Saimaa University of Applied Sciences
Technology, Lappeenranta.
Mechanical Engineering and Production Technology

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Design of a Hydraulic Pump for Construction with Additive Manufacturing Tools

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Abstract

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Saimaa University of Applied Sciences
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The objective of the study was to design a hydraulic pump suitable for additive manufacturing and thereby prove both the process of 3D printing hydraulic equipment and the concept of usage of such hydraulic equipment. The design was supposed to meet an arbitrary set of requirements that was developed in cooperation with the instructor.

The study was conducted in Saimaa University of Applied Sciences and it also received material and equipment support from the laboratory and the workshop staff. Computer-aided modelling and FEA simulations were conducted with the use of SOLIDWORKS and related software products.

The results of the study were a configuration of a hydraulic system with associated machinery and components and a developed 3D model with a set of defined properties, dimensions, features that were calculated according to the requirements which were discussed in the initial phase of the work. The designed pump was prepared to be constructed and potentially used in a real practical application.

The results can be used to continue research of additively manufactured hydraulic machinery. It may be possible to rapidly construct temporary substitutions for malfunctioning or failed hydraulic components and reduce idle time.

Keywords: additive manufacturing, hydraulics, pump, 3D printing

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List of terminology

AGMA – American Gear Manufactures Association
AM – Additive manufacturing
BLDC – Brushless, Direct Current
CAD – Computer Aided Design
DC – Direct Current
FDM – Fused Deposition Modelling
FEA – Finite Element Analysis
FEM – Finite Element Method
FFF – Fused Filament Fabrication
OTI – Open to Inlet
OTIO – Open to Inlet and Outlet
OTO – Open to Outlet
PD – Positive Displacement
UAS – University of Applied Sciences

1 Introduction

There is no doubt that additive manufacturing has a good public image nowadays despite of a gradual yet predicted decrease of inflated expectations from its capabilities. It is still popular among start-ups, researches and prosumers and demand on the technology persist. As the technology seemingly attains a mature phase of its development, more often companies are going to adopt it and integrate into their business. (Linden, 2003).

The idea of this study takes its origins in an article dedicated to a project of a group of engineering students from the University of Rhode Island. They designed, constructed, and tested a stabilisation platform that would allow them to negate turbulent sea conditions and to use a 3D-printer on-board. As one the students specified, the project was directed to aid work of research ships that were located far from shores and might be need for timely replacement of any piece of equipment. (Tia Vialva. 2019)

Discussion of benefits of additive manufacturing and advancement in researches of mobile AM systems were combined with personal experience of maintenance of production equipment, including hydraulic systems, and resulted in a seemingly original and potentially competitive concept.

1.1 Goal

The goal of the project is to estimate the manufacturability of a hydraulic oil pump with usage of additive manufacturing tools that are potentially available for an advanced category of consumers, also known as prosumers. The pumps performance has to be assessed to define how practical it is. The pump should be designed with a thought in mind that it should be possible to construct it with the use of the equipment provided by Saimaa UAS. The pump should be designed in a way that it can successfully perform a predefined task to prove its applicability.

1.2 Objectives

The first objective of this study is to give an exposition into the aspects of the relevant to the project professional- and prosumer-grade additive and rapid prototyping techniques; to provide a summary of basics of pumping principles and hydraulic fluids, to describe core aspects of the designed elements of the pumping unit. The provided reading material should be sufficient for a reader of this thesis work to comprehend the following practical studies.

The second objective is to conduct a research on materials for printing and hydraulic fluids. Mechanical and physical properties of the selectable materials should fulfil requirements of the design and their chemical must comply with limitation raised by nature of hydraulic fluids and vice versa.

The third objective is to design a hydraulic pump, including the selection of the pump’s configuration, materials and hydraulic fluid, and the creation of a CAD model. The design should be done with respect to limitations due to available equipment have to be considered.

Finally, the pump has to be constructed according to the developed design and tested in laboratory conditions. For these purposes, the AM machines and other equipment of Saimaa UAS should be used.

1.3 Scope

The study is focused on the development and refinement of the theoretical basement of a potentially functional pump design. The study has the following restraints:

- Time limits of 3 months
- Financial limitation, although the university covers expenditures on manufacturing equipment, material feedstock, and other related costs.
- In-depth analysis of fluid mechanics is omitted due to excessive complexity
- Technical limitations of the equipment. A computer system in the disposal of the author is not capable of running bleeding edge simulation software

and therefore, for example, simulations of advanced morphological simulations were not considered within the study.

Importantly, while the CAD design is set to eventuate in a working prototype, detailed manufacturability and applicability to a real-life scenario are beyond the scope of the study.

1.4 Structure

The study consists of two main parts. The first part is theoretical and provides a short overview of related concepts, machinery, and technologies that would aid further reading of the further sections. The second part is practical and includes the application of designing methods from the early stages of the design process to laboratory tests.

At the end, the section “summary and discussion” provides highlights of the work, draws a conclusion to the study and considers what improvements can be done and how the topic can be further explored.

2 Theory and background of the study

Additive manufacturing is a process of joining material, as in opposite to subtractive processes such as milling, turning, etc. and formative processes like forging, stamping, etc., to make physical parts from a digital model.

The technique has its origin in rapid prototyping that was used to quickly produce a physical representation of a future product, mainly with the purpose of communicating the idea of a design in a common and easy to understand way. A great advantage of rapid prototyping against other techniques was in a shorter time of manufacturing, higher precision and automatization than, for example, making a prototype curved from wood or machined from a solid steel billet. This application stays relevant for AM today.

With the improvement of the method and overall development of the technology, the usage of AM extended from conceptual design to functional models. The AM enabled to examine not only the form but also the tolerances required for assembly purposes and how the products would work, in other words, its function. (Gibson 2010, p. 3.)

The work has the aim of constructing a hydraulic pump according to a design, developed with use of additive manufacturing and computer-aided design software. As a CAD tool, SOLIDWORKS is chosen due to it being common and well-known amidst engineering students of Saimaa UAS. Its versatility provides the required capacities to create a detailed and precise model and an integrated file management system is supplied with an option to transfer proprietary file formats, e.g. .SLDPRT, .SLDDRW, etc., into commonly used formats among which is .STL (derived from "STereoLithography") (Gibson 2010, p. 3). The printers that are provided by the university for this work conveniently utilise .STL format in preparation to printing.

2.1 AM machines

At the time this research was conducted, the lab had in its disposal 2 3D-printers: Stratasys Objet30 Prime and BCN3D Sigma (Figure 1). The following section contains descriptions of the printers and explains the core principles of the utilised printing techniques.

Figure 1. Stratasys Object30 Prime (on the left) and BCN3D Sigma (on the right) (BCN3D Technologies 2019 & Stratasys Ltd. 2016)

The work of the former is run with a help of Stratasys' proprietary technology called PolyJet which is a variation on material jetting (ISO SFS-EN ISO/ASTM 52900:2017:en, p. 29). The print head jets microparticles of a pair of photopolymeric materials onto a building platform – similarly to how inkjet printing works, as Stratasys remarked in their info brochure. The first materials – primary material – constitute a planned design and eventuates in the ready part (Figure 2). The second material is used as a support material and removed during the manufacturing process. After the materials are disposed onto the building platform, an ultraviolet curing lamp instigates the formation of connections between the particles as well as a strengthening of the structure. With the next step, the building platform descends by a certain decrement (here, the precision of the change of the height defines possible layer thickness) and the print head spreads a new layer, starting the cycle of operations again. When the part stands readily on the building platform, it can be removed and proceeded to removal of the support material. (Stratasys, 2016)

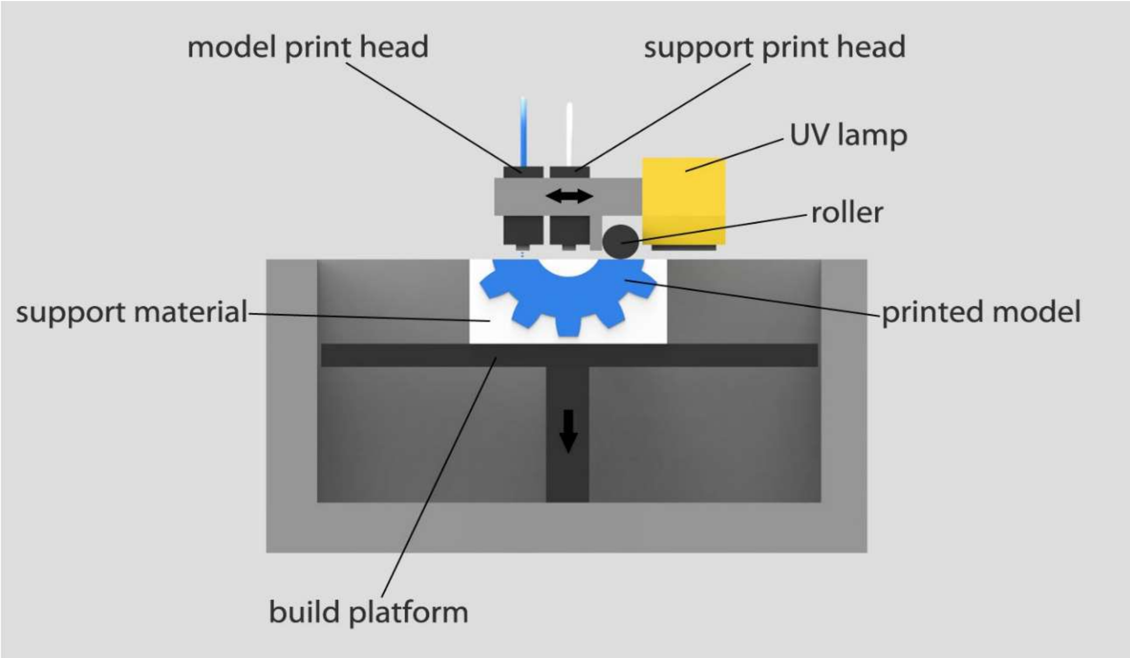


Figure 2. A scheme of Polyjet printing process (The Technology House/Sea Air Space 2019)

BCN3D Sigma, on the other hand, is an example of a printer utilizing material extrusion technique or material extrusion (also commonly referred to FDM – Fused Deposition modelling – or FFF Fused Filament Fabrication) (SFS-EN ISO/ASTM 52900:2017). Figure 3 depicts a scheme of the process. A plastic filament a) is fed through a head b) where it heats up and melts, thereby it can be controllably extruded onto the build surface (usually the last deposited layer of the part c)) or built platform e). The platform lowers down before a new layer is deposited. Support structures d) can be used to aid the construction of overhang geometries. The size of the part is defined by the dimensions of build volume f).

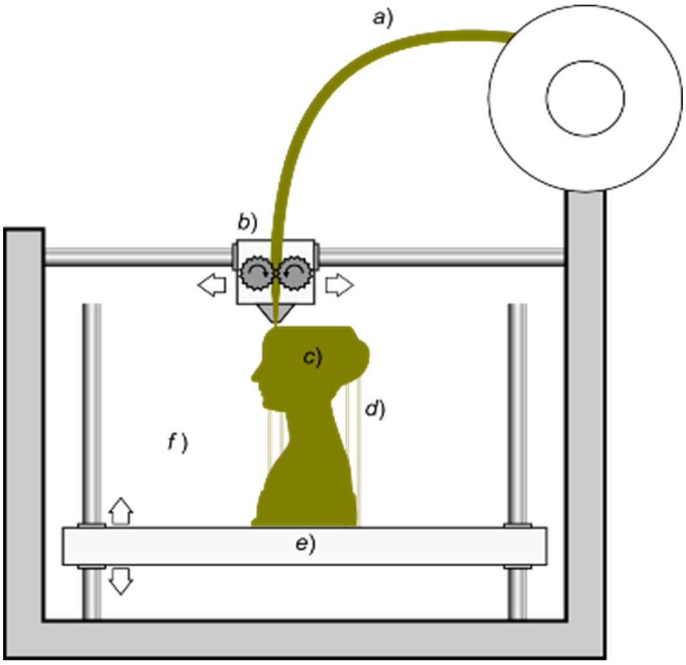


Figure 3. A schematic representation of the material extrusion process (Scopigno R. et al.. 2017.)

Modern printers utilising material extrusion process have been greatly developed with the help of RepRap programme. RepRap.org is a community project that is focused on the development of a self-replicating machine and where lots of researches, engineering professionals and mere enthusiasts shared their opensource works available for others to use (Reprap.org 2019). As an outcome, there was established a diverse and, more importantly, rather informal community of professionals, semi-professionals, and hobbyists who share a passion for 3D printing.

An important aspect of this community is that knowledge and expertise are often spread freely via the Internet – results of tests are published on forums; pioneers create tutorials and collect tips on things from how to set up a printer to advanced methods of reprograming slicing software for further optimisation for specific geometries, etc. Therefore, the most relatively reliable sources with a good reputation within the community shall fairly be considered as academically unreliable due to falling out from the conventional referential materials. To be mentioned, this issue has hindered significantly the study as proper articles on the related topics are scarcely available.

A brief comparison of the printers was written in a form of a table and listed below. All technical properties of Opject30 Prime were gathered from an official data sheet that could be found on Stratasys web-page and all technical data of Sigma were found on a corresponding web-page. A short summary of the technical characteristics is depicted in Table 1 below.

	Stratasys Object30 Prime	BCN3D Sigma
Technology	Material Jetting	Material Extrusion
Maximum build size	294x192x148.6 mm	210x297x210 mm
Accuracy	0.1 mm	0.0125 mm/mm or 0.001 mm/mm (z-axis)
Resolution	x-axis: 600 dpi (~0.04 mm), y-axis: 600 dpi, z-axis: 1600 dpi,	-
Layers height	Down to 16 μm	0,05 - 0,5 mm (depending on the nozzle diameter)
Materials	Proprietary materials produced by Stratasys Ltd.	PLA / ABS / Nylon / PET-G / TPU / PVA / Composites / Others
Advantages	Simple setup Little to low post-processing is required High surface quality High reliability High control over printing conditions in the enclosed build chamber Support materials that are dissolvable in water are available	Simple setup Little to low post-processing is required Simple maintenance Support of 3rd party materials Affordable price Two extruders may double productivity or allow parallel usage of support material
Disadvantages	Proprietary material feedstock system Difficult maintenance in case of malfunctioning Relatively high price of the machine	Noisy work Poor temperature control without an additional enclosure – problems with printing materials with high melting temperatures

Table 1. Comparison of Stratasys Opject30 Prime and BCN3D Sigma

2.2 Materials

Predominantly, thermoplastics polymers are used as materials for material extrusion. Their chemical composition, consisting of the relatively independent macromolecules, allows them to undergo multiple cycles of heating and cooling without significant damage. As a consequence, thermoplastics can be recycled which is a large advantage for purposes of additive manufacturing: drawn into coils and reeled onto spools, filaments have the right aptitude for melting, being deposited into shapes and solidifying again. (Biron 2018)

The university provided this study with a series of construction materials for both printers mentioned in the previous section. The following Table 2 depicts them with grouping by the corresponding AM machines.

StrataSys Object30 Prime	Rigid Opaque (Vero family), RGD720, RGD525
BCN3D Sigma	ABS, PVA+, Flex, Ninjaflex Sapphire, PETG, PLA, PLC

Table 2. Materials that are available for the 3D printers

Data on properties of all the aforementioned materials were collected from various sources. Predominantly, technical documentation of associated material manufactures was used as a source of the information. A combined table of the obtained data was created and used for this study. It can be found in appendices (Appendix 2).

Due to a lack of data about behaviour in oils of the available thermoplastics, a simple test was planned. The personnel of the laboratory gave their opinion that testing of materials is not reasonable since they are not chemically resistant. However, the actual possibility or, in the contrary, impossibility did not seem clear, therefore a simple test was conducted to empirically identify how the polymers would react with the oil. The details and results of the test are discussed in the later section 3.7.

2.2.1 Infill

Infill is a parameter or a measure of the presence of small voids that were deliberately added to the print-out to control the amount of used feedstock. The higher the value of infill, the less porous the printed part is. The term is relative to porosity (ISO 52900_2015). The creation of hollows volumes inside the parts helps spare the feedstock, reduces the duration of printing and weight of the part. Infill also influences the strength performance of the printed bodies. An example of infill patterns is depicted in Figure 4 below.

