Design, build and testing of a concentrating solar dish system

By

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Declaration

This project is Zhenzhou Feng’s own account of his research, the word count for all parts of the thesis excluding references, and appendices is 14207.
Abstract

This study describes the retrofitting of an old satellite dish as a concentrating solar dish system and investigates the factors that influence its performance concerning power generation. The satellite dish retrofit is viable for future development. The scheme of the project is divided into modelling, simulation, and optimisation. Modelling and simulation for the retrofitted satellite dish have been carried out by using System Advisor Model (SAM) and Franco’s program. According to the simulation results generated by SAM, the target dish can collect a maximum power of 2.987 MWh in January and a minimum power of 0.967 MWh in June. In this thesis, the effects on the system performance of several factors are studied, with factors including solar radiation resources, size of the dish, reflector material, focal point diameter, and cavity receiver thermal losses. Additionally, the theoretical design and analysis for a beta-type Stirling engine that integrates the retrofitted dish are provided as well. The performance of the engine can be enhanced by adopting the appropriate number of heater and cooler tubes to reduce the pumping losses in the regenerator and using more effective fluid. According to the simulation results that were generated by Franco’s program, the optimised peak power of the Stirling engine reaches almost 2.7kW and operates with a thermal efficiency of 43% at an engine speed of 64 Hz. A generator can be connected to keep the Stirling engine running at a constant speed once it reaches its maximum power. The retrofitted system can be used to power a water pump for irrigation purpose.
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Chapter 1: Introduction

Since ancient times, humans have used the sun to dry clothes or objects and make food. As a renewable energy source, solar energy boasts four significant advantages compared to traditional energy:

1. Universal: The sun can shine on earth with no terrain limit. It can be found everywhere, whether on land or ocean, mountain or island. Moreover, it can be acquired and utilised directly, it is easy to collect and there is no need for its transportation.

2. Persistent: According to the nucleation rate, it is estimated that the hydrogen in the core of the sun can sustain itself for over 5.4 billion years [1]. Given that Earth can survive for about 1.75 billion years [2], it can be concluded that solar energy is inexhaustible for Earth’s energy needs.

3. Enormous: Each year, the solar energy on Earth’s surface is equivalent to 130 trillion tons of coal [3], which means the sun alone can provide more energy than humankind demands.

4. Potential economic benefits: Power generation from solar energy can reduce expenses, such as the cost of silicon crystal photovoltaic conversion technologies. In addition, solar energy infrastructure can still generate power for several hours after the sunset through thermal storage technology.

Nowadays, solar energy can be used through solar thermal conversion, photovoltaic conversion, or a combination of both. Solar thermal power generation, an essential technique which takes advantage of the concentration of solar radiation, performs a significant and irreplaceable role in the generation of renewable energy. The four key configurations that dominate the solar thermal power market are as follows:

(1) Parabolic trough systems,

(2) Solar tower systems,

(3) Solar dish systems, and

(4) Linear Fresnel systems.
Among these systems, the Stirling solar dish system has been characterised as the most advanced since it operates at a high efficiency [4] and has an inherent hybrid capability. A solar-to-electric conversion efficiency (29.4%) was introduced in a previous study [5]. Moreover, in addition to solar irradiance, fossil fuel and biomass material can be also hybridised with the solar dish system. However, at this stage, this system is still in the research process so its cost can be reduced for eventual commercial availability.

This study aims to describe the retrofitting of an old satellite dish as a concentrating solar dish system and investigate the factors that influence its power generation performance. Available analyses for solar potential and each part of the solar dish system are taken into consideration in this thesis. In Chapter 2, a review that covers the various components of solar systems and background information about solar systems is presented. Chapter 3 introduces the theoretical calculations and retrofit scheme for the satellite dish system and prepares the parameters for further simulation. The performance of the retrofitted solar dish system is simulated by using the System Advisor Model and Franco’s program along with the different designs for the solar dish collector, receiver, and Stirling engine are presented in chapter 4, methods to optimise the system are offered with simulations and comparisons in chapter 4 as well. Finally, the conclusions are presented in Chapter 5.
Chapter 2: Literature review

As the oldest power on Earth, the Sun annually emits a massive scale of energy. Without anything obstructing most of the Sun’s energy, that energy travels through the solar system and flows into interstellar space. However, a tiny fraction of the Sun’s energy, $1367\text{W/m}^2$ approximately, after passing unhindered through interplanetary space, hits outside of the Earth’s atmosphere [6]. Humans have utilized solar power since ancient times. Socrates (469–399 B.C.) used passive solar energy when designing the orientation of his house. He pointed out that the porticos are penetrated by the Sun’s heat in winter, but shaded almost completely by the roof in summer if the house is designed with a southern aspect [7]. According to some ancient accounts, Archimedes (287-212 B.C.), the famous Greek mathematician and philosopher, burned wooden Roman fleets by utilizing catoptrics when Marcellus besieged Syracuse in 212 B.C. [8]. Figure 1 illustrates the branches of solar-powered thermal production systems, solar-powered electric production systems, and the integration of both [9]. This chapter presents a review of the literature concerning devices that harness solar energy for thermal and electric production.

![Figure 1 Solar-radiation harness strategies](image-url)
2.1 Thermal production

With most of the radiation in a wavelength range of 0.3 to 3 µm [6], solar water-heating systems and solar air-heater systems are the most common applications that aimed at thermal production, which are utilized in society. Renewable Energy Policy Network released the data in 2010; it says that about 70 million houses worldwide equip with SWH systems [27]. Solar air-heater systems are favored as well since they are designed to collect the maximum amount of solar energy at the minimum cost [28].

2.1.1 Solar water-heating system

As one of the most popular solar thermal applications, a solar water-heating system converts solar radiation into heat and transmits that heat into the fluid passing through the system [10]. Figure 2 displays the classification of solar water-heating systems. They are divided into active and passive systems. The active system is composed of an open-loop system and a closed-loop system. In an open-loop active system, fluid is heated by the collectors and stored in a tank. In case of solar water heater, hot water can then be pumped to service households directly [10]. In a closed-loop active system, a heat exchanger is used to transfer the heat from the storage tank to the household water [10]. The passive system consists of a thermosiphon and an integrated collector storage system. Regarding the operating principle of the thermosiphon system, the water in the flat-plate collector continues to be heated and expands, which results in the water becoming less dense and rising to the top of the storage tank [10]. As households use hot water, cold water is injected into the bottom of the tank to refill it, which keeps the system running [10]. Regarding an integrated collector storage system, this is composed of several metal tanks, insulated boxes, glazed covers, and auxiliary heaters [10]. Solar radiation is absorbed by the surfaces of the metal tanks, which heat the cold water inside [10]. It should be noted that the metal tanks are connected in a series; an outlet at the top of one of the
tanks is connected to the inlet at the bottom of the next tank [10].

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Figure 2 Classification of solar water-heating systems [10]

2.1.2 Solar air-heater system

Solar air heaters (SAHs) are expected to have a wide potential application in areas such as solar drying and solar heating [11]. Solar air heaters can absorb solar radiation and convert it into thermal energy, and the air flows through the SAH is heated [12]. Tanta University researched the thermal performance of three differently shaped single-flow SAHs [12]. Figure 3 provides a diagrammatic view of the cross-section shapes – circular, semi-circular, and half-circle plus isosceles triangle – for the three tested SAHs. Fabricated metal frames are used to maintain the cross-section shapes.

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Figure 3 Diagrammatic view of the cross-section shapes of tested materials [12]

The thermal performance results of these three SAHs are demonstrated in Figures 4 and 5. Figure 4 presents a comparison of thermal efficiency, as a function of solar irradiation, of the three tested SAHs. The thermal efficiency for the SAH is directly related to increased solar irradiance. In Figure 4, the circular solar air heater is the most outstanding configuration compared with the two other designs, as it can transfer more heat [12].

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Figure 4 Thermal efficiency as a function of solar irradiation [12]
In addition, Figure 5 compares thermal efficiency as a function of the air-flow rate among the three tested SAHs [12]. Similar to the results from Figure 4; the circular shape is the best, followed by the semi-circular, then the half-circle plus triangle.

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*Figure 5 Thermal efficiency as a function of the air-flow rate [12]*

### 2.2 Electricity production

With the extra-terrestrial solar radiation of 1367w/m² [6], Photovoltaic (PV) power generation and solar thermal-power generation are two of the most advanced techniques that employ devices to harness solar power to generate electricity. Research shows the approximate price of the electricity production from PV system, which is from 0.1 to 0.3$/kWh [29].

#### 2.2.1 Photovoltaic systems (PVS)

Photovoltaic cells are made from semiconductor materials and can perform as electrically neutral at low temperatures. As soon as they are excited by the Sun, however, they act as conductors [13]. Figure 6 is a schematic drawing of the energy bands for electrons in a solid [13]. In a valence band, the electrons remain in a steady state, whereas in a conduction band, the electrons are excited. To conduct a current, the minimum energy the electron needs to absorb is depicted as the forbidden gap [13]. Currently, silicon-based solar cells dominate the market, but other materials may displace silicon in the future.

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*Figure 6 Schematic drawing of the energy bands for electrons in a solid [13]*

*Crystalline silicon photovoltaic cells*

These first-generation PV cells are manufactured from bulks of solar-grade silicon with a
thickness ranging from 100 to 200 um [14]. They are categorized into mono-crystalline PV cells and poly-crystalline PV cells. Both types of crystalline silicon PV cells have an excellent market share (30% and 54% respectively) as their performance is outstanding under standard test conditions. However, with a temperature coefficient of 0.5%, neither is recommended for operating at elevated temperatures. In addition, neither can maintain their power output when they are partly shaded [14].

**Thin-film photovoltaic cell**

The second generation of PV cells is thin-film PV. They are cheaper than crystalline silicon PV cells because they don’t require ingot techniques [14]. The most common configurations of thin-film PV cells are amorphous silicon, copper indium gallium selenide, and cadmium telluride (CdTe). With a temperature coefficient of 0% and a maximum power conversion efficiency of 21%, CdTe is the most outstanding product of these three configurations [15].

