress for all construction (Çelik, Topakçı, Yılmaz & Akıncı, 2007)

This study provides a benefit in terms of prediction and designing of the most suitable agricultural machines with low cost and higher advantages.

Yıldız (2016) studied the swing arm used in the front suspension system of the automobiles was optimized by topology and shape optimization respectively. It was modelled hundred the swing arms using Catia design software. After this; considering boundary conditions, finite element model has been created and the stress values have been obtained. Objective and constraint functions have been developed, with radial basis functions and Matlab codes that belong metamodeling. At this technique, design dimensions are input and the stress values and the weight of the swing arm are output. Afterwards, mathematical model has been created. Interior search algorithm has been run. Based on the results, the model of the swing arm has been improved reducing the weight 28% according to the initial model. This study shows that, interior search algorithm can be applied structural optimization of vehicle components effectively.

CHAPTER THREE

LUBRICATION SYSTEMS IN ENGINES

3.1 Definition of Lubrication System

Lubrication system is one of the main factors affecting the engine. This system provides a flow to the clean oil at the correct temperature, with an appropriate pressure obtained in each part of the engine. The oil is sucked from the sump by the pump and then sent to the oil filter and then to the main bearings and the oil pressure gauge. The oil passes through the feed holes in the main bearings to the holes on the crankshaft and connecting rod bearings. The oil distributed by the crankshaft is taken from the bearings of the piston pin and cylinder walls. Excess oil is scraped by the lower ring on the piston. The excess oil then flows back into the sump. Figure 3.1 shows general view of an engine lubrication system.

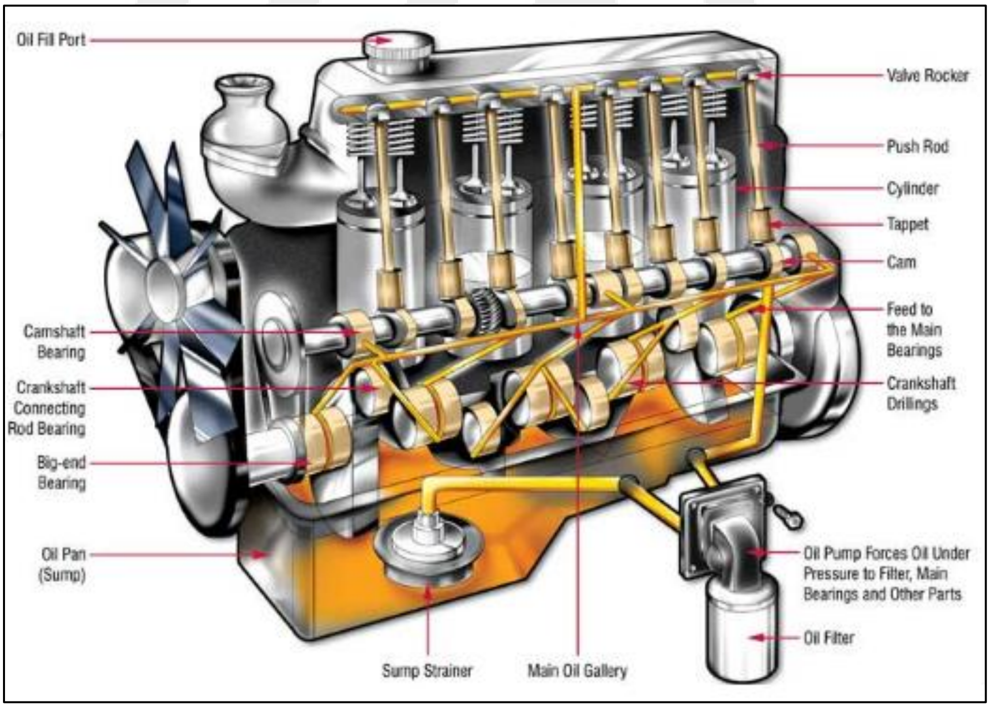

Figure 3.1 General view of an engine lubrication system (Lubrita, 2017)

Lubricants are one of the most important factors affecting the efficiency and life of an engine or machine. No matter how perfect the parts of the motor (machine) that move on each other, their surfaces will still not be smooth depending on the molecular structures of the parts of the work. The figure 3.2 shows the surface of two parts viewed with a magnifying glass.

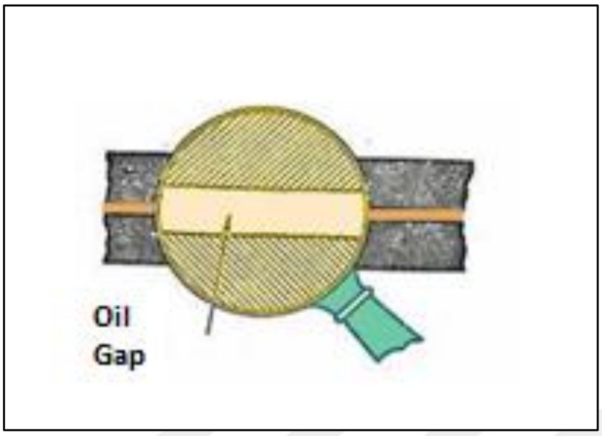

Figure 3.2 Oil film of bigger picture

Oil film is necessary to prevent the parts from rubbing directly to each other in order to facilitate movement, to get more efficiency and to extend the life of working parts. A suitable liquid is used to reduce friction between the two body layers. This fluid is called lubrication.

3.2 Benefits of the Lubrication System

3.2.1 Reduces friction

Each of the parts in the engine is connected to each other and abrasion occurs during working. Due to the use of engine oil these abrasions are minimized. While the engine is running, wear occurs in particles at micron level during the operation.

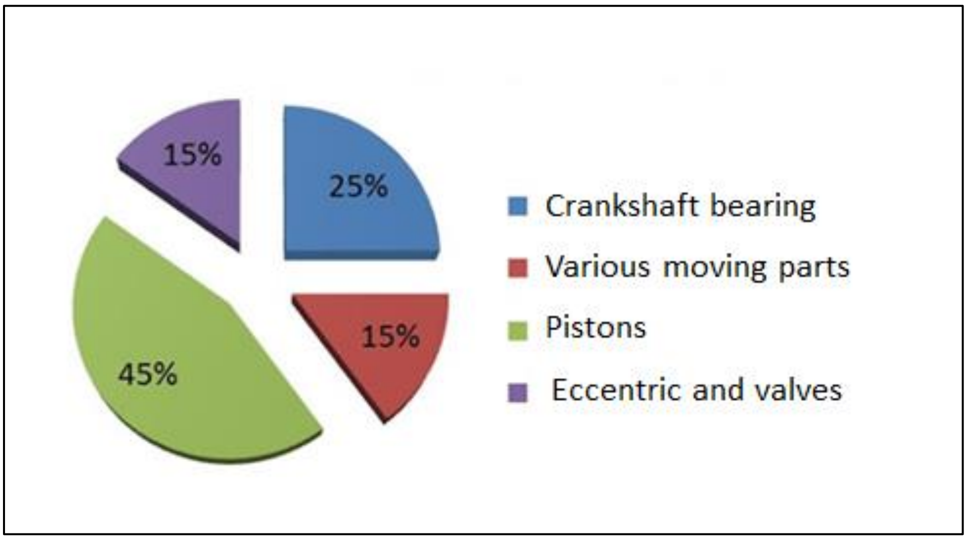

Figure 3.3 Wear rate of engine components (Sekizsilindir, 2017)

3.2.2 Cleans the engine

While the engine is running, abrasions occur due to the connection of parts. Although these abrasions are at micron level; these will cause burrs and impurity between the components after a certain period of time. These particles damage the surfaces of the parts. Pressurized oil; which circulates in the engine, cleans the engine and it provides to collect the burrs and impurity into the oil sump.

3.2.3 Reduces the heat in the engine

Due to the abrasion between the parts in the engine, the heat of the parts increases while the engine is running. The heat is transferred to the engine oil. Thus the lifetime of an engine is increased.

3.2.4 Allows the engine to work quieter

Because of the gaps between the parts inside the engine; a lot of crashes and friction occur. This issue causes vibration and harshness. Oil engine that is circulating in the system behaves like a thin film that covers the parts. So, vibration and harshness is reduced.

3.3 Lubrication System Components

Lubrication system consist of oil sump, oil sump plug, oil sump gasket, oiling channel and holes, oil filter and oil pump, mainly. Figure 3.4 shows General view of Lubrication system components.

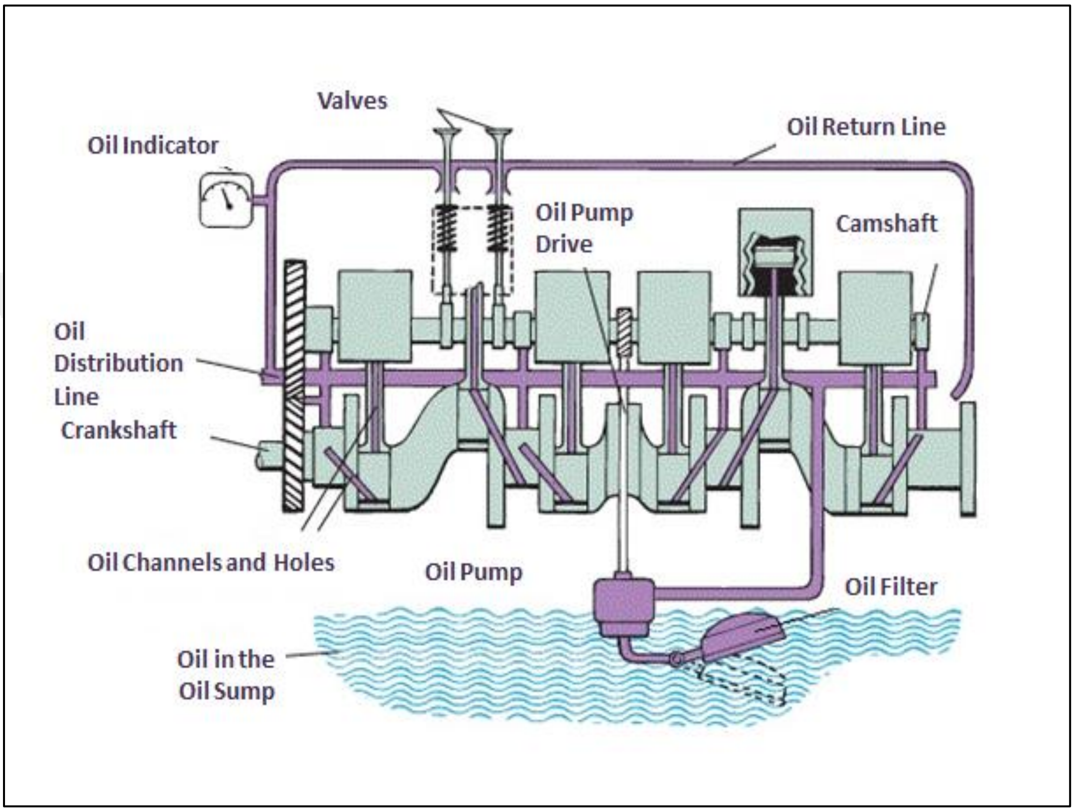

Figure 3.4 Sub-components of lubrication system (Sekizsilindir, 2017)

3.3.1 Oil Sump

Oil sump is the place where the oil accumulates. Oil sump capacity changes from 4 liters to 10 liters depending on the size of the engine.

3.3.2 Oil Sump Plug

It is necessary to remove oil from oil sump to the engine. This part; which looks like a thick bolt, serves as a stopper by screwing it into the oil sump (Figure 3.5).

Figure 3.5 Oil sump plug (Personal archive, 2019)

3.3.3 Oil Sump Gasket

Oil sump gasket, is assembled at the junction of the oil sump and the engine block and prevents oil leakage. Oil sump gasket can be of various type. One of them is given in Figure 3.6

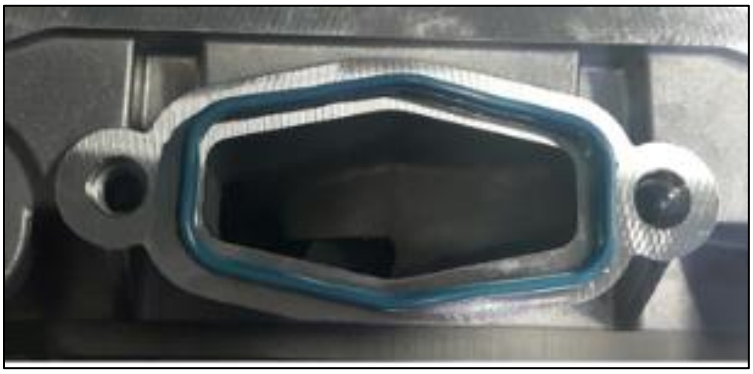

Figure 3.6 Oil sump gasket (Stackpole International Fluid Power Solutions archive, 2019)

3.3.4 Oil Channels and Holes

The oiling channels and holes provide the necessary pressurized oil that comes from the oil pump to the camshaft bearings, the valve springs, the crankshaft bearing and the other moving parts.

3.3.5 Oil Filter

It filtrates particles at micron level, which is circulated in the system by oil pump continuously and prevents the possible engine damages.

3.3.6 Oil Pump

Function of an engine oil pump is to supply oil pressure to all engine areas to secure engine function and life time.

A mechanical device that converts mechanical power into hydraulic energy such as flow and pressure is called a hydraulic pump. At this point, the fluid in the system is considered to be incompressible (Wikipedia, 2017).

The tasks of the pump can be listed as:

- Flow from low pressure to high pressure
- Flow from a lower level to a higher level
- Flow at a faster rate

Pumps can be classified according to their working principles, the liquid they handle and their orientation in space. It is possible to classify the pumps in many ways. However, the pumps are classified according to the three main categories considering the energy added to the fluid.

- Open Screw Pumps
- Kinetic Pumps
- Displacement Pumps

3.3.6.1 Open Screw Pumps

This type of pump is often used in waste water plants and irrigation channels (Piping engineering, 2017).

3.3.6.2 Kinetic Pumps

This type of pump is supplied with continuous energy to accelerate to values greater than those occurring during discharge of the liquid in the device. Thus, the velocity reduction in or out of the pump results in an increase in pressure.

- *Centrifugal Pumps*

Mechanical devices used to transfer various liquids are called centrifugal pumps. These pumps operate according to the centrifugal force principle. They convert the energy supplied by the movers, such as the electric motor, steam turbine, to the energy in the pumped liquid. Figure 3.7 presents general view of centrifugal pump.

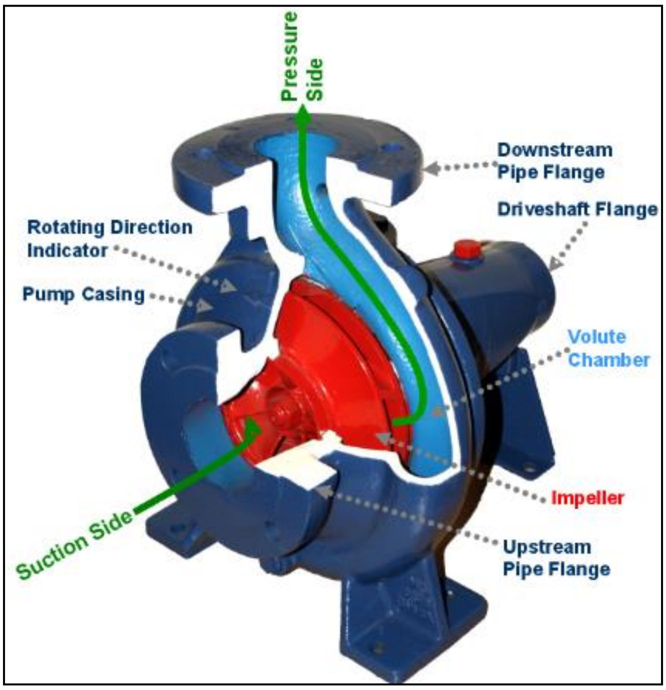

Figure 3.7 General view of centrifugal pumps (Centrifugal pump overview, 2017)

Centrifugal pumps has a casing, an impeller and the assembly for the impeller to move smoothly within the casing consisting of mechanical components. The impeller rotates giving a liquid accelerating. So speed is converted to pressure and flow by the casing (Intro to pumps, 2018).

The parts of the centrifugal pump can be listed as:

- Pump shaft
- The sealing mechanism
- Structural components
- Wear surfaces

The volute casing is designed to guide the flow out of the impeller in order to convert the fluid flow's kinetic energy into static pressure. Figure 3.8. shows the section view of centrifugal pump.

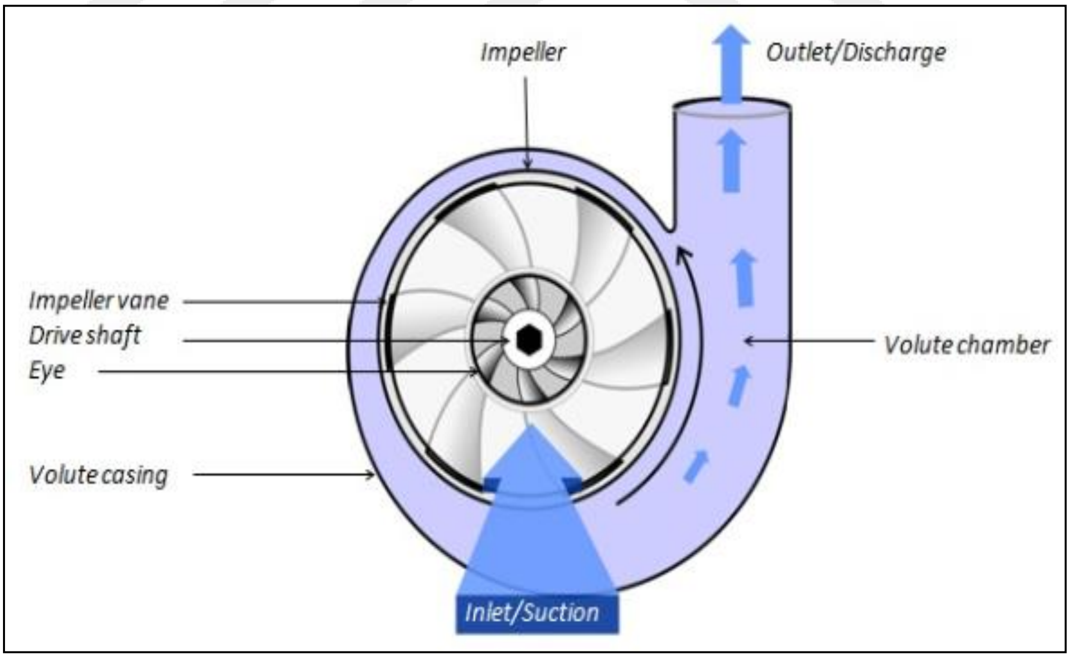

Figure 3.8 Section view of centrifugal pumps (Centrifugal pump overview, 2017)

Centrifugal pumps must be combined with a drive for trouble-free operation. Centrifugal pump units vary widely according to their application areas.

A centrifugal pump unit comprises at least two components (a drive and a pump). Usually the electric motor is the drive in a centrifugal pumping system. However, sometimes it can also be operated by other drives such as steam turbines or natural gas engines. Figure 3.9 shows the most common pump unit.

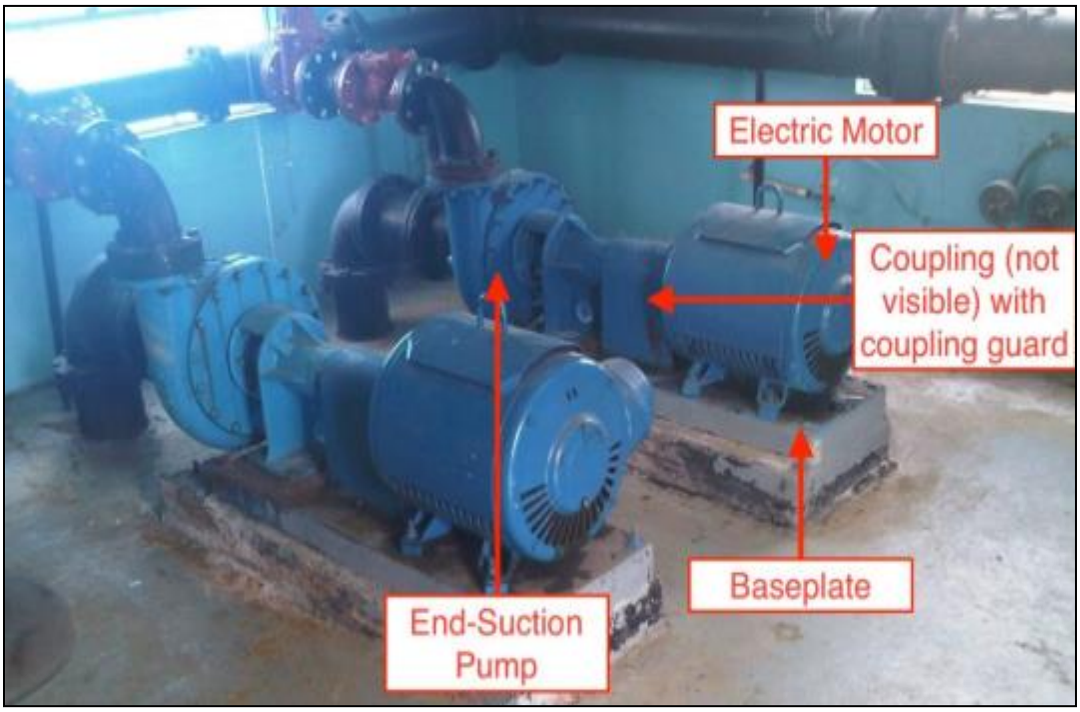

Figure 3.9 Frame-mounted end-suction pumping unit (Intro to pumps, 2018)

Advantages of centrifugal pumps can be listed as follows:

- Wide range of capacities, flow and pressure
- Low maintenance and first cost
- Quiet operation
- Simple operation

Their disadvantages can be listed as follows:

- Not suitable for high viscous liquids
- Generally they don’t have the capabilities of high pressure applications.
- They can’t ensure high pressures without changes in design.

3.3.6.3 Displacement Pumps

Displacement pumps convey the medium through closed volumes. The pump supplies the fluid with mechanical energy by accelerating the medium and thus increasing its kinetic energy.

As a rule, reciprocating pumps for incompressible media have no internal compression, so that the pressure increase is only due to the resistance of the downstream motor. The drive energy (depending on the speed and the drive torque) is transferred by the pump to the fluid (leads to an increase in pressure and a certain oil flow). Schematic view of a pump with drive and coupling is given Figure 3.10.

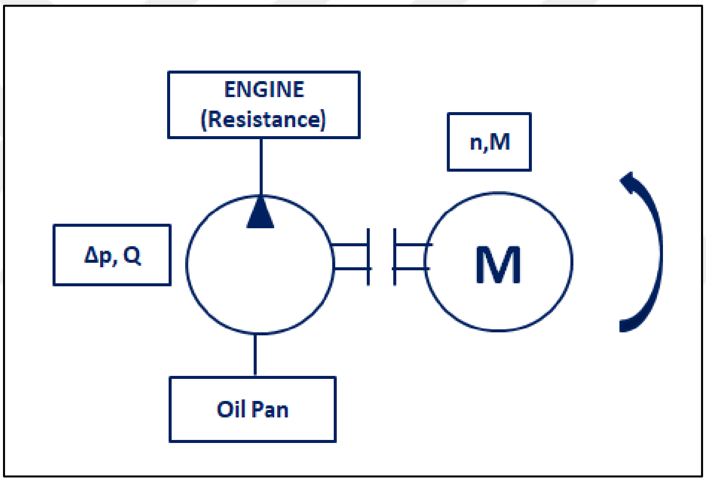

Figure 3.10 Schematic diagram of a pump with drive and coupling (Böwing, 2009)

Drive Power: $$P = 2.\pi.\frac{n}{60}.M \tag{3.1}$$

Pump Power: $$P = \Delta_{p.}Q \tag{3.2}$$

This pump moves the fluid by trapping a constant amount of force that confines the volume to the discharge pipe.

Some positive type of displacement pumps operates decreasing cavity on the discharger side and expanding cavity on the suction side. Liquid flows into the pump as the cavity on the suction side expands and the liquid flows out of the discharge as the cavity collapses.

3.3.6.3.1 Rotary Lobe Pumps. A variety of industries use rotary lobe pumps, in especially in food applications. The reason of it being popular is that rotary lobe pumps have reliability; high efficiency, corrosion resistance, sterilize-in-place and good clean-in place characteristics. These pumps can be; multi lobe, tri-lobe, bi-wing and single (Figure 3.11).

a) Bi-wing b) Tri-lobe c) Multi Lobe Rotary Pumps

Figure 3.11 Rotary lobe pump variables (Wrightflowtechnologies, 2018; Indiamart, 2018; Carotek, 2018)

These pumps have large pumping pockets and are non-contacting. So they can be used without damaging solids such as Olive or Chery. Beside this, it is useful for high viscosity liquids. Since having a reversible flow, they can operate dry for a long time.

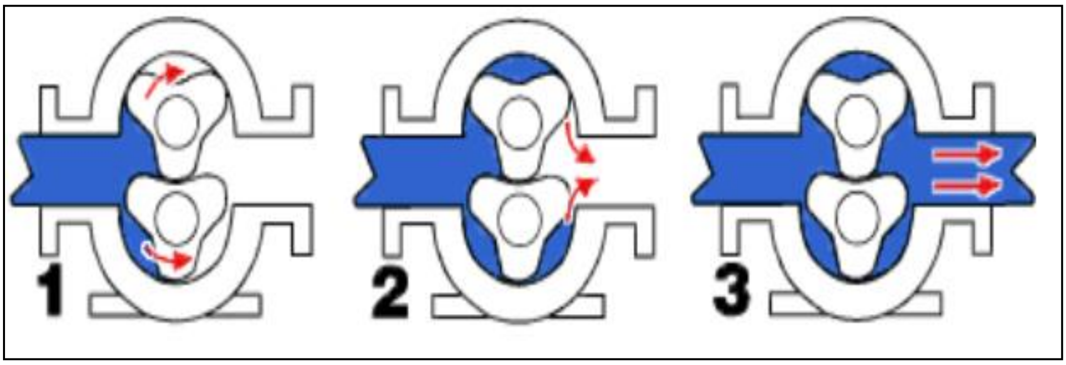

Figure 3.12 Schematic view of working principle of rotary lobe pumps (Pumpschool, 2018)

When the lobes are positioned as shown in figure 1; it creates an expanded volume on the inlet side of the pump and fluid flows inside the pump. As the lobes rotate; the fluid moves to pocket of lobes and there is no leakage between the lobes. Finally, rotating lobes force the fluid through the outlet side under pressure.