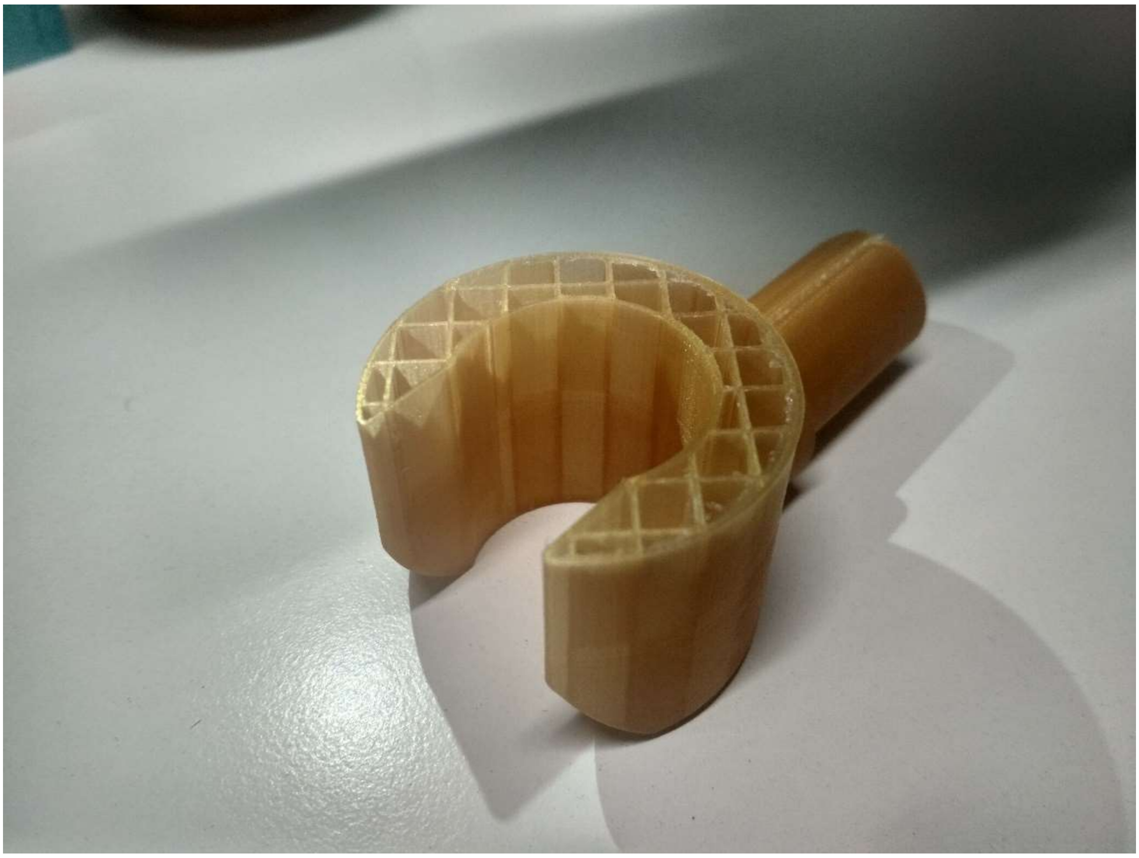


Figure 4. A printer LEGO hand with visible internal rectangular structure (Dobryakov 2019)

During the process of system set-up, users can define how dense the part should be and a slicer software generates an internal structure with voids. Typically, it is created as a checkered pattern or alike, e.g. rectangular, triangular, hexagonal (honeycomb), etc. The latter is considered the most efficient in terms of material use efficiency (as was proved by Thomas Hales (2001)) but at the same time it requires more time to construct in comparison to rectangular pattern. (Tyson 2017).

2.3 Hydraulic fluid

Hydraulic fluids play a role of the medium where power from hydraulic components is transferred. They serve as lubricants of internal moving parts, coolants preventing dangerous overheating of hydraulic machinery, and as a protection against risks of corrosion. Additionally, the hydraulic fluid may carry particles of dirt to integrated filters where they are collected and removed from the system; this prevents a risk of abrasive wearing of the machinery from the inside and prolongs its service life.

The university supported the study with a canister of hydraulic oil that had been produced by TEBOIL. Figure 5 demonstrates the label on the canister.

Figure 5. Hydraulic oil 46S that was used in the study

The markings that are displayed on the label correspond to technical standards and denote certain characteristics of the contained oil. These markings can be explained as follows.

ISO VG 46 – Kinematic viscosity of 46 centistokes (46*10-6 m2/s) ± 10%

DIN 51524 part 3 (HVLP) - lubricants have additives that protect from corrosion, oxidation and wear, plus additives increasing their viscosity index (VI >140, pressure >100 bar). They are intended for universal application; however, the biggest advantage is provided when used in external hydraulic systems. (P. Markevičius ir Ko).

ISO 6743-4 HV – according to SFS-EN 6743 (part 4), that provides a reader with a classification of Family H (Hydraulic systems) of fluids. HV designates that the refined mineral oil has improved viscosity /temperature properties along with improved anti-rust and anti-oxidation properties (HL) and improved anti-wear properties (HM). The standard also specifies that typical fields of applications for such oils are construction and marine equipment.

SS 15 5434 AV – Swedish standard. Unfortunately, no appropriate reference nor definition was found.

2.4 Hydraulic pump

The content of the following section of the study was based on the 1st chapter, 4th paragraph ("Classification of Pumps") of Pump Characteristics and Applications (2nd edit.) written by Michael Volk (pp. 1-49).

A pump is a machine that is used to move liquid in a piping system and to increase the liquids' pressure. The latter is done for multiple reasons including to overcome static elevation, friction forces in the piping system, and/or resisting pressure of a targeted pressurised vessel, and to increase the speed of the liquid.

2.4.1 Kinetic principle

In kinetic pumps, energy is added to the liquid to increase its velocity. Later, when its velocity is reduced, according to Bernoulli's principle, corresponding pressure rises up. Centrifugal pump is the prime example of utilisation of this pumping principle. (Karasik, p. 5).

2.4.2 Centrifugal pump

A centrifugal pump uses a rotary impeller as a driving component of the pump to create flow. The rotating blades of the impeller act on the liquid with centrifugal force (hence giving the type its name) transmitting it energy and moving towards the discharge side of the pimp. Centrifugal pumps are primarily used to pump liquids with low viscosity (e.g. water). Centrifugal pumps are in favour for application where high flow rate and abrasive solution compatibility is required.

2.4.3 Positive displacement

In a positive displacement (PD) pump the fluid is displaced by periodic addition of energy through direct force applying to one of the movable volumes of the fluid. The energy causes the pressure to increase until it is enough to move the fluid through the outlet.

A principle of work of a PD pump can be generally described as follows. Three elementary actions constitute the pumping sequence: Open-to-inlet (OTI), closed-to-inlet-and-outlet (CTIO), and open-to-outlet (OTO). During the operation cycle, the rotary and stationary parts of the pump act to define a volume, sealed from the outlet and open to the inlet and this volume increases as the rotor moves. After that, a part of the before-mentioned volume is sealed through the action of the rotor from the inlet, while remaining closed to the outlet. Thus, this volume is considered as CTIO. This phase of the cycle continues only for a short period. Next, the volume gets open to the outlet. For a good pumping action, OTI should grow continuously and smoothly, CTIO should remain constant, and OTO should shrink continuously and smoothly as well. Importantly, the liquid must not be in a position that is open to both outlet and inlet simultaneously (Karassik et al., p 3.80).

Reasons to choose positive displacement pumps

- The pump can tolerate liquids with high viscosity
- The pump is self-priming

- It can tolerate high pressure
- It can work even at low flow
- High efficiency
- It is able to work at low velocity
- Low shear stress
- The pump has the capability to handle fragile solids
- The accurate, repeatable flow measurements are possible
- Constant flow/variable system pressure
- It can generate a two-phase flow

2.4.4 Internal gear pump

Internal gear pumps are similar to external gear pumps. They move and pressurise liquid by meshing and un-meshing of gear teeth. With an internal gear pump, a rotor features internally cut teeth and meshes with a driven gear having external teeth that is mounted eccentrically. Figure 6 below shows a scheme of a generic internal gear pump.

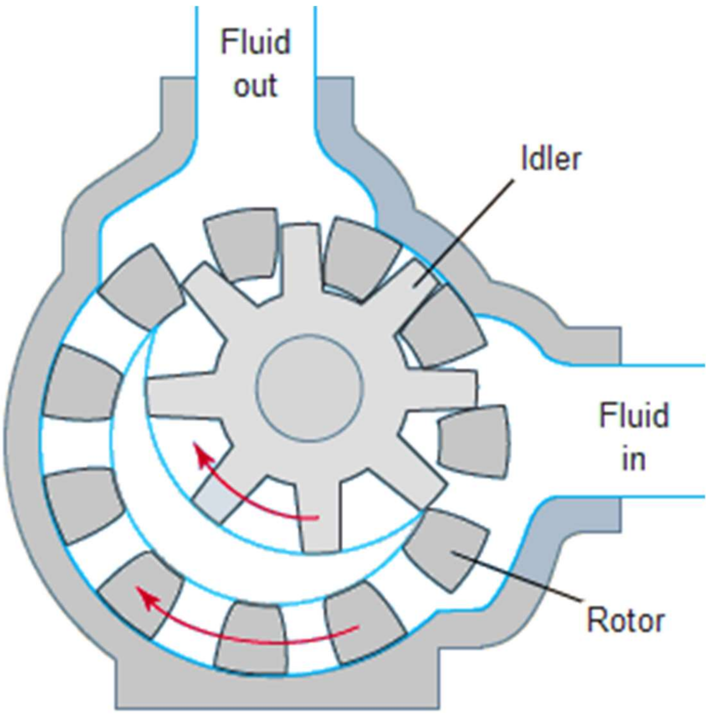


Figure 6. Internal Gear Pump – Interior-bearing type. Arrows indicate direction of flow and movement of gears (Jenner 2012)

The advantage of these types lays in the fact that it has only one seal for one protruding shaft, it is relatively inexpensive, the capability of rotating gears in either direction resulting in a reversible flow. Among disadvantages, there is an issue of one bearing submerged in the pump liquid and exposed to the effects of the latter, a requirement to support overhang load, and the inability to pump liquid containing abrasives or solids in most cases.

2.4.5 Rotary lobe pump

A rotary lobe pump is a PD pump. This pump's working principle resembles an external gear pump: the liquid flows between the rotor lobe surfaces which almost touch each other as they rotate. This provides continuous sealing and resembles the respective aspect of external gear pumps. What makes it different, is that one lobe cannot be driven by the other one and therefore the pump requires an additional power train to supply torque directly to the lobes.

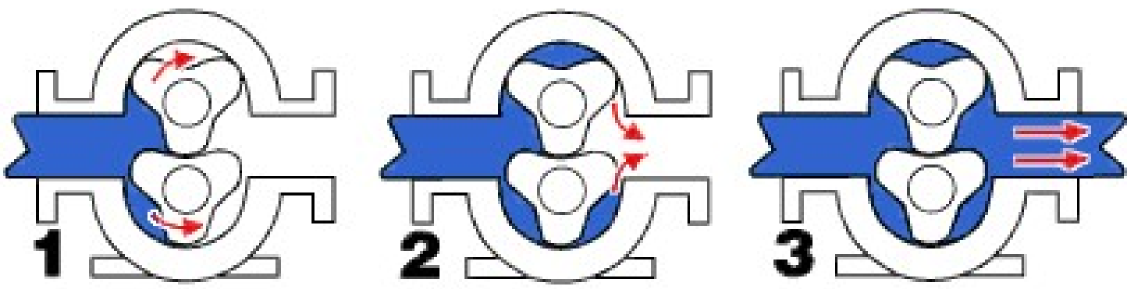


Figure 7. Scheme of a generic rotary lobe pump: 1. the liquid is flowing into the cavities that are created by the rotating lobes, 2. the liquid is moving in the cavities between the lobes and the casing of the pump to the outlet side of the pump, 3. the liquid is continuously flowing through the outlet port (Pump School 2017)

Due to the slow rotation speeds and consistent clearances between the lobes they are mostly compatible with viscous liquids and have found use in food processing for liquids containing solids or particles that have to be preserved. This type of pumps is subjected to pressure pulsation and also shows high slip when used with low-viscosity liquids. The pump's design is challenged with the requirements implementing timing gears.

2.4.6 Screw pump

A progressive cavity pump features a single-threaded screw, also called a rotor, that turns inside a double-threaded stator that is made of elastomeric material. The screw and the stator have an interference fit. Rotation of the screw causes cavities to form at the suction end of the stator; as the screw continues its rotation another cavity opens further along the axis of rotation and the previous one closes in. This axial progression of cavities goes from one end of the stator to the other and the liquid that is trapped in the formed cavities is pumped (an example is shown in the following Figure 8).

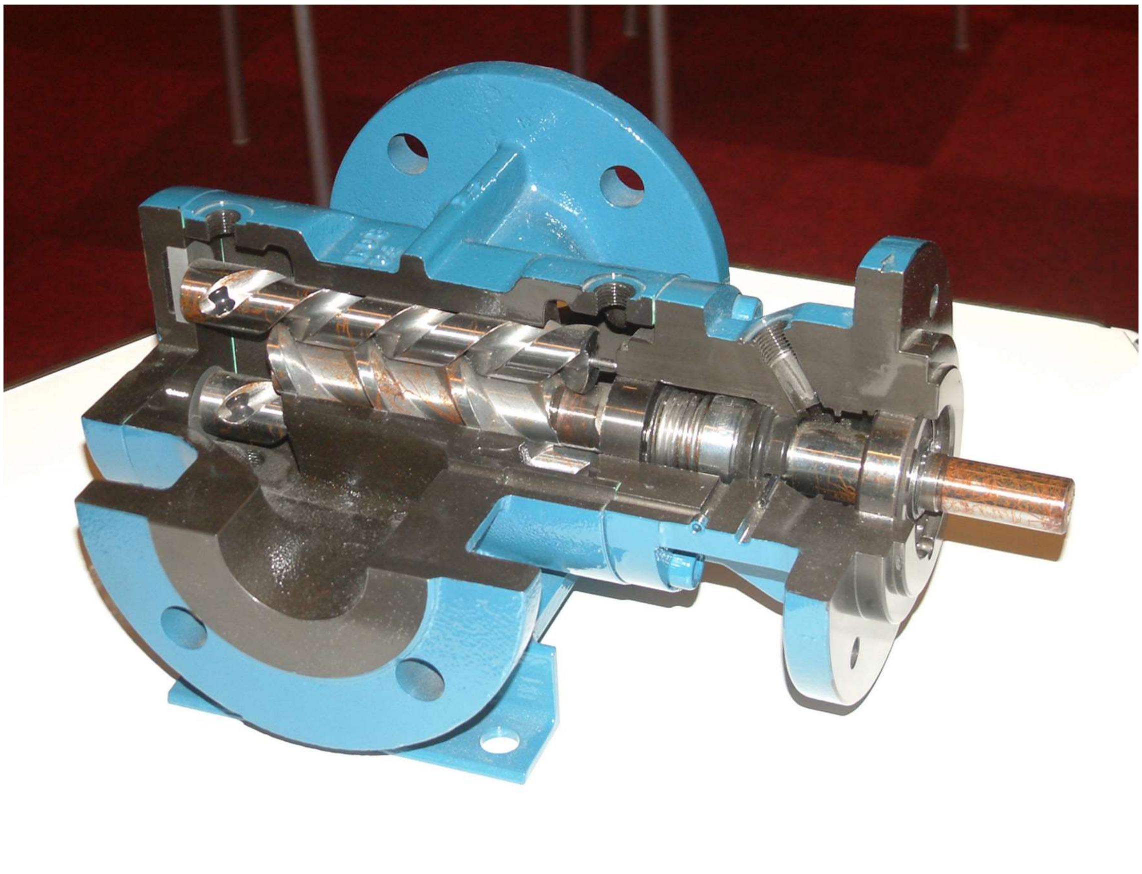


Figure 8. Photo of a crew pump (S.J. de Waard 2019)

Advantages of progressive cavity pumps include an ability to pump liquids with high viscosity, fragile solids and abrasives, little effect of pulsation. Disadvantages of the type are: it typically requires a lot of floor space (large length), relatively high cost of parts with one needs to be replaced, and that the pump requires high torque to start (due to its length) and thus the motor is large and more expensive than in other types of pumps of a similar performance.

2.4.7 Piston pump

A piston pump is a type PD pump. A reciprocating piston with a high-pressure seal extracts and retracts along a pressure chamber. It can be used to pump either liquids and gases and is preferred in an application where linearity with sleep is important. A system of linkages connecting the drive with the piston and the seal should be of concern when designing such pump. A simple scheme of a generic piston pump is given in Figure 9 below.

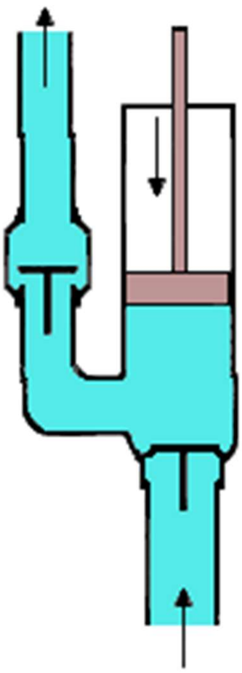


Figure 9. Scheme of a generic piston pump (KVDP 2010) (Edited)

2.4.8 External Gera Pump

For this study a decision was made that the design would be based around a type of external gear pumps. Such a decision was supported by an argument, that this type is relatively simple in its construction and in the mechanics of the pumping process. That would allow for a deeper focus on manufacturing and material selection aspect of the work than on dealing with fluid mechanics.

Advantages of external gear pumps include the fact that they operate at relatively high speeds, and output pressure is relatively high and, since it belongs to a category of positive displacement pumps, outlet flow rate stays constant. The work of the pump is free of significant noise and it is almost free of pulsation.

A disadvantage of external gear pumps is that they usually require instalment of 4 bearings or bushings to support rotation of the gear shafts and by that increasing the number of components. Another significant downside of the pumps is that they are not suitable for application where pumped liquids contain abrasives due to the risk of disrupting tight internal clearances unless a timing gear is used to negate the risk.

2.5 Working principle of external gear pump

Three elementary actions constitute the pumping sequence: Open-to-inlet (OTI), closed-to-inlet-and-outlet (CTIO), and open-to-outlet (OTO). During the operation cycle, the rotary and stationary parts of the pump act to define a volume, sealed from the outlet and open to the inlet and this volume increases as the rotor moves. After that, a part of the before-mentioned volume is sealed through the action of the rotor from the inlet, while remaining closed to the outlet. Thus, this volume is considered as CTIO. This phase of the cycle continues only for a short period. Next, the volume gets open to the outlet. For a good pumping action, OTI should grow continuously and smoothly, CTIO should remain constant, and OTO should shrink continuously and smoothly as well. Importantly, the liquid must not be in a position that is open to both outlet and inlet simultaneously (Karassik et al., p 3.80).