**State-of-the-art photovoltaic cells**

These photovoltaic cells are usually composed of organic and organo-metallic material [15]. There are many types of state-of-the-art PV cells, which include Perovskite PV cells, Dye PV cells, and concentrated PV cells. Currently, these designs are still in the research stage [15].

2.2.2 Solar thermal-power technologies

Solar thermal power generation (especially concentrated solar power) has received high attention as well. It can directly utilize the heat from the sun and has an excellent quality thermal storage system at a low price. Depending on the collector, CSP is divided into four categories: parabolic trough collectors, power towers, linear Fresnel, and parabolic dish collectors. However, all four CSP devices operate by harvesting the Sun’s heat and passing it on to fluid, which drives turbines to generate electricity.

**Parabolic trough collectors**

The composition of parabolic trough collectors is divided into two parts: collector and receiver. The receiver is a linear, vacuumed glass tube that is assembled at the focal axis. The collector
is made of curved, highly reflective, and smooth material that can reflect and focus sunlight onto the receiver. Additionally, the collector sits on a tracking system that can harness the maximum power from the Sun [16].

**Linear Fresnel technology**

In general, linear Fresnel technology is similar to parabolic trough collectors. Instead of using parabolic collectors, however, linear Fresnel adopts flat, linear collectors, which reduces the cost. Unfortunately, the gaps between the collectors negatively impact the overall efficiency. In addition, since the sunlight is reflected and focused on the linear evacuated tube by some of the collectors, it is not necessary to design tracking devices for this type of CSP system [16]. The Indian Government believes that the advantages of this system outweigh its disadvantages. In 2014, the Reliance Power’s STE project (125MW), which uses this technology, operated successfully [17].

**Power towers**

A power tower is another technology that utilizes solar thermal energy to generate electricity. A power tower system consists of two main parts: thousands of reflectors and a tower receiver. Compared with parabolic trough collectors, the heliostat has been replaced with a flat mirror, and a two-axis Sun tracking system has been adopted [16]. Furthermore, a power tower usually has a hot storage system. For example, there are two storage tanks utilized in the Crescent Dunes plant [18]. One aims to store the molten salt pumped from the receiver (usually 565°C), the other tank stores the cool salt (roughly 280°C) and pumps it to the receiver. Therefore, the stored molten salt is released into a heat exchanger at night to produce steam and drive a steam generator [18].

**Parabolic solar dish system**

The solar dish system is widely regarded as the most effective CSP technology for converting solar energy into electricity. For this reason, it has received much attention. However, John Ericsson, who is thought to be the first to integrate a parabolic dish with a Stirling engine in
the 1880s, predicted that the value of the solar dish system would not be fully reflected until coal is exhausted [19]. He pointed out that it would be the high expense that would prevent the solar dish from spreading and becoming popular rather than its technical challenges, which up to now has proved true [19]. Nonetheless, many advanced configurations and optimizations have been made to improve both its economic and operational performance. Furthermore, considering the growing greenhouse effect, which is primarily caused by the burning of coal, the solar dish system can play a significant role in mitigating environmental problems. In this chapter, we focus primarily on reviewing the Dish-Stirling system design, which is separated into a parabolic collector, a receiver, and a Stirling engine.

**Parabolic collector**

The factors that affect collector performance has been researched for many years. The factors are summarized and categorized into three groups: the material of the collector, the diameter of the solar dish, and its rim angle. Regarding the first factor, some researchers have investigated the performance of three parabolic solar dishes whose collectors are each made from different materials. They state that greater thermal efficiency is achieved by adopting highly efficient reflective materials and optimizing the shape of the collector [20]. Another key factor is that many solar dishes are designed or built in various sizes. In 1994, the Australian National University (ANU) constructed a 400m² solar dish system. The solar dish was designed in such a configuration to achieve a higher conversion efficiency; however, the capital cost of the dish was significant. Later, ANU proposed a 500m² solar dish composed of 380 spherical glass-on-metal laminate mirrors, which utilizes the mirrors as part of the structure to reduce costs [21]. Regarding the small size of the solar dish, instead of utilizing a curved glass mirror or a polished aluminum mirror, Palavras and Bakos refitted a 2.85m (diameter) satellite dish with a polymer mirror film. The receiver reached 300°C, and the collector is not only cheaper but also lighter than similar systems [22]. The rim angle of a solar dish, $\phi_{\text{rim}}$, can be calculated by
\[ \frac{f}{d} = \frac{1}{4\tan\left(\frac{\phi_{\text{rim}}}{2}\right)} \]

where \( f \) is the focal length of the dish in meter and \( d \) is the dish diameter in meter.

Figure 7 demonstrates the relationship between the rim angle (or the ratio of \( f/d \)) and a specific dish diameter \( d \). As the \( \phi_{\text{rim}} \) (rim angle of the dish) decreases, the parabola dish becomes flat, and its focal length and \( f/d \) ratio grow [23].

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*Figure 7 Segments of a parabola [23]*

Regarding the effects of the rim angle on solar dish performance, Sebastián Mendoza and Oscar Almazan concluded that a solar dish with a rim angle of about 45 degrees could achieve maximum solar concentration, as Figure 8 illustrates [24].

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*Figure 8 Variation of solar concentration [24]*

**Stirling/dish system receiver**

Generally, external and cavity receivers are the most common configurations adopted by parabolic dish systems. The former has a wider absorb direction than the latter, but the latter performs with lower heat-loss rates. For this reason, cavity receivers have received intense attention, and have been widely utilized in dish/Stirling systems. To demonstrate the factors that affect the useful heat supplied to the receiver, an equation is provided below [25].

\[
Q_{\text{useful}} = I_{b,n} A_{\text{app}} E(\cos \theta_i) \rho \varphi \tau \alpha - A_{\text{rec}} [U(T_{\text{rec}} - T_{\text{amb}}) + \sigma F(T_{\text{rec}}^4 - T_{\text{amb}}^4)]
\]

\( A_{\text{app}} \): collector aperture area
Working temperature

While the efficiency of the heat engine has a direct proportion to its operating temperature, the useful heat gain by the receiver varies inversely with the receiver temperature. The equation shows that, as the receiver temperature grows, heat loss increases; accordingly, the total useful heat is reduced. To enhance its performance, $A_{rec}$, $U$, and $F$ are three key considerations of receiver design.

Cavity cover transmittance

To reduce heat loss caused by convection, a fused quartz window is utilized to cover the aperture of the receiver [25]. However, such integration can reduce the incoming solar resource. Usually, the material used in the cover has a transmittance of approximately 0.9.

Emittance, absorptance, and reflectance of the receiver

Selective coatings are preferred in this area as they have different properties in different parts of the electromagnetic spectrum [25]. They can absorb well in the solar spectrum, and be a poor emitter of radiation in the thermal radiation section of the spectrum. Accordingly,
radiation loss is reduced. In addition, when we talk about photon energy conservation, the following formula should be considered.

\[
\text{Transmittance} + \text{reflectance} + \text{absorptance} = 1
\]

A selective surface has an outstanding absorptance, which can absorb the incoming energy efficiently with an absorptance of over 0.9 [25].

Cavity and receiver material selection

Under normal conditions, the operation temperature of a dish/Stirling system can increase to 700-degree C from the ambient temperature, and drop to the ambient temperature again after it is shut down at night. This significant temperature shift can lead to shape changes in both the cavity and the receiver [25]. Regarding failures, thermal fatigue, oxidation resistance, and the life cycle of materials are the most basic considerations for material selection [25].

Stirling/dish receiver sample

Figure 9 depicts a direct illumination receiver for a Stirling thermal motor. It uses a directly illuminated heater tube and can generate 720-degree C gas [25], and its peak efficiency reaches 90% [25]. Figure 10 presents another type of heat pipe receiver [25]. With a smaller aperture area, it is more suitable for our satellite dish structure, and heat loss is reduced as well. Unlike the former configuration, the latter uses heat-transfer fluid to transmit energy, which can keep the operation temperature stable [25]. As a result, although its efficiency is lower than the former, it is the widely favored design [25].

Stirling/dish system heat engine

In a solar dish system, different heat engines are selected according to the heat cycle. The
Stirling cycle and Brayton cycle, which are the ideal operating principles of a Stirling engine and a gas turbine respectively, are two of the most advanced technologies adopted by the solar dish system. In the literature review, only the design of a Stirling engine is investigated.

**Ideal Stirling cycle**

An ideal Stirling cycle is composed of four steps: compression, displacement, expansion, and displacement. These are demonstrated in Figure 11 [25]. In the first step, working gas in a cold area is compressed in the constant temperature compression process (1-2). The area of a-1-2-b in the p-v and T-s diagrams (Figure 11) represents the work demand by the cold piston and heat transfer from the working gas in the cold area respectively. In the second step – the constant volume heating process (2-3) – the hot piston moves synchronously to the right with the cold cylinder to keep the volume of the working chamber in the cylinder unchanged. During this process, as the cooler working gas in the cold area passes through the regenerator to the hot area, it absorbs the heat in the regenerator and its temperature rises. According to the p-v diagram, no work is done in this process. The heat supplied by the regenerator is represented by the area of b-2-3-c in the t-s diagram. In the next process (3-4), the working gas is heated and expands to push the hot piston. The work done in this process is represented by the area of a-b-3-4 in the p-v diagram. In the final process, the hot piston moves synchronously to the left with the cold piston to keep the volume of the working chamber unchanged. During this process, while the working gas is shuttled from the hot area to the cold area, the regenerator stores part of the heat. Kinematic Stirling engines and free-piston engines are designed based on the principles outlined above.