Advantages of rotary lobe pumps can be listed as follows:

- Non-pulsating discharge
- No metal-to-metal contact
- Long term dry run
- Pass medium solids

Their disadvantages can be listed as follows:

- Reduced lift with thin liquids
- Needs two seals
- Needs timing gears

3.3.6.3.2 Gear Pumps. In gear pumps, the pressure is below the atmospheric pressure in the suction port, that is the system pressure. This pressure difference causes the pump casing, bearings and gears to be subjected to large axial and radial forces. The magnitude of these forces vary in proportion to the gear geometry, gear dimensions, flow rate and working pressure. Here, the parameters can be defined as follows:

- V_0: Volume of fluid filled into a tooth (cm^3)
- V: The amount of theoretical fluid that the pump presses in one cycle (lt/min)
- Q_T: Theoretical flow rate - the amount of fluid that the pump presses at a specific cycle (lt / min)
- Q_e: The actual flow rate of the pump in a certain period (l / min)
- V: Volumetric efficiency
- η_T: Total yield
- n: Speed (rpm)V
- m: Module (cm)
- b: Thread width (cm)
- Z: Number of teeth
- d_t: Diameter of circle
- α_0: Grip angle (°)
- Δ_P: Working pressure (bar)
- P: Pump power (kw)

The working pressure can be found in the following equations depending on the flow rate and the drive power.

$$V_0 = \frac{\pi * d_1 * m * b}{z} \quad [cm^3] \tag{3.3}$$

$$V = 2 * Z * V_0 \quad [cm^3] \tag{3.4}$$

$$Q_T = V * n = \frac{\pi * d_1 * 2 * m * b * n}{1000} \quad \left[\frac{lt}{min}\right] \tag{3.5}$$

$$Q_e = Q_T * \eta_V \quad \left[\frac{lt}{min}\right] \tag{3.6}$$

$$\Delta_p = \frac{P * \eta_T * 600}{Q_e} \quad [bar] \tag{3.7}$$

In gear pumps, large surface areas in the pump body cause high axial and radial forces with high pressure in the pump. The design of the pump body with the optimum geometry that can withstand these forces is an important factor for optimum gear selection, material gain and maximum efficiency from the pump. The positive transmission of this type of pumps causes the gears and the pump housing to undergo different hydraulic pressures and related forces at the same time. The pressure distribution from the inlet to the outlet of the gear pump is shown in Figure 3.13 when the pump is in operation, the fluid rotating when the threads are rotated and rotated in contact with the single teeth. Here;

β: Rotation angle

The pressure is starting to rise with reference to the point, can be expressed as follows.

$$P_i = P_{max} * \frac{\beta}{\pi} \tag{3.8}$$

In addition to the above equations; It is important to determine the optimum inlet-outlet diameter for the inlet and outlet of fluid to the pump body. These diameters;

- Q_e (lt/min): Effective flow rate,
- V (m/s): Speed,
- d (mm): hole diameter can be found with the following equation.

$$d = 4.607 * \sqrt{\frac{Q_e}{V}} \quad [mm] \tag{3.9}$$

- *External Gear Pumps*

External gear pumps are similar to rotary lobe pumps. These pumps deliver liquid between the rotors and the housing and are advantageous for incompressible fluid. Since there is less friction between the moving parts, mechanical energy is very efficient at this type of pumps. Prinbciple of the operation of an external gear pump is given Figure 3.13 and pump gears and pressure distribution is given in Figure 3.14

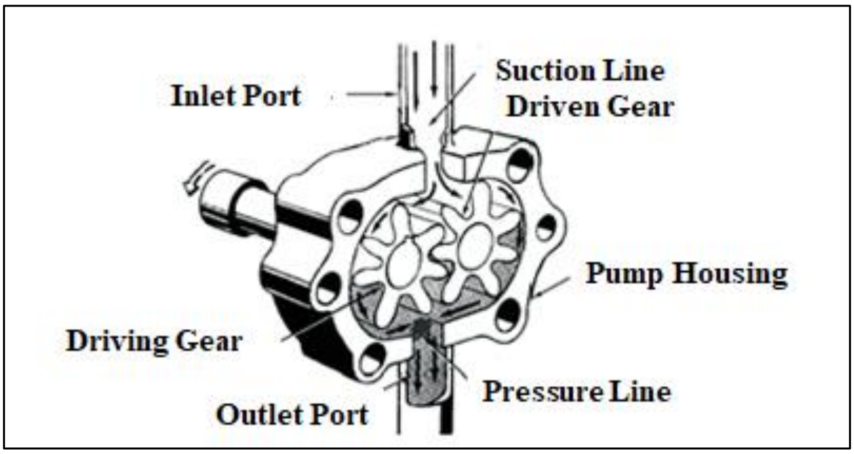

Figure 3.13 Principle of the operation of an external gear pump (Çelik et al., 2007)

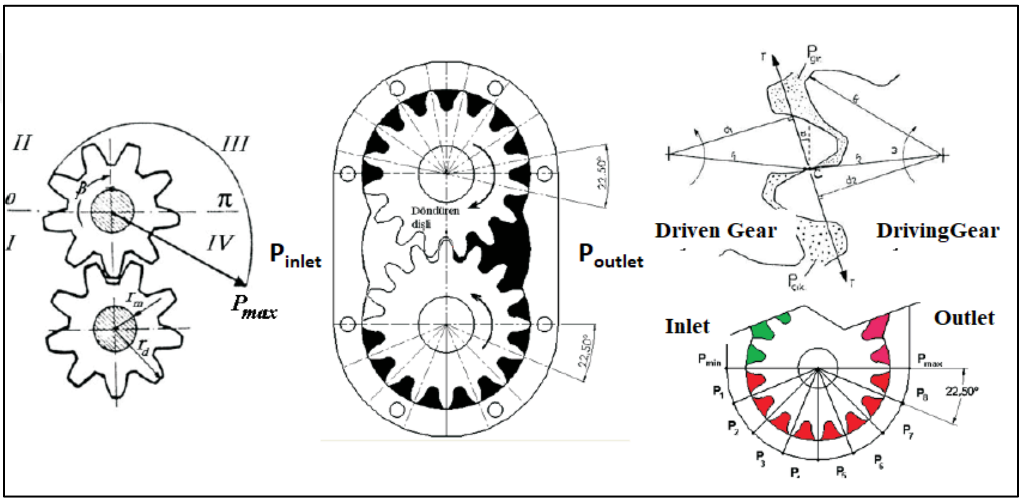

Figure 3.14 Pump gears and pressure distribution on the body (Çelik et al., 2007)

In external gear pumps, the housing is base part of the system. During the operation of the pump, the pressure force acting on the internal surfaces of the pump body where the fluid is in contact and the gear bearings in the body, the bearing forces applied to the bearings due to the pressure difference at the inlet and outlet causes the deformation of the pump. As shown in Figure 3.14, with the assumption that there is no fluid leakage, we can say that the pressure has increased linearly (Çelik et al. 2007).

Here, the I. zone is the inlet pressure zone and zero pressure is accepted. But in practice, especially in high-pressure pumps, the pressure below the atmospheric pressure in this region causes cavitation in the pump due to the high difference in the inlet and outlet pressures. To prevent cavitation, the pump inlet is supplied by another pump of low pressure (Çelik et al., 2007).

II. zone is low pressure zone and the body is affected by gear bearing forces inside it.

III. zone is affected increasingly by flow pressure between the teeth of gear, coming closer to IV. zone

IV. zone is the region under the influence of the outlet pressure.

Advantages of external gear pumps can be listed as follows (Pumpschool, 2018):

- High production volumes
- High pressures
- Large sealing surface between gear and housing
- Both axial and radial filling, thus better packaging and installation options in the engine
- Good mechanical efficiency
- Cost savings through common parts in the gears and shafts, and small gear size

Disadvantages of external gear pumps can be listed as follows (Pumpschool, 2018):

- No solids allowed
- Strong mechanical stress in the root of the tooth
- Fixed end clearances

- *Internal Gear Pumps*

Internal gear pumps consist of double-rotor system. Internal gear pumps belong to the double-rotor systems. Double rotor means that two interlocking elements, where inner and outer gears, rotate together, whereby the inner rotor rotates in the outer rotor. The system is arranged eccentrically. The eccentricity corresponds to half a tooth height. In tooth engagement, the wheels engage with each other and on the opposite side of the tooth, head seal slide the respective tooth heads with respect to each other.

Some advantages of internal gear pumps can be listed as follows (Pumpschool, 2018):

- Easy to maintain
- Only two moving parts
- Non-pulsating discharge
- Low net positive suction head
- Operates well in either direction
- Excellent for high-viscosity liquids

Their disadvantages of external gear pumps can be listed as follows (Pumpschool, 2018):

- Overhung load on shaft bearing
- One bearing is used only that pump manufactured
- Medium pressure limits
- In general needs moderate speeds

Internal gear pumps have two types named as sickle-free internal gear pump and sickle internal gear pump.

- *Sickel Internal Gear Pumps*

This type of pump has a crescent shape between ge-rotors. Since this crescent shape increases the sealing surface, these pumps can also be used for higher pump pressures. The number of teeth of the outer ge-rotor is more than the minimum two teeth, the number of teeth of the inner ge-rotor:

$$Z_i - Z_a - 1 = X \qquad X > 2 \tag{3.10}$$

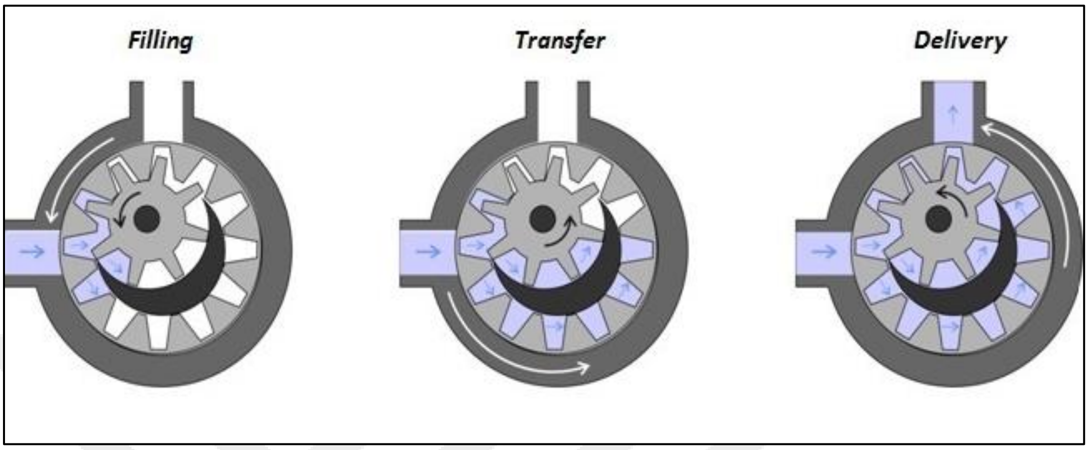

Figure 3.15 Sickel- internal gear pump operation

Fluid fills from the inlet port between idler teeth and the rotor. The arrow specifies the direction of the fluid and pump. Fluid goes through the pump between the teeth of the gear-within a gear principle. This operation is shown in Figure 3.15. The sickle shape divides the fluid and operates between inlet and outlet port. When the chambers are filled with fluid fully, the sickle shape force the fluid out of the outlet port.

- *Sickel Free Internal Gear Pumps*

This type of pump can also be named as ge-rotor pumps. The number of the teeth of the inner ge-rotor (Z_i) is smaller by one tooth than the number of the outer ge-rotor (Z_a). The space formed by the extra tooth in the outer ge-rotor determines the volumetric displacement.

$$Z_i = Z_a - 1 \tag{3.11}$$

Figure 3.16 Schematic view of ge-rotor pump (Gerotor Design Study, 2018)

The ge-rotor pump is a compact and simple pump with only two moving elements as can be seen from Figure 3.16. Due to eccentricity; when the inner ge-rotor and outer ge-rotor start to rotate, these two parts move away from each other. This issue increases the volume between the ge-rotors and it creates negative pressure. The fluid enters from the suction port and goes through between the outer rotor and the inner rotor. The liquid enters between the threads in the pump. The inner and outer geo-rotor teeth are completely knitted to form a seal from the transmission and suction ports. Thus, the liquid is forced out of the delivery port. Figure 3.17 presents components and pressure regions of ge-rotor pump.

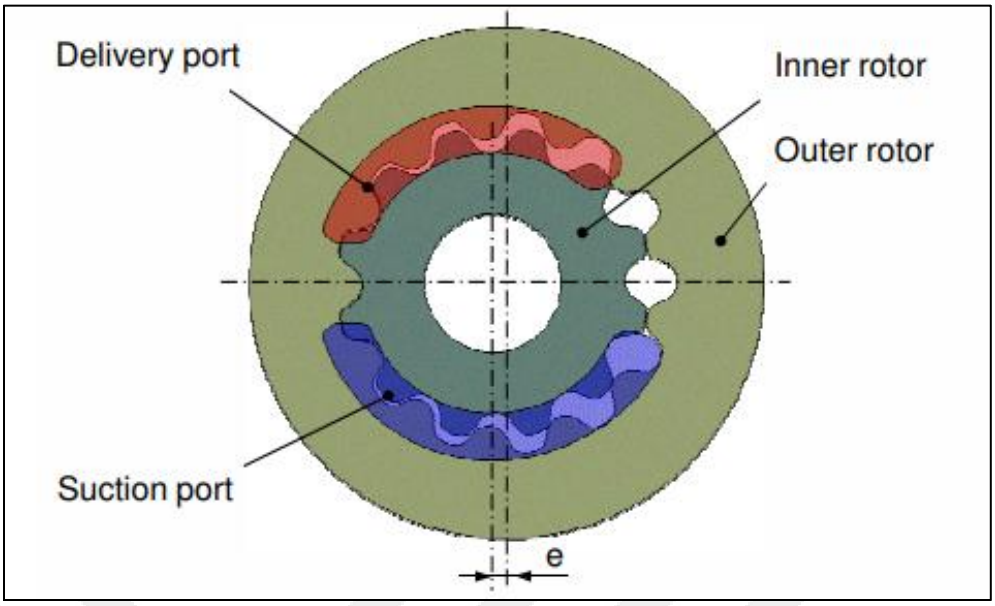

Figure 3.17 Components and pressure regions of ge-rotor pump

Advantages of sickle-free internal gear pumps are; only two parts to move, operating well in either direction, constant and even discharge, quiet operation, only one stuffing box, making to operate with one direction of flow with either rotation high speed.

Their disadvantages are; fixed clearances, one bearing running in the product pumped, no solids allowed, overhung load on shaft bearing, medium pressure limits.

3.3.6.3.3 Vane Pumps. Vane pumps can be subdivided into:

- Balanced (multi stroke) or unbalanced (single-stroke) pumps
- Constant or variable displacement pumps

- *Balanced Vane Pumps*

Two mutually opposite displacement chamber are arranged around the rotor, which are each connected to a suction kidney and a pressure kidney. Suction and pressure kidneys are also connected to each other and to the corresponding sockets. Due to cross-sectional widening between the suction and pressure kidneys, a negative pressure is created, which sucks in the oil. Due to the symmetrical arrangement of the displacement chambers of the rotor, the pump eliminates the bearing side loads. That’s why, this type of pumps are called as balanced vane pumps. General aspect of balanced vane pumps are given in Figure 3.18

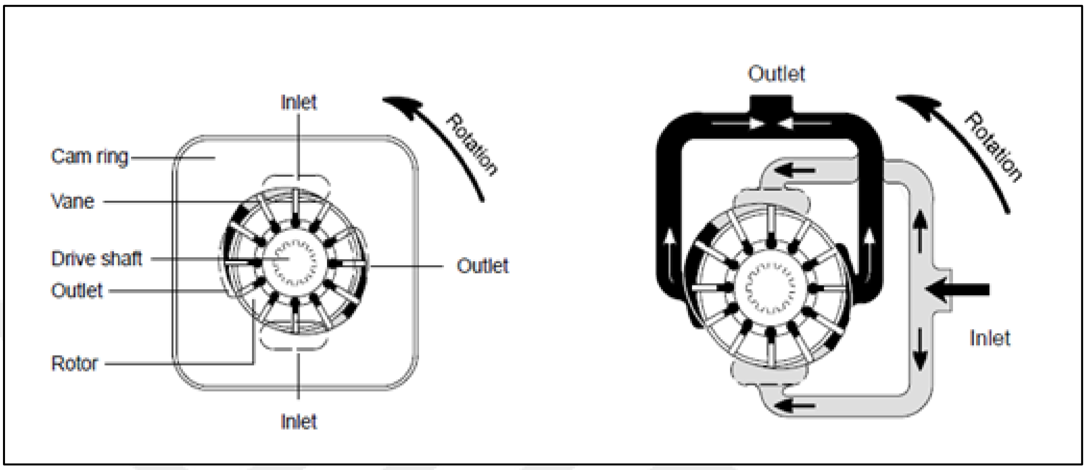

Figure 3.18 Balanced vane pump

Advantages of balanced vane pumps can be listed as follows:

- Due to having less wear and tear, it provides higher service life
- Low noise
- Low mechanical load due to hydraulic relief

Disadvantages of balanced vane pump can be listed as follows:

- Constant delivery volume
- More complicated design
- Higher manufacturing cost because of complex structure

- *Unbalanced Vane Pumps*

Unbalanced vane pumps mainly consist of rotor and slide (cam ring). There is a rotor which moves inside the slide and each radial slot has vane, which is free to enter or exit the slots because of centrifugal force.

The vanes or blades are designed to coincide with the surface of the cam ring during rotation of the rotor. As the rotor rotates, the centrifugal force pushes the vanes towards the surface of the cam ring. The vanes divide the space between the cam and the rotor ring into the small chambers. During the first half of the rotor rotation, the volume of these chambers raises, thereby the pressure reduces. This is the suction process. During the second half of rotor rotation, the cam ring moves the vanes back into the slots and occurs the reduction of the trapped volume. This positively removes the trapped fluid through the outlet port. In this pump, all pump action takes place in the chambers located on one side of the rotor and shaft, and so the pump is of an unbalanced design.

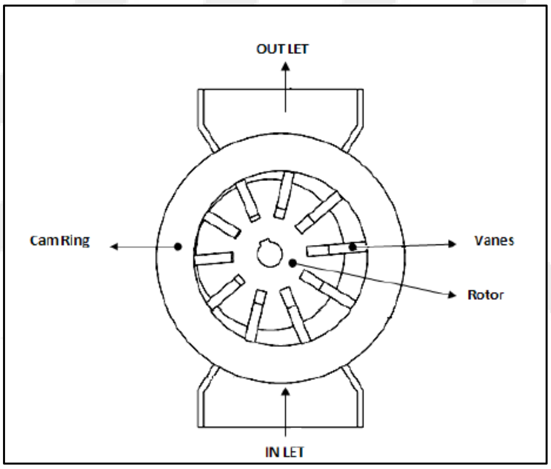

Figure 3.19 Schematic view of a variable vane pump

Advantages of unbalanced variable vane pumps can be listed as follows:

- Low wear
- Possibility of volume variability
- Low noise

3.4 Operating System of Oil Pumps

Oil is absorbed from oil sump via suction pipe and transfered to pressure line with the aid of rotor and vanes. This situation is shown in Figure 3.20.

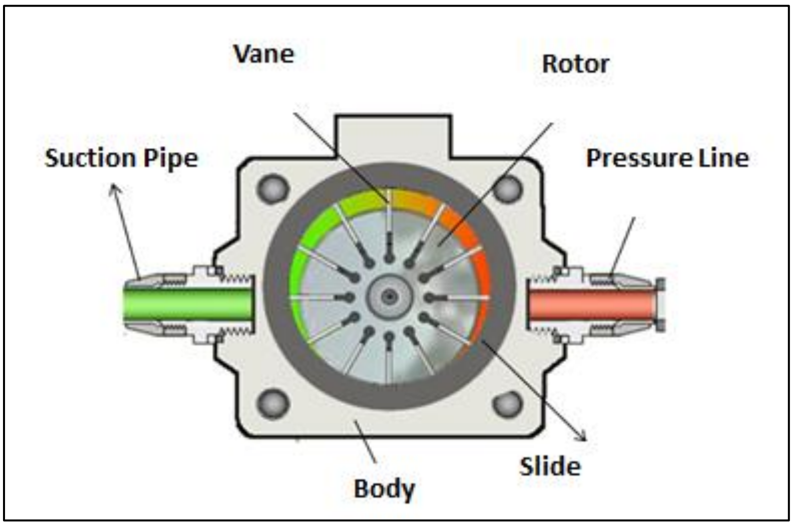

Figure 3.20 Oil pump operation representation (Akışkan Gücü Derneği [AKDER], 2017)

The main task of the oil pump is to provide the amount of oil that the engine needs. The function of an engine oil pump is to supply oil pressure to all engine areas to secure engine function and lifetime. The parts that are oiled by the oil pump is given in Figure 3.21

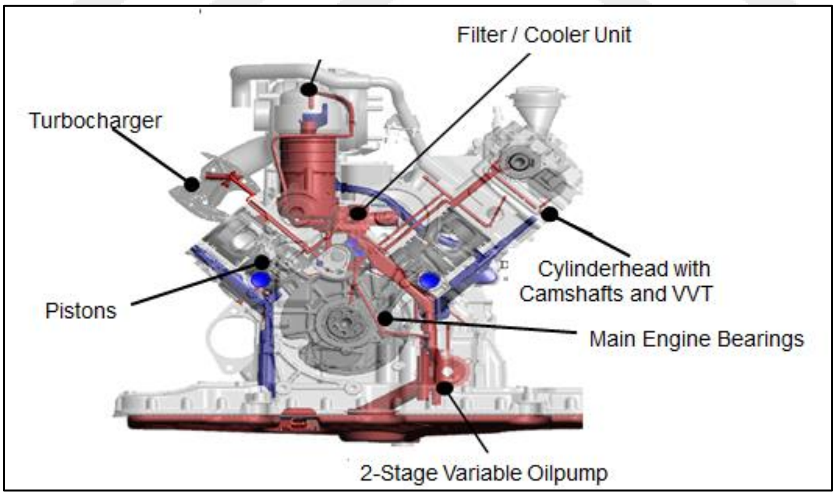

Figure 3.21 The units, lubricated by the lubrication system

3.4.1 The Regulation

A significant difference between the oil requirement of an internal combustion engine and the delivery characteristic of an oil pump designed as a positive displacement pump indicates that the oil pump should be controlled. This is particularly evident in pumps that require a large oil requirement even at low engine speeds.

If the pump supplies more oil than the engine needs, in order to keep the system in safe situation, the bypass channel opens and more oil comes back to the inlet side. The control valve consists of a ball and a spring. Spring pre-load and channel diameter are the decisive parameters for relieve pressure. The bypass channel is shown in figure 3.22 with a blue rectangular shape.

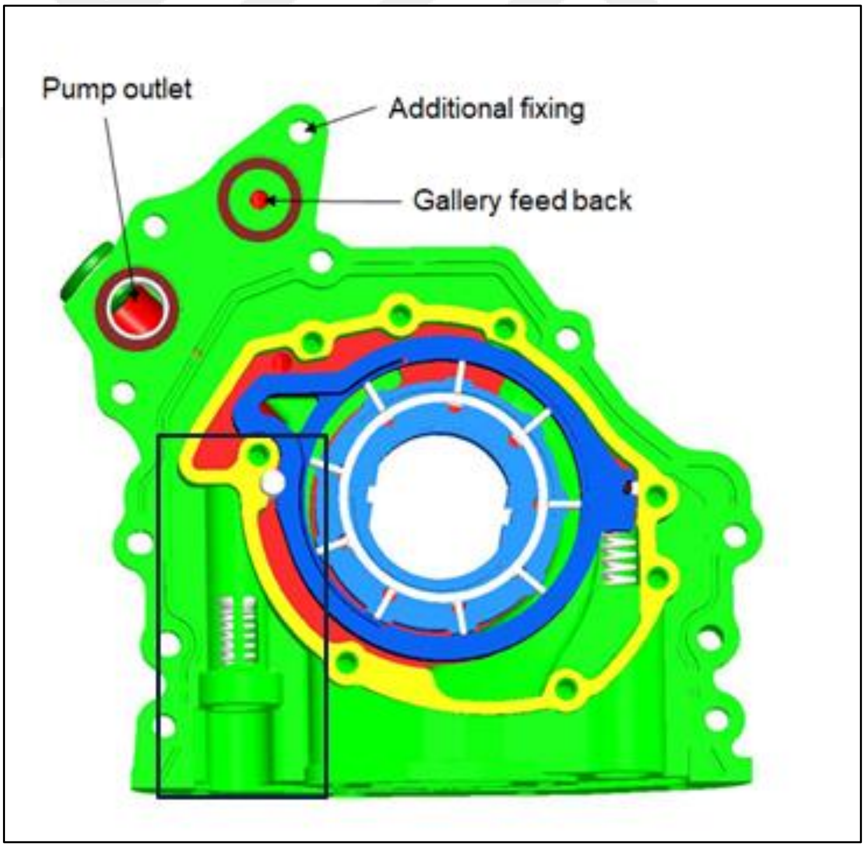

Figure 3.22 Control spring and bypass channel in an on-axis type oil pump (Stackpole International Fluid Power Solutions archive, 2018)

The amount of instantaneous oil required by the engine during operation is indicated by a red line in the graph. Constant volume pumps supply more oil that the engine needs when the rpm become higher. This case is represented by the blue line in the graph in Figure 3.23 Such a working condition increases both fuel consumption and loads too much stress on motor. For this reason, variable volume oil pumps have been developed that respond instantly and can self-adjust by engine revolution, thus energy gain is obtained. Point in the Figure 3.23 is the point at which the slope of the line changes. It is possible to obtain the regulation point with different designs.

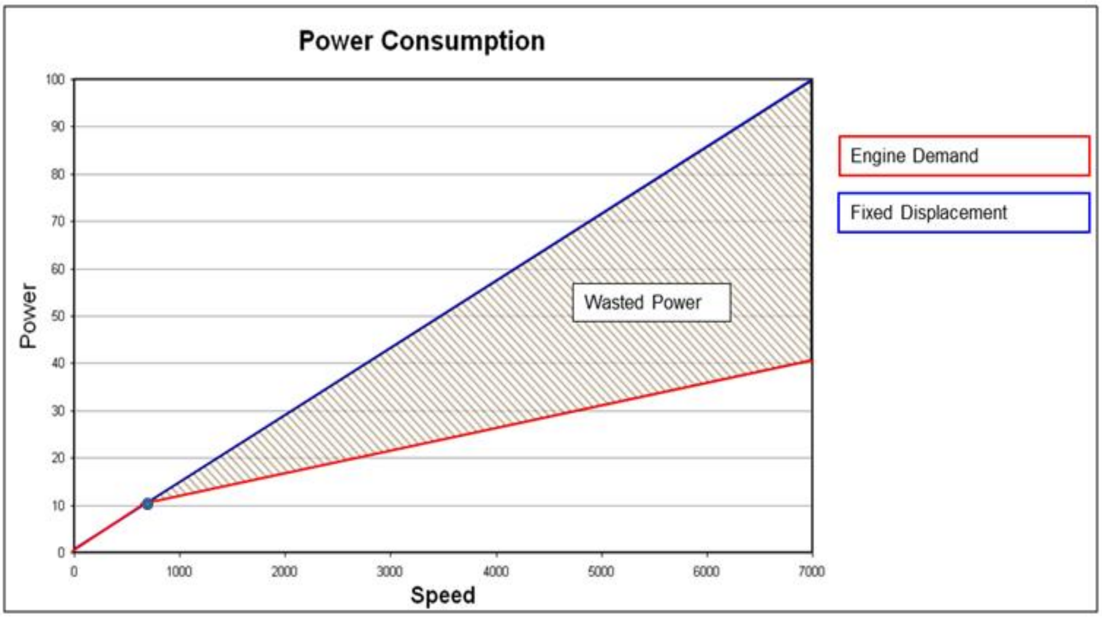

Figure 3.23 Graphical demonstration of the power consumption and speed for fixed pumps (Stackpole International Fluid Power Solutions archive, 2018).

Different oil pump models are developed. Variable volume oil pumps are divided into three groups according to their operating principle:

- Single Stage Variable Vane Pump
    - Outlet Pressure Controlled Variable Vane Pump
    - Gallery Pressure Controlled Variable Vane Pump
- Two Stage Variable Vane Pump
- Fully Variable Vane Pump

3.4.1.1 Single Stage Variable Vane Pump

Single stage variable vane pump have power savings. The bevel of the green line changes at the regulation point and yellow marked region shows the power savings in Figure 3.24.

