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Design of an Energy Storage System for Pacific Islands

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Design of an Energy Storage System for Pacific Islands

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Problem

The islands in the South Pacific Ocean mostly use a diesel generator to produce electricity. To replace a generator by solar panels an energy storage system is needed to keep power and energy constant during day and night. This system should provide energy for at least 10 households.

Storage Systems

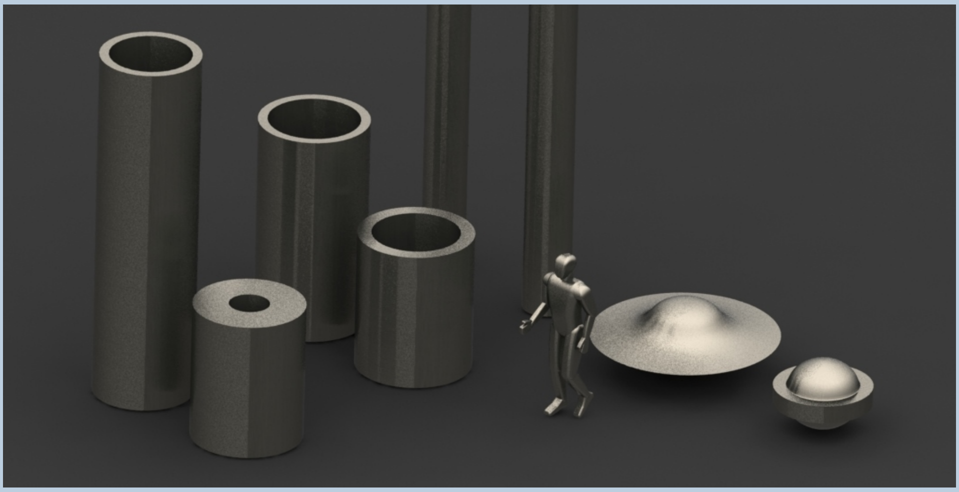
Multiple different storage systems have been reviewed and all differ in principle and specifications. The five main principles are: mechanical, chemical, electromagnetic, hydrogen and thermal energy storage. Out of these storage systems the flywheel is selected for its high power density $[KW/m^3]$, good specific power $[W/kg]$, high efficiency [%], long lifespan, high cycle life and low environmental impact. Also the maturity of the system is high for the flywheel since it has been used by people for ages.

Design Specifications

Size: 1.5m x 2m x 2m
Weight: <4000 kg
Capacity: 10 kW
Energy stored: 25 kWh
Cost: 36 100 FJD (US$ 17 028)
Production in the Pacific: Flywheel rotor and casing
Maintenance: >5 years without maintenance
Lifespan: >15 years
Environmental impact: 80% should be recyclable

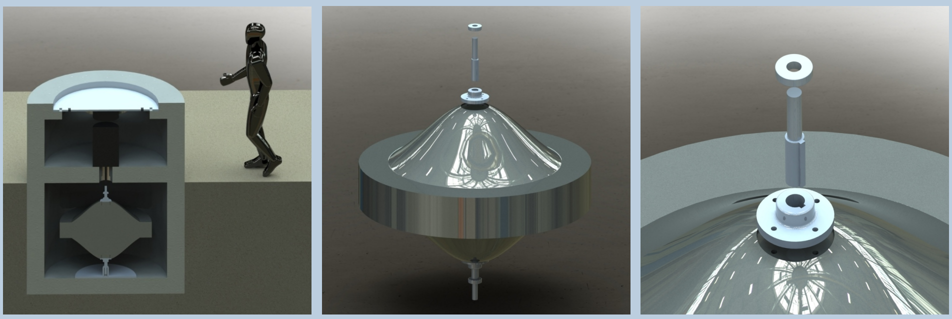
Flywheel Shapes

The flywheel rotor material is commonly made from metals or composites. The shapes of different flywheels can be seen in the picture below. Here different shapes are combined with different materials to find the best design. During the design it is important to know how much energy can be stored per mass. This can be calculated by equation: $W^M = K\frac{\sigma_u}{\rho}$. Here K is the shape factor which depends on the shape of the flywheel. The highest shape factor $K = 1$ is from the use of an equal stress disk. This shape has been used for the design.



Design

The design focuses on the flywheel rotor, shaft and bearing, since these are the most critical points of the design. The flywheel will be an equally stressed disk with a rim at the end. The design: $radius = 0.525m$, $height = 0.752m$, $mass = 2140kg$, $Energy_{stored} = 25kWh$ and $\omega_{max} = 10\ 000rpm$. The chosen bearing is the SKF QJ 305 N2MA. It is advised to use magnetic support for the bearing and a low pressure environment to keep the losses acceptable.



Conclusions

It is possible to design a flywheel that fits in the design specifications. A more detailed design is needed to see if the flywheel is really beneficial. Advantages: low environmental impact, lifespan and production in the South Pacific. Concerns: mass, safety and price.

Contents

Chapter 1

Introduction

Global warming and the rise of sea level are world scale problems. Of course this will affect the bigger continents but it will have even more effect on smaller islands, like those in the South Pacific. As some islands have a maximum elevation of less than five meters above sea level [1] a small raise of the sea level can have big consequences.

This is one of the main reasons for the Pacific Island Countries and territories to make targets for the implementing of sustainable energy. These targets cooperate with different government targets to provide the rural areas with electricity in the pacific. Electricity is not always a common good in the South Pacific. The current method of electrification of these islands is to provide these people first with a diesel generator and when electricity is a common good, the villages switch to renewable energy. However it should be from great good when the step of a diesel generator is skipped. Most Pacific Islands have a high level of solar renewable energy resources and a medium level of wind, hydro and geothermal resources [2]. Unfortunately these resources are not constant in their energy delivery. If the energy generated from these resources could be stored it would provide a significant improvement to the users. It will use the production spikes to store energy for less productive times. This would increase the benefits of renewable energy and could provide a 100% renewable solution for smaller electricity grids, like it is the case in Tuvalu. This island utilizes renewable energy only since 2012 [3].

The project is about to designing an energy storage system which can provide a small village from continual electricity. There are many different solutions for the storage of energy but not all suit the pacific conditions. Most islands are small and have rural surroundings which makes it difficult to reach the village and is expensive to connect with an electrical grid [4]. This report will give an overview of the situation on several islands and the possible solutions which will electrify the rural areas in a more renewable way. When the storage system is selected, specifications for a design will be listed and a design matching to these specifications will be made. At the end will be decide if this project should be continued.

In this report the prices are given in US dollars (US$) and in Fijian dollars (FJD). The exchange rate which is kept during the report is 1.00 US$ =2.12 FJD. The original price is given before the brackets and the exchanged price within the brackets. For example 3US$ (=6.36FJD).

Chapter 2

Pacific islands

In this chapter an outline will be given on the need of energy in the rural pacific islands. Some of the islands have an electricity connection but a lot of the smaller islands do not have a connection jet. Most islands which have electricity make use of diesel generators. First will be focused on the Pacific islands: which countries are there, what are their political behaviours and what are their needs. Most of the South Pacific countries are developing countries. This project is not the first renewable project in the Pacific, a lot of projects have been executed, some more successful than others.

2.1 The Pacific Island Area

Most people know the Pacific countries from their white beaches and small honeymoon islands. The tourists that look closer know that there are over thousand more islands which are inhabited by people, who sometimes live in very rural conditions. A lot of these islands do not have electricity or only have electricity a few hours per day when the generators are turned on. In figure 2.1 the map of the South Pacific countries is shown with their territorial waters.

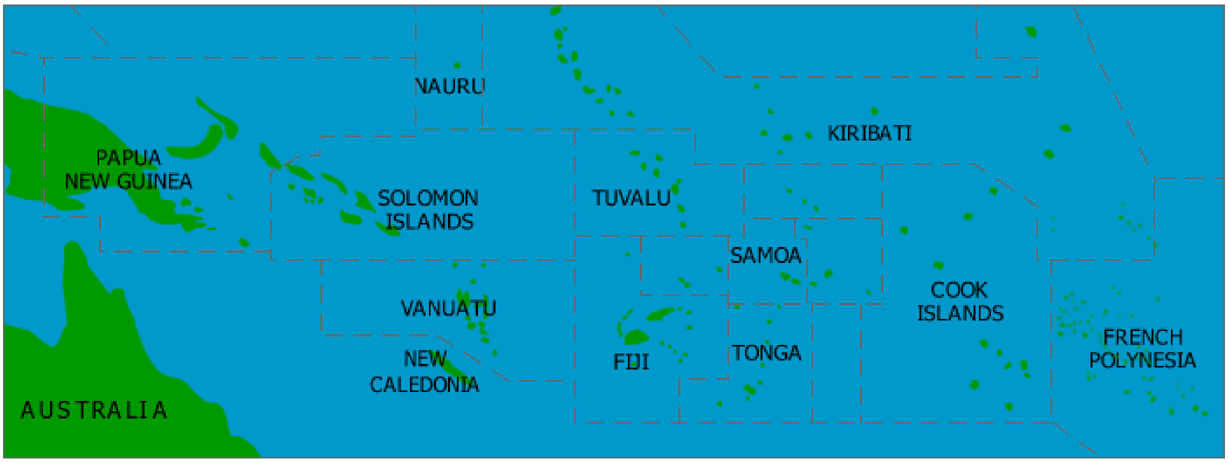


Figure 2.1: South Pacific Countries territory with Palau, Micronesia and the Marshall Islands more to the North [source: www.southwestpacific.com]

The differences between the South Pacific countries are big. Not only the population differs from 7 million in Papua New Guinea to 9847 in Tuvalu. Also the GDP per capita shows large differences. The Cook islands have with 13 478US$ (=28 575 FJD) almost nine times the GDP per capita than the GDP of the Solomon Islands 1 517US$ (=3 216 FJD). This results directly into the differences in development of the country and with this the access to power for the inhabitants. The correlation between the Electrification rate and the GDP per capita can be seen in figure 2.2. Here can be seen that the islands with lower GDP per capita also often have a low electrification rate. The same goes for the higher group of GDP per capita. In general three groups can be recognized in electricity development. Low levels of access in Papua New Guinea (10%), Solomon Islands (14%) and Vanuatu (17%). Medium levels of access: Federated States of Micronesia (54%), Kiribati (63%), Republic of Marshal Islands (80%) and Fiji (89%). And the high level of access in Tonga, Cook Islands, Samoa, Palau Tuvalu Nauru and Niue with all an access to power of 95%-100% [2], [5]. The marker shape per country in figure 2.2 indicates the population size, a square(■) is for 500.000+ inhabitants, triangle(▲) for 10.000-500.000 inhabitants and the circle(•) for a population smaller than 10.000 inhabitants.

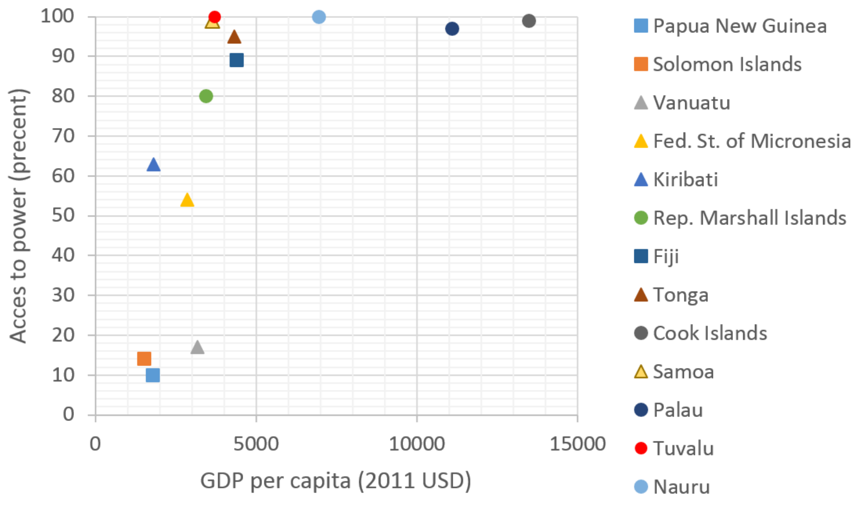


Figure 2.2: The access to power to the GDP per South Pacific country (2011) [2], [5]

The problem with expanding the current electrical grid is most of the time that it is difficult and expensive to reach the rural areas. The reason for low electrification level is not always purely financial but sometimes it is combined with political issues. Some of the Pacific countries use a fixed electricity tariff which is sometimes even below the actual costs. This makes it not attractive for third parties to invest in the electrification. Most of the time, governments use subsidies to cover the gap but this makes the situation unstable. These subsidies have to be paid upfront and are from high order because of the rural area. Two key requirements for expanding access to electricity in Pacific SIDS have been identified in the paper of M. Dornan [5]: "The first is funding. Funding is crucial to rural electrification, whether through extension of the electricity grid or installation of off-grid systems." "The second requirement for expanding access to electricity in Pacific SIDS is the reform of institutional arrangements. It is also needed in order to ensure that off-grid systems that are installed remain in a workable condition."

2.2 The Pacific Conditions

In the Pacific there are specific conditions that influence a project which is done in and by the Pacific. Of course the first is climate and earthquakes. Some of the islands in the South Pacific lay close to the fault lines of the Earth combined with the rain season where the change on hurricanes is high. Both of these conditions should be taken serious in a design. Also production and knowledge is something to keep in mind in the design process because in the rural areas, people with the right knowledge are not always close by. The inhabitants live isolated and replenishment of goods does come weekly or even less frequently. So it would be useful if the maintenance could be done by local people which are not schooled in this job. Next to this it must be taken into account that most of the goods which are delivered on the island will never leave the island again. Environmental impact is from great importance because this is also the environment where the people get most of their food from. If there is a defect which cannot be fixed by the local people, schooled employees should come. They should be located in central civilized areas. The transport of the employees would take more time than normal and could take days to weeks.