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*Figure 11 Basic processes of an ideal Stirling engine [25]*
Kinematic Stirling engines

Normally, kinematic Stirling engines are categorized into three configurations: \( \alpha \), \( \beta \), and \( \gamma \). \( \alpha \) type Stirling engine has two cylinders, one of them operates as a hot cylinder and the other operates as a cool cylinder, these two cylinders have almost the same size. \( \beta \) type Stirling engine has only one cylinder, but the two ends of the cylinder operate as hot end and cool end respectively. \( \gamma \) type Stirling engine has two cylinders, the displacer cylinder is quite smaller than the other cylinder. Although the physical layout of each configuration is different, all three have similar operating principles. The power piston and the displacer in these configurations are mechanically connected to a rotating crankshaft that drives the engine. Numerical kinematic Stirling engines are designed with a large capacity, such as the United Stirling USAB 4-95 engine (27.1kWe, Figure 12) and the USAB 4-275 engine (52.5kWe, Figure 13). In terms of kinematic Stirling engines with a small capacity, the Stirling Power Systems/solo v-160 engine was tested in 1991 (9kWe), and a 2.4kWe (Figure 14) beta-type Stirling engine is in development [26].

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [25]

*Figure 12 USAB 4-95 engine [25]*

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [25]

*Figure 13 USAB 4-275 [25]*

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [26]

*Figure 14 Beta-type Stirling engine [26]*
2.3 Retrofit schemes similar to our project

To improve our retrofit scheme, two examples which are similar to our project are demonstrated. N.D. Kaushika retrofitted a satellite dish for steam generation purpose with an aluminum frame and chose polymer film coated with a silver-aluminum alloy at its back as its reflector [30]. A semi-cavity and a modified cavity receiver are investigated to enhance its thermal performance. As a result, a solar to steam conversion efficiency of around 70-80% is performed at 450-degrees C [30]. In order to investigate the zeolite desorption by using the retrofit satellite dish, Polymer mirror film without a coat is chosen as its reflector by I. Palavras [31]. As a flat aluminum plate absorber is integrated (instead of a cavity receiver), an average value of overall heat loss coefficient of approximately 163 W/m² K is showed in his design, but it still reaches temperatures of more than 300-degrees C [31]. A retrofit experiment that transform the normal plate receiver to a cavity receiver is given in Appendix. Figure 15 gives the key parameters of the N.D.Kaushika’s dish and I.Palavras’s dish.

<table>
<thead>
<tr>
<th></th>
<th>N.D. Kaushika</th>
<th>I. Palavras</th>
</tr>
</thead>
<tbody>
<tr>
<td>Focal length of the dish</td>
<td>0.96m</td>
<td>1.02m</td>
</tr>
<tr>
<td>Aperture diameter of dish</td>
<td>2.405m</td>
<td>2.85m</td>
</tr>
<tr>
<td>Distance from a point on the rim to focus</td>
<td>1.34m</td>
<td>1.52m</td>
</tr>
<tr>
<td>Dish rim angle</td>
<td>65 degree</td>
<td>70 degree</td>
</tr>
<tr>
<td>Focal image width</td>
<td>0.14m</td>
<td>0.18m</td>
</tr>
<tr>
<td>Concentration ratio</td>
<td>295</td>
<td>250</td>
</tr>
<tr>
<td>Reflector material</td>
<td>polymer film coated with a silver-aluminium alloy at its back</td>
<td>polymer mirror film</td>
</tr>
</tbody>
</table>

Figure 15 Geometry of two reference solar dishes

Chapter 3: Theoretical design of the system

In this chapter, the entire retrofit project is introduced. To do so, the project is divided into the design of the tracking system, solar dish collector, solar dish cavity, and a β Stirling engine. A
preliminary analysis of the tracking scheme and an embedded system methodology were provided by a group of industrial computer students [32]. For this reason, only a summary of the development of the sun tracking system is given in this thesis. The remaining three parts of this chapter are described in detail with equations, and all parameters that are necessary for further simulation are also specified.

3.1 Tracking system

The pedestal of our solar dish system is located in the Renewable Energy Outdoor Test Area (shown in Figure 16). A group of industrial students built a dual-axis tracking system for the satellite dish based on the pedestal [32]; Figure 17 shows their design. Theoretically, a tracking system for the solar dish can be accomplished by azimuth elevation or polar tracking. The former rotates the dish with a large spur gear (parallel to the earth) and the parabolic dish base (perpendicular to earth). The latter tracking system rotates the plane, which remains parallel to the rotation of Earth, at a rate of 15° per hour, and the other axis adjusts the plane, which stays perpendicular to the polar axis plus or minus 23.5° throughout the year. The group adopted the azimuth elevation design, but they were unable to install the prototype as per the
design because they encountered manufacturing and machining issues [32].

![Image of Pedestal in the ROTA area](image)

*Figure 16 Pedestal in the ROTA area*

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [32]

![Image of Detailed analysis drawing of the tracking system](image)

*Figure 17 Detailed analysis drawing of the tracking system [32]*

The Stirling Energy System (SES) adopts a similar design to the design generated by the group of industrial students in SunCatcher, it tracks the sun in a way of azimuth elevation as it generates power by converting concentrated solar thermal energy through a Stirling engine [33]. The sketch design of the SES SunCatcher can be seen in Figure 18 [33]. It is composed of a pedestal, facet support structure, mirror facet, tracking drivers, boom, and a power conversion unit. A dish controller is equipped inside of the pedestal, and drivers help the dish to track the sun in both the azimuth and elevation directions. As a result, the collector can gather maximum solar energy during the day.

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3.2 Parabolic Concentrator

A solar dish concentrator, which is configured with mirrors, sits on a support structure that tracks the sun to gain maximum solar radiance by pivoting on both the azimuth and elevation axes. Factors that affect the performance of a parabolic dish been researched for many years; features such as collector diameter, dish focal length, dish rim angle, and concentrator reflectors represent the main considerations in many studies. These aspects of our satellite dish are explained in this section so they can be applied for further simulation.

3.2.1 Collector geometry factors

Theoretically, direct solar radiation is reflected into the focus area where a receiver sits by a parabolic dish concentrator. A significant amount of radiation is reflected and transmitted to the receiver if the aperture area of the solar dish is designed appropriately. The key factors that need to be addressed when designing a solar dish collector are aperture diameter, dish depth, and rim angle.

To calculate the rim angle of the satellite dish, the aperture diameter and dish depth must be measured; for our dish, these measurements are 3.2 m and 0.5 m, respectively. The rim angle of a solar dish indicates its curvature such that a collector with a smaller rim angle has a gentle slope. As a parabolic curve can be represented as

\[ y = ax^2 \]

Where

\[ f = \frac{1}{4a} \]

Because the dish diameter is 3.6 m and its depth is 0.5 m, point (1.8,0.5) on the curve is represented by the equation above. As a result,
\[ a = 0.154, \quad f = 1.62m \]

The rim angle can then be calculated by using the equation given below:

\[ \psi_{rim} = \tan^{-1}\left( \frac{f}{d} \right) \left( \frac{2 \times \left( \frac{f}{d} \right)^2 - \frac{1}{8}}{2 \times \left( \frac{f}{d} \right)^2} \right) \]

Where \( f \) is the focal length of the dish (1.62 m) and \( d \) is the dish diameter (3.6 m), which means the \( \psi_{rim} \) is equal to 58.103°.

### 3.2.2 Collector system errors

Ideally, the radiation should be reflected and focused into a point. However, as some imperfections occurred due to system errors, the beam is spread. These errors were caused by several factors, including an error in the mirror slope, receiver alignment discrepancies, the difference in the mirror’s reflectivity to the spectrum, tracking sensor errors, and tracking driver errors. The intercept factor, which is the ratio of solar radiation intercepted by the receiver to the solar radiation reflected by the collector [34], is heavily influenced by these errors. The total error is the standard deviation of the errors, which can be presented as follows:

\[ \sigma_{tot} = \sqrt{(2 \sigma_{slope})^2 + \sigma_{sensor}^2 + \sigma_{drive}^2 + \sigma_{align}^2 + (2 \sigma_{reflect})^2 + \sigma_{sun}^2} \]

Ricardo et al. designed a solar dish collector with a total error of 8 mrad (milliradian)[35]. Furthermore, Mancini et al. noted that the solar dish collector of the WG Associates (WGA) collector system is manufactured with lower errors (4 mrad), which contributes to an intercept factor of more than 99% [36]. The errors of the two examples of retrofitting satellite dish to a solar dish, which is introduced in Section 2.3, are more significant than 8 mrad. The reasons for this are presented in Section 3.2.2.2.

**Beam spread without errors**

In the case of our dish, nonparallel sun rays are reflected and spread by the collector. For this reason, the receiver has to be designed with a diameter larger than the length of the beam...
spread. Otherwise, the intercept factor decreases. In the absence of errors of the collector system, the length of the beam spread caused by nonparallel sun rays can be calculated by

$$\Delta r = 2p \times \tan \left( \frac{\varepsilon}{2} \right)$$

Where $\varepsilon$ is the size of the sun’s disc seen from earth (shows in Figure 19), which can be calculated by

$$\varepsilon = 2 \tan^{-1} \left( \frac{D}{2d} \right) = 0.533 \ degrees$$

![Figure 19 Sun's disc as seen from earth](image)

The length between any point on the parabolic surface and the focal point in metres is represented by $P$, and $\Delta r$ is the length of the beam spread perpendicular to the centre line of the reflected sun rays in metres. The value of $P$ can be calculated by

$$P = \frac{2f}{1 + \cos(\psi)}$$

Where $f$ is the focal length in metres and $\psi$ is the angle from $0^\circ$ to the dish rim angle. As a result, the length of the beam spread increases as the sunlight is reflected from the collector vertex to its outer edges. For our satellite dish, $P_{\text{max}}$ is equal to 2.12 m and $\Delta r$ is equal to 0.02 m.

