June 2021

Advanced Exergy Analysis of a WhisperGen Stirling Engine System

Akash Mathew
mathe117@uwindsor.ca

Advanced Exergy Analysis of a WhisperGen Stirling Engine System

by

Akash Mathew

A Major Research Paper
Submitted to the Faculty of Graduate Studies
through the Department of Mechanical, Automotive and Materials Engineering
in Partial Fulfillment of the Requirements for
the Degree of Master of Applied Science
at the University of Windsor

Windsor, Ontario, Canada

2021

Advanced Exergy Analysis of a WhisperGen Stirling Engine System

by

Akash Mathew

APPROVED BY:

____
N. Eaves
Department of Mechanical, Automotive and Materials Engineering

____
D. Ting, Co-Advisor

Department of Mechanical, Automotive and Materials Engineering

____
G. Reader, Co-Advisor

Department of Mechanical, Automotive and Materials Engineering

April 15, 2021

DECLARATION OF ORIGINALITY

I hereby certify that I am the sole author of this major paper and that no part of this major paper has been published or submitted for publication.

I certify that, to the best of my knowledge, my major paper does not infringe upon anyone's copyright, nor does it violate any proprietary rights, and that any ideas, techniques, quotations, or any other material from the work of other people included in my major paper, published or otherwise, are fully acknowledged in accordance with the standard referencing practices. Furthermore, to the extent that I have included copyrighted material that surpasses the bounds of fair dealing within the meaning of the Canada Copyright Act, I certify that I have obtained a written permission from the copyright owner(s) to include such material(s) in my major paper and have included copies of such copyright clearances in my appendix.

I declare that this is a true copy of my major paper, including any final revisions, as approved by my major paper committee and the Graduate Studies office, and that this major paper has not been submitted for a higher degree to any other University or Institution.

ABSTRACT

With the increasing demand for energy, it has become imperative to look for better and more efficient systems. Combined heat and power systems can provide the required power output while also utilizing the waste heat to increase the overall efficiency of the system. This study analyzed a 1 kW WhisperGen Combined Heat and Power system powered by a Stirling engine. A thermodynamic analysis of the engine was performed to understand the losses occurring within it. In addition, the WhisperGen engine was analyzed in terms of its exergy losses through an advanced exergy analysis on the different components of the engine. The first law efficiency of the engine was calculated to be around 67%. The exergy efficiency of the engine was 30%. The combustion chamber contributed nearly 84% of the total exergy destruction occurring in the engine. While on the other hand, the exhaust heat exchanger had the lowest exergy destruction making it the most efficient component with 45% exergy efficiency. The advanced exergy analysis provided insight into how much each component can be improved. The emphasis should be on improving the combustion chamber, since the avoidable exergy destruction in the chamber was more than 60%. Reducing the avoidable exergy destructions in the components would increase the exergy efficiency of the engine by at least 15%.

DEDICATION

I dedicate this work to my parents, Mr. Mathew Thomas and Mrs. Ansamma Mathew, and my sister Ms. Amala Mathew.

ACKNOWLEDGEMENTS

I would like to thank the Almighty for guiding me along every step of this work. I extend my sincere gratitude towards Dr. David Ting and Dr. Graham Reader for giving me the opportunity to work at this wonderful university. I am grateful for all their guidance during the course of my studies.

I am also indebted to Mr. Brue Durfy, who was always there whenever I had any questions regarding testing and working the engine. I would also like to extend my sincere thanks to Jan Barmentloo of Off Grid Energy, New Zealand for supplying the parts required for the engine.

I am forever obliged to everyone in the Turbulence and Energy Laboratory for all their help. I am also thankful to the Department of Mechanical, Automotive and Materials Engineering for their financial support in the form of my Graduate Assistantship.

I would also like to express my thankfulness to my family for their unconditional support and their confidence in me during my studies. Finally, I cannot thank my friends enough for being there for me, especially Mr. Binto Baiju and Dr. Shiju Easo John, during the entire duration of my master's program.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF ABBREVIATIONS/SYMBOLS

CHP	Combined Heat and Power
C_p	Specific Heat Capacity (kJ/kg·K)
$\dot{E}$	Energy Transfer Rate (kW)
EHE	Exhaust Heat Exchanger
h	Specific Enthalpy (kJ/kg·K)
LHV	Lower Heating Value
$\dot{m}$	Mass Flow Rate (kg/s)
m	Mass (kg)
N	Engine Speed (rpm)
P	Pressure (Pa)
P_{mean}	Mean Pressure (Pa)
$\dot{Q}$	Heat Transfer Rate (kJ/s)
R	Ideal Gas Constant (J/K·mol)
s	Specific Entropy (kJ/kg·K)
SHE	Secondary Heat Exchanger
T	Temperature
t	Temperature Ratio
V	Volume (m^3)
W_C	Compression Energy (kJ)
W_E	Expansion Energy (kJ)
$\dot{W}$	Work (kW)
X	Volumetric Ratio
y	Molar Fraction

Greek

α	Standard Chemical Exergy (kJ/kmol)
η	Efficiency
ε	Specific Exergy (kJ/kg)
$\acute{\varepsilon}$	Exergy Rate (kW)
$\acute{\varepsilon}_d$	Exergy Destruction Rate (kW)
$\acute{\varepsilon}_f$	Fuel Exergy Rate (kW)
$\acute{\varepsilon}_p$	Product Exergy Rate (kW)
$\acute{\varepsilon}_d^{AV}$	Avoidable Exergy Destruction Rate (kW)
$\acute{\varepsilon}_d^{UA}$	Unavoidable Exergy Destruction Rate (kW)
θ	Crank Angle
ϕ	Chemical Exergy Factor

Subscripts

a	Air
c	Compression
co	coolant
ch	Chemical
DC	Dead Volume at Compression End
DE	Dead Volume at Expansion End
E	Expansion
e	Exit
f	Fuel
H	Hot
i	Input

l	Loss
o	Output
R	Regenerator
SC	Swept Volume of Compression
SE	Swept Volume of Expansion
th	Thermal
w	Water

CHAPTER 1
INTRODUCTION AND LITERATURE REVIEW

Energy demands have been escalating in recent years. Between 2000 and 2018, energy production has increased by 1.5 times [1]. Although there has been an increase in the use of renewable sources, much of the energy is still produced from fossil fuels. In 2018, the contribution of fossil fuels to the total global energy production was around 81% [2].

In spite of the significant increase in energy production, electricity still remained inaccessible to 29% of the world population in 1990. In 2000, this percentage dropped to 23%. Over the past two decades, there have been tremendous efforts made by many countries, especially India and China, to close this gap. Despite of all the efforts, there is still about 11% of the world population that does not have access to electricity, according to data published in 2019 [3]. The main difficulty governments face is connecting rural areas to the main grid. Numerous alternative modes of electricity generation have been adopted to tackle this issue. Many experts have suggested remote electricity generation to tackle this issue [4, 5]. One way to accomplish this is by using co-generation systems, more commonly known as combined heat and power systems (CHP).

1.1 Combined Heat and Power Systems

The notion of combined heat and power generation is not new. It was implemented by authorities in cities in the United States as early as the 19th century [6]. In the latter half of that century, civic authorities set up plants to provide electricity to the people, using the heat from these plants to supply hot water and space heating for homes and offices. Different countries, like the United Kingdom, tried to replicate these, but it was not until the early

20th century that it were implemented [6]. This process of providing heat for households was labeled “district heating” in countries like Germany, Russia, Finland, and others. As time went by, these countries, especially the United Kingdom and Finland, started investing massively in CHP technology. Concerns for the environment have also contributed to the adaptation of CHP units, as they can achieve an efficiency of more than 80%, which helps to reduce emissions [7].

The purpose of co-generation is to generate the required power output by utilizing the energy of the fuel, then making good use of the low-grade thermal energy from the waste heat that results from the combustion process. The different types of co-generation systems include dual generation systems and tri-generation systems. CHP systems are also used to meet seasonal demands. There are systems that are used for both heating and cooling, depending on the need; these systems also produce power simultaneously and, thus, are called tri-generation systems. The common barriers associated with CHP systems are their capital cost and legislative challenges [6]. CHP systems use different types of prime movers for their operation. Some of the common prime movers used in CHP systems are tabulated in Table 1.1 [8]. Internal combustion engines have a wide range of power output when used as the prime mover. Gas turbines and steam engines are usually preferred when the required power output is greater than 500 kW. Stirling engines are favored for low power applications because of their high efficiency and fuel flexibility.

Table 1.1 Prime Movers in CHP Systems

Technology	Power Range
Stirling Engine	1 kW – 50 kW
Internal Combustion Engine	5 kW – 10 MW
Microturbine	30 kW – 250 kW
Steam Turbine	500 kW – 100 MW
Gas Turbine	2 MW – 200 MW

CHP systems employing Stirling engines had better heat and power ratios when compared to traditional boilers [9]. Omid et al [10] analysed different prime movers for CHP systems – like the Rankine cycle, Stirling engine, and microturbines – for exergy losses, finding that the maximum exergy destruction occurs in the combustion chambers and boilers of these systems.

Initially, when CHP units first came into existence, internal combustion engines were commonly incorporated as prime movers to generate the required power output. Later, there was a renewed interest in using Stirling engines as prime movers for low power applications due to their low noise and higher efficiency [11, 12]. Heffner [13] looked at the noise levels during the operation of a 3 kW Stirling engine. He found that, when compared to internal combustion engines, there was a reduction of 40 db in the airborne noise. Stirling engines also have an advantage over the other prime movers, as they are external combustion engines. This facilitates the use of different sources of energy, like solar, geothermal, waste heat, and the combustion of various fuels. Solar-powered Stirling engines have been at the forefront of the discussion, amongst others [14, 15].

1.2 Stirling Engine

The Stirling engine is an external combustion engine that works on the Stirling cycle. Figure 1.1 is a flowchart highlighting the major events in the development of Stirling engines. The Stirling cycle was invented in 1816 by Robert Stirling, and the first Stirling engine was developed in 1818 by Robert and James Stirling [16]. The engine developed, which had a rated power of 2 HP, was used for pumping water from an Ayrshire quarry. The air vessels used for the engine, however, had a problem with overheating, which led to the engine shutting down abruptly. The first successful analysis of the Stirling engine was performed by Gustav Schmidt of the German Polytechnic Institute of Prague in 1871 [16]. The first development stage of the Stirling engine ended due to the advancement of internal combustion engines.