2.2.1 Vunivau Solar Home Systems

In 2002 a program was funded by the governments of Fiji and Japan to increase the electrification of the rural areas. The installed solar systems had the size of 100Wp with five lights and one power point. A total of 250 systems were sold during the project [6]. The program employed RESCO, Rural Electricity Service Company, with the Fijian government as owner. The users in the rural areas paid a fee-for-service for the use of the system. This was a monthly tariff payment to rent the system which costs US$14 (=29.68 FJD) per month from which US$0.5 (=1.06 FJD) goes to the post office and US$13.50 (=28.62FJD) to the service company. The first survey under local users of the systems was positive. Unfortunately a more recent survey showed that in 2009 about 80% of the systems were out of service due to no availability of monitoring or maintenance services. The amount of defects could have been decreased if users were trained to do simple maintenance. Concluded out of this project can be that the factors to success are [6]:

- Private ownership to reduce the maintenance cost and overuse of the system.
- Smart subsidies to improve the affordability of the systems for the low incomes.
- Link the system with productive use of the system. This increases the durability and decreases the maintenance.
- Regular maintenance and monitoring to increase lifetime and reduce failure.

2.3 The Pacific Villages

As been described in the earlier section, villages in the Pacific are located in rural areas. This project does not want to focus on one specific village or island. This report will use two types of villages, a small and a big village. The small village has around 10 households in a very rural area with one society building, for example a chapel, community room or small office. The big village has a population of 100 households with ten society buildings. This village lays in a less rural area and with this it is easier to get an engineer to repair if the source is defect. Both villages already use generators to produce energy but people in the big village are more familiar with electricity. To give an idea of both villages two examples can be seen in figure 2.3 & 2.4.

Figure 2.3: Aloa Bay, Solomon Islands (source: Google maps)

Figure 2.4: Village, Rere area, Solomon Islands (source: Google maps)

When a non-electrified village starts to use electricity the inhabitants do not have many electric products. The only need they have is for a few light sources. When people get used to the usage of energy and trust it, their energy usage will develop. In documents of the United Nations is described that Pacific rural households have an entry energy use level rate of 0.2 kWh/day and an end level rate of 2 kWh/day [7]. In this document is estimated that these households exist out of 5 persons. A level rate of 0.2 kWh/day means a household at the entry level can have 2 saving energy lamps for 8 hours per day, while a household at end level is equivalent to 1 fridge, 1 mobile charger, 1 radio, 1 glow lamp, 2 saving energy lamps and 1 ventilator on average use. The specifications of both villages can be seen in table 2.1. From data of several rural area projects [3], [8] can be concluded that the capacity of the small village is around 10 kW and the big village is 200 kW.
The amount of energy which needs to be stored depends on the demand but also on the production. The production in solar panels is not constant. It depends on the sun power and the length of the daylight per day. The weather center in Nadi, Fiji, has measured that the difference in daylight through the year is two hours. With 13 hours of daylight in December and 11 hours in June the length of the solar production time is quite constant through the year. In Europe is stated that on a cloudy day the solar power is dropped to an production of only 10-25%. The average production of a solar panel in the Pacific is 1kW

per 10 squared meters. This gives an average daily production of 3.5 kWh. It is estimated that the storage should be at least 2 days without any production. Since the solar panels are always producing a low percentage of the energy the reserve power will be more than 2 days. Since both villages currently have a diesel generator this can be used to load the storage system when a long period of cloudy weather is present. This is likely to happen only a couple of times a year.

	Small Village	Big Village
Population	10 households	100 households
Average power use	1 kWh/household p day	2 kWh/household p day
Society buildings	1 buildings	10 buildings
Average power use	2.5 kWh/building p day	10 kWh/building p day
Total power use	12.5 kWh p day	300 kWh p day
Storage time with low input	2 days	2 days
Capacity	10 kW	200 kW
Area of solar panels	40 m^2	800 m^2
Total Energy storage	25 kWh	600kWh

Table 2.1: Two different villages with their specifications

Chapter 3

Energy storage

In this chapter an overview will be given of the current situation on the developments of energy storage systems. Which options are there and what are their strong and weak points? After this overview an outline will be given of the current projects that are built in the world. This is done because a lot of the projects in the first overview are only theoretically worked out or in a developing state. These storage options should be kept in mind for later projects but are not of any use for this report. The result gives us more input for the comparison between the options of the energy storage for the Pacific islands. There are two different sizes of islands used for this comparison. A small Pacific island with around 10 households and a bigger pacific community with around 100 households. At the end of the chapter they will be compared and conclusions can be drawn.

3.1 Energy storage systems

The application of energy storage can be roughly divided in three groups based on their storage duration and purpose.

- Power Quality
  Duration $<$1min: Used for smoothing and transient stability.
- Bridging Power
  Duration=1min-1h: Used for emergency backup and wind power smoothing.
- Energy Management
  Duration$>$1h: Mostly used during peak shaving and time/seasonal shifting .

Not only the application can be divided, also the field of energy storage can be divided into five areas: Mechanical, Chemical, Electromagnetic, Hydrogen and Thermal energy storage systems. All the areas are fundamentally different and use different techniques. The terms and names of the systems are directly cited from a paper by A. E. K. Siraj Sabihuddin and M. Mueller [9].

3.1.1 Mechanical Energy storage

Pumped Hydroelectric Systems (PHS), Compressed Air Energy Storage (CAES) and Flywheel Energy Storage (FES) systems are the best known solutions of mechanical energy storage. They all make use of the storage of potential energy or kinetic energy.

Pumped Hydroelectric Systems

Pumped Hydroelectric Systems (PHS) are currently the most dominant systems when it comes to storing massive quantities for long periods. The method is quite basic: there are two reservoirs, one on high ground and one on low ground. This can be seen in figure 3.1. When energy is needed, the system uses water from the upper reservoir through a turbine to generate electric energy from the potential energy. The water is pumped back up again when there is an overgrowth of electricity. PHS systems have a long life, low operation and low maintenance costs which make them suited for big projects. The disadvantage is that it needs certain geographical terrain and besides the height difference the area also has to be large. When the area is a problem a gravity power module is an option. Here the same principle is used but then on lower scale and not only water is pumped upwards but also a piston. This system is closed and much smaller than the PHS. Both systems can be used for grid stabilization and long-term energy storage [9].

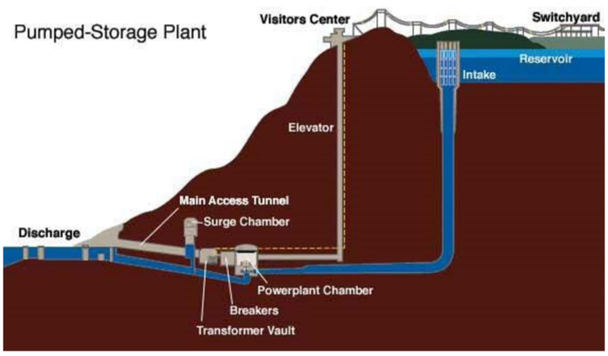


Figure 3.1: A Pumped Hydroelectric System [10]

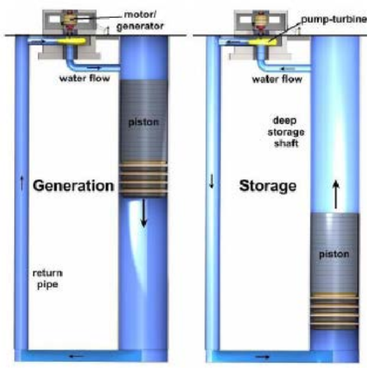


Figure 3.2: Gravity power module [10]

Compressed Air Energy Storage

The Compressed Air Energy Storage (CAES) is the second contender for large scale storage. The working principle is quite similar to a PHS system. When energy needs to be stored, air from outside the system will be pushed into the system which creates an increase in pressure and will be released through a turbine when electricity is needed. While most projects use underground caves, smaller scale systems use pressurized tanks but these are expensive. Figure 3.3 gives a schematic overview of the process. The compression phase's heat in salt caves can result in round-trip efficiencies of 90%-95%. The response time is between 5-15 min. [9], [10].

Flywheel

The last mechanical energy storage is a Flywheel Energy Storage (FES). Energy is stored in a rotating mass. Devices are composed of five main subsystems: flywheel, electrical machine, power converter, bearing and containment chamber. These components can be seen in figure 3.4. FES systems are mostly used for short-term storage because of their very low reaction time. This gives them a high efficiency. They have high specific power and power densities for the energy, fast response time and long life. The two main disadvantages are the high self-discharge rates and safety. The self-discharge rate lays around the 3%-40% capacity per hour. The latest improvements have reduced the losses below the 0.1% of superconducting solutions. But these improvements can not yet been used on large scale.

Overall these mechanical energy storage system is fairly mature and commercially tested. While PHS and CAES store for a long time with high capacity, FES still operates on a relatively small scale and for short durations [9], [10].

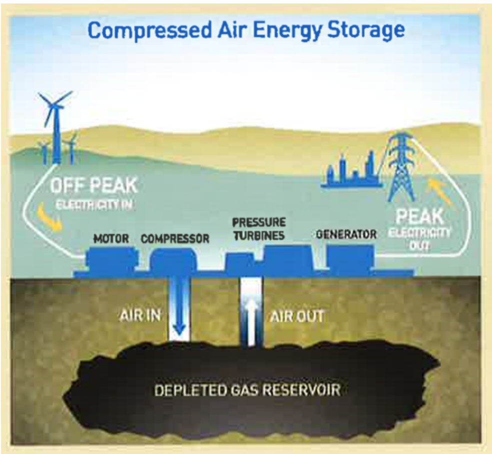


Figure 3.3: Compressed Air Storage System [10]

Figure 3.4: Flywheel system [11]

3.1.2 Chemical Energy Storage

Chemical storage has by far the greatest diversity of commercial energy storage products today. Next to this it is the best researched and developed field. These storage systems can be divided into: typical batteries, molten salt and liquid metal batteries, metal-air batteries, fuel batteries and flow batteries. While fuel and flow batteries produce energy on fuel and gas next to storing and are almost generating instead of storing they are excluded from this report.

Typical batteries

These batteries have a big arsenal of different combinations and each has its own advantages. A select range of these batteries are: zinc silver oxide (ZnAg), alkaline zinc manganese dioxide (ZnMn), lead aid (Pb-Acid), lithium ion (Li-Ion), nickel metal hydride (NiMH), nickel cadmium (NiCd), nickel iron (NiFe) and nickel zinc (NiZn) devices. The working principle of all these batteries is similar, with a positive (cathode), negative (anode) electrodes, the electrolyte and the separator electrons are stored. The measuring of the state-of-charge is an issue which gives some complications. Differences between these batteries can be mostly found in their energy density and the temperature they produce while storing. Despite big improvements on recycling most batteries still have a big environmental impact [9].

Liquid Metal, Molten Salt and Metal-Air Batteries

In the last years there has been a rise of higher temperature batteries that utilize molten salts and liquid metals. The molten salt batteries are mostly containing Sodium Sulpher (NaS) and Sodium Nickel Chloride (NaNiCl). These batteries utilize liquid/molten salts as electrolytes which also plays the part of electrodes. The electrodes are separated by a solid membrane separator. The biggest issue is the thermal management and because of their liquid state the process should be continued to prevent the molten salts to become solid. NaS batteries normally operate at 300-350°C. To maintain the temperature heating systems are needed. The operation of these systems are mostly the reason for efficiency drop. Next to this also safety and environmental issues are disadvantages of these batteries. There are some working examples in Japan and in the United States. The liquid metal electrodes are in a very early stage of development and have not yet seen any commercialized application. This is why they are left out of this report. Metal-air batteries have replaced the second electrode with an air electrode. The two mainly used are zinc-air (Zn-Air) and iron-air (Fe-Air). They offer one of the highest energy of most storage systems in general but most batteries suffer from an extremely poor cycle and shelf life. Zn-Air variations take care of these points: they are cheaper, environmentally better and have a longer storage life. Like molten salts, metal air batteries are sensitive to temperature changes. The latest improvements applied to iron-air (Fe-Air) batteries. Include cycle life of 5000 cycles and efficiency improvements of up to 80%. Which makes them a really good option for large scale energy storage [9].

3.1.3 Electromagnetic

First electromagnetic energy storage was limited to capacitors and inductors. Recent development of super-capacitors and superconductors has extended the use of capacitive and inductive technologies to larger scale applications. Superconducting Magnetic Energy Storage (SMES) refers to their use for energy storage in the magnetic field of an inductor which is superconducting. The SMES system has four components: the superconductor, the refrigeration, the containment vessel and the power converter. The main usage of a SMES system is for short-term power quality and stability. They have a very long lifespan, cycle life, high efficiency and fast response [9], [10]. The main issue is that the capital costs per KWh are massive. Next to this the cooling costs are high but can be lowered by the use of high temperature superconductors. With some more developments it could be a good option instead of the flywheel. The Electro Double Layer Capacitor (EDLC) that uses a combination of porous separators and electrolyte in place of a dielectric. While they operate at low voltages but have a high capacity, long cycle life and wide operating temperatures [9], [11].

3.1.4 Thermal Storage

Thermal storage is not new on the energy storage market. They have been used extensively in home temperature stabilization as can be seen in figure 3.5. Also on industrial scale thermal power plants and district heating applications have been developed. The use of thermal storage and heat recovery has allowed power plant efficiencies to increase up-to 60% for natural gas plants. Also for solar energy this type of storage is very useful. There are three types of storage that are in front: Sensible Thermal Energy Storage (STES), Latent Thermal Energy Storage (LTES) and reversible Chemical Thermal Energy Storage (CTES). These systems have always three components: the thermal material, the heat exchanger and the containment system [11].

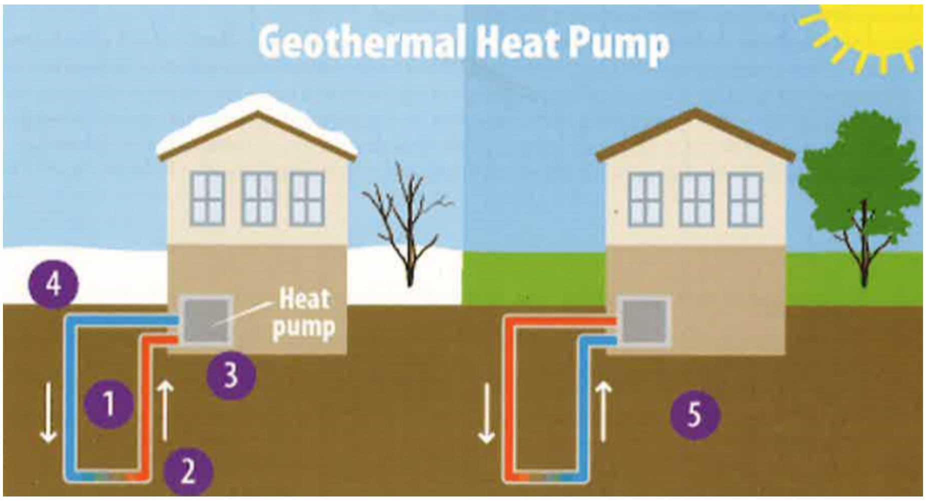


Figure 3.5: Geothermal Heat pump [11]

Sensible Heat Storage

There are two types of Sensible Thermal Energy Storage (STES) systems, active and passive. In passive systems the storage medium is fixed and heat is transferred through a passive heat transfer mechanism. While in active the storage medium is circulating through the system. This could be compared to most heat exchangers. Active systems are slightly cheaper because there are no different tanks needed for cold and hot substances.
Note that these systems differ from the molten salt batteries. For temperatures between 0 °C and 100 °C water is the most used option. STES systems are the cheapest form of thermal storage and work best on short-term storage. In figure 3.6 the work principle of an active molten salt thermal energy storage can be seen [9].