**Beam spread with errors**

The errors strongly influence the length of the beam spread. As shown in table in Section 2.3, the focal image width of the retrofit satellite dish made by N.D. Kaushika [30] and I. Palavras [31] are 0.14 m and 0.18 m, respectively. The beam spread, including collector errors, can be
calculated by

$$\Delta r = 2p \times \tan \left( \frac{n \sigma_{\text{tot}}}{2} \right)$$

where

$$p = \frac{2f}{1 + \cos(\psi)}$$

As a result, $P_{\text{max}}$ for Kaushika and Palavras’s dishes are 1.35m and 1.52m, respectively. Then, the $n\sigma_{\text{tot}}$ for Kaushika and Palavras’s collectors can be calculated. It should be noted that the focal image width is $w_n$, and the $w_n$ for Kaushika and Palavras’s dishes is 0.14 m and 0.18 m, respectively. The calculation for $w_n$ is as follows:

$$w_n = \frac{\Delta r}{\cos(\psi)}$$

As the rim angles for Kaushika and Palavras’s dishes are 65° and 70°, respectively, the $\Delta r$ for their collectors are 0.059 m and 0.062 m, respectively. When $n$ equals 4, 95% of the reflected solar radiance incident within the length of the beam spread. As a result, the $\sigma_{\text{tot}}$ for Kaushika and Palavras’s collectors are 10.92 mrad (0.626°) and 10.19 mrad (0.584°), respectively. Regarding our satellite dish, the total error of our collector is assumed as 11 mrad (0.63°). As a result, the beam spread including collector errors should be 0.093 m, and the focal image width is 0.176 m, which means the receiver diameter should be designed as at least 0.176 m.

**Summary of the satellite dish parameters**

A summary of the satellite dish parameters is given below. Also, a system diagram is demonstrated in Figure 20 to correspond with some of the parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Aperture diameter (m)</td>
<td>3.6</td>
</tr>
<tr>
<td>2. Dish depth (m)</td>
<td>0.5</td>
</tr>
</tbody>
</table>
### 3.3 Cavity receiver for solar dish

A cavity for a solar dish system is designed to achieve two primary goals: one is to absorb solar radiation as much as possible, and the other is to transfer the absorbed energy to the working fluid. Due to the existence of nonparallel rays, the imperfectly shaped concentrator, and the manufacturing errors, a focal area is distributed at the theoretical point of focus. To achieve the two goals, the cavity for solar dish systems should be sized to just cover the focal area to reduce radiation and convection loss. Up to now, receivers are the core components of parabolic dish solar thermal power generation systems, including directly illuminated and indirectly heated receivers (heat pipe receiver). The former collects sunlight to directly
illuminate the heat exchanger tube of the heat engine, while the latter transfers the solar energy to the engine through liquid sodium in the receiver.

### 3.3.1 Direct illumination receiver

Recently, much research on receivers has focused on further reducing costs and improving reliability and efficiency. A directly illuminated receiver bends a cluster of the heat exchanger tubes of the Stirling engine into a coil. Then, when the collected sunlight directly illuminates the coil surface (that is, the surface of each heat exchanger tube), the working medium in the tubes flows and absorbs the energy of the sunlight to reach a higher temperature and pressure, thus operating the Stirling engine. However, due to the instability of the solar radiation and low machining accuracy of the reflector, the heat flux on the heat exchanger tube is unstable and uneven, which results in an unbalanced supply of temperature and heat in the cylinders of the multi-cylinder Stirling engine [25].

### 3.3.2 Heat pipe receiver

An indirectly heated absorber transfers heat to the receiver of the Stirling engine by the evaporation and condensation process of liquid metal according to the phase-change heat transfer mechanism of liquid metal. This process improves the efficiency of heat engines with its strong isothermal condition. Its working medium is mainly liquid alkali metal sodium, potassium, or a sodium-potassium alloy. There are two indirect heating modes: the pool boiling receiver and the heat pipe receiver. The former heats the liquid metal pool with the solar energy gathered on the surface. Then, the vapour is condensed on the heat exchange tube of the Stirling engine, and finally heat is transferred to the working medium in the heat exchanger tube while the condensate returns to the liquid metal pool due to gravity. Its structure is simple, and the processing cost is low, but it has a particularly high requirement for sealing. The heat pipe receiver, on the other hand, adopts a capillary wick structure to distribute the liquid metal evenly on the heating surface. This surface is arched, and above it
lies the wick. The structure of the wick can take a variety of forms, such as stainless steel mesh or metal felt. Its working principle operates similarly to the pool boiling receiver.

### 3.3.3 Receiver heat losses consideration

Before the thermal energy enters through a power conversion unit (a Stirling engine in this case), the cavity receiver of the solar dish system accounts for most thermal loss. Under a constant temperature, the thermal loss in the cavity receiver is closely related to the conduction loss through the receiver and cavity wall, convection loss from the cavity, and radiation loss to the ambient air. The most significant loss is caused by radiation at around noontime. The convection loss increases during morning and evening as the receiver inclines toward the side. Conduction losses amount to only a small fraction of the total losses. To improve the overall efficiency of the solar dish system, a cavity receiver of a suitable size has to be designed to minimise the thermal losses.

**Conduction losses**

Although conduction loss only account for the minority of the total thermal losses on the receiver, it increases with its operating temperature. The factors that influence the thermal losses through conduction are the temperature difference between the interior and exterior of the cavity, thermal conductivity of the cavity material, average thickness of the cavity, and exterior area of the cavity. The total conductive losses on the cavity receiver can be calculated by

\[
q_k = \frac{k A \Delta T}{L} = \frac{\Delta T}{R_k}, \quad R_k = \frac{L}{KA}
\]

Where \( k \) represents the thermal conductivity of the cavity material in W/m-k, \( A \) is the exterior area of the cavity in m², \( \Delta T \) is the temperature gradient in °C, and \( L \) is the average thickness in m of the cavity receiver. In a solar dish system, conduction losses are easily controlled by adding insulation material into the cavity wall. Some previous research suggested that an average thermal conductivity of 0.061 W/m-k should be used as reference, and ceramic fibre
insulation can achieve this figure [37]. As the average thickness of our cavity receiver is 60 mm and its aperture diameter is 176 mm, the exterior surface area of our target cavity can then be calculated as

\[ S = 0.296\pi \times 0.15 = 0.139 m^2 \]

Therefore, the calculation shows that at a constant L, k, and ∆T, q_k increases with the growth of the exterior area of our target cavity. Conduction losses maintain a linear relation to the temperature difference between the internal and external cavity surfaces.

Convection losses

The convection losses in the solar dish cavity receiver account for a significant proportion of the total heat losses. Wu et al. gave some investigation on convection heat loss of a fully open cylindrical cavity with different boundary conditions [38]. Tests can be carried out by utilising electrical heating on various surfaces of the cavity wall to investigate the factors that influence heat loss on the target cavity receiver. The cavity receiver used in the test by Wu et al. is shown in Figure 21.

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [38]

*Figure 21 Cavity receiver used in Wu’s test [38]*

Three scenarios of heat loss were tested under constant heat flux and different inclination angles; the three cases are (1) bottom surface heated, (2) side surface heated, and (3) all surface heated [38]. Figures 22, 23, and 24 specify the test results of the three cases. By comparing the results from these three figures, it can be seen that the third case (3) lost more heat than other two cases. With the increase of the tilt angle of the receiver, convection heat loss dropped significantly while both radiation heat loss and conduction heat loss increased. As convection losses occupy more than half of the total heat losses when the inclination angle of the cavity receiver is less than 45°, it is necessary to determine the reason for this.
Convection heat losses can be divided into natural convection losses and forced convection losses.

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [38]

*Figure 22 Case (1), heat loss vs inclination [38]*

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [38]

*Figure 23 Case (2), heat loss vs inclination [38]*

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [38]

*Figure 24 Case (3), heat loss vs inclination [38]*

**Natural convection losses**

From Wu’s research, it can be concluded that the cavity temperature, cavity receiver aperture inclination angle, and wind velocity all impact the convection heat loss of the receiver. Normally, the convection losses are most noticeable in the early and late hours of a day as the tracking system has to orientate the collector directly toward the sun, which means the cavity receiver must be adjusted in a more horizontal direction in the morning and afternoon and an almost vertical at noon. Yeh et al. indicated that the resulting natural convective loss is a function of the cavity receiver orientation [39]. The blue flow represents the denser fluid, and red and orange flow refers to less dense fluid. Figures 25 suggest that with a constant Gr, the stagnation area that represents the less dense fluid increases with the growth of the inclination angle. In Figure 25, the cavity receiver has an inclination angle of 0 degree firstly, blue fluid clearly enters a cycle and nearly dominates the zone in the cavity receiver, which
represents a significant natural convection heat loss. Although the zone occupied by red fluid grew when the cavity inclines 30 degrees, the blue fluid still pushed away almost half of the red fluid, which indicates that natural convection losses are still significant. The situation improved while the cavity receiver inclines up to 60 degrees; the natural convection losses are reduced consistently. Finally, there is no red fluid leakage when the receiver inclines to 90 degrees, which represents the smallest convection heat loss.

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*Figure 25* Performance of the cavity for different angle (0 degree to 90 degrees) [39]

Clausing [40] suggested that the natural convection losses of a cavity receiver can be represented by

\[ Q_a = \rho_{amb}V_aA_aC_p(T_c - T_{amb}) \]

Where \( \rho_{amb} \) is the air density at the operation temperature, \( A_a \) is the aperture area of the cavity receiver in the research (but with the inclination angle changes, it should be the active area with specific orientation), \( C_p \) is the specific heat of the air at the operation temperature, and \( T_c \) is the temperature of the air as it leaves the aperture.

Effective flow velocity is expressed as \( V_a \) and can be calculated as

\[ V_a = 0.5 \sqrt{V_b^2 + \left(\frac{V_{wind}}{2}\right)^2} \]

The buoyancy induced velocity is shown as \( V_b \) and is determined by

\[ V_b = \sqrt{g\beta(T_c - T_{amb})L_a} \]

Where \( \beta \) is the expansion coefficient of the air at operation temperature.

From the above equations, it can be concluded that natural convection losses change with air density, which means these losses vary with altitudes. Moreover, with the increase of the
cavity aperture area, the convection heat losses increase linearly as well. Again, as mentioned previously in this thesis, the cavity receiver has to be sized just larger than the length of the coal image to reduce natural convective losses. Finally, the temperature difference between the internal cavity and the ambient air contributes to thermal losses as well.

