2006:212 CIV

MASTER'S THESIS

Hybrid Transmission

Design of the Electric Oil Pump

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2006:212 CIV • ISSN: 1402 - 1617 • ISRN: LTU - EX - - 06/212 - - SE

Abstract

Hybrid vehicles are a step towards more environmentally friendly way of transportation. This type of vehicles will play an important role in the foreseeable future to decrease the overall dependence on oil reserves but also to reduce the combustion engine's negative impact on e.g. pollution and noise in urban areas. Hybrid propulsion systems can be accomplished in several different ways, but the common idea is to add an electrical motor for additional tractive power. This also gives possibility to recuperate the kinetic energy which is otherwise lost when braking.

The hybrid transmission under development uses pressurized oil for gear shifting operations, clutch activation and cooling. Conventional automatic transmissions usually have a hydraulic pump driven by the combustion engine for pressurizing the oil. In the hybrid transmission, oil pressure is necessary also when the combustion engine is not running. In this Master's Thesis a design proposal for an electric oil pump for the transmission are developed. The goal has been to design a pump concept able to operate in all possible driving modes for a hybrid vehicle. The concept was designed considering capacity and pressure requirements, winter temperature performance, efficiency and package space.

Three different hydraulic pump concepts has been developed and analyzed. A current transmission oil pump has also been analyzed for reference purposes. The different concepts have been extensively evaluated by methods described by Pahl & Beitz [5].

The chosen concept is a single stage spur gear pump driven by an electric motor. The volume displaced at each revolution is $5cm^3$ and the output flow can be varied by altering the rotational speed of the electric machine. The output produced by the pump can by an intelligent control system be precisely matched to the flow demanded by the hydraulic system in the transmission. Compared to a conventional transmission pump where the output is constrained to the engine speed, less power is needed because of the demand oriented flow production. A reduction in driving power of up to 46% for a highway cruise cycle can be achieved by incorporating the electro-hydraulic pump concept into a transmission. This corresponds to a total fuel saving of approximately 1.8%. The overall efficiency for the pump concept developed is 85% under optimal operation conditions, for the reference pump, overall efficiency is 64% under the same conditions.

Packing size of the proposed design is strongly dependent on electric motor size since the motor is 5 times larger than the actual pump unit. This is due to the cold operation performance of the pump where low temperatures increase viscous friction dramatically and increases torque loss. If the cold operation performance was irrelevant and the motor could be designed for optimum operation conditions, the size of the motor could be reduced by 40%. The complete package including pump and electric motor is however substantially larger than the reference pump design, even if designing for optimum operation conditions.

Sammanfattning

Hybridfordon är ett steg närmare mer miljövänliga transporter. Denna typ av fordon kommer att spela en viktig roll i en inte allt för avlägsen framtid för att minska oljeberoendet samt minska förbränningsmotorns negativa effekt på exempelvis luft- och ljudföroreningar i stadsregioner. Hybridsystem kan byggas på en mängd olika sätt, den vanliga idén är dock att lägga till en elektrisk motor för ytterliggare drivningskraft. Detta ger också en möjlighet att återta den kinetiska energin som annars är förlorad under inbromsning.

I växellådan som är under utveckling används trycksatt olja för att växla, koppla och för att kyla roterande delar. I konventionella automatlådor trycksätts oljan av en hydraulpump som är driven av förbränningsmotorn. En hybridväxellåda behöver trycksatt olja även när förbränningsmotorn inte är i drift. I detta examensarbete utvecklas ett förslag på elektriskt driven hydraulpump för en hybridväxellåda. Målet var att utveckla ett pumpkoncept som klarar drift i alla möjliga körfall för ett hybridfordon. Konceptet är utvecklat med hänsyn till kapacitet- och tryckkrav, vintertemperaturer, verkningsgrad och storlek.

Tre stycken hydraulpumpskoncept har utvecklats och analyserats. En befintlig växellådspump har också studerats för referenssyfte. De olika koncepten har blivit utförligt granskade med metoder beskrivna av Pahl & Beitz [5].

Det valda konceptet är en enkel kugghjulspump driven av en elektrisk motor. Deplacementet är $5cm^3$ och flödet kan varieras genom att variera hastigheten på elmotorn. Det producerade utflödet styrs av ett intelligent kontrollsystem som matchar utflödet med flödet som hydraulsystemet i växellådan kräver. Jämfört med en konventionell växellådspump, där utflödet är bundet till förbränningsmotorvarvtalet, behövs mindre effekt eftersom utflödet styrs av flödesbehovet från hydraulsystemet. En besparing på upptill 46 % av drivkraften under en motorvägscykel kan uppnås genom att använda en elektrohydraulisk pump i växellådan. Detta motsvarar en bränslebesparing på cirka 1.8 %. Den totala verkningsgraden för pumpkonceptet är 85 % under optimala förhållanden, motsvarande siffra är 64 % för referenspumpen.

Storleken på det föreslagna konceptet är starkt beroende av elmotorstorleken eftersom elmotorn är cirka 5 gånger större än själva pumpmodulen. Detta beror på vintertemperaturkravet då låga temperaturer ökar den viskösa friktionen dramatiskt och därmed ökar momentförlusten. Om vintertemperaturkravet skulle vara irrelevant och att motorn kunde designas efter optimala förhållanden kunde storleken på motorn reduceras med 40 %. Det kompletta paketet, med elmotor och pump, är dock betydligt större än referenspumpen även om pumpen skulle designas efter optimala förhållanden.

Preface

This master thesis report is the written result of 20 weeks work and final part of the Master of Science program in Mechanical Engineering at Luleå University of Technology. The project was performed between January 2006 and May 2006 at GETRAG FORD Transmissions Sweden AB, located in Gothenburg, Sweden.

My thanks go to my supervisor Professor Roland Larsson at the Division of Machine Elements at Luleå University of Technology for his guidance and support and also my supervisors at GETARG FORD Transmissions, Annika Nissen and Lars Bergkvist for their support throughout the thesis work. I would also like to thank the entire Rotating Parts Design Group for invaluable help and assistance and also Dr. John Lord and Sture Johansson at the Analysis & Verification department.

Jonas Carlsson
Gothenburg, May 2006

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1. Introduction

Chapter one gives an introduction to GFT where this thesis work has been done, some background information about hybrid transmissions and a definition of the current problem which is to be solved.

1.1 GETRAG FORD Transmissions

GETRAG FORD Transmissions is a global transmission manufacturer and specialist in integrated transmission development. The company produces both manual and automated manual transmissions and have around 3,900 employees. It was founded on 1 February 2001 as a joint venture between the transmission specialist GETRAG and Ford Motor Company. The yearly production volume is about to exceed 2 million transmissions by the end of 2006. The expanding company is investing in the powertrain technologies of the future like modern automatic transmissions and electrical drives.

Customers are among others: Ford, Volvo and Land Rover

1.2 Background

Automobiles have made great contributions to the modern society by satisfying the needs for mobility. The automotive industry and the industry serving it represent the spine of the world's economy and employ the greatest share of the working population [1]. However, air pollution, global warming and the rapid reduction in the Earth's petroleum resources are major problems related to the world of mobilization. The reduction together with political instability in many of the oil rich regions has led to a severely increased oil price over the last couple of years, and as it looks today, it will keep rising.

In recent decades, the research and development have focused upon developing high efficiency, clean and recyclable transports. Electric vehicles, hybrid electric vehicles and fuel cell vehicles have been typically proposed to replace the conventional vehicle incorporating only an internal combustion engine for propulsion. This thesis will focus on the hybrid electric vehicle and the transmission integrated in it.

A hybrid vehicle utilizes at least two different energy storages, hence the word hybrid. Many car manufactures are working hard to put hybrid cars on the market. One reason is that the customer demand for hybrid cars are growing, due to rising fuel prices, tax benefits for hybrid vehicles and an increased environmental concern.

1.3 Hybrid Electric Vehicle

A HEV is one that offers a mix, or hybrid, of two sources of propulsion power [8]. The two sources of propulsion power are the internal combustion engine and an electric motor, or motors. For the purpose of recapturing part of the braking energy that is wasted as heat in a conventional ICE vehicle, a hybrid drive train usually has a bidirectional energy source and converter. Figure 1 shows the concept of a hybrid drive train and the possible different power flow routes.

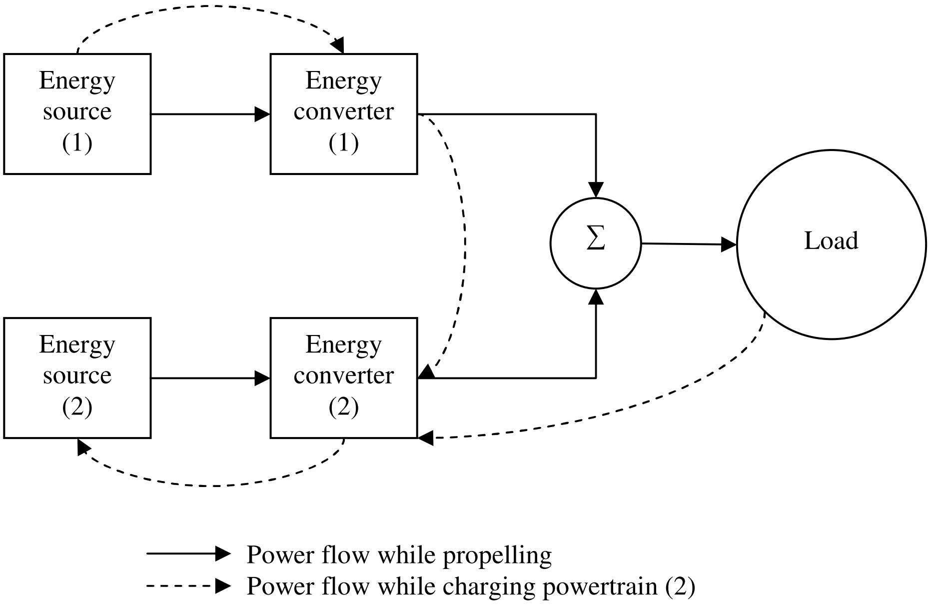

Figure 1: Illustration of hybrid electric drive train

There are many available patterns of combining the power flows to meet load requirements. Eshani [1] describes nine different examples:

1. Power train 1 alone delivers power to the load
2. Power train 2 alone delivers power to the load
3. Both power train 1 and 2 deliver power to the load at the same time
4. Power train 2 obtains power from load (regenerative braking)
5. Power train 2 obtains power from power train 1
6. Power train 2 obtains power from power train 1and the load at the same time
7. Power train 1 delivers power to the load and to power train 2 at the same time
8. Power train 1 delivers power to power train 2, and power train 2 delivers power to load
9. Power train 1 delivers power to load, and load delivers power to power train 2

The varied operation modes in a hybrid vehicle create more flexibility over a single power train vehicle. With proper configuration and control, applying the specific mode for each special operating condition can optimize overall performance, efficiency and decrease emissions.

1.4 Architectures of Hybrid Electric Drive trains

The architecture of a hybrid vehicle can be classified into two basic types: series and parallel. This has to do with the connection between the components that define the energy flow routes. However, Eshani [1] suggests a new classification due to the complexity of some architecture. Therefore, HEVs are now classified as four kinds: series hybrid, parallel hybrid, series-parallel hybrid and complex hybrid. The former two are shown in figure 2 below.

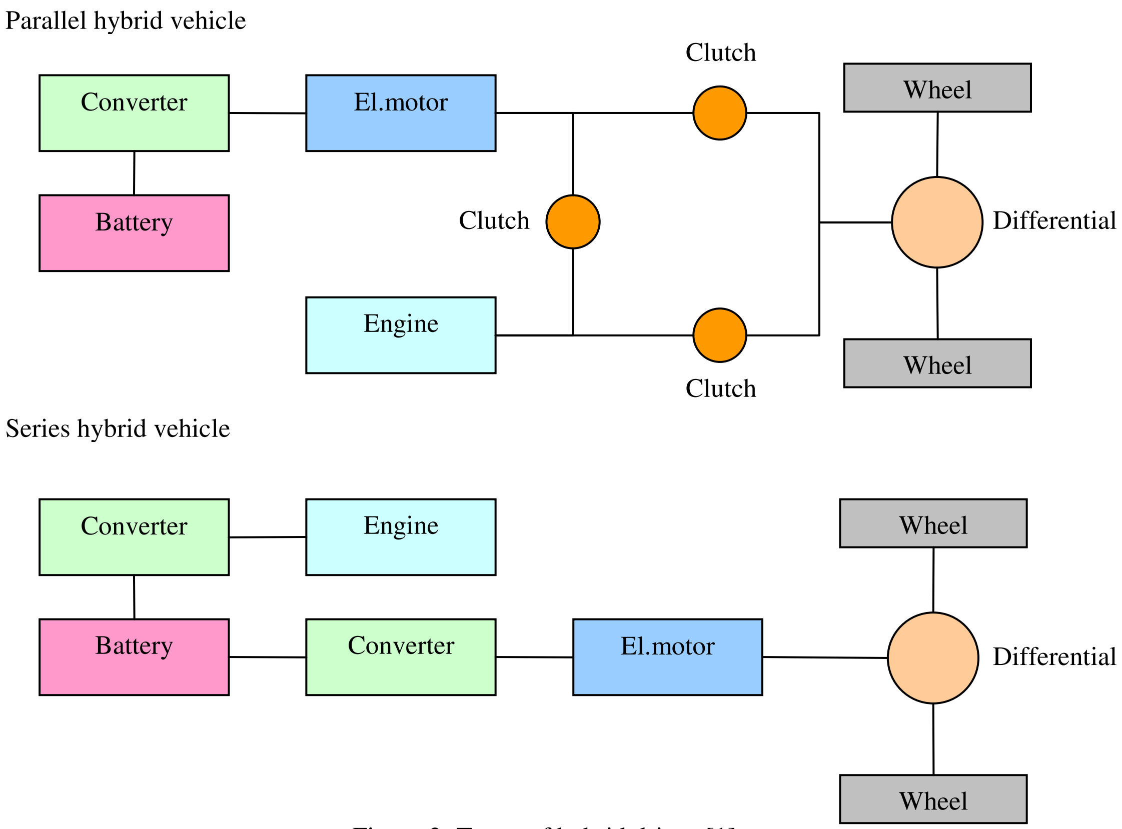

Figure 2: Types of hybrid drives [1]

1.4.1 Parallel Hybrid Drive trains

The parallel hybrid is a combination of drive systems. The ICE (engine) is mechanically connected to the wheels via a clutch and a gearbox. The working point of the hybrid can be chosen relatively freely with the help of the electrical machines, i.e. the speed of the engine is chosen with the gearbox and the torque with the electric motor. There are three options available: pure electric operation, pure ICE operation and a combined operation when the electric drive absorbs or delivers power to improve the ICE operating point. To achieve peak tractive power, both the ICE and the electric motor are used.

1.4.2 Series Hybrid Electric Drive Trains

The series hybrid has no mechanical connection between the ICE (engine) and the wheels. The ICE working point, i.e. speed and torque, can therefore be chosen freely, but at the expense of many energy conversions. The thermal energy is converted into mechanical energy in the ICE, and thereafter, in the power converter, turned into electric energy. The generator charges the battery that in its turn supplies the electric traction motor. On its way the energy also passes power electronics twice. These many energy conversions affect the system efficiency in a negative way.

1.5 Task

The task given for this thesis work is to design an electric oil pump for a hybrid transmission. Oil in a transmission is used for several purposes: lubrication, cooling and hydraulics. Pressurized oil is necessary for gearboxes utilizing hydraulics for gear shifting operations and clutch activation. A hydraulic pump driven by the combustion engine is normally used for this purpose. In a hybrid vehicle, oil pressure is required also when the combustion engine is not running. This is why a design concept for a hydraulic pump is necessary.

1.6 Goal

The goal of the thesis is to design a hydraulic pump able to operate in all possible driving modes of a hybrid vehicle. The concept should be designed considering capacity and pressure requirements, winter temperature performance, efficiency, packaging space and cost. The proposal will also contain a system sketch, package model and relevant calculation data.

1.7 Limitations

- The capacity and pressure requirements are based on data from the hydraulic system used in the GFT base transmissions. This is because of the hybrid transmission under development will be a carry over from the current transmissions. The hydraulic package will therefore by certainty be somewhat similar to the current package.

- The design solution should not trespass any approved patents or patent applications.

- The design concept will not present a highly innovative pump design. Work has been made in order to verify the operability of the design in all driving modes and to optimize the efficiency.

- The actual strategies for driving the pump in the most efficient way will not be considered.

2. Theory

This chapter describes some of the basic theory incorporated in automotive transmissions. The current Powershift technology and some hydraulics and fluid dynamics are also described.

2.1 Automotive transmissions

When developing a vehicle transmission the main goal is to convert power from the power source into vehicle traction as efficiently as possible over a wide range of road speeds. This has to be done ensuring a good compromise between the number of speeds, grading performance, acceleration and fuel consumption. Since the torque/speed profile of the ICE is not suited to be used in motor vehicles, transmissions are needed for the final output. This is to approximate the ideal engine characteristics with constant maximum engine power over the entire engine speed range. Clutches serve to adapt engine speed, transmissions serve to adapt both speed and torque.

Lechner [2] suggests the following classification of passenger car transmissions into the following main designs and types:

- Conventional 4-6 speed transmissions
- Semi-automatic transmissions
- Fully automatic transmissions
    - Conventional 3-speed to 5-speed automatic transmissions (consisting of torque converter and rear-mounted planetary gear)
    - 3-speed to 6-speed automatic countershaft transmissions
- Mechanical continuously variable transmissions

2.2 Powershift

The Powershift design is based on a manual gearbox with two layshafts, see figure 3. Unlike a manual the two clutches in the gearbox are linked to two input shafts and the shift and clutch actuation is controlled through a mechatronic module integrating the electronic and hydraulic elements. Like in a tiptronic type step-automatic transmission, the driver can change the gear manually or leave the shift lever in a fully automatic D (comfort oriented, shifting at lower engine speeds) or S (performance oriented, shifting at higher engine speeds) mode. The shift itself is always automatically handled by the shift and clutch actuators. Two clutches are linked to separate input shafts. Clutch 1 (red) is linked through an inner solid shaft with the gears 2, 4 and 6. Clutch 2 (blue) is linked through an outer hollow shaft with the gears 1, 3, 5 and reverse. The input shaft from the engine is connected through a damper with outer plates of both clutches. When starting the engine the first gear is engaged. As clutch 2 is open there is no torque transfer to the wheels. When clutch 2 is being closed the outer plates of clutch 2 are getting into slipping contact with the inner plates smoothly starting to transfer engine torque through the hollow shaft, gear set, to the differential and finally the wheels. In parallel the second gear is already pre-selected, which can be done, as clutch 1 does not transmit any torque at this moment. When shifting from first into second gear, full forward thrust is ensured as clutch 2 disengages at the same speed and progression as clutch 1 engages. When clutch 1 is fully engaged, third gear can be pre-selected, as now clutch 2 does not have to transmit torque. For fast shift times the next gear that is linked to the open clutch is generally pre-selected. Choosing the right gear is secured through complex algorithms in the transmission control unit (TCU) adjusting shift pattern and shift speed to the individual driver behavior. A car equipped with a Powershift gearbox will reach the same top speed as the manual transmission equipped car. However, Powershift will have higher acceleration as there is no torque interrupt during the shift and it can provide a 10% fuel efficiency improvement due to the optimized shift pattern to run the engine continuously in the best efficiency area [9].

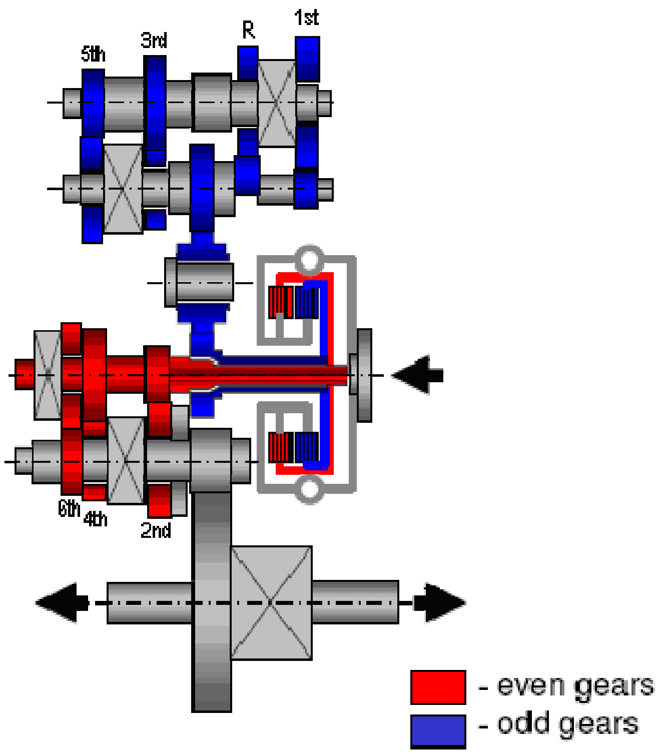

Figure 3: GETRAG FORD Transmissions SPS6 Powershift [16]

2.3 Hydraulic power transmission

The hydraulic package in the transmission converts mechanical energy from the internal combustion engine to fluid energy using a hydraulic pump. The fluid energy is guided through orifices in the valve body to solenoids and actuators which convert the energy back to mechanical energy by gear shifting operations. The clutches are also engaged and disengaged by hydraulic power. In addition, thermal cooling of individual parts in the transmission is achieved by oil flow supplied by the pump.

Transmission oil pumps are driven either directly from the input shaft or at some speed ratio relative to the input shaft. A transmission oil pump must be designed on order for the pump to deliver sufficient flow at low engine speeds, but must also avoid over pressurization of the lubrication system at high engine speeds. Because of the risk of over pressurization usually a pressure activated bypass circuit is used to limit the flow and resultant pressure at high engine speeds. The relief is internal to the pump and usually redirects excess oil flow back to the inlet port of the oil pump. The oil pressure relief is necessary to regulate the oil flow to the system, but there is an associated energy penalty. Figure 4 provides some basic insight into how much energy the system is taking from the input shaft. At the point where the flow from the pump goes into recirculation, the oil pump is converting just enough mechanical power into hydraulic power for the system to work. As the input shaft and oil pump speed increase, the oil pump is continuing to convert additional power, not because the system needs more power, but because the pump is re-circulating flow internally. The ideal pump would deliver an equal amount of oil to that which the system is demanding. This could ultimately reduce the power consumed by the system.

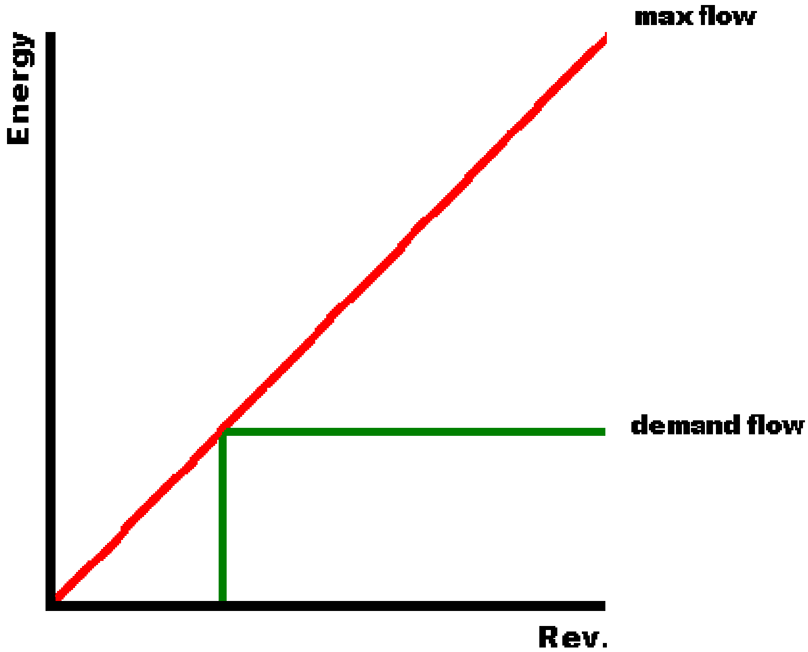

Figure 4: Flow produced by the pump and flow demanded by the system

A reduction in pumping power will result in better fuel economy as the transmission will require less power to rotate and hence the engine will require less fuel to operate. Figure 5 shows the potential for ultimate fuel savings by reducing the power consumption from the auxiliaries.

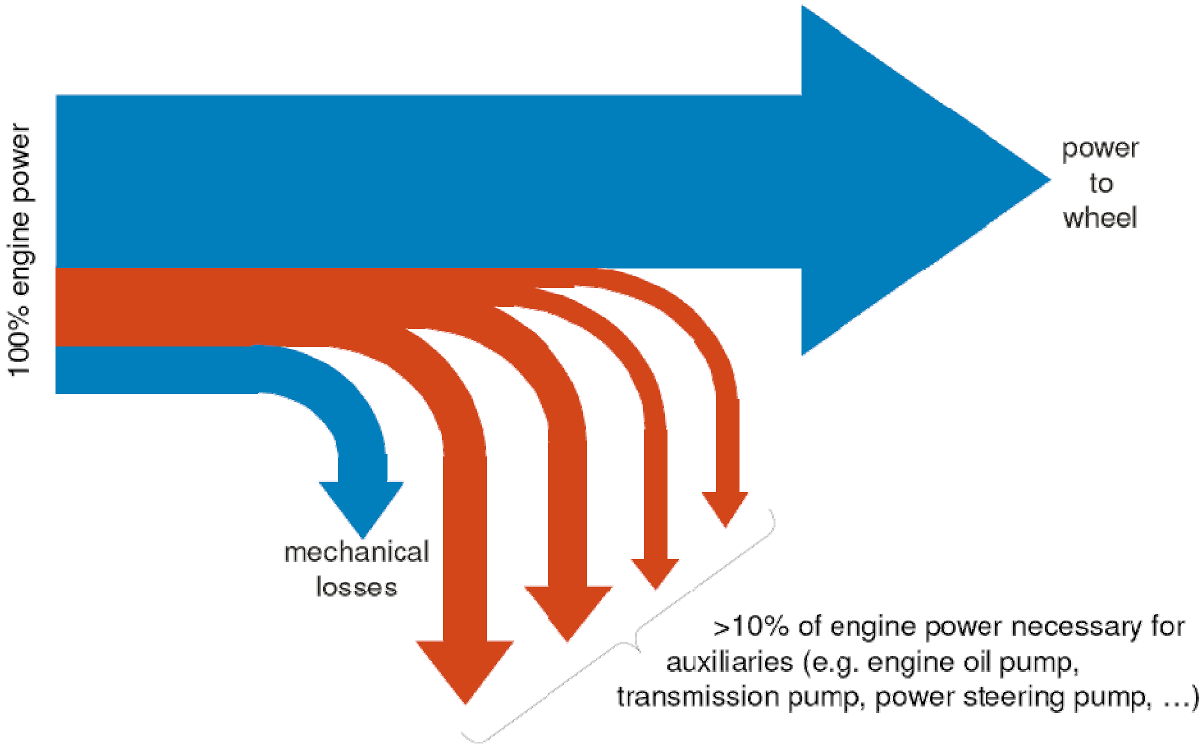

Figure 5: Auxiliary driving power [13]

2.3.1 Pump types

Pumps are classified in two main categories, positive displacement and non-positive or impeller based pumps. This thesis will focus on positive displacement pumps. The two most common pump types in automotive transmissions are the gear- and the vane type. This is because of the rigid and inexpensive design associated with the two. Unlike the gear pump, the vane type can be made with a variable displacement. This means that the vane pump can vary its output flow while the gear pump displaces a fixed quantity of fluid per revolution. Since the output is positive for a given operating speed, it is not significantly affected by resistance to flow. Instead, the resistance to flow will dictate the pump output pressure. This is the most important characteristic of a positive displacement pump.