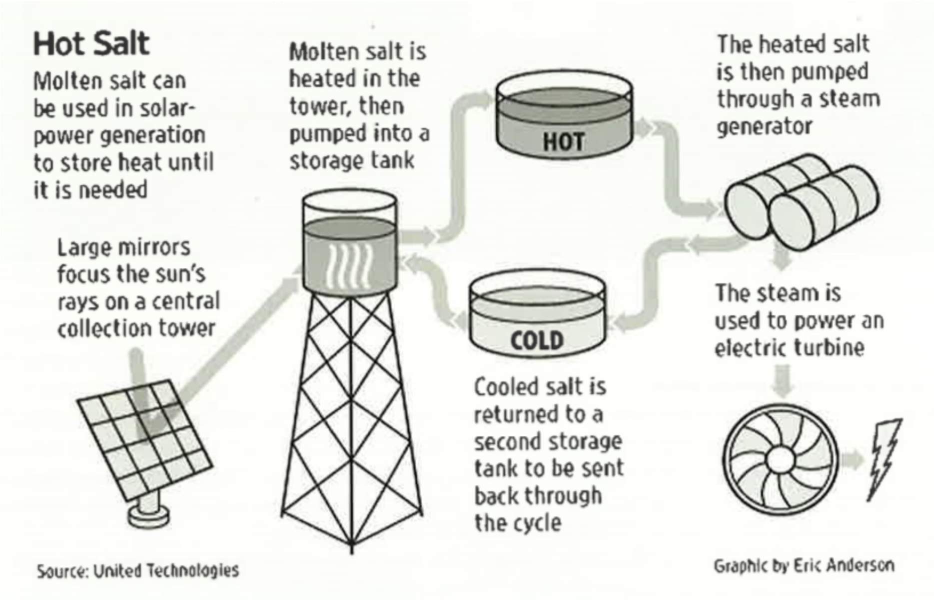


Figure 3.6: Thermal storage by molten salt [11]

Latent Heat Storage

Latent Thermal Energy Storage (LTES) utilizes the heat absorbed during phase transition of energy storage. Mainly there are three types of materials commonly used: Organic, inorganic and eutectic. Organic materials are popular but also more expensive. Inorganic materials include salt hydrates and metallic compounds. These are mostly cheaper but suffer from chemical decomposition after several cycles. The materials can experience transitions between solid, liquid and gas. The solid-liquid phase transitions are most commonly used due to storage capacity and ease of containment [9].

Reversible Chemical Reaction Heat Storage

The reversible chemical reaction heat thermal energy storage (CTES) is the third and last thermal energy storage system. Not the least it is the most efficient storage medium of the three. There are three main types of CTES systems: heat pump, heat pipe and heat of reaction systems. Chemical heat pumps utilize adsorption and desorption of a vapour/liquid into a solid substance. This reaction can be endothermic or exothermic. Chemical Heat Pipes (CHP) are very similar but it uses reactions based on acid and bases. Chemical Reaction Heat (CRH) operates by breaking down compounds into their constituent pars via an endothermic process. All these three storage systems are now in their research stage. They have a potential for long-term storage. The disadvantages are the precisely controlled temperatures and pressures [9].

3.2 Storage systems analysis

Section 3.1 gave an outline of the most used energy storage systems. These systems are, as has been said before, not all relevant for this project. The analysis is split into two parts, a small village with around 10 households which need a storage device with a capacity of 10kW and a bigger village with 100 households which need a 200kW storage device. In chapter 2 can be read more about the specifications of both villages. For both villages a schemetic overview of the best suitable storage options can be found in Appendix A. The data in this appendix are from multiple systems [9]. Of these systems the minimum, maximum, average value and quantity of each category are presented.

For the smaller village all the storage systems have been filtered on capacity, lifespan and efficiency. The capacity of all the systems should at least have a minimum of 10kW and a maximum average capacity of 2MW. This is because when the average of all the data is too high it is likely to say that these systems are not optimal for a small size solution. Next to this the average lifespan should be at least ten years and the average efficiency should be higher than 80%. There are four solutions that satisfy all these conditions: the Flywheel, Li-Ion battery, Supercapacitor and a Reaction Heat storage system. These are presented in a schematic view in figure A.1.

It can be concluded that when the village wants a storage system which provides energy during the renewable energy time shift it is advisable that the system should at least have a technical maturity of 'mature'. This is because the production is meant to be taken care by pacific companies and the maintenance by local citizens. This makes a reaction heat storage system and a supercapacitor less attractive. Also the environmental impact is of great importance. Most of the time when devices go to the islands they never leave it again.

The last two options, the flywheel and the Lithium Ion battery differ on a few subjects. The Lithium Ion battery has a lower self-discharge rate, lower capital cost and most importantly the application for which it is used is most of the time energy management. The flywheel on the other hand has a higher specific power, power density, higher cycle life, lower power capital cost and the environmental impact is much lower.

For the bigger island all the systems have been filtered in the same categories: capacity, lifespan and efficiency. The capacity of all systems should have at least a minimum of 200kW and an average capacity between 5kW and 50MW. This is for the same reason, when the average differs too much from the preferred capacity it is likely not the best option. Again the average lifespan should be at least ten years and the average efficiency should be higher than 80%. This comes to a total of five storage systems which could be an option for the bigger island. More detailed information can be found in figure A.2. Again the flywheel, lithium ion and supercapacitor are an option while the sodium sulpher and superconductor are new storage systems. The supercapacitor is not chosen for the same reason as in the small village. The Lithium Ion battery does not differ on many points but when compared to the sodium sulpher it costs less and has less environmental impact. When we compare the flywheel, sodium sulpher battery and the superconductor, the costs for the superconductor are at least ten times as high as the flywheel and the sodium sulpher battery costs even more. This makes it not a suitable option for the pacific. When we compare the last two options, the sodium sulpher battery and the flywheel, the impact on the environment and the technical maturity of the flywheel is better. Also it has a higher specific power, higher power density, higher efficiency, higher lifespan, higher cycle life and lower capital cost. While for the sodium sulpher battery the energy capital cost is a factor 40 lower and the used application is energy management. Which of these two options is more suitable for the bigger village it depends on the needs of the village. A combination of both should also be a reasonable option.

3.3 Worldwide used storage projects

3.3.1 Diesel generator

Currently most of the villages use diesel generators to produce electricity. When the Energy storage system is installed it should take over the tasks of the generator. The generator will be used as a backup in case of bad weather conditions. In figure 3.7 two examples are given of used diesel generators. It can be clearly seen that there are big differences in size. While the 10kW generator can be transported on a trailer at the back of a car, the bigger 200kW generator is completely different in size. This is also typical for the storage systems and important to keep in mind during the selection an design process.

Figure 3.7: 10kW Perkins diesel generator and 200kW John Deere diesel generator

3.3.2 Energy storage exchange project

The website www.energystorageexchange.org is made in 2011 by the Sandie Corporation in collaboration with the U.S. department of Energy [12]. The website is made for the exchange of knowledge concerning the current energy storage projects. Companies can share their newest projects here or update their existing projects. This interactive website has already registered 1306 projects worldwide with a total of 185 GWatt storage. These projects are not always built yet. For example they can be contracted or in building stage. In figure 3.8 a screenshot can be seen of the website to give an idea of the project.

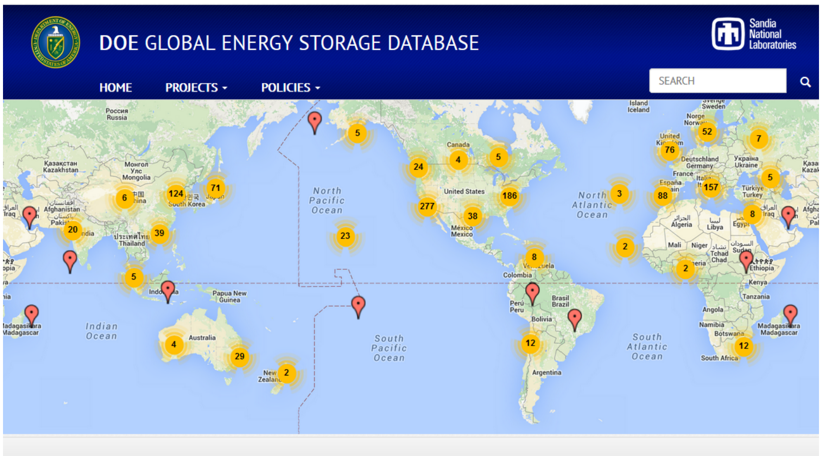


Figure 3.8: A screenshot of the website: www.energystorageexchange.com [12]

These projects have been filtered on their development status: "operational" and a minimum length of energy storage of four hours. These projects can be seen in table 3.1. Between brackets can the amount of projects be seen which are already used for renewable energy time shifting. The number in front of the brackets gives the total number of all projects which were filtered. Those projects are not used for renewable time shifting does not mean they cannot be used for it that is why they are included. Unfortunately not all the information on projects is complete so the total does not always match with the specific used method for the storage system.

	<250kW	250-999kW	1-10MW
Electro-Chemical	**41 (13)**	**12 (3)**	**33 (3)**
Flow Battery	12 (6)	1 (-)	2 (1)
Lithium Ion	13 (1)	2 (1)	8 (1)
Lead acid	7 (3)	5 (-)	-
Sodium Based	1 (1)	3 (1)	23 (1)
Nickel Based	3 (-)	-	-
Thermal Storage	**87 (2)**	**14 (1)**	**27 (5)**
Ice Thermal	84 (-)	8 (-)	12 (1)
Chilled water	1 (-)	5 (-)	11 (1)
Molten Salt	-	-	1 (1)
Pumped Hydro	**1 (1)**	-	-
Electro-Mechanical	-	**1 (1)**	**1 (1)**
Compressed Air	-	1 (1)	1 (1)
Hydrogen	**-**	**1 (1)**	**1 (-)**

Table 3.1: Used energy storage with: storage capacity >4 hours, operational status and between brackets used for renewable time shift.

In table 3.1 can be seen that the Electro-Chemical and Thermal Storage dominate the operational energy systems. Where Ice thermal storage, Flow batteries and Lithium Ion batteries dominate in the lower capacity region, the Sodium based battery, Ice thermal and Chilled water dominate in the high region. This matches the conclusion made in section 3.2 where in the lower region the Lithium Ion battery was found as one of the best options. The high use of Ice Thermal storage could be explained by the projects of Ice Energy® in California. They use energy stored in ice which will be used later for their air-conditioning. So there will not be any electricity gained out of the storage again. The same goes for chilled water energy storage system but then stored in water.

When the results in table 3.1 are compared with the results found in the earlier section even here in both the Lithium Ion and Sodium based battery are highly used. The use of thermal storage did not come out the previous section. This is because the Ice Thermal storage and chilled water storage do store energy in water or ice for the use of cooling systems and do not make electricity out of the energy.

3.3.3 Tesla Powerwall

The Tesla power wall came out in 2015 and is Tesla's newest invention. It is an lithium-ion battery which can be used for single households and can be connected to the electricity grid. The storage has a volume of 86cm by 130cm by 18cm and can store up to 7-10kWh. With an efficiency of 92% and a warranty of ten years it is one of the first home storage systems. The system is build in such a way that more capacity and more banks can be connected to create higher power and more energy. For a price of US$ 3000-3.500 (=6 360-7420 FJD) it is available this year. In the pacific this could be used for more than one household.

Chapter 4

Concepts

4.1 Concept Choice

This project is focusing on the small village of 10 households with an average size of 5 persons per household. This village has one community building and needs an energy storage system with a capacity of 10 kW. When all the possible energy storage systems are reviewed it can be concluded that it looks like the Lithium-Ion battery has the best specifications. In figure A.1 can be seen that the Lithium-Ion battery is strong on the specific energy [Wh/kg], energy density [Wh/m^3], efficiency, low self-discharge rate [%/day] and low energy capital cost [US$/KWh]. But a big point of interest is the environmental impact which is higher in the Pacific since most products never leave the Pacific islands after purchase. The production of the battery in the Pacific will be a problem because all components need to be bought overseas. That is not the case with a flywheel energy storage. Because of its material and the maturity of the system it is possible to produce in the pacific also the environmental impact is much lower then from a battery. This is why in this project the flywheel has been chosen to continue with.