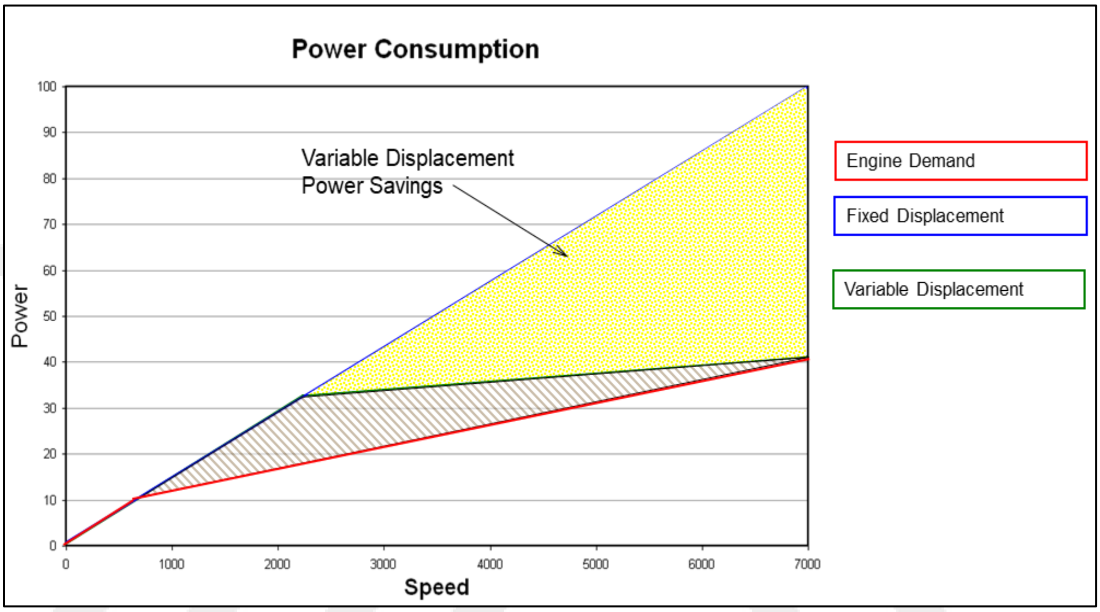

Figure 3.24 Graphical demonstration of the power consumption and speed for variable pumps (Stackpole International Fluid Power Solutions archive, 2018)

The point of the bevel of the green line changed is named as regulation point. This regulation point is arranged by the movement of the slide. (cam ring) Pressure and engine speed graph is given in Figure 3.25.

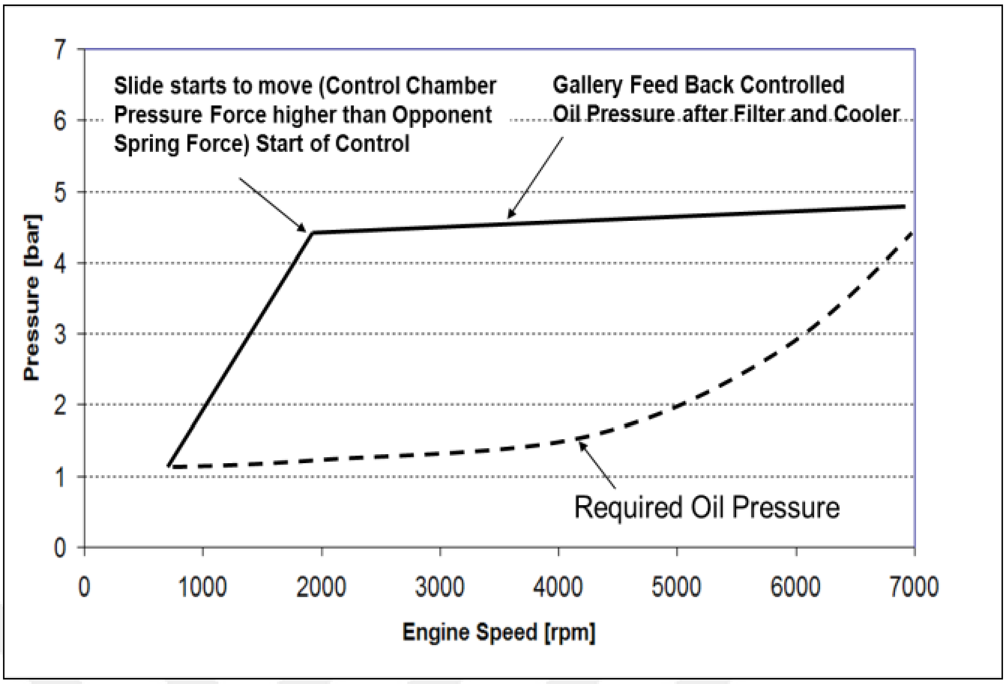

Figure 3.25 Pressure changes inside of the pump during engine working (Stackpole International Fluid Power Solutions archive, 2018)

3.4.1.1.1 Outlet Pressure Controlled Single Stage Variable Vane Pump. When the oil pump feeds the necessary oil to the relevant points, if the pressure at the oil pump outlet is high, the oil returns to the oil pump through the channels in itself. This condition pressures the slide part and compresses the spring.

How much the slide will compress the arc is directly proportional to the amount of energy it will impose on the pressure between the rotor and the slide is reduced and this is called regulation.

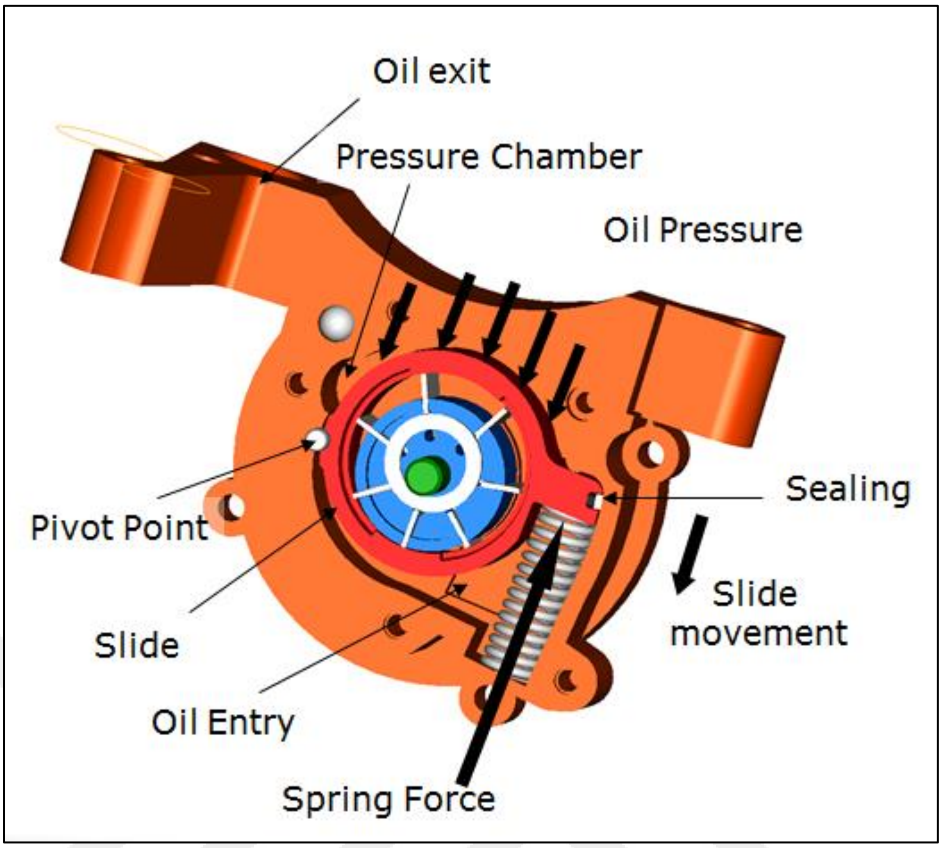

Figure 3.26 Single stage variable vane pump outlet pressure controlled (Stackpole International Fluid Power Solutions archive, 2018)

The oil pressure at pump exit pushed the slide against the spring. If the pressure is too high, the slide moves and reduces the pump displacement and the oil flow. The oil pressure is controlled by the spring force.

Outlet pressure controlled variable vane pump-oil pressure and flow over speed range graph is given Figure 3.27.

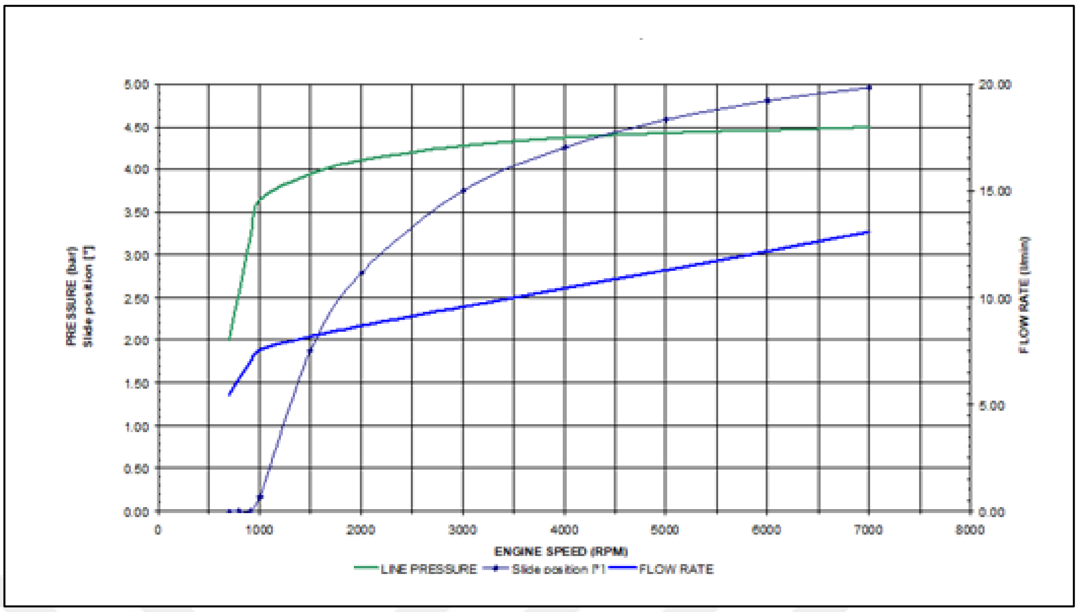

Figure 3.27 Outlet pressure controlled variable vane pump-oil pressure and flow over speed range (Stackpole International Fluid Power Solutions archive, 2018)

Advantages of single stage outlet pressure controlled variable vane pumps are:

- Simple and robust design principle for variable displacement
- Simple housing design, outlet pressure working directly on the slide
- Cost effective

Disadvantages of single stage outlet pressure controlled variable vane pumps:

- Pump pressure level needs to be set to worst case conditions, this means at low temperature, to cover the pressure losses through filter/cooler to main gallery and secure minimum pressure level. At high temperature, pressure level could be higher than necessary.

3.4.1.1.2 Gallery Pressure Controlled Single Stage Variable Vane Pump. The pressure from the engine gallery is transmitted via a control channel into the pump control chamber and exerts a force on the slide. As soon as the exerted force is greater than the opponent spring force, the slide starts to move around the pivot point and decreases flow and pressure. So a closed loop control of the gallery pressure is a function of the spring. Therefore, the system is controlled mechanically/hydraulically with the spring as a control element and the gallery pressure as actuator. General aspect of the components of gallery pressure controlled variable vane pump is given in Figure 3.28 and forces inside this pump is given in Figure 3.29 and Figure 3.30.

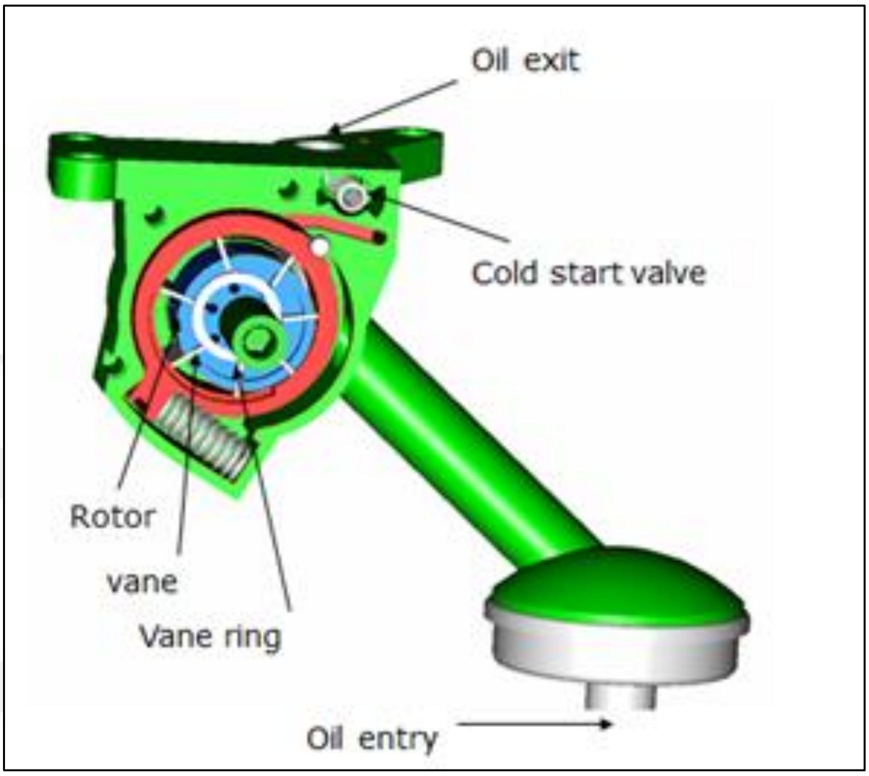

Figure 3.28 Demonstration of the components inside the gallery pressure controlled variable vane pump (Stackpole International Fluid Power Solutions archive, 2018)

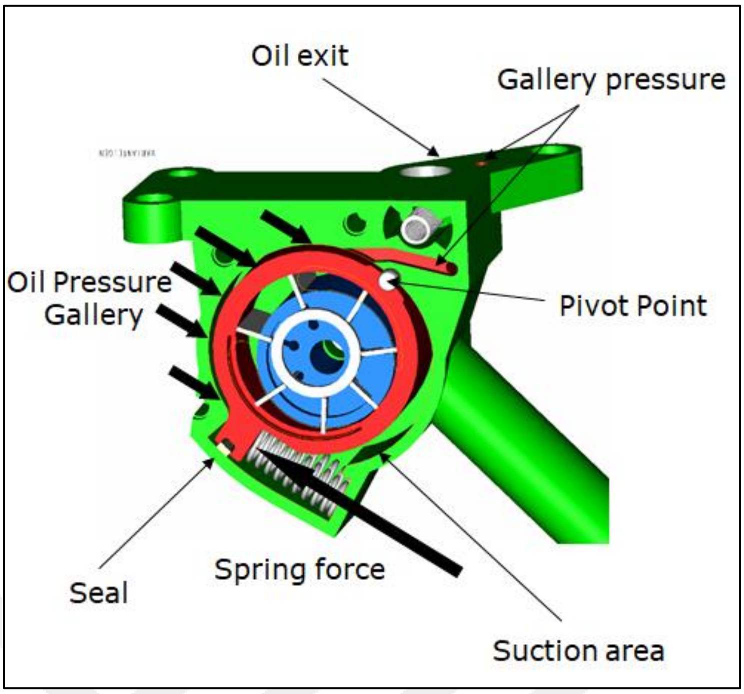

Figure 3.29 Forces inside the single stage gallery pressure controlled variable vane pump with related zones (Stackpole International Fluid Power Solutions archive, 2018)

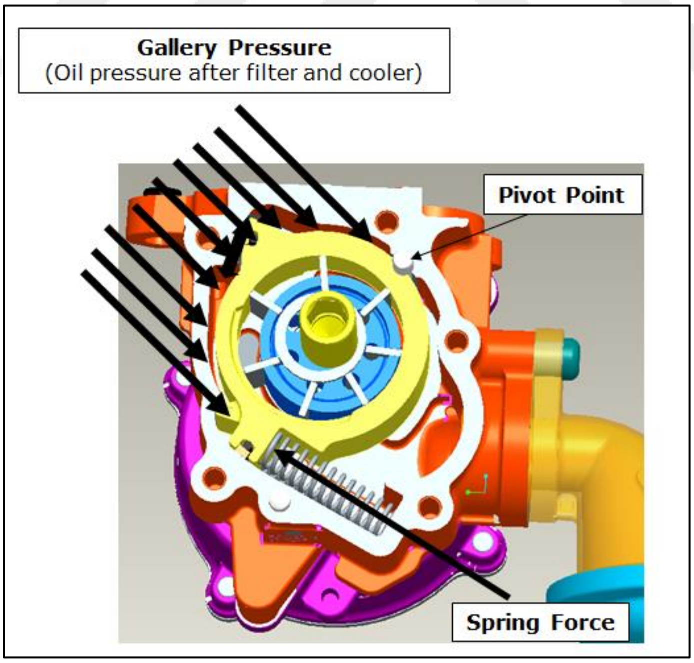

Figure 3.30 Forces inside the single stage gallery pressure controlled variable vane pump (Stackpole International Fluid Power Solutions archive, 2018)

Schematic view of working principle of gallery pressure controlled variable vane pump is given Figure 3.31

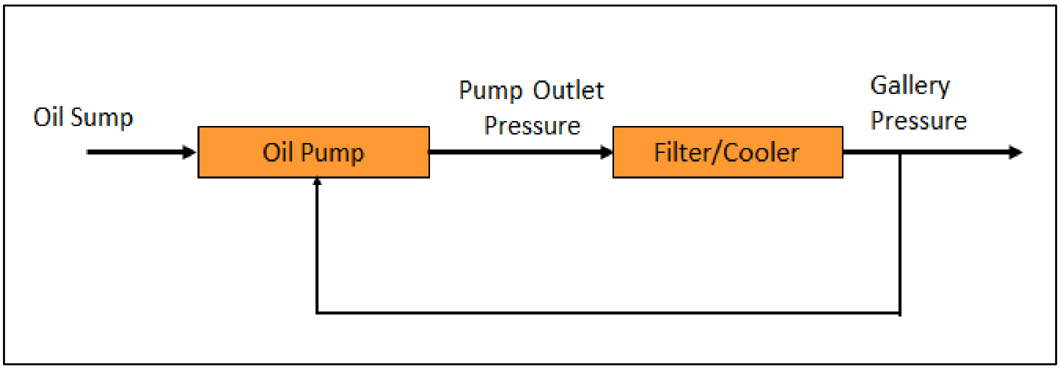

Figure 3.31 Schematic view of oil pump control at gallery pressure controlled variable vane pump. (Stackpole International Fluid Power Solutions archive, 2018)

Functional parameters of an oil pump such as flow rate, volumetric efficiency, line pressure etc is given at the graph in Figure 3.32

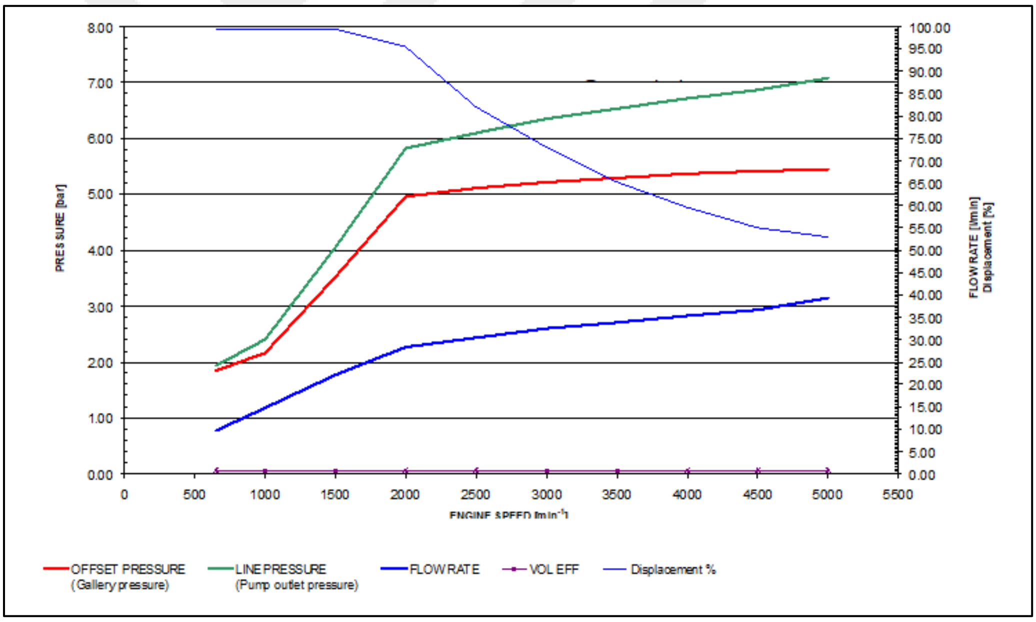

Figure 3.32 Gallery pressure controlled variable vane pump- pressure and flow over engine speed (Stackpole International Fluid Power Solutions archive, 2018)

3.4.1.2 Two Stage Variable Vane Pump

The variable vane pump can be adjusted to two pressure levels by using two control channels, feeding two chambers working on the slide. The pressure level can be significantly reduced at low engine speed and changed to a higher pressure level when required. The system is controlled by an electric valve, connected to the engine ECU (Electronic Control Unit). Advantage of this is a significant reduction of drive torque at low engine speed.

The pressure from the engine gallery is transmitted via a control channel into the pump control chamber and exerts a force on the slide. As soon as the exerted force is greater than the opponent spring force, the slides starts to move around the pivot point and decreases flow and pressure. So a closed loop control of the gallery pressure is a function of the spring. Therefore, the system is controlled mechanically/hydraulically with the spring as a control element and the gallery pressure as actuator. To adopt the pressure at low engine speed closer to the required pressure, a second switch chamber is feed also with gallery pressure oil and reduces again the oil flow and pressure by turning the slide. Forces inside the two stage gallery pressure controlled variable vane pump is given in Figure 3.33.

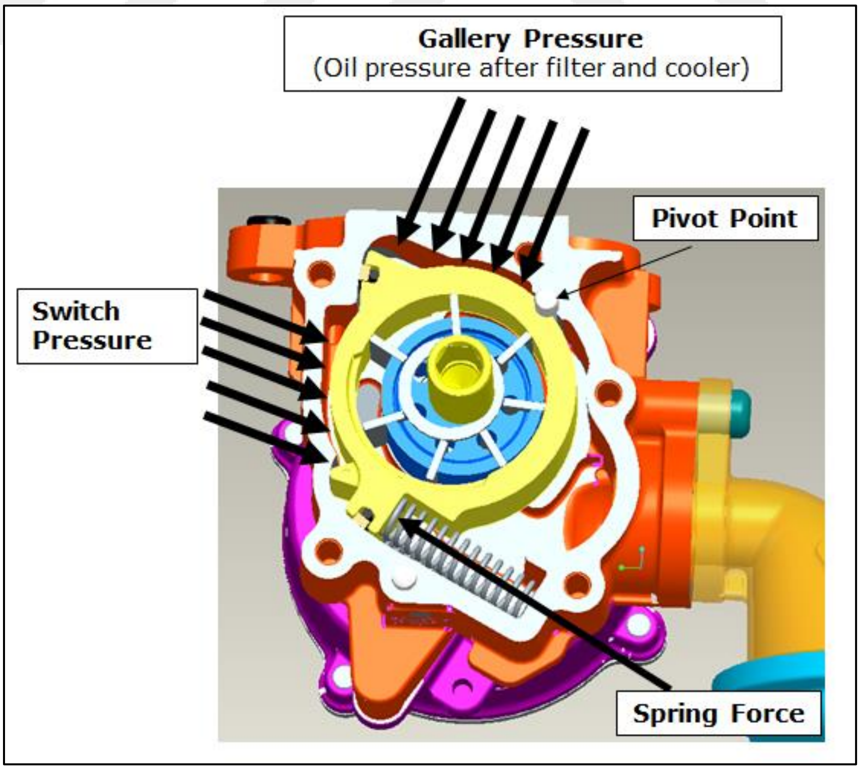

Figure 3.33 Forces inside the two stage gallery pressure controlled variable vane pump (Stackpole International Fluid Power Solutions archive, 2018)

The graph of the amount of oil that the two-stage variable volume oil pump supplies to the system as required by the engine is indicated by the orange line in the graph that is given in Figure 3.34. As seen, there are two regulating points where the slope of the curve changes.

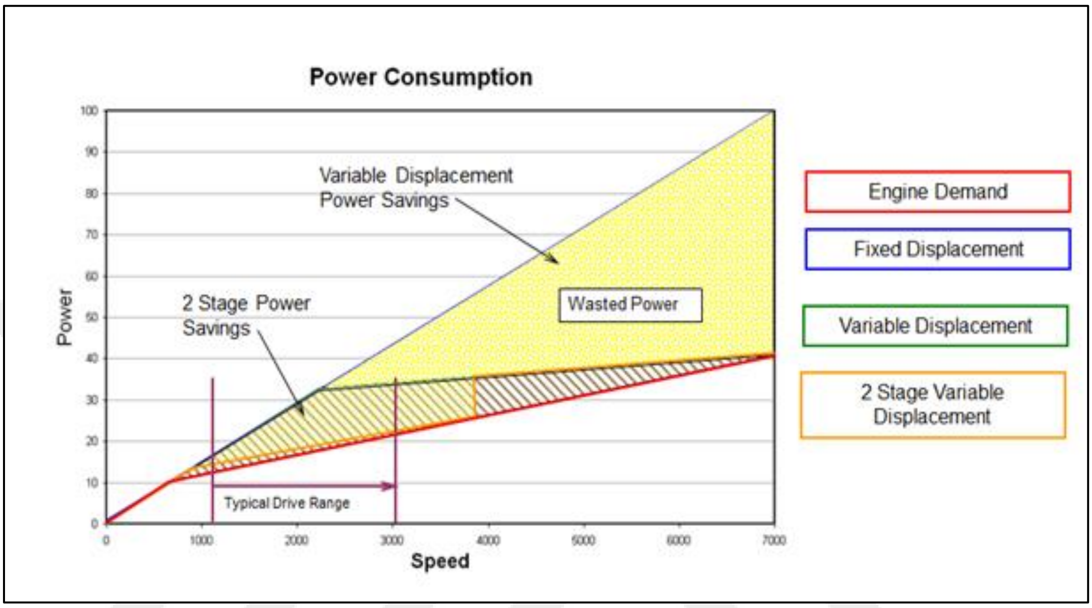

Figure 3.34 Graphical demonstration of the power consumption and speed for two stage variable pumps (Stackpole International Fluid Power Solutions archive, 2018)

The effect of the two chambers of the pump on the pressure change is shown at the graph in Figure 3.35 and Functional parameters of the pump is given in Figure 3.36

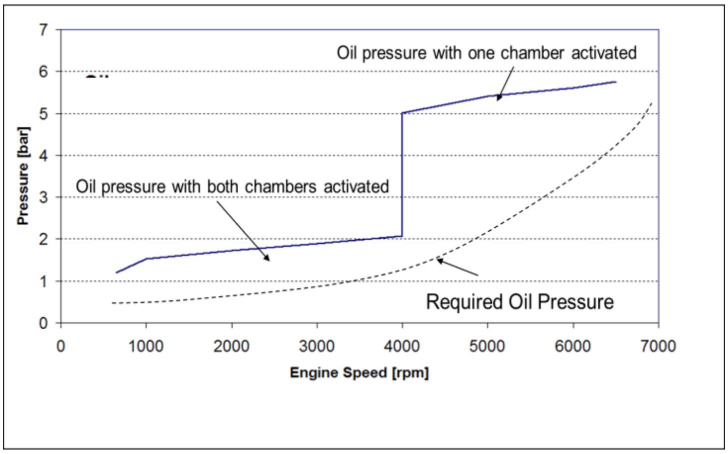

Figure 3.35 Pressure- engine speed relation graph of two stage variable vane pump (Stackpole International Fluid Power Solutions archive, 2018)

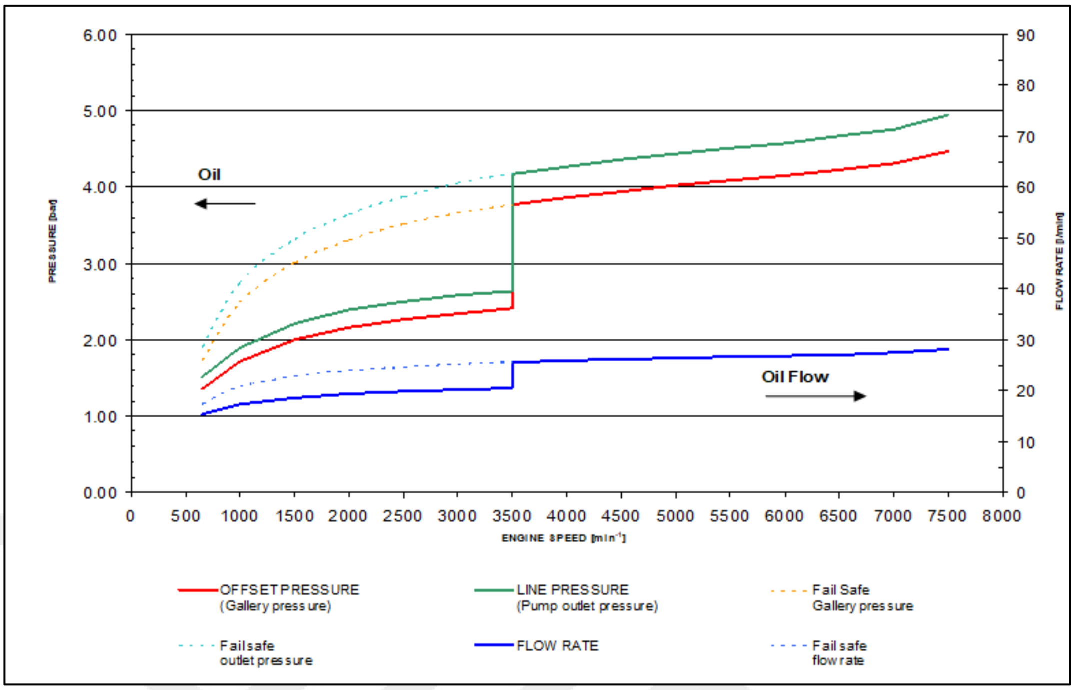

Figure 3.36 Graphical demonstration of functional parameters of a pump (Stackpole International Fluid Power Solutions archive, 2018)

Schematic view of working principle of two-stage gallery pressure controlled variable vane pump is given in Figure 3.37.