**Forced convection losses**

Forced convection losses are a function of wind speed. If the wind blows directly into the aperture of the cavity receiver, the forced convection losses significantly decrease the performance of the system. In 1993, Ma [41] completed a project to investigate the factors that influence the forced convection losses. The tested receivers had a larger aperture diameter (0.46 m) than our target cavity receiver (0.29 m). Similar to Yeh’s experiment, receivers were placed at a constant interval from 0° to 90° to observe the forced convection losses. Wind was generated by a machine with controllable speeds. Ma divided the forced convection losses into the side-on and head-on tests (shows in Figure 26) [41]. In the side-on test, the wind direction was adjusted to be parallel to the aperture of the cavity with speeds of 6, 8, and 20 mph; whereas in the head-on test, the wind blew directly into the receiver with speeds of 15 and 24 mph [41].

As the writer does not have the copyright, the picture is omitted from the electronic version of the thesis. The picture can be found in reference [41]

*Figure 26 Indication of side-on and head-on wind directions [41]*

The changes of the temperature of the heater tubes and convection losses were determined by using an organic fluid. To measure the natural convection losses of the receiver, Ma implemented six steps, and the results concur with Stine and McDonald’s natural convection correlation [41]. To calculate the total convection losses, Ma noted that the forced convection
losses of the receiver should be added to the natural convection losses [41].

**Side-on wind forced convection losses**

As introduced above, side-on wind blows parallel to the aperture of the receiver. Ma indicated the relation between the forced convection heat transfer coefficient and wind speed, which can be calculated as follows [41]:

\[ h_{\text{side-on}} = 0.1967 v^{1.849} \]

Where \( h_{\text{side-on}} \) is the convection coefficient in \( W/(m^2 \cdot k) \) and \( V \) is the wind speed in m/s.

From this equation, the only factor that influences the side-on wind forced convection losses is the wind speed, which means the cavity orientation, air density, and temperature do not impact it.

**Head-on wind forced convection losses**

Different from the side-on wind forced convection losses, head-on wind forced convection loss is defined as when the wind blows perpendicular to the aperture of the cavity receiver; by this definition, this type of loss is highly dependent on the inclination of the cavity receiver. In Ma’s research, the calculation of the head-on wind forced convection losses is given by [41]

\[ h_{\text{head-on}} = f(\theta)V^{1.401} \]

\[ f(\theta) = 0.1634 + 0.7498 \sin(\theta) - 0.5026 \sin(2\theta) + 0.3782(3\theta) \]

Where \( V \) is the wind speed in m/s, \( \theta \) is the inclination angle in degree, and \( h_{\text{head-on}} \) is convection coefficient in \( W/(m^2 \cdot k) \).

It is clear that both side-on and head-on wind losses increase with the size of cavity.

**Radiation losses**

Heat loss caused by radiation represent a large part of the total heat loss in the receiver. In Abbasi-Shavazi et al.’s research, under known conditions of temperature distribution inside the cavity receiver, valid assumptions were made to calculate the radiation losses [42]. Figure 27 demonstrates the differing radiation losses of the cavity receiver under different calculation methods of the receiver temperature [42]. When calculating the top or bottom surface
radiation losses, the measured temperature is used as a reference, and the temperature
distribution of the collector is assumed to be axially symmetrical [42]. A curve that fits the
experiment data is then used to calculate the radiation losses by adopting the radiosity
network method [42]. As the temperature of the top surface is higher than the bottom side,
its radiation losses are more severe. Area-weighted temperature is the average temperature
of the top and bottom side of the cavity receiver. The radiation loss calculated based on this
temperature is higher than the other two cases because their temperature distribution in the
cavity is assumed to be constant. It is interesting to note that radiation loss increases with the
increase of the cavity inclination, whereas convection loss lowers while the cavity inclination
increases.

As the writer does not have the copyright, the picture is omitted from the electronic version of
the thesis. The picture can be found in reference [42]

*Figure 27 Radiation losses as a function of cavity inclination [42]*

To calculate the radiation loss, an equation was determined by Incropera and DeWitt in year
2002 [43].

\[
q_{\text{radiation}} = \varepsilon_{\text{cav}} \cdot \sigma \cdot A_{\text{ap}} \cdot (T_{\text{cav}}^4 - T_{\text{amb}}^4)
\]

where

- \( \varepsilon_{\text{cav}} \) represents the emissivity of the cavity, the value of which is almost equal to the absorbance
  of the cavity (should smaller than 1.0);
- \( \sigma \) is Stefan Boltzmann’s constant, which is \( 5.67 \times 10^{-8} \text{W/(m}^2\text{K}^4) \);
- \( A_{\text{ap}} \) is the aperture area of the cavity receiver in \( \text{m}^2 \);
- \( T_{\text{cav}} \) is the average interior cavity temperature;
- and \( T_{\text{amb}} \) is the ambient temperature.
Again, radiation losses increase as the size of the cavity aperture area increases.

**Sizing of a receiver**

As previously mentioned, the receiver aperture diameter should be slightly larger than the length of the focal image to best reduce thermal losses. Additionally, although a heat pipe receiver is more advanced than a direct illumination receiver, the latter is simpler to build. Therefore, the design of a small direct illumination receiver, which is similar to the one used in Wu’s test, was adopted for this study. Figure 28 visualises the dimension and layout of the target cavity. It should be noticed that to increase the absorber surface area, the internal cavity wall is composed of heater tubes, which contain working fluid inside.

![Figure 28 Sketch design of the cavity](image)

3.4 Stirling engine for solar dish

The Stirling engine designed for this project is an external heat engine. While the absorption surface of the cavity receiver collects heat energy from the sun, the receiver transmits the energy to the piston heater head. As the solar resource comes from outside of the Stirling engine and the working medium is sealed inside the cylinder, no exhaust gasses are generated during the operation of this Stirling engine; it is thus a closed-loop system. Because Stirling engines are still not commercially available at the time of writing, only the theoretical design of a beta-type Stirling engine for our target solar dish, which follows the Franco’s method [44], is explained in this section.
3.4.1 Geometry of the target Stirling engine

To design a Stirling engine, it is necessary to obtain the geometrical measurements of the target Stirling engine. In Franco’s method, the key features of the Stirling engine that must be determined are the piston swept volume in the expansion space in cm$^3$, piston swept volume in the compression space in cm$^3$, minimum expansion space in cm$^3$, minimum compression space in cm$^3$, phase angle in degrees, volume of regenerator in cm$^3$, cross-section area and wetted perimeter of each heater and cooler tube in cm$^2$ and cm, length of each heater and cooler tube in cm and their numbers, heater and cooler temperature in k, average pressure and gas constant in Pa and J/(kg.k), Cp and Cv of working gas in J/(kg.k), and viscosity in N.s/m$^2$ [44]. In this section, parameters that are required for Franco’s method are determined for a beta-type Stirling engine.

First group of parameters

According to Franco’s method, the measurements that are required for the first group include the swept volume in the expansion and compression spaces, minimum volume in expansion and compression spaces, and phase angle [44]. In the expansion process of a Stirling cycle, working fluid that is pushing to the hot end heats up and expands, and the piston is then driven outward. The volume that is swept by the piston during this process is defined as the piston swept volume in the expansion space. In the compression process, most of the expanded working fluid is injected into the cool side. The working fluid is then cooled and contracted, which drives the piston inward. Similarly, the swept volume in the compression space is the volume that is swept by the piston in the cooling process. Minimum volume in expansion and compression spaces are the dead volume (unswept volume) in the expansion and compression spaces. The phase angle is a constant angle that helps the displacer leads the power piston. However, as a beta-type Stirling engine only has one cylinder in this design and the swept volume is highly dependent on the crank angle, it is difficult to directly calculate the piston swept volume in the expansion space and compression space. Base on Franco’s method, it is
better to determine the changes in the expansion space and compression space as a function of the crank angle first, then the piston swept volume in the expansion space and compression space can be determined from the curve of the two equations [44].

For a beta-type Stirling engine, the expansion space can be calculated by

\[ V_e = V_{\text{min},e} + \frac{1}{2} X_d A_{de} - \frac{1}{2} X_d A_{de} - \frac{1}{2} X_d A_{de} \sin(\theta + \alpha) \]

And the compression space can be determined by

\[ V_c = V_{\text{dead},c} + \frac{1}{2} X_p A_p + \frac{1}{2} X_d A_{dc} - \frac{1}{2} X_p A_p \sin(\theta) + \frac{1}{2} X_d A_{dc} \sin(\theta + \alpha) \]

Where \( A_{de} \) is the displacer area, \( A_{dc} \) is the area of the displacer minus the area of displacer rod, \( A_p \) is the power piston area, \( X_d \) is the stroke of the displacer, \( X_p \) is the stroke of the power piston, \( V_{\text{min},e} \) is the dead volume in the expansion space, \( V_{\text{dead},c} \) is the dead volume in the compression space, and \( \alpha \) is the phase angle. Figure 29 presents the change of expansion and compression spaces as a function of the crank angle. The swept volume in the expansion space and compression space (313 cm\(^3\) and 376 cm\(^3\)) and the minimum volume in the expansion space and compression space (237 cm\(^3\) and 260 cm\(^3\)) can then be observed from the graph.

![Graph showing the change of expansion and compression space as a function of the crank angle.](Image)

*Figure 29 Pistons performance*

**Second group of parameters**

The second group of data includes the regenerator void volume, flow cross-section area, the
wetted perimeter, flow length, and number of heater and cooler tubes [44]. This group of data is related to the design of the regenerator and its heater and cooler. It is not necessary to integrate a regenerator to the beta-type Stirling engine, but the performance of the Stirling engine is improved if a regenerator is equipped with it. Figure 30 shows the outline of a regenerator; the cooler and heater are composed of some small heater or cooler tubes whereas the regenerator is usually filled with of a dense matrix that consists of stacked metal screens [44].