4.2 The flywheel

The flywheel is a mechanical energy storage system that has been used for a long time. Most of the time it is used for a short energy storage to take the fluctuations out of the electrical grid. Because a flywheel is made of simple composites and metals it is considered as an environmental friendly way to store energy. The principle is that energy is stored in a rotating mass that speeds up. This increases the kinetic energy. By reducing the speed the kinetic energy is released and the flywheel can power the grid. A basic layout is given in figure 4.1. The energy that can be stored in a flywheel can be calculated by eq. 4.1.

$$W = \frac{I\omega^2}{2} \tag{4.1}$$

Where ω is the angular velocity $[rad/sec]$ and I is the moment of inertia $[kgm^2]$ which is known for a solid cylinder as eq. 4.2. From these equations can be seen that for the energy

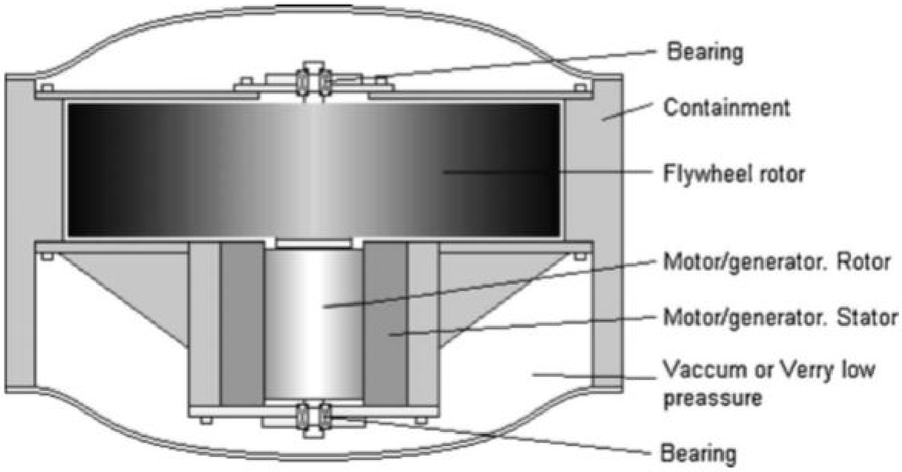


Figure 4.1: Basic layout of a flywheel energy storage [13]

three factors are important: the mass m, radius r and angular velocity ω.

$$I = \frac{1}{2}r^2m \tag{4.2}$$

When the flywheel rotor is designed it is important to know how much of the stored energy will be lost due to other factors. The biggest losses in most flywheels are in the bearings, air friction from the flywheel rotor and motor efficiency. To make these losses as small as possible or even cancel them. The design need to be improved. Most flywheels use a vacuum to cancel the air friction of the rotor but this requires more cooling. This is why most low speed flywheels use a low vacuum and not a high vacuum so the flywheel benefits from both worlds.

4.3 Requirements

Now that all these facts are known the requirements for the flywheel design can be listed. These can be seen in table 4.1. Also an explanation of each individual requirement is given.

- Size: The size of the load space of a trailer is 1.5m x 2m x 2m which will most likely be used for transport in rural areas.
- Weight: Most pickups can carry a trailer that has a maximum load of 4000 kg.
- Capacity & Energy stored: These specifications are described in section 2.3.
- Cost: The average paid cost in the Pacific are around US$ 0.40/kWh [14]. Combined with a depreciation of 5 years and a subsidy of 50% comes to the total cost of US$21000 (=44 520 FJD). The solar panels are estimated at 12 000 FJD (=US$5660) This makes the available budget 36 100 FJD (=US$17028).

	Requirements
Size	1.5m x 2m x 2m
Weight	prefer $<$4000kg
Capacity	10 kW
Energy stored	25 kWh
Cost	36 100 FJD (=US$ 17 028)
Production in the Pacific	Flywheel and casing
Maintenance	$>$5 years without any maintenance
Lifespan	$>$15 years
Environmental impact	80% should be recyclable

Table 4.1: Requirements for the flywheel to design

- Production in the Pacific: Flywheel and casing.
- Maintenance & Lifespan: In the rural area it is important that the maintenance period and lifespan are from a descent period due to the long travel times to the area.
- Environmental impact: ”What comes to the island never leaves the Pacific islands.” Is what local citizens say on Fiji. This makes it important that the storage system can be recycled after 15 years of use.

Chapter 5

Detailed Design

The design of a flywheel can be split in two different groups of components which needs to be designed separately. The first group is the flywheel rotor itself. Because this rotor gives the specifications for the second group, the shaft of the flywheel. The shaft will be designed afterwards. This shaft will support the rotor and makes sure no fluctuation in other planes are possible during the rotation. When all these components are designed losses can be calculated and the designs can be optimized with the newly gathered data.

5.1 Flywheel Design

For the design of the flywheel rotor two important decisions have to be made. These decisions concern the material and shape of the rotor. But not all the shapes are possible to make with each material. The four most common shapes and their materials can be seen in figure 5.1. In this figure is also the shape factor $K[-]$ of each flywheel shape listed. This factor gives combined with the maximum work strength σ_u $[Pa]$ and the density ρ_v $[kg/m^3]$, the energy per mass W^M $[J/kg]$ by equation 5.1.

$$W^M = K\frac{\sigma_u}{\rho} \tag{5.1}$$

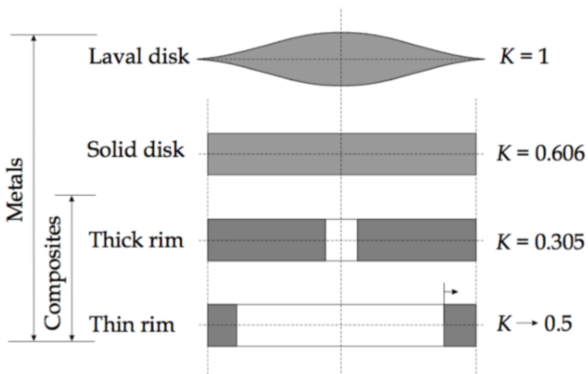


Figure 5.1: Shape factor of common flywheel shapes [15]

It is estimated that a maximum speed drop of 50% is allowed [13]. Since the working strength is related to the square angular speed ω^2. The active energy in the rotor is 75% of the total stored energy in the flywheel. From the requirements is known that this 'active energy' should be equal to 25 kWh. The maximum rotation speed for a low speed flywheel is 10 000 rpm [10]. This results in a minimum rotational speed of 5 000 rpm.

In this report three possible materials are covered: High tensile Steel AISI 4340, S2-glass and Carbon T1000. The two last materials are composites and are only applicable in the design of a thick or thin rim. The high tensile steel has also been tested in a Laval disk. The solid disk has been taken out of consideration since the Laval disk has all the advantages of the solid disk and less disadvantages. The specifications of all the three materials are listed in table 5.1. Where the maximum allowable working stress σ_u is calculated by dividing the yield strength by a safety factor n of 1.5 for metal material. For the composite materials the ultimate tensile strength is divided by a safety factor n of 2.0.

	Density ρ	Working Stress σ_u	Poisson ratio ν	Price
Steel AISI 4340	7830 $[kg/m^3]$	486 $[MPa]$	0.29	1.00 $[\$/kg]$
S2-glass	1920 $[kg/m^3]$	735 $[MPa]$	0.22	24.60 $[\$/kg]$
Carbon T1000	1520 $[kg/m^3]$	975 $[MPa]$	0.20	101.80 $[\$/kg]$

Table 5.1: Material specifications [source: www.matweb.com] & [13]

Thick Rim

There are two stress components that are important in the design of a flywheel rotor. The radial stress presented in eq. 5.2 and the hoop stress expressed by eq. 5.3 [13].

$$\sigma_r(r) = \frac{3+\nu}{8}\rho\omega^2\left(r_0^2 + r_1^2 - \frac{r_0^2 r_1^2}{r^2} - r^2\right) \tag{5.2}$$

$$\sigma_\theta(r) = \frac{3+\nu}{8}\rho\omega^2\left(r_0^2 + r_1^2 + \frac{r_0^2 r_1^2}{r^2} - \frac{1+3\nu}{3+\nu}r^2\right) \tag{5.3}$$

Where ρ is the mass density $[kg/m^3]$, ω is the rotational speed $[rad/sec]$, ν is the Poisson ratio, r_0 is the outer radius of the rotor $[m]$, r_1 is the inner radius of the rotor $[m]$ and r represent any radius within the rotor $[m]$. It can be seen from simulations that the hoop stress in a thick rim rotor is much higher than the radial stress. Next to this the hoop stress is at its maximum at r_1. When designing on the hoop stress of the inner radius the total stored energy W $[J]$ can be calculated by: $W = \frac{1}{2}I\omega^2$. Where I is the moment of inertia $[kgm^2]$. The best rotor configurations are calculated per material and are listed in tabel 5.2. Here the active energy storage is kept on 25kWh and a max. rotational speed of 10 000rpm.

	r_1 $[m]$	r_0 $[m]$	h $[m]$	m $[kg]$
Steel AISI 4340	0.050	0.260	7.645	12 370
S2-glass	0.468	0.617	1.5	1 460
Carbon T1000	0.217	0.594	1.5	2 190

Table 5.2: Configurations of a thick rim flywheel with $W = 25kWh$

Thin Rim

When designing a thin rim flywheel the procedure is similar to the hollow disk. The simplification which can be done is that due to the small crossing area the hoop stress can be taken as constant. Then eq. 5.3 can be simplified to eq. 5.4. [15]

$$\sigma_\theta = \rho r_1^2 \omega^2 \tag{5.4}$$

With this equation the best configurations of a cylindrical flywheel can be calculated and are shown in table 5.3.

	r_1 $[m]$	r_0 $[m]$	h $[m]$	m $[kg]$
Steel AISI 4340	0.135	0.23	13.10	11 920
S2-glass	0.49	0.59	2.29	1 488
Carbon T1000	0.44	0.54	3.85	1 804

Table 5.3: Configurations of a thin rim flywheel with $W = 25kWh$

Equal stress Design

The equal stressed disk is also known as the Laval disk. As has been shown in figure 5.1 this shape is due to material properties only made from metals. The main reason for the usage of the Laval disk is the high shape factor K which results in a higher energy storage per mass. The shape of an equally stressed disc can be obtained from the known equations of equilibrium and compatibility written in terms of stresses for linear, isotropic material with constant characteristics [16].

$$\frac{d(\sigma_r r h)}{dr} - \sigma_\theta h + \rho\omega^2 r^2 h = 0 \tag{5.5}$$

$$(\sigma_\theta - \sigma_r)(1+\nu) + r\frac{d\sigma_\theta}{dr} - \nu r\frac{d\sigma_r}{dr} = 0 \tag{5.6}$$

If $\sigma_r = \sigma_\theta = \sigma_u = const$ then from eq. 5.5 can be the profile $h(r)$ $[m]$ computed. This profile is given in eq. 5.7. Eq. 5.6 satisfies the conditions.

$$h(r) = h_c e^{-B\chi^2} \tag{5.7}$$

With dimensionless constants $B = \frac{\rho\omega^2 r_0^2}{2\sigma_u}$ and $\chi = r/r_0$.

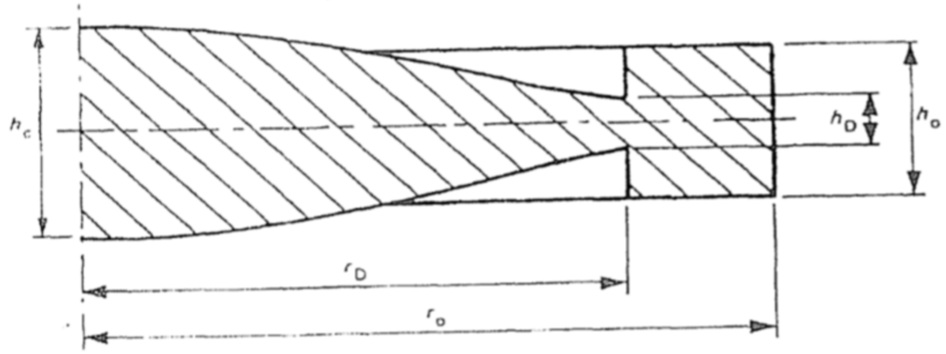


Figure 5.2: Constant-stress disc with constant-thickness outer rim [16]

The problem with this profile is that the radius goes to infinity. To create a boundary on the infinite profile a rim is needed at the end of the profile [16]. By creating a rim also the shape factor goes down. With $\beta = r_D/r_0$ as the ratio between the beginning of the rim and the outer radial and $\alpha = h_0/h_D$ as the ratio between the thickness of the rim to the connection point. Both can be seen in figure 5.2. The radial thickness β depends not only on B but also on α. [16]. This relation is shown in eq. 5.8 and by this the relation to shape factor can be determined and is shown in eq. 5.9.

$$\beta = \sqrt{\frac{1}{B\alpha}\left[\alpha - 1 + 2\sqrt{\frac{\alpha^2 B(B-1+\nu)}{(1-\nu)^2} + \frac{(\alpha-1)^2}{4}}\right] - \frac{1+\nu}{1-\nu}} \tag{5.8}$$

$$K = \frac{1 + [\alpha B^2(1-\beta^4)/2 - B\beta^2 - 1]e^{-B\beta^2}}{1 + [\alpha B(1-\beta^2) - 1]e^{-B\beta^2}} \tag{5.9}$$

In table 5.4 are the two best configurations for a steel equal stress disk with a rim and with a small rim been shown. The profile of the flywheel rotor with rim can be seen in figure 5.3.

	r_0 $[m]$	h_c $[m]$	m $[kg]$	h_D $[m]$	r_D $[m]$
Steel AISI 4340 with a small rim	1.0	0.737	1934	-	-
Steel AISI 4340 with rim	0.525	0.752	2140	0.195	0.391

Table 5.4: Configurations of a Laval disk flywheel with $W = 25\ kWh$

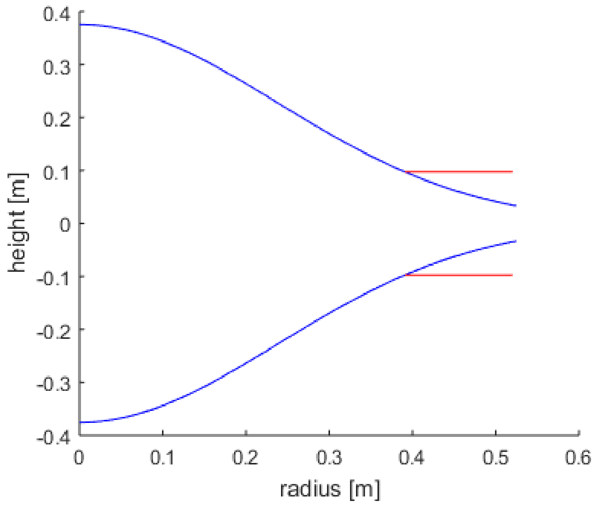


Figure 5.3: The profile which results from eq. 5.7 and in red added the rim.

Conclusion

A model of all the designs from this section can be seen in figure 5.4 on scale in comparison to a human. In the requirements is stated that the design should fit in a box of 1,5m x 2m x 2m. Only four of these designs will fit in this box. Since all four designs deliver the same power and work at the same rotation speed. Cost and production are the next criteria. Since the cost per kg differs a lot per material (table 5.1) but the total weight per design does not that significant. It can be concluded that the Steel AISI 4340 equal stress disk is the best option. The bigger rim will add some extra weight but decreases the area by a factor 4 this will make the production more easy. With the Laval disk with rim we will continue in this report.