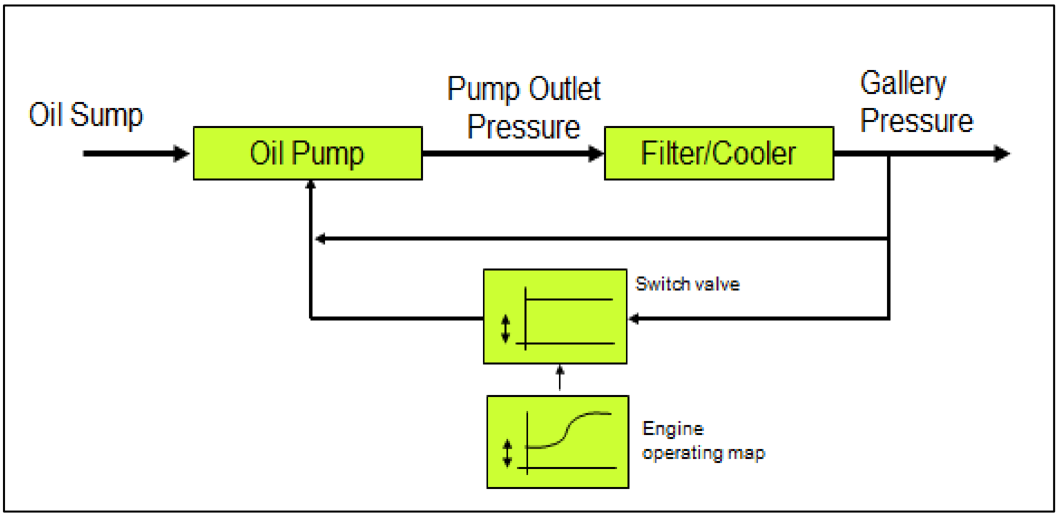

Figure 3.37 Schematic view of oil pump control at two-stage gallery pressure controlled variable vane pump (Stackpole International Fluid Power Solutions archive, 2018)

Advantages of two stage gallery pressure controlled variable vane pump are:

- Significant improvement in drive torque at low engine speed due to reduced pressure level compared to standard variable pump
- Robust control system in the pump, electric valve can be integrated in the pump housing, into the gallery feedback channel, or into the engine block

Disadvantages of two stage gallery pressure controlled variable vane pump are:

- Additional electric valve required, connection to engine ECU needed
- Cost increase due to electric valve.

3.4.1.3 Fully Variable Vane Pump

Pressure from the gallery is transmitted via a proportional valve into the control chamber and exerts a force on the slide. As soon as the exerted force is greater than the opponent spring force, the slides starts to move around the pivot point and decreases flow and pressure. So the gallery pressure is a function of the engine operating map. Therefore the control is electrically/mechanically/hydraulically with the operating map as control element and the chamber pressure as actuator. The pressure can be adjusted on the required particular pressure above the smallest gallery pressure. Fully variable vane pump with proportional control is shown in Figure 3.38 and schematic view of working principle of this pump is given in Figure 3.39.

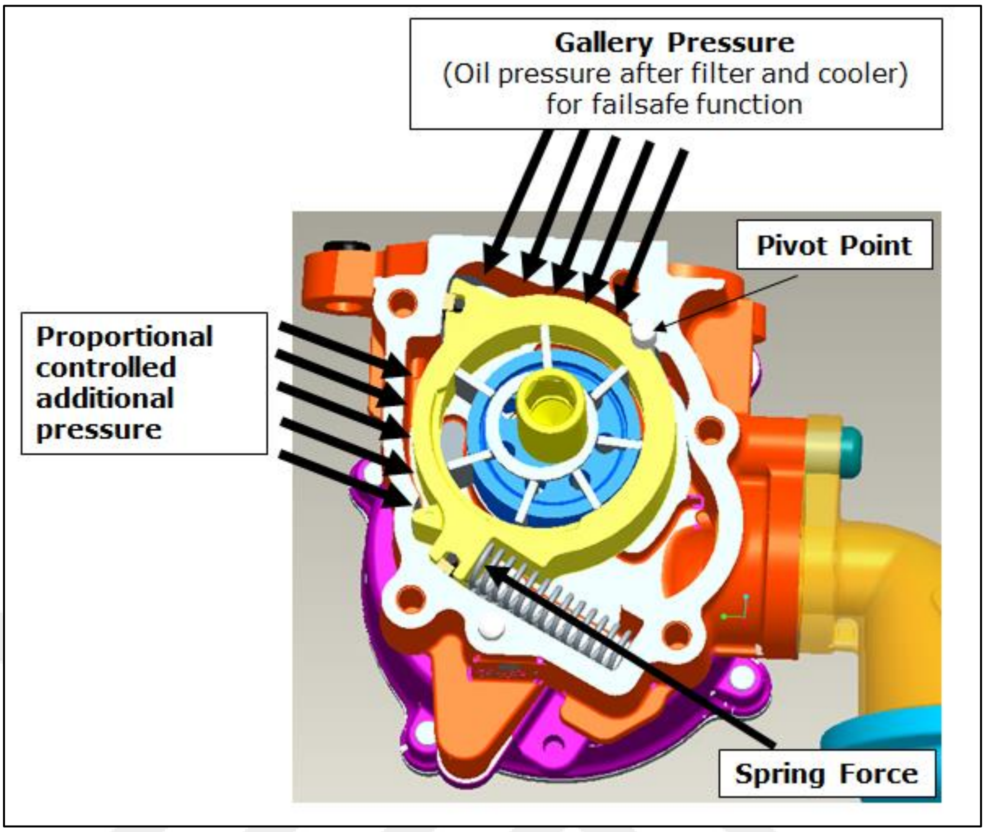

Figure 3.38 Fully variable vane pump with proportional control (with gallery feedback for fail safe) (Stackpole International Fluid Power Solutions archive, 2018)

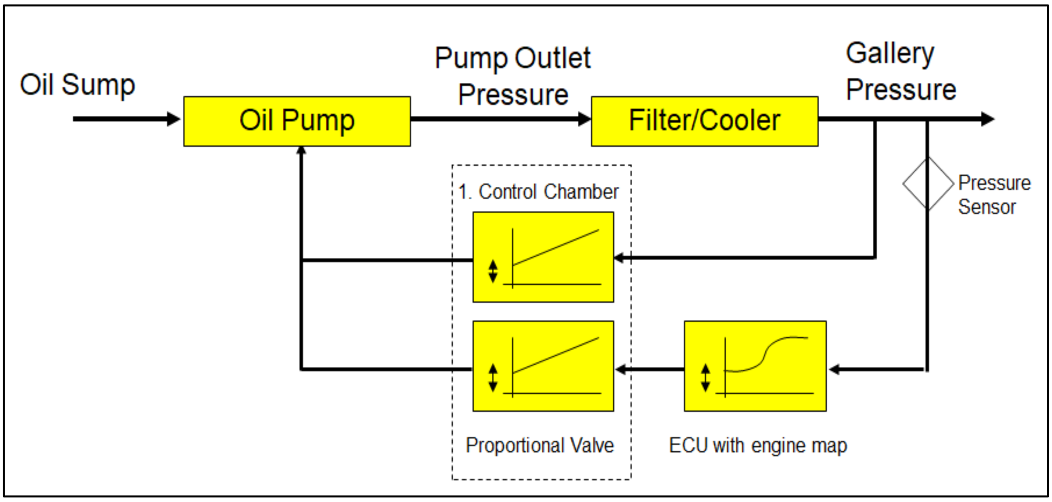

Figure 3.39 Schematic view of fully variable oil pump with proportional valve (Stackpole International Fluid Power Solutions archive, 2018)

The 1st control chamber with gallery pressure controls mechanically/hydraulically the oil pump on a constant maximum pressure level. The proportional valve controls the pressure in the 2nd control chamber controlled by an electrical current according to the engine map, by using the pressure sensor value as input. The fully variable oil pump is based on the principle of a two stage gallery feedback controlled oil pump. The 1st chamber has gallery pressure, working against a defined spring force and it controls the pump pressure on a level just above the minimum required engine pressure. The 2nd control chamber is fed by an additional pressure coming from the proportional valve and it moves the slide to get the required output pressure. The proportional valve is controlled by an electrical current generated in the Electrical Control Unit (ECU) and controlled by the engine map. With a failure in the ECU, the pump works like a single stage gallery feedback controlled oil pump. Different types of control can be realized.

There are three control variants can be listed as follows:

- Open loop with proportional controlled chamber pressure
- Closed loop with proportional controlled chamber pressure
- Closed loop with proportional controlled chamber pressure in the 2nd control chamber and gallery pressure in the 1st control chamber

First option is the cheapest solution, and no pressure sensor required. Engine map in ECU controls the pump pressure proportional to the control parameters. No adoption to manufacturing tolerances and pressure loss over life time is required.

At the second option, a pressure sensor in the gallery channel gives the actual pressure to the ECU. The engine map in the ECU controls the pump outlet pressure proportional to the control parameters and the actual gallery pressure. Full compensation of manufacturing tolerances and lifetime caused that the pump does not work properly.

Third option can be caused a failsafe function, the pump acts like single stage gallery feedback controlled oil pump as explained in second option.

Proportional valve gives a similar pressure curve that the engine needs at fully variable vane pump; this situation is given in Figure 3.40

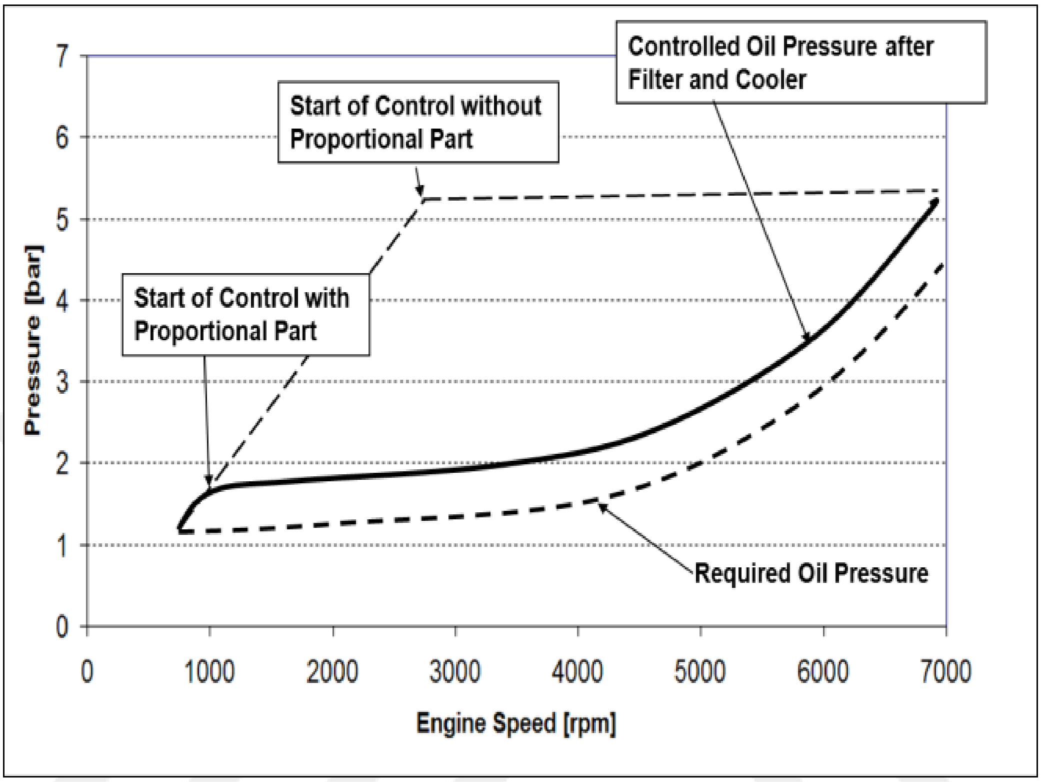

Figure 3.40 Fully variable vane pump with proportional valve (Stackpole International Fluid Power Solutions archive, 2018)

3.5 Design of Oil Pumps

Here the design of two-stage variable vane pump will be discussed .The basis for the design of any pump are the functional requirements given by the engine. These describe the pump-specific parameters that are necessary for the first determination.

Functional Requirements:

- Speed Range : from n_{engine}=500 (1/min) to n_{engine} =5000 (1/min)
- Translation: i= n_{pump}/n_{engine} =1
- Fully variable adjustable flow rate with regulation to almost zero delivery of 10% of the maximum delivery rate :6 (1/min) at n_{Engine}=5000 (1/min)

- Minimum hot idle flow rate: 6.6 (1/min) at speed n_{Engine}=500 (1/min)

The first parameter to be determined for each pump is delivery volume per revolution. For this; the minimum hot idle condition is decisive. Since the viscosity decreases with increasing temperature, the oil becomes more fluid and the internal leakage of the pump increases. Because of the oil flow losses in the pump, volumetric efficiency decreases. Therefore, the hot idle describes the most unfavorable operating point, in which the operational safety must be given. The variation of the kinematic viscosity of the oil with respect to temperature is given in Figure 3.41.

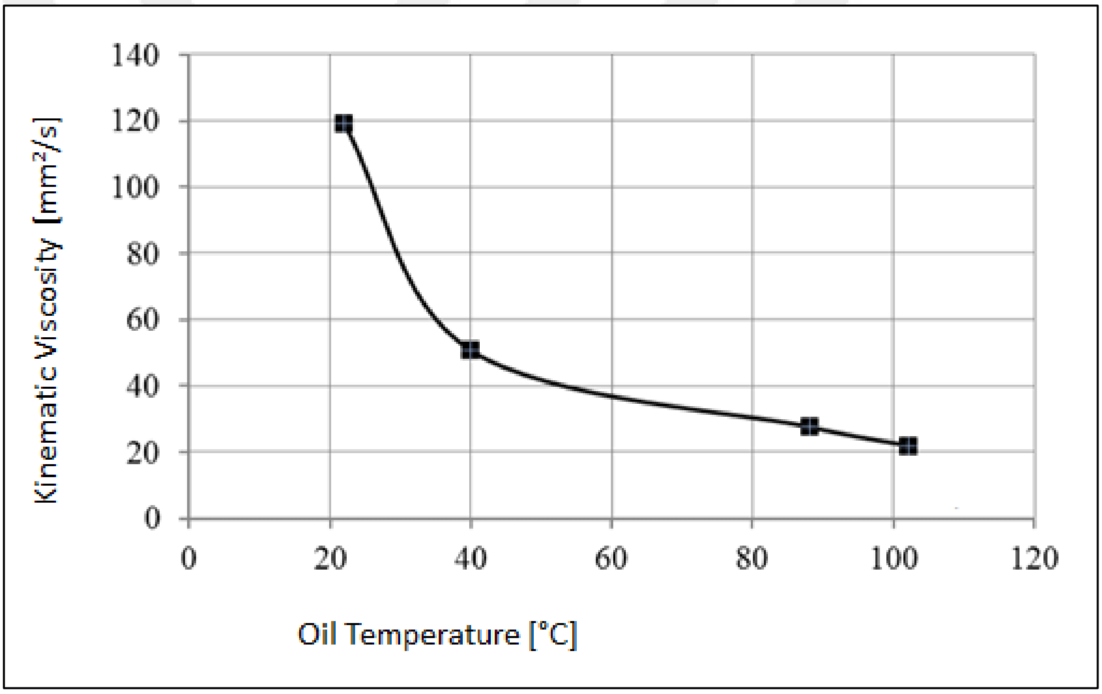

Figure 3.41 Dependence of the viscosity on the temperature (The viscosity class: 5W30) (Böwing, 2009)

For the engine at 6.6 1/min at speed 500 1/ min relative oil pressure 1.1 bar and oil temperature: 140 °C and oil type: 5W30 is decisive.

The pressure 1.1 bar, represents the minimum pressure that ensures the function of the hydraulic elements. The delivery volume per-revolution is calculated as follows in the hot idle condition.

$$V = \frac{Q}{n_{pump} * \eta_{vol}} \tag{3.12}$$

It is choosen here a volumetric efficiency 70%.

$$V = \frac{6.6 * \frac{1}{min}}{500 * \frac{1}{min} * 0.7} = 18.86 \frac{cm^3}{rpm} \tag{3.13}$$

This delivery volume applies to a balanced vane pump with maximum eccentricity of slide and rotor. Based on these specifications, geometrical dimensions are created to maintain the delivery volume with 19 cc/ revolution. Height of rotor, outer diameter of rotor, inner diameter of rotor, number of vanes and vane width are the design parameters of the rotating group.

The maximum delivery volume is achieved when the slide and rotor are the most eccentric to each other. Maximum eccentricity of slide and rotor is given in Figure 3.42.

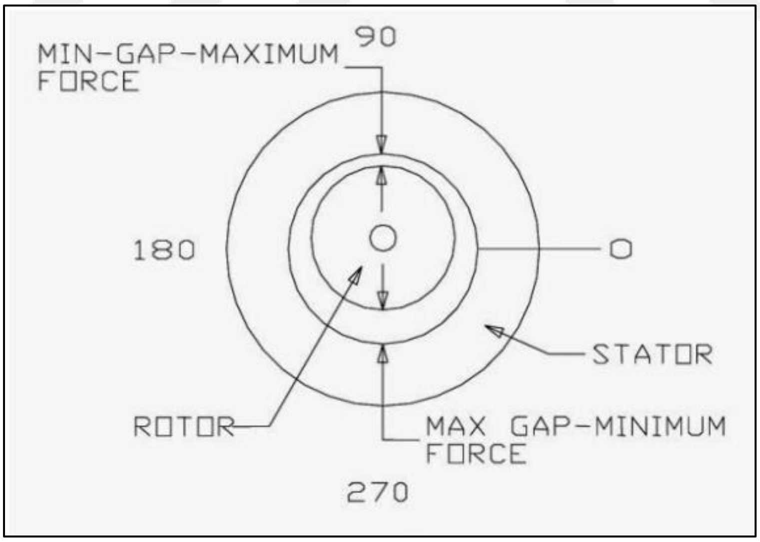

Figure 3.42 Maximum eccentirictу of slide and rotor

In the low pressure stage, a pressure of 1.5 bar is necessary for the function of the engine. To determine the control spring, spring rate and pre-load forces in the pump are considered. Forces inside the pump is shown in Figure 3.43.

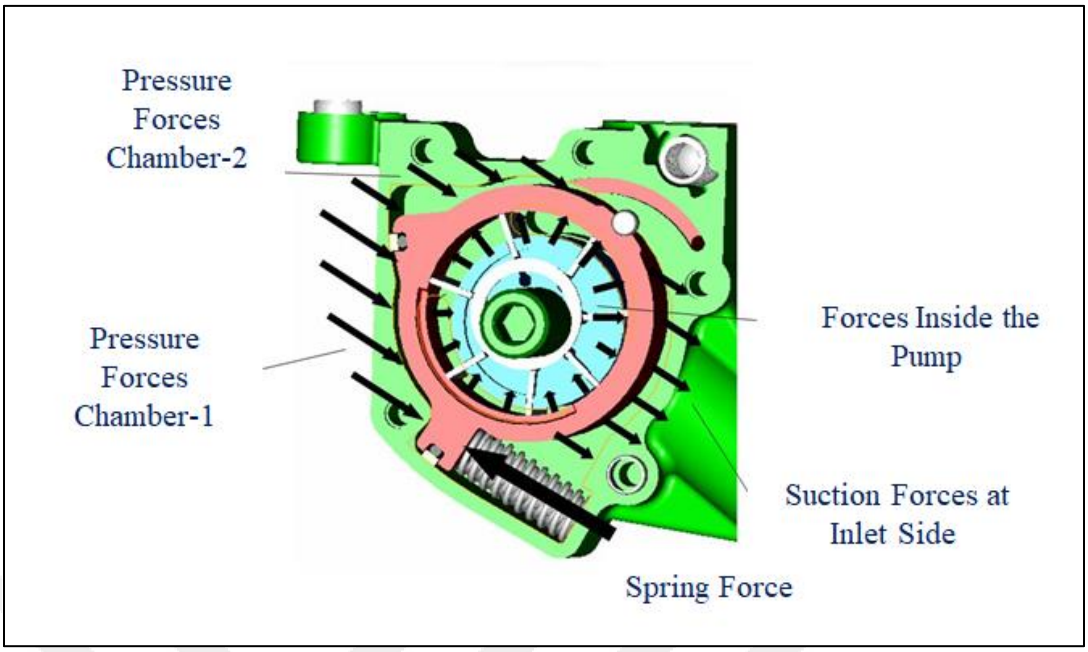

Figure 3.43 Demonstrations of forces in the pump

Due to installation space date, the maximum lever arm and the lever arm of the spring are decisive for low pressure stage. The moment equilibrium is shown below:

- Moment of the spring

2nd moment of the control chamber

$$\sum M = M_{chamber,LPL} - M_{spring} = 0 \quad (3.14)$$

$$M_{chamber,LPL} = F_{chamber,LPL} * \frac{1}{2} * l_{chamber,LPL} \quad (3.15)$$

$$F = P * A \quad (3.16)$$

$$M_{chamber,LPL} = P_{LPL} * A_{Proj} * \frac{1}{2} * l_{chamber,LPL} \quad (3.17)$$

$$A = h * l \quad (3.18)$$

$$M_{chamber,LPL} = P_{LPL} * h_{rotor} * l_{chamber,LPL} \quad (3.19)$$

$$M_{spring} = F_{spring} * l_{spring} \quad (3.20)$$

$$\sum M = P_{LPL} * h_{rotor} * \frac{1}{2} * l^2_{chamber,LPL} - F_{spring} * l_{spring} = 0 \quad (3.21)$$

$$P_{LPL} * h_{rotor} * \frac{1}{2} * l_{chamber,LPL}^{2} = F_{spring} * l_{spring} \tag{3.22}$$

$$F_{spring} = \frac{P_{LPL} * h_{rotor} * \frac{1}{2} * l_{chamber,LPL}^{2}}{l_{spring}} * (F_1) \tag{3.23}$$

With the above equation; spring pre-load force can be calculated at the low pressure level situation. Until reaching 1.5 bar in the low pressure stage, the adjusting ring does not move. With this preload, with the same calculation of the lever arm for the high-pressure stage can now be determined.

$$\sum M = P_{HPL} * h_{rotor} * \frac{1}{2} * l_{chamber,HPL}^{2} - F_{spring} * l_{spring} = 0 \tag{3.24}$$

$$P_{HPL} * h_{rotor} * \frac{1}{2} * l_{chamber,HPL}^{2} = F_{spring} * l_{spring} \tag{3.25}$$

$$l_{chamber,HPL} = \sqrt{\frac{F_{spring} * l_{spring}}{P_{HPL} * h_{rotor} * \frac{1}{2}}} \tag{3.26}$$

The regulation is achieved by a parallel connection of two springs. Spring rate can be taken based on empirical relations.

Up to now, the design of the pump has been based on a hot idling (stroke volume) as well as the diversion points for HDS and NDS (spring preload). To match the appropriate bias of the spring and the spring rate, the associated geometric data of the spring, e.g. free length, it is necessary to consider another point in the map.

It is required minimum promotion of 10% of the maximum value. This corresponds to a delivery volume of 1.89 l/min and an adjustment angle of 0.57°.The displacement of the adjusting ring and thus the compression of the spring results from below relation (Böwing, 2009).

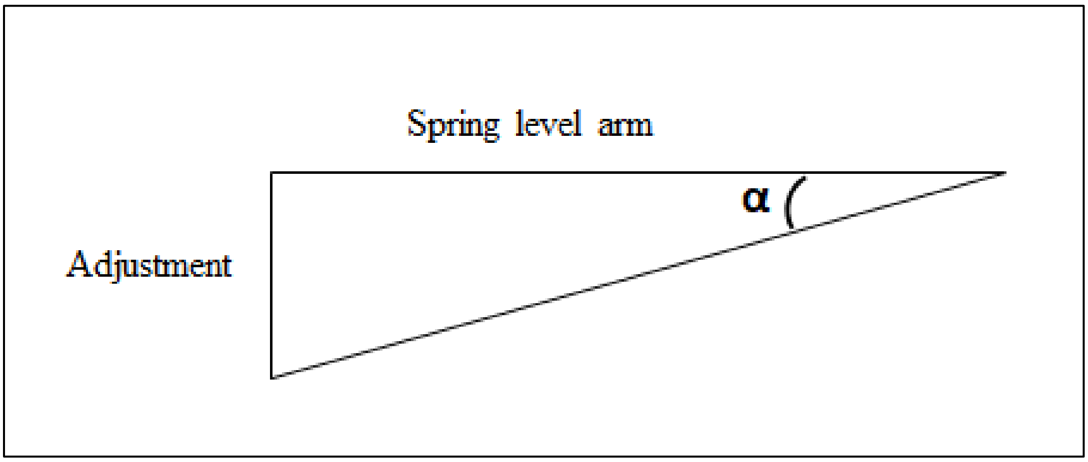

Figure 3.44 Relation between spring level arm and adjusment.

$$tan\alpha = \frac{\Delta l}{l_{spring}} \tag{3.27}$$

$$\Delta l = l_{spring} * \tan(\alpha_{max} - \alpha\ ) \tag{3.28}$$

It follows with the spring rate:

$$\Delta F = \Delta l * n_{spring} * R_{spring} \tag{3.29}$$

$$F_2 = F_1 + \Delta F \tag{3.30}$$

From the calculated forces and length difference between the free length of the spring (l_0) and the two operating points can be determined.

$$\Delta l_1 = \frac{F_1}{2.R} \tag{3.31}$$

$$\Delta l_2 = \Delta l_1 + \Delta l \tag{3.32}$$

One of the parameters that has a high impact on pump performance is the chambers, which are located on the body, cover and slide, called porting, to create the closed volume within the pump. The term of porting includes the closing times of the suction and the pressure conveying chamber. The most determinant factor is the position of the porting at both cover and housing side. Suction and pressure portings are shown in Figure 3.45.