The most widely used positive displacement pump is the gear pump. This is largely due to the simplicity and robustness of their design. There are two basic types of gear pumps. The first type is the external gear pump. It is termed this because the gear teeth are external to hub surface of the gear. The most common type of an external gear pump is the spur gear pump, see figure 6. The second type of gear pump is an internal gear pump. Here an external gear is replaced with an internal gear where the gear teeth are internal to the hub of the gear. The most common type of an internal gear pump is the Gerotor pump. The typical construction of an external gear pump consists of two or more spur gears, a drive shaft and a housing to hold it all. The spur gears are set in the housing between the low pressure side and the high pressure side of the pump. These sides are termed the inlet or suction side and the outlet or pressure side respectively. The first of the two gears is connected to the drive shaft, this gear is termed the drive gear. The second gear is called the idler gear, it meshes with the drive gear and is driven by it.

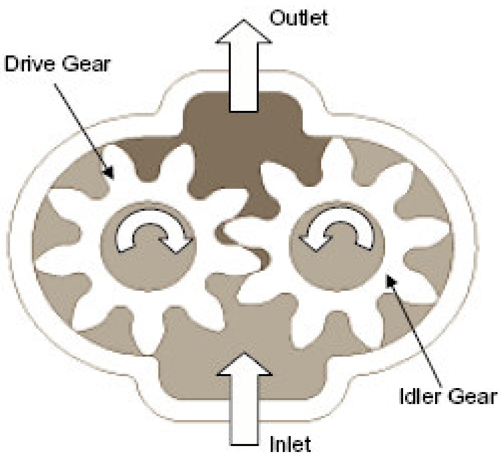

Figure 6: External gear pump and the operational principles associated with the pump [10]

When the drive shaft rotates it in turn rotates the drive gear which then drives the idler gear. As the gears rotate, the gear teeth unmesh on the inlet side and mesh on the outlet side of the pump. The volume on the inlet side increases as the gear teeth unmesh. This causes a reduction in pressure to a partial vacuum. Atmospheric pressure forces oil into the inlet chamber. Oil is then transported around the housing from the inlet side to the outlet side, via pumping chambers created by the gear teeth cavities and the housing. The volume decreases on the outlet side as the gear teeth mesh. This force the oil to be discharged from the pump as the fluid cannot travel back to the inlet side. It should be noted that the actual pumping occurs in the meshing area of the gear teeth. The volumetric displacement of this pump is determined by the size of the teeth cavities.

The second most common type of positive displacement pump is the vane pump. The basic construction of a vane pump consists of a cam ring, vanes, a slotted rotor, a drive shaft and a housing to hold it all. The slotted rotor sits inside the cam ring and is eccentric to it. By varying the eccentricity of the rotor the displacement can be altered. Figure 7 shows the construction as well as the basic operating principle of a typical vane pump. The openings between the vanes, rotor and cam ring forms the actual pumping chambers. As the rotor begins to rotate, at a specific speed, the vanes will be forced out to contact the cam ring. Centrifugal force will ensure constant contact of the vanes with the ring forming a positive seal. As the rotor spins, on the inlet an increasing volume is created for one half of each rotation. On the inlet side the pumping chambers are increasing in size which creates a partial vacuum. Atmospheric pressure forces oil into the chambers. Oil is then transferred to the outlet side where a decreasing volume is created. As the pumping chambers decrease in size, oil is discharged form the pump via the outlet. The inlet and outlet ports provide a perpendicular fluid passage to the rotor face, thus keeping the inlet and outlet flow separated. The inlet port is positioned in the location where increasing volume is formed, where as the outlet port is positioned in the location where decreasing volume is formed.

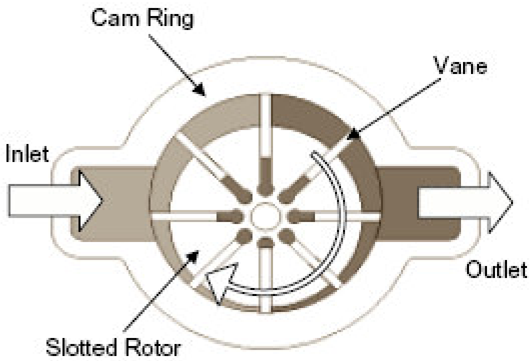

Figure 7: Vane pump section and the operation of the system [10]

The gear type pump has several advantages over the vane pump. These advantages are higher output flow, higher speed capability, higher mechanical efficiency, simpler serviceability, greater tolerance to contamination, and finally lower cost. The vane pump is comparable to the gear pump in terms of output pressure level. However, a vane pump has the advantage of higher volumetric efficiency and the ability of a variable displacement design compared to the gear pump.

2.3.2 Losses in hydraulic power transmission

Owing to viscous friction of the hydraulic oil, imperfect sealing of the elements performing the displacement and mechanical friction of the moving parts in the pump, hydraulic power transmission can only be realized at the expense of losses [3]. The power input in every hydraulic machine is divided into the following parts:

$$P_b = P_h + P_p + P_{vol} + P_{mech}, \qquad (1)$$

where P_b is the power input, P_h is the effective power, P_p is the hydraulic power loss produced by friction of the fluid, P_{vol} is the volumetric power loss produced by the clearance losses and P_{mech} is the mechanical power loss.

The power losses are taken into account with the efficiency. Hence the losses arising during hydraulic power transmission are incorporated with the hydraulic, volumetric and mechanical efficiency. The total efficiency is therefore the product of these three efficiencies:

$$\eta_{tot} = \frac{P_h}{P_b} = \eta_p \cdot \eta_{vol} \cdot \eta_{mech}, \qquad (2)$$

where η_{tot} is the total efficiency, η_p is the hydraulic efficiency, η_{vol} is the volumetric efficiency and η_{mech} is the mechanical efficiency.

Since this thesis is focusing on the pumping element in the hydraulic system, the hydraulic power loss produced by viscous friction will not be investigated. This is because of the complexity and limited knowledge in such analysis. The validity of the results found from such analysis would also be highly uncertain because of the high variability in the characteristics of the parameters controlling the hydraulic power loss. The type of flow through the pump is expected to be highly turbulent generated by the flow rate and sudden changes of sectional area. This will make calculations even more complex.

2.3.3 Volumetric efficiency

Volumetric efficiency is defined as:

$$\eta_{vol} = \frac{Q_k}{Q_b} = 1 - \frac{\Delta Q_v}{Q_b}, \qquad (3)$$

where ΔQ_v is the flow rate of the clearance loss, Q_b is the flow rate measured on the inlet of the pump and Q_k is the flow rate on the outlet of the pump. The clearance losses are a result from flow between the pumping element and the housing. This flow is driven by a combination of an externally imposed pressure gradient and the motion of the pumping element. This flow are considered a two dimensional steady flow between parallel plates, see figure 8.

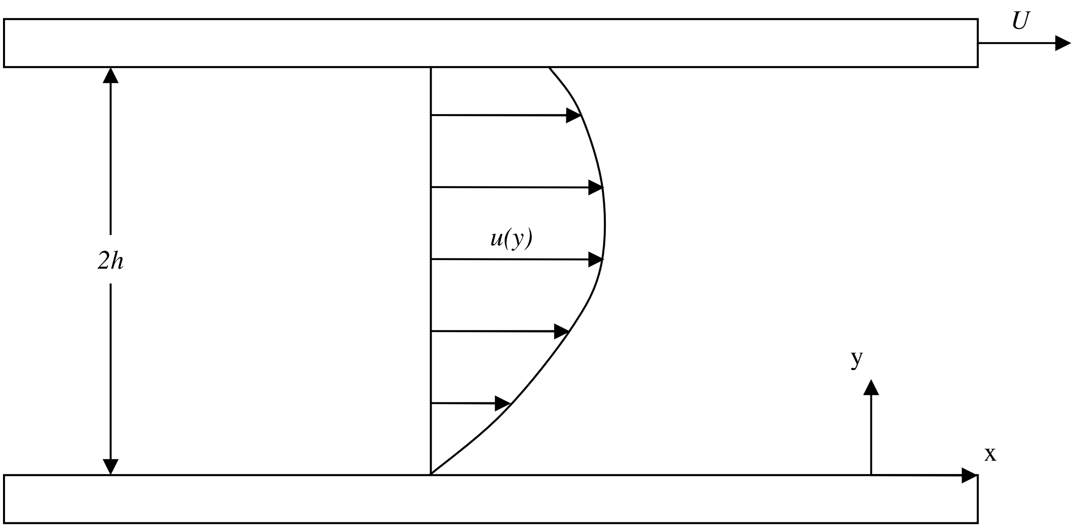

Figure 8: Flow between parallel plates

Since the pressure gradient is constant [4], the velocity profile becomes:

$$u = \frac{yU}{2h} - \frac{y}{\mu}\frac{dp}{dx}\left(h - \frac{y}{2}\right). \qquad (4)$$

The first term is this equation is the Couette flow. This flow is driven by the motion of the moving plate alone, without any externally imposed pressure gradient, see figure 9. Because of the velocity dependence of the Couette flow, this flow will change direction according to the velocity. The last term is called the Poiseuille flow. This flow is driven by the externally imposed pressure gradient through two stationary walls, see figure 10.

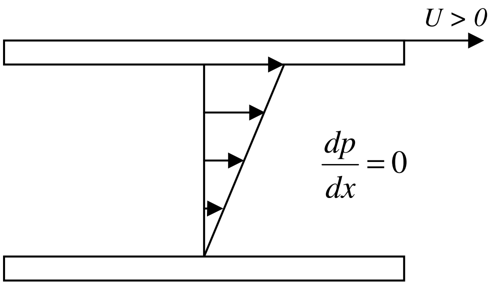

Figure 9: Plane Couette flow

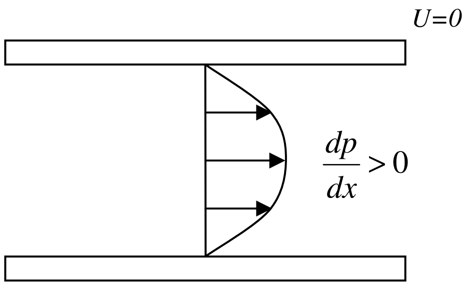

Figure 10: Plane Poiseuille flow

The flow rate per unit width of the clearance is given by:

$$\Delta Q_v = \int_0^{2h} u dy = Uh\left[1 - \frac{2h^2}{3\mu U}\frac{dp}{dx}\right]. \qquad (5)$$

For the vane and spur gear pump design, this implies that the clearance losses will depend principally on the velocity of the rotating parts and the clearance between the rotor and the housing and the teeth or vanes and the housing wall. Also the viscosity of the oil plays a fundamental role in the overall efficiency. The volumetric losses for a spur gear pump design also incorporate a term which defines the volume of oil trapped between the teeth when meshing. This volume of oil will be delivered back to the low pressure side when the gears unmesh. The volume will be dependent on the free space between tooth tip and tooth root of the meshing gears. Since *dx* is defining the pressure gradient, the term *dx* will also have a have high influence of the flow rate between the housing and the rotational parts.

2.3.4 Mechanical efficiency

Mechanical efficiency is defined as:

$$\eta_{mech} = \frac{M_{th}}{M_b} = \frac{M_{th}}{M_{th} + M_f}, \qquad (6)$$

where M_{th} is the theoretical output torque, M_b is the torque on the input shaft supplied by the driving source and M_f is the torque loss due to friction. Due to the clearance previously described, the rotating parts and the housing's surfaces will be separated by hydraulic oil, here acting as lubricant under pressure. If it is assumed that full film lubrication is present at all operating conditions the friction is caused only by viscous shear of the oil, see figure 11.

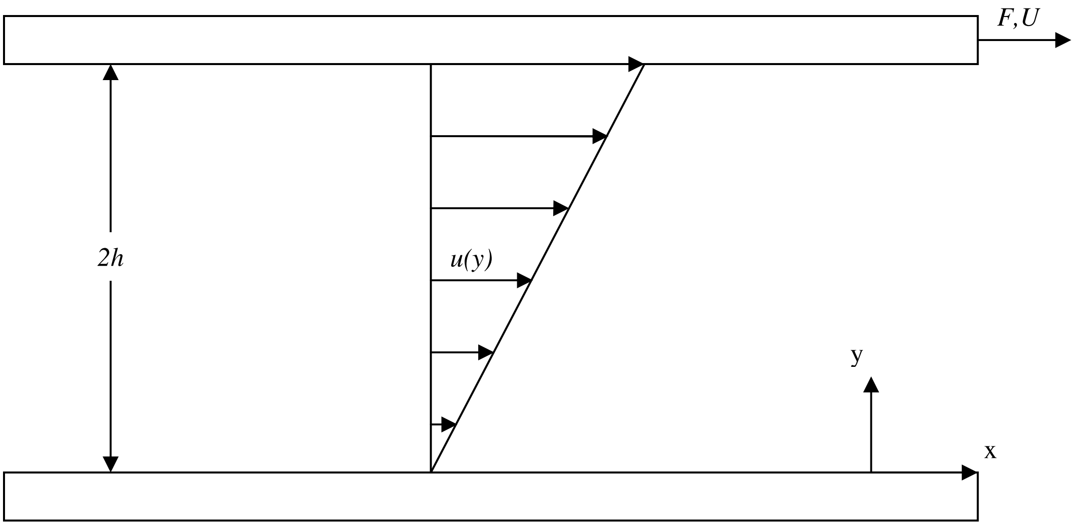

Figure 11: Viscous shear of oil between two parallel plates

The transmission under development utilizes the Newtonian fluid BOT341 for lubrication, coolant and pressure media. For a Newtonian fluid the magnitude of the shear stress, τ, along the surface is related to the velocity gradient by the linear relation:

$$\tau = \mu \frac{du}{dy}, \qquad (7)$$

which is called Newton's law of friction. Here the constant of proportionality, μ is known as the dynamic viscosity of the oil.

The force that must be applied to the plate in order to receive the velocity U is then:

$$F = \frac{A\mu U}{2h}, \qquad (8)$$

where, A is the are area of contact for the moving surface. But since the surface is rotating, the velocity, U, will increase with the radius from the axis of rotation. This is defined as the linear equation:

$$U(r) = \omega dr, \qquad (9)$$

where ω is the angular velocity. The torque loss due to viscous friction then becomes:

$$M_f = \frac{\mu}{2h}\int_{r_0}^{r_1} AUdr. \qquad (10)$$

2.3.5 Cavitation

Cavitation can be defined as the breakdown of a liquid medium under very low pressures. The absolute pressure above the vapor pressure available at the pump inlet must always exceed the absolute pressure above the vapor pressure required by the pump. The creation of vapors and the following collapse of the vapor bubbles upon reaching the high pressure side of the pump will cause cavitation. The phenomena forces the liquid into the vapor voids at high velocities and produces local pressure surges of high intensities imposed on the pump surfaces. These forces can exceed the tensile strength of the material of the pump, causing pitting of the gears and erosion of the vanes and housing walls. In addition, cavitation causes noise, vibration, and a loss of output flow. The risk of oil cavitating in the system can be reduced by placing the pump in close connection with the sump. This will increase the net pressure in the inlet of the system. Increasing the inlet diameter and simplifying the channel layout between the pump and sump can also reduce the risk of cavitation.

With any rotary pump, as the cavities rotate, they present a void for the incoming fluid to fill. This void is available for a fixed amount of time, and the fluid in the suction chamber must accelerate to fill this void in the available time. The higher the fluid viscosity, the more energy is required to accelerate the fluid to fill the void. If the fluid cannot fill the void in the time available, a partial void and cavitation will occur. By proper design of the inlet, fluid will get more residence time to fill the void, and with larger internal passages and ports, entry losses will be reduced as well.

3. Method

In this chapter the systematic approach of the work is outlined. The approach is schematically illustrated as point 2-5 in appendix I and is taken from [17]. The methods used and the processes involved in each point will be described briefly below.

3.1 Design space exploration

The project started off with some intensive search for information about the technology involved in the task. This was done by literature studies and article search in databases, also, much information was available within GFT. In addition to the information search through articles and in literature, patent databases were scanned in order to find related technology. This resulted in better understanding of the current problem but also draw some limitation due to existing patents and patent applications. The information gathered during this phase was analyzed and documented in a feasibility report which has proven to be a practical guide through the rest of the project.

3.2 Roadmap

This phase of the project was aiming to clarify the given task in more detail before starting the product development. The purpose of this was to collect information about the requirements that have to be fulfilled by the design, and also about the existing constraints and their importance. This resulted in a formulation of a *requirement list* that focused on, and was tuned to, the requirements and the requests of the design. The concept design phase and subsequent phases was based on this list that had to be updated continuously during the project.

3.3 Concept design

This was the divergent phase of the project where a lot of concepts and ideas were generated. Several methods for encouraging the innovative mind that are associated with generation of ideas were employed. The simplest and the most used during the project involved critical discussions with the colleagues at GFT. Other methods with an intuitive bias such as Brainstorming have also been used in order to generate possible solutions to different problems. The use of brainstorming generated a flood of new ideas which later was reviewed with the aim of finding potential solutions.

The divergent phase was followed up by a convergent phase. This phase reduced the number of solutions that were found during the divergent phase. Care was taken not to eliminate valuable working principles, often it is only through their combination with others that an advantageous design or structure will emerge. This has been learned from previous experience from design work. All totally unsuitable design proposals were eliminated. Three different design proposals where found not to be unsuitable and these was further elaborated and evaluated.

The three different designs were comprehensively evaluated in order for decision making. A final evaluation using criteria that where more detailed and quantified was performed. This evaluation involved an assessment of technical and economical values. The results from this assessment was a selection of the design concept that was the most promising but which could nevertheless benefit from, and could be further improved by, incorporating ideas and solutions from the others. By appropriate combinations and the elimination of weak links, the best design concept could be obtained.

3.4 Detail design

This was the phase of the design process in which the most suitable concept was turned into a product. It involved the definition of the arrangement, form, dimensions and properties of all the individual parts which was carefully documented. With the aid of CAE-tools all the ingoing parts could closely be examined and the optimum design specification could be met. A mathematical model describing the moving parts was made in order to optimize the design parameters for optimum efficiency. With the optimum design parameters defined, development of solid body models and the final assembly structures could be made using a CAE-system. For mathematical studies and optimization, Matlab was used. For modeling of individual parts and creating assemblies, NX3 was used.

4. Concept design

In order to evaluate and finally determine the most suitable concept for the application previously described, this section gives a comprehensive definition of three different concepts as well as the pump and system requirements.

4.1 Hydraulic pump requirements

Since the aim for this thesis is to design a hydraulic pump concept suitable for hybrid transmissions, efficiency and package size have been extensively examined. Moreover, requirements such as flow capacity and pressure requirements, winter temperature performance and costs have also been taken into account. Table 1 below shows the complete requirement specification.

Requirements hydraulic pump	
Requirement	
Pressure, high pressure system	3,7-18 bar
Pressure, low pressure system	0-20 bar
Max. flow, pump side	34l/min
Max. flow, high pressure system	27l/min
Min. flow, high pressure system	2l/min
Max. flow, low pressure system	32l/min
Min. flow, low pressure system	4,5l/min
Max. temperature	150℃
Min. temperature	-30℃
Max. viscosity (kinematic)	3,2cSt @150℃
Min. viscosity (dynamic)	1510cP @ -26℃
Bulk modulus	assume incompressible
Pour point	<-40℃
Service life	>5000h
Price	low

Table 1: Requirement specification

The hydraulic system and the flow requirements of the system have to be examined closely in order to tune the design and maximize the efficiency of the pump. The system's requirements will be described in more detail in sections below. The current pump in the SPS6/MPS6 transmission will also be examined and considered a datum in order to evaluate the current concepts.

4.2 System requirements

The flow generated by the hydraulic pump is guided and distributed to the consumers by the valve body. The consumers are made up by the clutch control system, the shift control system, the solenoid fed, the clutch cooling system and the oil cooler and lubrication system. These are fed by the Line Pressure (LP) and Clutch Cooling Flow (CCF), see figure 12.

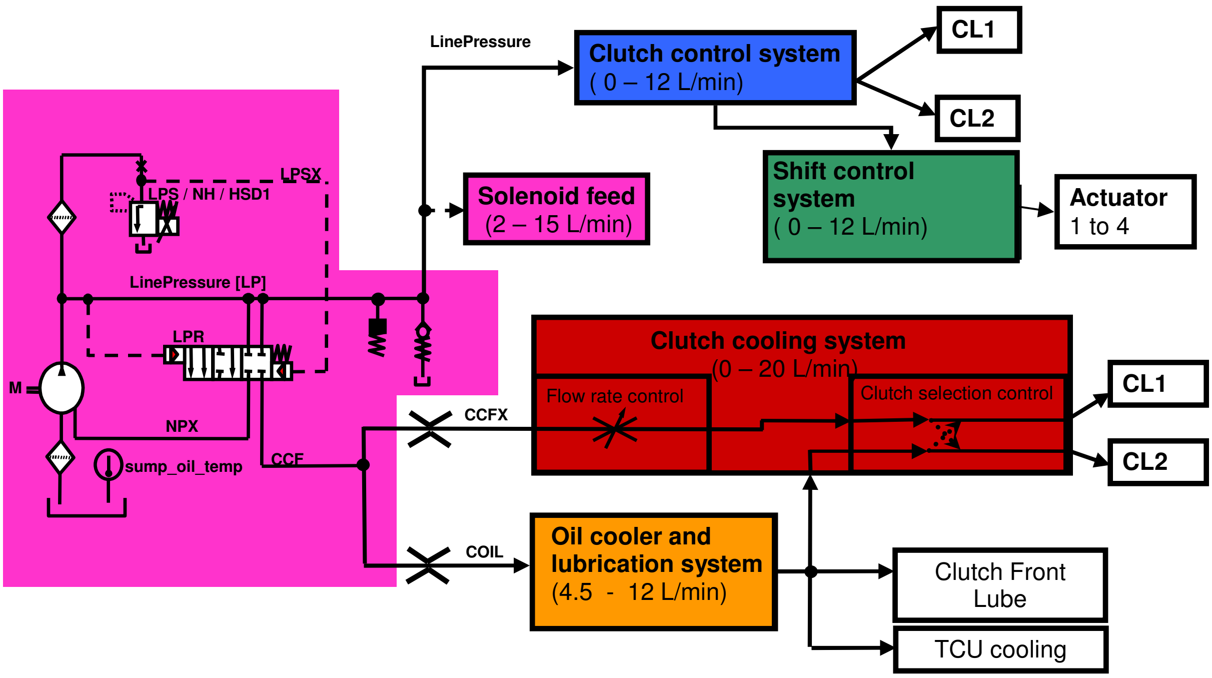

Figure 12: Flow management [14]

The flow management is designed to give priority of flow between the LP-system and the CCF [14].

The flow priority is set to be in descending priority order:

- Line Pressure (LP). Clutch and gear engagement and priority control
- Clutch Cooling Flow (CCF). Thermal management for launch, creep, stall, and hill hold operations
- Clutch cooling system controls priority between CCFX and COIL
- Recirculation flow (NPX)

The LP-system includes the clutch control system, the shift control system and the solenoid fed and requires a maximum flow of 27l/min at approximately 20bar. The CCF-system requires a maximum flow of 32 l/min at approximately 10bar. The clutch cooling flow feed the clutch cooling system, CCFX, and the oil cooler and lubrication system, COIL. The flow rate ratio between CCFX and COIL is variable from 0:1 up to 4:0 and is controlled by a flow rate control valve in the clutch cooling system. The oil cooler have a thermostatically controlled bypass valve which closes at oil temperatures above 75°C. The oil cooler and lubrication system also contains a pressure filter with an internal bypass valve which bursts at 15bar.

The minimum flow demands for the system at 10bar are assumed to have characteristics shown in table 2.

COIL		**CCF**		**CCFX**		**LP**		**System Flow demands**
Trans. lube	3l/min	Clutch lube	1l/min	Leakage	1l/min	Leakage	4,3l/min	
Clutch lube	1l/min	Leakage	0,3l/min					
Leakage	0,15l/min							
Flow demand for **COIL**	4,15l/min	Flow demands for **CCF**	1,3l/min	Flow demands for **CCFX**	1l/min	Flow demands for **LP**	4,3l/min	10,75l/min

Table 2: Minimum flow demands at 10bar

In the flow demands table presented above, the minimum flow is 10.75 l/min due to leakage and lubrication. This value is strongly dependent on the system and the value is subject to change due to future changes in the hydraulic system and number and type of clutches used. The value above is derived assuming a double wet clutch assembly.

The flow required by the system is determined by the functional states which is a function of the schematic of the system. Currently there exist 110592 states and all are ultimately dependent on the operating parameters such as driving situation, operating temperature and load. The functional state for shift operations within 50ms requires a flow of approximately 6 l/min to fill the actuator cylinder. During gear shifts COIL gets no flow, hence the total required flow for shift operations are 12.6 l/min. For clutch operations within 100ms a flow of approximately 5.5 l/min is required to fill the chamber. Likewise as for the gear shift operation, COIL gets no flow and hence the total required flow for clutch operations are 12.1 l/min. These values are only considered as guidelines due to possible changes in the future system. Flow required (Q_{req}) are defined as:

$$Q_{req} = Q_{CCF} + Q_{LP} . \qquad (11)$$

4.3 Current pump

The current pump in the SPS6/MPS6 transmission is a single stage spur gear hydraulic pump. The pump drive shaft is mechanically connected to the input shaft in the transmission by a gear ratio of 1:1.11. The volume displaced at each revolution is 14cm^3 and since the spur gear design, the displacement is constant over the full rev. range (500-6000 rpm). The drive shaft protrudes from the front plate where it is sealed by a shaft seal. The shaft has a pressed on pump gear and is supported by two needle bearings, one in the pump body and one in the front plate. The idler gear, which also is supported by two needle bearings, is pressed on a dowel pin. Figure 13 shows an exploded view of the pump assembly.