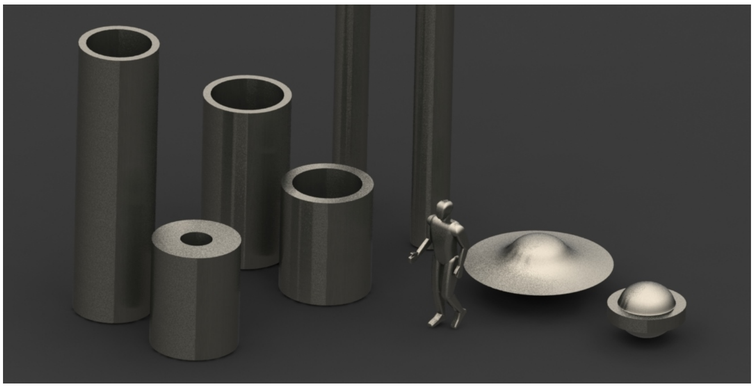


Figure 5.4: The flywheel designs with at the back four steel designs, then the two S2-glass plus human and in front the Carbon T1000 designs.

5.2 Shaft and Bearing Design

The flywheel can be positioned horizontal and vertical. Both configurations have no influence on the efficiency or stored energy of the flywheel. But for safety reasons it can be reasoned to place the flywheel underground. In case of a collapse the freed energy will be lost in the ground and not in flying particles through the air. For this situation a vertical configuration is more save since the swing area is all covered by ground. A vertical shaft has two options it can go through the flywheel or connects the top and bottom by two different shafts. The single shaft is the most common used shaft. It is easy in production and the chances for a misalignment in the flywheel are low. The big disadvantage is that for an equal stressed disk the allowed stress is halved when the shaft goes through the flywheel [16]. Two shafts on both ends do not have this disadvantage and so the flywheel does not have to increase in size.

Bearing Selection

In a flywheel design a lot of different types of bearings can be used. For high speed flywheels it is recommended to use magnetic or superconducting bearings to reduce the high frictional losses. Since this design is a low speed there are less frictional losses and mechanical bearings become also an option. Capital cost and simple production methods are of great importance in this design and both are lower for mechanical bearings. For this reason mechanical bearings are chosen for this project.

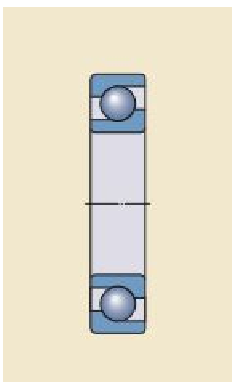


Figure 5.5: Angular contact bearing [17]

The most important requirements are the rotational speed of 10 000 rpm and the estimated axial load of 25 kN from gravity force of the flywheel rotor and connected components. From the SKF catalogue [17] can be seen that, for a high axial load and high rotational speed, angular contact ball bearings are the most recommended bearings. This bearing is shown in figure 5.5. The angular contact bearings are divided in two types: Single row angular contact ball bearings and Four-point contact ball bearings [17].

Both the single row bearing and the four-point bearing types have bearings that fullfill all the recommendations. To compare these bearings the resulting friction is important. This can be calculated by eq. 5.10 [17]. Where M $[Nmm]$ is the frictional moment, μ $[-]$ is the constant coefficient of friction for the bearing, P $[N]$ is the equivalent dynamic bearing load and d $[mm]$ is the bearing bore diameter. The coefficient of friction is slightly higher for the four-point bearing but the bearing bore is smaller.

$$M = 0.5\mu Pd \tag{5.10}$$

From this equation can be concluded that the best option for the bearing is the four point contact ball bearing with a bearing bore diameter of 25mm, QJ 305 N2MA. More specifications can be found in table B.1 in appendix B.

Shaft Design

For the shaft design it is important to know the torque and moment force which are applied on the shaft. Since it is decided to make use of a vertical shaft the moment forces can be assumed as zero. The required torque force applied on the shaft can be calculated by eq. 5.11. Where $P\ [W]$ is the power which needs to be delivered to the households and is known from the requirements of the design and $\omega_{min}\ [rad/sec]$ is the minimal operational angular rotation speed. This results in a torque force $T = 19\ Nm$

$$T_m = \frac{P}{\omega_{min}} \tag{5.11}$$

When the torque is known, stresses can be calculated by the resulting torque and momentum on the shaft. These relations are given in eq. 5.12 and eq. 5.13 [18].

$$\sigma_a = \left[\left(\frac{32K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}T_a}{\pi d^3}\right)^2\right]^{1/2} \tag{5.12}$$

$$\sigma_m = \left[\left(\frac{32K_f M_m}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}T_m}{\pi d^3}\right)^2\right]^{1/2} \tag{5.13}$$

With K_f and $K_{fs}\ [-]$ as the fatigue stress concentration factors, $M_m\ [Nm]$ the mid-range applied momentum, $T_a\ [Nm]$ the alternating applied torque and $T_m\ [Nm]$ the mid-rage applied torque. To see if the stresses are not fatigue for the shaft there are different criteria that can be used. This research will focus on the Gerber criteria. The Gerber criteria is a measure of the average behaviour of ductile materials on fatigue resistance. For the shaft design the Gerber failure criteria are being used. The criteria is given in eq. 5.14 [18].

$$\frac{n\sigma_a}{S_e} + \left(\frac{n\sigma_m}{S_{ut}}\right)^2 = 1 \tag{5.14}$$

With n the factor of safety, $\sigma_a\ [Pa]$ the alternating stress amplitude, $S_e\ [Pa]$ the endurance limit, σ_m as the midrange stress and $S_{ut}\ [Pa]$ as the ultimate Tensile strength. The endurance limit is the ultimate tensile strength of the shaft relative to the amount of rotations it will make during its life. The more rotations the shaft makes the lower the endurance limit but it can be assumed that the endurance limit goes to 50% of the ultimate tensile strength when the rotations are between 10^6 and ∞. With the equations for the alternating stress amplitude and midrange stress given in eq. 5.12 and eq. 5.13 [18]. The Gerber criteria eq. 5.14 can be rewritten into eq. 5.15.

$$d = \left(\frac{8nA}{\pi S_e}\left\{1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}}\right)^2\right]^{1/2}\right\}\right)^{1/3} \tag{5.15}$$

With:

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} \tag{5.16}$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} \tag{5.17}$$

The equation can be simplified by the assumption that the load and torque are constant during the operation of the flywheel. In reality these forces will fluctuate but it will fluctuate so slow that it can be assumed that the shaft behaves as it is under constant worst case conditions. This makes $M_m = 0$ and $T_a = 0$. Because of the vertical shaft $M_a = 0$. This results in $A = 0$ and $B = \sqrt{3}K_{fs}T_m$. Which will results at last in eq. 5.18.

$$d = \left(\frac{16n\sqrt{3}K_{fs}T_m}{\pi S_{ut}}\right)^{1/3} \tag{5.18}$$

Now that the equation is simplified the shaft can be designed. The material which is used for the shaft will be SAE 1020 HR. With a tensile strength $S_{ut} = 380MPa$. For production reasons a filling with a radius of 1mm is assumed to be reasonable. From figure 5.6 the value for K_{ts} can be found. To find the K_{fs} in figure 5.7 q_s can be found. q_s and K_{ts} give in equation 5.19 K_{fs}. With this d can be calculated from 5.18.

$$q_s = \frac{K_{fs} - 1}{K_{ts} - 1} \tag{5.19}$$

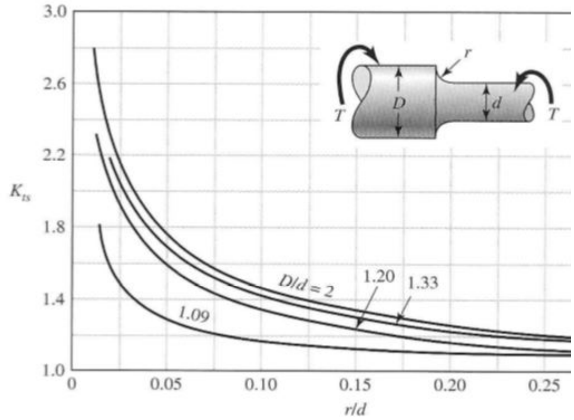


Figure 5.6: K_{ts} to the shaft dimensions [18]

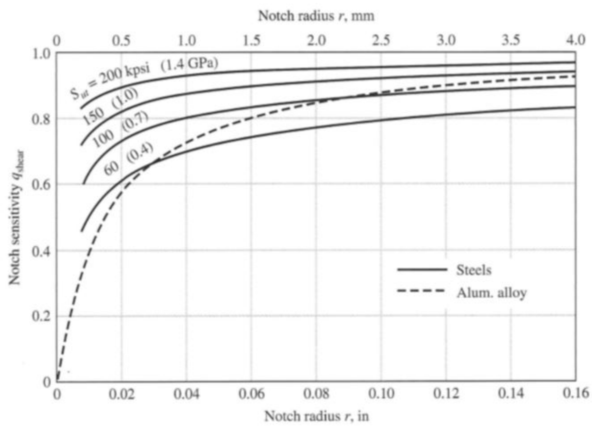


Figure 5.7: evaluation q_s factor [18]

From these figures and equations can be concluded that the shaft diameter should be at least 19.3mm. These calculations are done with a radius filling of 1mm and a safety factor of 10. Because 19.3 mm is smaller than the inner bearing diameter of 25mm. The shaft design will be based on the bearing. The shaft dimensions are presented in app. C with the same radial, safety factor and factor between d/D. It is safe to use.
Next to stress, buckling can be a problem. The maximum force before the risk of buckling occurs can be calculated by eq. 5.20. Here F_{max} $[N]$ is the maximum applied force before the chance of buckling occurs, E $[Pa]$ is the youngest modulus, D $[m]$ is the outer diameter, d $[m]$ is the inner diameter and L $[m]$ is the unsupported length. It is assumed that the shaft is a cylinder since the bottom of the shaft is only supported by the bearing and not on the inner diameter.

$$F_{max} = \frac{\pi^3 * E * (D^4 - d^4)}{32 * (0.5 * L)^2} \tag{5.20}$$

This results in a F_{max} that is a factor thousand higher than the gravity force of the flywheel. It can be concluded that no buckling occurs.

Flange

To connect the shaft with the flywheel rotor a flange is needed. The flange distributes the torque force from the shaft through the flywheel by four bolts. The requirements of the flange can be seen in app. C. The flange is designed following the usual proportions for a flange [19]. These can be seen in figure 5.8. Here can be seen that the outside diameter of the hub $D = 2d$, the length of the hub $L = 1.5d$, the pitch circle diameter of bolts $D_1 = 3d$, the outside diameter of flange $D_2 = 4d$ and the thickness of flange is $t_f = 0.5d$. At last there is chosen for a four bolt to ensure a good connection.

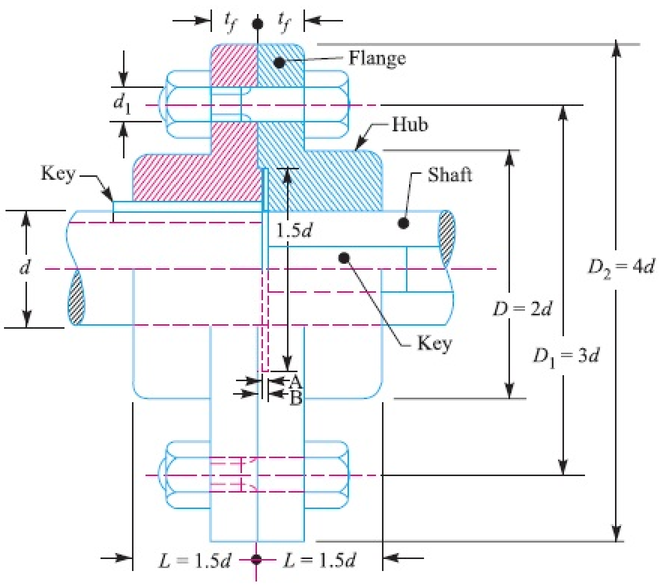


Figure 5.8: Typical flange proportions [19]

To check if these usual proportions are adequate it is important to know if this flange can manage the stresses that result out of the given torque. This can be done by five formulas eq. 5.21 - 5.25 [19]. The first formula is based on the torque that is transmitted through a hollow shaft. It is assumed that the hub is a hollow shaft and so this formula can be used. Here τ_c is the allowable shear stress for the flange material. For commonly used cast iron $\tau_c = 8MPa$ [19].

$$T_{1max} = \frac{\pi}{16} \tau_c \left(\frac{D^4 - d^4}{D}\right) \tag{5.21}$$

The second and third equation are based on the key which is inside the shaft. After fixing the length of the key in the shaft the induced shearing and compressive stresses may be checked. Here the maximum torque is calculated based on the maximum of these two stresses. Here $L\ [m]$ is the length of the key, $w\ [m]$ is the width of the key, $\tau_k\ [Pa]$ is maximum allowable shear stress for the key material. For most key materials $\tau_k = 40MPa$. [19].

$$T_{2max} = L\ w\ \tau_k\ d/2 \tag{5.22}$$

The third equation is based on the compressive stresses. Here t is the thickness of the key and σ_c is the allowable compressive stress for the key material. Which can be assumed for most key materials $\sigma_c = 80MPa$ [19].

$$T_{3max} = L\ t/2\ \sigma_c\ (d/2) \tag{5.23}$$

The fourth equation is based on the shearing stresses within the flange. The torque is the circumference of hub, times the thickness of the flange, times the shear stress of flange times the radius of the hub.

$$T_{4max} = \pi\ D\ t_f\ \tau_c\ D/3 \tag{5.24}$$

The last equation to check is the maximum torque the bolts can transmit before failure occurs. Here d_1 is the diameter of the bolt, $\tau_b\ [Pa]$ is the maximum allowed shear stress in the bolt and n is the number of bolts that is used in the flange.

$$T_{5max} = \frac{\pi}{4}(d_1)^2\ \tau_b\ n\ D_1/2 \tag{5.25}$$

Now that we have equations for the maximum torque in the hub, key, flange and bolts the results can be calculated. These are listed in table 5.5. It can be concluded that the standard proportions of the hub are thick enough for this design with a safety factor of at least 3. Which suits perfect for this project.