In the late 19th and early 20th centuries, Stirling engines were largely forgotten, only produced for small ventilating fans and toys [17]. There was a renewed interest in Stirling engines before the Second World War. Around that period of time, in 1937, Philips Research Laboratory developed a small unit of a Stirling engine that could generate 16 W of electricity [18], which is considered a major development in Stirling engines. The effectiveness of the heat exchangers used in Stirling engines plays a crucial role in the overall efficiency of a Stirling engine [19].

In 1993, a Stirling engine battery charger was developed. In this study, emphasis was given to identifying the key problems associated with Stirling engines. The authors developed a mechanism called the wobble yoke mechanism to be used in the battery charger [20]. A modified version of this mechanism has been used in the WhisperGen Stirling engine. Stirling engines are more notably used in micro-CHP units with a power output of 1 kW to 15 kW, as well as in small-scale units with power output up to 100 kW [21, 22]. There have been many manufacturers that have come forward with their own forms of CHP systems incorporating Stirling engines, most notably WhisperGen Tech, Ballard Power Systems, and Yanmar Holdings, among others. These systems have a power output ranging from 1 kW to 100 kW. There are different Stirling engines available in the market depending on their configurations. The Stirling cycle and the different Stirling engines are discussed in the following sub-sections.

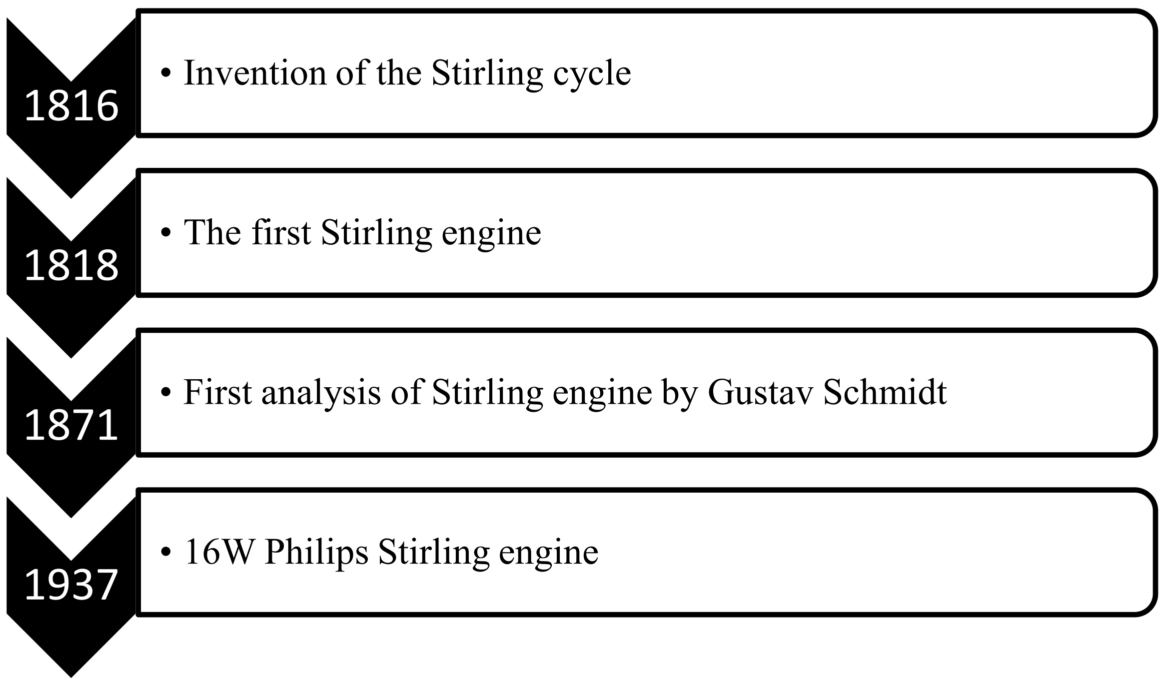

Figure 1.1 Flowchart of Stirling Engine Development

1.2.1 Stirling Cycle

This section describes how a Stirling cycle works, as well as its important characteristics. A Stirling cycle is a closed thermodynamic cycle consisting of four different processes, as shown in Figure 1.2. The figure shows an ideal Stirling cycle with perfect regeneration. The colors red and blue indicate the change in temperature of the operating gas inside the Stirling engine. The figure on the left shows the pressure-volume variations during the cycle, and the figure on the right shows how the entropy changes with respect to the temperature change during the process. $\dot{Q}_h$ is the rate of heat added from the cold side of the engine. $\dot{Q}_c$ is the rate of heat that is released from the fluid to the cold side. $\dot{Q}_{Ro}$ and $\dot{Q}_{Ri}$ are the heat rejection rate and the rate of heat addition to the regenerator, respectively. T_h and T_c are the hot side and cold side temperatures, respectively. A detailed description of the process is presented below.

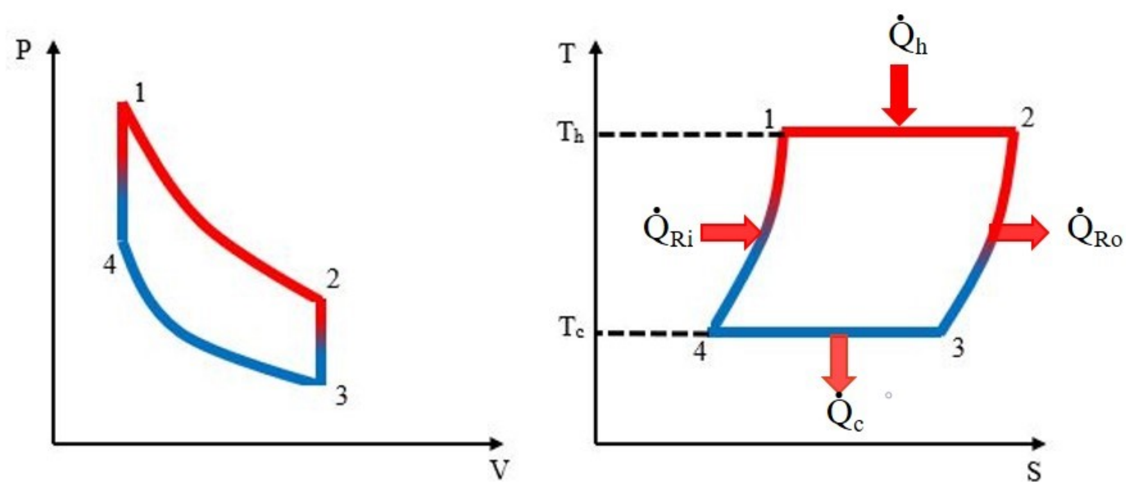

Figure 1.2 P-V and T-S Diagrams of Stirling Cycle

<u>Process 1-2</u>. The process from 1 to 2 is the expansion process. When the expansion happens, heat is added to the fluid under an ideally constant temperature process. Therefore, Process 1-2 is an isothermal heat addition process.

<u>Process 2-3</u>. The fluid exchanges its heat with the regenerator in Process 2-3.

The volume of the fluid remains the same, making this process an isochoric heat removal.

Process 3-4. Process 3-4 is a constant temperature heat removal process. During this, the fluid in the chamber compresses, hence, heat is exchanged with the cold side to maintain the temperature of the fluid. This makes the process an isothermal compression.

Process 4-1. During this process, the fluid absorbs the heat in the regenerator supplied during Process 2-3. The cycle is completed once the fluid returns to its initial stage [23]. The temperature difference between the hot side and the cold side is the crucial factor in the operation of the Stirling engine**.**

1.2.2 Stirling Engine Classifications

Stirling engines are usually classified by their cylinder and piston arrangements. According to their configuration, Stirling engines can be categorized as either an alpha Stirling engine, beta Stirling engine, or gamma Stirling engine. The three different configurations are shown in Figure 1.3.

The alpha configuration has two separate cylinders connected by a regenerator. The pistons are in these separate cylinders, as depicted in Figure 1.3. This is the most widely used configuration. In the alpha configuration, both pistons act as power pistons. The total work produced is the result of the motion of both pistons. Hence, the alpha configuration has a high power-to-weight ratio.

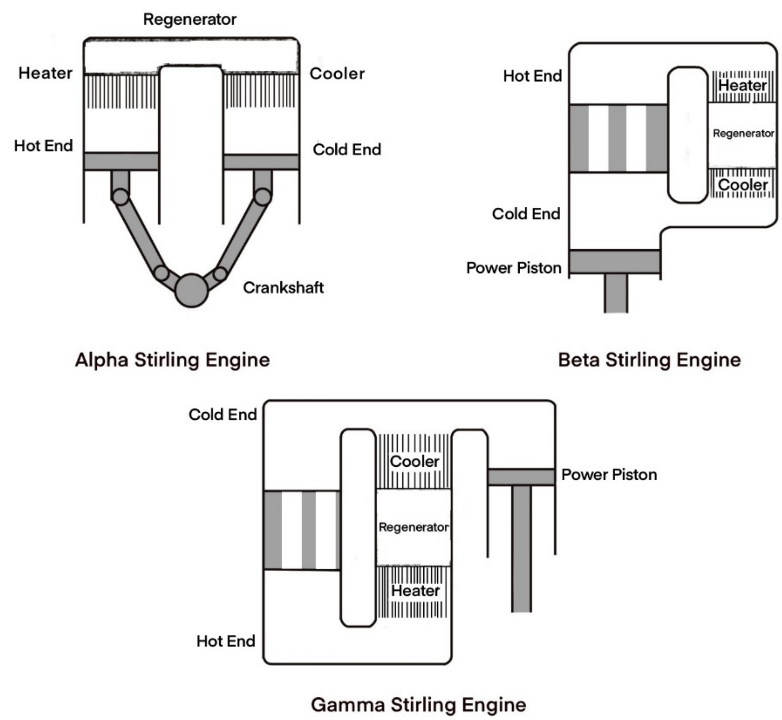

Figure 1.3 Stirling Engine Classification (Redrawn Based on [20]).

The beta configuration has one power piston and a displacer. Unlike the alpha configuration, beta engines have only one cylinder. Both piston and displacer are located in the same cylinder, along the same axis. The purpose of the displacer is to shuttle the gas between the hot side and the cold side, but the displacer does not produce any work output in this configuration. The beta configuration is a very compact design when compared to the others. The overlaps of piston strokes allow better use of the volume. Only using one cylinder also helps in reducing the weight of the engine.

The gamma configuration has the features of both the beta and the alpha. It also has a displacer and a power piston, like in beta engines. For this configuration, however, the displacer and the piston are in separate cylinders. This model provides a convenient separation between the heat exchangers associated with the displacer cylinder and the compression and expansion workspaces associated with the piston. The gas in the two cylinders can freely flow between them with the engine remaining a single body, making this arrangement mechanically simpler than the others [20].