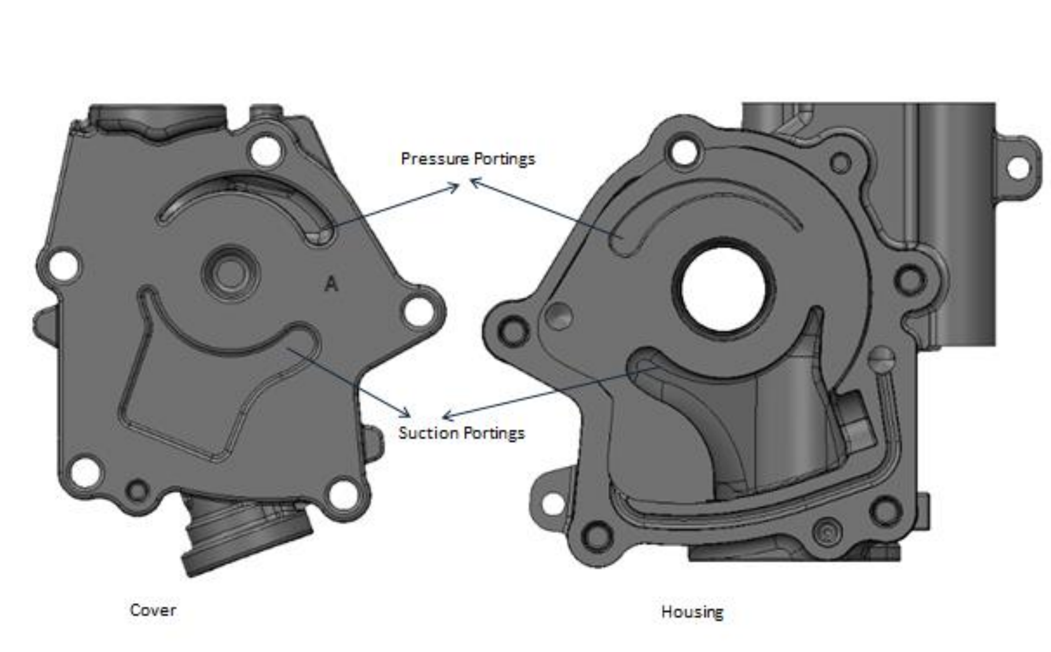

Figure 3.45 Position of the suction and pressure portings within the pump (Stackpole International Fluid Power Solutions archive, 2018)

Between portings there are transport and counter transport chambers. A displacement pump consists of a closed volume that includes transport chambers. For this reason, the angles between kidneys is always bigger than the angle that describes the conveying chamber that is surrounded by the vanes.

The angle enclosed by the vanes:

$$\alpha = \frac{360^0}{n_{vane}} = \frac{360^0}{7} = 51.43^0 \tag{3.33}$$

Angle between vanes are demonstrated in Figure 3.46.

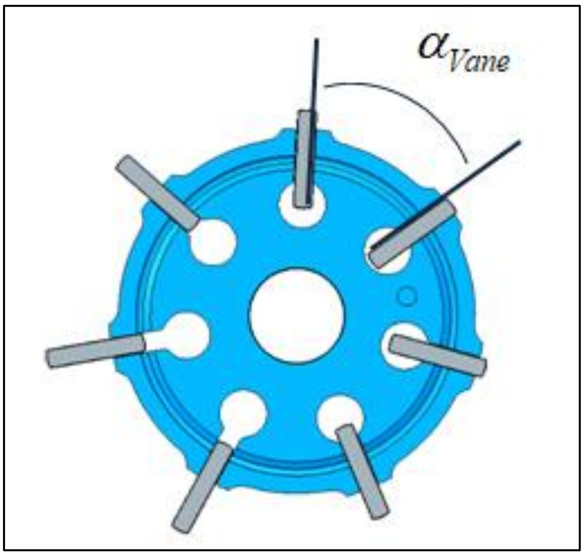

Figure 3.46 Angles between vanes in the rotating group of the pump (Personal archive, 2018)

Angles between suction and pressure portings in the cover are given in Figure 3.47.

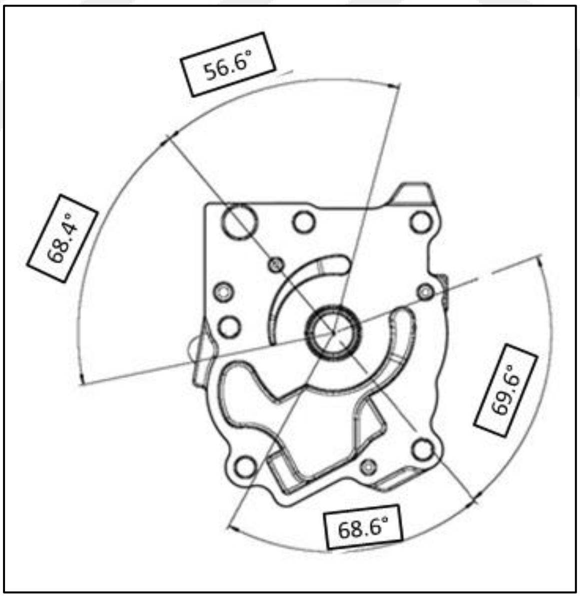

Figure 3.47 Angles of the suction and pressure portings in the cover (Böwing, 2009)

The pump is not always in the maximum eccentricity position during operation. Therefore; in order to distribute the forces on the pump, ports are created on the slide as shown in Figure 3.48.

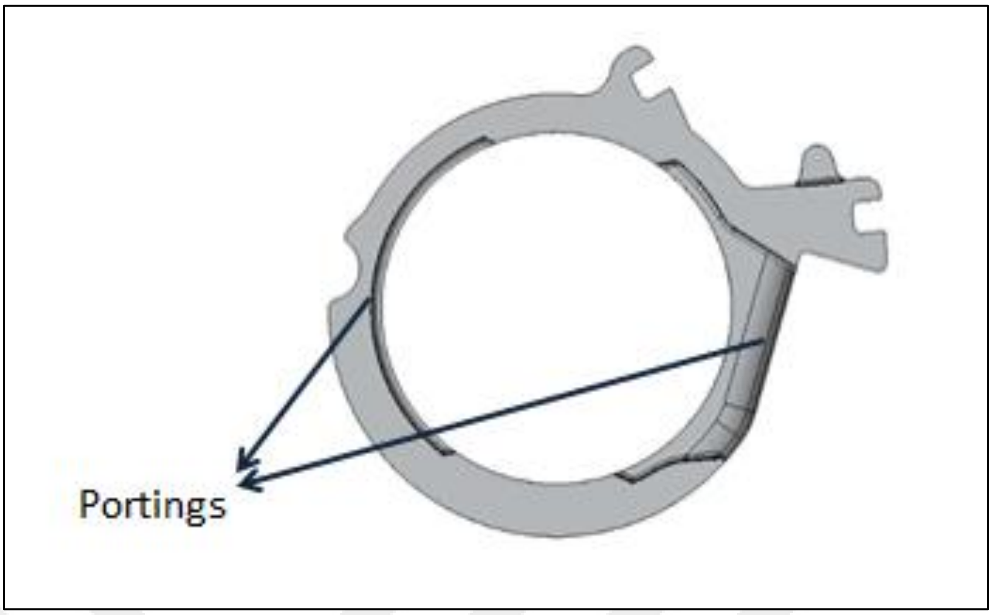

Figure 3.48 Demonstration ports in the slide (Personal archive, 2018)

When housing and slide assembled together, angles between porting is shown like that in Figure 3.49. Here, BDB means *“Bottom Dead Band”* and TDB means *“Top Dead Band”*.

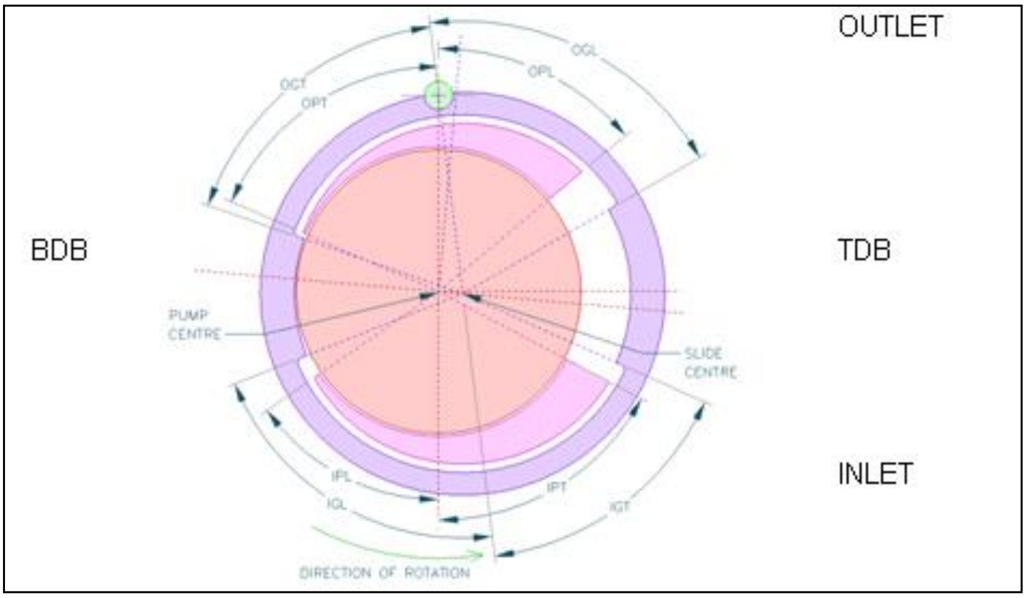

Figure 3.49 Portings angle of slide and housing (Böwing, 2009)

To investigate the effect of porting change on hydraulic pump performance and effect of porting modification on pressure pulses below modifications are done on the porting:

- Increase of opening time of inlet area of 10°
- Decrease of opening time of outlet area of 10°
- No chance on closing angle of pressure chambers

Table 3.1 shows the experiment set that includes the modification on the porting angles.

Table 3.1 Experiment set (Böwing, 2009)

	Description	Actual Timing Angle			Experimental Timing Angle		
		Min	Nom	Max	Min	Nom	Max
Slide	IGL: Inlet Groove Leading	71.0	71.5	72	77.7	78.2	78.7
	IGT: Inlet Groove Timing	67.9	68.4	68.9	75.3	75.8	76.3
	OGL: Outlet Groove Leading	70.1	70.6	71.1	62.8	63.3	63.8
	OGT: Outlet Groove Timing	69.6	70.1	70.6	62.1	62.6	63.1
Housing	IPL: Inlet Porting Leading	69.1	69.6	70.1	76.1	76.6	77.1
	IPT: Inlet Porting Timing	68.1	68.6	69.1	75.1	75.6	76.1
	OPL: Outlet Porting Leading	67.9	68.4	68.9	62.9	63.4	63.9
	OPT: Outlet Porting Timing	56.1	56.6	57.1	49.1	49.6	50.1

Due to the increased opening time of the sucking side, chamber filling will increase at higher engine. Decreasing the probability of cavitation and speed effects the decreases the influence of the inertia of the oil.This change can provide a loss of hydraulic efficiency at low engine speed. Gallery pressure value changing according to engine speed at different temperatures are given in Figure 3.50 and gallery pressure value at 125°C is obtained within reference limits as shown in Figure 3.51 for the actual pump.

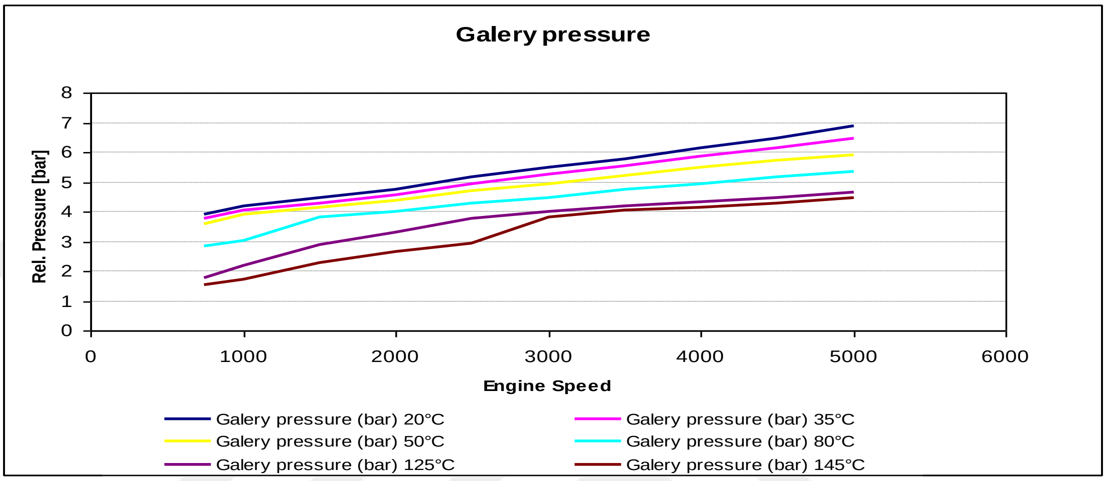

Figure 3.50 Comparison of gallery pressure at different temperatures of an actual pump (Böwing, 2009)

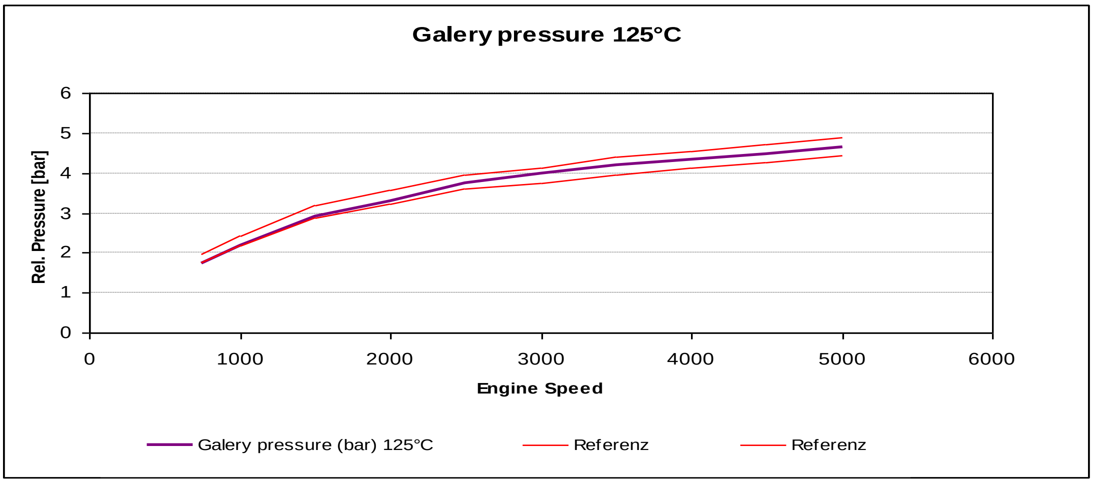

Figure 3.51 Gallery pressure of actual pump (Böwing, 2009)

Gallery pressure value changing according to engine speed at different temperatures are given in Figure 3.52 and gallery pressure value at 125°C obtained within reference limits as shown in Figure 3.53 for the experimental pump.

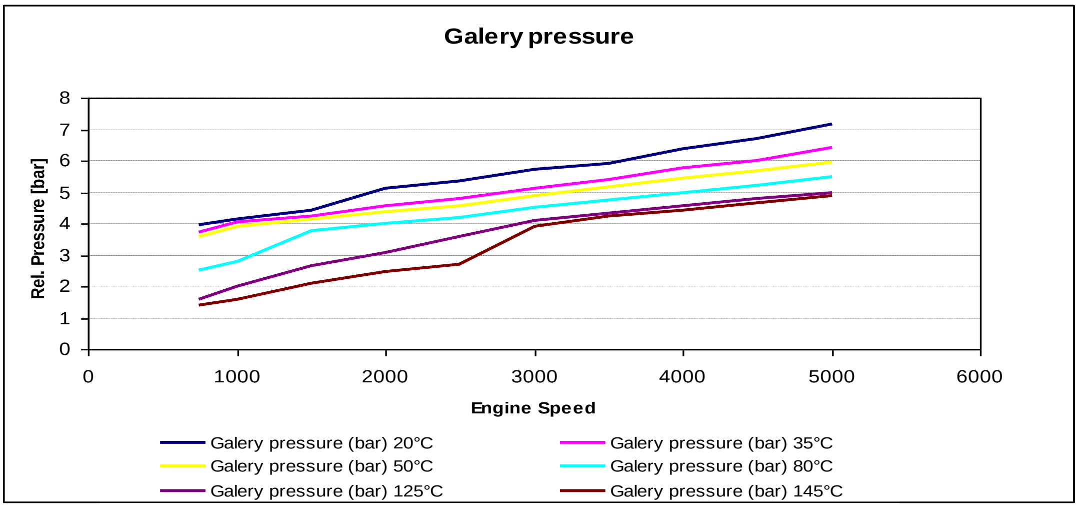

Figure 3.52 Comparison of gallery pressure at different temperatures of an experimental pump (Böwing, 2009)

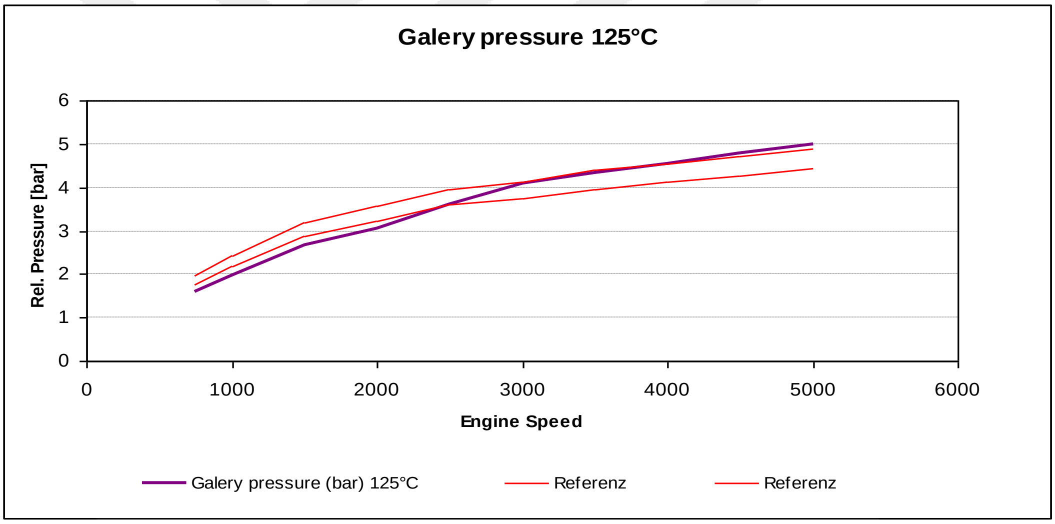

Figure 3.53 Gallery pressure of experimental pump (Böwing, 2009)

As a conclusion, the porting modifications show an influence on the hydraulic performance on the pump. As expected, a better filling of the pump in high speed range can be observed. A decrease of hydraulic efficiency can be observed at 120°C at 100 % pump displacement. The pump performance at 100% displacement drops below specification.

3.6 The Test Bench

The test bench simulates the oil circuit on the engine. The function of the pump is verified on the test bench. The following parameters can be recorded with test bench:

- Number of revolution,
- Torque,
- Temperatures
- Pressures,
- Volume flow, density and mass flow.

The following parameters can be approached in a controlled manner:

- Oil sump temperature,
- Pump outlet pressure,
- Oil flow,
- Gallery pressure,
- Number of revolution and turning direction,
- Measuring times and measuring cycles.

Figure 3.54 presents schematic view of a hydraulic funtional test rig.

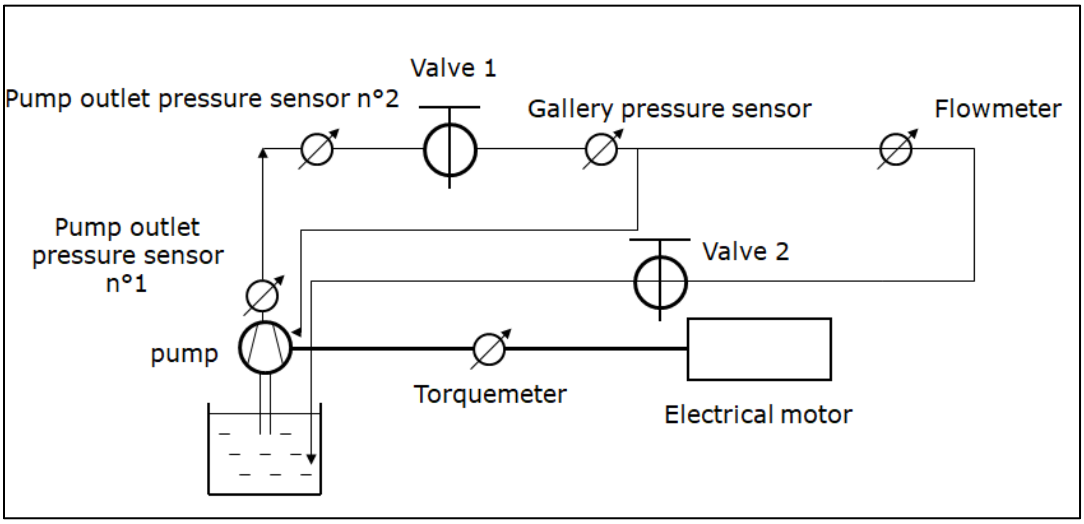

Figure 3.54 Circuit diagram of the functional test rig (Stackpole International Fluid Power Solutions archive, 2018)

The test bench consists of independent assemblies connected by pneumatic and hydraulic elements. Components of the test bench can be listed as follows:

- Drive module: Supports the drive motor and adapter plate.
- Oil conditioning: It is housed as a separate unit under the drive module and s the test oil from 20°C to 140°C.
- Frame: It covers the test bench and accommodates the measuring instrument.
- Oil pan: It provides the oil.
- PC: It is the central control of the test bench.
- Control cabinet: Contains the electrical distribution, inverter, emergency stop, etc.

An electric motor drives the pump. The tempered oil is absorbed from a free-flowing pool by the pump flowing into the control circuit. After the pump, a throttle is connected, which adjusts the pressure loss via the radiator / filter element installed on the engine.

- The valve 1 creates the pressure drop from pump outlet pressure to engine gallery, to simulate the filter / cooler pressure drop
- The valve 1 is controlled by the pump outlet pressure sensor 2.
- The valve 2 creates the engine pressure drop.
- The valve 2 is controlled by the gallery pressure sensor.
- A pressure drop between valve 1 and 2 and flow are input data's for the valve setting.
- The bench sets for each point and temperature the valves 1 and 2 to a specific percentage of opening to obtain the required pressure drop and flow
- These valve settings are defined for each speed and temperature
- Once the valve setting are defined, these valve settings are saved
- The valve settings are used as input data for all further measurements
- An output of further measurements are : pump outlet pressure, galley pressure oilflow, torque

The following metrics are included by default:

- Pump outlet pressure
- Gallery pressure
- Volume flow, formed from the mass flow and the density of the medium.
- Drive torque
- Drive speed

The test bench has a conditioning circuit which adjusts and regulates the desired temperature with the help of heating or cooling elements. The two circuits are independent of each other.

The test bench is controlled by a computer. It is possible to control according to most of the parameters listed above. Particularly noteworthy is simultaneous control of flow and pressure loss over the first throttle, since this is necessary for the creation of a customer-specific characteristic map. In addition to this pulsation measurement can be accomplished with test bench. The speed signal output by the test bench is recorded simultaneously with the pressure values. Since the PC used for the control and knife recording is not able to process the data in such a high-frequency manner, a separate measuring system is used here. The data can be report with the following graph (Figure 3.55).

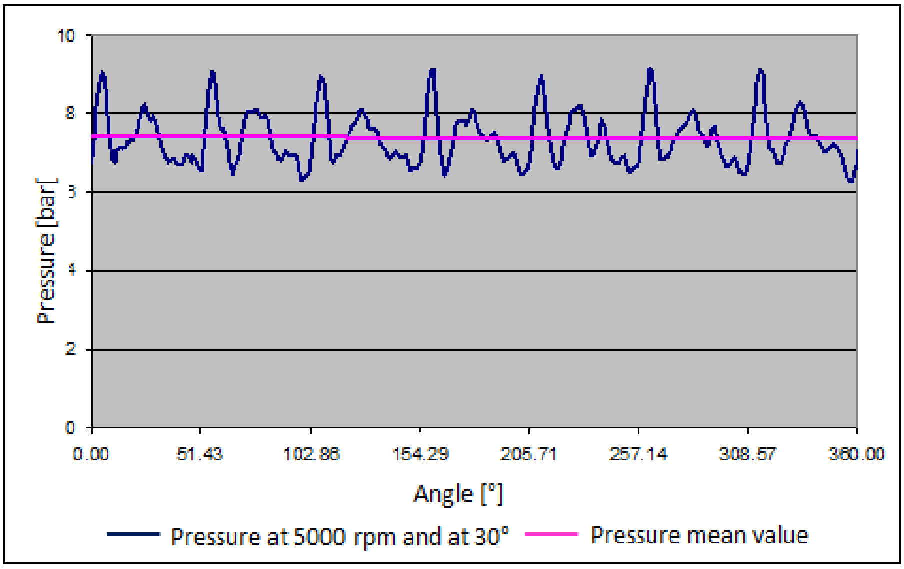

Figure 3.55 Pump output pressure pulsation over one revolution (Böwing, 2009)

CHAPTER FOUR
THICKNESS DETERMINATION OF OIL PUMP COVERS USING WITH FINITE ELEMENT METHOD

4.1 Introduction

In this section, thickness determination of an on axis type oil pump cover is studied using the finite element method.

Firstly, existing pump is studied with finite element analysis method and Von Mises Stress and displacement values through the normal direction of cover flat surface are evaluated.

Secondly, different cover models were created with different thicknesses. So that, new assembly parts are modelled for design set.

Thirdly, all design variables, objective function, design constraints are determined using the finite element analysis method using simulation module of Solidwork 2016 Software.

Finally; static analyses are run for all models that are participated in the design set.

Analyses are based on the previous experiences regarding to parameters that affects the pump performance.

Considering production constraint and the cost, the most appropriate cover model is selected.

4.2 Design Process Approach

Optimization is the action of getting the best results under certain conditions. In the maintenance, construction and design of any engineering system, engineers have to decide on several managerial and technological issues in several stages. The purpose of these decisions is to minimize the effort required or to maximize the desired benefit. In this context, the process of obtaining conditions that give the minimum or maximum value of a function is called optimization (Rao, 2009).

Any problem with which certain parameters need to be set in order to meet the constraints can be formulated as an optimization problem (Arora, 2012).

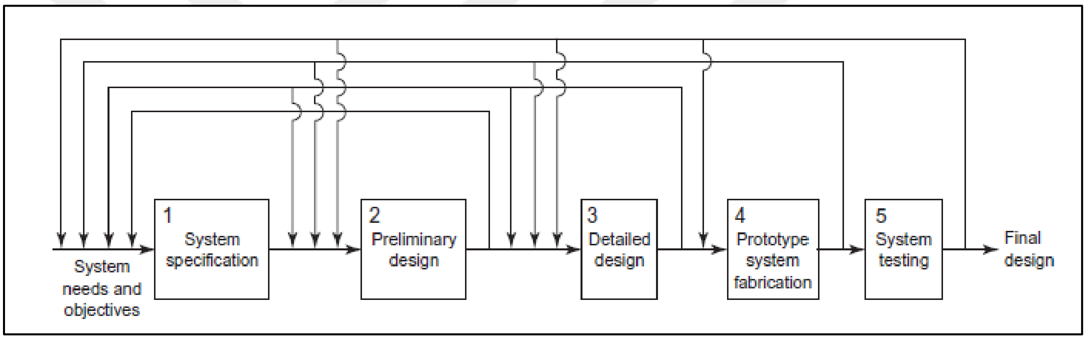

Figure 4.1 The design process evaluation model (Arora, 2012)

After 2010, traditional testing and prototyping has been replaced by a simulation-oriented design process, in order to reduce product development time and cost. This type of process, which reduces the need for these time-consuming and expensive physical prototypes, allows engineers to successfully predict product performance. The simulation-oriented product design process and the traditional product design process are compared in Figure 4.2.