$\mathbf{T_{needed}}$	T_{1max}	T_{2max}	T_{3max}	T_{4max}	T_{5max}
19 Nm	386 Nm	61.4 Nm	153.6 Nm	2470.6 N m	452.4 Nm

Table 5.5: Maximal torque forces in the different parts of the flange

5.3 Tolerances

The tolerances are an important part of the design. When tolerances are not used in the right way it can lead to failure during assembly or parts getting loose during use of the product. The table for the selection of fits and hole basis system [18] is used to find the right tolerances for the project. When assembling the components, the first components to connect are the shaft with the flange. Because this part will transmit all the torque forces from the motor to the flywheel a heavy press fit is needed. A heavy press fit means that the tolerances are so close to each other that the two pieces only fit when a heavy press is used. This results in the tolerances which are listed in table 5.6. For the connection between the flange and the shaft a heavy push fit is chosen. This means the tolerances slightly not fit. But can be connected by an easy tick with a hammer or easy controlled press. The last fit is that of the flywheel with the flange. This fit is not necessary for transmitting the torque. This is done by the bolts which connect the flange with the flywheel. To prevent an over-constrained situation where the fit blocks the bolts from fitting the tolerances of this connection is a course tolerance. All these tolerances can be seen in table 5.6.

	Dimension	Fit	Basic size	Min. Tol.	Max. Tol.
Flywheel	Flange connection	Course tolerance	128 mm	0 μm	+250 μm
Flange	Outer diameter	Course tolerance	128 mm	-200 μm	-450 μm
	Inner diameter	Heavy press	32 mm	0 μm	+43 μm
Shaft	Top diameter	Heavy press	32 mm	+25 μm	+59 μm
	Bottom diameter	Heavy push	25 mm	+21 μm	+28 μm
Bearing	Inner diameter	Heavy push	25 mm	0 μm	+15 μm

Table 5.6: Minimum and maximum tolerances per part of the shaft design

5.4 Losses

In section 4.2 the loss components have already been discussed. Now the flywheel is designed these losses can be estimated and when they are out of proportion changes have to be made in the design. First an estimation of the air friction will be made and after this the friction in the bearing will be calculated and conclusions will be made.

Air friction

The friction which is caused by the speed difference between the flywheel rotor and the air can cause high frictional losses. The equation which is known for a rotating circular disk in air is presented in equation 5.27 [20]. The flywheel in this design is not a circular disk but the values give an estimation of the proportion of the losses. This is why only global conclusions can be made later when a more detailed design is made better simulations are needed. In the equation the power loss $P_{a,l}$ $[W]$ with ρ_a as density of a gas $[kg/m^3]$, β_a the dynamic viscosity of the gas $[Pa \cdot s]$, ω the angular speed $[rad/s]$, r radius of the flywheel $[m]$ and h the height of the flywheel $[m]$.

$$P_{a,l} = 0.04 \cdot \rho_a^{0.8} \cdot \beta_a^{0.2} \cdot (\omega \cdot r)^{2.8} \cdot (2 \cdot r)^{1.8} \cdot (\alpha + 0.33) \tag{5.26}$$

With:

$$\alpha = \frac{h}{2 \cdot r} \tag{5.27}$$

To compare four different gases it is assumed that the temperature in the flywheel stays at 50°C. The four gases which have been chosen are Air, Ammonia, Helium and Hydrogen. These gases are chosen for their low density and low viscosity at 50°C. In figure 5.9 can be seen that when the pressure is lowered the power loss will decrease. It can also be seen that when the flywheel rotates at atmospheric pressure in air; the power loss will be 261kW. This is over 10 times the storage capacity of the system. An combination of other gases and lower pressure brings a solution to the problem and could bring the losses to a low level. In figure 5.10 shows the air friction depending on the rotational speed. For this example is chosen for a Helium gas at a pressure of 0.001 bar. This is within the range of mechanical pumps which can go to 10^{-5} bar. It can be concluded from this graph that when the motor stops the flywheel will slow down in 48 hours from maximum speed to the minimum speed. 6.1 kWh is lost due to air friction. This is an average of 0.13 kW. Note that hydrogen looks a better option than Helium although the safety with pure Hydrogen is a big issue and it is not advised to used. Helium is not flammable like Hydrogen and is in this case a better option.

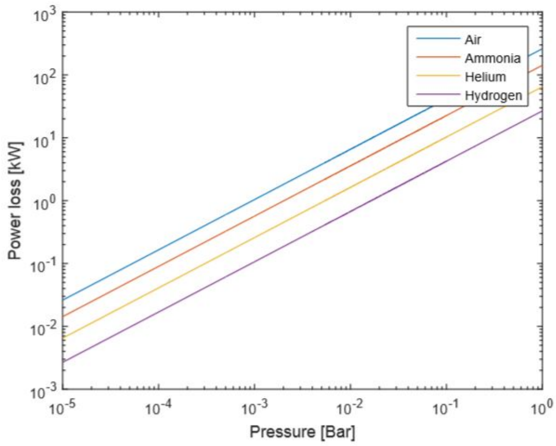


Figure 5.9: Extra torque force due to the air friction.

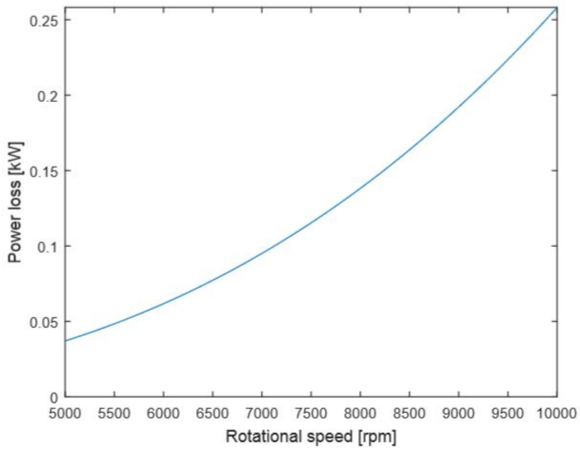


Figure 5.10: Air friction on flywheel Helium at 10^{-3} bar.

Bearing losses

Since in the previous section is concluded that low pressure conditions are needed around the flywheel rotor. The bearing needs to be sealed to keep the bearing clean and to protect the lubricate from flowing out of the bearing due to pressure differences. For an angular contact ball bearing three sealing options are provided by SKF [17]. The first option is placing a shield between the inner and outer cylinder of the bearing. Shields are made of sheet steel and cover both sides of the bearing. The second option are contact seals. Contact seals have the same position as shields. Seals are made of NBR and are reinforced with a sheet steel insert. The last option is an external sealing which will seal the bearing area around the bearing so no sealing is needed between both rings. Which of these options is more beneficial should be investigated in a later and more detailed design.

The second biggest loss of the flywheel will be the bearing on the bottom shaft. In the section shaft design eq. 5.10 has been used to pick a bearing with the lowest loss. This is a rough estimation of the frictional moment. The exact frictional moment can be calculated by summiting the rolling frictional moment M_{rr}, sliding frictional moment M_{sl}, frictional moment of seals M_{seal} and the frictional moment of drag losses M_{drag}. The equations for all the individual losses can be found in the catalogues of SKF [17]. The calculated total frictional moment results in a $M = 1.41Nm$. From this can be calculated that the power loss will be 1.4kW. Which is an increase of 5.6% of the stored energy per hour. This is a high value since in section 5.5 will be concluded that the axial force on the bearing is too low. The later advised magnetic support of 87.5% of the weight of the flywheel would bring the losses down to 170W. Which is a reduction of the max stored energy by 0.68% this is acceptable.

5.5 Safety and Lifetime

Safety

Safety is a big issue when energy is stored in the flywheel. Since the flywheel rotates with 10 000 rpm, with a radius of 0.525 m the outer particles of the flywheel have a speed of 550 m/s. To ensure flywheel componants would not go flying around in case of a failure, safety measures are needed. A good first safety layer around the flywheel would be the vacuum surrounding. But since a low pressure around the flywheel rotor is an option a second layer of safety is needed. There is chosen to place the flywheel in a concrete case. This case will be placed underground so all the energy that come lose when a failure occurs will be covered. Particles will first hit the concrete wall and second the loose ground behind the wall. This will fully protect the citizens of the village. The situation can be seen in figure 6.1.

Bearing lifetime

The last point of interest is the life time of the bearing. This can be calculated by the SKF rating life [17]. This method is done by equation 5.28 which gives the life time of the bearing under a certain load. Here a_1 is the life adjustment factor for reliability which is 0.55 for a reliability of 96%, a_{skf} is the SKF life modification factor which can be found trough tables in the catalogue of SKF [17], C is the basic dynamic load rating $[kN]$ and P the equivalent dynamic bearing load $[kN]$. For a vertical shaft with only a axial load, the equivalent dynamic load can be calculated by $P = 1.07F_a$.

$$L_{nm} = a_1 \cdot a_{SKF}(\frac{C}{P})^3 \tag{5.28}$$

From the SKF rating life can be concluded that this bearing under the load of the flywheel rotor will live at least 8.8 million revolutions. For most bearings this would be a good amount but since the flywheel rotates with 5 000-10 000 rpm. The life time is less than a day. Other bearing options discussed in earlier sections were also analysed but did not have a higher life time. In most flywheels with mechanical bearings the bearing is supported by permanent magnets. These magnets support the axial force and by this lowers the equivalent dynamic bearing load P. A support of 87.5% of the load would give the bearing a lifetime of 418 days. The magnetic support should be investigated in a further design.

5.6 Dynamic model

The dynamic model of the flywheel can be used to simulate the behaviour of the flywheel and to design a controller for the rotational movement. One of the important simulations is to look for eigenfrequency's and eigenmodes. In section Losses are the two most important losses discussed. These where the bearing frictional losses and air frictional losses. For the dynamic model the pressure around the flywheel rotor is taken at 10^{-3} bar Helium and the magnetic supported on the bearing will be 22 kN. This makes the resulting axial force from 25kN reduced to 3kN on the bearing. The bearing frictional moment force M_b $[Nm]$ can be seen in eq. 5.29. This formula is estimated by given data from the catalogue of SKF [17] and the function is derived from these data point.

$$M_b = 5.4 \cdot 10^{-6} \cdot \omega + 0.106 = \alpha\omega + \beta \tag{5.29}$$

With the equation $T = \frac{P}{\omega}$ eq. 5.27 can be rewritten and simplified to eq. 5.30. Where T_{air} $[Nm]$ is the torque force by the air friction on the flywheel rotor.

$$T_{air} = 0.1393 \cdot \rho_a^{0.8} \cdot \beta_a^{0.2} \cdot r^{4.6} \cdot (\alpha + 0.33) \cdot \omega^{1.8} = \gamma \cdot \omega^{1.8} \tag{5.30}$$

When the summation of moment forces around the ax of the flywheel rotor is made. Eq. 5.32 can be obtained. Where $M(t)$ $[Nm]$ is the resulting moment from the electric motor.

$$\sum M = M(t) - M_b - T_{air} \tag{5.31}$$

$$J\ddot{\theta} = M(t) - \alpha\omega - \beta - \gamma \cdot \omega^{1.8} \tag{5.32}$$

This can be rewritten to the standard form eq. 5.33.

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} \dot{\omega} \\ \dot{\theta} \end{bmatrix} = \begin{bmatrix} \alpha\frac{1}{J}\omega - \beta\frac{1}{J} - \gamma\frac{\omega}{J}\sqrt[5]{\omega^4} \\ \omega \end{bmatrix} + M(t)\begin{bmatrix} \frac{1}{J} \\ 0 \end{bmatrix} \tag{5.33}$$

With moment of inertia $J = 219.3kg \cdot m^2$ for the flywheel rotor, found by Solidworks. The dynamic model will not be used in this research but can be used for a continued research to give an assumption for the behaviour of the flywheel.

Chapter 6

Conclusions and Recommendations

6.1 Conclusions

A rough design of the flywheel has been made. The design has a Laval disk with the height of 0.752 m, radius of 0.525 m and mass of 2140 kg. This flywheel rotor is held in place by two shafts which are connected by flanges to the flywheel. Both can be seen appendix C. The bearing QJ 305 N2MA is chosen and fullfills all the specifications except the lifetime, see recommendations. The losses in the flywheel are high but with the modification of a vacuum and magnetic support on the bearing they can be dealt with. The safety of the flywheel is guaranteed by an underground concrete box so flying particles in case of a failure are captured in the ground. After this research can be concluded that designing a flywheel meeting the specifications is possible. Still, further research is needed to make the flywheel design more complete and more efficient. Compared to the Tesla Powerwall the flywheel will be more expensive, heavier and bigger than the Tesla Powerwall. On the other hand the flywheel storage creates employment in the Pacific, has far less impact on the environment and has potential to a longer life time.

6.2 Recommendations

After this research a few recommendations can be made. These are advised to be the next step to make sure that the flywheel would be beneficial in use.

- Losses: A more detailed research needs to be done to calculate all the losses and to calculate the efficiency of the flywheel.
- Bearing lifetime: The bearing lifetime is far too low. An option to solve this problem is by a magnetic support on the bearing.
- Motor design: The motor design needs much more detail to see what is needed and what are the costs. The dynamic model is a good start. Also a stability analysis is needed.
- Lastly, the interaction between the flywheel and the solar panels should be evaluated.