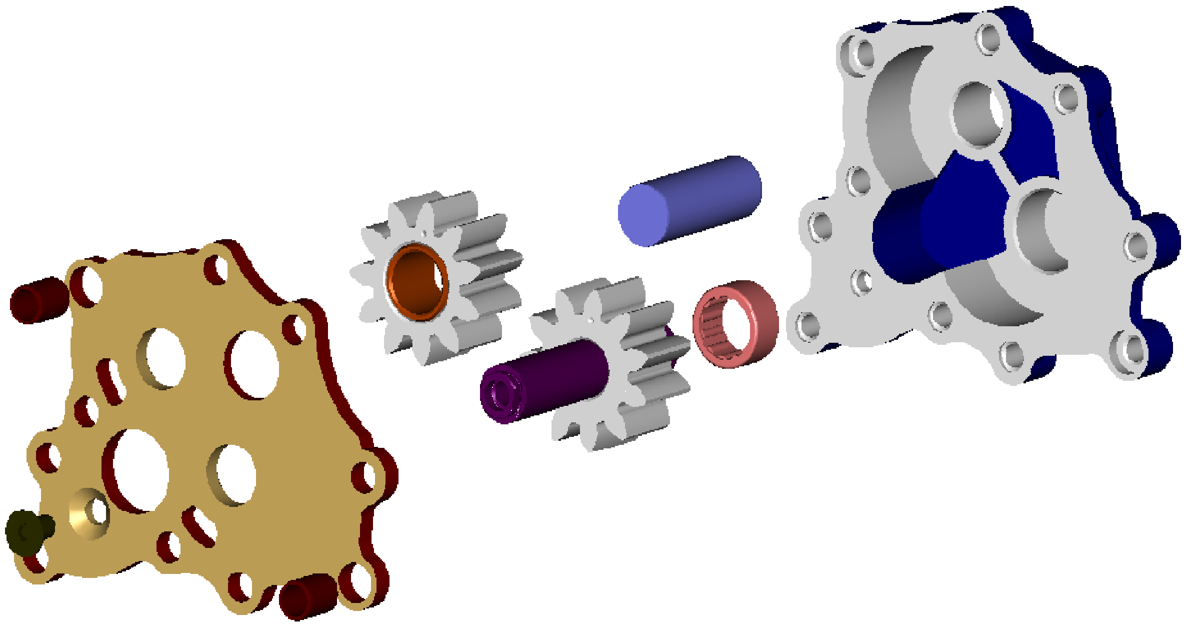

Figure 13: Exploded pump assembly

The suction side of the pump has one inlet entering the pump's body from the filter and three inlets are in connection with the recirculation port entering through the front plate. This allows the fluid's inertia to decrease the risk of cavitation during high speed operation. The pressurized oil is discharged through a single circular port on the front plate. The discharge port is equipped with pressure relief grooves in order to keep pulsation and noise emissions at a minimum.

Volumetric efficiency studies have been conducted in lab by B. Strerath [15] and the approximated results gained from these can be found in table 3. The measurements were performed at 120°C and viscosity of oil was 16cSt.

Pump [rpm]	700	1000	2000	3000	4000	5000	6000
Pressure[bar]				Efficiency [%]			
1	88	90	94	95	/	/	/
3	82	85	89	93	95	96	95
5	76	81	87	92	92	94	94
8	64	74	85	89	91	93	93

Table 3: Volumetric efficiency at different pump speeds and pressures

Measurements concerning the mechanical efficiency have also been conducted in lab. The results found in table 4 shows the mechanical efficiency at 80°C at different pressures and speeds.

Pump [rpm]	555	1110	1665	2220	3330	4440	5550
Pressure[bar]				Efficiency [%]			
5	67	68	69	69	66	60	53
8	67	68	70	70	71	67	62
10	63	69	72	73	72	70	65
12	65	69	72	71	73	71	68
15	/	/	/	/	/	69	72

Table 4: Mechanical efficiency at different pump speeds and pressures

The total efficiency has from these data been approximated for 5 and 8bar and reduced to equal conditions, see figure 14.

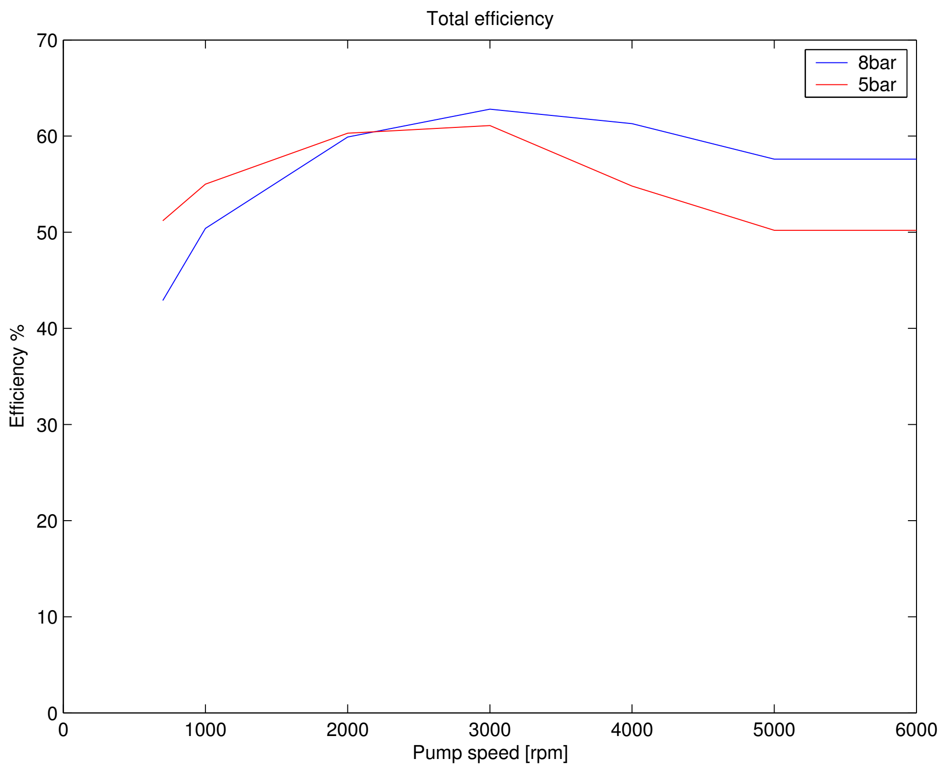

Figure 14: Total efficiency at different pump speeds

Since the displacement of the pump is constant and the ratio of rotation between pump and engine is fixed, the flow Q, will be directly proportional to the engine speed. However, due to volumetric losses, Q will not be equal to the flow produced, (Q_{out}). See table 5.

Motor [rpm]	500	1000	1500	2000	3000	4000	5000	6000	7000
Pump [rpm]	555	1110	1665	2220	3330	4440	5550	6660	7770
Theoretical flow, Q [l/min]	7,77	15,54	23,31	31,08	46,62	62,16	77,7	93,24	108,78
Actual flow, Q_{out} @ 8bar[l/min]	5,75	13,21	20,75	28,28	43,36	57,81	72,26	85,78	97,90

Table 5: Theoretical and actual flow rate

As mentioned earlier, the flow produced by the pump will be proportional to the engine speed. The required flow (equation 9) for the system will not have such a relation to the engine speed and hence an excess flow (Q_{ex}), will be produced. The excess flow will be dependent on the flow produced and the flow demanded by the system to operate, i.e. clutch cooling flow (Q_{CCF}) and the actuators in the high pressure system (Q_{LP}). Hence the excess flow is defined by:

$$Q_{ex} = Q_{out} - Q_{CCF} - Q_{LP} \qquad (12)$$

It is assumed that hydraulic friction depending on the design of the recirculation channels, velocity and type of flow will reduce the pressure to the initial value at the low pressure side of the pump. Hence the recirculation flow can be considered a loss of hydraulic power. In order to approximate the excess flow at different driving situation, three different flow profiles for the maximum required flow demands by the system has been constructed. The three different driving cycles are:

1. driving in 16% slope with trailer
2. aggressive city driving
3. high speed cruising

The different profiles are shown in figure 15 together with the theoretical flow produced by the pump.

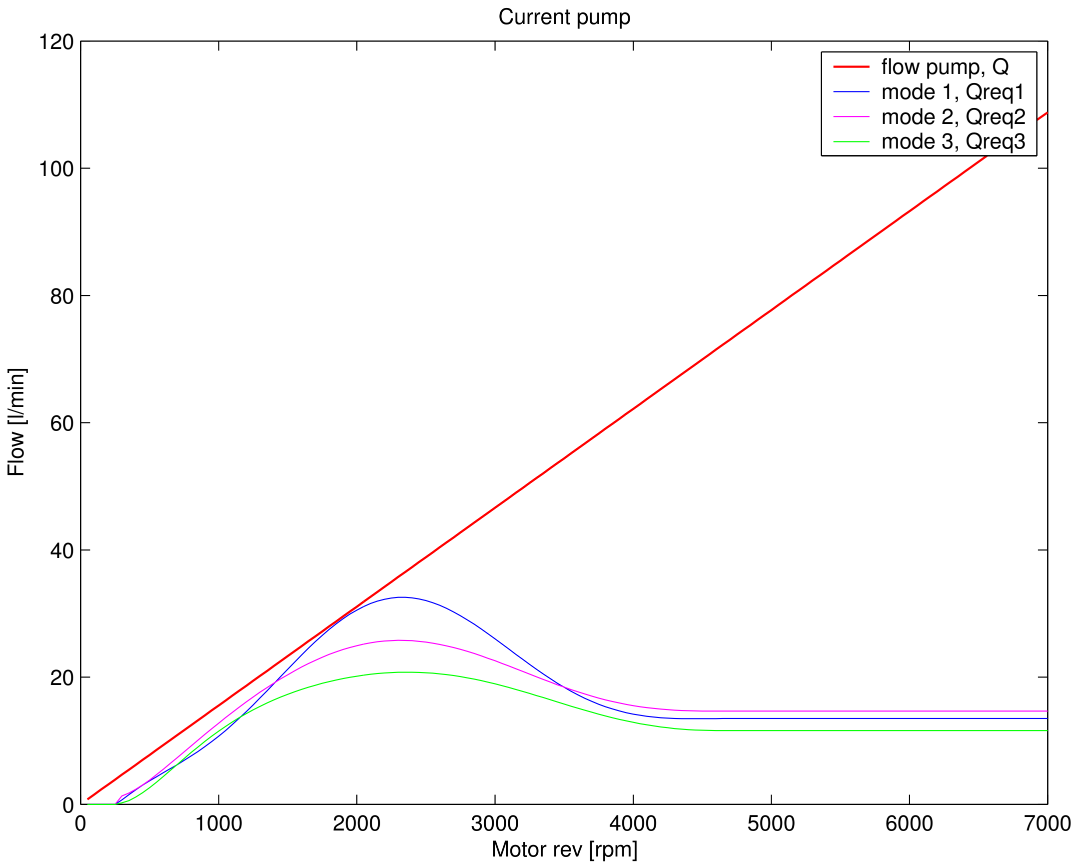

Figure 15: Flow profiles versus motor speed

The simulations were conducted in Matlab using data for the three different driving cycles stated in the list above for a standard diesel engine. The data are collected during long term tests and can be used in the simulation by calling the collected data file. Figure 16 shows the excess flow produced for the different cycles in the time domain.

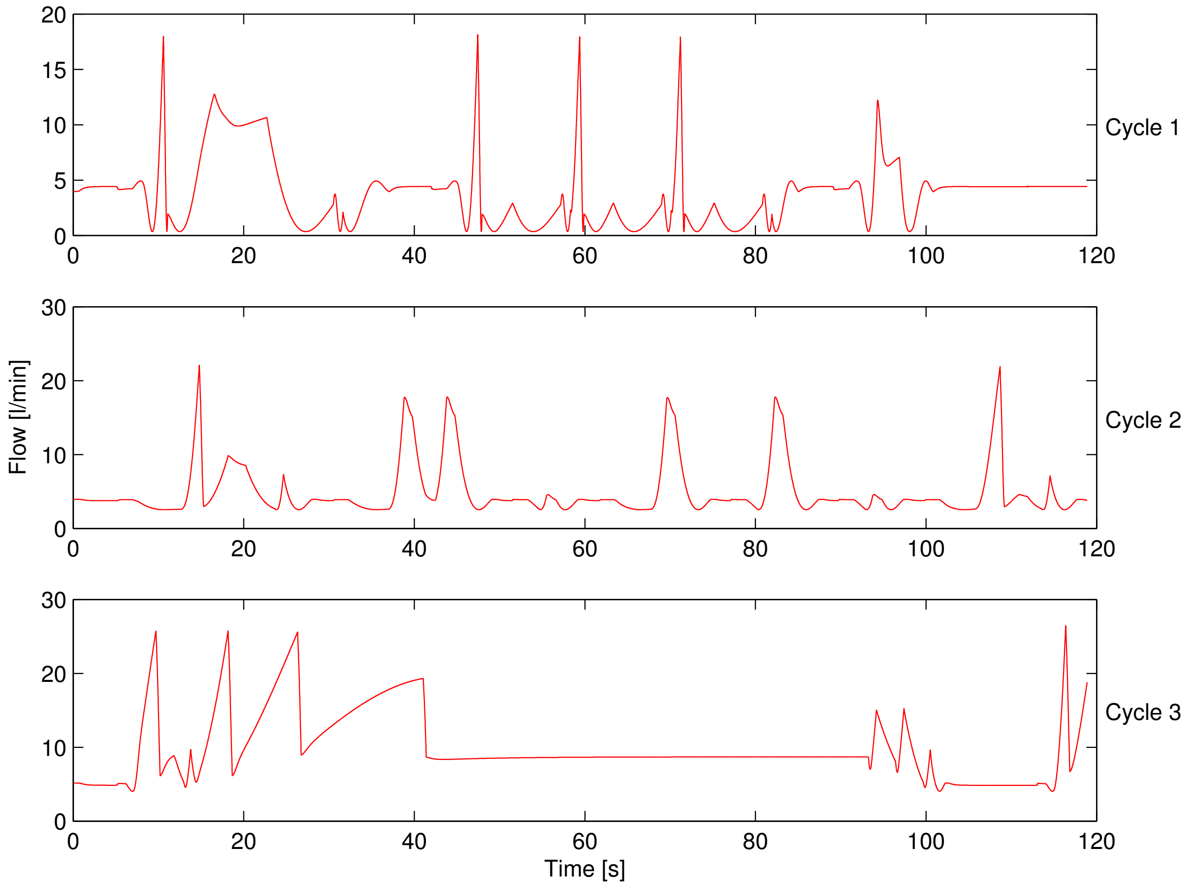

Figure 16: Excess flow produced during different driving cycles

It is clear that the hydraulic flow produced by the pump at each time step exceeds the flow required for the system to operate. For cycle 1, the average excess flow is 3.8l/min, in cycle 2, average excess flow is 4.9l/min and in the last cycle, average excess flow is 9.8l/min. This flow will, as stated before, be circulated back to the pumps inlet through the recirculation channel.

4.4 Concept 1

Concept 1 consists of a single multi vane hydraulic pump with a variable displacement. The drive shaft of the pump is connected by a gear linkage to the input shaft in the transmission. The design of the pump allows for the displacement to be changed automatically during operation. This is done by changing the eccentricity of the pump chamber. The movement required for the operation is done by an internal actuator in the housing of the pump. Flow produced is in this concept dependent on the speed of the input shaft and the displaced volume. In order to manage all possible driving situations the vane pump must be supported by an electro-hydraulic gear pump connected to the system in parallel. The mechanically driven vane pump must be placed in close connection with the input shaft and close to the sump in order to minimize the risk of cavitation. The electrically driven pump can be placed in a suitable location, preferably in connection to the sump for the same reason. Figure 17 gives a basic over view of the system.

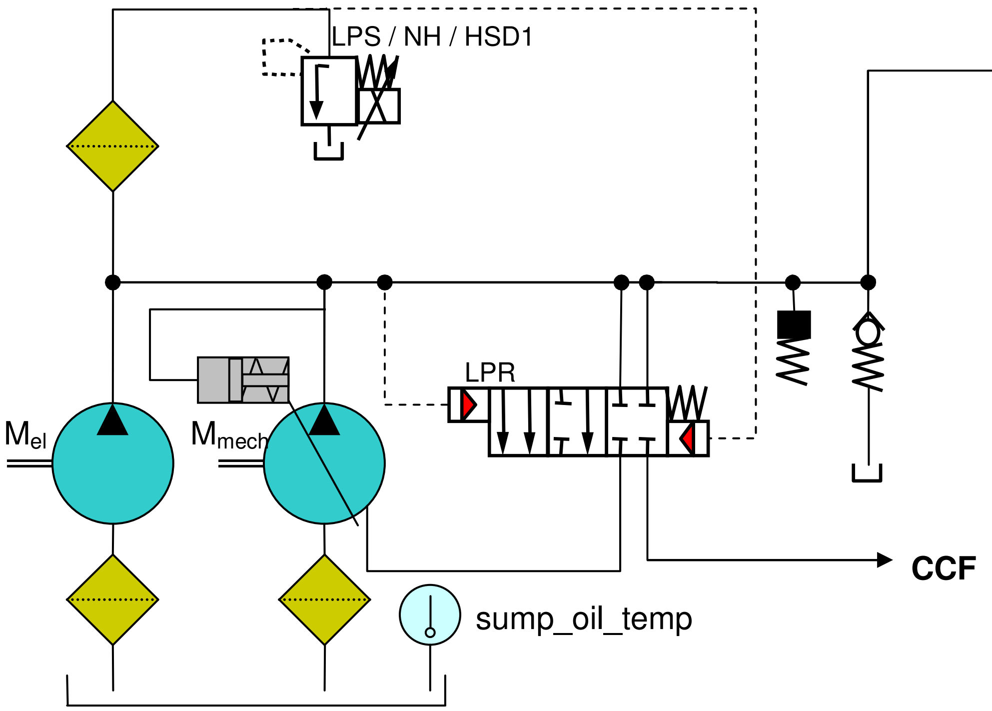

Figure 17: System over view

In order to design and approximate the efficiency of the concepts a mathematical model was developed. Parameters controlling the most suitable dimensions for optimum efficiency were calculated. In this design proposal there exist two pumps in parallel and hence two different efficiency studies were performed. In order to simplify the calculations the same mathematical model was used for both the pumps, only the dimensions and initial conditions were changed. This gave an approximately value sufficient enough to measure the difference of the concepts. In the model both the mechanical efficiency (η_{mech}) and the volumetric efficiency (η_{vol}) was studied, see appendix II

Since the eccentricity of the displacement chamber is maneuvered by an actuator operating under system pressure, the operation may not respond to sudden changes in flow demand. This can cause production of excess flow during short intervals but also cause lack of flow.

During situations when flow demand is larger than flow produced by the vane pump the electro-hydraulic pump will support.

4.5 Concept 2

Concept 2 consists of two single stage spur gear pumps connected in parallel in the hydraulic system. One pump is mechanically driven by the input shaft in the transmission and the other one is an electro-hydraulic pump with the same design as the one described in concept 1. Flow produced is dependent on the speed of the input shaft and the speed of the electro-hydraulic pump and the displaced volume. The two pumps are identical in design, only the power supply is different. Figure 18 gives a basic over view of the system.

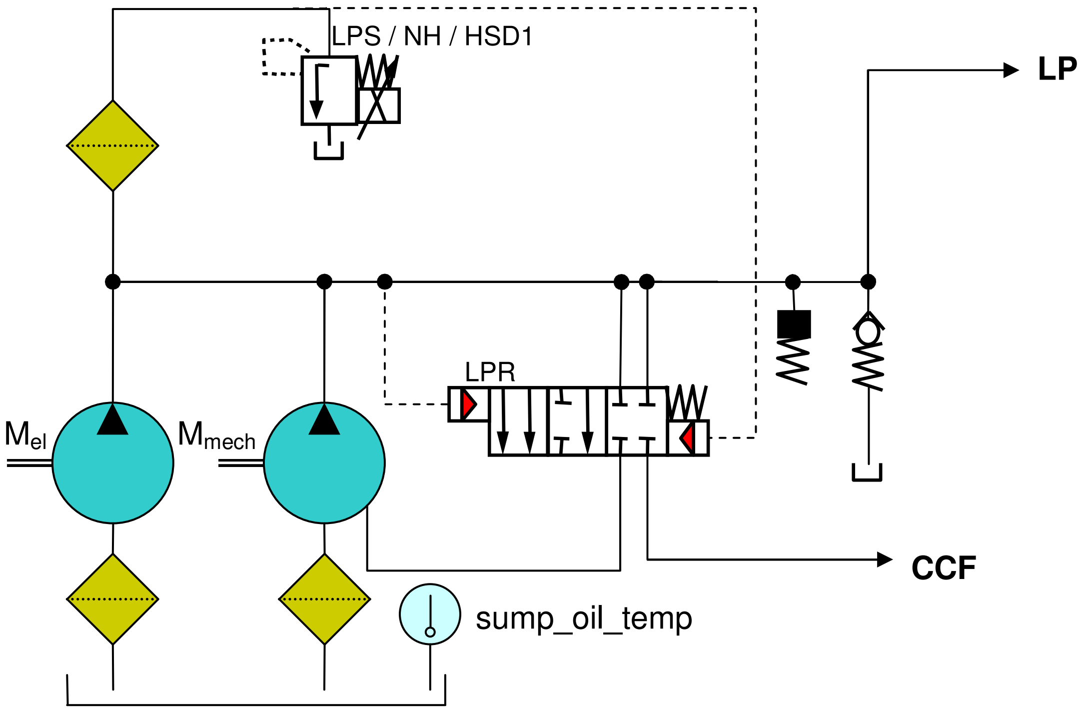

Figure 18: System over view

As for the previous concept, the optimum design parameters were calculated and the same mathematical models and formulas as previous were used. The optimum parameters and losses are presented in appendix II.

4.6 Concept 3

Concept 3 consists of a single stage spur gear hydraulic pump driven by an external electric motor. The design of the pump does not allow variable displacement but the produced flow can be controlled by varying the speed of the electric motor. Since there is no mechanical linkage between the rotating parts in the transmission and the pump unit, the package can be placed at a suitable location internal or external on the transmission casing. However, in order to prevent cavitation and large frictional losses in the system, the pump must be placed in close connection with the sump. Due to high temperatures and vibration levels, a robust design with effective sealing technology is required. A major part of the system cost is determined by the electric motor, therefore an efficient pump requiring a smaller motor makes sense. The concept must be tailor made so suit the limited packaging space. Figure 19 gives a basic over view of the system.

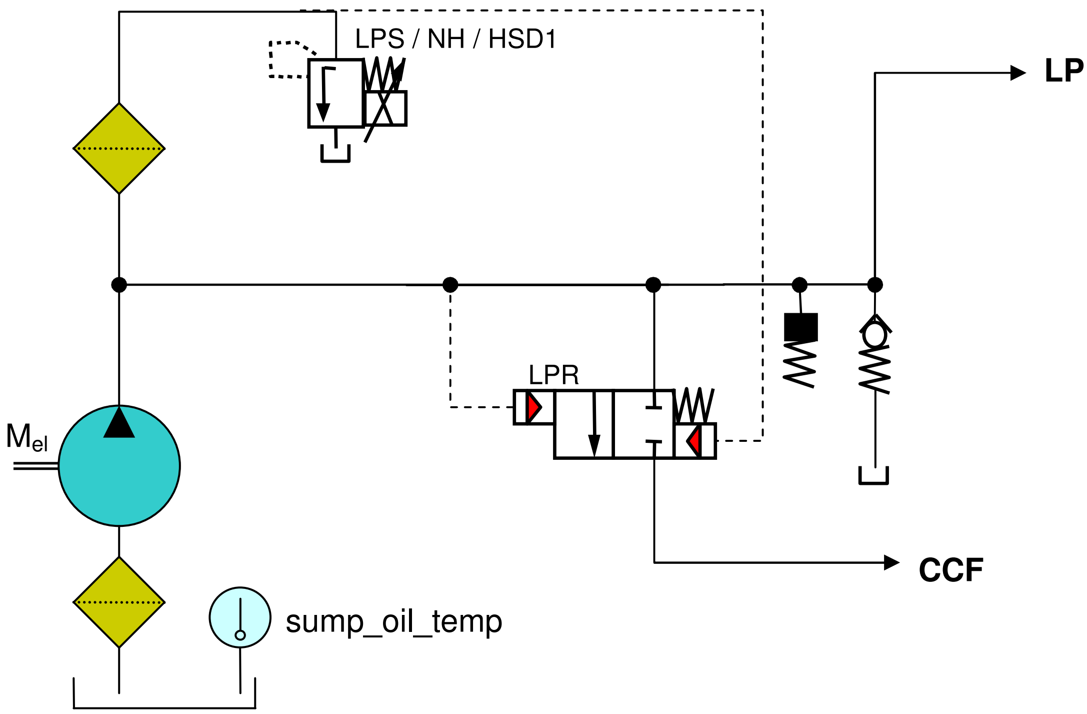

Figure 19: System over view

The development process will follow the same steps as presented earlier. This design will put higher requirements on the electric motor due to the high operation speeds during high flow demands from the system. It also requires a sophisticated control system for controlling the speed since the flow delivered is proportional to the speed of the pump. An ideal control system should control the pump unit to deliver enough flow and pressure for optimum performance of the hydraulic system. No excess flow will be produced during ideal operation, hence the power loss due to excess flow will be zero. The over all efficiency of the system will therefore be determined by parameters including mechanical friction, volumetric losses and electrical efficiency. The initial efficiency calculations done for this concept is shown in appendix II. No considerations are taken for the efficiency of the electric machine. Since there are no need for a mechanical link between pump and transmission, the gears supplying mechanical power to the existing pump in the transmission can therefore be removed. This currently occupied space can be used to accommodate new features related to the hybridization of the transmission. A tailor made design concept will keep the packing space a minimum and allow for further improvements of the efficiency.

5. Evaluation of concepts

This chapter presents the evaluation process of the three pump concepts according to methods by Pahl & Beitz [5]. The process is described in detail in appendix III.

5.1 Identify evaluation criteria for the hydraulic pump

In order to objectively evaluate the different pump concepts a number of criterions have been developed. This step is based, first of all, on the requirements list. During concept development, variants that were found to be unsuitable in principle were eliminated. The requirement list consists of criterions that must be fulfilled by the design and since these are static they cannot be evaluated. The evaluation criterions consists therefore of parameters that improves performance, size and price. During the conceptual phase it has been difficult to put actual figures to the cost. Nevertheless, both technical and economic aspects have been considered and evaluated. The criterions are illustrated in figure 20.

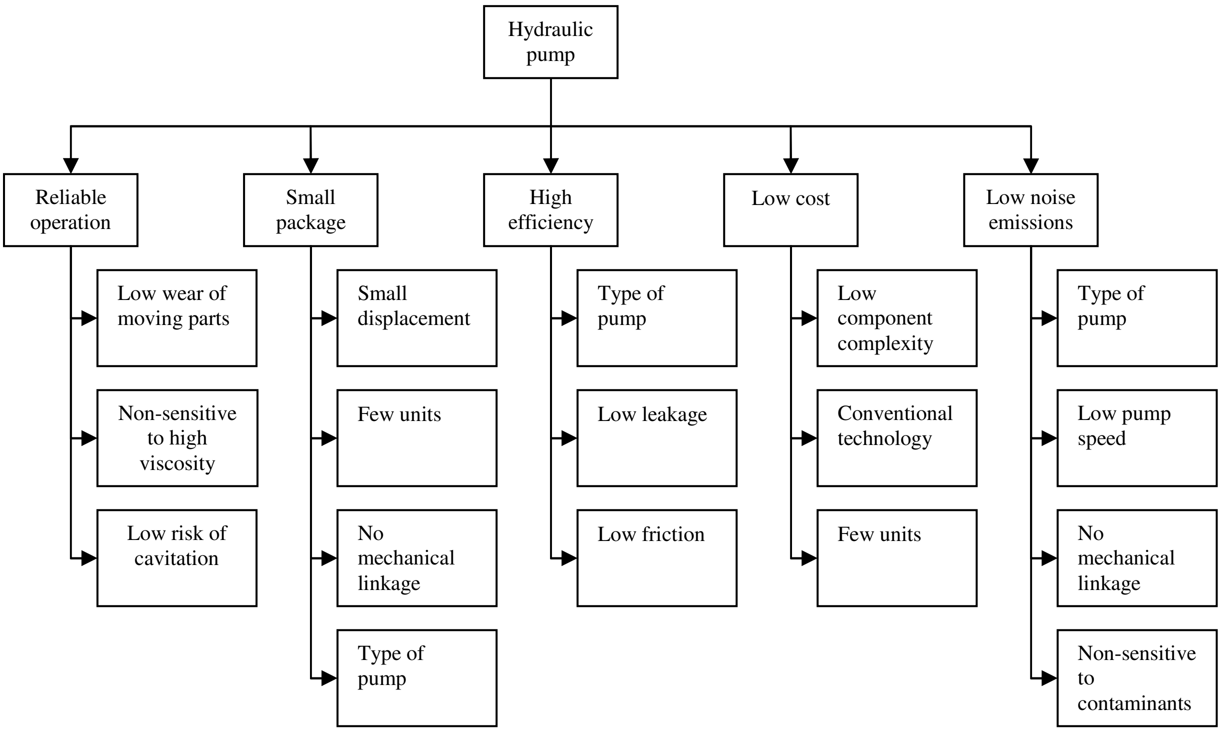

Figure 20: Criteria of the hydraulic pump

5.2 Weights of the criteria

In order to find out how important the different criteria are they have been weighted with a three-level grade. This type of weighting gives an objective estimation of how important each criterion is. The result from the weighting is illustrated in figure 21.