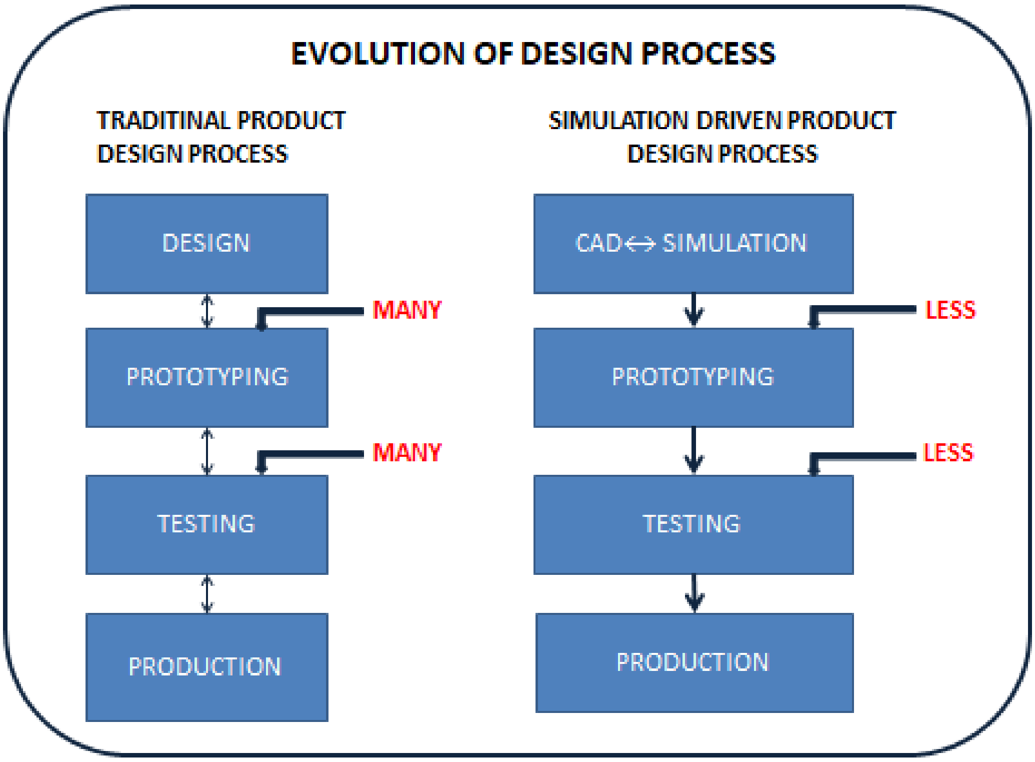

Figure 4.2 Traditional product design process ans simulation-driven product design process

4.3 Conceptual Analysis Steps of Design Determination

Conceptual analysis for a design determination using FEA shows that the logical steps and sub-steps presented below are essential:

- Definition of problem
- Building the model
- Material property assignment
- Boundary conditions and design variables specification
- Mesh generation & Running Simulation
- Post-processing (getting results)
- Validation (checking)

4.3.1 Definition of Problem

In this study, thickness determination of an on axis type oil pump cover that is used for passenger cars has been studied with finite element analysis method using with Solidwork Simulation Program. The oil pump that is studied consists of variable components. Full component list of the oil pump (bill of material) is given at Table 4.1 and exploded view of the pump for the main components is given at Figure 4.3.

Table 4.1 Bill of material of the on-axis type variable vane pump

COMPONENT DEFINITION	QUANTITY
HOUSING	1
SLIDE	1
VANE-RING	2
ROTOR	1
VANE	9
SLIDE SEAL	1
SLIDE SEAL SUPPORT	1
PIN	1
COVER	1
BOLT	6
RELIEF SPRING	1
CUP	2
RELIEF PLUNGER	1
CONTROL SPRING	1
CUP	1
O-RING	2
PIN	1
CONTROL VALVE	1
BOLT FOR VALVE	1

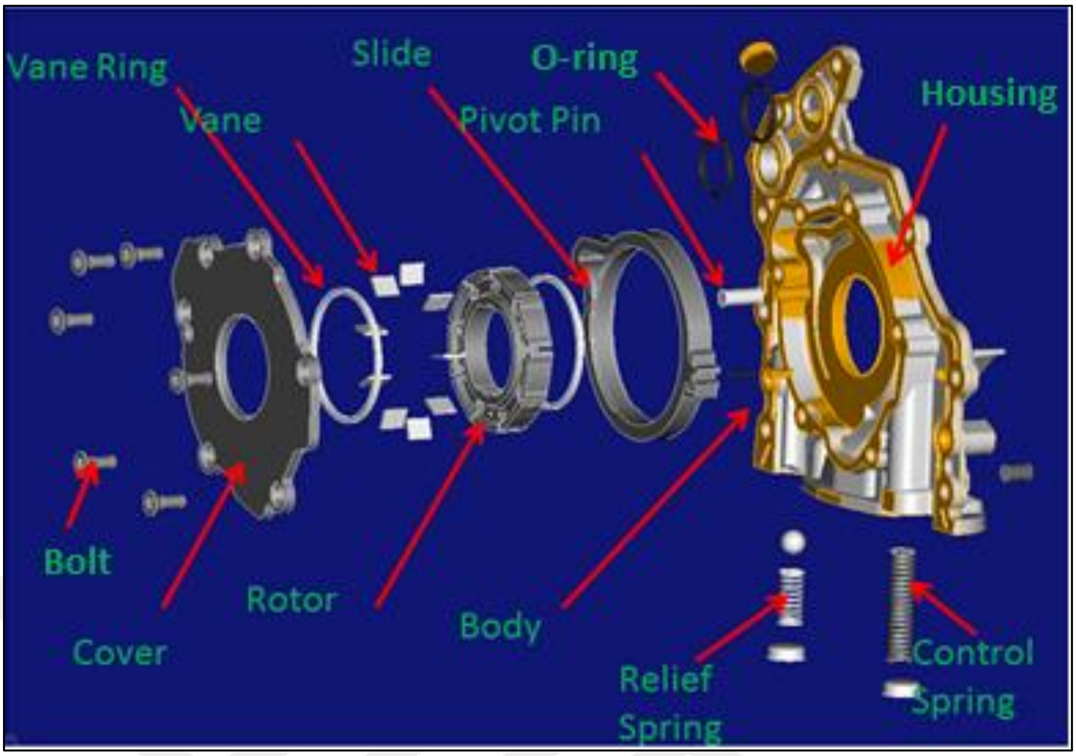

Figure 4.3 Exploded view of an on-axis variable vane pump (Stackpole International Fluid Power Solutions archive, 2018)

4.3.2 Building the Model

The models that are used for the optimization analysis, housing and cover have been built with Creo Parametric 4.0 design software. Due to being the model of housing parts is very complex and it has small geometrical features on it such as round, chamfer; the housing model is simplified for the appropriate mesh generation and then, it has been converted to the step model for the Solidworks Simulation analysis program. Since the softwares are employed to different models which are separetely designed and analyzed, there has been observed some problems regarding the surface continuity. The model has been repaired for the surface continuity and cleaned with the possible surfaces and edges that may impair the mesh structure.

Inner surface of cover part has been separeted with "*Split Line*" operation in order to define to pressure, contact surface and the region that rotor part has a contact. Figure 4.4 shows the surface separeted cover model for the analysis.

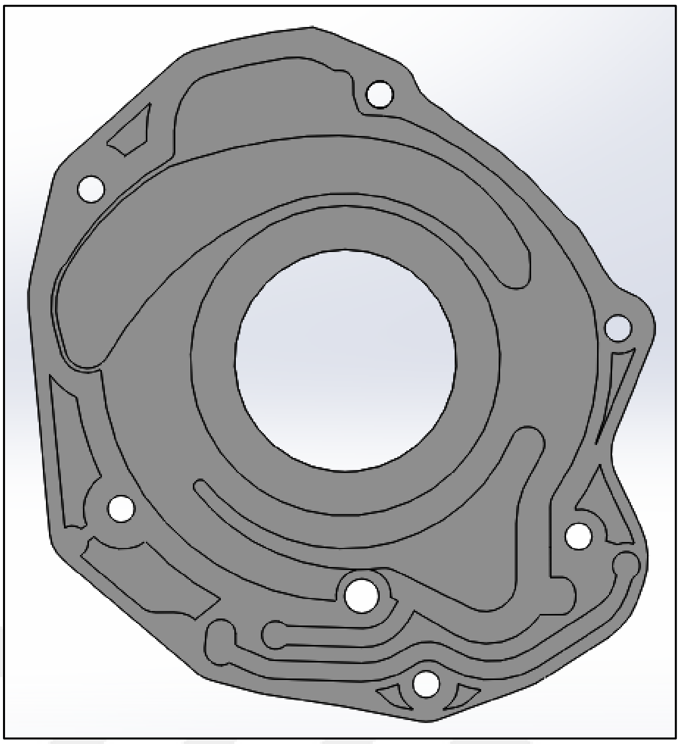

Figure 4.4 Surface separeted cover model

Since the model is not axisymmetrical, it can not be simplified with taking half or quarter of the pump. For this reason, whole geometry is taken for the analysis.

Following the improvements, the models have become available for analysis.

4.3.3 Material Property Assignment

Solidworks Simulation program has a library that includes several materials for the analysis. Since the materials that are used for the parts don't participate in the Solidwork Library, they have been defined as customized materials inside the program.

Housing part is manufactured with aluminum cast material defined as EN AC 46000-F AlSi9Cu3(Fe)DF. Cover part is manufactured with stainless steel defined as AISI 1015 Carbon Steel and bolt parts are manufactured with material defined as ISO 898 Grade 10.9.

Table 4.2, 4.4 and 4.6 respectively presents mechanical properties of the housing, bolt and cover. Table 4.3, 4.5 and 4.7 respectively show chemical properties also.

Table 4.2 Mechanical properties of housing part

Housing Material	**EN AC 46000-F AlSi9Cu3(Fe)DF**	
Aluminum Alloys	Mechanical Properties	SI-N/mm^2 (Mpa)
Property	Value	Units
Elastic Modulus	73000	N/mm^2
Poisson's Ratio	0.33	N/A
Shear Modulus	28000	N/mm^2
Mass Density	2800	kg/m^3
Tensile Strength	270	N/mm^2
Yield Strength	160	N/mm^2
Thermal Expansion Coefficent	2.10E-05	/K
Thermal Conductivity	100	W/(m-K)
Specific Heat	880	J/(kg.K)

Table 4.3 Chemical properties of housing part.

CHEMICAL PROPERTIES OF EN AC 46000-F AlSi9Cu3(Fe)DF		
ELEMENT	SYMBOL	WEIGHT %
Aluminum	Al	79.7 to 90
Silicon	Si	8.0 to11
Copper	Cu	2.0 to 4.0
Iron	Fe	0 to 1.3
Zinc	Zn	0 to 1.2
Mangesium	Mg	0.050 to 0.55
Manganese	Mn	0 to 0.55
Nickel	Ni	0 to 0.55
Lead	Pb	0 to 0.35
Titanium	Ti	0 to 0.25
Chromium	Cr	0 to 0.15
Tin	Sn	0 to 0.15
Residuals	-	0 to 0.25

Table 4.4 Mechanical properties of bolt part

Bolt Material	ISO 898 Grade 10.9	
Aluminum Alloys	Mechanical Properties	SI-N/mm^2 (Mpa)
Property	Value	Units
Elastic Modulus	210000	N/mm^2
Poisson's Ratio	0.3	N/A
Shear Modulus	80000	N/mm^2
Mass Density	7800	kg/m^3
Tensile Strength	1000	N/mm^2
Yield Strength	900	N/mm^2
Thermal Expansion Coefficent	1.20E-05	/K
Thermal Conductivity	45	W/(m-K)
Specific Heat	4800	J/(kg.K)

Table 4.5 Chemical properties of bolt part

CHEMICAL PROPERTIES OF ISO 898 GRADE 10.9		
ELEMENT	SYMBOL	WEIGHT %
Carbon	C	0.25 to 0.55
Phosphorous	P	0 to 0.025
Sulfur	S	0 to 0.025
Boron	B	0 to 0.003

Table 4.6 Mechanical properties of cover part

Cover Material	AISI 1015 Carbon Steel	
Steel	Mechanical Properties	SI-N/mm^2 (Mpa)
Property	Value	Units
Elastic Modulus	19000	N/mm^2
Poisson's Ratio	0.29	N/A
Shear Modulus	73000	N/mm^2
Mass Density	7900	kg/m^3
Tensile Strength	390	N/mm^2
Compressive Strength		N/mm^2
Yield Strength	235	N/mm^2
Thermal Expansion Coefficent	1.20E-05	/K
Thermal Conductivity	52	W/(m-K)
Specific Heat	470	J/(kg.K)
Material Damping Ratio		N/A

Table 4.7 Chemical properties of cover part

CHEMICAL PROPERTIES OF AISI 1015 CARBON STEEL		
ELEMENT	SYMBOL	WEIGHT %
Iron	Fe	99.13 to 99.57
Carbon	C	0.13 to 0.18
Manganese	Mn	0.30 to 0.60
Sulfur	S	0 to 0.050
Phosphorous	P	0 to 0.040

4.3.4 Design Constraints and Design Variables Specification

In this study; objective function is defined as minimizing of the cover thickness. At the begining of any project, allowable maximum weight of the pump is defined by the customer. One of the main target of the designers is keep the weight of the pump as much as minimum.

Keeping the weight of the pump is an important parameter in addition to providing weight advantage, as it reduces the unit cost per product in parts such as the housing and cover.

Objective Function: Minimizing of the Cover Thickness - Min f(X)

Variables:

X: Thickness of the Cover Part

X_1= 4.7 mm X_2=4 mm, X_3=3 mm

Y: Material Type of Cover

Y_1 = AISI 1015 Carbon Steel (Original Material)

Y_2= EN AC 46000-F AlSi9Cu3(Fe)DF (Alternative Material)

Constraints:

Operating Temperature: 125°C

Maximum Inlet Pressure: 1 Bar

Maximum Pressure for Rotating Area: 2.55 Bar

Maximum Outlet Pressure: 5.1 Bar

Bolt Pre-loads: 17 Nm

Flatness value of assembly surface of the cover part is defined as 30 microns at the manufacturing technical drawings. Based on the previous experiences, in case of flatness value being higher than 30 microns, this causes an oil leakage at the pump and it does not work properly. In detail; the pump can not supply pressured oil to the relevant zones of the engine. For this reason, maximum deflection value in the normal direction of the cover can not be higher than 30 microns.

Rotor inside the pump rotates at counter clockwise with the rotational movement of the crankshaft. The rotational movement of the rotor produces a vacuum effect on the inlet side of the pump. So that, the oil is suctioned from the oil sump to pump. This corresponds to 1 bar – atmospheric pressure and it is defined on the both housing and cover model (Figure 4.5 and 4.6).

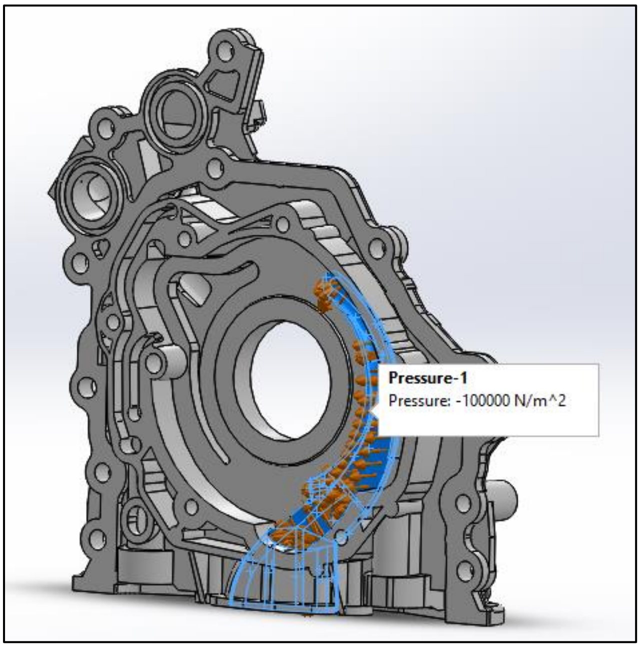

Figure 4.5 Definition of inlet pressure (vacuum impact) at the housing part

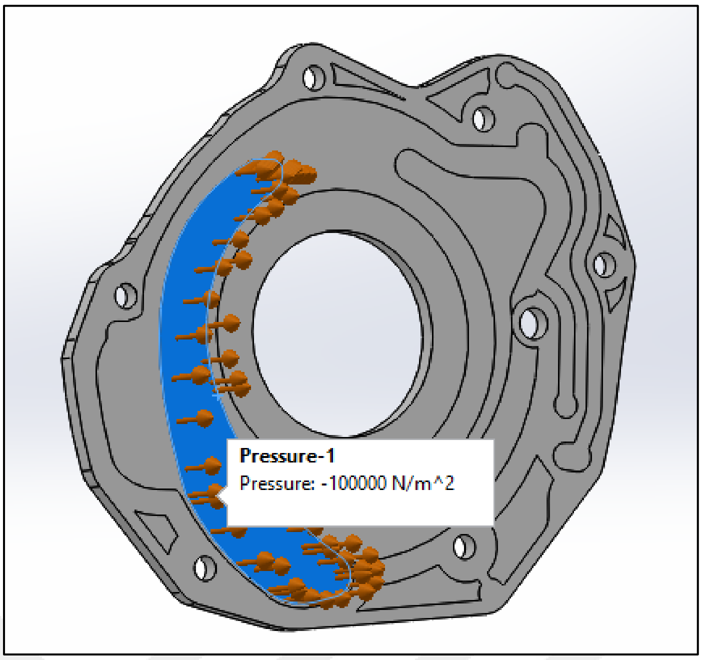

Figure 4.6 Definition of the inlet pressure (vacuum impact) at the cover part

The amount of oil that is needed by the engines changes according to the engine speed. This is a parameter that specifies the functionality of the pump. For this reason; required pressure value at the outlet side of the pump and oil flow at the working temperature is determined by the customer and all values are written at the customer drawings or specification sheets.

Since the on axis pumps are driven by the rotational movement of the crankshaft directly, speed ratio is defined as “1:1” This means that, the speed of the engine equals with the speed of the pump. The specified pressure values are written in the Table 4.8.

Table 4.8 Required pressure values at the outlet side of the pump

Rev/Min	Pump Outlet Pressure (bar)
750	1.4
1000	1.7
1500	2.3
2000	3
2500	3.6
3000	4.5
3500	4.8
4000	5.1
4500	5
5000	4.9

Pump functionality is checked on the machines that are named as “Hydraulic Function Test Rig” that simulates the working principle of the engine.

The pump functionality is simulated on the test rig device working with 2 minutes at each engine speed and the outlet pressure is measured. The test is ended after running 2 minutes at 5000 rpm. Since the part is under a maximum pressure of 5.1 bar, this value is applied to the outlet region of both housing and cover (Figure 4.7 and 4.8).

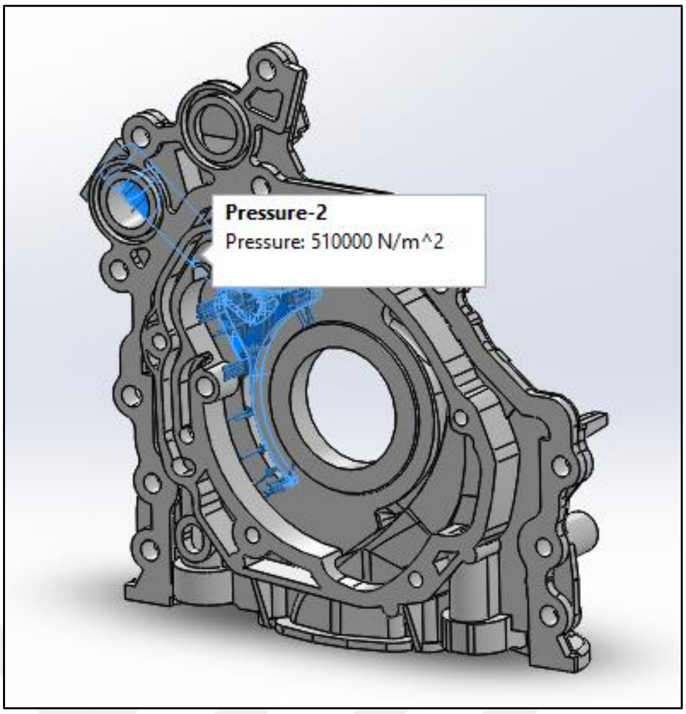

Figure 4.7 Definition of outlet pressure at the housing part

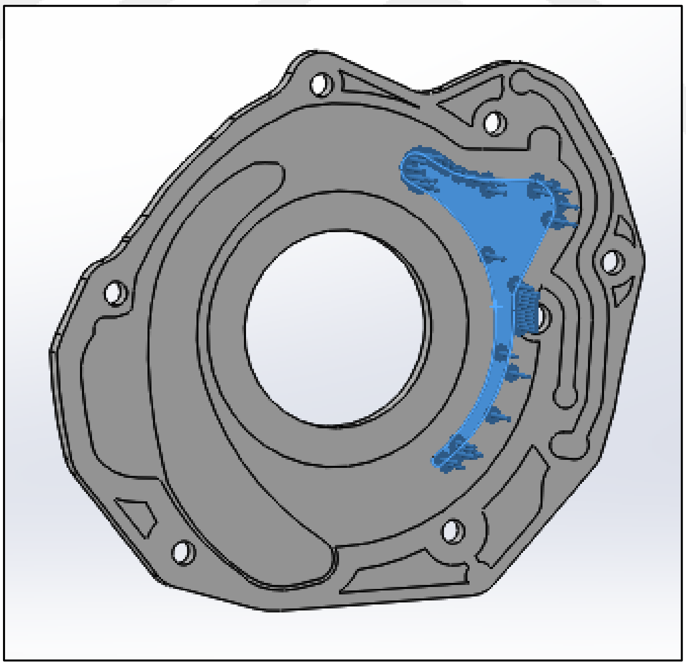

Figure 4.8 Definition of outlet pressure at the cover part

Considering serial production conditions, it is not possible to check the pressure value at the rotational area inside the pump and at the gallery channels. This requires, destructive testing with adding valves at the pump regions that is intended to check the pressure values. This test validates the pressure value at the rotational side is approximately half of the outlet pressure value. So the pressure that affects the rotor side is calculated as shown in the Table 4.9 and applied on the model.

Table 4.9 Approximate pressure values on the rotational area

Rev/Min	Rotational Area Pressure (bar)
750	0.7
1000	0.85
1500	1.15
2000	1.5
2500	1.8
3000	2.25
3500	2.4
4000	2.55
4500	2.5
5000	2.45

Since the maximum value that affects the rotational area is 2.55 bar, this applied to the both housing and cover part (Figure 4.9).

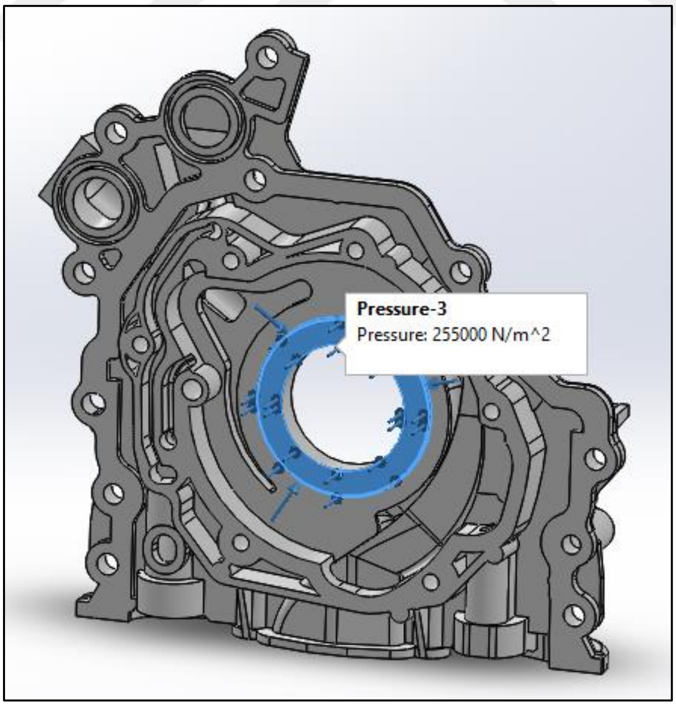

Figure 4.9 Definition of the maximum pressure value of the rotational area on the housing part

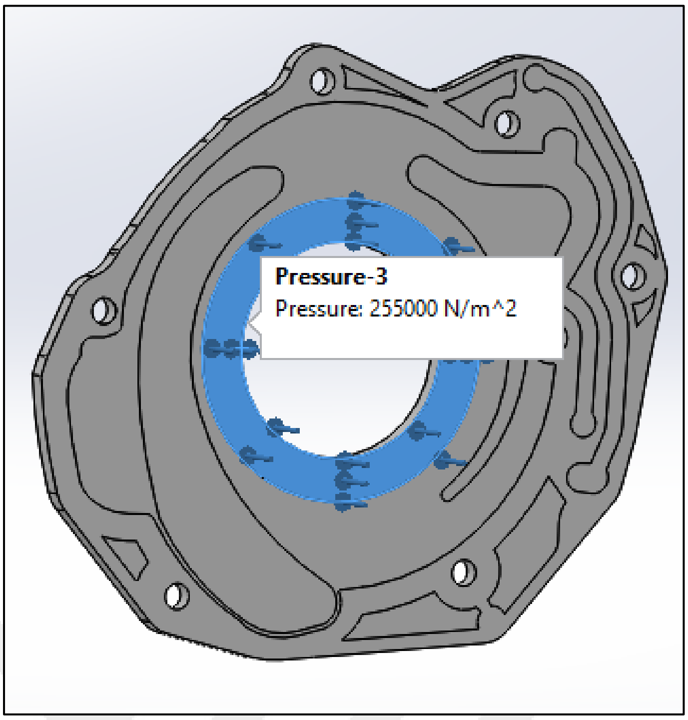

Figure 4.10 Definition of the maximum pressure value of the rotational area on the cover part

Working temperature (125°C) is applied to the specified zones that oil has a contact at the housing and cover. Figure 4.11 and 4.12 present definition of working temperature.