![Regenerator sketch design](image)

The flow cross-sectional area is the interior aperture area of a single tube in cm$^2$. The wetted perimeter is the interior perimeter of a single tube in cm, while the flow length is defined as the total length of all tubes in cm. As the interior diameter of the heater and cooler tubes are suggested within 2 mm to 5 mm and the ratio of the length of a single tube to the interior diameter is 80:1, 30 heater and cooler tubes with an interior diameter of 3 mm and a length of 240 mm were selected. The heated volume of the heater then can be calculated as [44]

$$V_{heater} = A_{heater}L_{heater}N_{heater}$$

Where $A_{heater}$ is the flow cross-section area of a single heat tube (0.071 cm$^2$), $L_{heater}$ is the flow length of each heater tube (24 cm), and $N_{heater}$ is the number of heater tube (30). So, $V_{heater}$ is 51.12 cm$^3$.

Similarly, the cooled volume can be calculated as [44]

$$V_{cooler} = A_{cooler}L_{cooler}N_{cooler}$$

Where $A_{cooler}$ is the flow cross-section area of a single heat tube (0.071 cm$^2$), $L_{cooler}$ is the flow length of each heater tube (24 cm), $N_{cooler}$ is the number of heater tubes (30).
Therefore, $V_{\text{cooler}}$ should be 51.12 cm$^3$ as well.

The wetted perimeter of the heater and cooler tubes can be calculated as

$$P = \pi d = 0.942\text{cm}$$

Based on Franco’s method, the void volume of the regenerator can be determined by [44]

$$V_{\text{regenerator}} = V_{\text{heater}} + V_{\text{cooler}}$$

In this case, $V_{\text{regenerator}}$ is 102.24 cm$^3$.

**Third group of parameters**

In this section, the heater and cooler temperatures are determined. The cooler temperature is 293.15 K, but it is difficult to determine the heater temperature. Based on the two examples in Section 2.3, the temperature of the heater is assumed as 793.15 K.

**Fourth group of parameter**

The average pressure in the Stirling engine affects its performance. To calculate the average engine pressure, hourly engine pressure is simulated by SAM (System Advisor Model), and the average value is generated by the Microsoft Excel program (1,082,019.647 Pa). In this thesis, an average pressure of 1,000,000 Pa is adopted.

**Fifth group of parameter**

The use of different working fluids in a Stirling engine results in different power outputs. In our design, helium was chosen as the working fluid in the Stirling engine. The gas constant for helium is 2,077 J/kgK; $C_p$ is 5,200 J/kgK; $C_v$ is 3,123 J/kgK; and viscosity is 0.00003 Ns/m$^2$ [44].

**Stirling engine simulation input data summary**

A summary of key factors of the target Stirling engine is given below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Swept volume in expansion space (cm$^3$)</td>
<td>313</td>
</tr>
<tr>
<td>Swept volume in compression space (cm$^3$)</td>
<td>376</td>
</tr>
<tr>
<td>Minimum volume in expansion space (cm$^3$)</td>
<td>237</td>
</tr>
<tr>
<td>Minimum volume in compression space (cm$^3$)</td>
<td>260</td>
</tr>
<tr>
<td>Parameter</td>
<td>Value</td>
</tr>
<tr>
<td>--------------------------------------------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>Phase angle (degrees)</td>
<td>90</td>
</tr>
<tr>
<td>Regenerator void volume (cm$^3$)</td>
<td>102.24</td>
</tr>
<tr>
<td>Flow cross-section area of each heater tube (cm$^2$)</td>
<td>0.071</td>
</tr>
<tr>
<td>Flow cross-section area of each cooler tube (cm$^2$)</td>
<td>0.071</td>
</tr>
<tr>
<td>Wetted perimeter for each heater tube (cm)</td>
<td>0.942</td>
</tr>
<tr>
<td>Wetted perimeter for each cooler tube (cm)</td>
<td>0.942</td>
</tr>
<tr>
<td>Flow length of each heater tube (cm)</td>
<td>24</td>
</tr>
<tr>
<td>Flow length of each cooler tube (cm)</td>
<td>24</td>
</tr>
<tr>
<td>Number of heater tubes</td>
<td>30</td>
</tr>
<tr>
<td>Number of cooler tubes</td>
<td>30</td>
</tr>
<tr>
<td>Heater temperature (K)</td>
<td>793.15</td>
</tr>
<tr>
<td>Cooler temperature (K)</td>
<td>293.15</td>
</tr>
<tr>
<td>Average pressure (Pa)</td>
<td>1000000</td>
</tr>
<tr>
<td>Gas constant (J/[kg.k])</td>
<td>2077</td>
</tr>
<tr>
<td>Cp of working gas (J/[kg.K])</td>
<td>5200</td>
</tr>
<tr>
<td>Cv of working gas (J/[kg.K])</td>
<td>3123</td>
</tr>
<tr>
<td>Viscosity (N.s/m$^2$)</td>
<td>0.00003</td>
</tr>
<tr>
<td>Regenerator constant eps1 (dimensionless)</td>
<td>0</td>
</tr>
<tr>
<td>Regenerator constant eps2 (dimensionless)</td>
<td>0</td>
</tr>
<tr>
<td>Scale factor to attain average desired pressure at desired engine speed (default = 1.0)</td>
<td>1</td>
</tr>
</tbody>
</table>

Figure 31 shows the integration of the cavity receiver and the target Stirling engine. The main issue in this design is the problem of sealing the working piston. Most of the researchers noted the addition of rings for the working piston to improve its sealing performance. The simulation
Chapter 4 Simulation results and optimizations

The performance of the target Stirling dish system is affected by many factors. The Stirling engine power output increases with increasing solar intensity, which means any improvement to enhance the incoming solar radiation is beneficial to the system operations. Also, optimisations related to the Stirling engine can improve the system performance as well. In this section, SAM (System Advisor Model) is used to analyse the target system performance. The most important aspect is that the expansion temperature of the Stirling engine is set as constant in SAM. However, it should vary sinusoidally along with the operation of two pistons. Moreover, SAM fails to provide a way to add a regenerator to improve the Stirling engine performance. Besides, SAM can only simulate the performance of the Stirling engine at a constant speed. For this reason, the simulation results generated by SAM are not especially accurate; they act more like a reference. The program provided by Franco simulates the Stirling engine in a more accurate manner, but it does not have a function to simulate the dish collector and cavity performance. As a result, the location and solar resource, dish collector performance, reference power output of the Stirling engine, and cavity receiver performance were simulated with SAM, whereas Franco’s program generated a more detailed account of the performance of the Stirling engine.
4.1 SAM (System Advisor Model) simulation results

The simulation results generated from SAM are divided into four parts in this section, which are location and resource, collector, receiver, and Stirling engine.

4.1.1 Location and resource

Figure 32 was taken from SAM and it gives the location and solar resource of the city of Perth. As is shown in Figure 32, Perth is located at -31.93° N 115.95° E, which means there is ample opportunity to harvest solar energy. With an elevation of 29 m, the air density stays in a normal condition, which does not lead to an increase of convection loss. Additionally, the mild annual average temperature and wind speed contribute to maintaining the performance of the system as well.

![Figure 32 Target place location and resource](image)

4.1.2 Collector

The selection of the reflector material that is configured on the collector can have a significant impact directly on the simulation result. In Figure 33, some solar reflector materials are shown [45]. Among these products, most of the reflectors are made of aluminium as aluminium is highly reflective. Therefore, the reflector that is composed of aluminium was selected to operate as a reflector in our project as well.

As the writer does not have the copyright, the picture is omitted from the electronic version of
the thesis. The picture can be found in reference [45]

Figure 33 Properties of reflector materials [45]

Figure 34 shows the mirror parameters that were input into SAM, including a reflectivity of 90%. Total mirror area is the parabolic surface area of the satellite dish in m$^2$. Projected mirror area is the area of mirror that is projected on the parabolic collector in m$^2$. In some cases, this area is less than the total mirror area for two possible reasons: the shape of the reflector cannot fully cover the collector and the pedestal of the dish also occupies some area. For our design, the use of spherical aluminium reflectors is assumed, and they just barely cover the collector to collect as much solar radiation as possible.

<table>
<thead>
<tr>
<th>Mirror Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Projected Mirror Area</td>
<td>10.9278 m$^2$</td>
</tr>
<tr>
<td>Total Mirror Area</td>
<td>10.0278 m$^2$</td>
</tr>
<tr>
<td>Reflectance</td>
<td>0.9 (0.1)</td>
</tr>
</tbody>
</table>

Figure 34 Collector parameters

Figure 35 displays the collector performance for 12 months in a year. In this section, the results of the collector thermal power incident and produced collector thermal power are explicated. The x-axis represents the hour of the day whereas the y-axis is the kW that are collected by the collector. The collected and produced thermal energy always reach their peak during noontime in each month. From Figure 36, it is clear that the maximum power incidents on the collector in the month of January (2.987 MWh) and the minimum power is collected in June (0.967 MWh). As a result, the collector also produces the most power in January (2.680 MWh) and the least power in June (0.870 MWh). The difference between the collector thermal power incident and output is caused by system errors and the reflectivity of reflectors, but the results are acceptable.
4.1.3 Receiver

Our direct illumination receiver as modelled in SAM is described in this section. Solar radiation reflected from the collector is directly absorbed by absorber tubes that contain the working fluid. Differing from the cavity receiver used in Wu’s test, our design uses long absorber tubes to replace the heating coil in the internal side of the cavity wall, which supplies a larger absorber area. The cavity receiver simulation result is given in Figure 37. The receiver thermal power input is exactly consistent with the thermal power produced by the collector. However, because of the existence of radiation, convection, and conduction thermal losses, there is a gap between the receiver thermal power output and receiver thermal power input. The most
significant thermal losses appear in December, which is 0.638 MWh (shows in Figure 38). However, although there are great thermal losses, the power that is transmitted to the Stirling engine is still significant, and it measures at 2.156 MWh in January and 1.746 MWh in December.