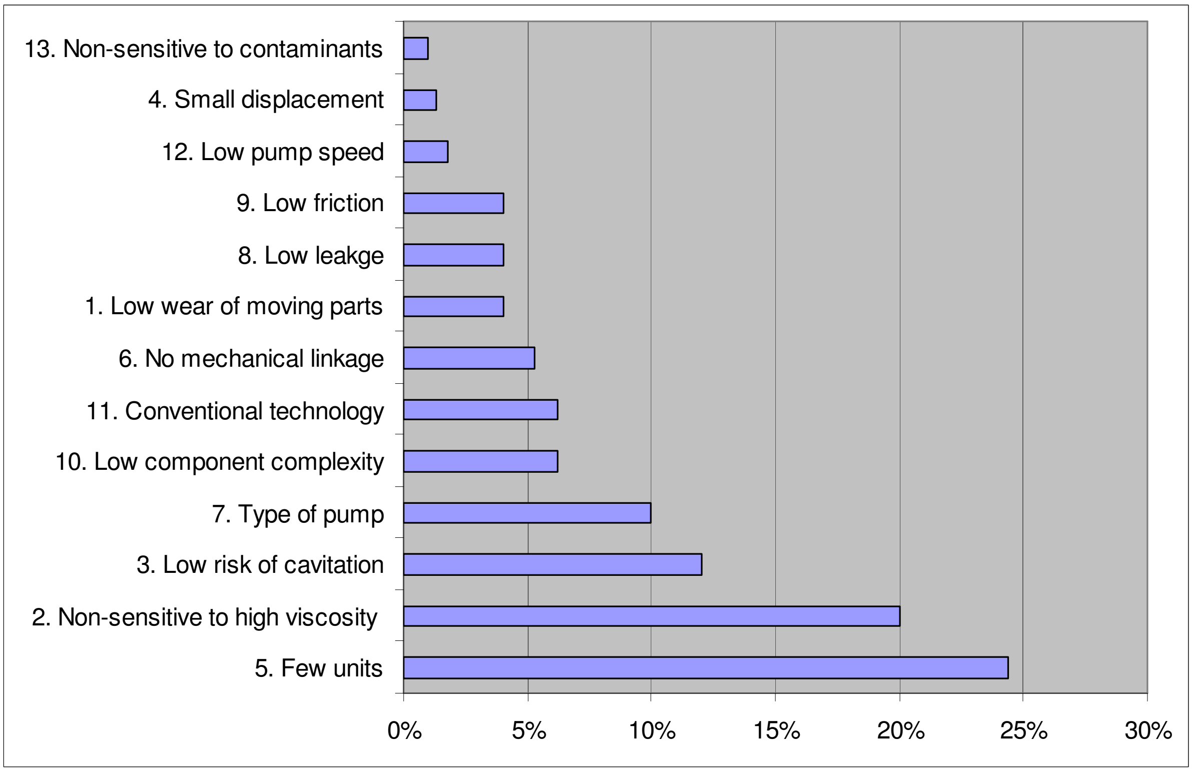

Figure 21: Chart showing the result gained from the grading of the hydraulic pump design

5.3 Evaluation of the proposed concept designs

The three different concepts have been weighted with the current pump as a reference for each criterion. This gave an overall value for each of the concepts where the concept with the highest score is the best suited for further development. The grades are:

-1 = Does not fulfill the criterion as good as the reference.
0 = Fulfill the criterion as good as the reference.
1 = Fulfill the criterion better than the reference.

The weightings and the criteria are illustrated in table 6.

Criteria	Grade				Weighted grade			
	Current pump	Concept 1	Concept 2	Concept 3	Weight	Concept 1	Concept 2	Concept 3
1. Low wear of moving parts	0	1	1	1	0,040	0,04	0,04	0,04
2. Non-sensitive to high viscosity	0	1	1	0	0,200	0,2	0,2	0
3. Low risk of cavitation	0	1	1	1	0,120	0,12	0,12	0,12
4. Small displacement	0	1	1	1	0,013	0,013	0,013	0,013
5. Few units	0	-1	-1	0	0,244	-0,244	-0,244	0
6. No mechanical linkage	0	0	0	1	0,053	0	0	0,053
7. Type of pump	0	1	0	0	0,100	0,1	0	0
8. Low leakage	0	1	1	1	0,040	0,04	0,04	0,04
9. Low friction	0	1	1	1	0,040	0,04	0,04	0,04
10. Low component complexity	0	-1	0	0	0,062	-0,062	0	0
11. Conventional technology	0	-1	-1	-1	0,062	-0,062	-0,062	-0,062
12. Low pump speed	0	0	0	-1	0,018	0	0	-0,018
13. Non-sensitive to contaminants	0	0	0	0	0,010	0	0	0
Sum					**1,00**	**0,185**	**0,147**	**0,226**

Table 6: Criteria and overall score for the three different design concepts

From table 6 one can conclude that concept 3 is the design that best fulfill the criteria. The main reason for this is that concept 3 only consists of one pumping unit. The only two drawbacks for this concept are its pump speed and the relatively new technology of electro-hydraulics in the automotive industry.

The only difference between concept 1 and 2 is the design of the mechanically driven pump unit. This is also reflected in the overall score where the design proposal for concept 1 uses a more suitable type of mechanically driven pump. Apart from this the three concepts performs equally in almost all criteria although concept 3 is more advantageous because of the previously stated single unit design and lack of mechanical link. An explanation of each grade can be found in appendix III.

6. Results

This chapter gives comprehensive information about the details and performance of the electro hydraulic pump concept.

6.1 Detail design solutions

Extensive work has been made in order to define the optimum design parameters. This has mainly been carried out by mathematical models describing the moving parts in the pump and the losses arising during different operation modes. Parameters that have been studied are:

- Differential pressure
- Rotational speed
- Viscosity
- Clearance

The parameters have governed the design and the dimensions of the electro-hydraulic pump. An exploded view of the final design is shown if figure 22.

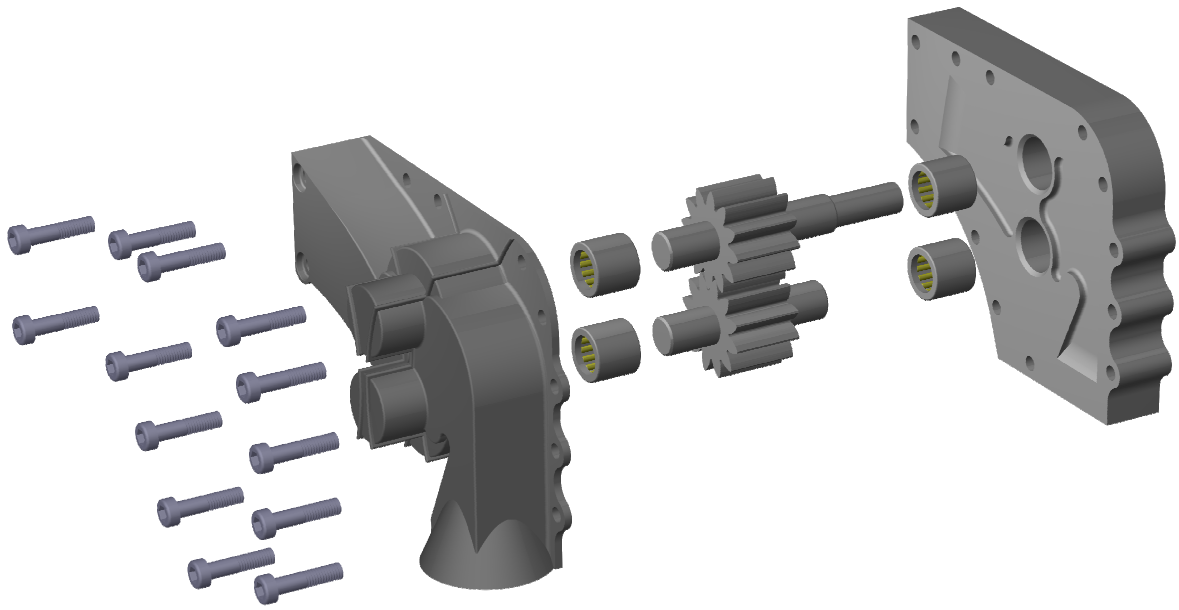

Figure 22: Exploded view of the pump assembly

6.1.1 Front plate

The front plate of the pump is mounted onto the electric motor. The drive shaft protrudes through the plate and into a connection with the motor drive shaft. For optimum sealing the drive shaft must be fitted with a shaft seal. There are two seating for the roller bearings supporting the dowel pin and drive shaft, one in the hole where the drive shaft protrudes and one in a circular pocket where the dowel pin is seated. The bearings are offset from the sealing surface in order to decrease friction and encourage lubrication. The sealing surface is a trade off between maximum sealing and minimum friction. The sealing surface is shown as the red section in figure 23.

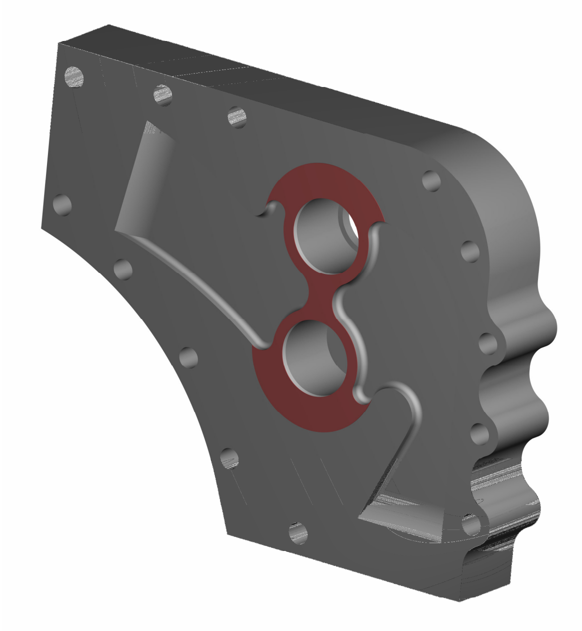

Figure 23: Front plate with sealed surface marked in red

There is always a sliding contact between the rotating gears and the surface marked in the figure. The oil interface between the front plate and the gears are assumed to be supported by full film lubrication. Care has been taken in order to make all corners and edges in the flow channel equipped with a smooth radius or edge blend. This in order for as smooth flow as possible. This implies also for the surface in the channel. Since there are no mechanical stresses of importance applied to the front plate, the component can be cast in a suitable material.

6.1.2 Gears

The gears in the pump are of high importance. Noise, vibration and short life are some of the penalties to be paid for gears imperfectly designed. The first of the two gears is connected to the drive shaft. This gear is termed the drive gear. The second gear is called the idler gear. It meshes with the drive gear and is driven by it. When the drive shaft rotates it in turn rotates the drive gear which then drives the idler gear. As the gears rotate, the gear teeth unmesh on the inlet side and mesh on the outlet side of the pump. An ideal meshing can only be achieved by a perfectly designed and manufactured gear unit. The profile of a gear tooth must be chosen with the following aspects in mind for ideal meshing:

- The gears must mate and mesh with a smooth uniform action.
- The tooth must have a section sufficiently strong for the applied loads.
- The tooth must be free from weakening undercuts.
- The tooth will mesh at the correct shaft centre distance.
- The profile of the teeth offers no manufacturing difficulties.
- The geometry provides an adequate tooth overlap.

The involute curve provides the most widely used profile for gear and teeth design. And since lack of knowledge for other teeth profiles, the involute profile was used. A comprehensive study of geometrical calculations, materials, strength calculations and service life approximations for the gears designed are presented in appendix IV. Figure 24 indicates some of the basic terms used in connection with gears and will be used in the rest of this work.

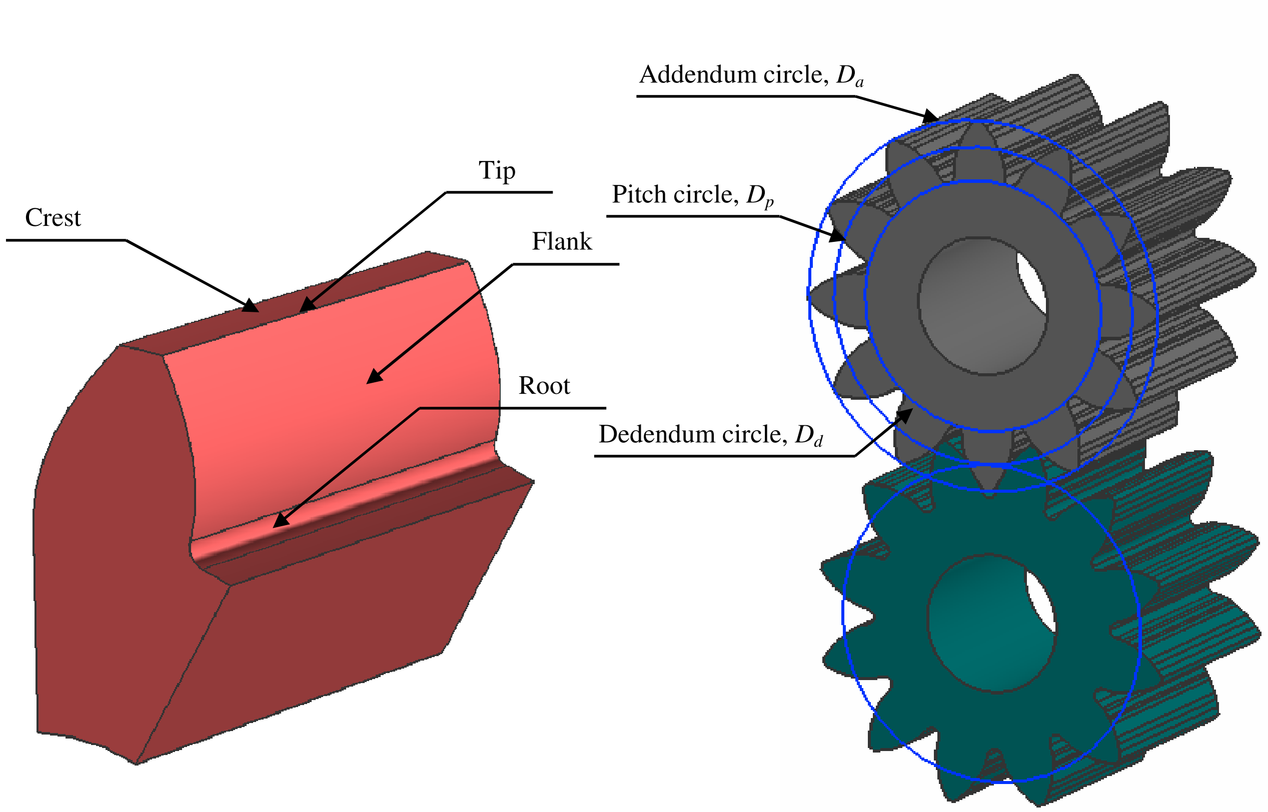

Figure 24: Nomenclature for gear tooth and meshing gears

For efficient running it is important that correct meshing of teeth is ensured, and this comes down primarily by establishing the correct centre distance for the shafts. In the design, there is tangency between the two pitch circles of the gears. This will hence be the centre distance between the shafts. The spur gear pump design displace a fixed volume of fluid each revolution. This volume is dependent on the free space between each tooth. The formula for calculating this volume is given by:

$$V = 2hD_p\pi b\,, \qquad (13)$$

where *2h* denotes the depth of the teeth, D_p the pitch diameter and *b* the gear width. The depth of the teeth is the distance from the tip of the teeth to the base circle.

Table 8 shows data from appendix IV defining the geometrical shape of the gear and the volume calculated using equation 13.

Number of Teeth, z	12
Pitch Diameter, D_p (mm)	23,0
Gear Width, b (mm)	18,0
Module, m	1,9
Addendum Circle, D_a (mm)	26,9
Base circle, D_b (mm)	19,2
Dedendum Circle D_d (mm)	18,2
Displacement, V (mm^3/prev)	5000
Displacement (cc/prev)	5,00

Table 8: Data regarding the dimensions of the gear

The complete gear set is shown in figure 25.

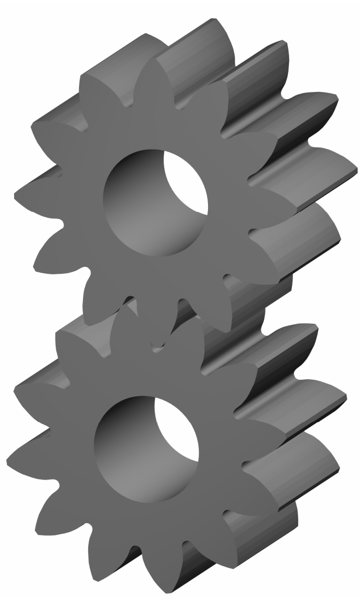

Figure 25: Gear set

6.1.3 Drive shaft and dowel pin

The drive shaft and the dowel pin are made of SS141450-1 carbon steel, typically used in shafts and other machine components. Since the purpose of the shaft is to transfer the torque supplied by the motor to the gears, the design requirements are strongly dependent on the torque. The gear is pressed onto the shaft in the assembly line and the interface between the gear and shaft must provide enough frictional force in order to transfer the torque but also allow for easy assembly. The frictional force is dependent on the frictional coefficient, the area of contact and the contact pressure and is defined by:

$$F = \mu p \pi d l \,, \qquad (14)$$

where μ is the frictional coefficient, p is the contact pressure, d diameter of shaft and l length of contact. The variance of the contact pressure is allowed to vary in an interval where the operability of the joint is guaranteed. The lowest pressure allowed is determent from the criteria where the shaft-gear interface is not slipping, that is:

$$p = \frac{1}{\mu \pi d l} \sqrt{\frac{4T^2}{d^2}}, \qquad (15)$$

where T is the transmitted torque.

The highest pressure allowed is determent from the criteria of structural strength. That is, the pressure is not allowed to exceed the yield strength for none of the parts. From the theory of structural integrity for solid shafts:

$$\sigma_{Y_{shaft}} = p \qquad (16)$$

and for hubs:

$$\sigma_{Y_{hub}} = p \frac{d^2}{D_d^{\,2} - d^2} \sqrt{1 + 3\frac{D_d^{\,4}}{d^4}}. \qquad (17)$$

The gear is considered a hub where the shaft hole is denoted d and the dedendum circle is denoted D_d.

The axial force required for pressing the gear onto the shaft is of course the same as the frictional force, see equation 14.

By assuming the frictional coefficient to 0.1, required and max allowable pressure and can be derived. The results are shown in table 7, using equations 15, 16 and 17.

Torque, T @-30℃ [Nm]	23,17
Shaft diameter, d [mm]	10
Dedendum circle, D [mm]	18,2
Contact length, l [mm]	18
Frictional coeff., μ	0,1
Min. allowable pressure, p [MPa]	81,9
Max. allowable shaft pressure, p_{shaft} [MPa]	>250
Max. allowable hub pressure, p_{hub} [MPa]	99,3

Table 7: Calculated data

As can be seen in the table, the contact pressure between the gear and the shaft are allowed to vary between 82 and 99.3MPa. Contact pressure above 99.3MPa will risk operability of the gear. For security against shaft-gear slipping of a factor 1.1, the minimum allowable pressure should be 90.1MPa. This requires 5.1kN of axial force when pressing the gear onto the shaft. The hole diameter in the gear should be approximately 12.5µm smaller than the shaft in order for correct pressure and grip. The drive shaft and the dowel pin are shown in figure 26 below.

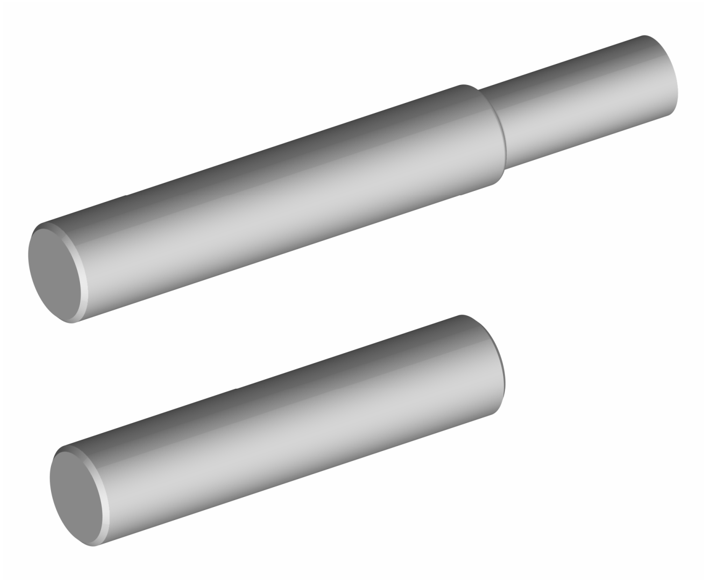

Figure 26: Shaft and dowel pin

The shaft and the dowel pin will also comprise the inner raceways for the needle bearings. This requires the shaft and dowel pin to have the same hardness and surface finish normally found on the bearing raceways. The following recommendations are found in the SKF catalogue *4703/IE, Needle roller bearings.* Raceways machined in associated components for drawn cup needle roller bearings must have a Rockwell hardness of between 58 and 64 if the load carrying capacity of the bearing is to be fully exploited. The raceway surface roughness should have a R_a value equal or greater than 0.2µm. The out of round should not exceed 25% and the deviation from cylindrical form should not exceed 50% of the actual diameter tolerance.

6.1.4 Bearings

The shaft and the dowel pin are supported by four needle rolling bearings in the pump housing and in the front plate. Needle roller bearings are roller bearings with cylindrical rollers which are thin and long in relation to their diameter. The radial space between front plate and drive shaft is small in order to improve efficiency of the hydraulic sealing. Needle rolling bearing is extremely suitable for arrangements where radial space is limited. Because of the practical and economical aspects of the production of housing and front plate, the bores accommodating the bearings are made simple. This requires the needle rolling bearings to support its own raceway. The drawn cup needle rolling bearing supplied from SKF have a thin-walled outer ring for this purpose. The bearings are pressed into the bore in the housing and front plate and will then be used directly on the shaft and dowel pin. The dimensioning of the bearings is governed by the shaft diameter and the leakage length between the dedendum circle of the gears and the accommodating bores. Also, since needle bearings only take radial loads, F_r will set limitations. The radial forces taken up by the bearings are a result from compressed oil trapped in the space between the meshing teeth, defined as F_{press}. The magnitude of this force is dependent on the pressure of the oil trapped and the area where the pressure is acting. Radial forces (F_{rad}) will also be present under the sequence of meshing. The total load taken by the bearings is therefore the sum of pressure governed load and the load exerted during meshing, see equation 18

$$F_r = F_{press} + F_{rad} \qquad (18)$$

The radial load for normal operation is approximately 410N, but extreme values during cold operation exceed 900N. If it is assumed that cold operation is limited this will not damage the bearing, and hence the bearing can be dimensioned for normal operation. The pressure governed force for 20bar is calculated to 80N, equation 18 then gives the force taken up by the bearing to 490N.

The basic load rating C is used for calculating the service life of the bearing. C indicates bearing load for a nominal bearing service life of 1 million revolutions. From the well known service life equation from SKF:

$$L_{10} = a_{SKF}\left(\frac{C}{P}\right)^p, \qquad (19)$$

where L_{10} is the nominal service life in millions of revolutions achieved or exceeded by 90% of a large batch of identical bearings, a_{SKF} a tabulated constant from SKF for different bearing types, C is the basic load rating, P is the equivalent dynamic bearing load and p is a constant depending on the bearing type. Since the type of bearing is a rolling bearing, p is set to 10/3.

The service life for the bearings expressed in millions of revolutions is calculated from the mean value of revolutions for the driving cycles defined in section 4.3. The mean value is just below 3500 rev/min and for a service life of 5000 hours, this is equal to 1050 million revolutions. Hence L_{10} must be equal or greater than 1050 in order for 90% certainty of fail safe operation.

The bearing life for the HK1012 bearing from SKF has been calculated with assist from the SKF online calculator [12]. The calculations are made assuming normal operating conditions where the temperature is 75°C and cleanliness factor of 0.5. This is because the leakage of hydraulic oil will lubricate and remove wearing particles. According to appendix V, the resulting nominal service life for the bearing is 2960 million revolutions. This exceeds the required service life by a wide margin. The HK1012 bearing is shown in figure 27.

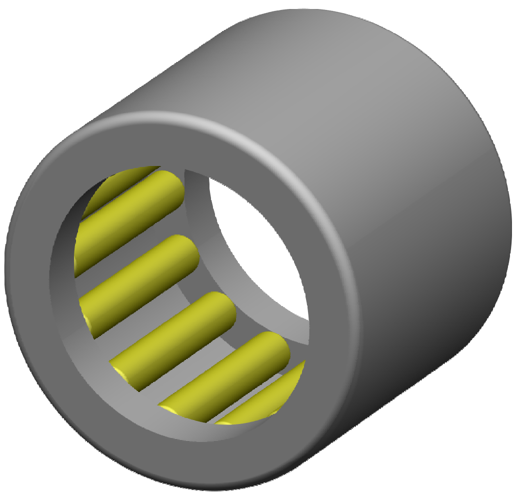

Figure 27: The HK1012 drawn cup needle roller bearing

6.1.5 Housing

The housing of the pump is the feature where all the rotating parts are accommodated. It is mounted onto the front plate making the house a sealed compartment. There are two bores where the bearings supporting the drive shaft and the dowel pin are located. The bearings are, like in the front plate, offset from the sealing surface. The sealing surface is designed, as previously mentioned in the front plate section, as a trade off between maximum sealing and minimum friction. Apart from the similarity with the front plate, the housing also seals the section where the crests of the teeth are in contact with the housing wall, this section is called the sealing wall. The oil interface between the housing and the gears are assumed to be supported by full film lubrication. On the discharge side of the sealing surface, two grooves are cut perpendicular to the surface, see figure 28. These grooves are made in order to decrease the pressure pulsations occurring when a filled tooth cavity discharges on the high pressure side of the pump. The pulsations can influence the operability of the system in a negative way and also cause noise emissions. Due to the pressure relief grooves, a tooth cavity can be pre-discharged moment before it opens up for the discharge and hence decrease the pulsations.