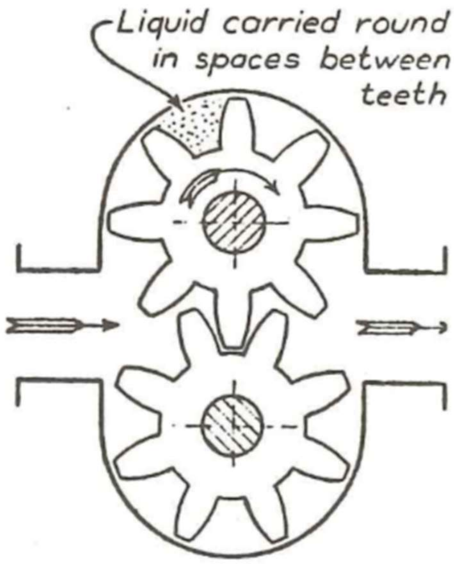


Figure 10. Scheme of a generic external gear pump (Trade & Technical Press 1974, p. 34)

In external gear pumps when the gears come out of mesh, they create expanding OTI volume. The liquid flows into opened volumes between the teeth and gets trapped there as the gears continue to rotate. Liquid travels around the outer radii of the gears in the CTIO volumes between the teeth and the casing. Liquid eventually leaves the meshing in OTO volume and is forced through the outlet port under pressure.

As was mentioned in the forgoing section, the work of external gear pumps is characterised by positive displacement. In a positive displacement pump, the fluid is displaced by periodic addition of energy through directly applying force to one of the movable volumes of the fluid. The energy causes the pressure to increase until it is enough to move the fluid through the outlet. (Volk 2005, pp. 5-7.)

The type is called *external* gear pump as its gears are mounted on two separate shafts and interlocked with each other. It is different from a related type of *internal* gear shafts, where one, smaller gear is interlocked with other gears and rotates inside of it. (Volk 2005, pp. 8-9.)

2.6 Components of external gear pump

2.6.1 Body

A body typically consists of a casing, a front plate and a back (rear) plate. It serves as housing and protection of internal components and keeps the pumping process enclosed, maintains internal pressure.

A casing of a pump is a base component of the assembly. It typically includes a hollow pumping chamber, inlet and outlet ports, holes for fasteners, and can have sockets for sealing components, e.g. gaskets. The back plate is a relatively plain component that has holes for fasteners, sockets for bearings, support shafts and seals, optional support elements, and sealing sockets in some design configurations. A front plate includes all aforementioned for a black plate, port for a rotor shaft, and additionally may feature integrated glands as an interface between the pump and an attached motor.

2.6.2 Gears

This section is primarily based on the content of *Pump characteristics and applications* by Michael Volk that is covered in pages 29-35.

As an element of pump design, gears play a role of rotors transferring mechanical power directly to the pumped liquid. In general, in external gear pumps gears move the liquid at the inlet side, move it in a cavity between the teeth and leave at the outlet side.

Spur gears have teeth parallel to the axis of their rotation. They are generally used to transmit motion from one shaft to a parallel shaft and this function remains in the pumps as well. Helical gears differ from spur gears by having teeth which are inclined to the axis of rotation. Both types are illustrated in Figure 11 below.

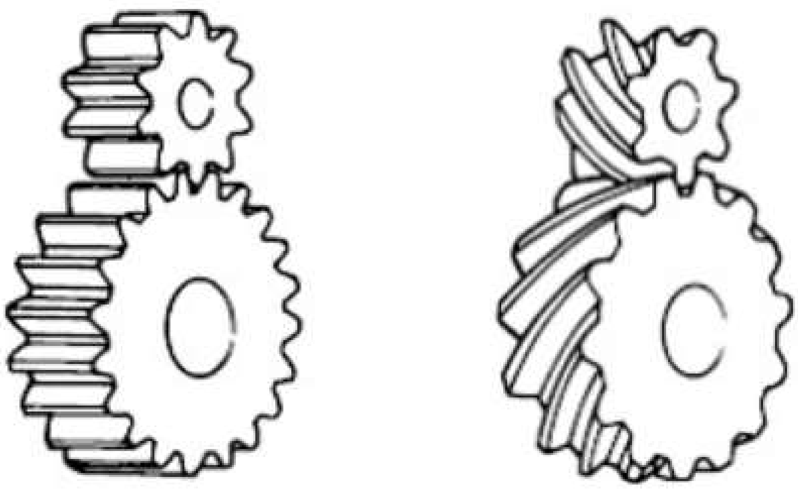


Figure 11. A sketch of spur and helical gears (*Kohara Gear Industry Co. 2015*)

Pumps with spur gear are generally louder than their counter-parts with helical or herringbone gears due to the protruding noise cause by “squeezing” of the fluid between the meshing gears. Some manufacturers add relief slots around the point of meshing to give the fluid space to flow in. This measure reduces the produced noise and increases the pump’s efficiency.

An additional advantage of spur gears is that only minimum axial thrust is inflicted to them.

When using helical gears, the geometry of the gears itself guides the fluid from trapped positions and eliminates a need in additional slots. A typical configuration with helical gears is efficient and quiet. Nevertheless, as the teeth are inclined relative to the axis of rotation, even radially directed loads are deflected and obtain components parallel to the axis and these gears suffer from axial forces against the pump thrust bushings. Consequently, axial wear leads to a swift decrease in performance and it is highly important that tight axial clearances are maintained. Due to this reason, helical gears also require corresponding bearing, capable of carrying axial loads.

Herringbone gears are claimed to be the most efficient but at the same time the most difficult to manufacture. They are also quiet and are less sensitive to axial loads. The complexity of the design could make the type a good candidate for AM design as one of its benefits is that almost every geometry is possible with quite similar ease (Rosen 2014).

The following Table 3 provides a comparison of spur and helical gears (in the next page).

In spur gear, the teeth are parallel to the axis of the gear.	In helical gear, teeth are inclined at an angle (called helix angle) with the gear axis.
Spur gear imposes only radial load on bearings.	Helical gear imposes radial load and axial thrust load on bearings.
Radial bearings that can handle radial load only can be employed here.	Bearings that can handle both radial and axial load must be employed here.
Spur gear can be used for parallel shafts only.	Helical gears can be used for parallel shafts or crossed shafts.
At the time of engagement, the entire face of the tooth comes in contact with the mating tooth.	Here engagement initiates with a point on tooth face and gradually extends across the tooth.
Mating teeth are subjected to impact loading.	Load on the mating teeth builds up gradually.
Sudden application of load increases vibration, especially in high speed condition.	Gradual loading on teeth minimizes vibration.
Spur gear produces noticeable noise due to sudden loading and vibration.	Operation of helical gear is noise free, even at high speed.
Teeth of the spur gear can be cut easily as it requires only two-dimensional motion.	Making teeth for helical gear is comparatively difficult as it requires three-dimensional motion.

Table 3. Comparison of spur and helical gears

2.6.3 Bearings

The primary purpose of bearings is to maintain centricity of the running clearance within practical limits. Bearings hold rotating components and transfer axial and radial loads from the source of the load to the supporting structures. In a case of a gear pump, bearings are used to support gear shafts and carry radial loads applied to them.

The practice of manufacturing of plastic molded bearing is rather common and such products are available in the market, meaning that it is potentially possible to additively manufacture them with proper materials. Considering that one of the main advantages of AM lays in disregarding geometrical complexities, bearings seem to be a good candidate for 3D printing. However, there are high requirements for tolerances and surface finish that ultimately eliminates the option of using laboratory-manufactured bearings in the pump design.

2.6.4 Sealing components

A commonly used sealing component is an O-ring. An O-ring is an object in a form of a torus, typically made of an elastomeric material. A simple O-ring seal consists of an O-ring itself and a groove/gland/cavity to contain the former. The principle of the seal is that under the effect of supplied mechanical pressure (or a squeeze) and positive pressure of the hydraulic system the elastomeric material yields and deforms, taking a shape of the glands geometry and filling up imperfections of the confining the O-ring materials. (Volk 2005, pp328-333.)

Because the system pressure assists the sealing, it leads to a relation: the higher the pressure inflicted on the seal, the more effective it is. However, it remains true only until the mechanical limits of the elastomeric material are exceeded. From then on, the O-ring will be extruded into cavities between matting cavities, expanding them and increasing the clearance. (Volk 2005, pp328-333.)

For the designed pump, the pressure was not expected to get close to the excessive level when the seal could fail. A bigger risk was associated with a lack of effective pressure which would cause penetrability of the seal.

The following limitations should be considered: high temperatures, high frictional speeds, cylinder ports, excessive clearance.

Another example of simple sealants is gaskets. A gasket is a mechanical seal that is cut in the form of the socket where it is placed and compressed between two mating surfaces to prevent leakages. Gaskets are usually made of elastic materials meaning that it allows some degree of deformation which is desirable to completely feel the gap between the surfaces. Different shapes of gaskets that are specific for an application can be cut from material sheets.

2.6.5 Motor

In hydraulics, motors are usually used as a power source. Typically run on electric power, motors transmit rotational motion to rotary components of the pumps.

For the purpose of this study, an electric motor was required and had to be ordered. A short description and comparison of the most common types of electric motors are given in this section. The content is based on reference materials prepared and provided by Groschopp, Inc.

AC motors run on alternating current that can be found in ordinary wall outlets. This property eliminates the need in a DC power supply or a rectifier. The motor uses the advantage of alternating current to do the commutation – a change in direction of the electric current and consequently in a direction of respective forces - and does not need brushes. There are many types of AC motors: the most wide-spread were examined.

Single phase induction motors utilize a single-phase oscillating magnetic field to induce torque on the rotor. Because the current is in one phase it cannot pull the magnetic rotor off its balance and make it rotate. It leads to a need in a capacitor to create a phase shift in starter wiring. Three-phase motor in return uses magnetic fields in multiple phases and manufactured with internal wiring that is pre-setup for the duty.

A DC motor, in contrast to its AC counterpart, is run on direct current. DC does not run in ordinary wall outlets and therefore needs to be obtained through additional devices, for instance, a rectifier. A rectifier converts AC power to DC.

To control a DC motor, there are a few options. For the first, a transformer is required to change an input voltage, thereby changing the speed of the DC motor. Alternatively, there is an option of using a DC speed controller that contains an armature with a controllable value of resistance. These are relatively simple way to control a motor. It can also be operated with a circuit with a PLC and relays – this method would involve more equipment and a work on PLC programing, which can be in fairly quickly if the method is known to the programmer.

A brief comparison of the motor types is given in Table 4 below. It is important to mention, that the comparison is not definitive and covers only relatively basic aspects of the motors and their work which are enough within the scope of this study.

Type	Advantages	Disadvantages
Three-phase AC motor	Good speed control (with controllers), compatible with speed controllers, has wider range of speeds than a single phase motor (with controllers)	Low efficiency, additional external devices are required to change the motor's speed, relatively high cost per unit of power
Single-phase AC motor	Long life with low maintenance, good speed maintenance over a wide range of torque, low starting current, compatible with gear boxes	Low efficiency (40%-70%), poor speed regulation, relatively high cost per unit of energy, the motor needs a capacitor to start
DC motor	Efficiency of 60-75%, compatible with gearboxes, can work with DC power without controllers, relatively cheap average acquisition cost	Depend on maintenance (mainly because of failures of brushes), cogging at low speeds (less than 300 rpm) may occur, full-wave rectified voltage lead to high power losses
Brushless DC motor	Long life, low need in maintenance, produces a little noise, gearbox compatibility, high power density	High price (mainly because of rare-earth metals that are used in the motor's construction)

Table 4. Comparison of different types of electric motors (Groschopp)

2.7 Slip

Also sometimes referred to as internal leakage, slip is the amount of pumped liquid that leaks from the OTO volume to the OTI volume per unit of the time. Generally, slip is caused by differential pressure between the OTO and the OTI volumes and its value or extensiveness depends on the differential pressure, geometrical clearance between the rotors and stationary components, and the viscosity index of the pumped fluid. (Karassik, p 3.140)

3 Pump design

When the original concept was generated and began to gain its details, a need for visual demonstration appeared. A layout drawing was prepared to depict the key components of a future pump and how they would be related to each other in space, define the direction in which the project would be developed, and to get a better idea of the general dimensions. The layout drawing that was approved by the instructor can be found in Appendix 1.

3.1 Assumptions

The designed pump had to be configured to fulfil an arbitrary goal in a performance test. The conditions for it were discussed with the instructor and set as represented in Table 5 below. The idea of the application was centered around an extracting piston that would have to move a load (expressed as force F) with a use of pressure being created by a pumped liquid.

Diameter D (m)	Load F (N)	Power P (W)	RPM
0.04	600	63	700

Table 5. Initial assumptions of several parameters of the hydraulic system

In the table*, Diameter* corresponded to a hydraulic piston's diameter to which the pump is connected and where the hydraulic fluid would be pumped to. *Load* stands for the force that is applied at the piston and is meant to be counteracted by the generated flow. *Power* is the rated power output of the attached electric motor transmitting torque to the driving gear. *PRM* stands for the assumed frequency of rotations of the motor. All the values except for RPM were not purely arbitrary but rather selected according to what equipment was available or could be acquired.

From these values, a series of parameters could be derived such as piston's area, created pressure and flow rate, and rotor's angular velocity. The values are given in Table 6 below.

Piston's area A (m^2)	Pressure p (Pa)	Flow rate Q (m^3/s)	Flow rate Q (l/min)	Angular velocity ω (rad/s)
0,00126	477464,8	0,00013	7,88	73,30

Table 6. Calculated parameters for the hydraulic system

The values from the table # above were calculated with a use of formulae listed below (Mäkelä 2009, pp. 18, 99). The formula 4 was used to calculate flow rate in more conventional unit, namely l/min. The formulae were adopted from *Mechanical and metal trades handbook (Fischer et al 2006, pp. 102-110).* The

$$A = \frac{\pi D^2}{4} \qquad (1)$$

$$p = \frac{F}{A} \qquad (2)$$

$$Q = P/p \qquad (3)$$

$$Q = \frac{P * 60000}{p} \qquad (4)$$

$$\omega = \frac{2 * \pi * RPM}{60} \qquad (5)$$

3.2 Load calculation

In order for a pump to work, an attached motor transmits torque to the moving elements of the pump (i.e. a rotor). In the case of an external gear pump, it is a driving gear mounted on a shaft, which may be connected to the motor in several ways or protrude from the motor itself. The torque causes the shaft and, consequently, the gear to rotate at a specific speed and can be calculated with a formula that is given below. (Budynas & Nisbett 2011, p. 706).

$$T = \frac{P}{\omega} \qquad (6)$$

Where P – power (W), T – torque (Nm), and ω – angular velocity (rad/s).

The driving gear mates with another, driven gear and the effect of the torque are transferred to the latter through a contact force at a point of contact. In Figure 12 (see below) the force transmitting the torque and the reaction force of the driven gear are denoted as F_{12} and F_{21} respectively and the subscripts marking the directions of the forces (e.g. F_{12} acts from the gear 1 to the gear 2). Each force is further balanced by a reaction forces of a shaft (F_{a1} & F_{b2}) on which the gears are mounted; all forces are resolved, and the sum of all forces is 0.((Budynas & Nisbett 2011, p. 706).

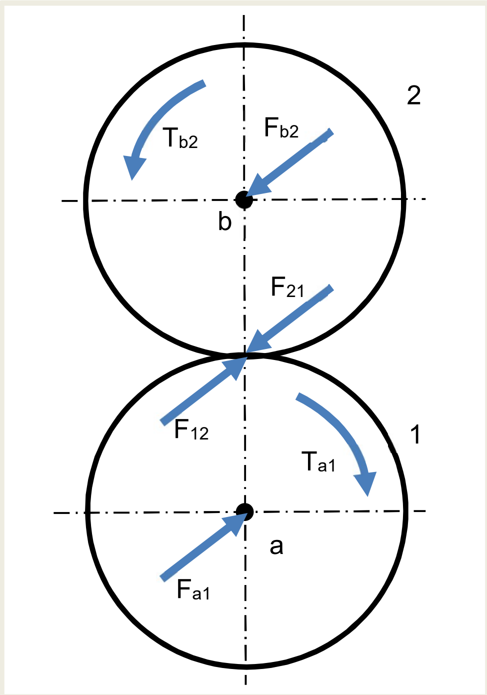


Figure 12. Free Body Diagram of the gears. The driving gear (1), the driven gear (2), the shaft to the motor (a), the supporting shaft for the driven gear (b). “F”s denote forces and “T”s denote torque

Quantitatively, the transmitted force or load can be found with a Formula 7 or alternatively with formula 8 below.

$$W_t = \frac{60000P}{\pi d n} \qquad (7)$$

Where W_t – transmitted force (N), P – power (W), d – gear diameter (mm), and n – rotation speed (rev/min).

$$W_t = \frac{P}{v} \qquad (8)$$

Where W_t – transmitted force (N), P – power (W), and v – pitch-line velocity (m/s).

Using the formulae 6 and 7 the values of torque and corresponding loads were defined. The results of the calculations are presented in the following Table 7.

	Torque (N/m)	Gear diameter (mm)	Transmitted load W_t (N)
Config. 1	0.85	40.0	42.74
Config. 2	0.85	40.0	42.74
Config. 3	0.85	20.0	85.49

Table 7. Results of calculations of torque and loads for different configurations of gears

The obtained values of transmitted loads or, in other words, loads between mating gears were later used to evaluate stresses that occurred in the gears.