![Figure 37 Annually receiver performance](image)

### 4.1.4 Stirling engine

To supply reference data, SAM was used to simulate the performance of the Stirling engine first. In SAM, the model that was used to simulate the Stirling engine was created based on the Beale curve-fit equation with temperature correction. The receiver simulation provides the data for the input power. Hourly average output power is generated based on several
considerations, which are the Beale curve-fit equation, pressure curve-fit equation, the engine displacement and operating speed, expansion space temperature, and compression space temperature. The rotational speed of the engine drive shaft was defined as 1500 rpm, and the expansion temperature was assumed to be 793.15 K. With a displaced piston volume of 0.000689 m$^3$, Figure 39 presents the hourly engine power output in kW for one year. The trend of power output is consistent with the receiver thermal power output. After the hourly data table was sent to Microsoft Excel, the average power of the Stirling engine was calculated, with a result of 1.455 kW. According to the simulation result displayed in Figure 40, the gross engine power output reaches its maximum in January, which is 0.7 MWh, and meets the minimum in July with 0.182 MWh. Additionally, it is clear that the power output from the Stirling engine significantly fluctuates in one day, which is harmful to some household applications.

![Figure 39 Annual Stirling engine performance](image)
More detailed results were generated from Franco’s program. Figure 41 shows the changes of power and the thermal efficiency of the Stirling engine as a function of engine speed. In the Stirling engine, helium is used as working fluid, which has better properties than air. According to the simulation result, the Stirling engine reaches its maximum power at an engine speed of about half of its maximum speed. Furthermore, a Stirling engine that operates at high speed does not necessarily operates with a high thermal efficiency. In contrast, better thermal efficiency is indicated when the Stirling engine operates at low speed and power. However, if it is necessary for the engine to generate more power, it has to operate at higher speed, although its thermal efficiency then decreases. As the working fluid shuttles back and forth in the regenerator and heater and cooler tubes, there are pumping losses in the Stirling engine; the losses grow with the increase of the shuttle speed. As a result, when the power of the Stirling engine drops to zero, the Stirling engine operates at its maximum speed with its lowest thermal efficiency. To achieve the maximum power (1388.959 W) for our system, the speed of the Stirling engine should be set at 23 Hz. This simulation result is similar to the result generated by SAM.
Figure 41 Power and thermal efficiency vs speed when helium is used as working fluid

To avoid the engine reaching its maximum speed (power output decreases to 0), a generator is connected to maintain its operation at 23 Hz. Figure 42 expresses the temperature and pressure performance of the Stirling engine. The maximum compression pressure is higher than the maximum dead volume pressure, which is followed by the expansion space pressure. The temperature fluctuates mildly in the expansion and compression spaces.
The regenerator performance is shown in Figure 43. The heat that is sent to the regenerator is slightly greater than the heat leaving the regenerator in a single cycle.

Figure 43 Regenerator performance at an engine speed of 23 Hz

Figure 44 displays the expansion space work generated by the Stirling engine per cycle (A₁, area of the blue circle, 158.49 J), whereas Figure 45 shows the compression space work per cycle...
(A₂, area of the yellow circle, 98.1 J). The Stirling engine net work per cycle can be calculated by $A_1 - A_2$, which equals 60.39 J.

Figure 44 Expansion space pressure vs expansion space volume

Figure 45 Compression space pressure vs compression space volume
4.2 System optimisation

In this section, the optimisation of the collector, cavity receiver, and Stirling engine to improve the overall system performance are explained, and the simulation of the optimised versions is detailed. In this step, the chosen method involved changing one of the input datum and observing the change in simulation results of the adjusted data. Then, the results of the simulation were analysed. Again, this chapter only indicates the reference of the changes in the Stirling engine power output because the model is not very accurate in SAM. The simulation results provided by SAM for the solar dish collector and cavity receiver, on the other hand, are reliable. More accurate simulation results of the Stirling engine were also generated by Franco’s program.

4.2.1 Optimisation for collector

To collect maximum solar radiation, improvements on the collector can be divided into two groups: methods to increase the mirrored surface area and possibilities to equip the reflector with higher reflection. The available mirror area on the satellite dish is 10.928 m², so spherical aluminium panels that just cover the full mirror area were chosen as the reflector to maximise the input solar radiation in our design. In this way, a higher incoming solar radiation can be collected by adopting a larger satellite dish to achieve a larger mirrored area. The tables in Figure 46 offers the results of two cases; the right side of Figure 46 demonstrates the simulation result of a solar dish with a diameter of 4 m with other parameters remaining constant, whereas the left part indicates the result of our target satellite dish. It is clear that both the power incident on the collector and the power entering the receiver from the collector can reach a higher level by increasing the size of the satellite dish.
Although the Stirling engine model supplied in SAM is not very accurate, the results are still reliable. A line chart is given to demonstrate the relationships between the dish diameter and Stirling engine annual energy output. Figure 47 presents the simulation results of the changes in the Stirling engine annual energy output as a function of the dish diameter.

Another way to maximise the solar radiation collection is to adopt better reflector panels. The left side of Figure 48 explains the results of replacing the reflector material with polymeric film non-metal mirrors with 0.98 reflectivity, and the right part of the figure is the result of our target design (0.9 reflectivity). Since the size of the satellite dish remains constant, the power incident on the collector of these two designs is the same. However, when a better reflector...
material is adopted, the power entering the receiver from the collector is higher.

![Figure 48 Optimised reflector performance (Left one) vs original reflector performance (Right one)](image)

A line chart which gives the changes of the Stirling engine annual energy output as a function of the reflectiveness of the reflector is given (shown in Figure 49). To reflect as much solar radiation as possible to the receiver, the reflector with high reflectivity (more than 0.8) has to be chosen.

![Figure 49 Optimised reflector performance](image)

4.2.2 Optimisation for cavity receiver

In our design, the aperture of the cavity is sized to just cover the length of the focal image to reduce heat losses. Additionally, the thickness of the cavity wall was designed at 60 mm to increase the insulation performance, and the solar absorption of the heater tubes absorber and cavity were designed to reach 0.9 to absorb as much power as possible. For further improvements in the design’s thermal performance, the cavity depth can be decreased to reduce the surface area of the cavity, which results in the decrease of thermal losses. In this section, the performance of a cavity with a depth of 50 mm is simulated. In Figure 50, the left
part of the figure is the performance of the optimised cavity receiver, and the right side shows the result of the target cavity receiver. It can be seen that the receiver output power increases, but also the thermal receiver losses decrease.

![Figure 50 Optimised cavity performance vs original cavity performance](image)

However, the thermal loss reaches its minimum when the depth of the cavity is set as around 30 mm. Less system energy output is obtained when the depth of the cavity is less than 30 mm. The changes of the system energy output as a function of cavity depth is shown in Figure 51.

![Figure 51 Optimised cavity performance](image)

### 4.2.3 Optimisation for Stirling engine

The optimisation for Stirling engines has been studied for many years. Nowadays, the additions of a regenerator and improvements to the crank mechanisms are widely used in Stirling engine design. In our design, the working piston and displacer were geared in a 90° phase shift by
linking their respective connecting rods to a yoke equipped on the centre of the flywheel. A sketch design is displayed in Figure 31. As can be seen, both the connection rods of displacer piston and working piston leave the cylinder through the cold area. It should be noticed that the work piston has to be designed with well with a heat sealing property. This type of crank mechanism is not the most effective, but it is the easiest design.

To optimise the performance of the Stirling engine, pumping losses can be minimised by adding the heater and cooler tubes in the regenerator. As mentioned in Franco’s research, friction losses exist in the heater and cooler tubes as the working fluid shuttles back and forth in the regenerator. Figure 52 shows the simulation results of a Stirling system with a different number of heater and cooler tubes equipped in the regenerator. The figure shows that the output power increases as the number of heater and cooler tubes increases. When the Stirling operates at 43 Hz, a maximum power of 1849.546 W is obtained when 180 heater and cooler tubes are used in the regenerator. The power of the Stirling engine begins to lessen when the total number of heater and cooler tubes is larger than 180.

<table>
<thead>
<tr>
<th>Engine speed (Hz)</th>
<th>17</th>
<th>20</th>
<th>25</th>
<th>28</th>
<th>30</th>
<th>33</th>
<th>35</th>
<th>37</th>
<th>40</th>
<th>41</th>
<th>42</th>
<th>43</th>
<th>44</th>
<th>45</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum output power (W)</td>
<td>1997.775</td>
<td>1598.021</td>
<td>1399.985</td>
<td>1299.626</td>
<td>1199.407</td>
<td>1099.188</td>
<td>999.969</td>
<td>899.750</td>
<td>799.527</td>
<td>699.190</td>
<td>599.895</td>
<td>599.205</td>
<td>598.902</td>
<td>598.699</td>
</tr>
<tr>
<td>Number of heater tubes</td>
<td>20</td>
<td>25</td>
<td>30</td>
<td>35</td>
<td>40</td>
<td>45</td>
<td>50</td>
<td>55</td>
<td>60</td>
<td>65</td>
<td>70</td>
<td>75</td>
<td>80</td>
<td>85</td>
</tr>
<tr>
<td>Number of cooler tubes</td>
<td>20</td>
<td>25</td>
<td>30</td>
<td>35</td>
<td>40</td>
<td>45</td>
<td>50</td>
<td>55</td>
<td>60</td>
<td>65</td>
<td>70</td>
<td>75</td>
<td>80</td>
<td>85</td>
</tr>
<tr>
<td>Total number of tubes</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td>70</td>
<td>80</td>
<td>90</td>
<td>100</td>
<td>110</td>
<td>120</td>
<td>130</td>
<td>140</td>
<td>150</td>
<td>160</td>
<td>170</td>
</tr>
</tbody>
</table>

*Figure 52 Optimised regenerator performance data*

A line chart that presents the changes in maximum Stirling engine power and its corresponding speeds as a function of some tubes used in the regenerator is shown in Figure 53. Both changes become less significant as increasing numbers of heater and cooler tubes are used in the
Another way to enhance the engine performance is to use a fluid that functions better. The working fluid that was used in the simulation is helium, which performs better than air. To further improve the Stirling engine power output, hydrogen can be used to fill the Stirling engine to act as the working fluid. The properties of air, helium, and hydrogen are supplied in Figure 54.