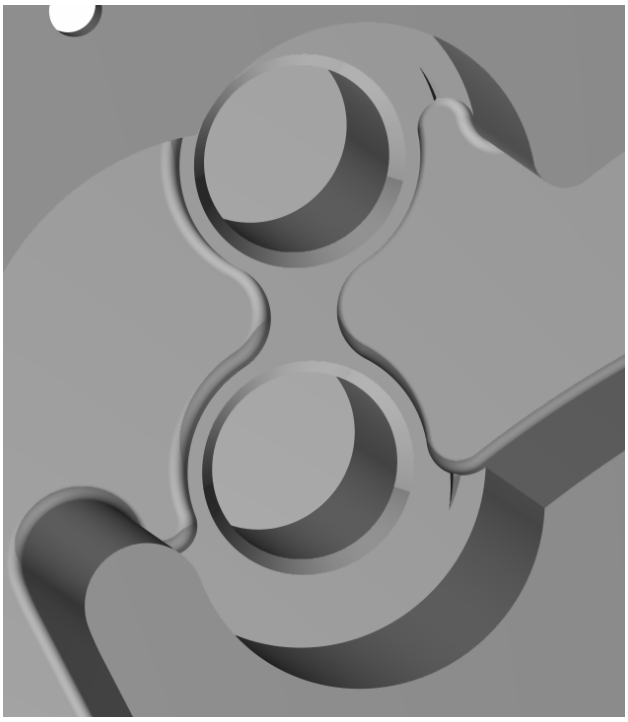

Figure 28: Sealing surface, sealing wall and pressure relief grooves

As can be seen in the figure the inlet port and discharge port are designed as wide as possible, this for keeping the velocity of the fluid at a moderate level even at high flow demands. All corners and edges in the flow channel are also designed with a radius or edge blend in order for the flow to be as smooth as possible. The surface in the flow cannels should also be made with care for the same reason. The housing is shown in figure 29.

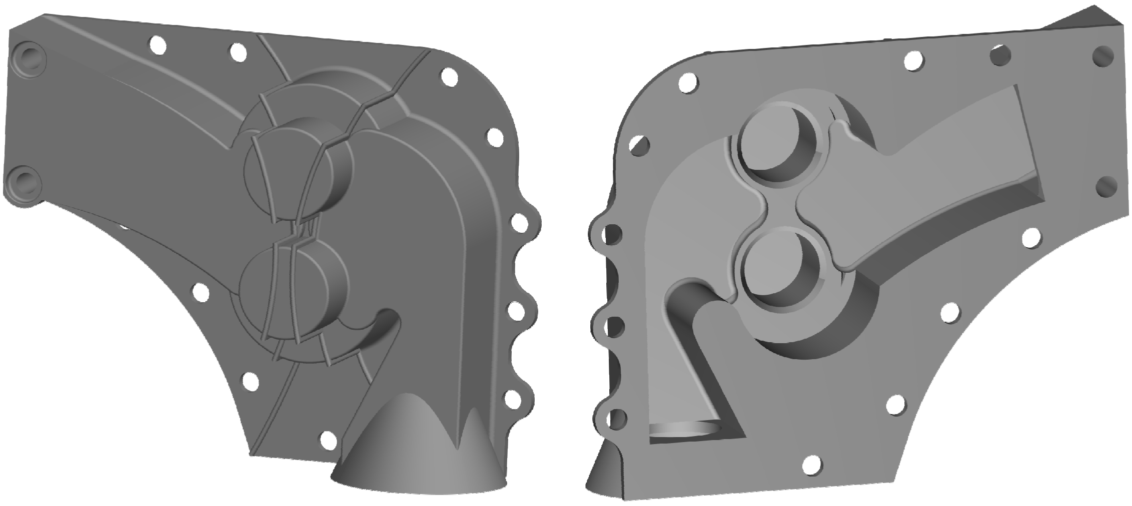

Figure 29: Front view and inside view of the housing

The fluid enters the housing through a filter connected on the port on the underside of the part. Fluid is then transported into the pre-pressure chamber where it fills the cavities in the gears. The fluid is pressurized and discharged in the discharge port. The pressurized fluid exits the housing through a port on the side. The housing can by advantage be made in a material suitable for casting. Figure 30 show the complete pump assembly with the housing connected with a filter. The total volume taken up by the pump assembly, excluding the filter is 0.21 liters.

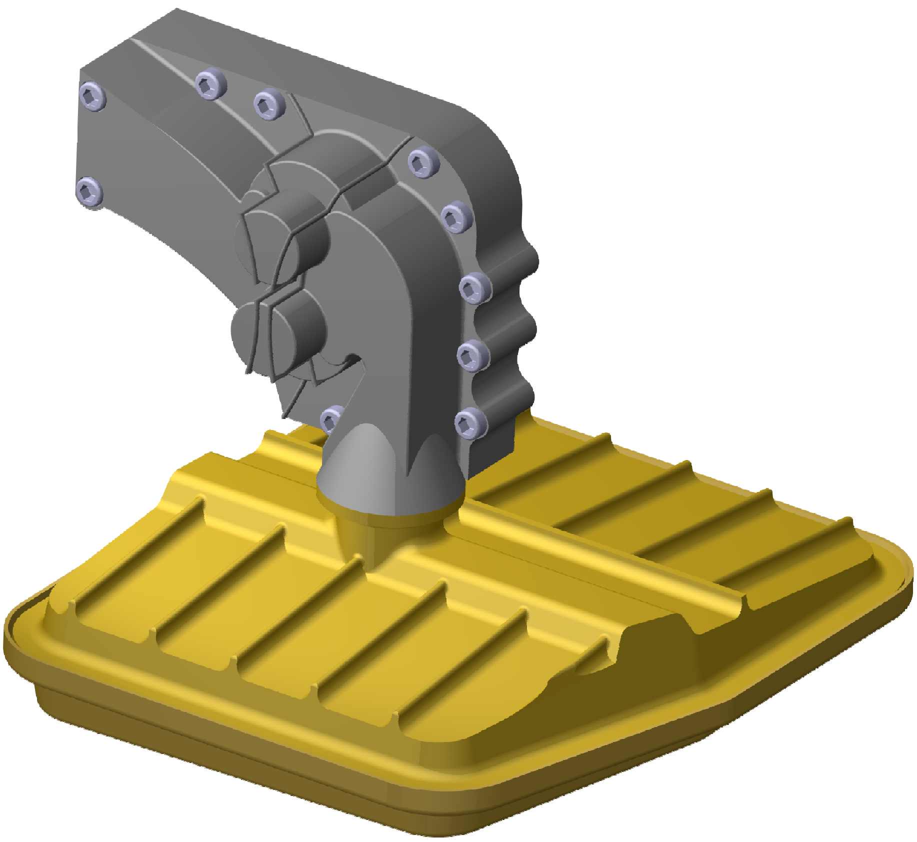

Figure 30: Pump assembly with filter connected to intake port

6.2 Efficiency

In order to estimate the power losses produced by the pump during different operating conditions a Matlab model was created. The losses analyzed in the model are due to imperfect sealing of the rotating elements in the pump and mechanical friction. The analysis is focusing on finding the total efficiency at different operating parameters. Since most design parameters are already defined within the design of the pump, viscosity, speed and differential pressure can be varied. The clearance plays a fundamental role in the overall efficiency. Since it is contradicting, small clearance results in less flow losses but increase the frictional losses and large clearance results in low frictional losses but increase the flow losses, an optimum has to be established. But, the optimum can only be established for specific operation modes and hence the most favorable clearance will vary during operation. However experimental investigations of the clearance of spur gear pump designs have shown that values between 5µm and 10µm can be taken as optimal for the full operating parameter range. And since the higher value is more suitable for high operational speeds the higher value is taken as optimal.

The analysis is divided into two parts, one describing the flow through the clearances and one describing the friction produced due to rotational movement of the pumping element. The first part is the origin of volumetric power losses (P_{vol}) and the latter one give rise to mechanical power losses (P_{mech}).

The flow between the clearances is described by two different types, the Couette flow and the Poiseuille flow. The Poiseuille flow is driven by the differential pressure existing between the outlet and inlet ports of the pump. The Couette flow is caused by the difference in velocity between the housing and the rotating gears, see equation 4 and 5. The velocity of the moving surface will increase along the radius of the gears, see equation 9.

The dynamic viscosity, μ, is of high interest since it has large influence on both the leakage and the friction between the rotating and stationary parts, see equation 5 and 8. The viscosity is a strong function of temperature. Figure 31 shows the characteristics of the viscosity for the transmission oil BOT341 currently used.

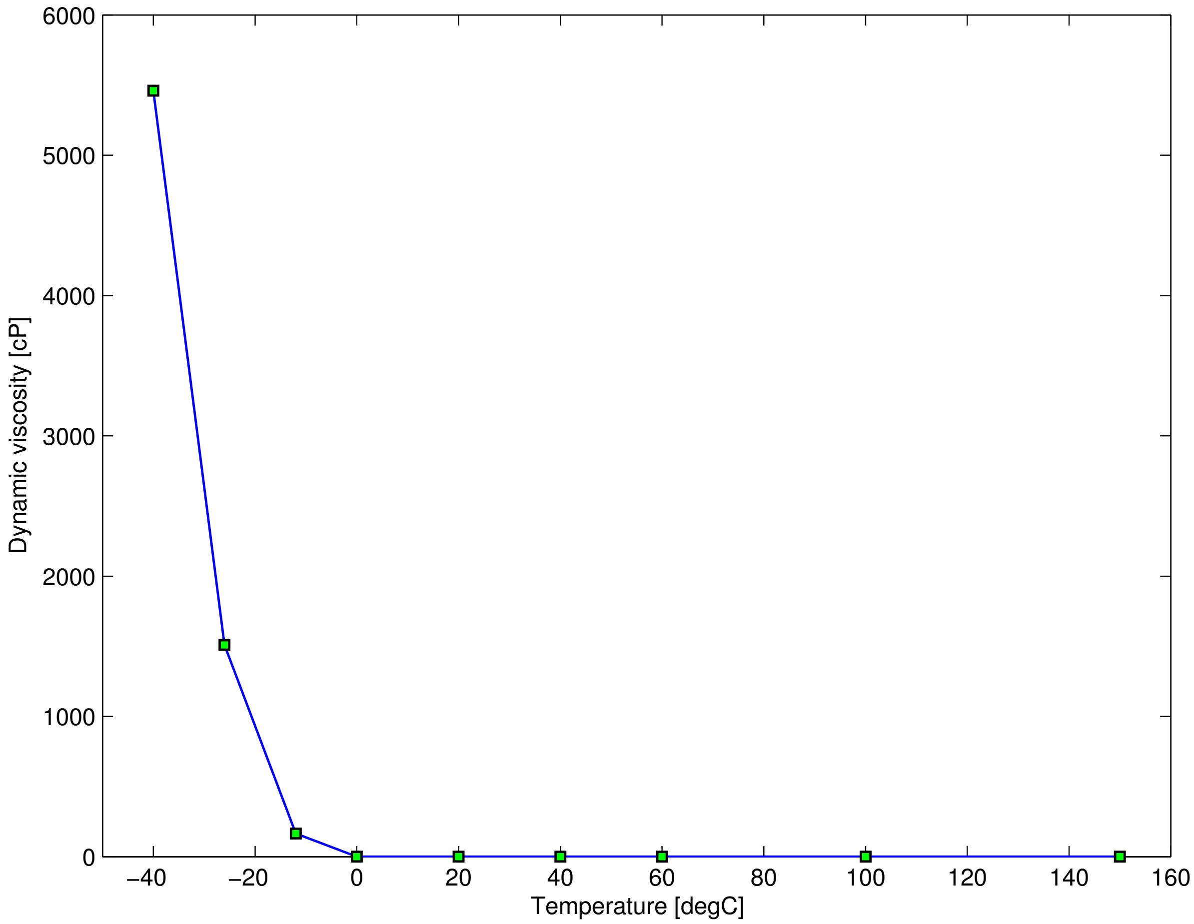

Figure 31: Chart showing the viscosity as a function of temperature

The pressure gradient, *dp/dx*, is a function of the differential pressure and the length of the contact between the housing and gear. The contact length for the planar face of the gear will decrease along its radius. The contact length along the sector where the teeth are in contact with the housing wall forms the sealing on the circumference of the gear.

Since the Couette flow is velocity dependent, it will change direction according to the velocity. On the meshing side of the gears the Couette flow will counteract with the Poiseuille flow, but on the housing side the flow will interact with the Poiseuille flow. This will result in part of the fluid being transported from the low pressure side into the high pressure side through the clearance. This will reduce leakage and will also increase lubrication and prevent serve wear of the sliding surfaces.

The flow rate through the clearances, 2 planar surfaces and 2 sectors, are shown in figure 32 for a differential pressure of 20bar and a rotating speed of 7100rpm. This operating condition is the worse case scenario and the largest losses are expected to occur during these extreme parameters.

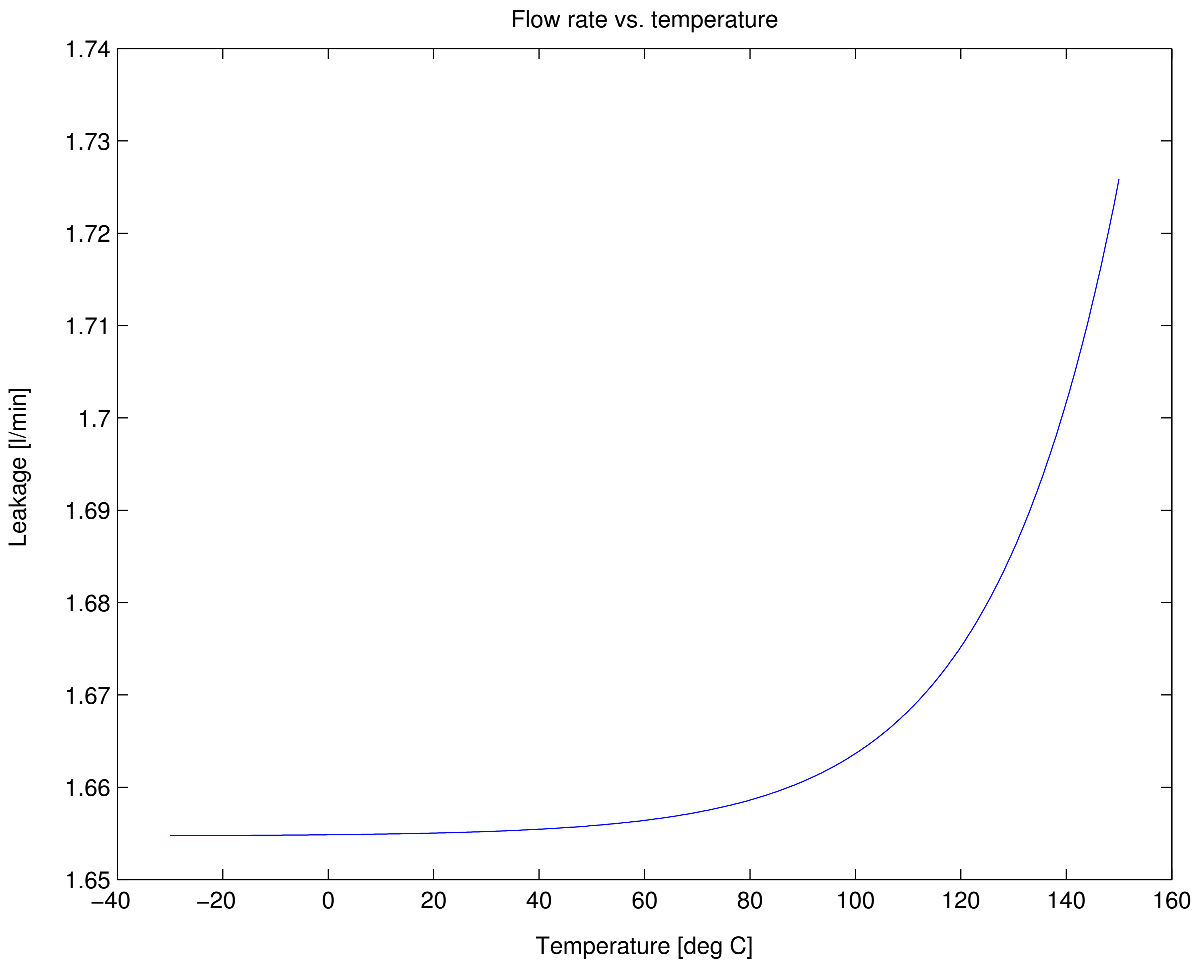

Figure 32: Total flow rate between the clearances at 10µm clearance, 20bar and 7100rpm

The figure shows that there exists a constant leakage although the viscosity is high, i.e. low temperature. This is caused by the fluid trapped in the space between the teeth when meshing. At elevated temperatures, the flow rate is increasing. This is due to the decrease in viscosity for the oil. As equation 3 implies, the leakage of oil will result in a decrease in volumetric efficiency, see figure 33.

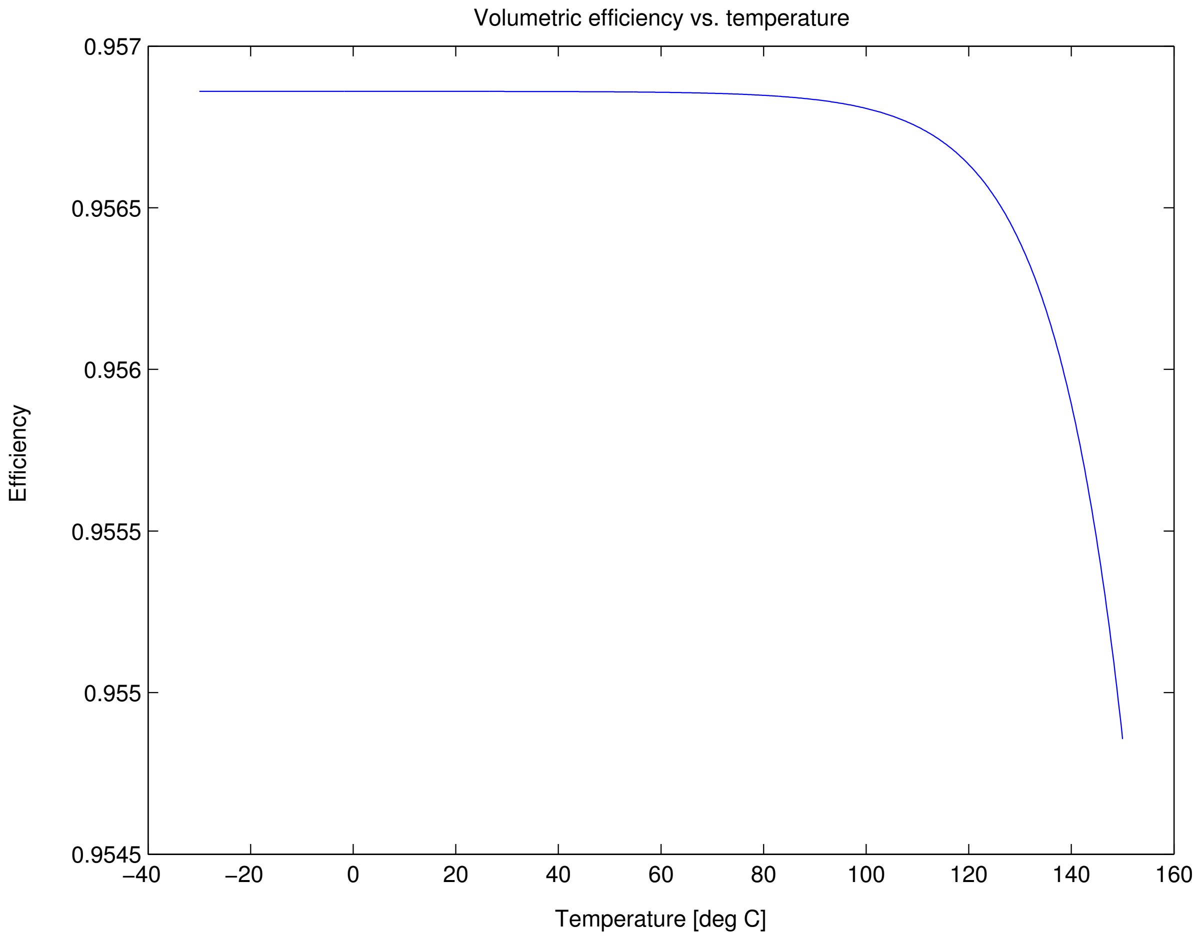

Figure 33: Volumetric efficiency at 10μm clearance, 20bar and 7100rpm

Frictional forces arise in the interface between the sliding surface and the oil film. As previously stated, full film lubrication is present in all operating conditions and hence friction is caused only by viscous shear of the oil. For a Newtonian fluid, as the BOT341 used, the magnitude of the shear stress, τ, along the surface is related to the velocity gradient, see equation 7. The force that must be applied on the shaft in order to receive the angular velocity is given by substituting equation 9 into equation 8. In order to solve this equation for different values of μ, the area of contact must be divided into small increments where the velocity and force can be assumed to be constant. For the contact along the sector where the teeth are in contact with the housing, velocity is constant and the area is assumed to be the sum of the area of the tooth tips in contact.

The force required to shear the oil film in the clearances will ultimately be supplied by a torque on the input shaft. This implies that there must be a loss of torque in order to rotate the gears, see equation 10. This torque is shown in figure 34 for 2 planar surfaces and 2 sectors for the worse case scenario defined earlier.

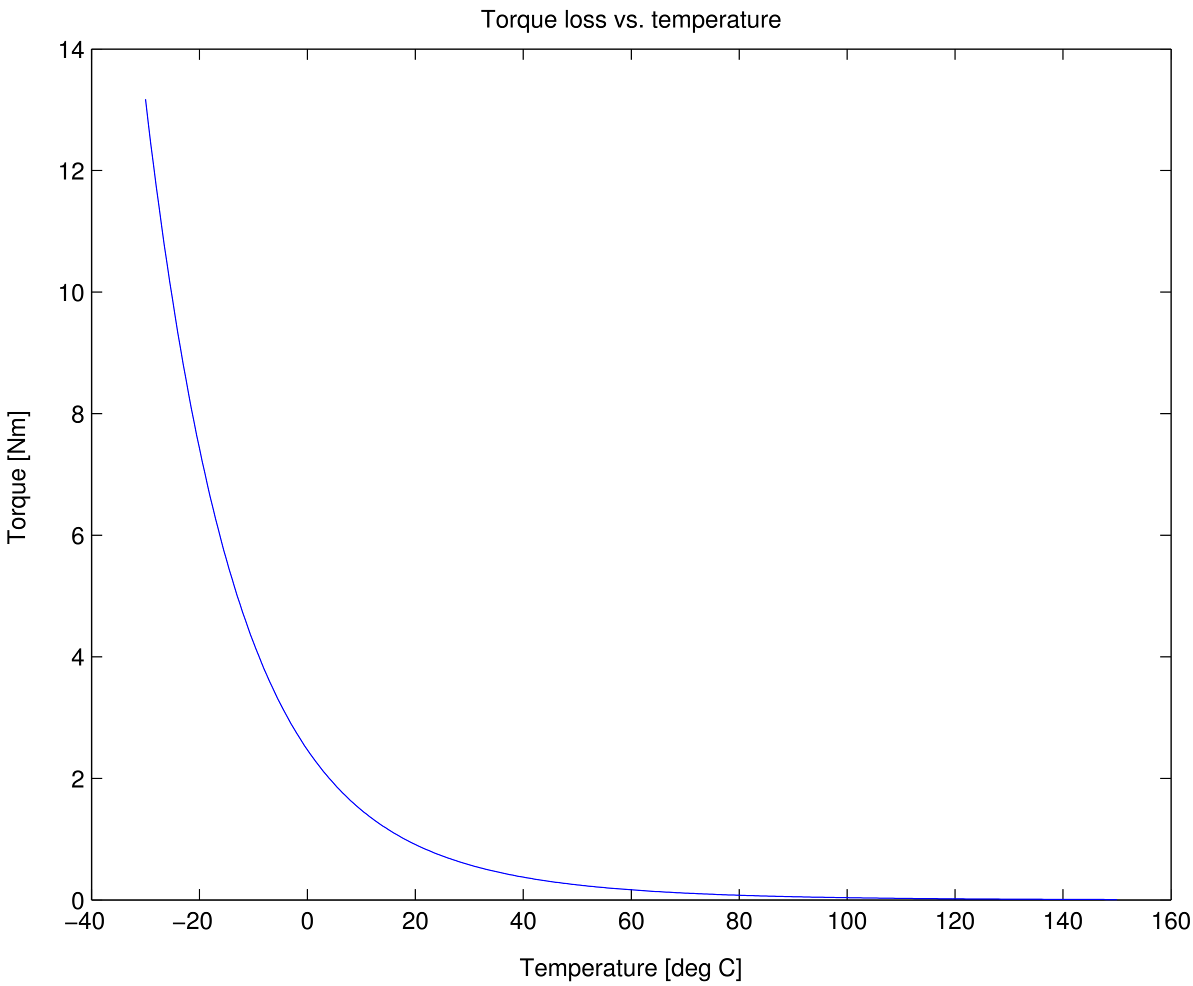

Figure 34: Torque loss cause by viscous shear at 10µm clearance, 20bar and 7100rpm

The figure implies that during low temperatures the force caused by viscous shear increases. During high temperatures when the viscosity is low, the shearing force is low and hence the required torque is low. Equation 6 says that the loss of torque will lead to a decrease in mechanical efficiency, see figure 35.