3.3 Flow rate calculation

The flow rate (or capacity) of a rotary pump is the quantity of fluid delivered by the pump per the unit of time. For incompressible liquids, the flow rate equals the total volume that is displaced by the pump per the unit of time excluding the slip. (Karassik, 3.140). The value of slip would be considered in calculations in the future section.

$$Q_c = Q_d - Q_s \qquad (9)$$

where Q_c – flow rate (capacity) (l/min), Q_d – theoretical or geometrical displacement (l/min), and Q_c – slip (l/min)

Displacement is a function of the geometry of the gear and time. Consider the former only. The theoretical displacement per revolution q is the volume of liquid kept in a volume created between a pair of the gears' teeth and internal geometry of the pump's body. To calculate a required displacement for the pump the following formula 10 was used:

$$q = \frac{Q}{RPM} \qquad (10)$$

Therefore, a required displacement of the pump would be 0,112*10^{-6} m^3 or 1,12 l.

Displacement can also be calculated based on the geometry of the gear.

In their handbook, J. Karassik and his co-authors suggested the following formula 11 (p. 3.141) to determine the value of the gear pump displacement per revolution.

$$2 * \frac{1}{2} * \frac{\pi}{4} * (OD^2 - ID^2) * W = q \qquad (11)$$

where OD – outside diameter (mm), ID – inner diameter of bore diameter (mm), and W – width of the gear (mm)

This allowed to proceed with the further calculation of the gears' geometry.

3.3.1 Gear mesh design

The following section describes the method by which the geometry of the gears was defined and the displacement of the gears was calculated. As follows, the section also describes details of gear profile design and how values important for hydraulics were acquired.

To begin with, for the purposes of this project the choice was made to use spur gears. Their primary advantage is relative geometrical simplicity that greatly aids the study and allows to avoid unnecessary time investments on deep researching. Following CAD modeling and FEA benefited from the relatively uncomplicated design as well. Spur gears would also simplify the further selection of load carrying bearings.

If rearranged, the formula 11 may be presented as shown below (Formula 12) and can be used to calculate the value of the width of the gears.

$$W = \frac{4 * q}{\pi * (OD^2 - ID^2)} \qquad (12)$$

Such a method, although simple and effective, might have been too generalised and significantly lacking precision. To define the required face width of a gear, it was decided to take an approach from defining a precise volume between the gear teeth and the sides of the pump's body during the OTIO step (or in other words, a displacement per a pair of teeth) with a help of CAD software. If the volume is known, the width can be calculated with respect to the required displacement that was calculated in the precious section (see pp 33-34).

To start with, arbitrary values of defining parameters – reference pressure angle, number of teeth, and reference diameter - were chosen. Using a general guidance for spur gear (Kohara Gear Industry 2015), the other values were defined with an exception of tooth depth and corresponding dedendum which were intentionally decreased for smaller gaps between meshing teeth: typically, addendum (height of the tooth above the pitch circle) is a little smaller than dedendum (height of the tooth below the pitch circle) but in this design the two are equal. Table 8 below demonstrates the calculated properties for each configuration.

Item	Config. 1	Config. 2	Config. 3
Module	4	5	2
Reference Pressure Angle (˚)	20	20	20
Number of Teeth	10	8	10
Centre Distance (mm)	40	40	20
Reference Diameter (mm)	40	40	20
Base Diameter (mm)	37.59	37.59	18.79
Addendum (mm)	4	5	2
Tooth Depth (mm)	8	10	4
Tip Diameter (mm)	48	50	24
Root Diameter (mm)	32	30	16

Table 8. Configurations of the gears

Note-worthily, configuration 1 and 2 share identical pitch or reference diameters; hence there are fewer variations to the overall pump's design between the two than the 3rd configuration demands.

To define the volume that is displaced between one pair of teeth (which would be equal to a half of OTIO volume), a CAD model was required to be created and assessed.

For the purpose of creating the CAD model, SOLIDWORKS was used. The gear had to be modelled and tested. The dimensions for the test were based on one of the proposed configurations, namely, on the 2nd configuration with 10 teeth and a reference diameter of 40 mm. The sketch of its tooth was based on an involute arc which is commonly used nowadays in gear meshing. The involute profile can be seen in Figure 13 below.

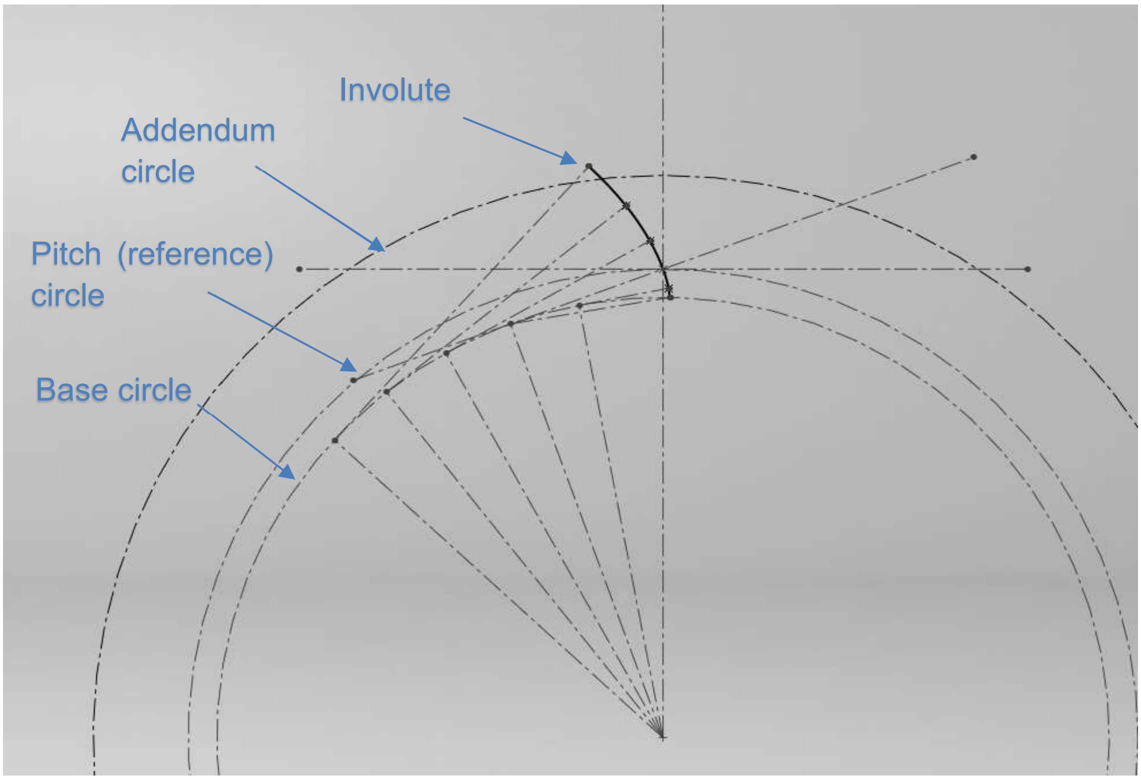


Figure 13. Construction of involute arc for a gear profile

The involute profile allows the meshing gears to produce constant angular velocity that in return leads to positive (constant) displacement. As the meshing teeth push against one other, the point of their contact occurs where their surfaces are tangent to each other. Remarkably, the tangential force is induced along a common normal that is also known as an *action line.* (Budynas et al. 2011, p.679 and p.775)

The root geometry or the flank of the teeth was lightly modified if compared to a typical one with a reciprocated truncated symmetric involute profile to minimise clearance of the meshing teeth during all the phases. The design of the tooth profile can be seen in Figure 14 below.

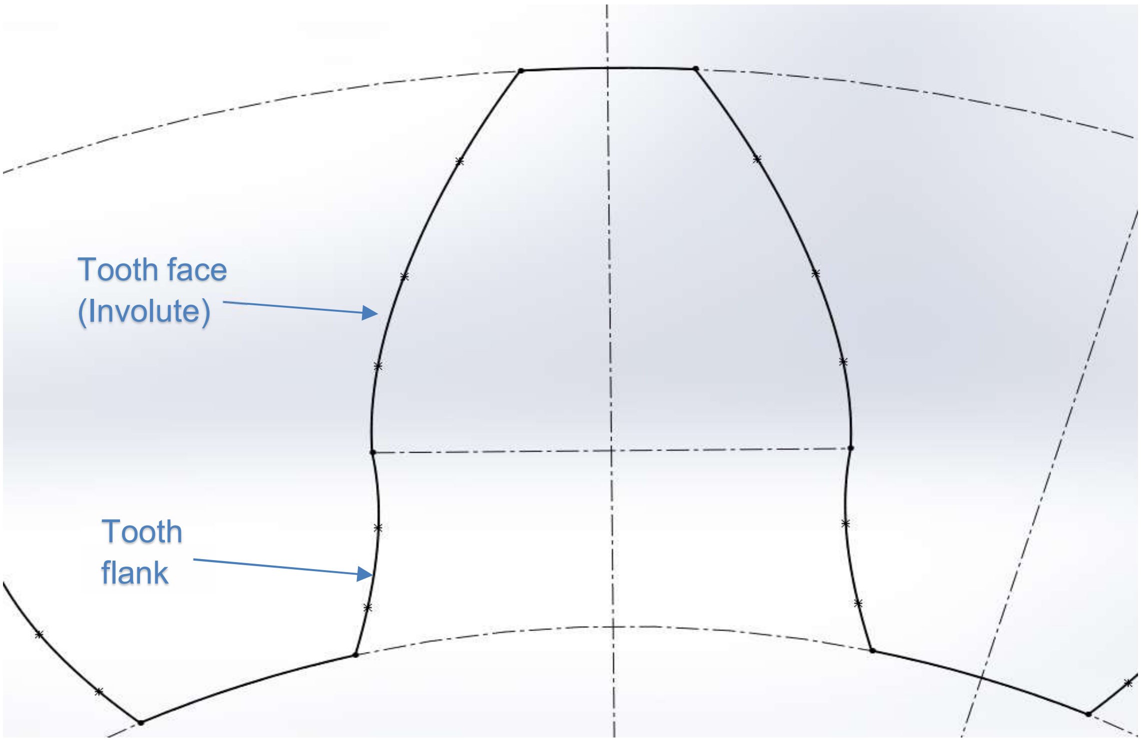


Figure 14. Teeth profile

The profile was copied to match the declared parameters and later the gear was extruded and tested with a simple motion study. After that, the tooth profile was masked to derive a sketch of the space between the teeth (Figure 15 below). The area of the body was then evaluated with a built-in measuring tool.

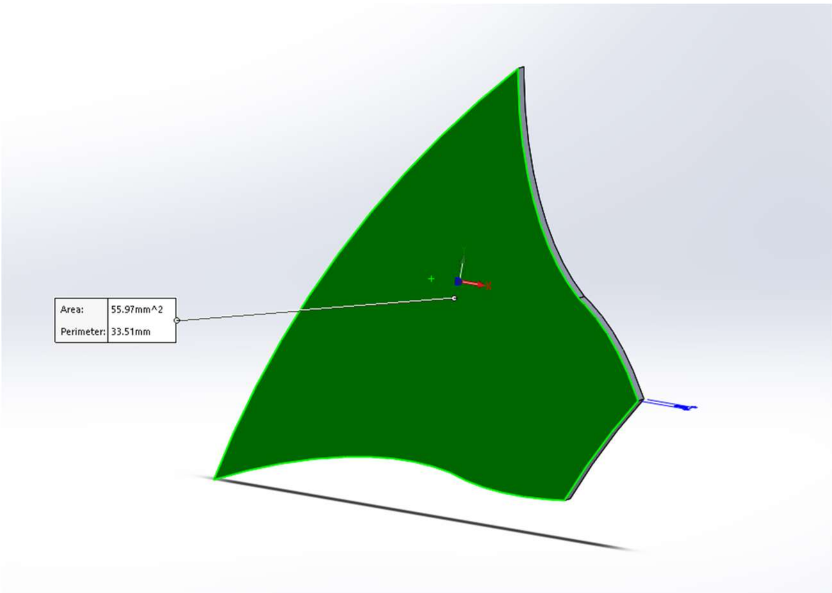


Figure 15. Area of the space between the teeth and the body or per tooth displacement. The text in the data box as follows: “Area: 55.97 mm^2; perimeter: 33.51 mm”

When the area was defined, the required tooth face width could be calculated by the formula 13 below.

$$W = \frac{q}{A * N} \qquad (13)$$

where q – displacement (m^3), A – are of the space between the gear teeth (m^2), and N – number of teeth

The same procedure was repeated for the rest 2 configurations and the respective values of the width were calculated. The results are presented in Table 9 below. Note that the face width is given in mm.

	Configuration 1	Configuration 2	Configuration3
q (m^3)	0,112*10^{-6}		
N	8	10	10
A (m^2)	8,91*10^{-5}	5,60*10^{-5}	1,4*10^{-5}
W (mm)	15,78	20,10	80,47

Table 9. Calculations of face width for the gears

The calculated value would be sufficient for meeting the set requirement for flowrate if the effect of internal leakages was neglected. Otherwise, slip had to be considered. A further section # covers slip in more details but at this point, it is only relevant to note that an assumption was made that the internal leakages of the pump would be equal to about 20% and, consequently, slip factor as 0,8 (Formula 19) of the calculated geometrical displacement. Therefore, to compensate the losses, the face width of the gears had to be recalculated with slip factor taken into account (Formula 14)

$$W = \frac{q}{A * N * \sigma} \qquad (14)$$

After the new series of calculations was conducted, the updated values of face width were as listed in Table 10 below.

	Configuration 1	Configuration 2	Configuration 3
Face width (mm)	19,73	25,12	100,59

Table 10. Face width with the effect of slip considered

3.3.2 Theoretical stresses

Contact stresses are usually the limiting factor in gear design. Contact between two meshing gears occurs along the action line, tangent to the gears' base diameters. According to American Gear Manufacturers Association (from now on AGMA), the gear work stresses can be defined by the following equation (see 15):

$$\sigma = W^t K_o K_v K_s \frac{1}{bm_t} \frac{K_H K_B}{Y_J} \qquad (15)$$

Where W^t is the tangential transmitted load (N), K_o is the overload factor, K_v is the dynamic factor, K_s is the size factor, P_d is the transverse diametral pitch, b is the face width of the narrower member (mm), K_H is the load-thickness factor, Y_J is the geometry factor for bending strength (which includes root fillet stress-concentration factor K_f), m_t is the transverse metric module.

The fundamental equation of contact stress (16) is

$$\sigma = Z_E \sqrt{W^t K_o K_v K_s \frac{K_H}{d_{w1} b} \frac{Z_R}{Z_I}} \qquad (16)$$

where Z_E is an elastic coefficient (N/mm^2), Z_R is the surface condition factor, d_{w1} is the pitch diameter of the driven gear (mm), Z_I is the geometry factor for pitting resistance.

The issue with the formulae 15 and 16 is that they contain multiple factors that are based on laboratory tests and measurements that provided reference tables for specific materials. (Budynas et al. 2011, pp. 746-747). In order to simplify calculations an alternative approach needed to be found.

In a technical handbook about gear stresses and vibrations, written by J. Smith (1983), the author had described a simplistic yet relatively rough approach through using Mean Hertzian contract stress. Xiaoyin Zhu gave a short guidance about Hertz's stress in his publication (2012). In Hertz's theory of contact, the focus is on contacts where tension forces are not allowed to occur in the area of contact. The following assumptions had to be made before proceeding with the method:

- Associated strains lay within the elastic limits of the involved materials
- The area of contact is small in relation to the entire body's geometry
- The contact surfaces are continuous
- The bodies are in frictionless contact

The first point was assumed to be true for the examined case although the strains had not been defined yet. The fourth assumption would be applicable to a conventional gear with precise surface roughness and geometry but the specific of this makes the quality controversial. Conventionally, the last assumption was made.

Figure 16 below depicts the action line EF with contact happening over the distance between points C and D. At some point X radii of curvature of the gear teeth are XE and XF.

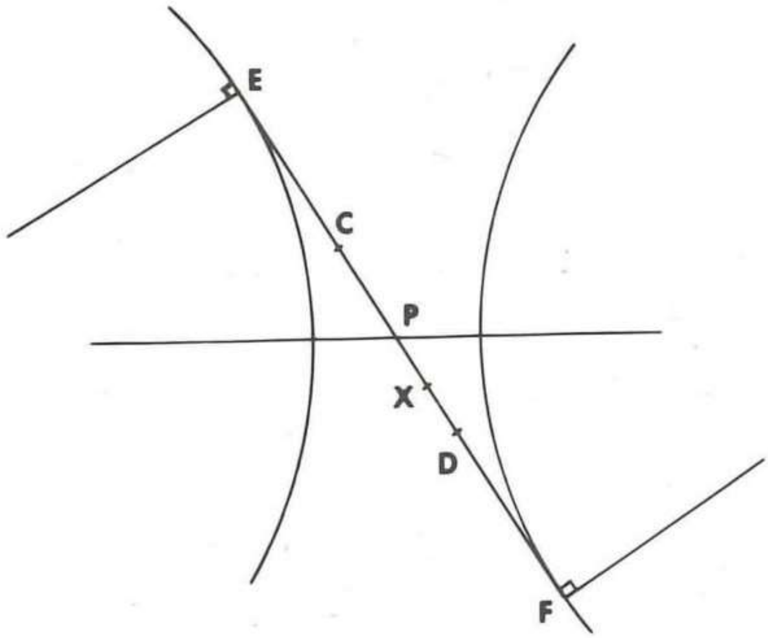


Figure 16. Radii of curvature of involutes (Smith 1983, p. 12)

Mean Hertzian contact stress can be calculated through the following formula. (Smith 1983)

$$\bar{q} = \frac{1}{4} * \left(\frac{P'\pi}{R} * \frac{E}{2(1-\nu^2)}\right)^{\frac{1}{2}} \qquad (17)$$

where E – Young Modulus, v – Poissons ration, and P' – the load per unit width, R – the effective radius of curvature at the contact.