Each of the aforementioned Stirling engine configurations have either one cylinder or two cylinders. A four-cylinder arrangement can be used to increase the total work output, which is shown in Figure 1.4. In the figure, ‘H’ represents the hot side, ‘C’ is the cold side and ‘P’ stands for piston. The four-cylinder double-acting arrangement is a variation of the alpha configuration. In this design, the four cylinders are interconnected with each other. This is achieved by connecting the expansion side of one cylinder to the compression side of another. As can be seen in Figure 1.4, the compression side of the first cylinder is connected to the expansion side of the second cylinder and so on. The compression side of the last cylinder is connected to the expansion side of the first cylinder and, thus, the configuration is completed, with regenerators between consecutive cylinders.

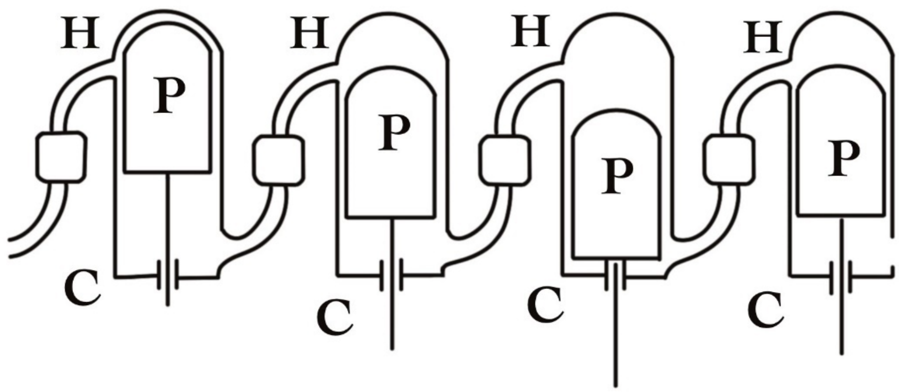

Figure 1.4 Four Cylinder Double-Acting Configuration (Redrawn Based on [20]) 'H' represents the hot side, 'C' is the cold side and 'P' stands for piston

A novel type of Stirling engine is the free-piston Stirling engine. Figure 1.5 shows a model of the first free-piston Stirling engine developed by William Beale. In this type, the piston and displacers are not connected using any mechanical linkages and, thus, reduce the friction generated in the engine [24]. The free piston also makes use of a displacer and power piston. The magnets and spring systems help to move the piston without any mechanical linkages, as shown in the figure. Water is used as a cooling medium to maintain the temperature difference.

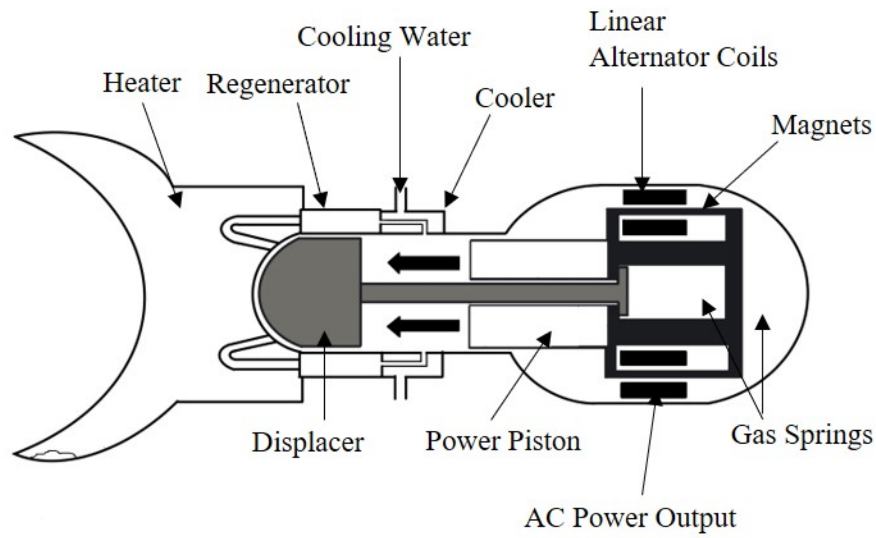

Figure 1.5 Model of Beale's Free-Piston Stirling Engine (Redrawn Based on [24]).

1.3 Objective

Over the years Stirling engines have been used for many purposes but has faced many difficulties in gaining momentum. It is evident that one of the best ways to utilize Stirling engines is to employ it in lower output applications. Using Stirling engine as the prime mover in a CHP system might well be the solution going forward for remote power generation. The WhisperGen Stirling engine is a type of CHP system that employs Stirling engine as the prime mover. There has not been enough analysis performed on the engine to understand the improvements that can be made. The study in this major paper is an attempt in contributing towards the missing work.

1. To analyze a 1kW WhisperGen Stirling engine and calculate the first law efficiency.

2. To perform an exergy analysis to identify the exergy destruction in each component and to calculate the exergy efficiency of the engine.

3. To carry out an advanced exergy analysis to recognize the exergy destruction that can be avoided and to identify the component that can be improved the most.

CHAPTER 2

WHISPERGEN STIRLING ENGINE

The WhisperGen Stirling engine is a micro combined heat and power system. The engine was developed as a co-generation system by a New Zealand-based firm, WhisperGen Tech, primarily for domestic applications. A 1 kW WhisperGen Stirling engine was tested by Farra et al [25] for different fuels like diesel, biodiesel, and ethanol. They found that ethanol and bio-oil can be used as an alternate fuel without compromising efficiency too much. The overall efficiency of the engine while using ethanol was found to be about 85.7%, compared to 85.4% when diesel was used. A similar model of the same engine was tested at the laboratory of Discrete Technology and Production Automation at the University of Groningen [26]. This study mainly dealt with evaluating the wobble yoke mechanism used in the engine. A control system approach was used to analyze the mechanism [26]. Increasing the stability of the mechanism would result in an increase in the overall performance of the engine.

2.1 Overview of WhisperGen Stirling Engine

The studied engine makes use of a double-acting alpha configuration for its operation and is fueled by diesel. Nitrogen is used as the working fluid for the working of the Stirling engine. The nitrogen gas shuttles between the hot side and cold side, thus helping the pistons to alternate. This movement of the pistons is used to generate the power output. The engine is inside an enclosure, as shown in Figure 2.1. The enclosure includes all the critical components for the engine’s work, like the burner, the Stirling engine, exhaust heat exchanger, and so on. The air blower and the fuel line are on the left side of the engine. The fuel tank is suspended above the engine to make use of gravity. The microcontroller is protected by the electronics enclosure. The oxygen sensor, FID (flame ionization detector), and the glow plug are installed on top of the burner to aid in the combustion process.

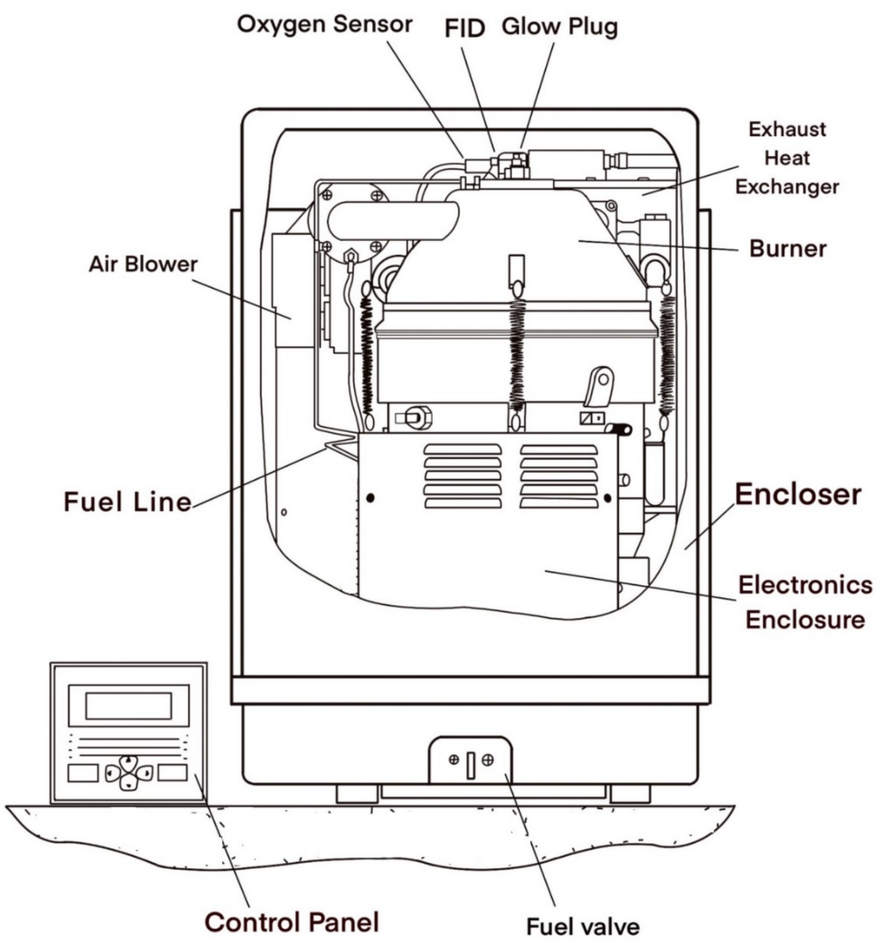

Figure 2.1 WhisperGen Stirling Engine (Redrawn Based on [27]).

The control panel on the left hand side of the engine can be used to remotely control and monitor the engine. The engine can also be connected to a computer, as it comes with its own software called Micromon for logging the data from the engine. The electronic enclosure houses the microcontroller. The specifications of the engine are shown in Table 2.1. The engine uses diesel fuel for its operation. The fuel pump used can deliver a maximum fuel supply of 1l/h. Nitrogen pressure is maintained between 2.6 MPa and 2.8 MPa. The engine is compact in size, with a maximum height of 85 cm.

Table 2.1 Specifications of WhisperGen Engine

Feature	Specification
Prime Mover	Four-cylinder alpha double-acting Stirling cycle
Fuel	No. 2 diesel
Fuel Consumption	1 l/hr maximum
Working Fluid	Nitrogen
Power Output	1 kW nominal
Engine Speed	1200-1500 rpm
Dry Weight	120 kg
Dimensions	390 mm (width) × 550 mm (depth) × 850 mm (height)

2.2 Components of WhisperGen Engine

2.2.1 Air Supply System

The air is drawn into the burner using a 12V DC swirling blower, which is shown as the air blower in Figure 2.2. A J-type thermocouple is fitted at the intake of air to measure the intake air temperature. The flow rate of the air is measured and controlled using the blower tachometer with an accuracy of 5 l/min using the pulse width modulation of the blower fan.