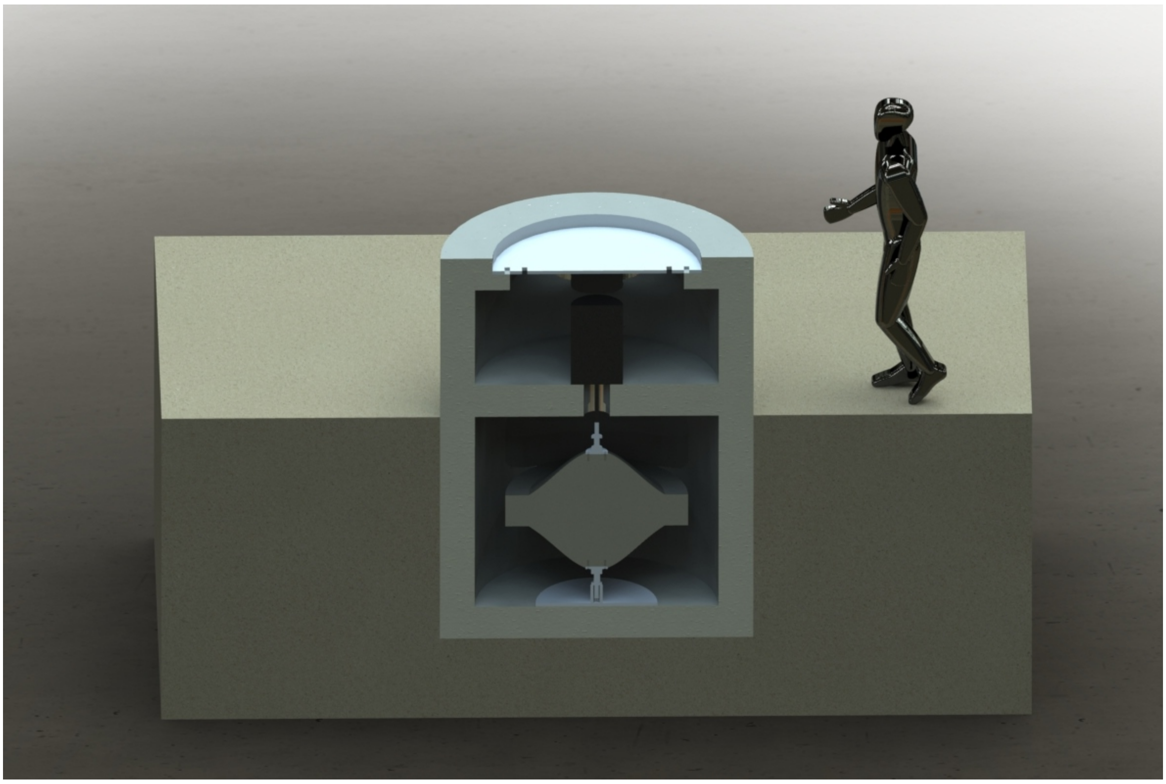


Figure 6.1: Flywheel installed on location

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Appendix A

Data of energy storage systems

		FES Flywheel	Li-Ion Lithium Ion	ELDC Supercapacitor	CTES Reaction Heat
Specific Energy [Wh/kg]	min	5	30	0,007	250
	max	200	300	85,6	250
	average	58,16	143,9	16,36	250
	numb	19	29	25	1
Energy Density [KWh/m3]	min	0,25	94	1	300
	max	424	500	35	300
	average	95,16	290	13,77	300
	numb	12	17	9	1
Specific Power [W/Kg]	min	400	8	5,44	
	max	30000	2000	100000	
	average	6592	606	8930,44	
	numb	12	22	24	
Power Density [KW/m3]	min	40	56,8	15	
	max	2000	800	4500	
	average	816,29	366	921	
	numb	7	4	5	
Efficiency [%]	min	70	70	65	75
	max	96	100	99	100
	average	89,36	89,8	91,33	87,5
	numb	22	17	12	2
Lifspan [yr]	min	15	2	5	
	max	20	20	20	
	average	17,5	10,13	11,43	
	numb	4	8	7	
Cycle Life [cycles]	min	10000	250	10000	
	max	100000	10000	1000000	
	average	41100	1018	302308	
	numb	10	19	13	
Self-Discharge Rate %/day	min	24	0,03	0,46	
	max	100	0,33	40	
	average	64,61	0,158	18,64	
	numb	7	9	7	
Scale [MW]	min	0,001	0	0	0,01
	max	10	3	5	1
	average	1,96	0,93	0,52	0,505
	numb	25	12	23	2
Energy Capital Cost [US$/KWh]	min	200	200	100	10,9
	max	150000	4000	94000	137
	average	12454	1110	19866	73,74
	numb	26	15	16	2
Power Capital Cost [US$/KW]	min	30,28	175	100	
	max	700	4000	800	
	average	296,14	2325	321	
	numb	16	9	10	
Application		Medium Power Q	Small/Medium Energy manag	Small/Medium Power Quality	Small/Medium Energy manag
Technical maturity		Mature	Mature	Proven	Proven/Develop
		6	6	4	2
Environmental Impact		Very Low	High/Med	Very Low	Uncertain
		1	5	1	2

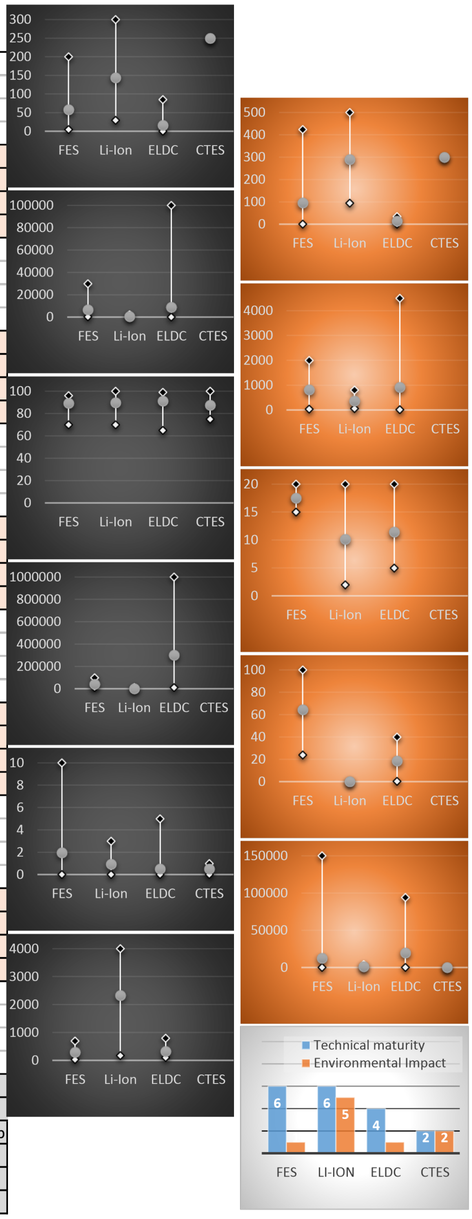
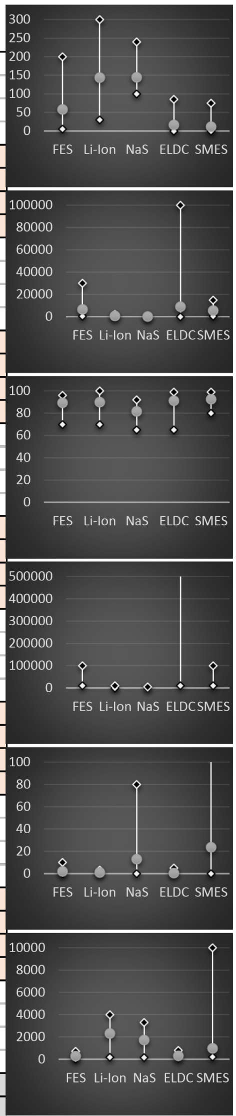


Figure A.1: Data of a FES, Li-Ion, ELDC and LTES storage system for 10 households [9]

		FES Flywheel	Li-Ion Lithium Ion	NaS Sodium Sulpher	ELDC Supercapacitor	SMES Super conductor
Specific Energy [Wh/kg]	min	5	30	100	0,007	0.27
	max	200	300	240	85,6	75
	average	58,16	143,9	145	16,36	11,79
	numb	19	29	10	25	11
Energy Density [KWh/m3]	min	0,25	94	150	1	0,2
	max	424	500	345	35	13,8
	average	95,16	290	213	13,77	4,99
	numb	12	17	10	9	12
Specific Power [W/Kg]	min	400	8	14,29	5,44	500
	max	30000	2000	260	100000	15000
	average	6592	606	176	8930,44	5600
	numb	12	22	9	24	5
Power Density [KW/m3]	min	40	56,8	1,33	15	300
	max	2000	800	50	4500	4000
	average	816,29	366	21,8	921	1457,5
	numb	7	4	5	5	4
Efficiency [%]	min	70	70	65	65	80
	max	96	100	92	99	99
	average	89,36	89,8	81,5	91,33	92,45
	numb	22	17	21	12	11
Lifspan [yr]	min	15	2	5	5	20
	max	20	20	20	20	30
	average	17,5	10,13	12,22	11,43	25
	numb	4	8	9	7	4
Cycle Life [cycles]	min	10000	250	1000	10000	10000
	max	100000	10000	4500	1000000	100000
	average	41100	1018	2771	302308	68000
	numb	10	19	12	13	5
Self-Discharge Rate %/day	min	24	0,03	0	0,46	1
	max	100	0,33	20	40	15
	average	64,61	0,158	8,01	18,64	7,5
	numb	7	9	5	7	3
Scale [MW]	min	0,001	0	0,01	0	0,01
	max	10	3	80	5	200
	average	1,96	0,93	13,1	0,52	23,56
	numb	25	12	16	23	22
Energy Capital Cost [US$/KWh]	min	200	200	150	100	500
	max	150000	4000	900	94000	1080000
	average	12454	1110	387	19866	125488
	numb	26	15	14	16	10
Power Capital Cost [US$/KW]	min	30,28	175	150	100	196
	max	700	4000	3300	800	10000
	average	296,14	2325	1736	321	981,56
	numb	16	9	13	10	16
Application		Medium Power Q	Small/Medium Energy manag	Medium Large Energy manag	Small/Medium Power Quality	Medium/Large Power Quality
Technical maturity		Mature 6	Mature 6	Proven 4	Proven 4	Proven 4
Environmental Impact		Very Low 1	High/Med 5	Medium/Low 3	Very Low 1	Low 2



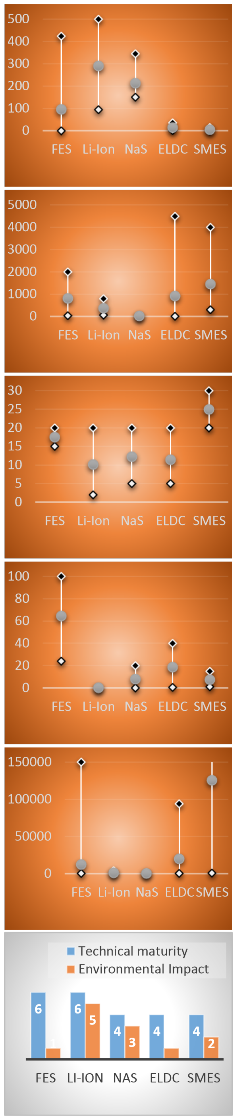


Figure A.2: Data of a FES, NaS, SMES and LTES storage system for 100 households [9]

Appendix B

Bearing QJ 305 N2MA

Principal dimensions	
d	25 mm
D	62 mm
B	17 mm
Basic load ratings	
Dynamic C	42.5 kN
Static C_0	30.0 kN
Fatigue load limit	
P_u	1.27 kN
Speed ratings	
Ref. Speed	15 000 rpm
Limiting speed*	20 000 rpm*
Mass	
m	0.29 kg
Dimensions	
d_1	34 mm
D_1	49 mm
$r_{1,2min}$	1.1mm
a	30 mm
Abutment and fillet dimensions	
d_{a-min}	32 mm
D_{a-max}	55 mm
r_{a-max}	1 mm

Table B.1: Bearing specifications [17]

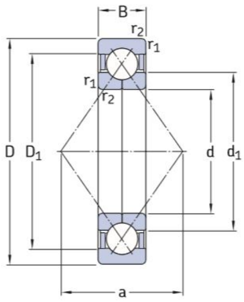


Figure B.1: Basic design [17]

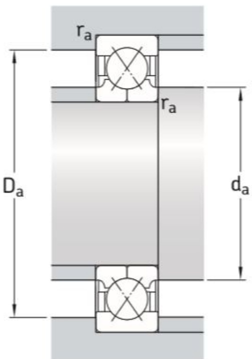


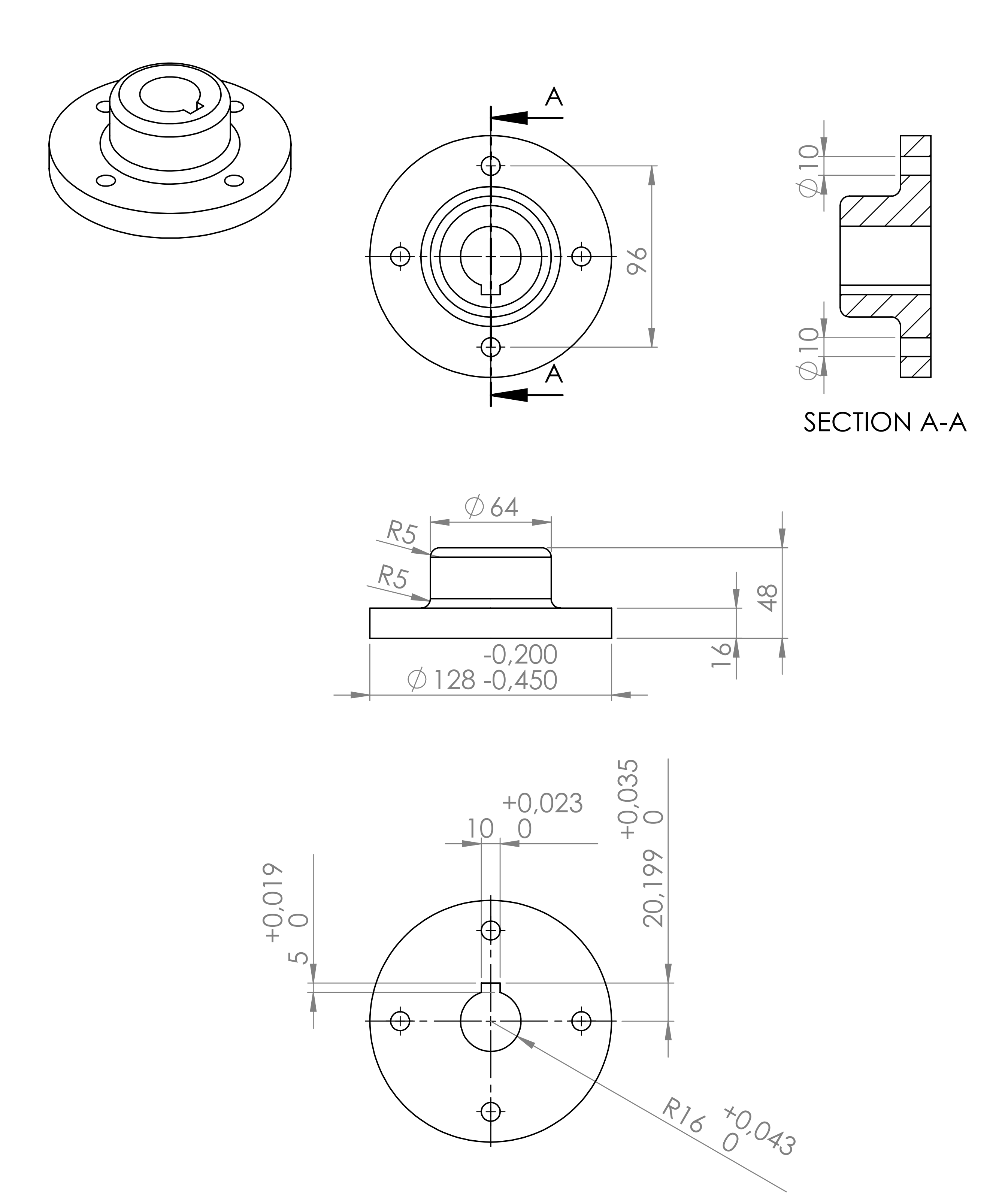
Figure B.2: Abutment and fillet dimensions [17]

*The limiting speed should be reduced to 70% when a vertical shaft is used.