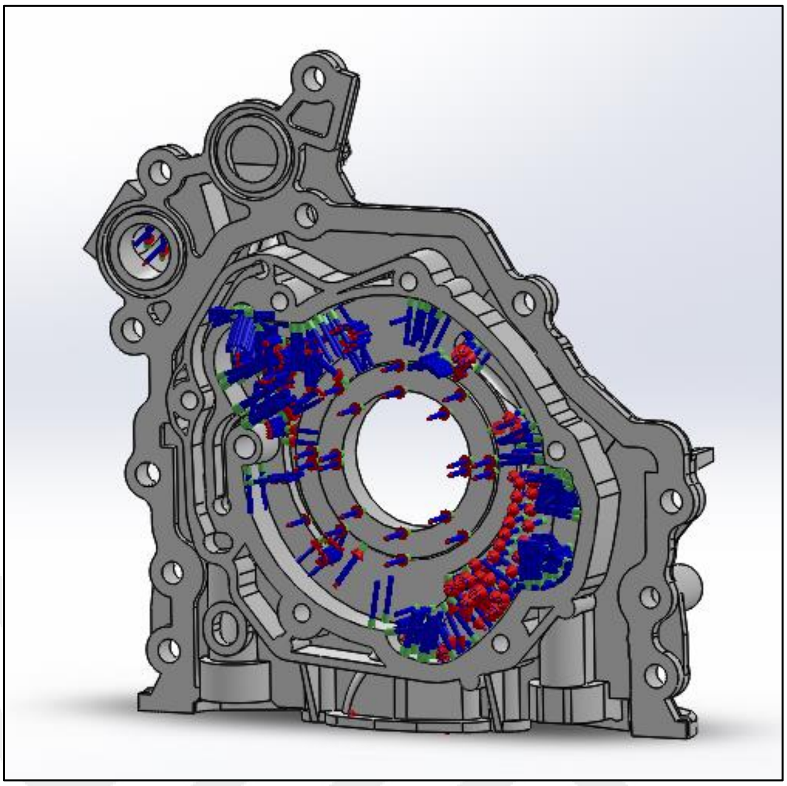

Figure 4.11 The surfaces that are affected by the working temperature at the housing part

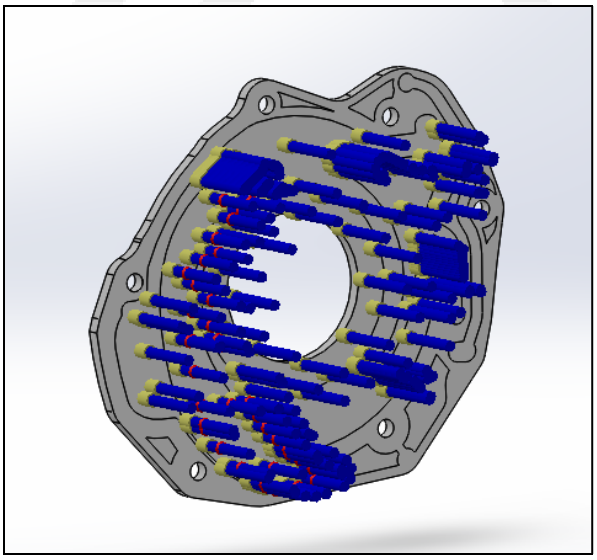

Figure 4.12 The surfaces that are affected by the working temperature at the cover part

Contact set is defined for the surfaces of cover and housing. Friction coffecient between stainless steel cover and aluminum housing is defined as 0.3 under vacuum conditions and "No penetration" option is selected. In case of material type of the cover is changes as aluminum alloy, same material with the housing part, friction coeffcient is defined as 1.35 under vacuum conditions.

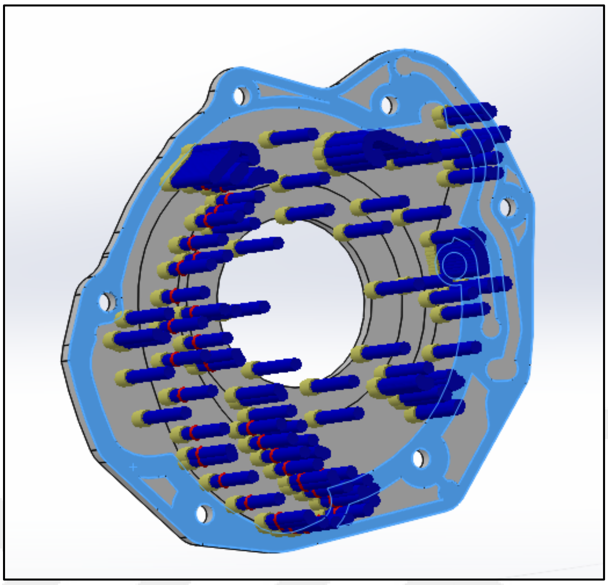

Figure 4.13 Definiton of contact surface on the cover part

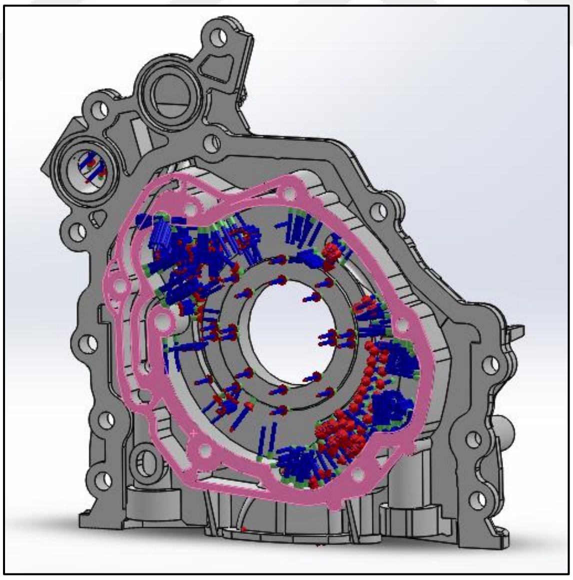

Figure 4.14 Definition of the contact surface on the housing part

M6-Screws are used for the assembly of the cover and housing. So all bolt holes restricted with 17 Nm torque value. Bolt material is selected as ISO Grade 10.9 and weight of each bolts is approximately 5.1 gram. Cover fits on the housing part with pin connection guiding. Although it allows rotational movement, it does not allow translation movement. Pin connection is defined to the model considering its own weight:7.7 gram. Bolt and pin connections are shown in Figure 4.15.

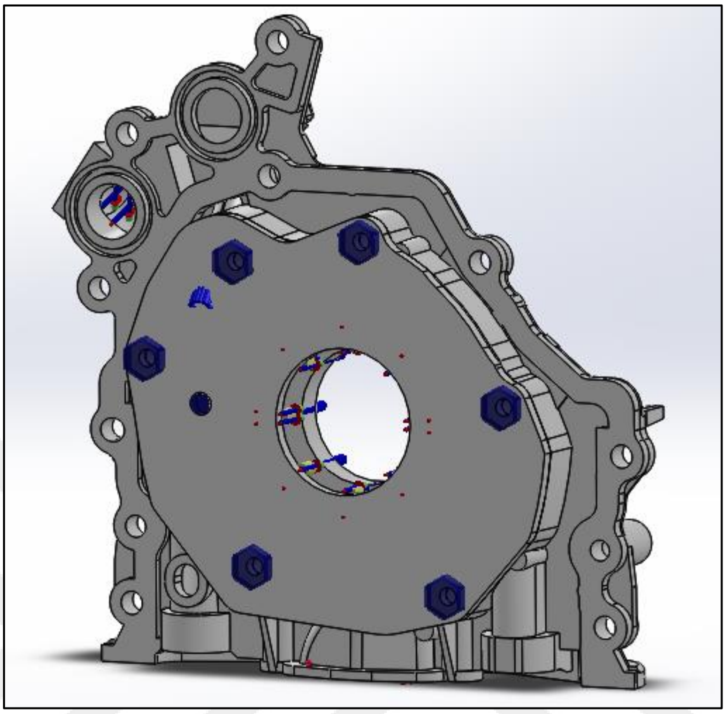

Figure 4.15 Bolt and pin conneciton on the assembly part

The engine connection holes on the housing part as defined as fixed. So that, it is not allowed to both rotational movement and translation movement in the direction of X, Y and Z. Since the oil sumb connection allows to the pump only translation movement, oil sumb connection holes on the housing are defined as “ slider” (Figure 4.14 and 4.15).

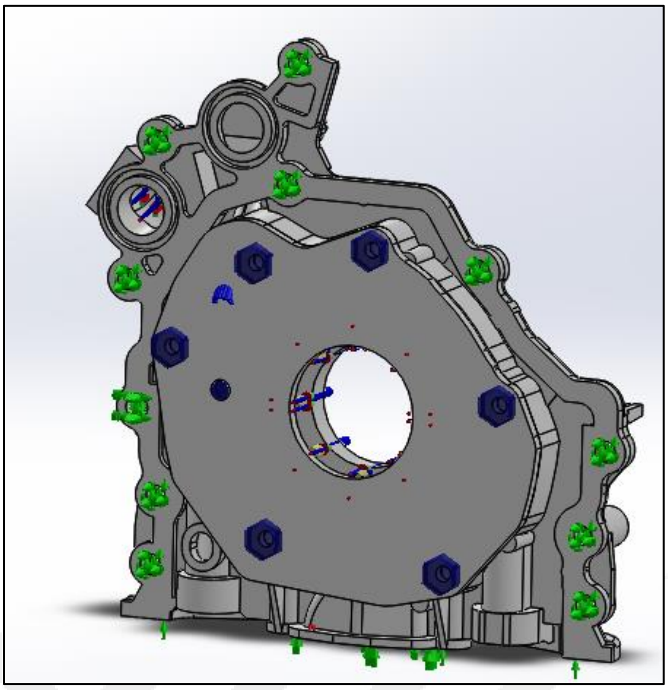

Figure 4.16 Definiton of engine connection holes on the housing part

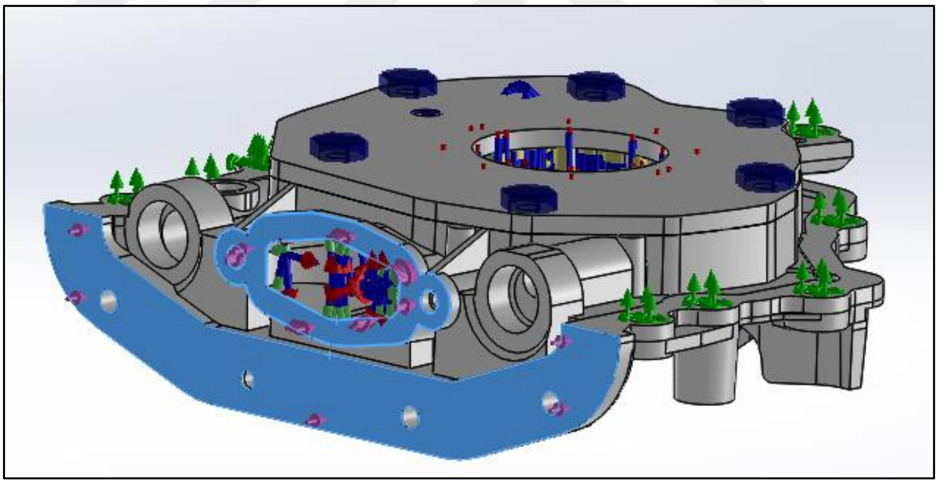

Figure 4.17 Definition of oil sump assembly surface on the housing part

Oil pump is driven by the rotaional movement of crankshat and it is connected with the rotor. So the hole at the center of the pump does not touch the surface of crankshats. It is remains as " free".

4.3.5 Mesh Generation and Running Simulation

After completing all definitions on the model, mesh operation is applied to the model. Due to complexity of the housing part, curvature based mesh type is applied to the model. Figure 4.18 presents mesh generated model.

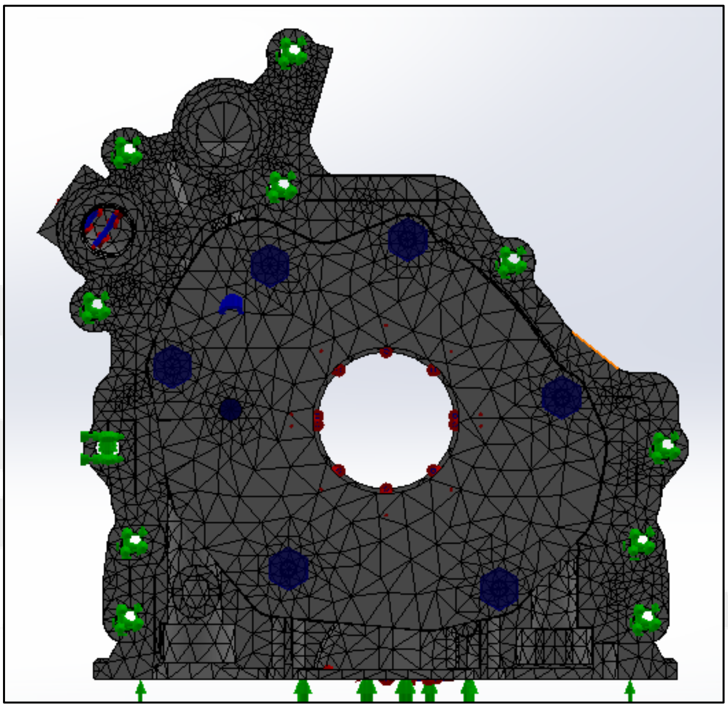

Figure 4.18 Curvature mesh generated model

Total number of nodes are obtained as 102371 and total number of element is obtained as 76027. Since the main target is defining deflection ratio of the cover, mesh control is made for the cover part. Figure 4.19 shows mesh control applied model.

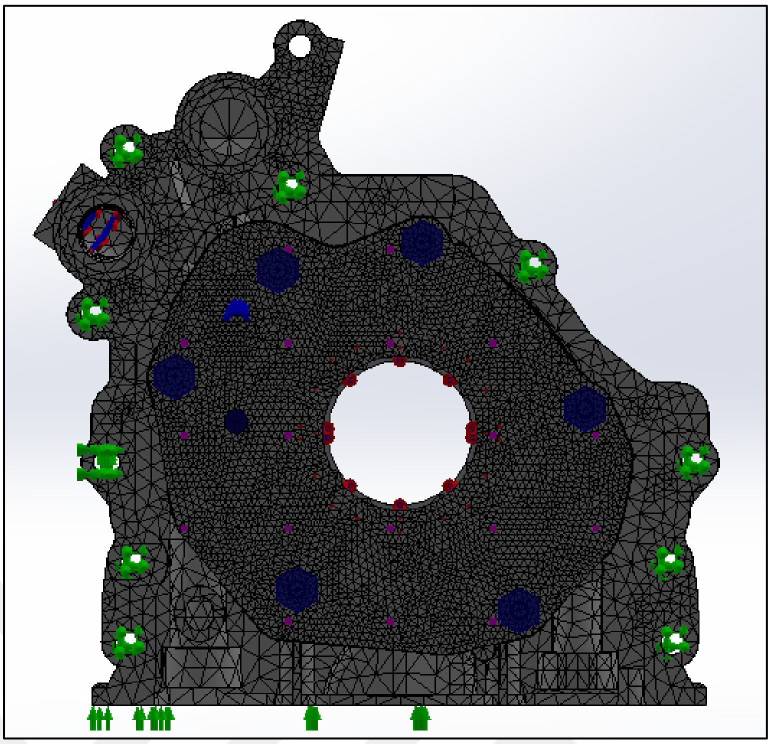

Figure 4.19 Mesh control applied model

After applying mesh control to the cover part, total number of nodes changed as 83724 total number of elements changed as 143499. Minimum size element is defined as approximately 1.709 mm and maximum element size is defined as 8.544 mm.

Table 4.10 presents design set of the analyses.

Table 4.10 Table of design variables

Design Variables (X,Y)	X_1= 4.7 mm	X_1= 4 mm	X_1= 3 mm
Y_1 = AISI 1015 Carbon Steel (Original Material)	Analysis-1	Analysis-2	Analysis-3
Y_2= EN AC 46000-F AlSi9Cu3(Fe)DF (Alternative Material)	Analysis-4	Analysis-5	

In Analysis 4 and Analysis 5; material type of the cover has beens changed as aluminum alloy and. So considering cast manufacturing constraint, a new cover model has been developed as shown in Figure 4.20.

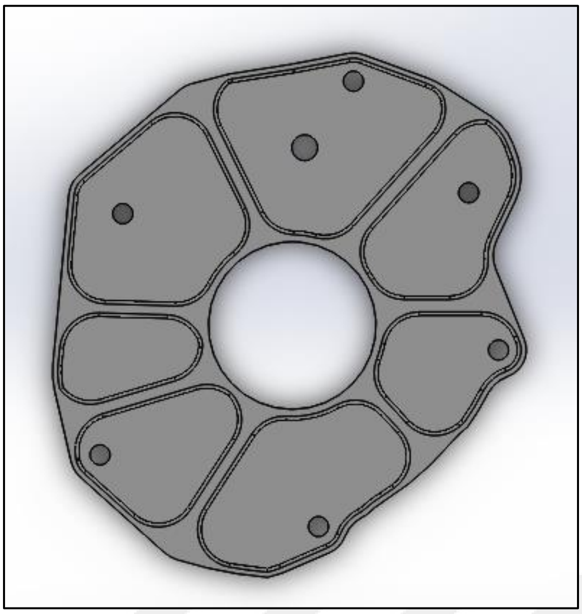

Figure 4.20 New cover model

Then; a new assembly model has been created with the new cover models and used for the analysis. Figure 4.21 shows mesh generated assembly model.

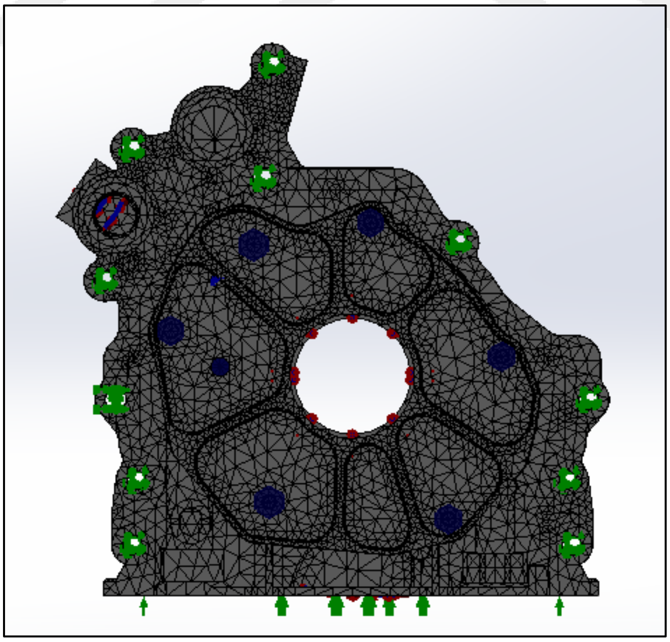

Figure 4.21 Mesh generated model assembly model with the new cover

4.3.6 Post-processing and Getting Results

When the mesh operation is finished, static analysis is run separately and the results are get for each analysis that is given in the Table 4.10 In this study; displacement value and stress distribution are evaluated.

CHAPTER FIVE
RESULTS

After defining all design constraints and boundary conditions, static analysis was run on assembly models that are participated in the design set and compared the results.

Previous manufacturing experinces show that, in case of deformaton of the cover being higher than 30 microns, a gap occurs between housing and cover parts and this creates a leakage problem and the pump can not perform accurately and reach the desired pressure. For this reason, flatness value is defined as 30 microns for the inner surface of the cover at the manufacturing drawings. Therefore, the deformationvalue of the cover is decisive in this study.

Firstly original cover part was analyzed with FEM and also the results are displayed. Figure 5.1 shows the resultant displacement of the original cover. Approximately 10 microns deformation is observed and this is the under limit value.

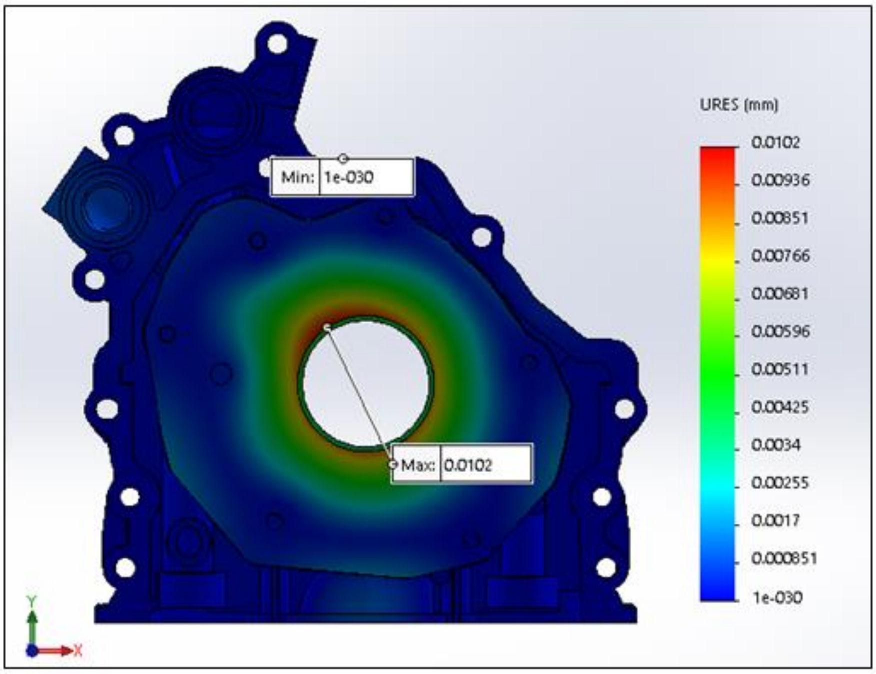

Figure 5.1 Resultant displacement of the original cover (Analysis1)

Since the displacements in the normal direction to the cover will have more effect on the internal leakage, the displacement values in the normal direction of the cover is displayed in Figure 5.2 The results shows that; there is not major distance between resultant displacement and the normal direction displacements. For this reason, for the other analysis, only resultant displacement is considered for the evaluation.

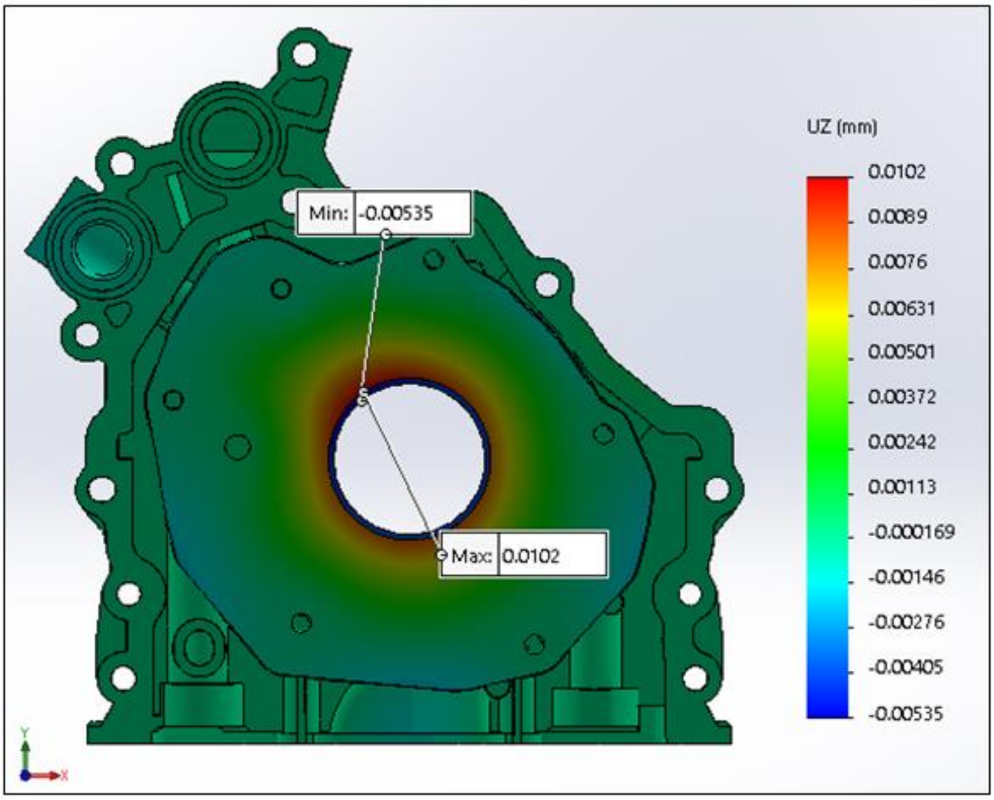

Figure 5.2 Deformation of normal direction of the cover (Analysis1)

On the other hand, Von Mises stress distribution values are evaluated and in order to confirm the parts work under yield point. The Von Mises stress distribution on the assembly part which contains original cover is given in Figure 5.3.

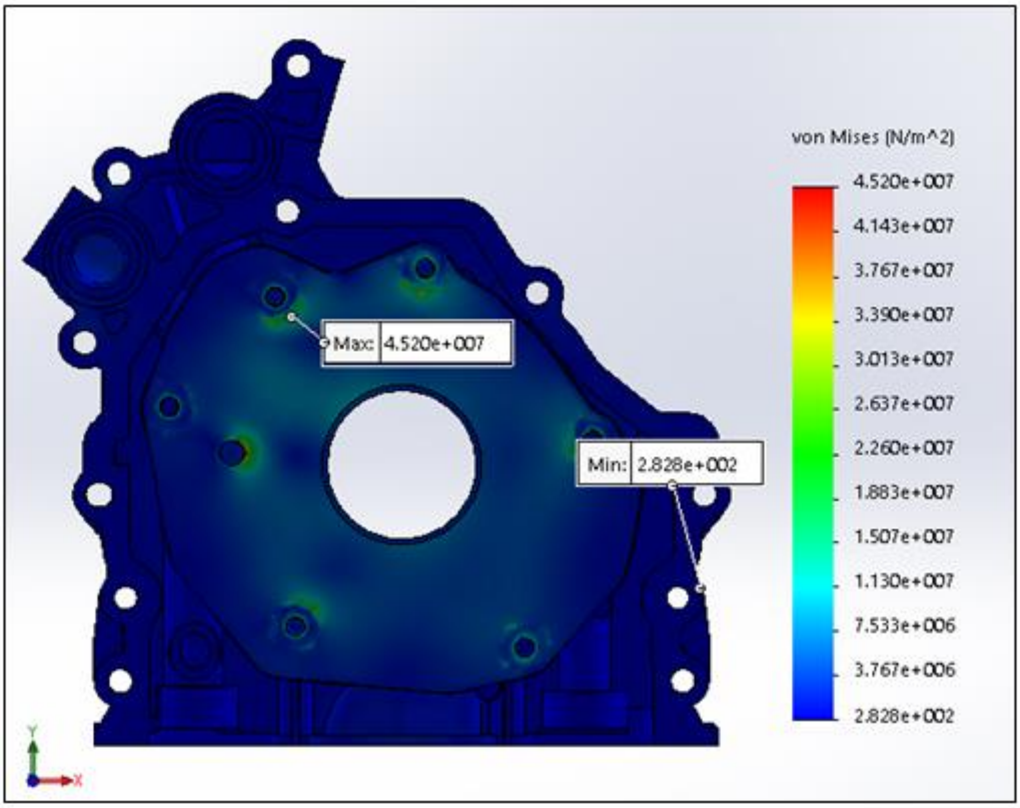

Figure 5.3 Von Mises stress distribution on the assembly part (Analysis 1)

Secondly, a new assembly part which was assembled with the cover that has a 4 mm thickness value and the resultant displacement values and the Von Mises stress distributions are evaluated (Figure 5.4 and 5.5).