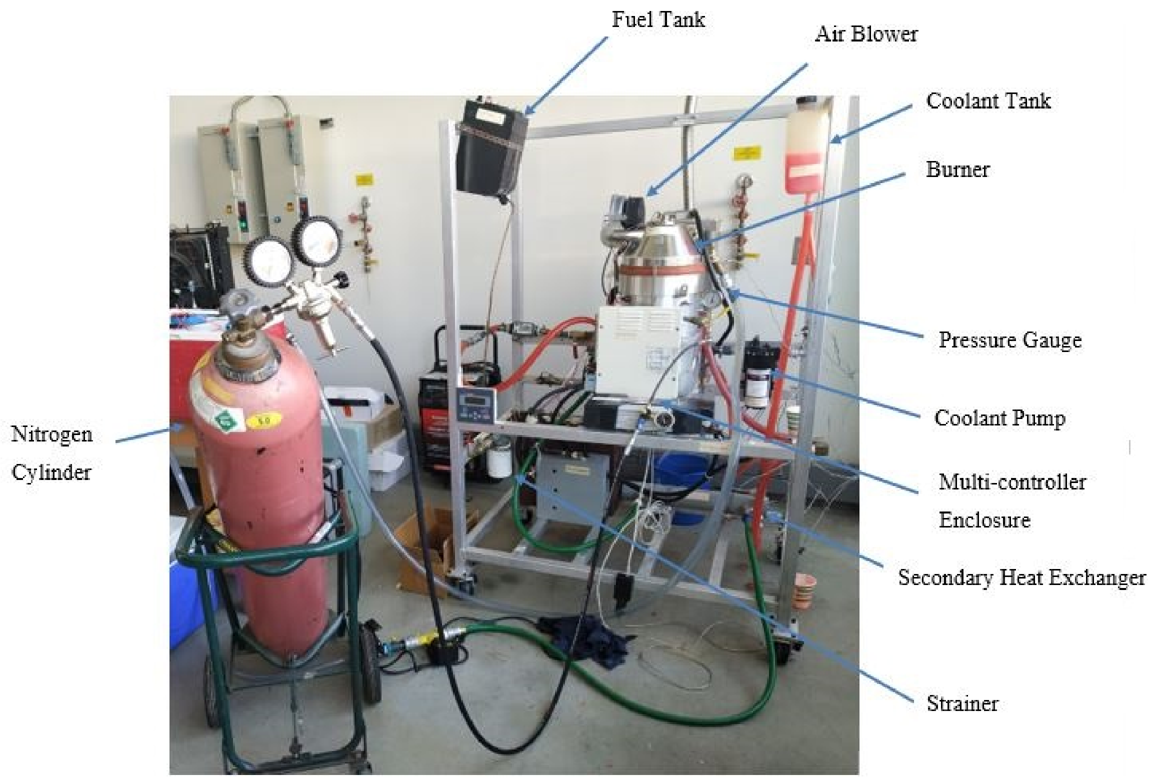

Figure 2.2 A Photograph of the Experimental Setup

2.2.2 Fuel Supply System

The fuel supply system consists of a fuel tank, isolation valves, filter, and 12V fuel pump. The fuel tank has a capacity of one gallon. The pump operates on a pulse width modulated signal from the controller in order to deliver the required amount of fuel into the evaporator. The fuel pump can supply up to 18 ml/min with 1 ml accuracy.

2.2.3 Coolant System

The coolant system consists of a primary coolant tank, along with a 12V DC pump, filter/strainer, and a 12V clamp element heater. The coolant used is a mixture of 50% glycol and 50% water. The coolant line is also fitted with mechanical valves and pressure relief valves. The secondary heat exchanger – a counter-flow heat exchanger – is also a part of the coolant system. This heat exchanger is used to cool down the coolant which is used as the cold side for the working of the Stirling engine. The secondary heat exchanger is placed below the alternator.

2.2.4 Exhaust System

The exhaust heat exchanger is a part of the exhaust system, which also consists of a condensate drain, exhaust tubing, and a draft fan. There are six J-type thermocouples fitted at critical locations to measure the respective temperatures. The main function of the exhaust system is treating the flue gases coming out of the combustion chamber. The condensate drain collects the water after it passes through the exhaust heat exchanger. The remaining flue gas is passed to the exhaust tubing, which goes to the hood.

2.2.5 Burner

The burner is one of the most important parts of the WhisperGen engine. The burner houses the combustion chamber, where the combustion of fuel and air takes place. The burner consists of a series of sheet metal shells welded concentrically. These shells act as a heat exchanger in that they help to preheat the incoming air. The burner sits above the Stirling engine, as shown in Figure 2.2. It is sealed with a high temperature ceramic sealant. An evaporator is fitted at the top of the burner, consisting of a fine mesh to filter the unburnt fuel and a glow plug to preheat the combustion chamber at the start of the engine's operation. The evaporator mixes the incoming air and the fuel vapor, which helps in their combustion.

2.2.6 Stirling Engine

The Stirling engine sits just below the burner, so the heat from the combustion chamber is used for its working. It consists of four cylinders arranged in the alpha configuration of the Stirling engine. Nitrogen is used as the working fluid in these cylinders, and is pressurized to around 2.8 MPa. A wobble yoke mechanism is connected to the four cylinders to convert their longitudinal motion into rotary motion.

2.2.7 Electrical System

The electrical system consists of an alternator, which uses the rotary motion produced by the wobble yoke mechanism, converting it into AC current. This three-phase AC current is converted into DC current with the help of rectifiers, and is then stored in a 12 V battery. All the commands are controlled by the microcontroller supplied with the engine, which can be controlled via a remote computer by using the Micromon software included in the engine.

2.3 Working and Experimental Setup

The WhisperGen Stirling engine incorporates a Stirling engine to generate the energy output. As a cogeneration system, the WhisperGen engine can produce an electricity output of 1 kW and a heat output of around 6.5 kW. The output from the Stirling engine is used to produce energy, which is converted into usable energy using an alternator before being stored in a 12 V battery. To better analyze the working of the engine, it has been divided into four main components: the burner, the Stirling engine, the exhaust heat exchanger, and the secondary heat exchanger.

2.3.1 Working of the Engine

The burner (or the continuous combustion chamber) lies above the Stirling engine, as shown in Figure 2.2. It is used to burn fuel (diesel) in the presence of air. The air is taken in by the blower at Point 1, and the fuel is injected into the chamber by a fuel pump. The fuel is then vaporized using an evaporator. The burner is made up of concentric shells near its top. The incoming air is preheated by the flue gases (exhaust gases) passing through these shells. The flue gases are then passed onto the exhaust heat exchangers, which cool them down before they exit the system through aluminum exhaust pipes attached to the exhaust heat exchanger.

Below the burner is the four-cylinder alpha configuration Stirling engine. The heat from the combustion chamber acts as the hot side for the Stirling engine. Coolant is passed through the bottom of the engine to remove heat from it. This side of the engine cools down and acts as the cold side, thus providing the temperature difference required for running the Stirling engine. The four pistons are connected to a wobble yoke mechanism underneath the Stirling engine. This mechanism is used to convert the movement of the pistons into rotary motion, which is used by the alternator to generate electricity. The arrangement of the mechanism is shown in Figure 2.3, where, 1, 2, 3, and 4 represent the four pistons.

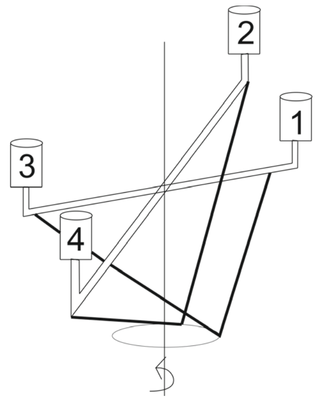

Figure 2.3 Wobble Yoke Mechanism

The coolant exiting the Stirling engine is cooled via a counterflow heat exchanger before it goes back into the coolant tank, completing the cycle. This is achieved by using cold water flowing in the opposite direction. After passing through the secondary heat exchanger, the laboratory water moves to the exhaust heat exchanger and extracts heat from the exhaust gases coming out from the burner. This helps in achieving the required temperature for the hot water. A water drain is placed below the exhaust heat exchanger to collect the condensed water from it.

The temperature to which the water must be heated is controlled by maintaining the exhaust temperature and the coolant temperature. The control system maintains the exhaust set point to control the hot water temperature. The system is designed to shut off once the engine reaches the exhaust set point, thus maintaining the temperature of the water. The system also monitors the temperature of the coolant throughout the working of the engine to better control the temperature. The engine also has several working modes to meet varying energy demands. All these activities are controlled by the microcontroller connected to the engine.

2.3.2 *Experimental Setup*

The WhisperGen engine was operated in the heat manage mode by setting the temperature to 52 ^{0}C, with the auto charge option turned off. All the required steps were performed before starting the engine, as instructed in the WhisperGen manual. The fuel pump had to be replaced, as it was not providing the required fuel flow rate.

The engine was run for a total time of 45 minutes. The first 20 minutes included the preheating and warming up stage. The engine was left to run for the next 25 minutes.

After the start command was initiated, the engine went through a pre-test procedure to check the functionality of the electronic components. After that was done, the combustion chamber was preheated by switching on the glow plug. The fuel pump was switched on by the microcontroller. The presence of a flame was monitored by the flame rod inside the burner, which was also displayed by the engine. In fact, the engine shuts off if the flame is not there.

The combustion process started in the burner, and the temperature in it started to build up until the required temperature was reached. At this point, the engine was cranked. The next stage is a crucial stage; after the cranking of the engine, the power output generated is monitored for a few seconds. If the required power output is not obtained after cranking, the engine starts the shutoff sequence by displaying the corresponding error code. After the cranking stage, the engine ran for 20 minutes. The shut off sequence was started once the command was received by the microcontroller. The fuel pump was turned off, and the air blower was operated at 100% to cool down the engine. The laboratory water kept on recirculating to cool down the engine. Various J-type and K-type thermocouples were used to log the temperature at the state points, as shown in Figure 2.4.