Appendix C

Drawings Shaft and Flange

The drawings start on the next page.



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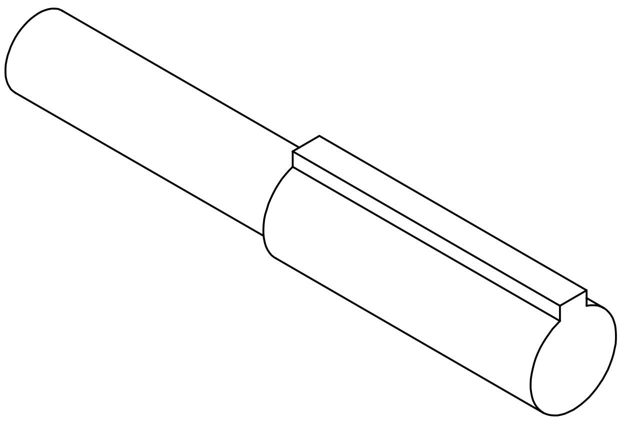
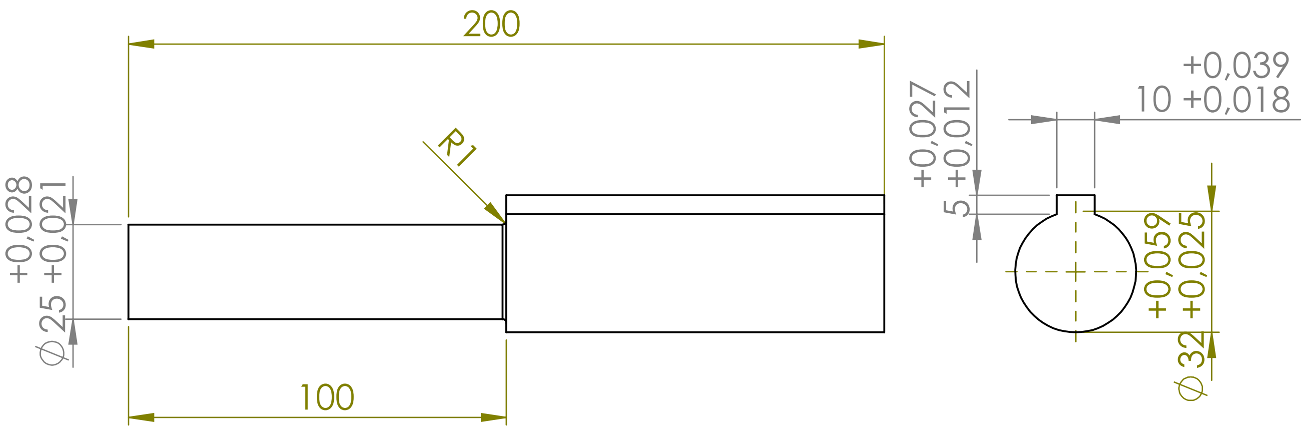
		CHECKED	--	SCALE	1:3
MATERIAL: Cast Iron --	SURFACE FINISH: --	TITLE: Flange			REV. 01
		DRAWING NO.: 1			
UNIVERSITY OF TWENTE.		FILE / PART NAME: flange			A4
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1	





PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	Niels Leijtens	DATE	27-10-2015
		CHECKED	--	SCALE	1:2
MATERIAL Cast Iron --	SURFACE FINISH --		TITLE Shaft		REV. 01
			DRAWING NO. 2		A4
UNIVERSITY OF TWENTE.			FILE / PART NAME shaft		
FACULTY OF ENGINEERING			DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1

Appendix D

Wind turbine analysis

As a side project a short wind turbine analysis have been done for two different locations in the South Pacific. The wind data is collected by the University of the South Pacific on the islands Tarawa and Abaiang. The used wind turbine is the Vergnet GEV MP R 275kW.

Figure D.1: Wind side Abaiang

Figure D.2: Wind side Tarawa

Tarawa and Abaiang

The data was collected by two wind towers on the exact locations Tarawa: N023 23.670' E173 26.750' and Abaiang N001 48.486' E173 02.126. These sites are both potential locations for a future wind farm. The data is collected by two sensors at a height of 34m. The sensors did not always give the same value. It is assumed that the sensor with the lower value was not rotating properly and so the sensor with the higher value is at that moment leading. The recovered data was automatically averaged to 10 min averages. All the data points sorted per month are plotted in D.4 and D.7. Here is the middle point the average point and the thick bar is the middle 25%-75%. The data of Tarawa included all months of the year while the Abaiang data was only of six months.
Figure D.5, D.6, D.8 and D.9 presents the probability of a certain wind speed in the period of the wind data. Through this probability the Weibull distribution is plotted. It can be seen that the difference between ten minutes averaged data and the hourly averaged data is small in the Weibull distribution. Only the high bars in the histogram flats out but this is logical behaviour when taking a bigger averaged time period.

Wind turbine

The wind turbine Vergnet GEV MPR 275kW is already used in a wind farm on Fiji [21]. During the period November-February the South Pacific has high change of getting one or multiple cyclones. This is why the wind turbines in this wind farm have the ability to be laid horizontally on the ground. [21] These cyclones can normally easily destroy a wind turbine. By taking them down in advance the wind turbine can handle up to four times the wind speed it can by standing [21]. The power curve of the Vergnet MPR 275kW can be seen in figure D.3. Other important data can be seen in the table next to this.

Vergnet MPR 275kW	
Rotor diameter	32m
Class	IV
Blades	2
Hub height	32m
Cut in speed	3.5 m/s
Cut out speed	25m/s
Max speed	37.5 m/s
For a period of	10min

Table D.1: Wind turbine specifications [22]

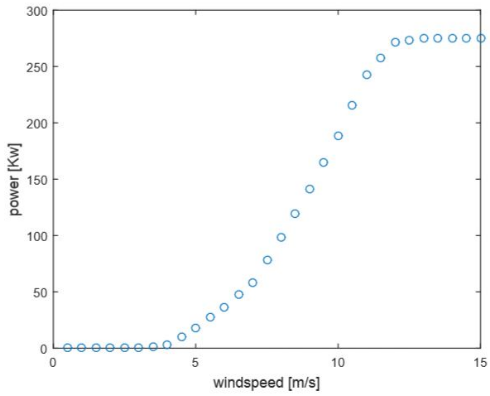


Figure D.3: Power curve of the GEV MPR 275kW

Conclusion

The wind turbine has been analysed on both locations. The wind data has been modified to the hub height. Since the height difference is only two meters and the area roughness class is between 0 and 1 the wind speed is not major changed. The calculated annual power, energy recovery factor and capacity factor are shown in table D.2. The influence of maintenance and down time during cyclones have not been added. The data of Abaiang is not annually but this data is partly from the high and low season. It can be seen that the energy recovery factor is above the preferable value. This means the wind turbine makes enough energy out of the available wind. Unfortunately the capacity factor is too low which means the wind turbine has a overcapacity. This can also be seen when the Weibull distribution and power curve are compared. The probability is around 4 m/s while the power curve begins to generate good power from a wind speed higher than 6 m/s. This is not a good fit. It can be concluded that another wind turbine with a lower cut-in wind speed could make more energy and has less overcapacity. It can be doubted if this is a profitable wind side.

	Annual energy production	Energy recovery factor C_E	Capacity factor C_f
Abaiang	483 MWh	0.334	0.201
Tarawa	343 MWh	0.312	0.1426
Minimum	N/A	0.2 (prefer: 0.25)	0.25 (good site 0.5)

Table D.2: Analytical wind turbine results

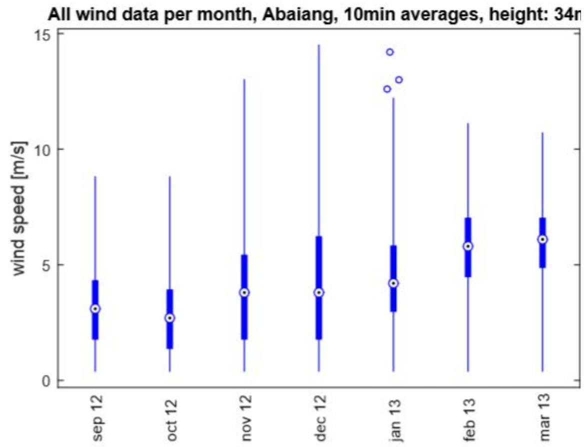


Figure D.4: All data plotted per month: Abaiang

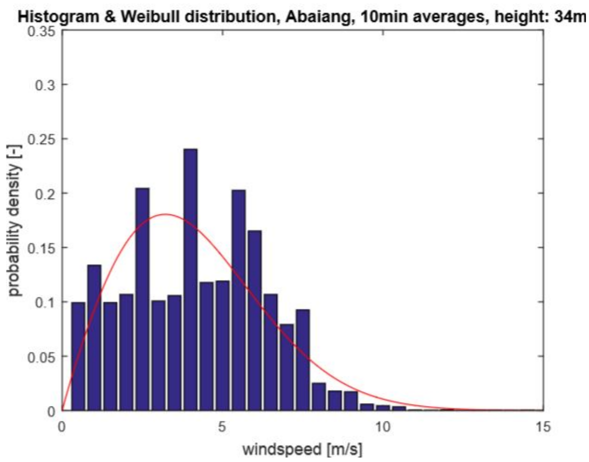


Figure D.5: Weibull distribution: Abaiang, 10min average

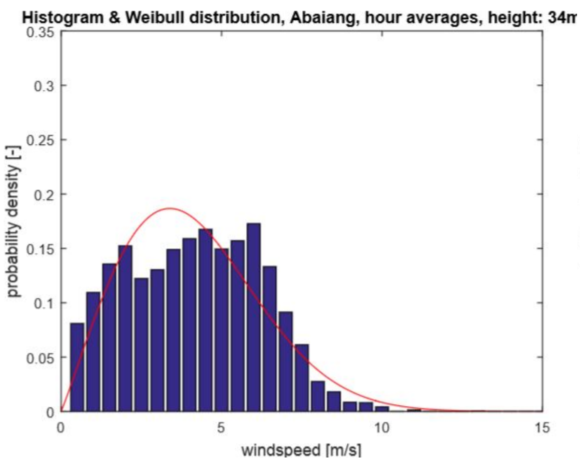


Figure D.6: Weibull distribution: Abaiang, hour average

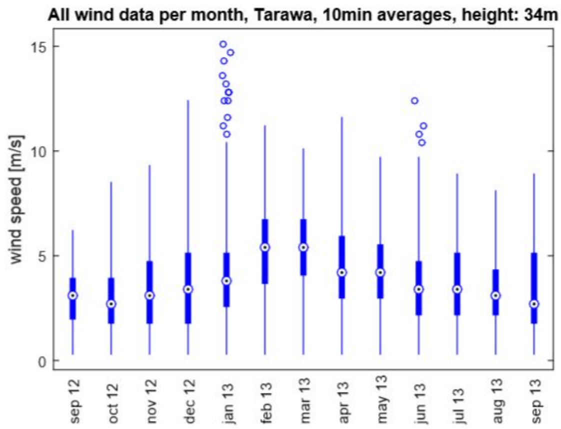


Figure D.7: All data plotted per month: Tarawa

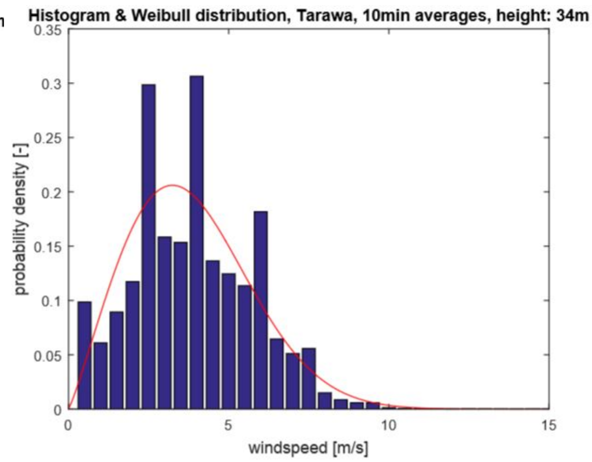


Figure D.8: Weibull distribution: Tarawa, 10min average

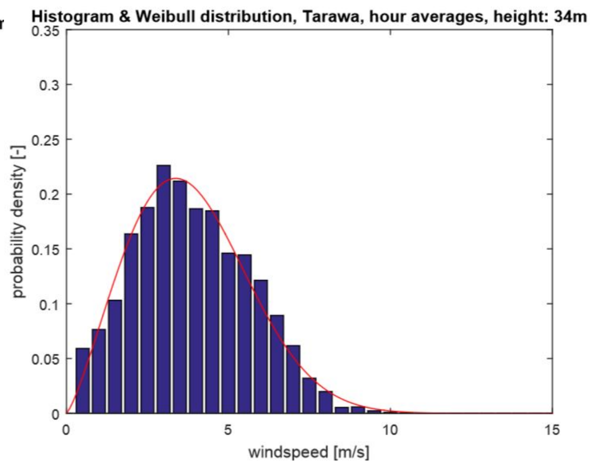


Figure D.9: Weibull distribution: Tarawa, hour average