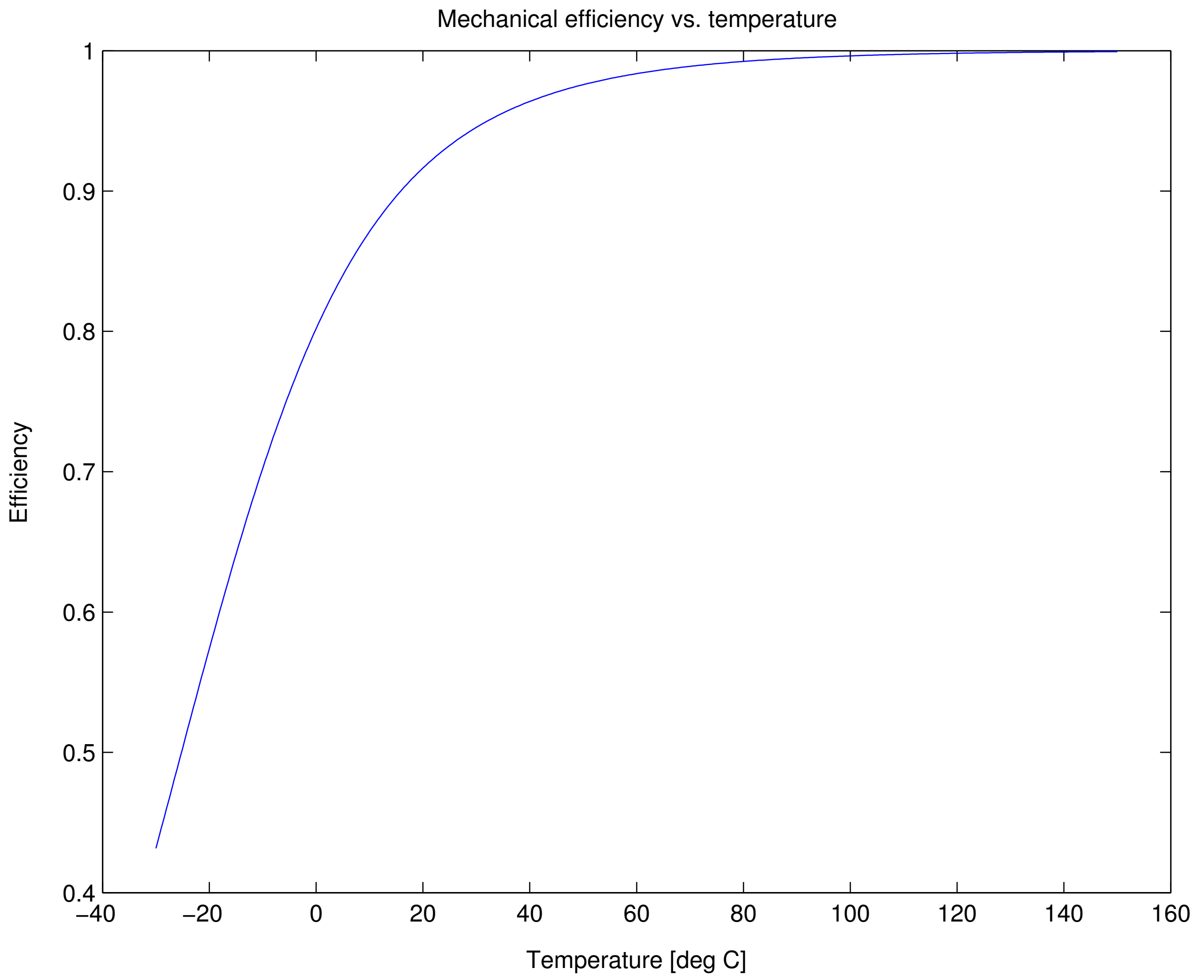

Figure 35: Mechanical efficiency at 10µm clearance, 20bar and 7100rpm

The losses arising during operation are incorporated with the volumetric and mechanical efficiency. From equation 2, the total efficiency is the product of these two efficiencies. Figure 36 shows the total efficiency for operation at 20bar and 7100rpm for different viscosities.

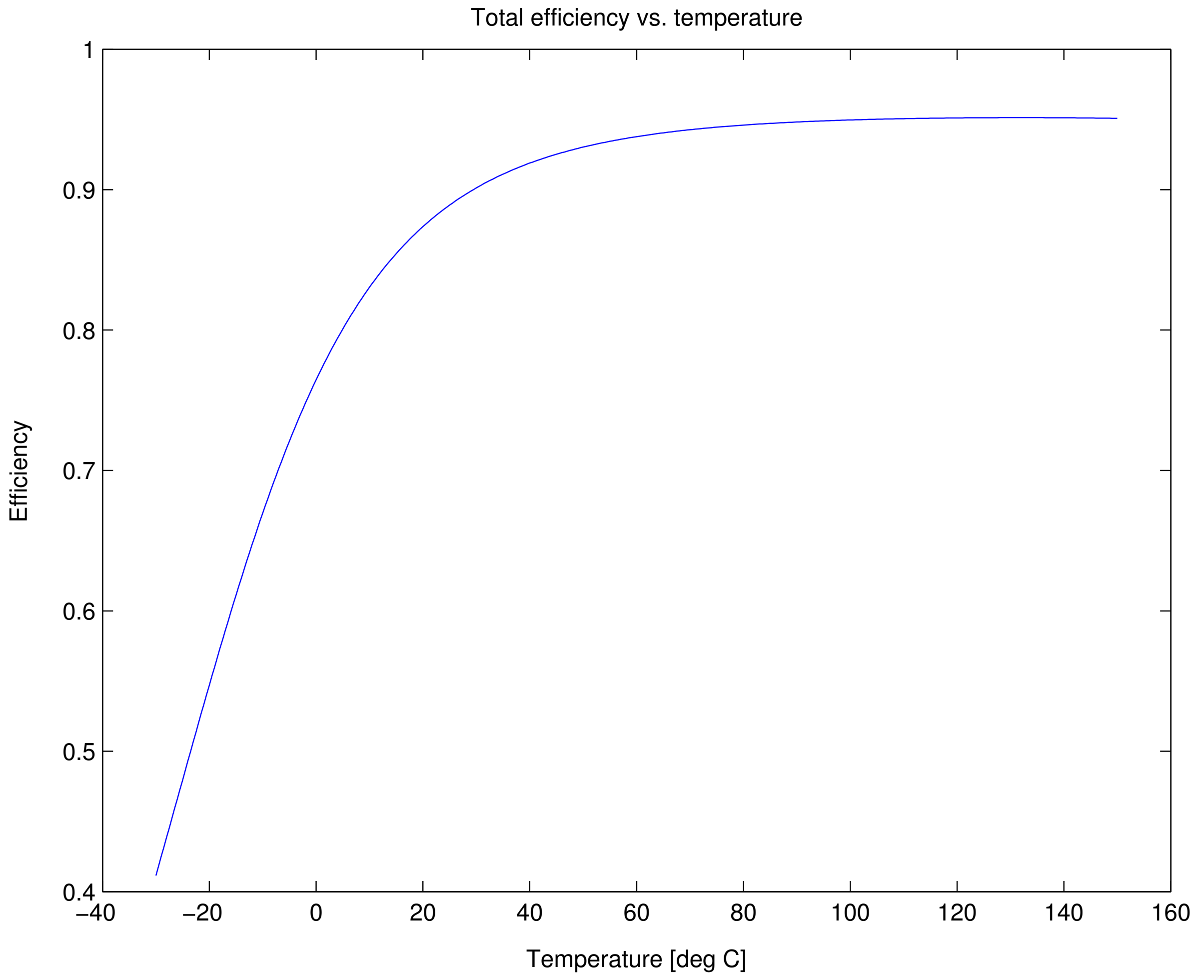

Figure 36: Total efficiency at 10µm clearance, 20bar and 7100rpm

6.3 Electric motor

The electric motor converts the electrical energy, stored in batteries or directly supplied from the generator, into mechanical energy. The mechanical energy is used to drive the gears in the pump in order for hydraulic energy conversion. The service life for the pump is expected to exceed 5000 hours without any maintenance of the ingoing parts. This anticipation will put high requirements on the electric motor. In the conventional DC motor, a mechanical contact forming the circuit between the electrical source and the set of electrical sections on the rotor (termed the commutator) are made up by brushes. As the rotor rotates on axis, the stationary brushes slides on the commutator making the material in the brushes, typically carbon, wear due to mechanical friction and electrical erosion. The brushes are an essential part in the motor and once worn down they must be replaced by a new set in order for the motor to operate. Typical service life for brushes in variable speed applications are according to [11] 2000 hours. A more suitable energy source for the pump is the brushless DC motor or the EC (electronically-commutated) motor. The motor offers maintenance free operation for a long service life, >5000 hours, and high efficiency, typically 90% depending on the power electronics used. This technology has replaced the mechanical contact between the brushes and commutator with electronically controlled commutation. The motor is equipped with a rotor position sensor, and is connected to the power source through its control and power electronics. The electric motor, control system and power electronics provides the capability to operate the pump at the required flow demand by the hydraulic system. This requires an intelligent control system which determines the rotation variation of the motor. After receiving signals from the TCU, the control system adopts a pre-established strategy which is based on the current hydraulic state and oil temperature and sends a signal to the motor for activation. In this way the motor rotation can be varied freely, from zero rotation to maximum rotation. For maximum flow demand by the system (Q_{req}=*34l/min*) the maximum speed required for the motor to produce enough output flow (Q_{out}) due to leakage is 7100 rpm. The size of the motor is strongly dependent on torque requirements. Since torque loss due to viscous shear of the oil is dependent on temperature, see figure 34, the cold condition performance will drive the physical size of the motor. In figure 37 the physical volume of the motor is shown as a function of maximum torque potential. The torque potential is the maximum torque that can be exerted by the motor. The figure also show the torque required for driving the pump as a function of temperature at 20bar.

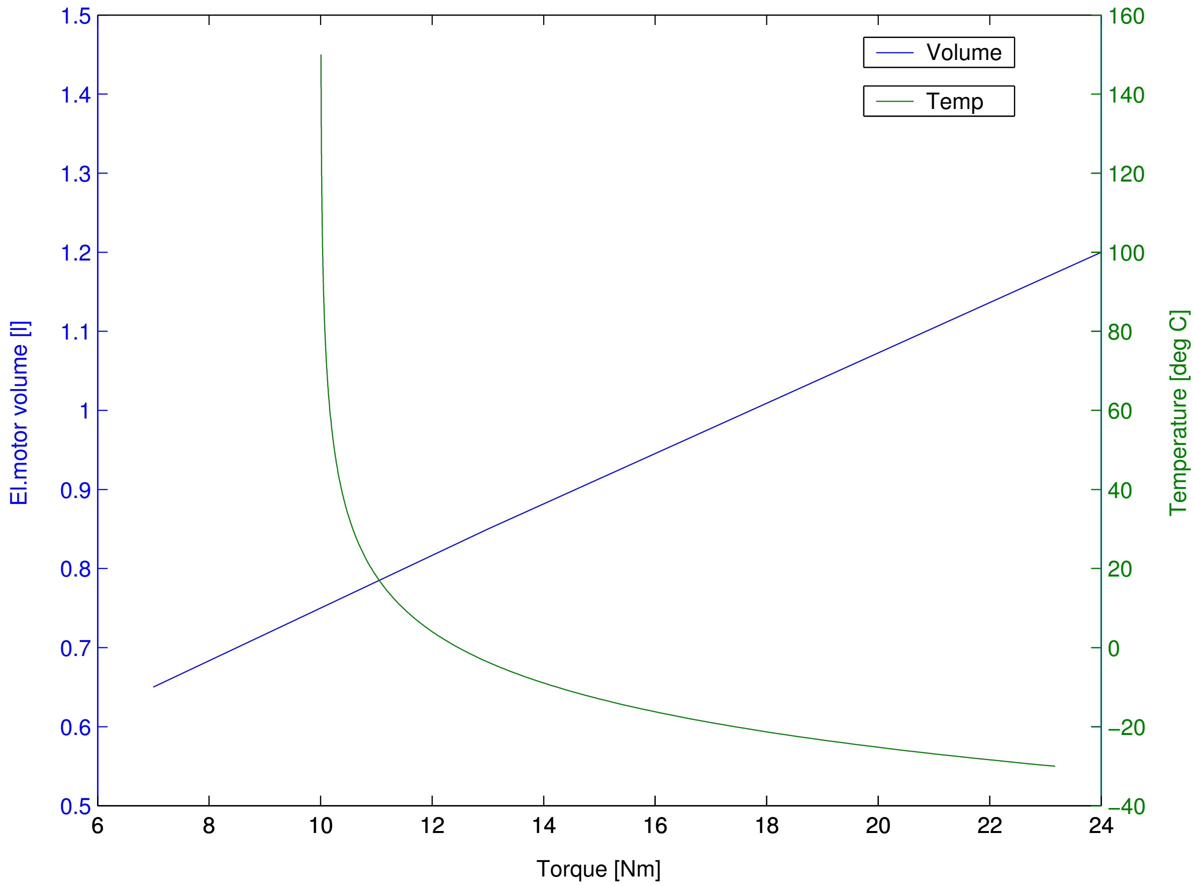

Figure 37: Electric motor volume and temperature as a function of torque at 10µm clearance, 20bar and 7100rpm

The required motor volume capable for driving the pump at -30°C at 20bar differential pressure is approximately 1.2 liters including bearings. If the cold operation performance was irrelevant and the motor could be designed for optimum operation conditions the size of the motor could be reduced by 40%. The complete package including pump and electric motor is substantially larger than the current design, even if designing for optimum operation conditions.

6.4 Cavitation potential

In order to predict cavitation initiation by theoretical or numerical analysis, one has to compare the calculated value of the pressure in a critical region of the flow to a threshold value, typically the vapor pressure. The method of calculation depends on the flow configuration. Since the flow in the pump dealt with in this thesis is assumed to be highly turbulent, experimental investigations have to be employed. However, in appendix VI the risk of cavitation has been assessed by non-dimensional parameters.

If it is assumed that there is no pressure drop in the inlet system and the reference pressure in the inlet system is 1bar, the non-dimensional pressure coefficient equals -0.05. Since it is estimated that the negative pressure coefficient is equal to the cavitation number at initiation, risk of cavitation will occur when the cavitation number equals 0.05. This corresponds to a peripheral velocity for the gears of 13000rpm. It should be noted that the analysis is defined using dynamical parameters and not geometrical ones.

6.5 Noise emissions

Because of the rotating motion of the gear pump vibrations will occur when in operation. Vibrations are normally caused by imbalance of the rotating components, but can also be caused by hydraulic flow through the valves in the valve body. It is obviously desirable to maintain sufficient low vibration levels in order not to interfere with vehicle drivability and ultimately prevent mechanical damage. Vibrations will also cause noise which is undesirable from the driver's point of view. In the design presented, energy is transmitted via the gear teeth to the oil. Because the number of teeth is finite, here 12, pressure pulsations will arise. Also, if the pump is operating above its design interval, cavitation can occur, leading to an unsteady state of flow. The pressure pulsations and unsteady flow may excite vibrations in the pump casing and package, with the result that noise may be transmitted to the surrounding environment. Due to careful design of the housing and gears, the influence of pulsations is reduced. In the proposed design, pressure relief grooves are incorporated in the housing in order for pulsations to be dampened. Since the overall dimensions of the pump are reduced in comparison with the current pump, energy conversion takes place in a much smaller volume. This will in turn reduce the noise emitted from the proposed design if compared to the current pump. The average speed of the electro-hydraulic pump will be lower than the current pump and since high speed operation has an unfavourable effect on the noise developed by the pump it follows that a reduction in average speed leads to lower noise and vibration levels. Additionally, the elimination of the recirculation port, will also lead to a reduction in hydraulic noise levels. The noise emitted by the electric motor will however be a source not present in the current design.

6.6 Fuel savings

When incorporating an electro-hydraulic pump into the transmission all the excess flow can be dismissed. As mentioned in earlier sections the excess flow is assumed to be a loss of power. This power is generated by the internal combustion engine in the vehicle which in turn converts power from the energy stored in the fuel. A reduction in pumping power will result in better fuel economy as the transmission will require less power to rotate and hence the engine will require less fuel to operate. Form figure 5 one can see that >10% of engine power are required for driving auxiliaries. From the three driving cycles defined in section 4.3 the current transmission pump acquires approximately 4% of engine power. This approximation can be done since the power exerted by the engine during the cycles is known and also the theoretical power absorbed by the pump. Figure 38 shows the power consumed by the current pump and the electro hydraulic pump during the three cycles. Working pressure is 20bar and oil temperature is 80°C. The efficiency of the electric motor is assumed to be 90% in all driving modes and pump efficiency is 94%, hence an overall efficiency of 85%. This assumes however that the electric machine is fed by electric power by the generator, see figure 1 and example 8. The current pump is assumed to have a total efficiency of 64% in all operation modes, see figure 14.

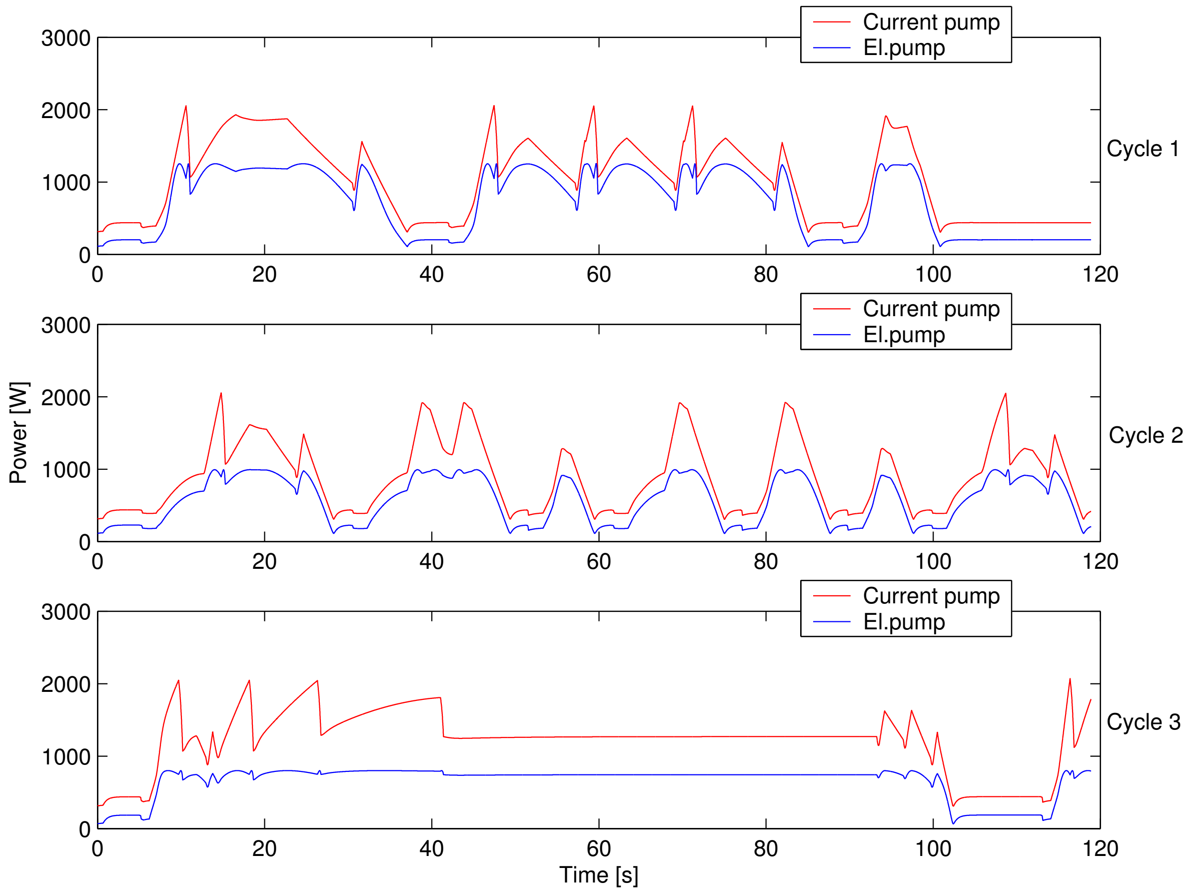

Figure 38: Power consumed during different driving cycles

The reduction in power when incorporating an electrically driven pump can clearly be seen. In cycle 1, the average power reduction is 35.3%, in cycle 2 the reduction is 38.7% and in cycle 3 the reduction is 46.9%. With the previously approximation that the current pump contributes with 4% of engine power consumption, total fuel savings can be estimated. The results are shown in table 9.

	Average power reduction	Estimated fuel savings
Cycle 1	33,8%	1,4%
Cycle 2	37,3%	1,5%
Cycle 3	45,7%	1,8%

Table 9: Power reduction and estimated fuel savings

7. Discussion

The proposed design in this thesis is based on the requirement list. This list was defined early in the project phase before any conceptual ideas could be developed. The requirement list is in turn based on the current Powershift transmissions. This puts limitations in implementing the design into the hybrid transmission. However, the hybrid is based on the GFT base transmission and the ambition is to make the hybridization a total carry over from the base. The hydraulic system would in that case be left unchanged. But, because of the traction motor incorporated into the transmission, the need for cooling of power electronics and rotation parts will increase. This cooling must be supplied by an increased cooling flow by the hydraulic system which ultimately puts higher demands on the pump.

The efficiency study performed shows some really good performance even during high speed operation. The efficiency model created shows the expected trends during low temperature operation and high temperature operation. The volumetric efficiency is expected to decrease with increasing temperatures. The model shows the same trend, however the decrease in efficiency is smaller than expected. This may be the result of small clearance and low differential pressure which reduces the Poiseuille flow. Also, no considerations are taken for the thermal expansion of the materials used. Since the housing and front plate is cast in one material and the gears are sintered, the thermal expansion coefficient may not be the same. This will for sure increase the clearance at elevated temperatures and hence increase the Poiseuille flow. At cold temperatures the mechanical efficiency is expected to be low and should increase with temperature. The prediction is confirmed by the model which shows that the efficiency is considerably low at low temperatures but increases with increasing temperatures.

Analysis concerning the cavitation behavior in the pumping chamber must be taken as a rough calculation. Lack of time and knowledge and the complexity of the problem set limitations. The only was to verify the calculations should be by CFD-analysis or by actually testing the pump unit in a lab. Both this two are however too time consuming and not realistic in this thesis work.

The fuel savings calculated should be taken as an approximation. This is because of the many approximations done in the calculations. The estimated overall efficiency only takes into account the efficiency of the electric machine and the pump unit, no considerations are taken for the efficiency of the power electronics, generator or battery supplying electrical power to the machine. The fuel saving analysis is done with data from driving cycles for a diesel engine. The same analysis for a gasoline engine would for sure result in even larger fuel savings. This because of higher engine speeds for the gasoline engine.

The size of the electric motor could be reduced if cold operation performance were dismissed. With concept 1 and 2, which work in parallel with a mechanically driven pump, the electric motor could be made smaller. This because the mechanically driven pump would pressurize the system at low temperatures and the electric pump would start up at higher temperatures. During the concept design phase some ideas concerning one single pump unit driven both mechanically and electrically via a freewheeling clutch, so that either source of torque could drive the pump, were investigated. This concept turned out to be patented and it was decided not to put more effort into this concept.

The cost aspect is not considered in this thesis, this because of the ambition of purchasing a complete pump package from a supplier. Define available suppliers have not been the aim for this thesis and hence no contacts for cost estimations have been done. Discussions concerning the material in the pump unit should be made with the supplier. It should be very interesting to mold the housing and front plate in some kind of polymer material. This may reduce the overall cost of the pump and also increase volumetric efficiency due to limited thermal expansion of the material.

8. Conclusions

The oil flow demand of the Powershift gearbox is dependent on the functional states of the hydraulic system. Since the functional states are determined by the schematic of the system, there exist 110592. In turn, the functional states are determined by operating parameters such as driving situation, operating temperature and load. The current technology for transmission oil pumps is over dimensioned fixed displacement pumps. Their dimensions are larger than required during normal operation to ensure functionality during extreme functional states at low engine speeds. This requires the use of recirculation ports for re-circling the excess oil not used by the system. Eventually, this causes bad efficiency at usual operation.

Concept 3 proved to be the most suitable solution for supplying oil pressure and flow to the hydraulic system in the hybrid transmission. Due to the electrification of the pump, cooling and hydraulic energy can be provided independently of the engine speed. Because of the electronically control, the oil flow supplied can be matched to the oil flow demanded, thus the pump size can be minimized and its performance optimized. Also, by precisely matching the oil flow, the overall oil quality is improved by minimizing re-circulation of the oil. This reduces aeration and heat generation in the oil thus extending its useful life. The optimized design and the electronically control of the pump supplying oil on demand, will keep the power consumption down and hence increase fuel efficiency. This is particularly true for high reving engines such as gasoline engines and during highway cruise when speed is high and consequently engine speed is high.

In a hybrid electric vehicle the electric pump is an essential part. This is because of the fact that when a hybrid vehicle stops in a light signal or in serve traffic the vehicle engine stops functioning, in this moment the electric pump continue supplying oil flow and pressure necessary to keep the hydraulic system operating and hence the transmission functioning. At start up, the system is already pressurized and gear shifting operations and clutch activation can be performed without time interruptions.

The electric pump package offer very flexible packaging options. The system can be remote mounted external to the transmission casing but also internal if packing space is sufficient in order to accommodate the package. External mounting yields easier installation and maintenance of the system. Designing for cold operation the electric motor must be considerably larger than when designing for optimum operation point. The torque required for starting the pump at low temperatures forces the design for large drivers and causes increased weight and packing matters. Although speed reduction units can be applied in order for employing smaller motors with higher speeds, the electro-hydraulic pump package is essentially larger than the current pump. Since a major part of the package cost is determined by the motor, reduction in motor size can decrease system cost.

9. Future work

This chapter gives some information and thoughts about this remaining work.

9.1 Pump unit

The design presented in this thesis is not a finished product. The pump unit has to be refined and adjusted in order to be fitted onto the transmission. It is not clear today where and how the unit is going to be placed, this because of the numerous parameters that defines the transmissions casing. Preferably, placement of the pump will be in close connection to the sump and valve body. Work have also to be made in order for determine the interfaces to the valve body and the hydraulic system as a whole. In order for getting a feel of the pump a physical model could be made by rapid prototyping. This would also help determine a suitable location of placement.

9.2 Electric motor

The control strategies of the electric motor must be clearly defined in order for minimal power consumption. This implies that the power electronics must have an interface to the TCU. Work must be made in order for identifying this interface and creating a suitable map for the motor. For economical reasons, the pump and electric motor must be supplied by an electro-hydraulic pump manufacturer. This makes the need for identifying a supplier and evaluating price and performance of the pump.