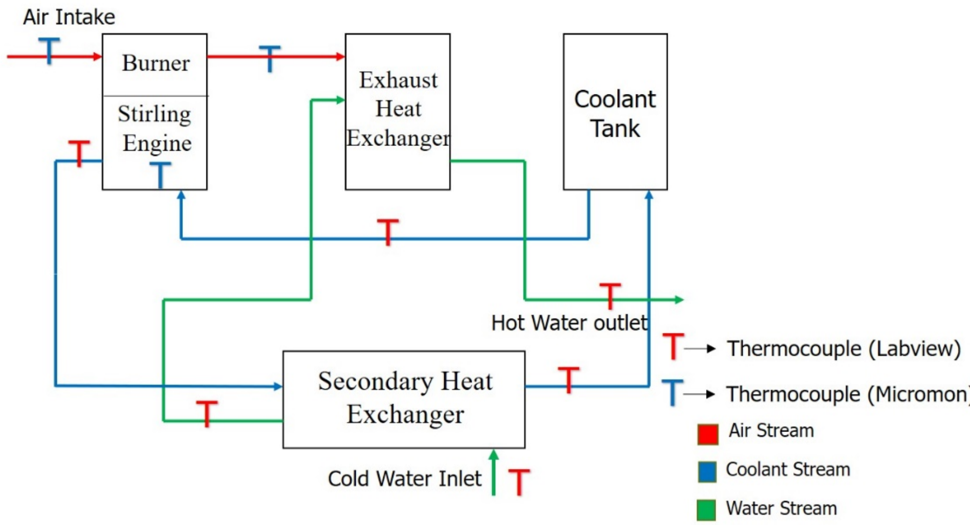

Figure 2.4 Thermocouple Location

In Figure 2.4, the blue 'T' represents the thermocouple wherein the data is logged by the Micromon software, while the red 'T' represents the thermocouples wherein the data is logged using LabVIEW. The experiment was conducted three times under identical conditions to get a better understanding of the experimental results.

CHAPTER 3

ENERGY AND EXERGY ANALYSES

The laws of thermodynamics govern the various processes occurring in nature and engineering systems. In this chapter, the different laws of thermodynamics are explained, and their implementation for the system under consideration is discussed.

3.1 First Law of Thermodynamics

The first law states that *energy can neither be created nor destroyed, but can be transformed from one form to another.* The significance of this statement is that, during a process, the total energy in the system is always conserved. Energy transferred from a system or into a system can be done through heat, work, and mass transfer. The change in energy of a system is equal to the energy entering the system minus that leaving the system,

$$d\dot{E}_{system} = \dot{E}_{in} - \dot{E}_{out} = [Q_{in} - Q_{out}] + [\dot{W}_{in} - \dot{W}_{out}] + [\dot{E}_{mass,in} - \dot{E}_{mass,out}] \quad (1)$$

where $\dot{Q}_{in}$ and $\dot{Q}_{out}$ are the rate of heat entering and leaving the system, respectively; $\dot{W}_{in}$ and $\dot{W}_{out}$ are the rate of work input and work output from the system, respectively; and $\dot{E}_{mass,in}$ and $\dot{E}_{mass,out}$ represent the energy associated with the rate of mass entering and rate of mass leaving the system, respectively. For any system or process, efficiency is the key factor. It not only helps to compare similar engines, but also indicates how the engine is performing. For the analysis in this paper, two different efficiencies have been defined: the engine's thermal efficiency and its overall efficiency. The thermal efficiency deals with the amount of heat energy that can be recovered from the engine. The thermal efficiency can be defined as

$$\eta_{th} = \frac{Q_{out}}{Q_{in}} \quad (2)$$

where $\dot{Q}_{out}$ and $\dot{Q}_{in}$ are the rate of heat energy leaving the system and rate of heat energy provided for the engine, respectively. The overall efficiency of an engine can be defined as the ratio of the net work output to the total input energy provided to the engine, that is,

$$\eta_{overall} = \frac{W_{net}}{Q_{in}} \quad (3)$$

Here, W_{net} is the net work output from the engine.

Let us consider the tube-and-shell secondary heat exchanger used in the system, as described in the previous chapter. Figure 3.1 is a magnified image of the secondary heat exchanger of the system. In the heat exchanger, the hot fluid (coolant) exchanges heat with the cold fluid (water) moving in the opposite direction. In Figure 3.1, $\dot{Q}_w$ is the rate of heat energy gained by the cold fluid ($\dot{Q}_{out}$), $\dot{Q}_{co}$ is the rate of heat energy lost from the hot fluid ($\dot{Q}_{in}$), and $\dot{Q}_L$ is the rate of heat loss from the heat exchanger.

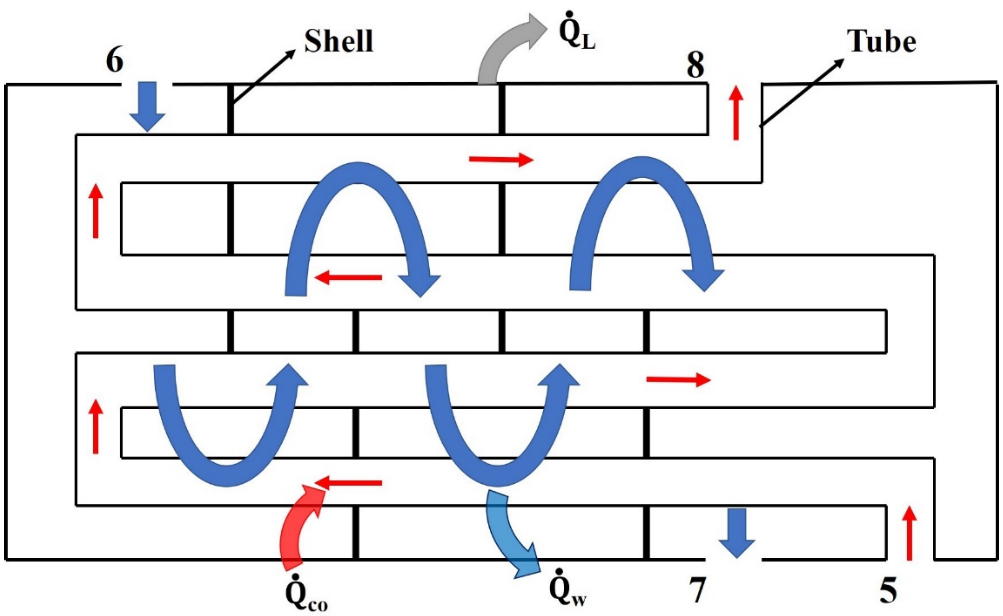

Figure 3.1 Shell-and-Tube-Heat Exchanger

The energy balance can be expressed as

$$Q_{co} - Q_{w} = Q_{L} \tag{4}$$

Let us consider the state points of Figure 3.1. Assume T_5, T_6, T_7, and T_8 to be 326 K, 303 K, 313 K, and 320 K, respectively. For the heat exchanger, the heat energy lost by the coolant,

$$Q_{co} = \dot{m}_{co} C_p (T_5 - T_8) \tag{5}$$

where $\dot{m}_{co}$ is the mass flow rate of coolant and C_p is the specific heat capacity of the coolant. The heat energy gained by water

$$Q_{w} = \dot{m}_{w} C_p (T_7 - T_6) \tag{6}$$

where, $\dot{m}_{w}$ is the mass flow rate of water and C_p is the specific heat capacity of water. For heat exchangers, the efficiency is usually defined as

$$\eta_{th} = \frac{Q_{co}}{Q_{max}} \tag{7}$$

where $\dot{Q}_{max}$ is the maximum heat transfer rate that can be achieved by the heat exchanger.

3.1.1 First Law Analysis of the System

Figure 3.2 shows a schematic for the WhisperGen Stirling engine. The temperatures of all the state points in Figure 3.2 are provided in Table 3.1. From the table, it can be found that the highest temperature recorded is at State Point 2, and, hence, this is a very crucial point for the operation of the engine.

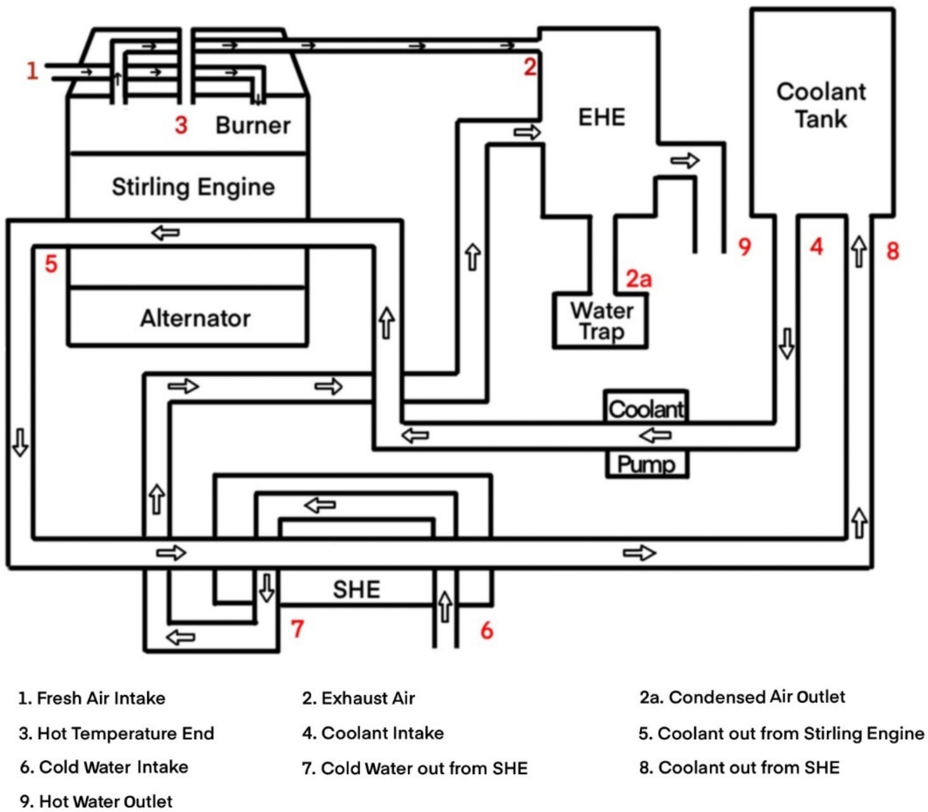

Figure 3.2 Schematic of WhisperGen Engine

Table 3.1 Temperature of State Points

State	Temperature (K)	Repeatability Range
Ambient Air (0)	298	-
Fresh Air Intake (1)	309	-
Exhaust Air (2)	666	664 – 668 K
Hot Temperature End (3)	630	627 – 633 K
Coolant Intake (4)	319	318 – 320 K
Coolant out from Stirling Engine (5)	326	324 – 328 K
Laboratory Water Intake (6)	303	301 – 305 K
Laboratory Water out from SHE (7)	313	311 – 315 K
Coolant out from SHE (8)	320	319 – 321 K
Hot Water Outlet (9)	315	313 – 317 K

Figure 3.3 shows the control volume for the first law analysis. Heat and mass interactions are considered as shown in Figure 3.3; $\dot{m}$, $\dot{Q}$, and $\dot{W}$ stand for mass flow rate, heat flow rate, and workflow, respectively, across the control volume. The subscripts a, co, f, h, i, l, o, and w represent air, coolant, fuel, heat, input, loss, output, and water, respectively. During the operation of the engine, it was found that the temperature around the burner increased by 3 ^{0}C. This contributes to the $\dot{Q}_l$ for the engine. The assumptions for the first law analysis are as follows:

1. Steady state operation;
2. No heat loss from the Stirling engine to the surroundings;

3. The temperature of the combustion chamber is equal to the hot side temperature;
4. There is no fouling in the heat exchangers;
5. Nitrogen operates as an ideal gas;
6. Changes in kinetic and potential energy are negligible.