R can be defined by 1/R =1/XE + 1/XF and is maximum when the contact between meshing teeth happens in the middle of the line EF or in other words in the pitch point, where the pitch circles of two meshed gears are tangent.

As in the examined application, the gears are identical, XE was equal to XF (further, both were replaced by X). That leads to the following definition of R (Formula 18).

$$R = \frac{XE * XF}{XE + XF} = \frac{X^2}{2X} = X/2 \qquad (18)$$

Where X is a half of the action line. The length of the former was evaluated with the use of SOLIDWORKS. Figure 17 below demonstrates the result for the 2nd configuration. Similar procedures were repeated for the rest 2 configurations.

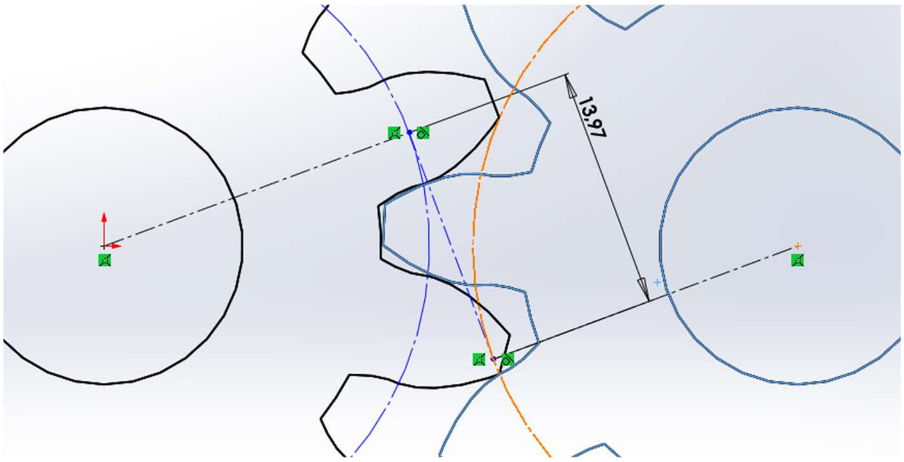


Figure 17. Evaluation of the length of the action line for the 2nd gear configuration

Regarding Poison's ration, the values for the examined plastics were found to be limited and difficult to obtain. As an author of an article from Engineering Edge (2019) suggested, the ratio of 0.35 for ABS could be also used for other materials alike and such an assumption was eventually made.

	Load per unit width (N/mm)	Effective radius of curvature (mm)	ABS	PVA+	PETG	PLA	PLC
v1	2.17	3.49	12.36	16.37	12.22	11.78	13.43
v2	1.70	3.49	10.95	14.50	10.82	10.44	11.90
v3	0.85	0.89	15.32	20.30	15.15	14.61	16.66

Table 11. Results of analysis of the mean Hertzian stress

The results of the calculation could be compared to the gathered data presented in Table 12 below. The data was collected from technical documentation and marketing brochures which were provided by respective manufacturers of materials for AM and freely available on the Internet. As it can be seen, for no material the estimated stresses would exceed a corresponding stress limit. That fact was taken into account as one prove of applicability of the design.

Material	Tensile (MPa)	Bending (MPa)
ABS	39	70.5
PVA+	78	N/A
Flex	N/A	N/A
Ninjaflex Sapphire	26	N/A
PETG	50	N/A
PLA	44	N/A
PLC	28	N/A

Table 12. Maximum stresses for different materials that are commonly used in 3D printing

The next step was to conduct a Finite Element Analysis (for now on an abbreviation FEA will be used) with the help of SOLIDWORKS Simulation. A single gear that had been mounted on a shaft was firstly examined. ABS was applied to the assembly as the only material in SOLIDWORKS material database that was featured in this study, thereby all inherited properties of the material such as Young's modulus, density, etc. were further used in computations. Later the results could be extrapolated on other materials of similar properties, thus on PLA, PLC, PETG and to some extent on more rigid PVA+. Table 13 below presents the abovementioned properties.

Model Reference	Properties	
	Name:	**ABS**
	Model type:	**Linear Elastic Isotropic**
	Default failure criterion:	**Unknown**
	Tensile strength:	**3e+07 N/m^2**
	Elastic modulus:	**2e+09 N/m^2**
	Poisson's ratio:	**0.394**
	Mass density:	**1020 kg/m^3**
	Shear modulus:	**3.189e+08 N/m^2**

Table 13. Material properties

Fixtures were added to the model (see Figure 18). Two cylindrical fixtures with limited axial and radial translation were added to the shaft by both sides of the gear to replicate the behaviour of bearings. One more fixture was added to a side of one gear tooth to replicate the mating contact between the gears.

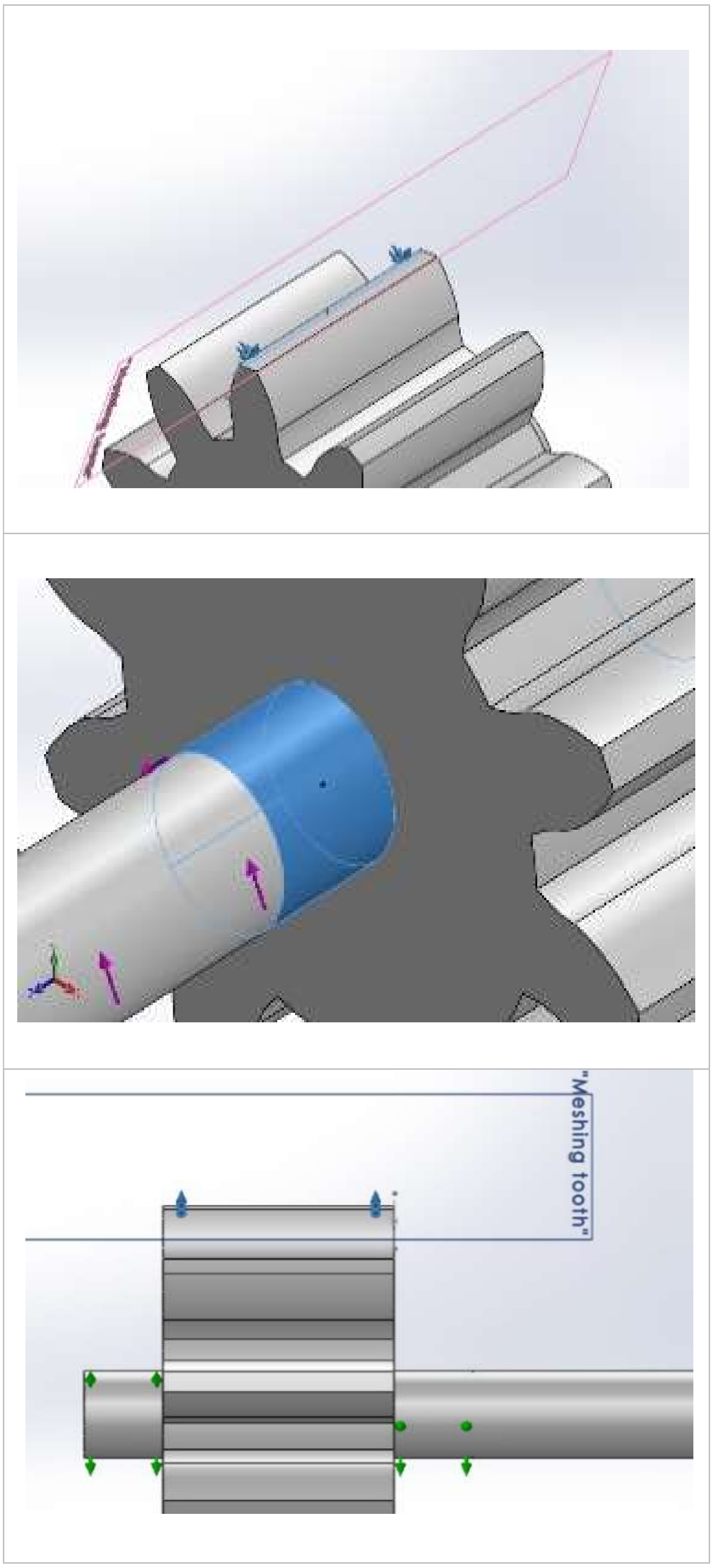


Figure 18. Fixtures

The shaft was subjected to torque with magnitude defined in section 3.2 and applied as shown in Figure 19.

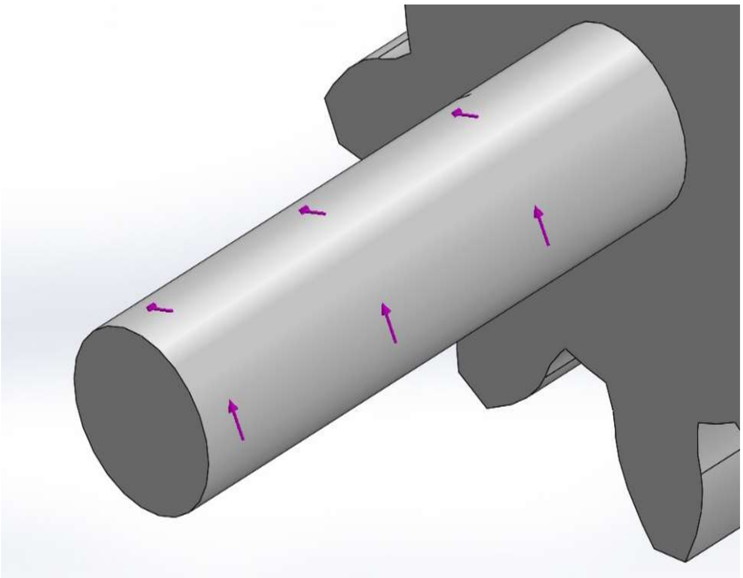


Figure 19. Applied torque

Meshing was applied to the assembly. Both parts were meshed with competitively – with mesh nodes without internal deformation, “node-to-node”. The maximum element size was equal to 4.8127 mm, the minimum – to 1.6042 mm.

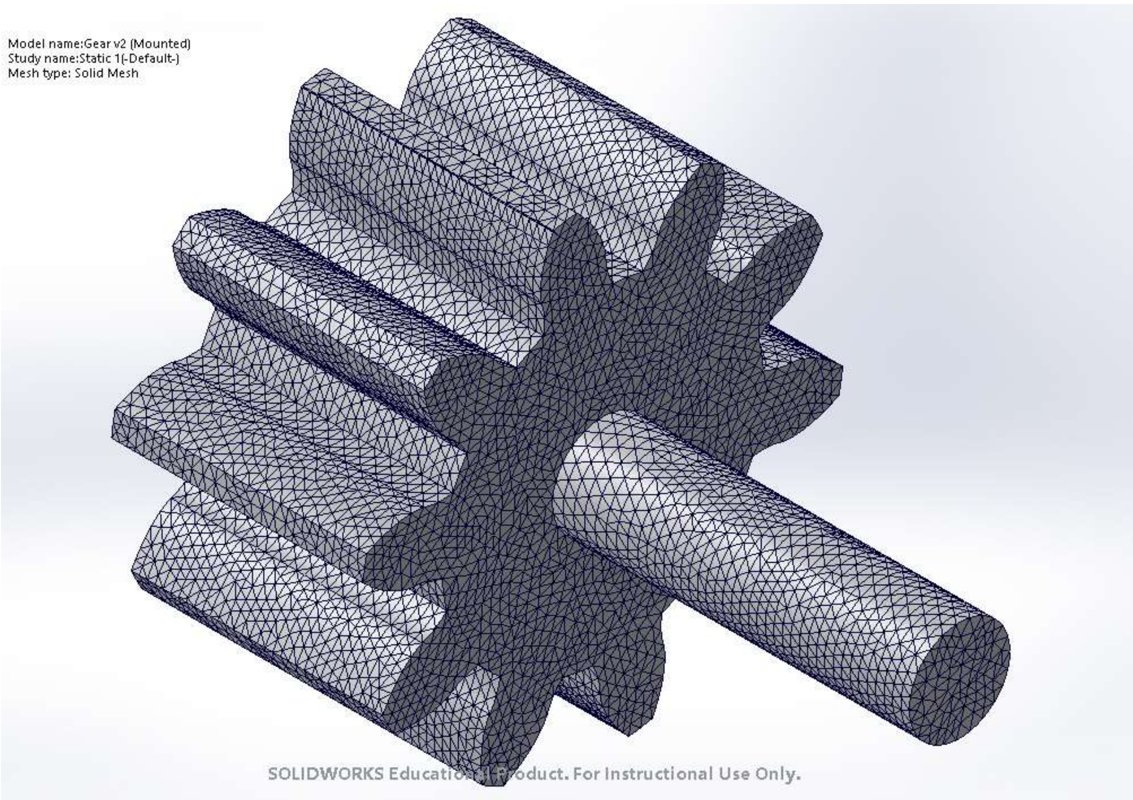
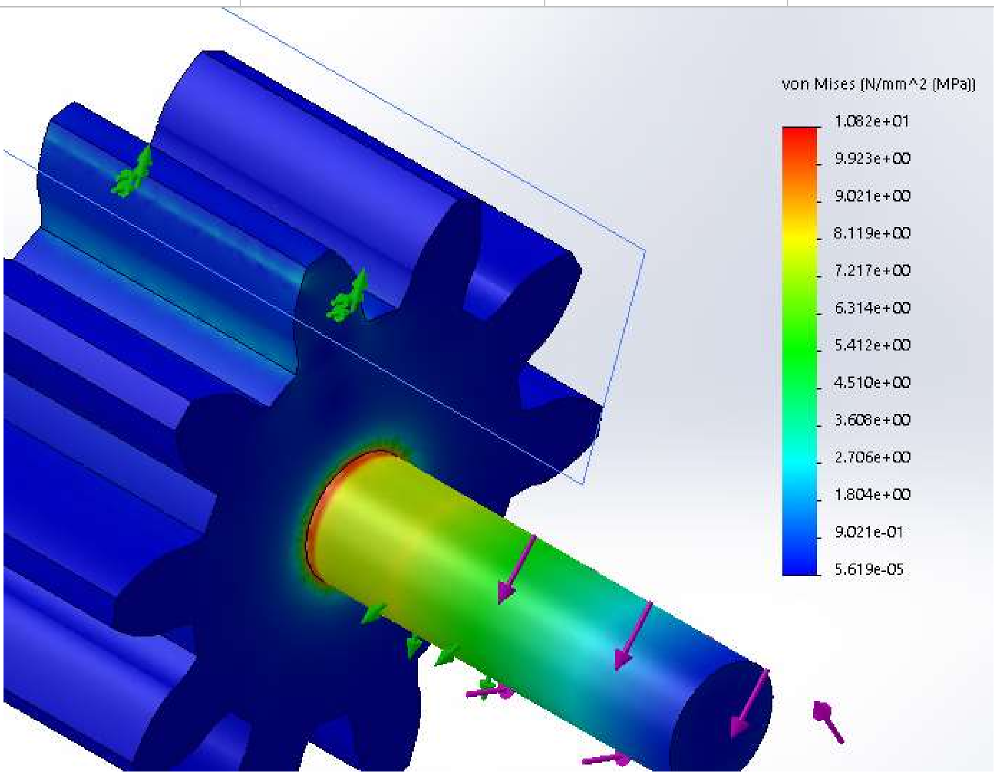


Figure 20. Meshed assembly

The results of the simulation are shown in Table 14 below. The maximum stress (market in the stress plot with a scarlet red colour) of 10.82 mm^2 occurred in the edge where the contact between the shaft and the gear happened. If compared to the ultimate stresses given in Table 12 (p. 47), the stress could be considered as low and not having potential to cause harm to the assembly during intended exploitation.

Furthermore, the value stress that occurred across the edge between the shaft and the gear could be decreased by modification of the associated geometry. An addition of a fillet would distribute the stress over a larger area and consequently reduce it.

Name	Type	Min	Max
Stress1	Von Mises Stress	5.619e-05 N/mm^2 (MPa) Node: 47	1.082e+01 N/mm^2 (MPa) Node: 42813



Gear v2 (Mounted)-Static 1-Stress-Stress1

Table 14. Results of FEA. Von Mises Stress

3.4 Slip estimation

For a new design, slip value is definable only with a use of professional flow simulation software. Such an analysis is beyond the scope of this thesis and therefore the exact value of slip of the pump will be determined through a laboratory testing.

Besides that, an arbitrary value may be assumed to adjust the values of the gears displacement and correct the geometry. According to experience in the field and with actual efficiencies of conventional external gear pumps, the value of the designed pump's slip was estimated to be equal to about 20% of the theoretical displacement. Slip factor was then defined as 0,8% according to an adapted from Wikipedia formula 19, that can be seen below.

$$\sigma = q'/q \qquad (19)$$

Where σ – slip factor, q' – actual displacement, and q – theoretical or geometrical displacement.

3.5 Infill

Parts that are produced by the technique of material extrusion are rarely completely solid and mainly include a hollow structure inside – infill. An online article that was published on Rigid.ink, suggests that strength properties of not fully dense (thus of structures that contain internal voids) structures are directly proportional to infill percentage. However, the exact function is not clearly defined: as a rule of thumb, 50% infill makes the part approximately 25% stronger than if the part of the same geometry had 25% infill and 75% infill is about 10% stronger than 50% infill. A simple estimation of the function was done based on the given description. The values were numerically compared, extrapolated to 100% infill and re-evaluated in respect to fully dense state. The results are shown in Table 15 and corresponding Figure 21 below. (Tyson, E 2017).