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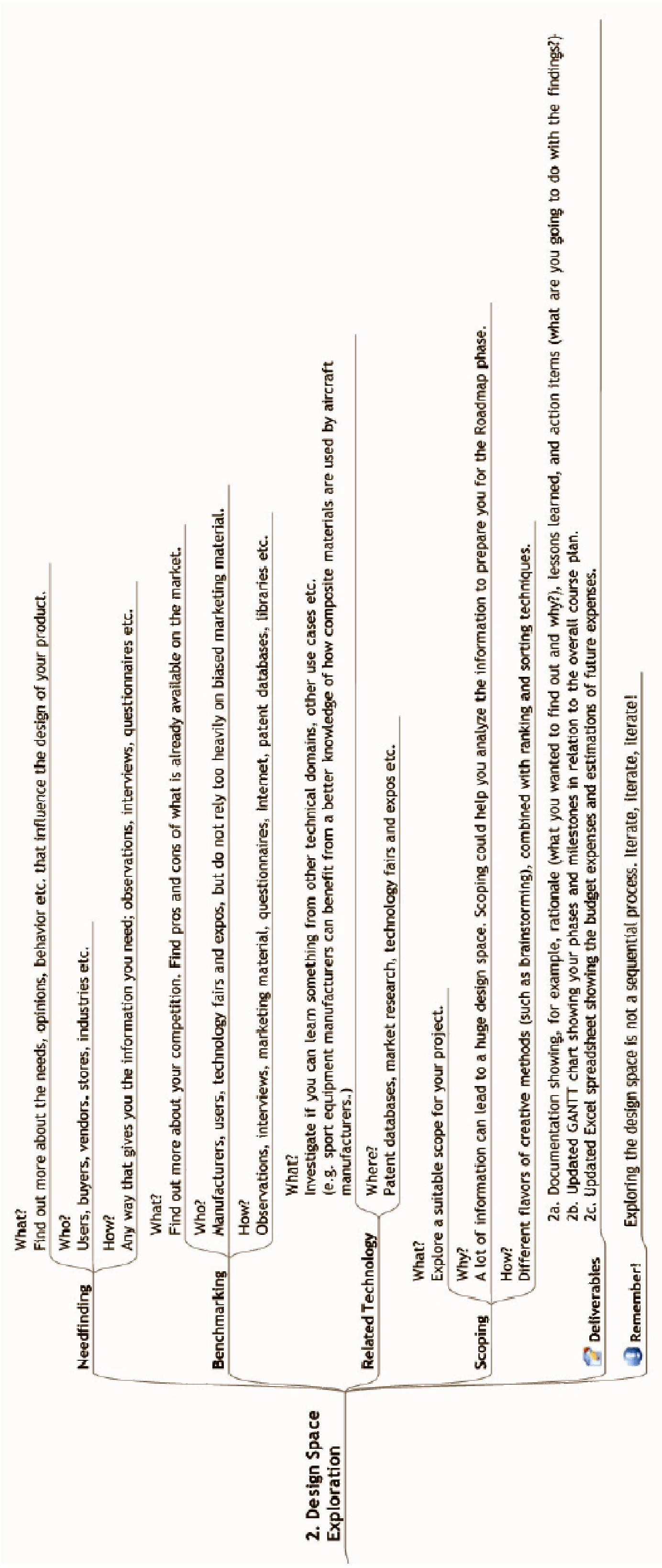
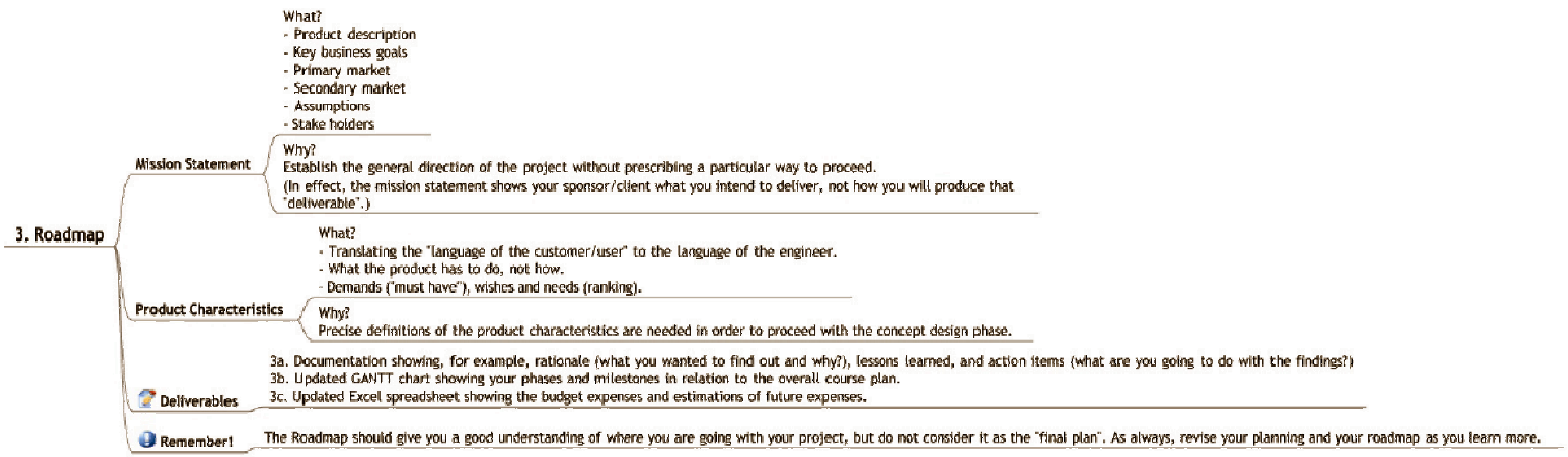
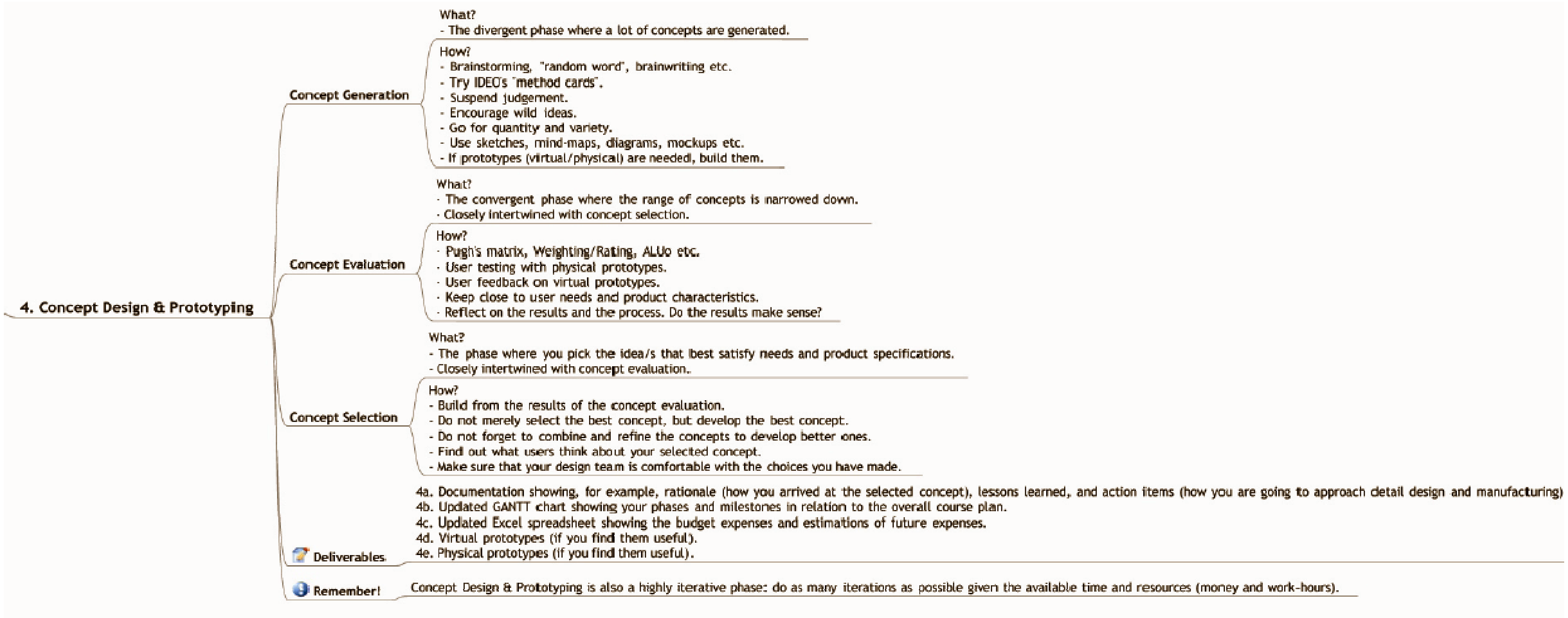
Appendices

Appendix I – Product Development




3. Roadmap
Mission Statement
What?
- Product description
- Key business goals
- Primary market
- Secondary market
- Assumptions
- Stake holders
Why?
Establish the general direction of the project without prescribing a particular way to proceed.
(In effect, the mission statement shows your sponsor/client what you intend to deliver, not how you will produce that "deliverable".)
Product Characteristics
What?
- Translating the "language of the customer/user" to the language of the engineer.
- What the product has to do, not how.
- Demands ("must have"), wishes and needs (ranking).
Why?
Precise definitions of the product characteristics are needed in order to proceed with the concept design phase.
Deliverables
3a. Documentation showing, for example, rationale (what you wanted to find out and why?), lessons learned, and action items (what are you going to do with the findings?)
3b. Updated GANTT chart showing your phases and milestones in relation to the overall course plan.
3c. Updated Excel spreadsheet showing the budget expenses and estimations of future expenses.
Remember!
The Roadmap should give you a good understanding of where you are going with your project, but do not consider it as the "final plan". As always, revise your planning and your roadmap as you learn more.


4. Concept Design & Prototyping
Concept Generation
What?
- The divergent phase where a lot of concepts are generated.
How?
- Brainstorming, "random word", brainwriting etc.
- Try IDEO's "method cards".
- Suspend judgement.
- Encourage wild ideas.
- Go for quantity and variety.
- Use sketches, mind-maps, diagrams, mockups etc.
- If prototypes (virtual/physical) are needed, build them.
Concept Evaluation
What?
- The convergent phase where the range of concepts is narrowed down.
- Closely intertwined with concept selection.
How?
- Pugh's matrix, Weighting/Rating, ALUo etc.
- User testing with physical prototypes.
- User feedback on virtual prototypes.
- Keep close to user needs and product characteristics.
- Reflect on the results and the process. Do the results make sense?
Concept Selection
What?
- The phase where you pick the idea/s that best satisfy needs and product specifications.
- Closely intertwined with concept evaluation.
How?
- Build from the results of the concept evaluation.
- Do not merely select the best concept, but develop the best concept.
- Do not forget to combine and refine the concepts to develop better ones.
- Find out what users think about your selected concept.
- Make sure that your design team is comfortable with the choices you have made.
Deliverables
4a. Documentation showing, for example, rationale (how you arrived at the selected concept), lessons learned, and action items (how you are going to approach detail design and manufacturing)
4b. Updated GANTT chart showing your phases and milestones in relation to the overall course plan.
4c. Updated Excel spreadsheet showing the budget expenses and estimations of future expenses.
4d. Virtual prototypes (if you find them useful).
4e. Physical prototypes (if you find them useful).
Remember!
Concept Design & Prototyping is also a highly iterative phase: do as many iterations as possible given the available time and resources (money and work-hours).

5. Detail Design & Manufacturing

What?
- The phase where you turn your concept into a product.
- Decreasing production cost without sacrificing product quality.

How?
- Use strategies from Design for Manufacturing (DFM) and Design for Assembly (DFA) etc.
- Other strategies (Design for X)? Safety, reuse, reliability, environmental concerns?
- Think about manufacturing costs, costs of components, assembly cost/time etc.
- Use CAD/CAE/CAM software tools to design and evaluate your product.
- Virtual prototyping enables quick evaluations and changes to your product, before it costs too much to correct the mistakes.

Deliverables

5a. Documentation showing, for example, rationale, lessons learned, and action items (how you are going to launch your product)
5b. Updated GANTT chart showing your phases and milestones in relation to the overall course plan.
5c. Updated Excel spreadsheet showing the budget expenses and estimations of future expenses.
5d. Virtual prototype.
5e. Physical prototype.

Remember!

9 times out of 10, manufacturing takes longer and costs more money than expected. Identify risks early!

Appendix II – Design parameters

This appendix summarizes the design parameters and the efficiencies for each concept.

Concept 1

In the mathematical model developed, both the mechanical efficiency and the volumetric efficiency were calculated. The mechanical efficiency was defined in the model as a function of rotational speed, clearance between housing and rotor and the viscosity of the oil. Volumetric efficiency was defined as the leakage losses (P_{vol}) and is a function of rotational speed, clearance viscosity and differential pressure and could be approximated by:

$$P_{vol} = \Delta Q_v \cdot \Delta p = Uh\left[1 - \frac{2h^2}{3\mu U}\Delta p\right] \cdot \Delta p \,, \qquad (1)$$

where U denotes the rotational speed, h denotes the clearance between the rotor and housing, μ denotes the dynamic viscosity of the oil and Δp the differential pressure.

With the assumption of pure viscous friction and constant clearance, the losses due to friction (P_{mech}) could be determined:

$$P_{mech} = M_f \omega = \frac{\mu}{2h} A \omega \Delta r^2 \,, \qquad (2)$$

where ω denotes angular velocity, A denotes the area of the rotor and Δr denotes the radius from the rotor axis.

The model was assumed to be valid for both types of pumps with some modifications due to the volume of oil trapped between the teeth when meshing in the spur gear pump design. The contact between the tips of the gear teeth and housing and vane tips and housing was also considered a difference in the model. The resulting dimensions and efficiencies are presented in table 1.

Vane pump		Gear pump	
Rotor diameter [mm]	32,4	Number of teeth	12
Rotor width [mm]	9,1	Pitch diameter [mm]	23,0
Eccentricity [mm]	2,5	Gear width [mm]	18
Pressure chamber diameter [mm]	37,4	Module	1,9
Number of vanes	5	Addendum circle [mm]	26,9
		Clearance circle [mm]	19,2
		Dedendum circle [mm]	18,2
		Max. shaft diameter [mm]	9
Displacement [mm3/rev]	5000	Displacement [mm^3/rev]	5000
Displacement [cm3/rev]	5	Displacement [cm^3/rev]	5
Clearance [mm]	0,01	Clearance [mm]	0,01
Viscosity @ 30℃ [mm2/s]	65	Viscosity @ 30℃ [mm2/s]	65
Rotational speed [rpm]	2200	Rotational speed [rpm]	2200
Angular speed [rad/s]	230,38	Angular speed [rad/s]	230,38
Differential pressure [bar]	8	Differential pressure [bar]	8
Mech. loss. [W]	12,966	Mech. loss. [W]	20,871
Vol. loss. [W]	0,821	Vol. loss. [W]	3,282
Total loss [W]	13,787	Total loss [W]	24,153
Total loss ~ 38 W			
Total efficiency @ 8 bar and 2200 rpm ~ 87%			

Table 1: Calculated parameters

It was clear that by decreasing the diameter of the rotors the peripheral velocities was dramatically reduced and hence the mechanical losses were decreased. This could ultimately increase the total efficiency. Since the displacement for both pump types is a function of rotor diameter, the displacement would decrease if diameter was decreased. In order for the concept to maintain the required flow rate, the speed of the electric pump should be increased. However, due to structural strength of the parts in the pumps, a further reduction of the overall dimensions would risk operability at low temperatures. This is valid for all the concepts in this appendix.

Concept 2

The same mathematical model as for concept 1 was used in order for approximating the efficiency of this concept. The results are found in table 2.

Gear pump mech.		**Gear pump el.**	
Number of teeth	12	Number of teeth	12
Pitch diameter [mm]	23,0	Pitch diameter [mm]	23,0
Gear width [mm]	18	Gear width [mm]	18
Module	1,9	Module	1,9
Addendum circle [mm]	26,9	Addendum circle [mm]	26,9
Clearance circle [mm]	19,2	Clearance circle [mm]	19,2
Dedendum circle [mm]	18,2	Dedendum circle [mm]	18,2
Max. shaft diameter [mm]	9	Max. shaft diameter [mm]	9
Displacement [mm^3/rev]	5000	Displacement [mm^3/rev]	5000
Displacement [cm^3/rev]	5	Displacement [cm^3/rev]	5
Clearance [mm]	0,01	Clearance [mm]	0,01
Viscosity @ 30℃ [mm2/s]	65	Viscosity @ 30℃ [mm2/s]	65
Rotational speed [rpm]	2200	Rotational speed [rpm]	2200
Angular speed [rad/s]	230,38	Angular speed [rad/s]	230,38
Differential pressure [bar]	8	Differential pressure [bar]	8
Mech. loss. [W]	20,871	Mech. loss. [W]	20,871
Vol. loss. [W]	3,282	Vol. loss. [W]	3,282
Total loss [W]	24,153	Total loss [W]	24,153
Total loss ~ 48 W			
Total efficiency @ 8 bar and 2200 rpm ~ 84%			

Table 2: Calculated parameters

Concept 3

The same mathematical model as for concept 1 was used in order for approximating the efficiency of this concept. The results are found in table 3.

Electro-hydraulic gear pump	
Number of teeth	12
Pitch diameter [mm]	23,0
Gear width [mm]	18
Module	1,9
Addendum circle [mm]	26,9
Clearance circle [mm]	19,2
Dedendum circle [mm]	18,2
Max. shaft diameter [mm]	9
Displacement [mm^3/rev]	5000
Displacement [cm^3/rev]	5
Clearance [mm]	0,01
Viscosity @ 30℃ [mm2/s]	65
Rotational speed [rpm]	4400
Angular speed [rad/s]	460,77
Differental pressure [bar]	8
Mech. loss. [W]	41,742
Vol. loss. [W]	4,583
Total loss [W]	46,325
Total efficiency @ 8 bar and 4400 rpm ~ 84%	

Table 3: Calculated parameters

Appendix III – Concept evaluation

In order to objectively evaluate the different pump concepts a number of criterions have been developed. This step is based, first of all, on the requirements list. During concept development, variants that were found to be unsuitable in principle were eliminated. The requirement list consists of criterions that must be fulfilled by the design and since these are static they cannot be evaluated. The evaluation criterions consists therefore of parameters that improves performance, size and price. During the conceptual phase it has been difficult to put actual figures to the cost. Nevertheless, both technical and economic aspects have been considered and evaluated. The criterions are illustrated in figure 1.

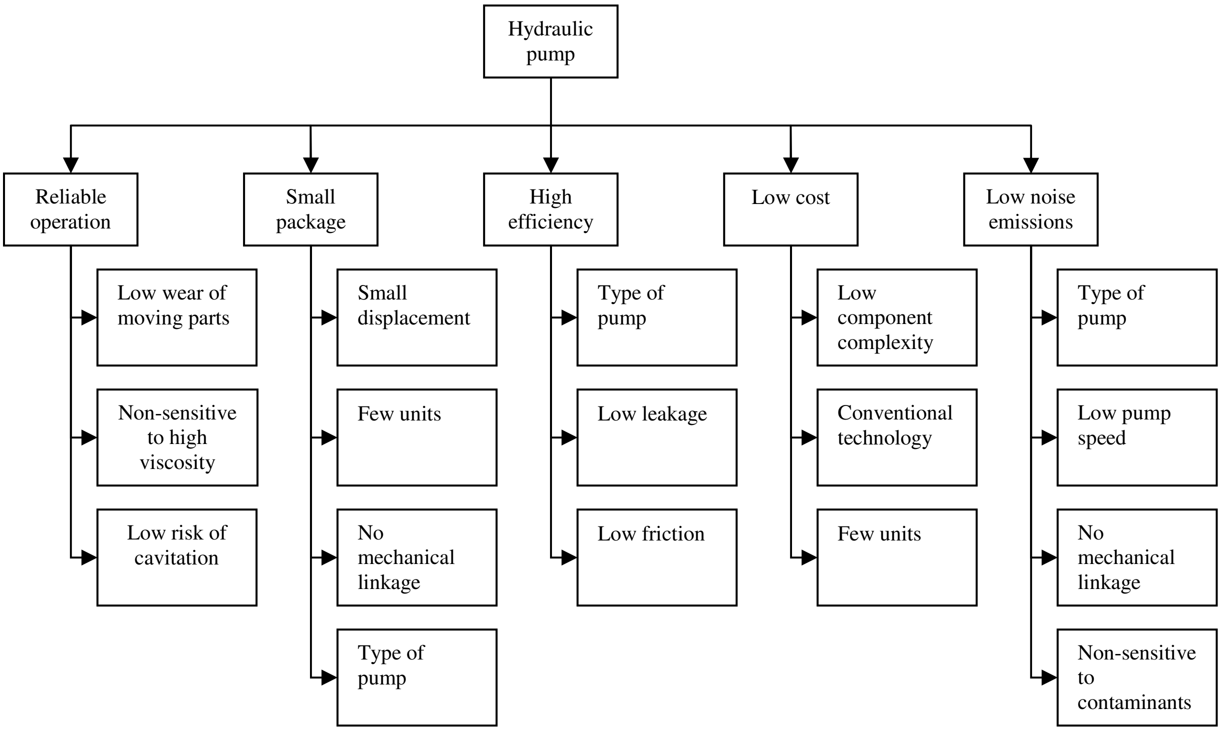


Figure 1: Criteria of the hydraulic pump

Weights of the criteria

In order to find out how important the different criteria are they have been weighted with a three-level grade.

0 = The horizontal criterion is less important than the vertical.
1 = The horizontal criterion is as important as the vertical.
2 = The horizontal criterion is more important than the vertical.

This type of weighting gives an objective estimation of how important the criteria are. The method is further described in Phal & Beitz [5].

Table 1 below shows how each of reliable operation, small package, high efficiency, low cost and low noise emissions is weighted.

	Reliable operation	Small package	High efficiency	Low cost	Low noise emissions	Sum	Weight
Reliable operation	1	2	2	2	2	9	0,360
Small package	0	1	2	0	2	5	0,200
High efficiency	0	0	1	0	2	3	0,120
Low cost	0	2	2	1	2	7	0,280
Low noise emissions	0	0	0	0	1	1	0,040
Sum						25	1,00

Table 1: Weights of reliable operation, packing size, efficiency, price and noise emissions

From the table it can be seen that reliable operation is the most important parameter of the pump design whereas noise emissions have a low weight and therefore not as important as the rest.

In order for the pump design to be reliable during service life the criteria of reliable operation and their weights are presented in table 2.

	Low wear of moving parts	Non-sensitive to high viscosity	Low risk of caviation	Sum	Weight
Low wear of moving parts	1	0	0	1	0,111
Non-sensitive to high viscosity	2	1	2	5	0,556
Low risk of caviation	2	0	1	3	0,333
Sum				9	1,00

Table 2: The weights of the criteria of reliable operation

To ensure that the pump package occupy a limited space it is important to know the weights of the package size criteria. The weights of these criteria are found in table 3.

	Small displacement	Few units	No mechanical linkage	Type of pump	Sum	Weight
Small displacement	1	0	0	0	1	0,063
Few units	2	1	2	2	7	0,438
No mechanical linkage	2	0	1	1	4	0,250
Type of pump	2	0	1	1	4	0,250
Sum					16	1,00

Table 3: The weights of the criteria of small package

The criteria high efficiency and its sub-criteria's weights can be seen in table 4.

	Type of pump	Low leakage	Low friction	Sum	Weight
Type of pump	1	1	1	3	0,333
Low leakage	1	1	1	3	0,333
Low friction	1	1	1	3	0,333
Sum				9	1,00

Table 4: The weights of the criteria of high efficiency

The criteria of cost and its weighted values are presented in table 5:

	Low component complexity	Conventional technology	Few units	Sum	Weight
Low component complexity	1	1	0	2	0,222
Conventional technology	1	1	0	2	0,222
Few units	2	2	1	5	0,556
Sum				9	1,00

Table 5: The weights of the criteria of low cost

To ensure low noise emissions the criteria of table 6 should be fulfilled.

	Type of pump	Low pump speed	No mechanical linkage	Non-sensitive to contaminants	Sum	Weight
Type of pump	1	0	2	1	4	0,250
Low pump speed	2	1	2	2	7	0,438
No mechanical linkage	0	0	1	0	1	0,063
Non-sensitive to contaminants	1	0	2	1	4	0,250
Sum					16	1,00

Table 6: The weights of the criteria of low noise emissions

Figure 2 illustrates the criteria tree and each weight.

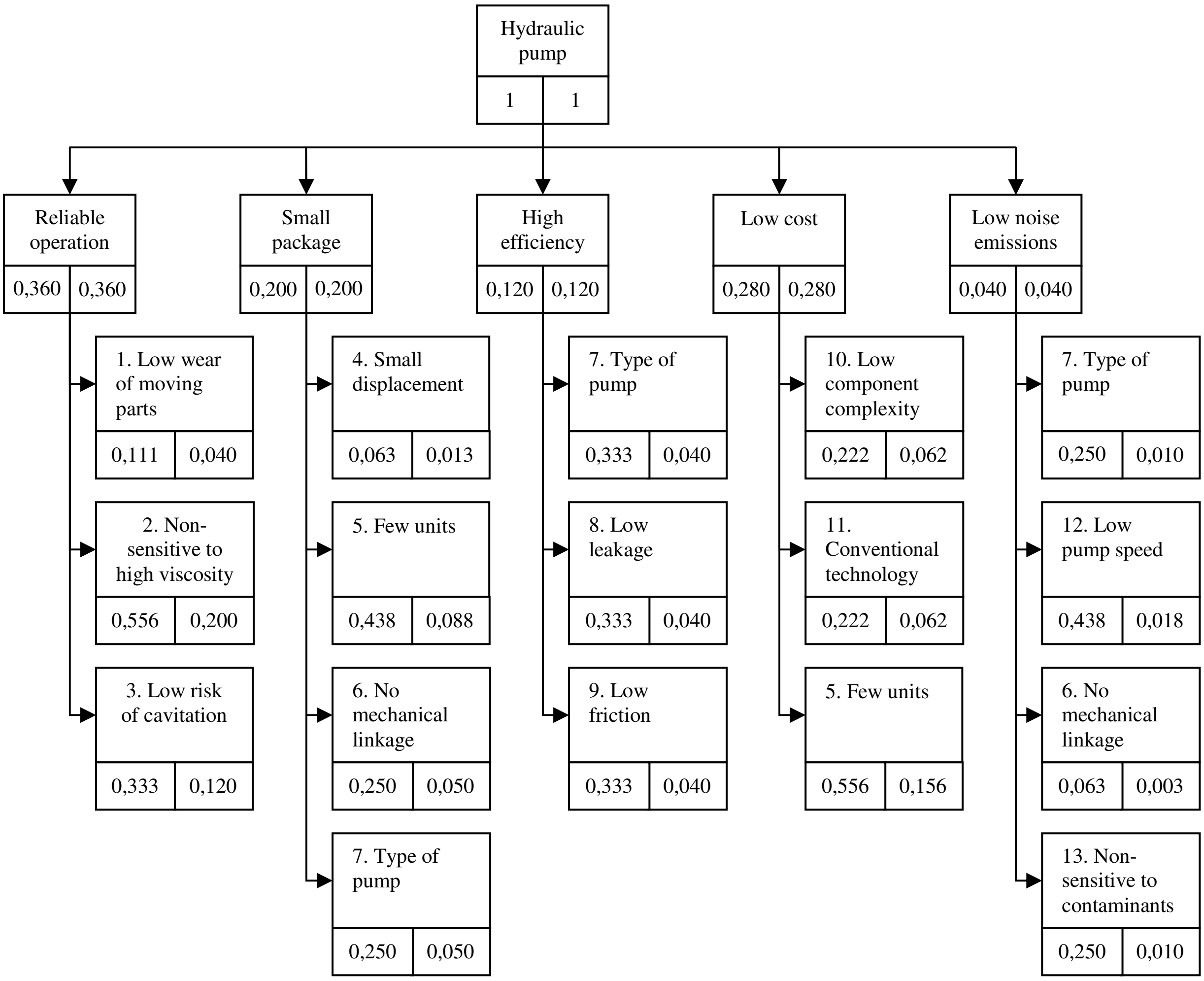


Figure 2: Criteria tree of the hydraulic pump package with their weights

The result from the weighting is illustrated in figure 3.