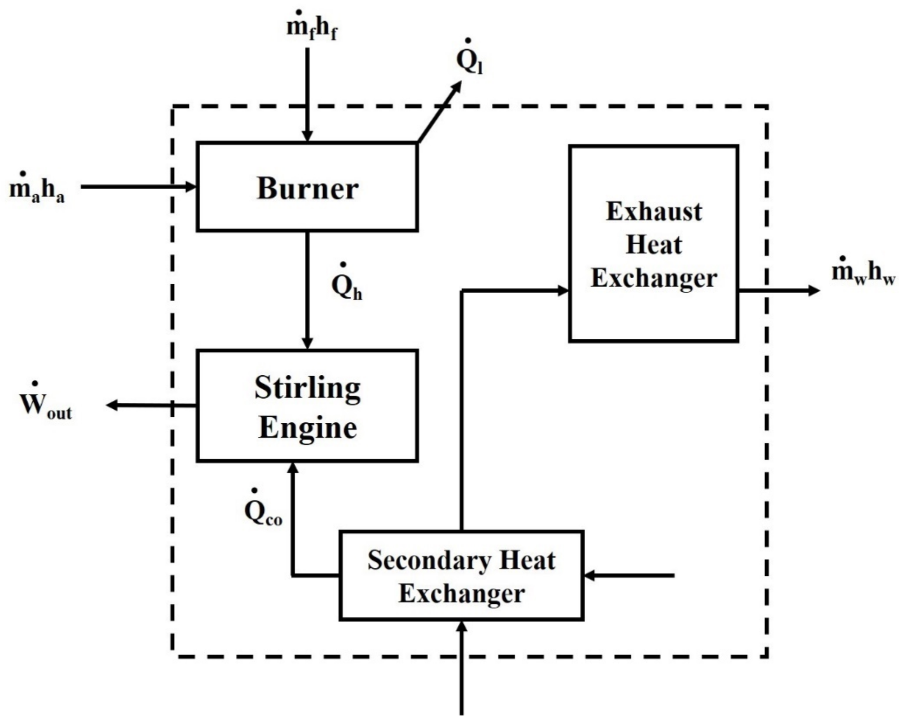

Figure 3.3 Control Volume for the WhisperGen Engine

According to the first law of thermodynamics, energy is conserved. For the control volume shown above, this can be mathematically expressed as

$$\dot{m}_a h_a + \dot{m}_f h_f - Q_l - \dot{m}_E h_E - \dot{m}_w h_w - \dot{W}_{out} = 0 \qquad (8)$$

where h is the specific enthalpy of the fluid and $\dot{W}_{out}$ is the power generated by the Stirling engine. The first law analysis can reveal the energy interaction during the process. The energy (or thermal efficiency) efficiency of the engine can be calculated as

$$\eta_{thermal} = \frac{Q_w}{Q_i} \quad (9)$$

where $\dot{Q}_w$ is the rate of heat energy gained by water and $\dot{Q}_i$ is the rate of heat energy supplied to the system.

The net-work (W_{net}) will include the electricity generated by the Stirling engine, along with the heat output. The total efficiency of the system can be calculated from

$$\eta_{total} = \frac{W_{net}}{Q_i} \quad (10)$$

3.2 Schmidt Analysis of the Stirling Engine

One of the primary methods to analyze a Stirling engine was proposed by Schmidt. This theory is based on the isothermal expansion and compression of the gas. The assumptions made in the Schmidt analysis are given below:

1. There is no pressure loss in the heat exchangers, and there are no internal pressure differences;
2. The expansion and compression processes take place isothermally;
3. The gas behaves like an ideal gas;
4. There is perfect regeneration;
5. The expansion space and compression space change according to a sine curve;
6. The expansion dead space maintains the expansion gas temperature, and the compression dead space maintains the compression gas temperature.

7. The regenerator temperature is an average of the expansion gas temperature and the compression gas temperature as expressed below.

The volume occupied by the gas changes as the process takes place, changing with respect to the crank angle (θ). The crank angle is 0 (θ = 0) when the piston is at the top dead center of the compression cylinder. For the WhisperGen model, a four-cylinder double-acting alpha Stirling engine is used. The expansion volume leads the compression volume by a phase angle of 90^0 (dθ = 90). The pressure and volume changes in the cylinder can be plotted to understand the behavior of the engine. The pressure can be calculated from

$$P = \frac{P_{mean}\sqrt{1\text{-}q^2}}{1 - q\cos(\theta\text{-}j)} \tag{11}$$

where θ is the crank angle , P_{mean} is the mean pressure in the engine during its operation which can be calculated from

$$P_{mean} = \frac{2\,m\,R\,T_C}{V_{SE}\sqrt{S^2\text{-}B^2}} \tag{12}$$

where, m is the total mass of the fluid inside the engine, R is the universal gas constant, T_c is the temperature at the compression side and V_{SE} is the swept volume of expansion side. q is defined as,

$$q = \frac{B}{S} \tag{13}$$

where,

$$B = \sqrt{t^2 + 2\ t\,v\ \cos d\theta + v^2} \tag{14}$$

where, t is the temperature ratio expressed as,

$$t = \frac{T_C}{T_h} \qquad (15)$$

where, T_h is the temperature at the expansion end, and v is the swept volume ratio expressed as,

$$v = \frac{V_{SC}}{V_{SE}} \qquad (16)$$

wherein, V_{SC} is the swept volume of compression side.

$$S = t + 2\,t\,X_{DE} + \frac{4\,t\,X_R}{1 + t} + v + 2\,X_{DC} \qquad (17)$$

where, X_{DE} is the dead volume ratio which can be calculated using Equation (18). X_R is the ratio of the regenerator volume to the swept volume of the expansion side and X_{DC} is the ratio of the dead volume of compression side to the swept volume of expansion side as expressed in Equation (19) and Equation (20), respectively.

$$X_{DE} = \frac{V_{DE}}{V_{SE}} \qquad (18)$$

where, V_{DE} is the dead volume of expansion space.

$$X_R = \frac{V_R}{V_{SE}} \qquad (19)$$

$$X_{DC} = \frac{V_{DC}}{V_{SE}} \qquad (20)$$

In Equation (11), ‘j’ is given by,

$$j = tan^{-1} \frac{v \sin d\theta}{t + \cos d\theta} \qquad (21)$$

The volume at each phase angle can be determined from

$$V = V_E + V_C + V_R \qquad (22)$$

where V_E, V_C, and V_R are the expansion, compression, and regenerator volumes at a particular phase angle, respectively.

$$V_E = \frac{V_{SE}}{2}\ (1+\cos\theta) + V_{DE} \tag{23}$$

$$V_C = \frac{V_{SC}}{2}\ (1+\cos\ (\theta+d\theta)) + V_{DC} \tag{24}$$

The values for the engine are given in Table 3.2. Figure 3.4 shows the P-V diagram for the Stirling engine. The graph is similar to the P-V diagram discussed in Chapter 1.

Table 3.2 Stirling Engine Specifications

Description	**Value**
LHV of Diesel	4279 kJ/kg
R (Gas Constant)	296.8 J/kg · K
V_{SE}	3.414E-05 m^3
V_{SC}	2.813E-05 m^3
V_R	1.745E-05 m^3
V_{DH}	9.180E-06 m^3
V_{DC}	8.269E-06 m^3
m_N	3.750E-04 kg
ε_R	0.63

The P-V diagram of the Stirling engine has been plotted based on the Schmidt analysis in Figure 3.4. The highest mean pressure for the engine was around 5.7 MPa after the constant volume heat addition process.

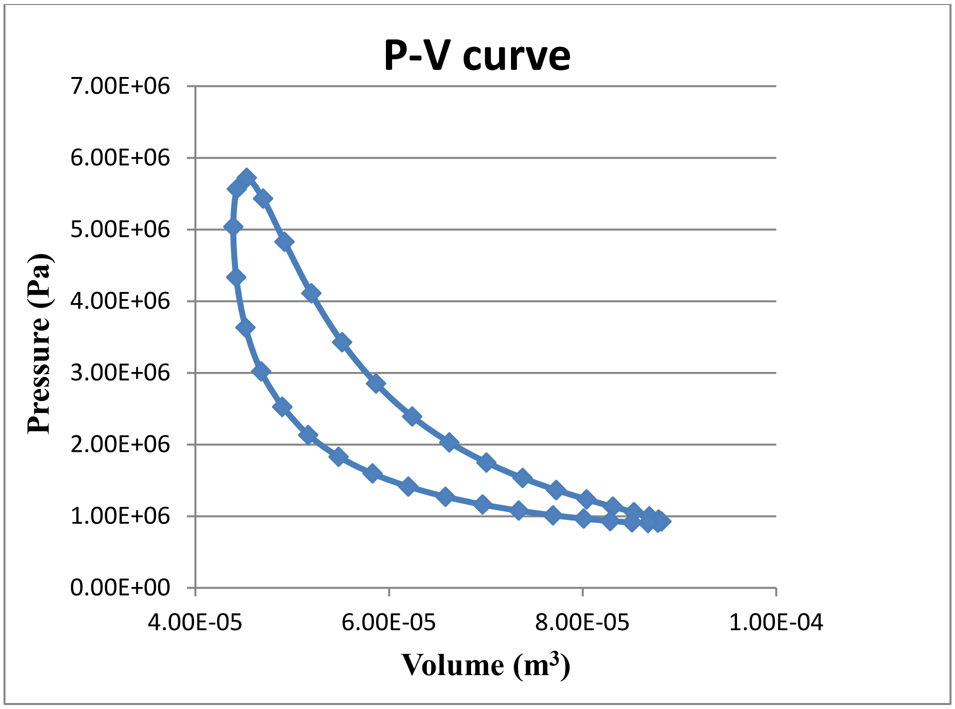

Figure 3.4 P-V Diagram for Stirling Engine

The indicated power for the engine can be calculated by,

$$\dot{W} = \frac{W_i \, N}{60} \tag{25}$$

where,

$$W_i = W_E - W_C \tag{26}$$

W_E and W_C are the expansion energy and compression energy, respectively. N is the speed of the engine in rpm, while W_i is the indicated energy of the engine.

$$W_E = \frac{P_{max}\, V_{SE}\, \pi\, q \;\; \sin(j)\, \sqrt{1-q}}{1 + \sqrt{1-q^2}\;\, \sqrt{1+q}} \tag{27}$$

where, P_{max} is the maximum pressure in the engine which can be calculated as follows,

$$P_{max} = P_{mean}\sqrt{\frac{1+q}{1-q}} \quad (28)$$

$$W_C = \frac{P_{max}\ V_{SE}\ \pi\ q\ t\ \sin(j)\ \sqrt{1-q}}{1+\sqrt{1-q^2}\sqrt{1+q}} \quad (29)$$

By solving Equations (26) to (29), the indicated power of the Stirling engine was found to be 1.35 kW.