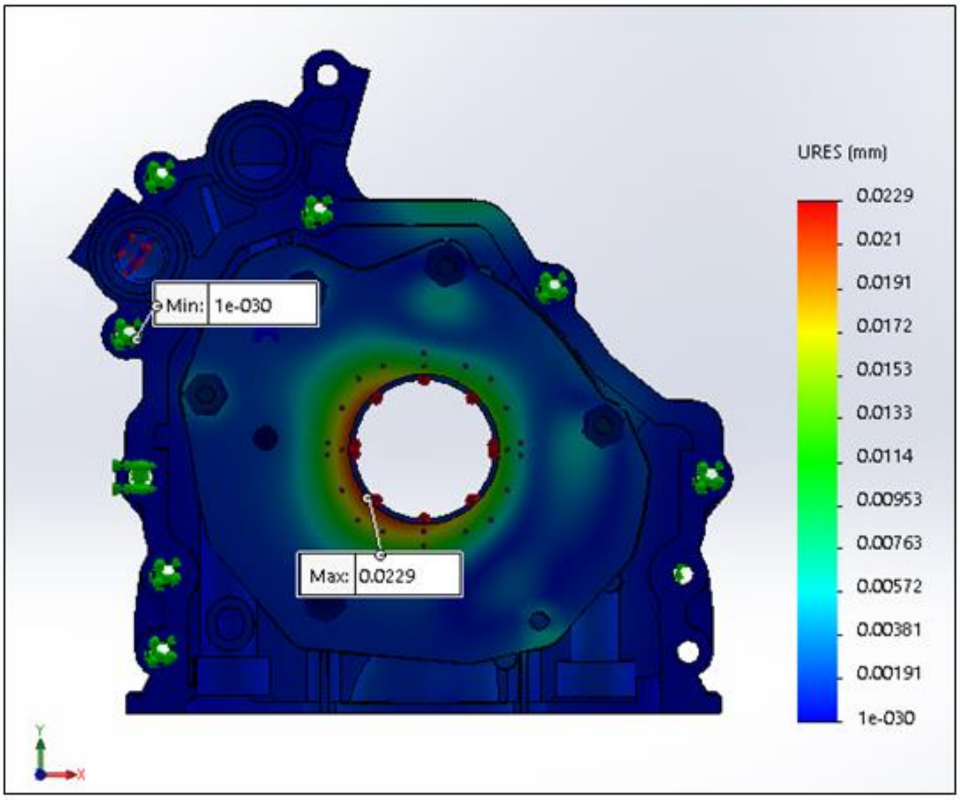

Figure 5.4 Resultant displacement value of the cover (Analysis 2)

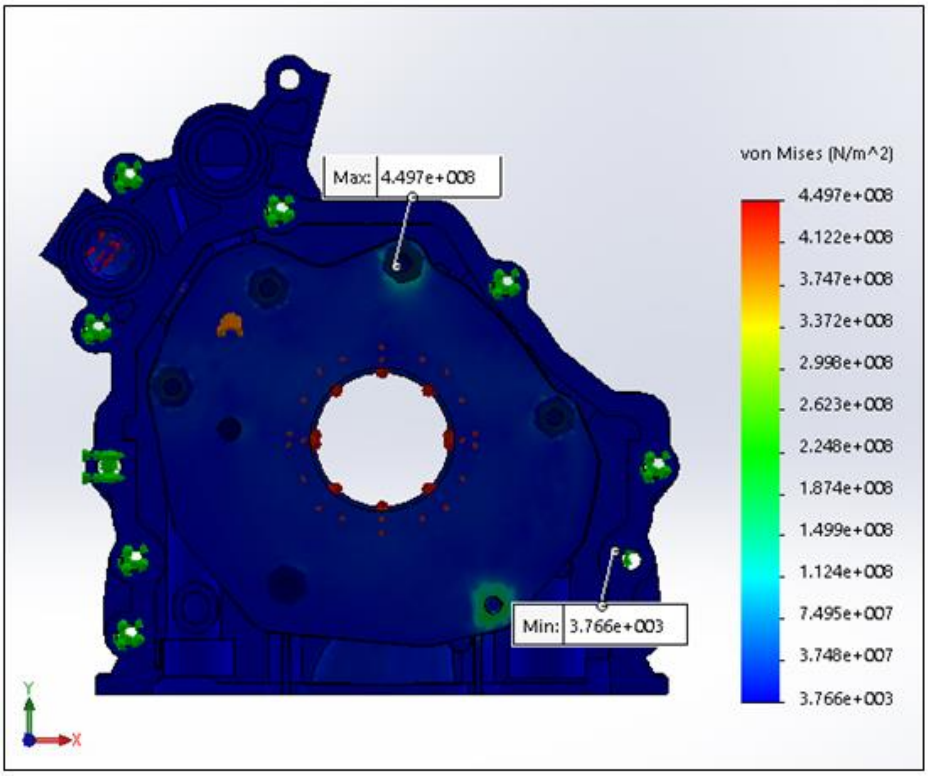

Figure 5.5 Von Mises stress distribution on the assembly part (Analysis 2)

Displacement value is obtained approximately 23 microns and this is under the limit.

Thirdly; analysis-3 was run and the results (Figure 5.6 and 5.7).

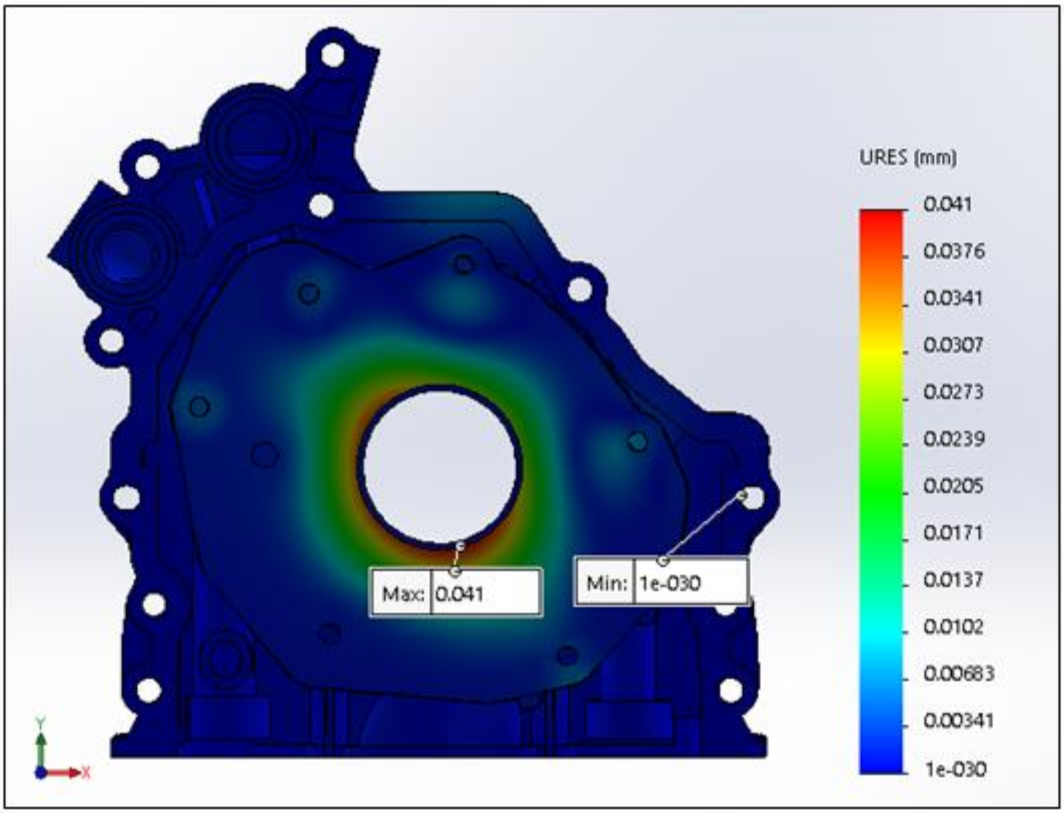

Figure 5.6 Resultant displacement value of the cover (Analysis 3)

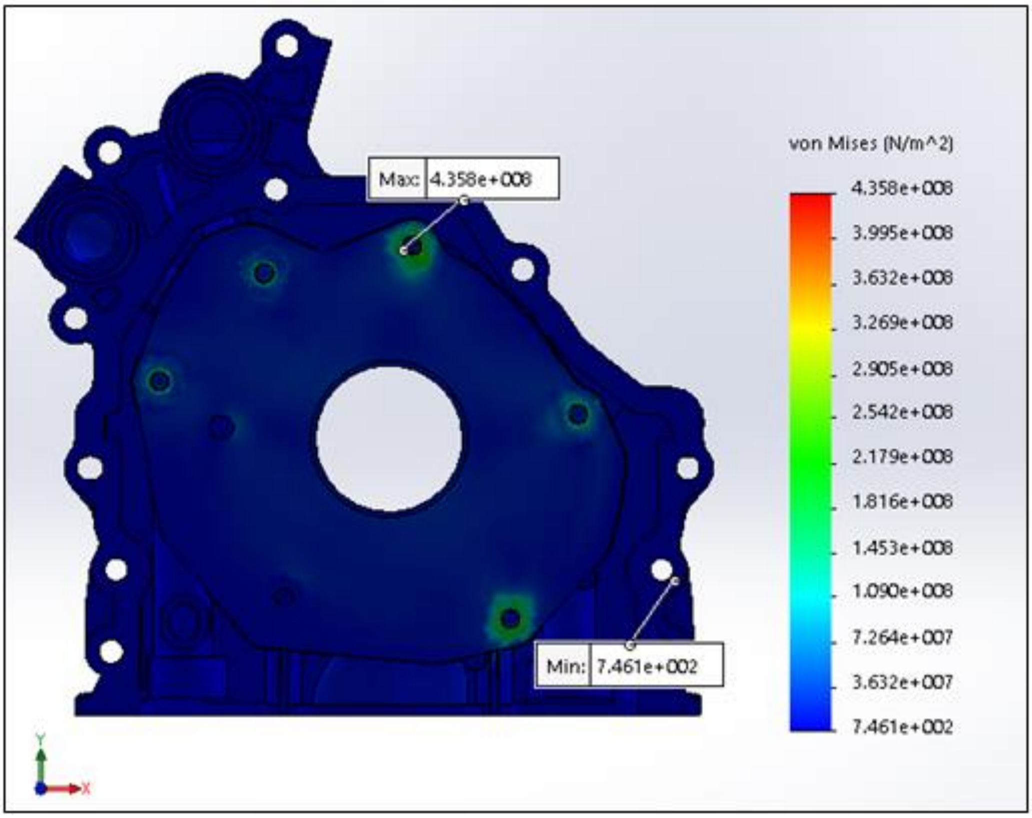

Figure 5.7 Von Mises stress distribution on the assembly part (Analysis 3)

Analysis 4 and the Analysis 5 includes a new cover with aluminum alloy material. Figure 5.8 and 5.10 present resultant displacement values and Figure 5.9 and 5.11 shows the Von Mises stress distribution on the assembly part.

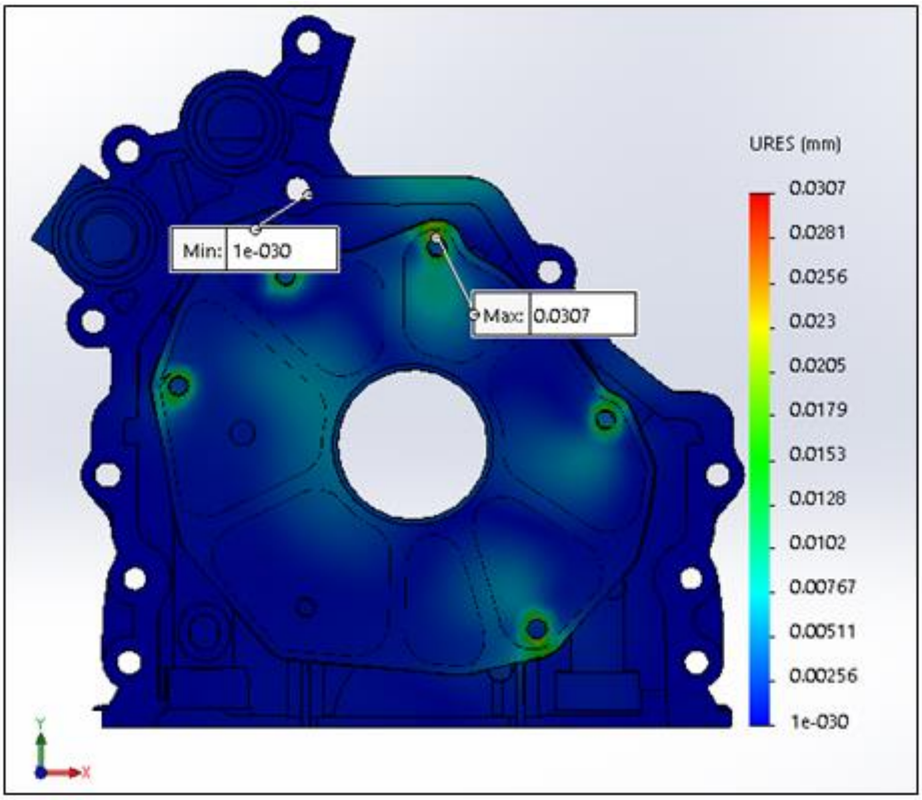

Figure 5.8 Resultant displacement values of cover (Analysis 4)

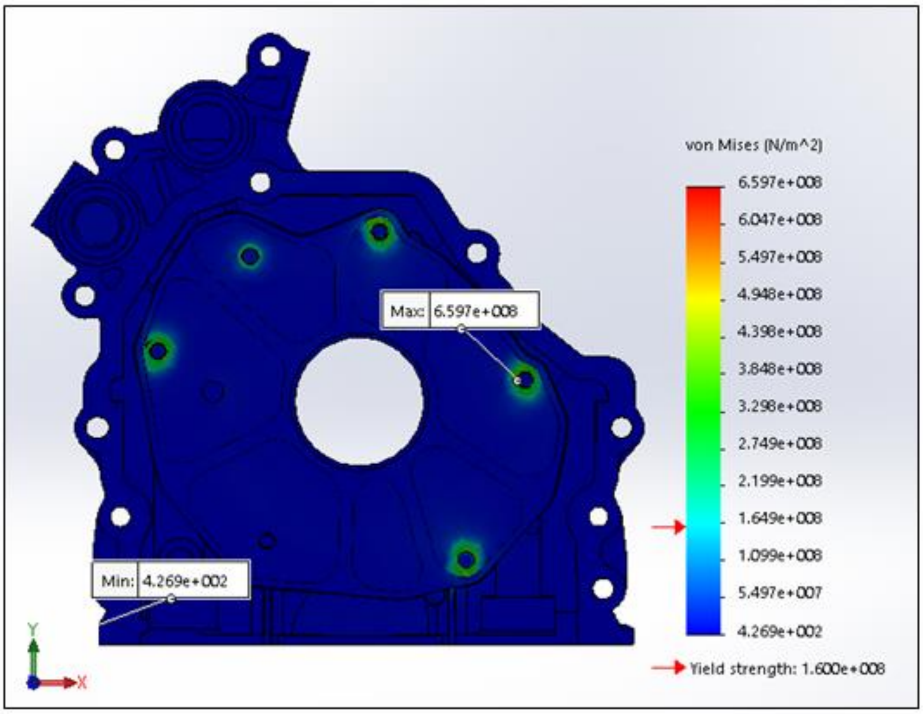

Figure 5.9 Von Mises stress distribution of the assembly part (Analysis 4)

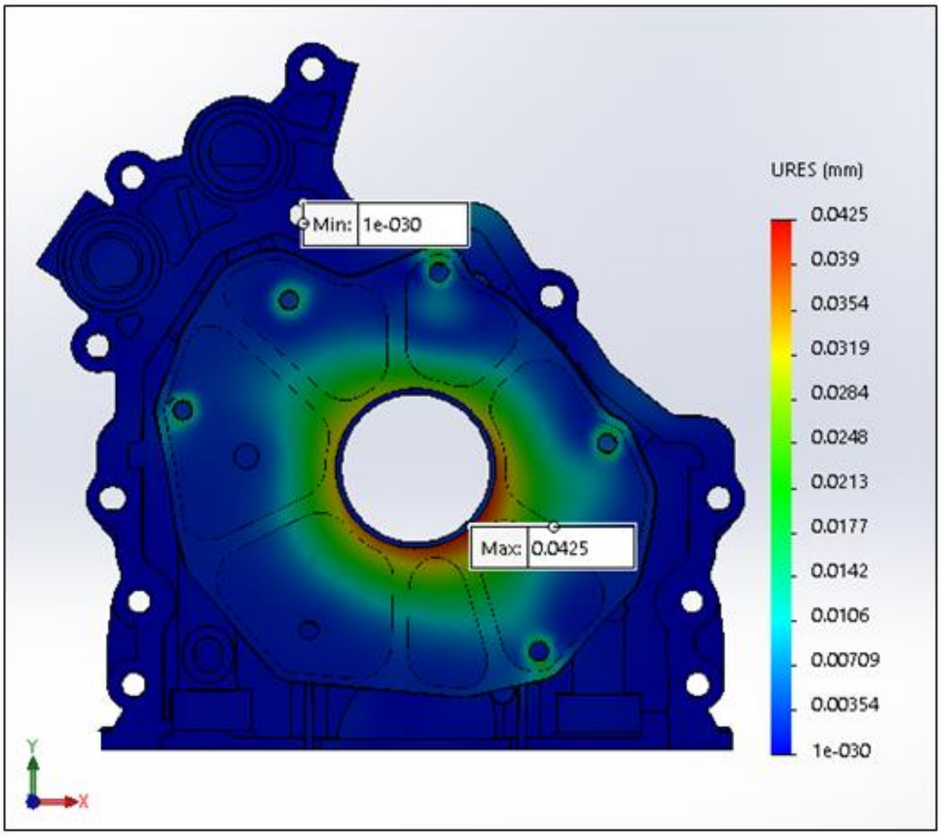

Figure 5.10 Resultant displacement values of cover (Analysis 5)

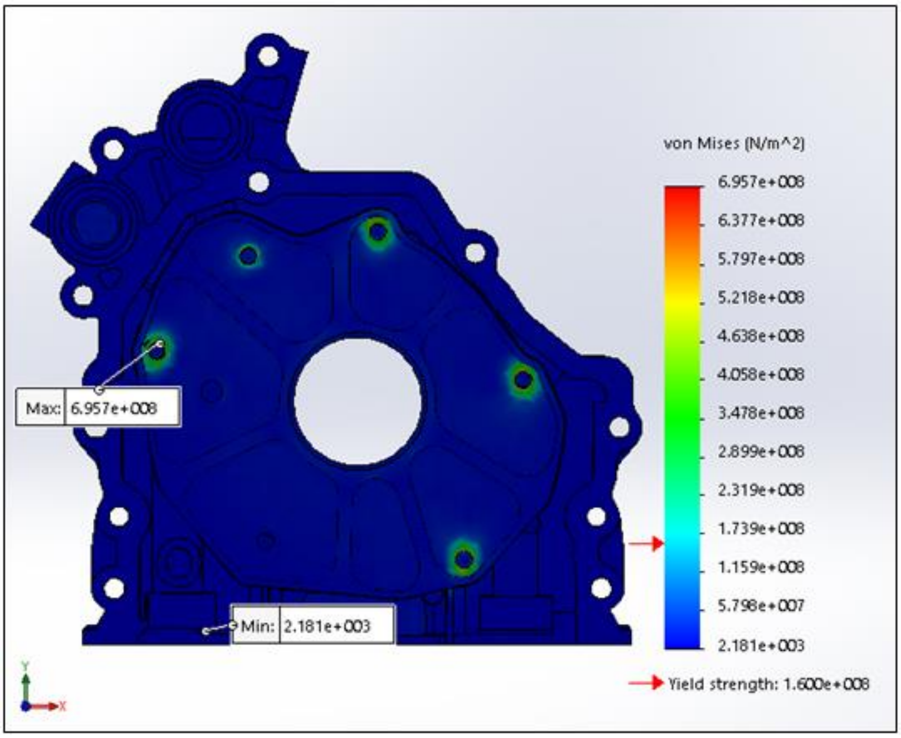

Figure 5.11 Von Mises stress distribution on the assembly part (Analysis 5)

Table 5.1 presents all results summarized.

Table 5.1 Analysis results of design set

STATIC ANALYSIS RESULT OF DESIGN SET						
Analyses	Cover Thickness Values	Maximum Resultant Displacement (mm)	Maximum Resultant Displacement (micron)	Maximum Von Mises Stress Value (N/ mm^2)	Yield Strength of the Cover Material (N/mm^2)	Cover Weight (gram)
Analysis-1	Cover: 4.7 mm- 1015 Carbon Steel	0.0102	10.2	4.25E+01	235	499.71
Analysis-2	Cover: 4 mm- 1015 Carbon Steel	0.0229	22.9	4.50E+01	235	425.28
Analysis-3	Cover: 3 mm- 1015 Carbon Steel	0.041	41	4.36E+01	235	318.96
Analysis-4	Cover: 4.7 Al- Aluminum Alloy	0.0307	30.7	6.60E+01	160	206.39
Analysis-5	Cover: 4 mm Aluminum Alloy	0.0425	42.5	6.96E+01	160	180.01

According to the analysis results cover thickness can be reduced to 4 mm provided that using the same material with original model. 74.43 gram weight reduction is obtained and this means approximately 15% weight improvement.

The resultant displacement value of the new cover models that are used for Analysis 4 and Analysis 5 exceed the specified displacement value (30 microns). Resultant displacement value of the Analysis 4 is too near to the limit values. For this reason; considering weight reduction advantage (36%), the model can be improved adding additional ribs.

CHAPTER SIX
CONCLUSION

In this study, thickness determination of an on axis type oil pump cover that is used for passenger cars has been studied with finite element analysis method.

Models have been created with Creo Parametric design software, considering manufacturing and assembly constraints and then the models have been simplified for appropriate mesh generation.

The static analyses of the current cover model and the new cover models for the design set have been studied with Solidworks Simulation 2016 design and simulation software.

As a result of the comparisons, positive results were obtained and it has been figured out that the thickness of the covers reduced to 4 mm using the same material. This means approximately 15% weight improvement. Although weight improvement is 36%, changing material type on the part with the same thickness gives limit resultant displacement values. Adding extra ribs can reinforce the cover structure and reduce the deformations on the part.

On the other hand, since the new cover models with aluminum alloy material would be design considering cast manufacturing necessity, this model includes some ribs around bolt holes. So this situation can bring some assembly difficulties considering serial production conditions. In addition to this manufacturing and part cost are the other important inputs that affects the final decision.

As a result of this study, it was understood that the thickness determination process had an effect on the final part design and the optimal design was achieved in a shorter time and closer to the intended model in the first stages of the design process.

Nevertheless, the manufacturing constraints of the derived conceptual model should also be taken into consideration for optimal design and it is understood that the prototype models and the design needs to be validated.

REFERENCES

2017 Automotiv industry trends (n.d.). Retrieved June 06, 2017 from http://www.strategyand.pwc.com

Arora, J.S. (2012). *Introduction to optimum design* (3rd Ed.). London: Elsevier Academic Press.

Asl, M. G., Torabi, R., & Nourbakhsh, S. A. (2009). Experimental and FEM failure analysis and optimization of a centrifugal-pump volute casing. *Engineering Failure Analysis, 16,* 1996-2003.

Böwing, O. (2009). *Die 2-stufige variable flügelzellen ölpumpe & einfluss des portings auf die statischen undbdynamischen drücke.* Diploma Thesis, Fachhochschule Aachen, Aachen.

Capriccioli, A., & Frosi, P. (2009). Multipurpose ANSYS FE procedure for welding processes simulation. *Fusion Engineering and Design, 84,* 546-553.

Centrifugal pump overview (n.d.). Retrieved July 02, 2017 from https://www.mi-wea.org

Çelik, H.K, & Topakçı, M. (2007). Çizelin yapısal ve işlevsel elemanlarında sonlu elemanlar yöntemi ile mukavemet analizi. *Tarım Makinaları Bilimi Dergisi, 3* (2), 111-116.

Çelik, H.K., & Uçar, M. (2007). Yüksek basınçlı dişli pompalarda gövdenin parametric tasarımı, gerilme analizi ve optimizasyonu. *Mühendis ve Makine, 49* (576), 15-24.

Dong, J., Bao, X. & Zheng X. (2012). Comparison of two methods of finite element modeling for elbows with unequal wall thickness. *Physics Procedia, 33*, 498-504.

Gerdemeli, İ., & Kurt, S. (2010). Finite element analysis of the tower crane. *Trends in the Development of Machinery and Associated Technology, 14*, 561-564.

Görener, A. ve Görener, Ö. (2008). Otomotiv endüstrisinin Türkiye ekonomisindeki yeri: Sektörel bir inceleme. *Elektronik Sosyal Bilimler Dergisi, 7* (26), 306-319.

Görener, A. ve Görener, Ö. (2008). Türk otomotiv sektörünün ülke ekonomisine katkıları ve geleceğe yönelik sektörel beklentiler. *Journal of Yaşar University, 3* (10), 1213-1232.

How the lubrication system works in an engine? (n.d.). Retrieved June 19, 2017 from http://www.lubrita.com

Hydraulic pump (n.d.). Retrieved June 19, 2017 from https://en.wikipedia.org

ISO 898-1 (2009 E) *Mechanical properties of fasteners made of carbon steel and alloy steel & bolts, screws and studs with specified properties classes, coarse thread and fine pitch thread*

İstanbul sanayi odası sektör raporu (2007). Retrieved February 20, 2018 from http://www.iso.org.tr

Kinetic pumps, centrifugal, peripheral, special pumps. Retrieved July 02, 2017 from http://www.piping-engineering.com

McKinsey (n.d.). Retrieved May 16, 2017 from www.mckinseycom.tr

Moaveni, S. (2015). *Sonlu elemanlar analizi teori ve ansys ile uygulamalar. (4th ed.).* (A. O. Ayhan, Trans.). Ankara: Palme Yayıncılık. (Original work published 2015)

Motor yağlama sistemi nasıl çalışır? (n.d.). Retrieved August 13, 2017 from https://www.sekizsilindir.com/2017/06/motor-yaglama-sistemi-nedir-nasl-calsr.html

Otomotiv Sanayi Derneği (n.d.). Retrieved February 24, 2018 from http://www.osd.org.tr

Paletli pompaların çalışma prensibi.(n.d.). Retrieved August 13, 2017 from http://www.akder.org/tr/makale/229-paletli-pompalar-n-cal-sma-prensipleri

Pumpschool (n.d.). Retrieved from October 10, 2018 http://www.pumpschool.com

Rao, S.S. (2009). *Engineering optimization theory and practice (4th ed.).* John Wiley: Sons.

Reddy, P. S.K., & Nagaraju, Ch. (2017). Weight optimization and finite element analysis of composite automotive drive shaft for maximum stifness. *Materials Today: Proceedings 4,* 2390-2396.

Rotary lobe pumps (n.d.). Retrieved October 10, 2018 from https://www.carotek.com

Rotary lobe pumps (n.d.). Retrieved October 10, 2018 from https://www.indiamart.com

Rotary lobe pumps. Retrieved October 10, 2018 from http://www.wrightflowtechnologies.com

Silori, P., Shaikh, A., KC, N.K. & Tandon, T. (2015). Finite element analysis of traction gear using ANSYS. *Materials Today: Proceedings 2*, 2236-2245.

T.C. Dış işleri bakanlığı. Retrieved December 24, 2018 from http://www.mfa.gov.tr

T.C. Strateji ve bütçe başkanlığı (2013). *10. kalkınma planı* 2014-2018. Retrived February 10, 2018 from http://www.sbb.gov.tr

Teker, E. ve Felekoğlu B. (2007). Dünya otomotiv endüstrisindeki küresel gelişmeler ve bu gelişmelerin Türk otomotiv endüstrisi üzerindeki etkileri. *Mühendis ve Makina 568*, 26-30.

The international organization of motor vehicle manufacturers [OICA]. Retrieved March, 11, 2018 from http://www.oica.net

Turgut, S., & Turgut, M., (2011). *Solidworks, simulation, motion, moldflow, solidCAM (4th ed.).* Ankara: Seçkin Yayıncılık.

Tübitak (2017). Parça sektörü otomotiv yan sanayii 2013 vizyonu draft rapor. Retrieved June 06, 2017 from http://www.tubitak.gov.tr

What is a centrifugal pump (n.d.). Retrieved from, June 19, 2017 https://www.introtopumps.com/pumps

Yayar, R., & Yılmaz, E., (2016). *Türkiye otomotiv sanayinin gelişimi*. Retrieved March, 11, 2018 from https://www.researchgate.net

Yanhong, C., & Feng, Z., (2011). The finite element analysis and the optimization design of the Yj3128-type dump truck's sub-frames based on ANSYS. *Procedia Earth and Planetary Science, 2,* 133-138.

Yıldız, A.R. (2016). Taşıt elemanlarının yapısal optimizasyon teknikleri ile optimum tasarımı. *Journal of Polytechnic*, *20* (2), 319-323.