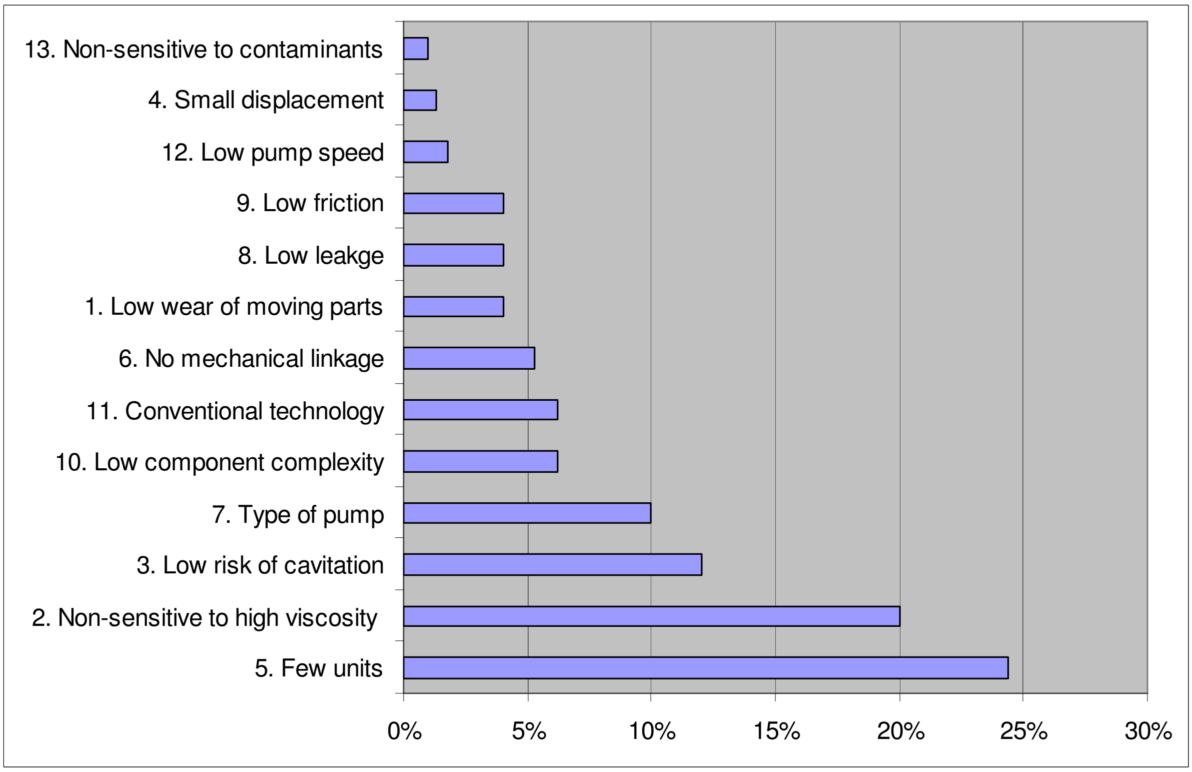


Figure 3: Chart showing the result gained from the grading of the hydraulic pump design

Evaluation of the proposed concept designs

The three different concepts have been weighted with the current pump as a reference for each criterion. This gave an overall value for each of the concepts where the concept with the highest score is the best suited for further development. The grades are:

-1 = Does not fulfill the criterion as good as the reference.
0 = Fulfill the criterion as good as the reference.
1 = Fulfill the criterion better than the reference.

The weightings and the criteria are illustrated in table 7.

Criteria	Grade				Weighted grade			
	Current pump	Concept 1	Concept 2	Concept 3	Weight	Concept 1	Concept 2	Concept 3
1. Low wear of moving parts	0	1	1	1	0,040	0,04	0,04	0,04
2. Non-sensitive to high viscosity	0	1	1	0	0,200	0,2	0,2	0
3. Low risk of cavitation	0	1	1	1	0,120	0,12	0,12	0,12
4. Small displacement	0	1	1	1	0,013	0,013	0,013	0,013
5. Few units	0	-1	-1	0	0,244	-0,244	-0,244	0
6. No mechanical linkage	0	0	0	1	0,053	0	0	0,053
7. Type of pump	0	1	0	0	0,100	0,1	0	0
8. Low leakage	0	1	1	1	0,040	0,04	0,04	0,04
9. Low friction	0	1	1	1	0,040	0,04	0,04	0,04
10. Low component complexity	0	-1	0	0	0,062	-0,062	0	0
11. Conventional technology	0	-1	-1	-1	0,062	-0,062	-0,062	-0,062
12. Low pump speed	0	0	0	-1	0,018	0	0	-0,018
13. Non-sensitive to contaminants	0	0	0	0	0,010	0	0	0
Sum					**1,00**	**0,185**	**0,147**	**0,226**

Table 7: Criteria and overall score for the three different design concepts

From table 7 one can conclude that concept 3 is the design that best fulfill the criteria. The main reason for this is that concept 3 only consists of one pumping unit. The only two drawbacks for this concept are its pump speed and the relatively new technology of electro-hydraulics in the automotive industry.
The only difference between concept 1 and 2 is the design of the mechanically driven pump unit. This is also reflected in the overall score where the design proposal for concept 1 uses a more suitable type of mechanically driven pump. Apart from this the three concepts performs equally in almost all criteria although concept 3 is more advantageous because of the previously stated single unit design and lack of mechanical link. An explanation of each grade can be found in table 8.

Criteria	Grade			
	Current pump	Concept 1	Concept 2	Concept 3
1	0	1; Moving parts are tailor made and design is focusing on optimum lubrication	1; Moving parts are tailor made and design is focusing on optimum lubrication	1; Moving parts are tailor made and design is focusing on optimum lubrication
2	0	1;Two units perform better since both displace volume	1; Two units perform better since both displace volume	0; One unit perform equally as current pump with same design
3	0	1; Two pumping units perform better since speed can be reduced	1; Two pumping units perform better since speed can be reduced	0; One unit perform equally as current pump with same design
4	0	1; Lower displacement will decrease gear radius and hence reduce peripheral speed	1; Lower displacement will decrease gear radius and hence reduce peripheral speed	0; Displacement will be lower but speed will increase
5	0	-1; Design needs two pumping units	-1; Design needs two pumping units	0; Only one unit is needed
6	0	0; Design needs mechanical linkage	0; Design needs mechanical linkage	1; No need for mechanical linkage
7	0	1; Design consists of variable displacement pump	0; No possibility to vary displacement	0; No possibility to vary displacement
8	0	1; Tailor made design with leakage compensator	1; Tailor made design with leakage compensator	1; Tailor made design with leakage compensator
9	0	1; Tailor made design for optimum lubrication	1; Tailor made design for optimum lubrication	1; Tailor made design for optimum lubrication
10	0	-1; Relatively complex design	0; Same design as reference	0; Same design as reference
11	0	-1; Incorporates electric machine	-1; Incorporates electric machine	-1; Incorporates electric machine
12	0	0; Same as reference	0; Same as reference	-1; Higher than reference
13	0	0; Same as reference	0; Same as reference	0; Same as reference

Table 8: Explanation of the different weightings

Appendix IV - Selection and analysis of gears

The typical construction of an external gear pump consists of two or more spur gears, a drive shaft, and a housing to hold it all. The spur gears are set in the housing between the low pressure side and the high pressure side of the pump. The first of the two gears is connected to the drive shaft. This gear is termed the drive gear. The second gear is called the idler gear. It meshes with the drive gear and is driven by it. There are many possible variations of the external gear pump type. They range from multiple gears, to different types of gears such as helical and herringbone. When the drive shaft rotates it in turn rotates the drive gear which then drives the idler gear. As the gears rotate, the gear teeth unmesh on the inlet side and mesh on the outlet side of the pump. An ideal meshing can only be achieved by a perfectly designed and manufactured gear unit. Noise, vibration and short life are some of the penalties to be paid for gears imperfectly designed and manufactured. This appendix will give the basic aspects of gear design and to recognize the limitations.

Tooth profile

The profile of a gear tooth must be chosen with the following aspects in mind:

1. The gears must mate and mesh with a smooth uniform action.
2. The tooth must have a section sufficiently strong for the applied loads.
3. The tooth must be free from weakening undercuts.
4. The tooth will mesh at the correct shaft centre distance.
5. The profile of the teeth offers no manufacturing difficulties.
6. The geometry provides an adequate tooth overlap.

The involute curve provides the most widely used profile for gear and teeth design. And so this profile will be used and described here.

Involute profile

An involute profile curve can be constructed by tracing the end of a cord unwound from the periphery of a circular disc, see figure 1. The contour of the involute curve is governed only by the diameter of the disk from which it is developed. There is no limit in length of the involute curve. Under working conditions, the contact between two teeth at the pitch point is pure rolling contact. Either side of that point, the contact is sliding and the rate of sliding constantly varies. The contact area of an involute gear is shown in figure 2.

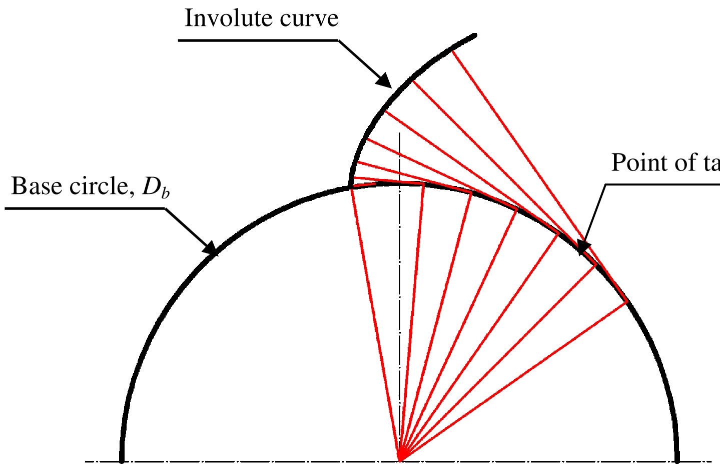


Figure 1: Developing an involute curve

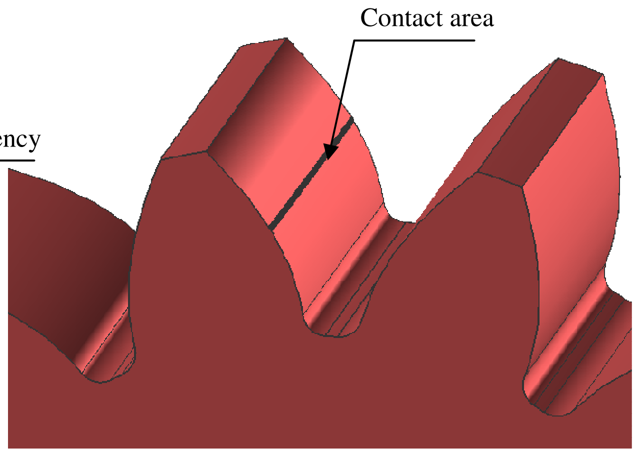


Figure 2: Contact area for involute gear

Figure 3 indicates some of the basic terms used in connection with the design of gears and gearing.

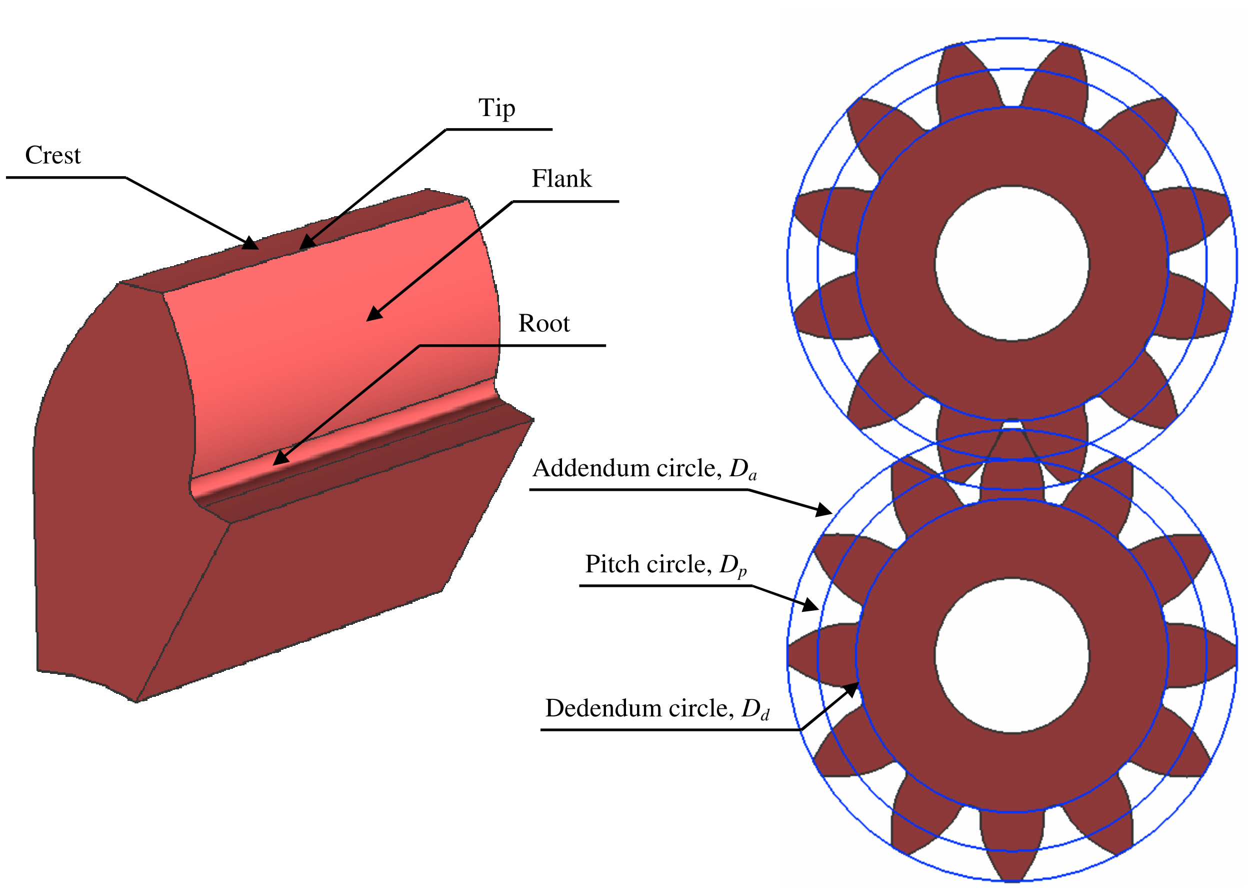


Figure 3: Nomenclature for gear tooth and meshing gears

For efficient running it is important that correct meshing of teeth is ensured, and this comes down primarily by establishing the correct centre distance for the shafts. If there is no tangency between the two dedendum circles of the gears, the teeth crests and root will get a different shape. Ultimately, if the centre distance grows larger the teeth tip will get a sharp profile and the root will get thicker.

As previously stated, the gear pump design displace a fixed volume of fluid each revolution. This volume is dependent on the free space between each tooth. The formula for calculating this volume is given by equation 1:

$$V = 2hD_p\pi b\,, \qquad (1)$$

where *2h* denotes the depth of the teeth, D_p the pitch diameter and *b* the gear width. The depth of the teeth is the distance from the tip of the teeth to the base circle. The base circle is defined by:

$$D_b = D_p - 2m\,, \qquad (2)$$

where *m* is the module which denotes the size of the gear. The module is defined as:

$$m = \frac{D_p}{z}\,, \qquad (3)$$

where z is the number of teeth.

When the pitch diameter and the module are known, the addendum circle diameter can be calculated by:

$$D_a = 2m + D_p\,. \qquad (4)$$

And also the dedendum circle can be calculated:

$$D_d = D_p - 2m * 1.25\,. \qquad (5)$$

Table 1 shows data for the gears.

Number of Teeth, *z*	12
Pitch Diameter, D_p (mm)	23,0
Gear Width, *b* (mm)	18,0
Module, *m*	1,9
Addendum Circle, D_a (mm)	26,9
Base circle, D_b (mm)	19,2
Dedendum Circle D_d (mm)	18,2
Displacement, *V* (mm^3/prev)	5000
Displacement (cc/prev)	5,00

Table 1: Data regarding the dimensions of the gear

The centre distance, if the offset between the pitch circles are zero, see figure 3, can be calculated by:

$$a = \frac{D_{p_1} + D_{p_2}}{2} = \frac{m(z_1 + z_2)}{2}. \qquad (6)$$

And since the pitch circle is the same for both gears in this case, the centre distance becomes D_p.

Strength of gears

With the geometrical values defined, it is now possible to calculate other physical parameters such as weight, size and forces acting on the gears. When the forces are known, other design considerations can be defined. These considerations include design of axis and type and size of bearings.
For example: torque can be expressed as:

$$T = F * \frac{d}{2}, \qquad (8)$$

where F is the force acting on the teeth and d is the diameter where the force is acting. From this expression one can se that if T is given, the force will increase if the diameter is decreasing. This gives the following contradicting rule:

- Smaller gear → larger force → larger axis diameter and bearings
- Larger gear → smaller force → smaller axis diameter and bearings

Since largest possible T is known and also d is known, F can easily be calculated. The structural integrity of the gears will depend upon the force applied. Figure 4 illustrate the areas where structural stress will appear.

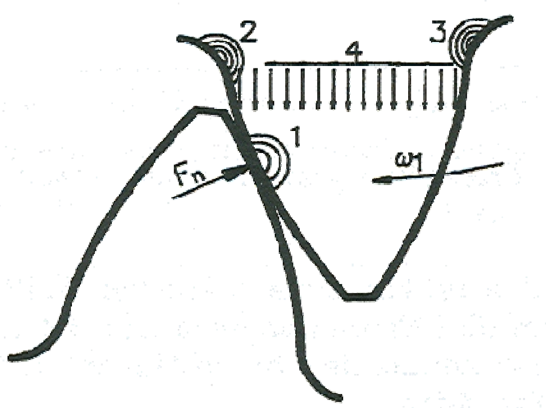


Figure 4: Structural stress areas [7]

Area 1 indicate surface pressure, σ_H, defined by Herts (see below), 2 and 3 are areas experience bending stress, at 2 tension stress and at 3 compression stress. This will also be dealt with in a later section. Area 4 indicates normal and shearing forces. These are however small relative to surface pressure and bending stresses and will therefore be neglected. Frictional forces appearing due to sliding between the surfaces of the flanks are also neglected. From figure 4, the normal force, F_n, acts on the flank under the angle α from the horizontal. F_n can therefore be expressed as forces in the xy-plane:

$$\begin{aligned} F_x &= F_{per} = F_n \cos\alpha \\ F_y &= F_{rad} = F_n \sin\alpha \end{aligned} \qquad (9)$$

Where F_{per} is the peripheral force and F_{rad} is the radial force. F_{per} is the force that transfers the torque.

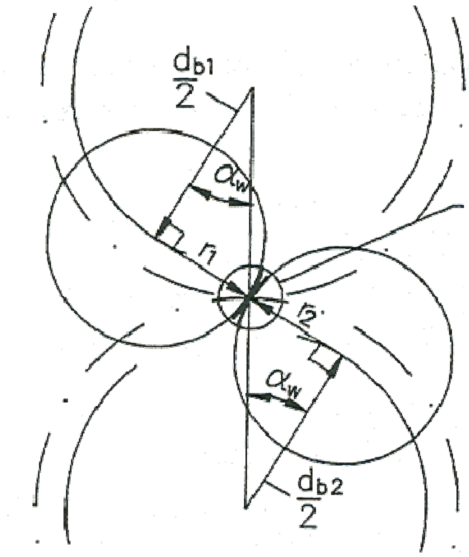
Calculations of load bearing capacity

The contact between two flanks when meshing will give rise to a contact pressure or a surface pressure. The material subjected to this contact will experience shear stresses. If the shear stresses exceed the materials shear strength, pitting can occur. If two cylindrical bodies with the same material and length are pressed against each other with the force F_n, Hertz contact pressure can be expressed as:

$$\sigma_H = \sqrt{\frac{F_n * E}{2\pi L(1-\nu^2}\left(\frac{1}{r_1}+\frac{1}{r_2}\right)}, \qquad (10)$$

where E = Young's modulus
L = length of contact
ν = Poisson's ratio
r_1= radius of cylinder 1
r_2= radius of cylinder 2

As previously stated, the contact between two teeth at the pitch point is a pure rolling contact. Either side of that point, the contact is sliding and hence can be treated as a uniform load. The contact at the pitch point will experience an EHD-contact (Elasto-Hydro-Dynamic) and therefore Hertz theorem can be applied. However, the radius of the cylinders in the formula must correspond to the radius of the evolvent profile on the rolling point, see figure 5.



In the rolling point, the tooth flanks are assumed to replace Hertz's cylinders. r_1 and r_2 corresponds to the cylinders radius.

Figure 5: Evolvent radius at rolling point [7]

From figure 5:

$$r_1 = r_2 = r = \sqrt{\left(\frac{D_p}{2}\right)^2 - \left(\frac{D_b}{2}\right)^2}\,, \qquad (11)$$

where $\alpha_w = \arctan\left(\frac{2r}{D_b}\right)$. (12)

In the DIN3990 standard, calculation of load-bearing capacity of spur gears, the permissible contact pressure for a straight tooth is expressed as:

$$k_{perm} = \frac{HB^2}{2560\sqrt[6]{n}}\,, \qquad (13)$$

where *HB* is the Brinell hardness and *n* denotes the rotational speed.

Resistance to pitting and wear due to excessive contact pressure is provided by the equation for S_w, which is given by:

$$S_w = \varphi * k_{perm}\,, \qquad (14)$$

where φ is a tabulated life factor, table 2.

Service life [h]	10	50	150	312	625	1200	2500	5000	10000	40000	80000	150000
Life factor, φ	2,82	2,15	1,79	1,59	1,41	1,27	1,12	1	0,89	0,71	0,83	0,57

Table 2: Life factor

Guidelines in the DIN3990 standard implies that for small gear pairs, $z < 20$, S_w should be greater or equal to 1.2 in order to avoid pitting. Selection of service life can be taken as the lifetime of the whole gearbox. The current requirements are 350000km, converted to lifetime this can be assumed to be 5000 hours if the car is run at an average of 70km/h during its lifetime.

For S_w equals 1.5 due to guidelines, and a life factor of 1, the minimum hardness of the gears are given by:

$$HB = \sqrt{\frac{S_w}{\varphi} 2560 \sqrt[6]{n}} \ . \qquad (15)$$

Teeth calculations for bending and tooth fracture

Area 2 and 3 in figure 4 indicate areas which will experience bending stress when meshing. Elementary beam theory can be applied in order to calculate bending stress and deflection. If the tooth is considered a rigid beam and a load is applied on the top of the flank, the first elementary case can be used:

$$\sigma_{Factual} = \frac{M_b}{W_b} = \frac{6F * h}{b * s^2}, \qquad (16)$$

where F denotes the applied force, h the height of the tooth, b the width and s the thickness.

If the pump is to deliver 20bar, the torque applied to the driven gear is given by:

$$T = P * V, \qquad (17)$$

where P is the pressure in Pa and V is the displacement in m^3/rev. Since T consists of a force and a distance, the force can be calculated if the distance is known. In order to relate this force to the force, F, in the first elementary case equation, the radius of the addendum circle is used. In the real case this force will not act perfectly peripherally on the circle but will consist of both a peripheral force and a radial force. However, a rough estimate is that all of the force will act peripheral and hence this force can be referred to as F in the elementary case equation.

The permissible stress at area 2 and 3 are given by the formulas in the DIN3990 standard. The permissible root stress is defined in the standard as:

$$\sigma_{Fperm} = \sigma_{F\lim} Y_{NT} Y_L, \qquad (18)$$

where σ_{Flim} is the permissible root stress, Y_{NT} is a tabulated life factor dependent upon the number of load changes and Y_L is a constant depending on the type of load.

Resistance for tooth fracture is calculated by equation 19:

$$S_F = \frac{\sigma_{Fperm}}{\sigma_{Factual}}. \qquad (19)$$

The value of S_F should be higher or equal to 1 if fracture is to be avoided.

Material

The material used in the gears in the current pump is sintered steel with internal name D39. The limited property data available is shown in table 3.

Sintered steel D39					
Density [g/cm^3]	Porosity [%]	Tensile strength [MPa]	Yield strength [MPa]	Young's modulus [MPa]	Hardness HB
>6,9	<11	730	440	120000	>200

Table 3: Properties of D39

Results

The calculated data for the gear with the dimensions presented in table 1 and material in table 3, is shown in table 4.

Torque @ -30°C [Nm]	23,17
Normal force, F_n [N]	1723
Peripheral force, F_{per} [N]	1436
Radial force, F_{rad} [N]	952
Contact pressure, σ_H [MPa]	795
Bending stress, σ_F [MPa]	83
r [mm]	6,4
α_w [°]	33,5
k_{perm}	1,5
S_w	1,5
φ	1
Rotational speed, n [rpm]	5000
Min. hardness, *HB*	130
Premissible bending stress, σF_{perm} [MPa]	308
Permissible root stress, σ_{Flim} [MPa]	440
Y_{NT}	1
Y_L	0,7
S_F	3,7

Table 4: Calculated values

From table 4 one can conclude that minimum hardness exceeds the hardness for the material currently used. By making the gears in a harder material, fail safe operation can be achieved for >5000 hours and also increase resistance of tooth fracture.

Appendix V – Bearing calculations

Bearing life

Every care has been taken to ensure the accuracy of this calculation but no liability can be accepted for any loss or damage whether direct, indirect or consequential arising out of the use of the calculation.

Select η_c: 0.5

Bearing	HK 1012
F_w, mm	10
D, mm	14
C, kN	5.39
P_u, kN	0.78
P, kN	0.490
n, r/min	3500
v, mm²/s	17

		L_{10}	2960	L_{10h}	14100
κ	0.899				
v_1	18.9				
a_{SKF}	2.09	L_{10m}	6170	L_{10mh}	29400

Old a_{23} method for comparison

a_{23}	0.829	L_{10a}	2450	L_{10ah}	11700

For grease lubricated bearings, please check the grease life. See section "Grease lubrication"
For calculation of two bearings on a shaft, see the program "SKF Bearing Select"
For calculation of the contamination factor η_c, see the program "SKF Bearing Select"

Appendix VI - Non-dimensional parameters

This appendix presents some approximations concerning the cavitation limit for the designed pump.

Cavitation number

In a hydraulic system liable to cavitate, such as a pump, p_r is the pressure at a conventional reference point r where it is easily measurable. Usually, r is chosen in a region close to that where cavitation inception is expected. If T is the operating temperature of the liquid and Δp is the pressure difference that characterizes the system, the cavitation number (also called Euler number) for a pump is defined by:

$$Ca = \frac{p_{inlet} - p_v(T)}{\rho V_p^{\ 2}}, \qquad (1)$$

where V_p is the velocity at the periphery of the gear.

It should be noted that the cavitation number is defined using dynamical parameters and not geometrical ones.

Cavitation number at initiation

The number σ_{iv} is the value of the parameter Ca corresponding to cavitation initiation at any point of the flow system. Cavitation appears because of either a decrease in pressure at the reference point or an increase in the Δp-value. Operation in non-cavitating conditions requires that:

$$Ca > \sigma_{iv}. \qquad (2)$$

The threshold σ_{iv} depends on all the usual factors considered in fluid mechanics such as flow geometry, viscosity, gravity, surface tension, turbulence levels, thermal parameters, wall roughness and the gas content of the liquid in terms of dissolved and free gases. In general, the smaller the value of σ_{iv} for a given system, the better behaved is the flow. When Ca becomes smaller than σ_{iv}, cavitation usually becomes increasingly developed. In many circumstances, particularly for the numerical modeling of cavitating flows, the following estimate is taken for σ_{iv}:

$$\sigma_{iv} = -Cp_{\min}, \qquad (3)$$

where Cp_{min} is the minimum pressure coefficient, which is normally negative. The pressure coefficient Cp at a point M is defined by the relation:

$$Cp = \frac{p_M - p_r}{\Delta p}. \qquad (4)$$

In this expression, p_r is the absolute pressure at the reference point. Two assumptions lie behind equation 4. First, cavitation occurs at the point of minimum pressure and second, the pressure threshold value is that of the vapor pressure. However, the estimated result must be considered cautiously.

If it is assumed that there is no pressure drop in the inlet system and p_r is 1bar as a reference in the inlet, the pressure coefficient equals -0.05. Since it was estimated that the negative pressure coefficient is equal to the cavitation number at initiation, σ_{iv} becomes 0.05. Hence, risk of cavitation will occur when Ca equals 0.05. This corresponds to a peripheral velocity of 13000rpm at a vapor pressure of 0.5bar at 40°C.