3.3 Second Law of Thermodynamics

The maximum useful work that can be obtained from a system at a given state in a specified environment (its surroundings) is termed as exergy, or available energy. The exergy of a system is at its maximum when it executes a reversible process. In this case, there are no irreversibilities in it, and the system can obtain the maximum work output.

A system cannot produce any work when it is in a dead state. A system is said to be in a dead state when it is in thermodynamic equilibrium with its surroundings. At the dead state, the temperature and pressure of the system is equal to the surroundings, it has no kinetic or potential energy, and it is also in chemical equilibrium with its surroundings. In a dead state, the system is no longer capable of producing any work with respect to the surroundings. When a system undergoes a process, exergy in the system is always destroyed. We can say that the irreversibilities in a system lead to exergy destruction, thereby proving to be a hindrance in the system achieving its maximum potential. Hence, it is very important that we identify these irreversibilities. This can be done by performing an exergy analysis of the system.

3.4 Exergy Analysis

Exergy analysis is an effective tool for understanding how much the system can be improved by reducing its inefficiencies [28], [29]. To illustrate the difference between the first law efficiency and the second law efficiency, let us consider Figure 3.5. We have two similar heat engines (Engine A and Engine B), both with the same thermal efficiency of 40%. The engines operate at different temperatures. In the case of first law analysis, both the engines appear to operate the same. If we consider the reversible conditions, we can calculate the Carnot efficiency of the engines by,

$$\eta_{Carnot} = 1 - \frac{T_L}{T_H} \tag{30}$$

where T_L and T_H are the cold temperature and hot temperatures of the engine, respectively, in Kelvin. The exergy efficiency can be calculated using,

$$\eta_{\varepsilon} = \frac{\eta_{th}}{\eta_{Carnot}} \tag{31}$$

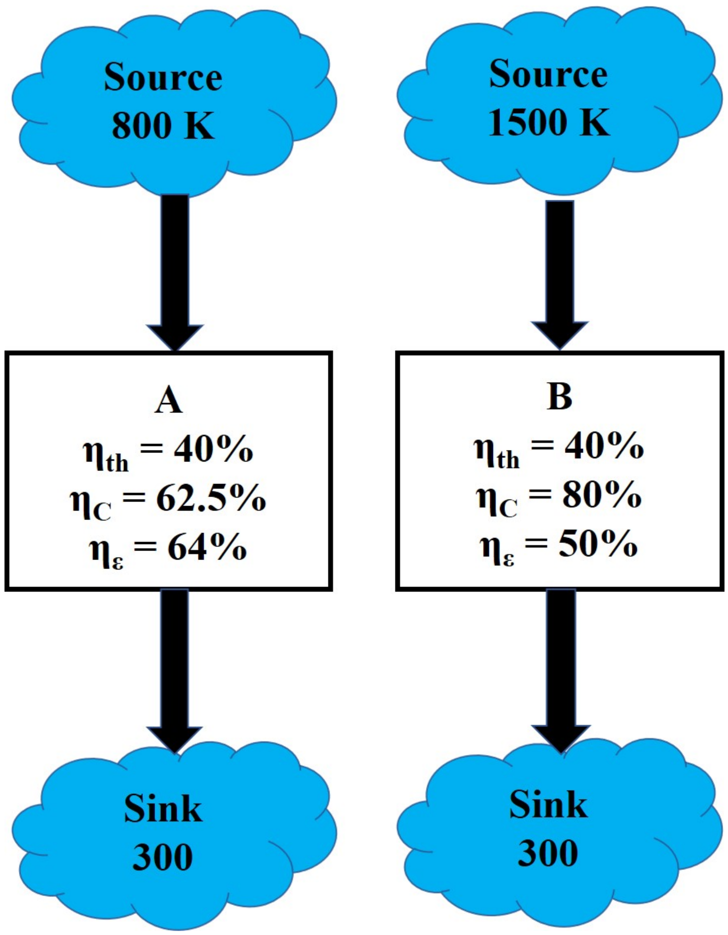

Figure 3.5 Heat Engines

We can see from Figure 3.5 that the Carnot efficiency of Engine B is higher than Engine A. This means that Engine B has a higher potential. This cannot be understood by the first law. From the exergy efficiency, we can see that the exergy efficiency of Engine B is lower than the exergy efficiency of Engine A. Thus, exergy helps to compare the engines with their actual potential. From the exergy efficiency, it can be said that Engine A has the potential to produce better quality output than Engine B. In the following section, we will discuss the conventional exergy analysis and the advanced exergy analysis by using the heat exchanger from the system as an example.

3.4.1 Conventional Exergy Analysis

For the tube-and-shell heat exchanger, the exergy interaction is as shown in Figure 3.6. The heat exchanger's exergy interaction is due to the flow of the fluid. Assume there is no work or heat interaction with the surroundings, the rate of exergy input (ε_{in}) is provided by the hot fluid, as shown,

$$\acute{\varepsilon}_{in} = \acute{\varepsilon}_5 - \acute{\varepsilon}_8 \tag{32}$$

The exergy gained by the cold fluid can be calculated by,

$$\acute{\varepsilon}_{out} = \acute{\varepsilon}_7 - \acute{\varepsilon}_6 \tag{33}$$

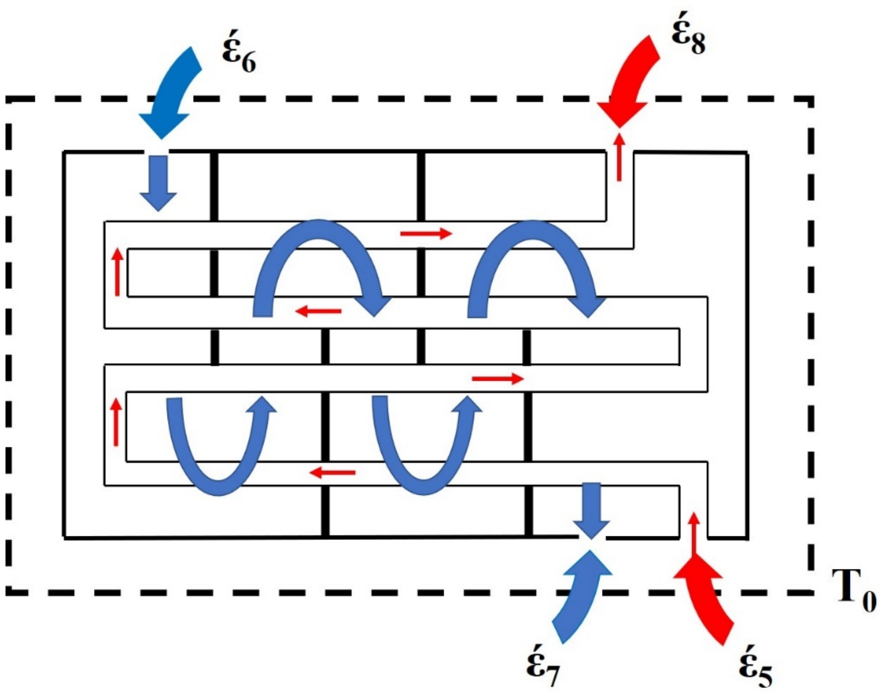

Figure 3.6 Exergy Interaction for a Shell-and-Tube Heat Exchanger

For State Point 5, the flow exergy can be given by,

$$\varepsilon_5 = (h_5 - h_0) - T_0(s_5 - s_0) \tag{34}$$

where T_0 is the reference temperature for the system. h and s are the specific enthalpy and specific entropy of the fluid at the given temperature. Similarly, the flow exergy of the other points can also be calculated.

In any practical process, exergy is always destroyed. In simple terms, exergy destruction is the difference between the exergy entering the system and the exergy leaving the system. Exergy destruction is always a positive value. At steady state condition, exergy destruction rate in a system can be expressed as,

$$\acute{\varepsilon}_d = \acute{\varepsilon}_{heat} - \acute{\varepsilon}_{work} - d\acute{\varepsilon}_{system} \tag{35}$$

$$\acute{\varepsilon}_d = \sum \left(1 - \frac{T_0}{T}\right) Q - (\dot{W}_{out} - \dot{W}_{in}) - (\acute{\varepsilon}_{flow2} - \acute{\varepsilon}_{flow1}) \tag{36}$$

where, $\acute{\varepsilon}_d$ represents the exergy destruction rate, $\acute{\varepsilon}_{heat}$ represents the exergy flow rate due to heat transfer to or from the surroundings, $\acute{\varepsilon}_{work}$ represents the exergy due to work and $d\acute{\varepsilon}_{system}$ is the change in exergy in the system. For the heat exchanger, the exergy destruction is the difference between the rate of exergy entering and rate of exergy leaving the system while the environment stays in the reference state. The exergy interaction for a system is shown in Figure 3.7. In the figure '1' is the initial state of the system, '2' is the final state of the system and $\dot{Q}$ represents the rate of heat transfer.