Infill	Strength * X
100%	1
75%	0.93
50%	0.84
25%	0.68

Table 15. Infill - strength dependence

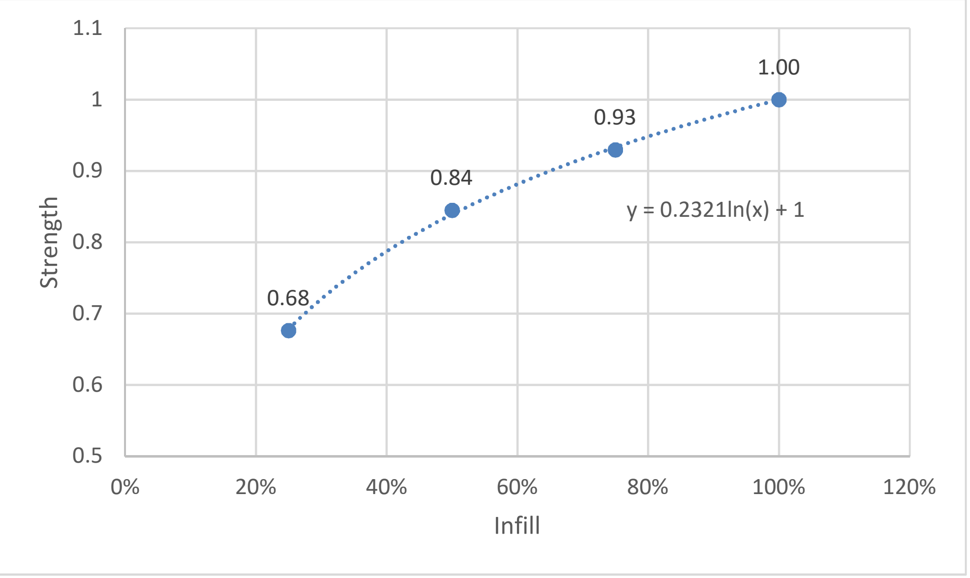


Figure 21. Chart of Infill – strength dependence. The chart features a logarithmic trendline with fitted formula

In order to use the results in further calculation several assumptions had to be made:

- The effect of outline thickness on strength properties is neglected.
- The body is considered to be isotropic and the material's performance is identical under loads, applied in different directions.
- The defined infill-strength ratio is not affected by exact geometry of the part (e.g. in especially thin elements where density would be 100% infill may have been disregarded)

The ratio was later used to define how much infill percentage would be required to keep the built part capable of carrying the load. Assuming that the value from

Table 11 could be applied to how properties of a part if they were built with the available AM machines would change, the following conclusion on suitable infill percentage could be made. The examined stresses were approximately 4 times lower than the maximum tensile stresses rated by manufacturers of the materials. Therefore, even with applied safety factor 2, 25% infill would be sufficient for the assessed application with a margin.

3.6 Oil test

The goal of the test was to experimentally define whether available polymers are susceptible to chemical reactions with mineral oils. Thereby, ensure that materials are suitable for the assessed application.

This is an empirical test without measurements. A formal study of an effect on polymers caused by contact with mineral oils, involving precise measurements of possible alterations of mechanical properties, chemical reactions, etc. are beyond the scope of the study. However, the aforementioned aspect of the test was taken into consideration in further discussion of the test's results.

Equipment of the test. The hydraulic oil and samples of thermoplastic polymers were provided by the personnel of Saimaa UAS. The glass beakers were provided by the members of a laboratory of LUT School of Engineering Sciences.

The test consisted of submerging of specimens into the mineral oil and following storying for a predetermined amount of time. The duration of exposure of every specimen was identical and has been arbitrary selected as 24 hours. The specimen was placed into a beaker (demonstrated in Figures 22 & 23 below) with 20 ml of mineral oil (brand Teboil) and stored for 24 hours. The oil was assumed to be of an ambient temperature of the room which was 23.2°C according to the room thermostat.

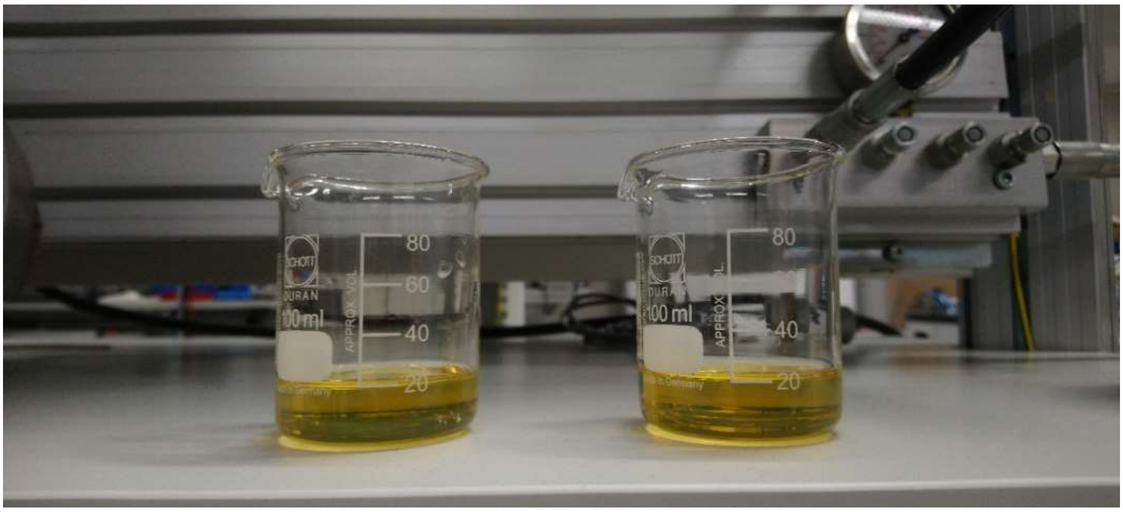


Figure 22. The beakers with the hydraulic oil. The volume of the oil in the beakers – 20 ml

After the predefined time period was over the specimen was removed, dried with a paper towel, and both the specimen and the remaining oil were assessed.

Figure 23. Setup of the test. The specimens of polymers have been submerged in the oil

The specimens were "standardised" by obtaining a length of about 35 mm. All specimens shared a universal thickness of 2.85 mm (diameter of cross-section area by the shortest side).

The following criteria were considered during the examination of the results of the test:

- a change in colour of the oil and/or the specimen
- a presence of precipitate in the oil, pointing at a chemical reaction of the two
- a change in surface appearance

- (may be inaccurate) a change in mechanical properties e.g. a decrease of tensile strength and stiffness

After a batch of the specimen was removed from the beakers, it was firstly cautiously dried with a paper towel and examined. The results are shortly denoted in Table 16 below.

	Change in surface appearance	Colour change	Precipitate	Changes in oil
PETG	-	-	-	-
ABS	-	-	-	-
PVA+	-	-	-	-
PLA	-	-	-	-
PLC	-	-	-	-
NINJAFLEX	-	-	-	-
FLEX	Permanent oily gloss	-	-	-

Table 16. Results of the test of submersion of the materials in the hydraulic oil. The symbol "-" here denoted a lack of any reaction between the oil and the submerged material

Unexpectedly, the specimens occurred to be rather inert to the oil and almost did not react with it. No precipitates were observed to be left in the beakers. Morphologically the specimens remained mostly intact, except FLEX: as a relatively soft and porous material it probably observed a significant amount of the oil which led to the appearance of an oily gloss on its surface.

The results of the test may be considered controversial. According to Benjamin Edwards, who had a post about the deterioration of chassis made of ABS and other related materials, ABS specimen had to suffer from being soluble in mineral oils due to the presence of esters in the latter. However, ABS yielded a negative result. Generally speaking, a mere fact that the specimens did not show reactance with oils raises the question of the legitimacy of the test.

As it was discussed in the beginning of this section, the test could suffer from a series of problem that had potential to affect the results:

- The test lacked precise measurements, e.g. the specimens' strength properties had been assessed neither before nor after the test. Therefore, no data could be qualitatively compared.

- The time limits restrained the test and the specimens were kept in investigated conditions for the duration which might not be enough for a reaction to
- The specimens were kept in the static conditions. An assumption can be made that, by analogy with the formation of a thin layer of aluminium oxide on the surface of aluminium parts on contact with oxidants, a reaction between the specimen and the oil could begin initially but later be suppressed by the formation of the “protective” layer of the top of the surface. During the actual exploitation, diverse stresses would appear in the materials, hence critically changing the conditions of the contact with a fluid.

Another reason for a lack of active reactions might be the used in the test oil. According to the markings on its label (see Section 2.3), it belonged to a family of hydraulic oils with increased anti- wear, -oxidation, and -rust properties. If precise chemical backing of the issue is to be left outside the scope of this study, it might be assumed that the oil does not have a predisposition to actively reacting with the tested polymers and thus reactions with the tested materials were going reluctantly and did not lead to any visible results.

3.7 Selection of other components

3.7.1 Motor

Popular internet resources and stores were browsed in a search for an offer. A few were found and then compared. After a discussion with the staff of the laboratory, who also had to place the order for the motor, one was acquired.

A decision was made to place an order on a brushless DC motor. Its electrical and mechanical diagrams can be found in Appendix 3. The main premise for that choice was the fact that variable frequency controllers (VFC) and a supply of DC current were available in the lab and could be used. That meant that a DC motor could be installed and operated with relative ease. Besides that, another factor was an appealing price that made the offer a good bargain. Table 17 below depicts the motor’s specifications.

Type by the number of phases	3 Phase
Rated Output Power	188 W
Nominal Voltage	24 VDC
No load Speed	4000 ± 10% RPM
Rated Torque	0.6 Nm
Rated Speed	3000 ± 10% RPM

Table 17. Specifications of the BLDC motor (ACT Motor GmbH)

The motor had to be connected to the pump through an interface. The shaft of the motor (see Appendix 3) featured a cut that would serve as a key. After considering several concepts a decision was to use a slotted end of the gear shaft. The shaft of the motor would be inserted into the gear shaft. That would allow to transmit the torque from the motor to the pump's rotor. A simple inspection of the stress was considered, and the results are presented in Figure 24 below. The analysis was conducted using a CAD model of the shaft, and ABS was applied as the material of the part. The slotted end was subjected to the torque and the opposite end was fixed.

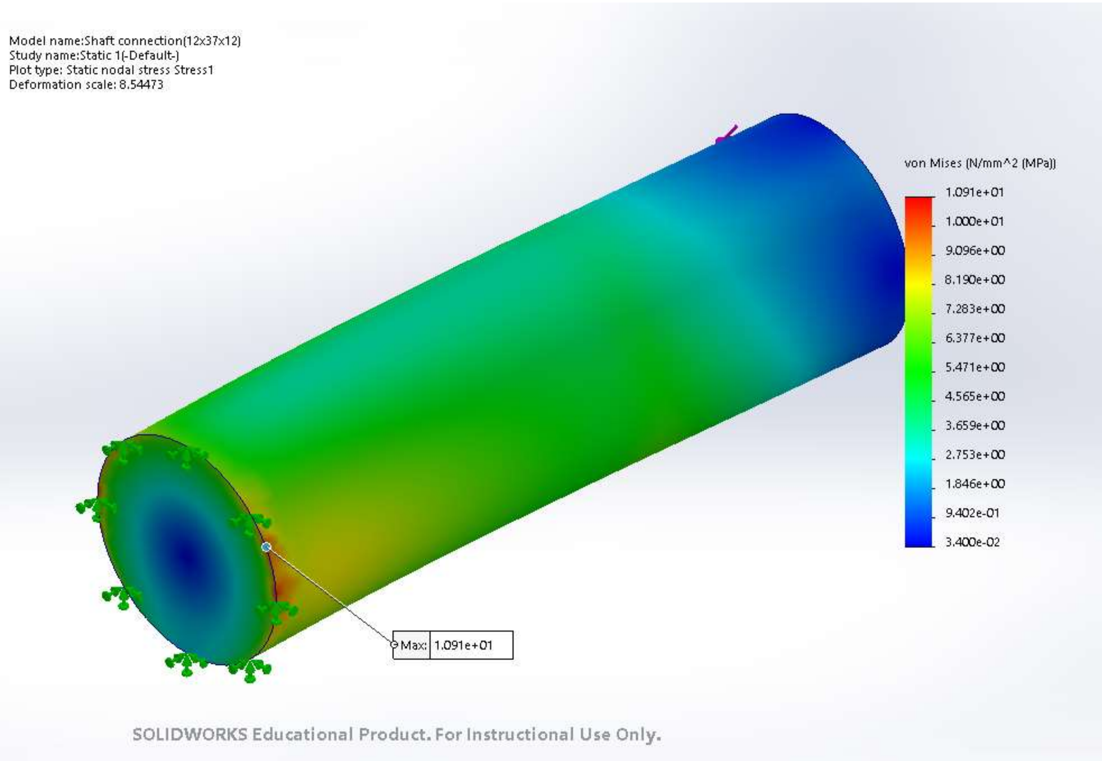


Figure 24. Von Mises stress on the gear shaft caused by torque from the inserted motor's shaft. Maximum von Mises stress is 10.9 N/mm^2 as denoted in the data box.

The defined stress of about 10 N/mm^2 would be 4 times lower than the ultimate tensile stress for ABS (see the table of material properties in the appendix 2) and therefore not critical for the part.

4 Proposed design

The following section describes the design of the pump that was proposed as the result of the project. The following Figure 25 demonstrates how all the designed components look within the assembly. Figure 25 also depicts fasteners (bolts, nuts, and threaded rods) within the assembly. A layout drawing of the pump can be found in Appendices (Appendix 4).

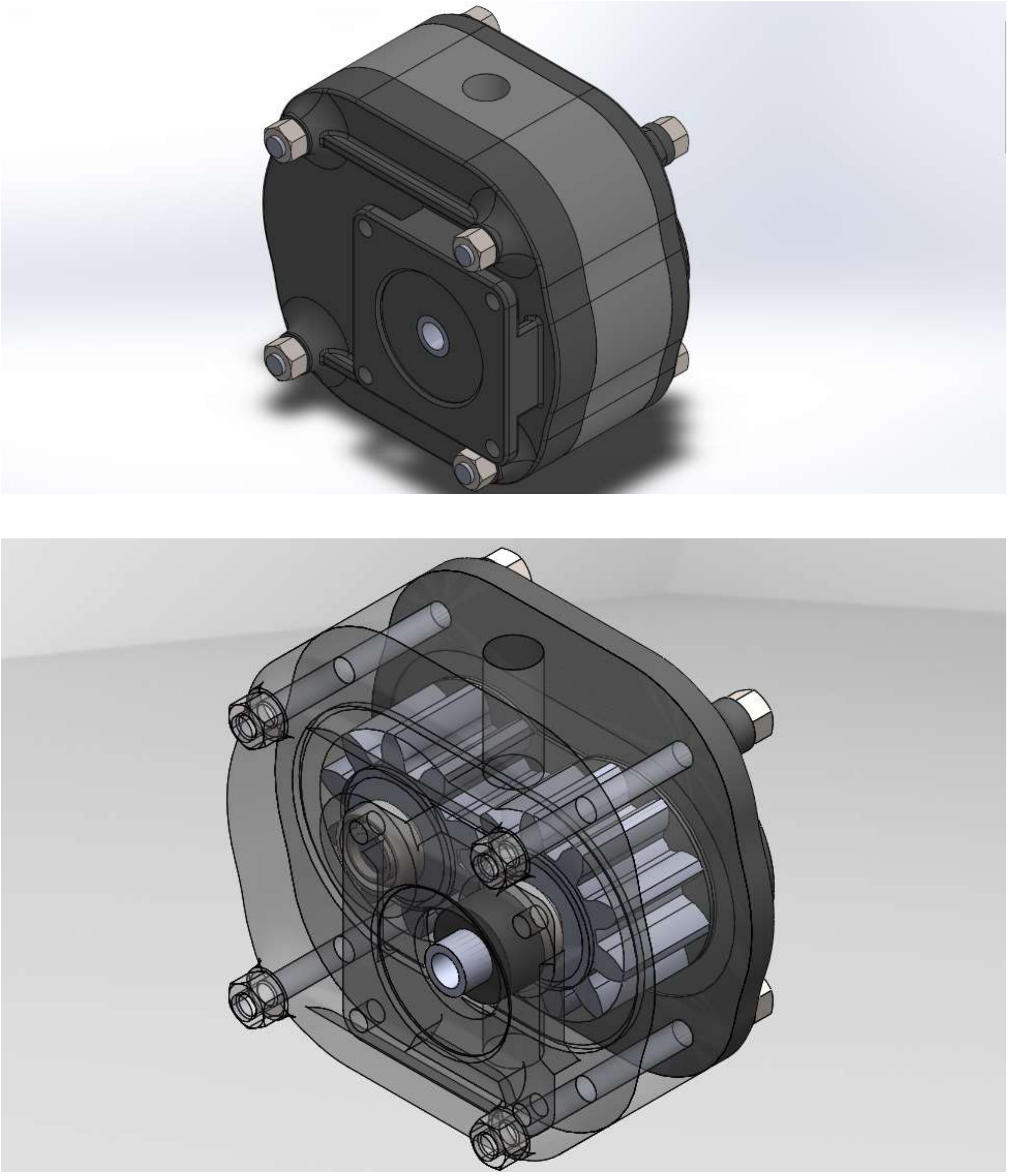


Figure 25. The pump assembly (with fasteners and with transparent front plate, casing, and fasteners)

4.1 Casing

The casing would be 100 mm high and 110 mm wide. The casing would have a thickness of 28 mm to fit the gears with a face width of 25 mm and additional mating elements of the back and front plates. 0.8 mm of the material was added to feature insert slots for gaskets.