<table>
<thead>
<tr>
<th></th>
<th>Air</th>
<th>Helium</th>
<th>Hydrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas constant (J/Kg.K)</td>
<td>287</td>
<td>2077</td>
<td>4120</td>
</tr>
<tr>
<td>Cp (J/Kg.K)</td>
<td>1007</td>
<td>5200</td>
<td>14310</td>
</tr>
<tr>
<td>Cv (J/Kg.K)</td>
<td>720</td>
<td>3123</td>
<td>10190</td>
</tr>
<tr>
<td>Viscosity (N.s/m^2)</td>
<td>0.00003</td>
<td>0.00003</td>
<td>0.000015</td>
</tr>
</tbody>
</table>

The simulation results are given in Figure 55, it is clear that the Stirling engine can generate more power if hydrogen is used as the working fluid.
4.3 Final design of the project

A sketch of the target system is shown in Figure 56. The Stirling engine sits in an insulated box that is painted in black. The cavity emerges from the aperture of the box, and the cavity aperture sits at the focal point. The receiver box is connected to the edge of the dish collector through two support connection rods. The plane formed by the two rods that are near to the cavity is perpendicular to the dish aperture. The dish collector that sits on the pedestal can always face toward the sun by adopting the tracking system designed by other students.
Regardless the cost of the project, the parameters for our most advanced design are given below.

<table>
<thead>
<tr>
<th>Tracking system</th>
<th>azimuth elevation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total system error (mrad)</td>
<td>11</td>
</tr>
<tr>
<td>Aperture diameter (m)</td>
<td>3.6</td>
</tr>
<tr>
<td>Dish depth (m)</td>
<td>0.5</td>
</tr>
<tr>
<td>Rim angle (degrees)</td>
<td>58.103</td>
</tr>
<tr>
<td>Focal length (m)</td>
<td>1.62</td>
</tr>
<tr>
<td>Collector system errors (mrad)</td>
<td>11</td>
</tr>
<tr>
<td>Beam spread with errors (m)</td>
<td>0.093</td>
</tr>
<tr>
<td>focal image width (m)</td>
<td>0.176</td>
</tr>
<tr>
<td>Cavity receiver aperture diameter (m)</td>
<td>0.176</td>
</tr>
<tr>
<td>Cavity receiver internal depth (m)</td>
<td>0.03</td>
</tr>
<tr>
<td>Cavity receiver length (m)</td>
<td>0.1</td>
</tr>
<tr>
<td>Parameter</td>
<td>Value</td>
</tr>
<tr>
<td>---------------------------------------------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>Cavity receiver thickness (m)</td>
<td>0.06</td>
</tr>
<tr>
<td>Absorber surface area (m²)</td>
<td>0.041</td>
</tr>
<tr>
<td>Swept volume in expansion space (cm³)</td>
<td>313</td>
</tr>
<tr>
<td>Swept volume in compression space (cm³)</td>
<td>376</td>
</tr>
<tr>
<td>Minimum volume in expansion space (cm³)</td>
<td>237</td>
</tr>
<tr>
<td>Minimum volume in compression space (cm³)</td>
<td>260</td>
</tr>
<tr>
<td>Phase angle (degrees)</td>
<td>90</td>
</tr>
<tr>
<td>Regenerator void volume (cm³)</td>
<td>306.72</td>
</tr>
<tr>
<td>Flow cross-section area of each heater tube (cm²)</td>
<td>0.071</td>
</tr>
<tr>
<td>Flow cross-section area of each cooler tube (cm²)</td>
<td>0.071</td>
</tr>
<tr>
<td>Wetted perimeter for each heater tube (cm)</td>
<td>0.942</td>
</tr>
<tr>
<td>Wetted perimeter for each cooler tube (cm)</td>
<td>0.942</td>
</tr>
<tr>
<td>Flow length of each heater tube (cm)</td>
<td>24</td>
</tr>
<tr>
<td>Flow length of each cooler tube (cm)</td>
<td>24</td>
</tr>
<tr>
<td>Number of heater tubes</td>
<td>90</td>
</tr>
<tr>
<td>Number of cooler tubes</td>
<td>90</td>
</tr>
<tr>
<td>Heater temperature (K)</td>
<td>793.15</td>
</tr>
<tr>
<td>Cooler temperature (K)</td>
<td>293.15</td>
</tr>
<tr>
<td>Average pressure (Pa)</td>
<td>1000000</td>
</tr>
<tr>
<td>Gas constant (J/[kg.k])</td>
<td>4120</td>
</tr>
<tr>
<td>Cp of working gas (J/[kg.K])</td>
<td>14310</td>
</tr>
<tr>
<td>Cv of working gas (J/[kg.K])</td>
<td>10190</td>
</tr>
<tr>
<td>Viscosity (N.s/m²)</td>
<td>0.000015</td>
</tr>
<tr>
<td>Regenerator constant &amp;eps1 (dimensionless)</td>
<td>0</td>
</tr>
<tr>
<td>Regenerator constant ϵ² (dimensionless)</td>
<td>0</td>
</tr>
<tr>
<td>Scale factor to attain desired average pressure at desired engine speed (default = 1.0)</td>
<td>1</td>
</tr>
</tbody>
</table>

With the parameters given in the table, the performance of the optimised system is shown in the Figure 57. With a thermal efficiency of 43.021%, the Stirling engine operates at 64 Hz and reaches its maximum power of 2.7 kW.

![Power and thermal efficiency vs Speed for Hydrogen](image)

*Figure 57 Final optimised Stirling engine power output performance*

Figure 58 shows the performance of the optimised Stirling engine. Compared with the design simulated in Section 4.1.4, it has more stable pressure in the expansion space, compression space, and dead volume.
The performance of the optimised regenerator is expressed in Figure 59. Compared with the regenerator introduced in Section 4.1.4, the optimised version transfers more energy to the compression and expansion spaces. The heat that is sent to the regenerator (positive) is slightly greater than the heat leaving the regenerator (negative) in a single cycle.
Figure 60 shows the optimised expansion space work generated by the Stirling engine per cycle (area of the blue circle, 97.194 J), whereas figure 61 shows the compression space work per cycle (area of the yellow circle, 55.413 J). The Stirling engine net work per cycle can be calculated by $A_1 - A_2$, which is 41.78 J, which is less than the design discussed in Section 4.1.4. However, the operation speed of the optimised system is 64 Hz. Therefore, the energy generated by the optimised Stirling engine in 1s is $41.78 \times 64 = 2,673.92$ J, which is larger than the energy generated by the previous one in 1s ($1,388.97$ J).
Figure 60 Final optimised expansion space performance

Figure 61 Final optimised compression space performance
Chapter 5 Conclusion and recommendations

The aim of the project is to design, build and testing of a solar dish system by retrofitting an old satellite dish. The modelling and simulation for retrofitting a satellite dish into a solar parabolic dish that operates at different times of the year in Perth are presented. It is viable to retrofit the old satellite dish as a concentrating solar dish system. After the key geometric factors of the target system were determined, thermal analysis of the solar dish system was generated by SAM (System Advisor Model) and the program made by Franco. In Perth, which is located in western Australia, the target solar dish Stirling engine generated a maximum energy about 0.714 MWh in January. This thesis first explained the key parameters of the satellite dish, such as the diameter of the parabolic dish concentrator, size of the aperture area of concentrator and receiver aperture length, the focal length of the parabolic dish, geometric concentration ratio, and rim angle. Then, it considered several factors that impact the performance of the solar dish, such as the size of the dish collector, system errors, collector reflectivity, receiver thermal losses and regenerator composition. It can be concluded that the incident solar radiation on the collector can be improved by increasing the size of the satellite dish and the area of the mirrored portion on it. Moreover, reflector panels with higher reflectivity can lead to a higher level of incoming solar power as well. The surface area of the cavity receiver also impacts its thermal losses. According to the simulation result, lower thermal losses can be achieved by decreasing the depth of the receiver. However, if the cavity receiver is too shallow, its power output decreases as well. The engine performance was analysed in detail using Franco’s program. Its performance can be improved by adopting the appropriate number of heater and cooler tubes to reduce pumping losses and by using more effective working fluid. The focus of the future work is to improve the sealing performance of the Stirling engine and find the appropriate material to retrofit the satellite dish and build the Stirling engine. Also, a heat resisting black paint should be chosen to coat the cavity receiver to improve its thermal absorption ability.
Appendix

To investigate the temperature performance of a cavity receiver, the receiver of the solar dish that was placed on the roof area of the engineering energy building was retrofitted with a cavity receiver. Figures 62 and 63 show the original receiver of the solar dish and its performance. The original receiver was composed of a metal plate absorber and a water tank attached to it. It can be seen that the receiver failed to collect all of the reflected solar radiation. To improve its performance, a cavity was built and attached to the absorber.

![Original receiver without insulation material](image-url)
Figure 63 Original receiver performance

Figure 64 shows the retrofitted receiver. It supplies a larger aperture area to absorb as much solar radiation as possible. To further improve its performance, it was painted in black (shows in Figure 65).

Figure 64 Retrofitted cavity receiver
Figure 66 shows the optimised solar dish receive at the roof area of the engineering energy building. Ideally, the cavity receiver can reach its maximum temperature if the test is conducted around 1pm. However, the tracking system of the solar dish failed to track the sun during that time as the sun was so high that the solar radiation could not be focused into the cavity aperture.
A temperature of 220° C was obtained from the thermal couple at 3pm. It can reach 300° C if the solar dish can track the sun during noontime. However, the black paint could not survive the high temperature, so it was cooked (shows in figure 67). Further improvements should be made regarding the paint issue.
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