The wall thickness is relatively shallow in comparison with many conventional external gear pumps as the associated with this particular design study loads were considered to be rather small and thermal effects insignificant; the casing was mainly required to maintain the pumping process enclosed and carry internal elements.

The inlet and outlet ports were configured to fit connectors that were used with hydraulic hoses Festo Didactic 1W-04 DN 06. These hoses were primarily used in the university's laboratory.

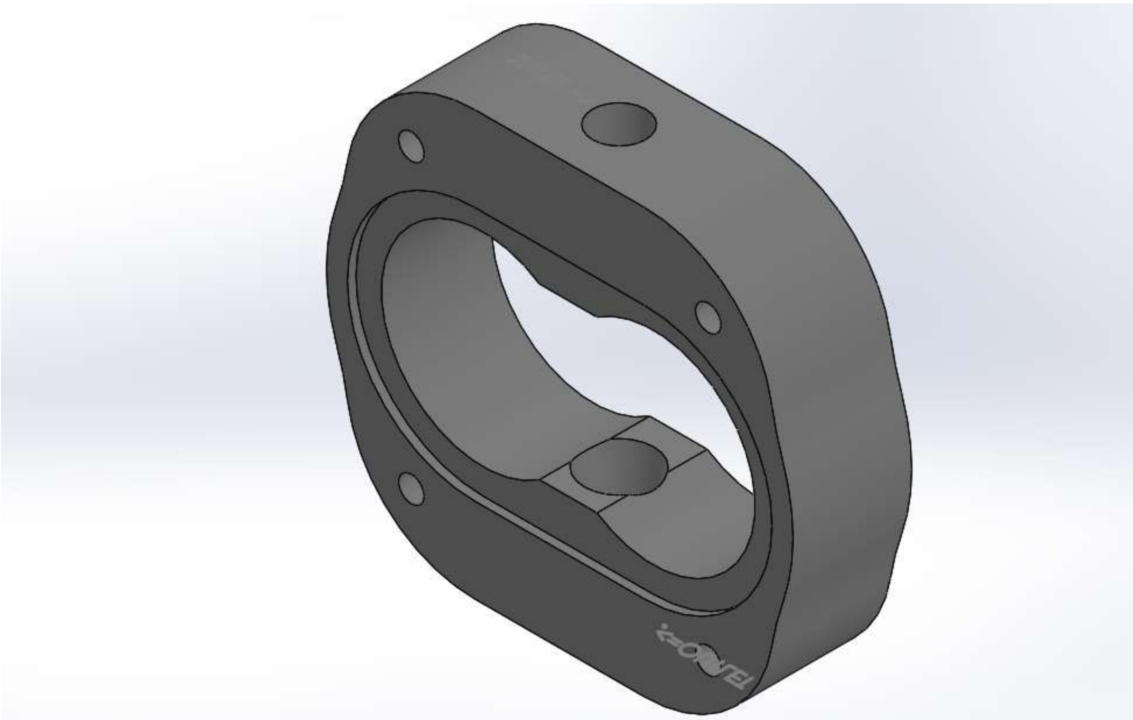


Figure 26. Proposed casing

4.2 Back plate

The back plate (shown in Figure 27) featured the contour of the casing-donor. The thickness consisted of 18 mm. The rear side included protrusions for fasteners and reinforcement for the bearing housing in the centre. It also had lateral ribs to increase the structural stiffness between the fastened points to prevent deformation in the case of exceeding thermal expansion of the plate during exploitation.

The front side featured housings for bearings, an insert slot for a gasket, and a relief slot. The latter was added to prevent risks of gears' failure due to trapped liquid between the meshing teeth. The housings were designed for bearings with dimensions of their bore diameter, outer diameter, and width equal to 12x28x8 (all in mm).

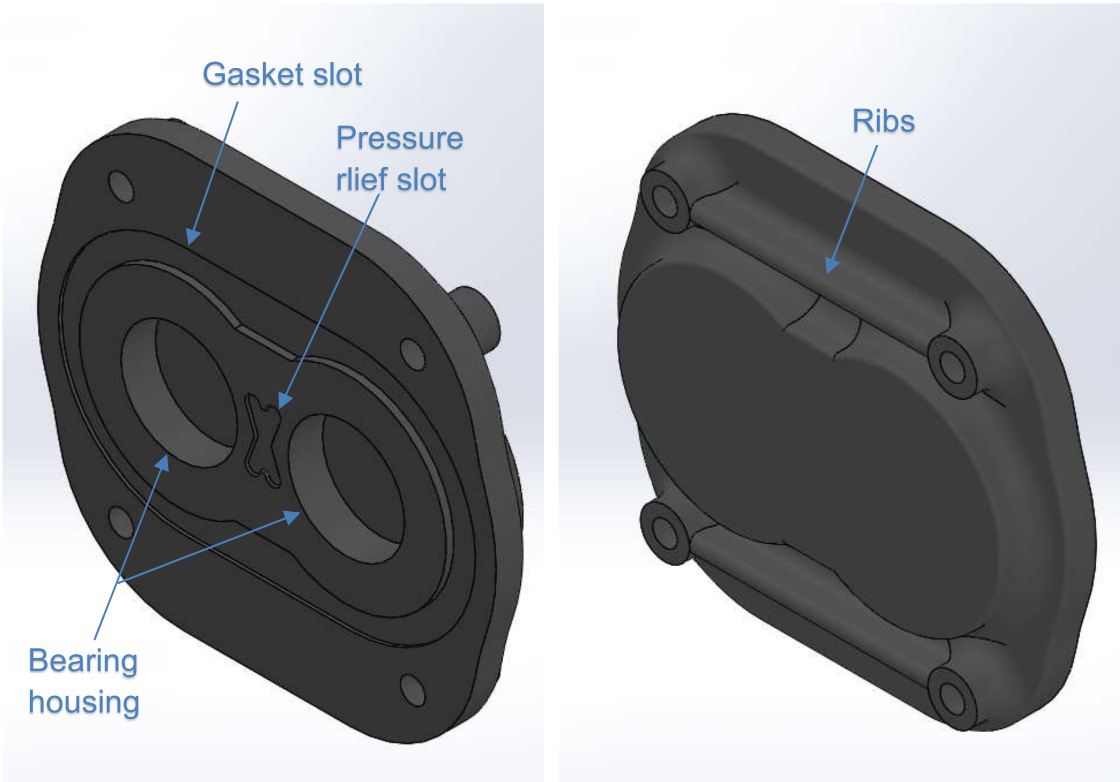


Figure 27. Back plate. The front view (left) and the rear view (right)

4.3 Front plate

The front plate (depicted in Figure 28) featured protrusions for fasteners, a housing hole for a shaft and a gland-like geometry for mating with a driving motor at the front side. The rear side had an oval protruding contour that was designed to be inserted into the casing, thereby reducing the possibility of leakages. Besides that, the side also featured a pair of housing for bearing and a slot for sealing around the shaft. The housings, as in the case of the black plate, were designed for a pair of bearings with the bore diameter, the outer diameter, and the width of 12x28x8 (in mm).

The gland is the interface with the motor was designed to fit the selected motor (see the section 3.7.1). The front face of it features 4 slots for fastening bolts (hole size 5.2), slot that matches circular protrusion of the motor's mating face, and a hole for the gear shaft.

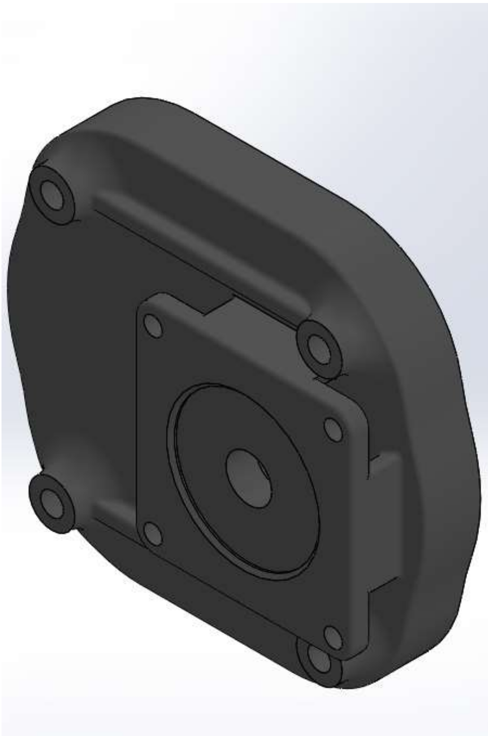
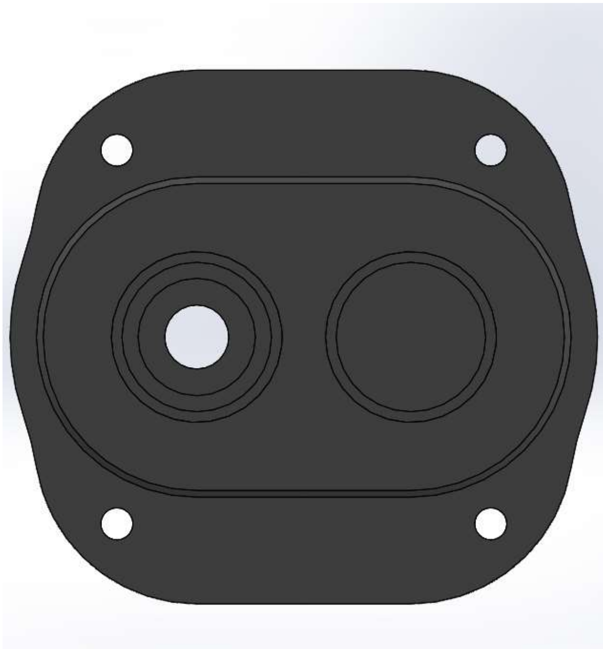



Figure 28. Front plate. The front view (left) and the rear view (right)

4.4 Gears

Gears would be mounted on the shafts as they are printed. A second configuration of the gears was taken as a base for the design and therefore the width diameter would be 25 mm. The pitch diameter would be 40 mm and the diameter of the addendum circle (distance between opposite teeth's tips) would be 48 mm. The gear would have 10 teeth and the pressure angle of 20˚. Figure 29 below depicts the gear.

The shaft also would include a slot at its end. The slot was designed to house the end of the motor's shaft which featured the same geometry as the slot. The decision to use this interface was supported by FEA, the results of which are presented in the section 3.7.1.

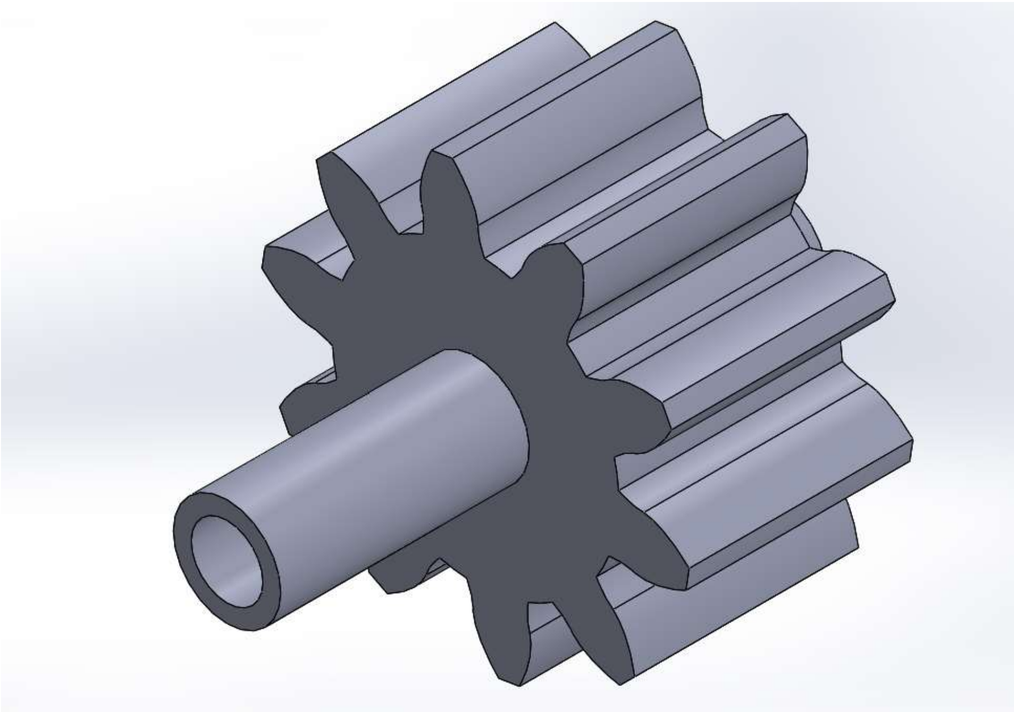


Figure 29. Proposed gear (driving)

The driven gear shared the geometry and dimension with the driving gear. The difference in the gears was in the length of the shafts (the driven shaft was sorter with 41.5 mm against 61 mm of the driving shaft) on which the gears were mounted and in absence of the slot at the end of the driven shaft.

5 Summary and discussion

The conducted study resulted in the creation of the CAD model of the external gear pump that complied with the predefined requirements. Initial assumptions of featured dimensions were adjusted and supported by the theoretical stress analysis and FEA simulation. Unfortunately, the initial goal of constructing the hydraulic pump according to the developed design was not achieved.

During the course of the project, multiple obstacles were faced and dealt. Original scope of the study did not include selection of an electric motor because an assumption was made that one could be found in the universities laboratories and used, but the actual situation appeared to be different. After the motor was delivered it had to be wired and connected to a speed/frequency controller which was thought to be free available for students. However, a head of the automation laboratory denied the access to it making a reference to safety issues. It meant that a PLC- based controller circuit had to be created and configured – a multitude of efforts to do that ended up in failures. This problem combined with the time restrains lead to a decision to cease the project prematurely at a stage when the manufacturing of the components and the construction of the pump-motor assembly were not completed.

The conducted study may be used to support other or further researches in a field maintenance works of hydraulic systems in remote areas where temporarily substitution of malfunctioning hydraulic units by rapidly manufactured counterparts or urgent replacement of laboriously manufacturable parts is required. Scenarios of failures of components with especially long lead time would benefit the most. With an example of a remote island with a developing economy, various mechanical devices can be manufactured “on-site” with only a limited inventory of universal printing materials.

Apart from commercial application, additive manufacturing of pumps, along with other AM goods, may become an aid in providing vital services to isolated areas. A hypothetical mobile printing station would be able to move around such areas and manufacture simple low-cost equipment for water wells or hydraulic levers.

Such an aid may give a significant impulse to development to less prosperous regions.

The results of the theoretical study could have been implemented in a real-life model build with printers provided by the university. To the greatest regrets of the author, the conditions of the available machines required maintenance and they could not be used for concurrent construction. It should be possible to recreate the designed pump and test in laboratory conditions to acquire actual empirical data about its performance and reliability and to the overall applicability. It would also prove the viability of the concept.

Another field to enlarge and improve this study could be the widening of the spectrum of the assessed materials and manufacturing techniques. An assumption of the author is that consideration and usage of selective laser sintering technique may greatly aid design freedom and the final properties of the pump. The technique is capable of creating geometries with good tolerances and surface tolerances.

Figures & tables

References

1974. Pump users handbook. Modern: Trade & Technical Press, p. 34.

BCN3D Technologies. 2019. BCN3D Sigma. https://www.bcn3dtechnologies.com/en/3d-printer/bcn3d-sigma. Accessed on the 7th of March 2019.

Brandt, M. 2017. Laser additive manufacturing: Materials, design, technologies, and applications. Amsterdam: Elsevier.

Budynas, R. G. & Nisbett, J. K. 2011. Shigley's mechanical engineering design. 9th ed. in SI units. New York: McGraw-Hill. (p.679, p. 706, p.707, pp.746-747, 2. p775)

Common Plastic Molding Design Material Specification. 2019. Engineers Edge, LLC. https://www.engineersedge.com/plastic/materials_common_plastic.htm. Accessed on the 23rd of April 2019.

Dobryakov, I. 2019. The photo “A printer LEGO hand with visible internal rectangular structure”.

Fischer, U., Gomeringer, R., Heinzler, M., Kilgus, R., Näher, F., Oesterle, S., Paetzold H., Stephan, A. 2006. Mechanical and metal trades handbook. Haan-Gruiten: Europa Lehrmittel. pp. 102-110.

Gibson, I., Rosen, D., Stucker, B. 2010. Additive Manufacturing Technology: Rapid Prototyping to Direct Digital Manufacturing. Springer Science & Business Media, LLC. pp. 1-6.

Groschopp, Inc. Motor Comparison. https://www.groschopp.com/products. Accessed on 28th of March 2019

Hales, T. 2001. The Honeycomb Conjecture. Discrete and Computational Geometry. 25 (1): pp. 1–22.

Jenner, C. 2012. Gear Pumps. ProcessPrinciples.com. http://processprinciples.com/2012/07/gear-pumps/. Accessed on the 11th of May 2019.

Kohara Gear Industry Co., Ldt. 2015. Calculation of Gear Dimensions. https://khkgears.net/new/gear_knowledge/gear_technical_reference/calculation geardimensions.html. Accessed on the 2nd of April 2019.

Kohara Gear Industry Co., Ldt. 2015. Sketch of Spur and Helical Gears. https://khkgears.net/new/gear_knowledge/introduction_to_gears/types_of_gear s.html. Accessed on the 7th of May 2019.

KVDP. Scheme of a generic piston pump. Wikipedia, The Free Encyclopaedia. https://commons.wikimedia.org/wiki/File:Piston_VS_Plunger_Pump.png. Accessed on the 10th of May 2019.

Lieneke, T., Denzer, V., Adam, G., Zimmer, D. 2016. Dimensional Tolerances for Additive manufacturing: Experimental investigation for Fused Deposition Modeling. 14th CIRP Conference on Computer Aided Tolerancing (CAT). Accessed on the 06th of May 2019.

Linden, A. & Fenn, J. 2003. Understanding Gartner's Hype Cycles. Gartner, Inc.

Mäkelä, M. 2009. Technical formulas: Basic formulas of mathematics, physics, chemistry and strength of materials, and SI system of units. 2nd rev. ed. Tampere: Tammertekniikka. pp. 18, 99.

P.Markevičius ir Ko. “Classification of DIN 51524 hydraulic oils”. Office address: Islandijos plentas, 217-11, LT49165, Kaunas, Lithuania. http://www.pmarkevicius.lt/en/useful/oil-and-grease-specifications/din-51502-classification/classification-of-din-51524-hydraulic-oils. Accessed on the 10th of April 2019.

Primacreator. PETG technical data sheet. https://primacreator.com/products/primaselect%E2%84%A2-petg?variant=61776055115. Accessed on 20th of March 2019.

Primacreator. PLA technical data sheet. https://primacreator.com/collections/pla/products/primavalue%E2%84%A2-pla?variant=61780172811 Accessed on 21th of March 2019.

Prospector. Acrylonitrile Butadiene Styrene (ABS) Typical Properties Generic ABS. https://plastics.ulprospector.com/generics/1/c/t/acrylonitrile-butadiene-styrene-abs-properties-processing. Accessed on 20th of March 2019.

Pump school. 2017. Scheme of a generic rotary lobe pump. Lobe Pumps. http://www.pumpschool.com/principles/lobe.asp. Accessed on 13th of May 2019.

RepRap contributors. ABS printing temperature. RepPar.org. https://reprap.org/wiki/ABS. Accessed on 20th of March 2019

RepRap contributors. RepRap. RepRap. https://reprap.org/mediawiki/index.php?title=RepRap&oldid=185157. Accessed on the 1st of May 2019.

Rosen, D. 2014. What are Principles for Design for Additive Manufacturing. Conference: 1st International Conference on Progress in Additive Manufacturing. DOI: 10.3850/978-981-09-0446-3_027.

Scopigno R., Cignoni P., Pietroni N., Callieri M., Dellepiane M. 2017. Digital Fabrication Techniques for Cultural Heritage: A Survey. Computer Graphics Forum 36 (1): 6–21. DOI:10.1111/cgf.12781. Accessed on the 5th of May 2019.

SFS-EN ISO/ASTM 52900:2017:en. Additive manufacturing. General principles. Terminology (ISO/ASTM 52900:2015).

Smith, J. D. 1983. Gears and their vibration: A basic approach to understanding gear noise. New York: Dekker. pp. 10-13.

Stratasys Ltd. 2016. Object30 Prime. Product brochure.

Stratasys Ltd. 2019. Object30 Prime. https://www.stratasys.com/3d-printers/objet30-prime. Accessed on the 7th of March 2019.

The Technology House/Sea Air Space. 2019. Polyjet (Objet). https://www.tth.com/3d-printing/objet-prototyping. Accessed on the 5th of May 2019.

Tyson, E. 2017. How to Use 3D print Infill Settings – Increase Strength, Save Filament. Rigid,Ink ttps://rigid.ink/blogs/news/optimum-infill. Accessed on 20th of April 2019.

Vialva, T. 2019. URI Students 3D Print at Sea with the Formlabs Form 2. 3D Printing Industry (3DPI). https://3dprintingindustry.com/news/uri-students-3d-print-at-sea-with-the-formlabs-form-2-147902/. Accessed on the 15th of February 2019.

Volk, M. 2005. Pump characteristics and applications. 2nd ed. Boca Raton (FL): Taylor & Francis/CRC Press. pp. 1-3, 5-49, 29-35, 328-333.

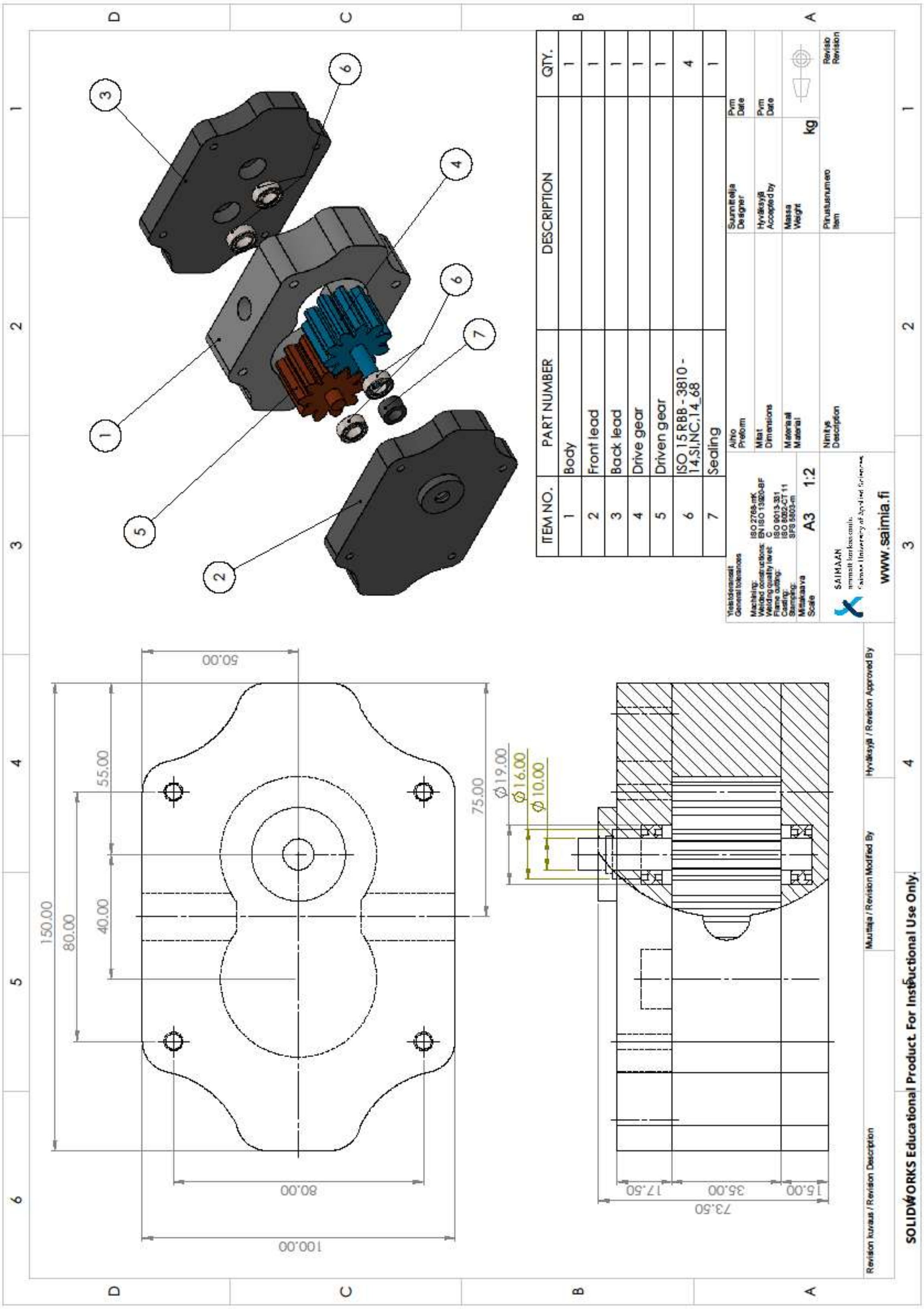
Waard, S. 2019. Screw Pump. Engineering360. https://www.globalspec.com/learnmore/flow_transfer_control/pumps/screw_pumps . Accessed on the 10th of May 2019.

Wikipedia contributors. 2018. Slip factor. Wikipedia, The Free Encyclopaedia. https://en.wikipedia.org/w/index.php?title=Slip_factor&oldid=835539385
Accessed on the 2nd of May 2019

Zhu, X. 2012. Tutorial on Hertz Contact Stress.

Appendices

Appendix 1. Layout drawing of the pump

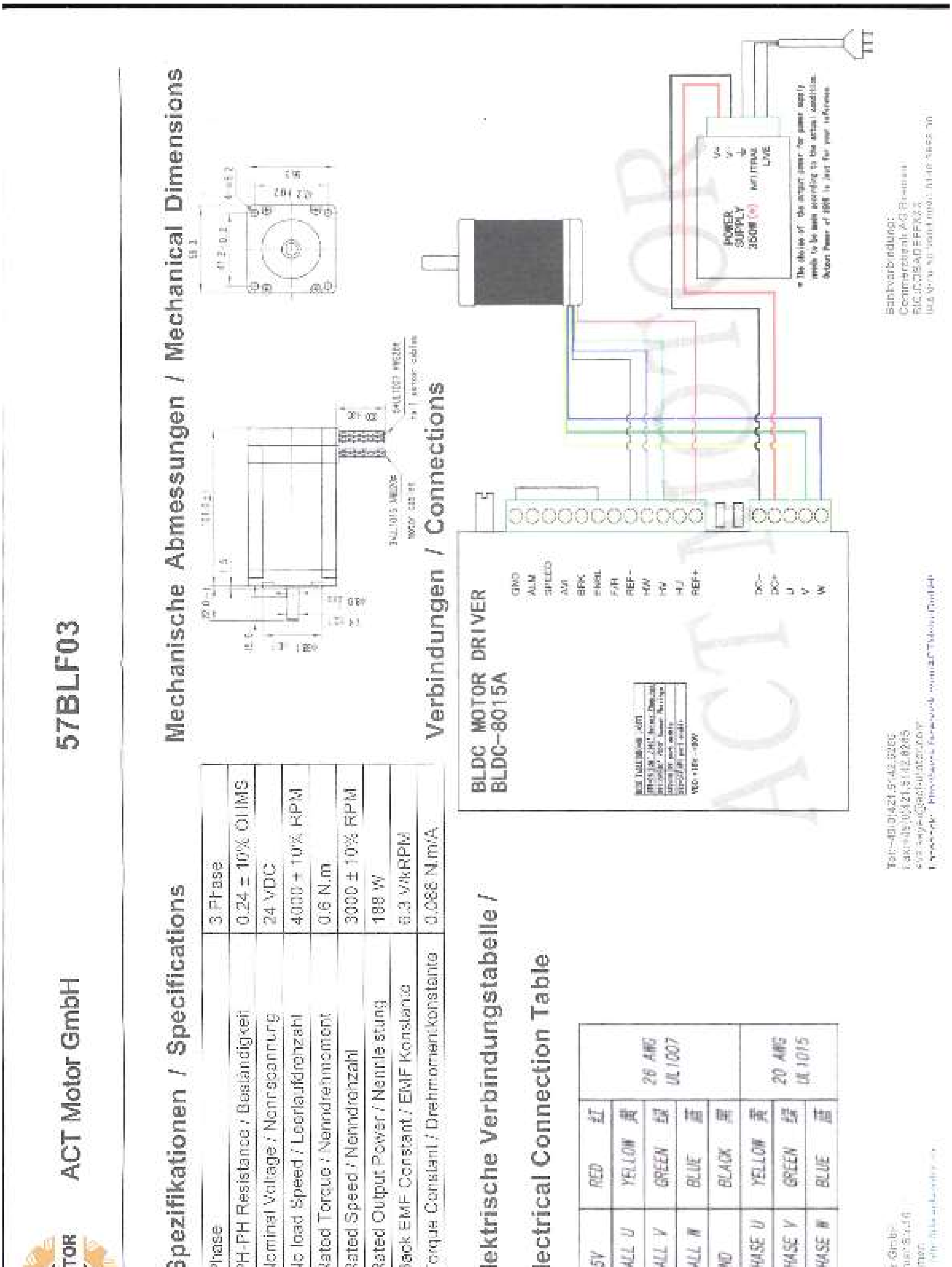


Appendix 2. Combined table of data on material properties

Criteria									Material
Temperature (0C)		Chem. Resist		Hardness		Strength (MPa)			
Melting	HDT (at 0.45MPa)	Ester	Water	Rockwell (D)	Shore (D)	E (Pa)	Bending	Tensile (u)	
N/A	45-50	N/A	N/A	73-76	83-86	2000-3000	75-110	50-65	Rigid Opaque(V
N/A	45-50	N/A	N/A	N/A	N/A	2000-3000	110	50-65	RGD720
N/A	63-67 (75-80	N/A	N/A	78-83	87-88	3200-3500	110-130	70-80	RGD525
225-245	N/A	0 (soluble)	N/A	N/A	N/A	N/A	70.5	39	ABS
163	N/A	N/A	0 (Soluble)	N/A	N/A	3860	N/A	78	PVA+
N/A	N/A	0	N/A	N/A	45	N/A	N/A	N/A	Flex
216	44	high (hydraulic oil)	High	N/A	85 (A)	1200	N/A	26	Ninjaflex Sapphire
N/A	70	N/A	1 (hydrophob.)	105	N/A	2150	N/A	50	PETG
245	N/A	N/A	N/A	N/A	N/A	2000	N/A	44	PLA

Density (g/cm^3)	Specific gravity	Temperature (0C)			
0	0	VST	Recommended Bed	Print t	Glass tr T
1.17-1.18	N/A	N/A	N/A	N/A	N/A
N/A	N/A	N/A	N/A	N/A	N/A
N/A	N/A	N/A	N/A	N/A	N/A
N/A	1.1	N/A	N/A	230	105
N/A	1.23	60,2	80-120	180-205	85 [1]
N/A	N/A	N/A	90-100	220-260	N/A
N/A	1.19	N/A	N/A	N/A	-35
N/A	1.27	N/A	45-60	195-235	70 [1]
N/A	1.03	N/A	N/A	180-210	60-65

Appendix 3. A diagram of BLDC motor 57BLF03 (ACD Motor GmbH 2019)



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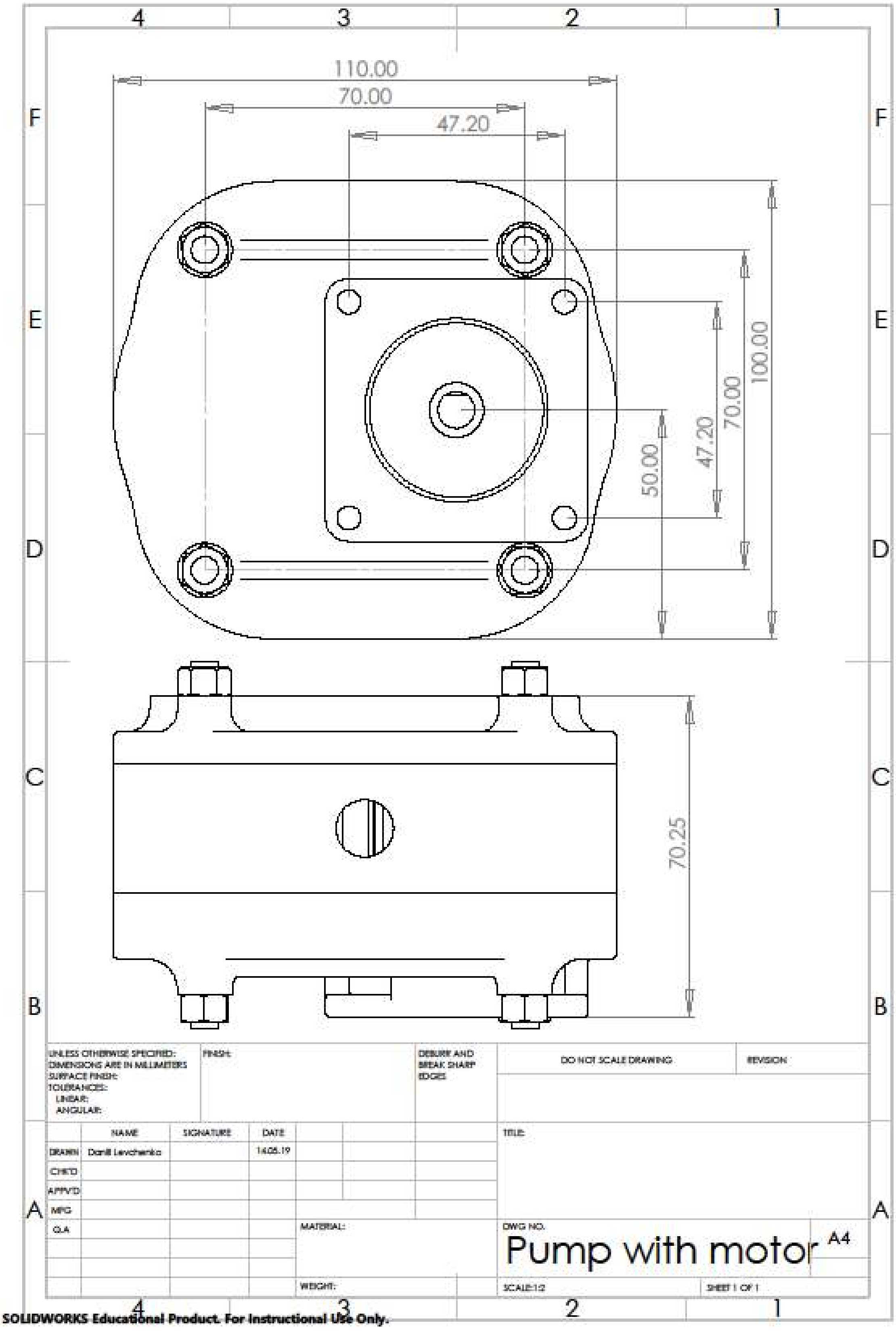
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Appendix 5. Render of the pump-motor assembly