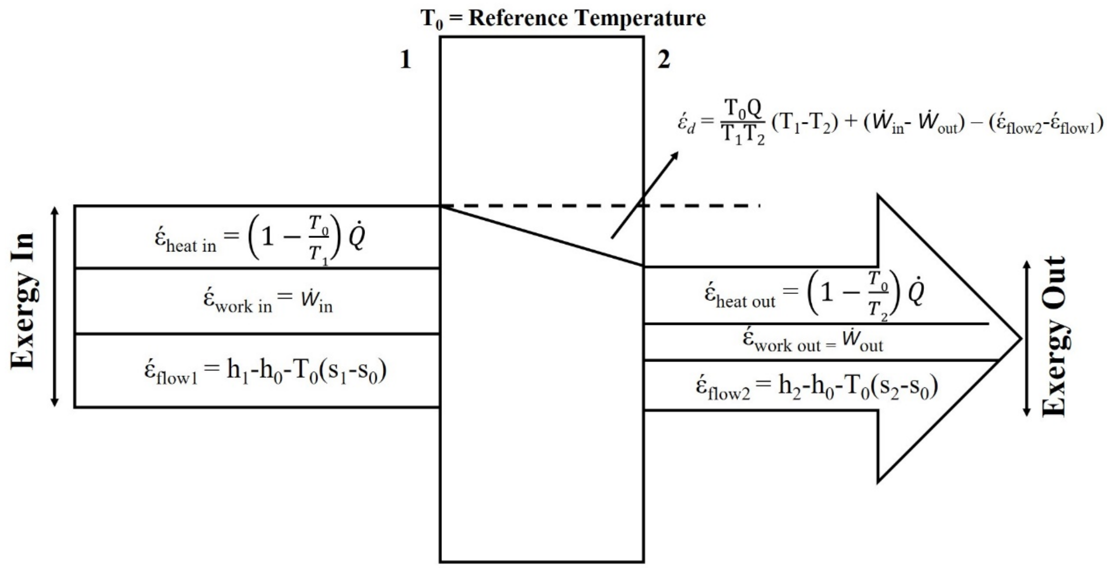

Figure 3.7 Exergy Interaction for a System

For the heat exchanger, we have

$$\acute{\varepsilon}_d = \acute{\varepsilon}_{in} - \acute{\varepsilon}_{out} \qquad (37)$$

where, $\acute{\varepsilon}_{in}$ is the rate of exergy entering the heat exchanger and $\acute{\varepsilon}_{out}$ is the rate of exergy leaving the heat exchanger. The second law efficiency is also known as the exergy efficiency. There are different definitions of exergy efficiency that can be used to analyze the system. For the purpose of this paper, the exergy efficiency is defined as the ratio of exergy recovered to exergy expended, that is,

$$\eta_\varepsilon = 1 - \frac{\acute{\varepsilon}_{destroyed}}{\acute{\varepsilon}_{expended}} \qquad (38)$$

For the heat exchanger, we have

$$\eta_\varepsilon = 1 - \frac{\acute{\varepsilon}_d}{\acute{\varepsilon}_{in}} \qquad (39)$$

3.4.2 Conventional Exergy Analysis of the Engine

For the exergy analysis, each component will be analyzed separately. The reference temperature, pressure, and air composition for the analysis are T_0 = 298K, P_0 =1atm, and air (N_2 [75.67%], O_2 [20.32%], H_2O [3.12%], and CO_2 [0.03%]), respectively, where '0' represents the reference state. For the exergy analysis, the engine was divided into the four main components. The different components are the burner or the combustion chamber, the Stirling engine, the exhaust heat exchanger, and the secondary heat exchanger. There are two exergies associated with the combustion chamber: flow exergy, due to the transfer of air and fuel into the control volume, and chemical exergy, associated with the combustion of diesel fuel inside the chamber. The chemical exergy associated with the exhaust gases released after the combustion process can be calculated using

$$\acute{\varepsilon}_{ch} = \sum y_i \alpha_i + RT_a \sum y_i ln y_i \tag{40}$$

where R stands for the universal gas constant, T_a is the ambient air temperature, α_i is the standard chemical exergy of a pure chemical compound i, y_i is the molar fraction of the components in the exhaust gases, and y_i^e stands for the molar fraction of the components of the exhaust gases in the reference temperature [30]. The fuel exergy is given by,

$$\acute{\varepsilon}_{fuel} = \phi \ LHV \tag{41}$$

where ϕ is the chemical exergy factor and LHV is the lower heating value of diesel. The chemical exergy factor,

$$\phi = 1.0401 + 0.1728 \frac{h}{c} + 0.0432 \frac{o}{c} + 0.2169 \frac{\alpha}{c} (1 - 2.0628 \frac{h}{c}) \tag{42}$$

where, c, h, o, and α are the mass fractions of carbon, hydrogen, oxygen, and sulphur, respectively [30]. Table 3.3 gives the exergy balance for all four components as shown in Figure 3.2.

The conventional exergy analysis helps to understand the amount of exergy destroyed in the system. It also helps to calculate the exergy efficiency, providing insight into the actual performance of the system. The analysis, however, is unable to identify the source of the exergy destruction or the amount of exergy destruction that can be avoided. It also does not give an idea about how much the system can be improved. This can be understood by performing an advanced exergy analysis, which is discussed in the following section.

Table 3.3 Exergy Balance for the Components

Component	Exergy Balance	Exergy Efficiency
Burner (Combustion Chamber)	$\acute{\varepsilon}_d = \acute{\varepsilon}_1 + \acute{\varepsilon}_Q + \acute{\varepsilon}_{flow} + \acute{\varepsilon}_{ch2} - \acute{\varepsilon}_2$	$\varepsilon_{exergy} = \frac{\acute{\varepsilon}_{product}}{\acute{\varepsilon}_{fuel}}$
Exhaust Heat Exchanger	$\acute{\varepsilon}_d = (\acute{\varepsilon}_9 - \acute{\varepsilon}_7) - (\acute{\varepsilon}_2 - \acute{\varepsilon}_{2a})$	$\varepsilon_{exergy} = \frac{(\acute{\varepsilon}_2 - \acute{\varepsilon}_{2a})}{(\acute{\varepsilon}_9 - \acute{\varepsilon}_7)}$
Stirling Engine	$\acute{\varepsilon}_d = (\acute{\varepsilon}_{3n} + \acute{\varepsilon}_{4n} + \acute{\varepsilon}_{4c})$ - $\acute{\varepsilon}_W$	$\varepsilon_{exergy} = \frac{\acute{\varepsilon}_W}{(\acute{\varepsilon}_{3n} + \acute{\varepsilon}_{4n} + \acute{\varepsilon}_{4c})}$
Secondary Heat Exchanger	$\acute{\varepsilon}_d = (\acute{\varepsilon}_5 - \acute{\varepsilon}_8) - (\acute{\varepsilon}_7 - \acute{\varepsilon}_6)$	$\varepsilon_{exergy} = \frac{\acute{\varepsilon}_7 - \acute{\varepsilon}_6}{\acute{\varepsilon}_5 - \acute{\varepsilon}_8}$

3.4.3 Avoidable and Unavoidable Exergy Destruction Analysis

To identify how much of the exergy destruction in the system can be avoided, we perform an advanced exergy analysis. For the advanced exergy analysis, the splitting of the exergy destruction is performed. In this splitting, the part of exergy destruction that can be avoided is deemed to be avoidable exergy destruction and the part that cannot be avoided is called unavoidable exergy destruction. This can be represented as,

$$\acute{\varepsilon}_{d,} = \acute{\varepsilon}_d^{AV} + \acute{\varepsilon}_d^{UA} \qquad (43)$$

$\acute{\varepsilon}_d$ is the total exergy destruction in the heat exchanger, $\acute{\varepsilon}_d^{AV}$ is the avoidable exergy destruction, and $\acute{\varepsilon}_d^{UA}$ is the unavoidable exergy destruction in the shell and tube heat exchanger [31].This has been illustrated in Figure 3.8. The figure shows that from the total exergy provided by the coolant, an amount of exergy is destructed during the process. The remaining exergy is gained by water. The exergy that is destroyed during the process can be divided into two part; avoidable exergy destruction in the heat exchanger and the unavoidable exergy destruction in the heat exchanger.

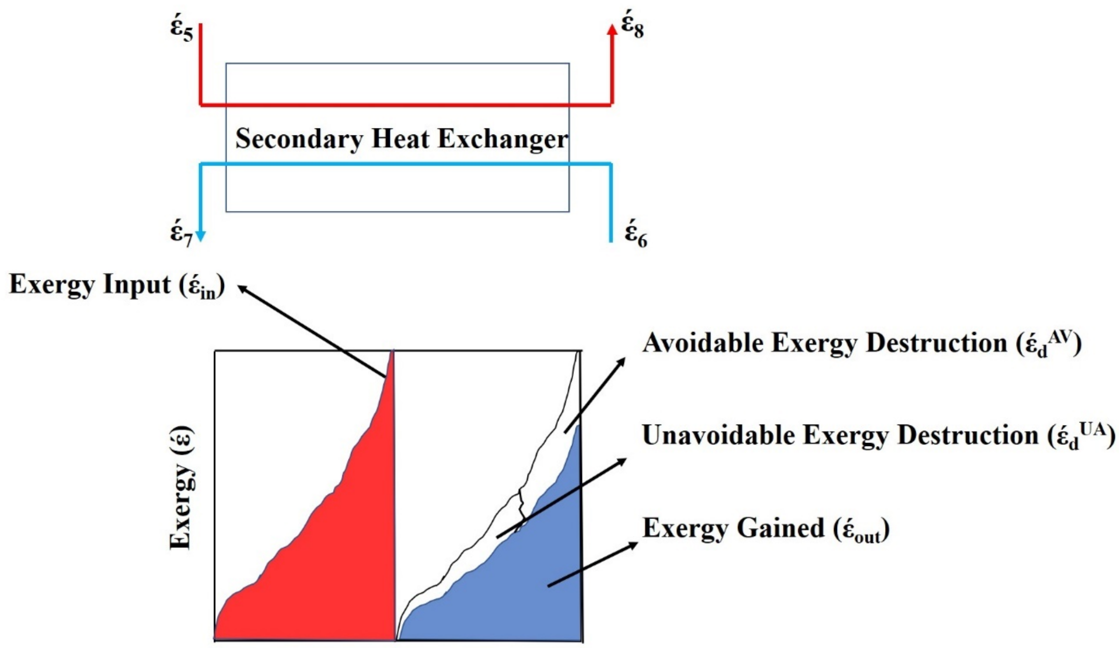

Figure 3.8 Exergy Destruction in a Heat Exchanger

The unavoidable exergy destruction is calculated by assuming the most reasonable operating conditions. To calculate the unavoidable exergy destruction for the heat exchanger, we consider that there is at least 10K of temperature difference between the fluid entering and leaving the system. In this case, the condition can be expressed as,

$$T_7 - T_6 \geq 10 \tag{44}$$

To calculate the unavoidable exergy destruction, we make use of the fuel-product rule [31]. For the fuel-product rule, the fuel term stands for the raw materials, primary energy required to make the product while the term product is defined with accordance with the aim of the purchase and what is expected from the component. Using the fuel-product rule, we can calculate the unavoidable exergy part by using

$$\acute{\varepsilon}_d^{UA} = \left(\frac{\acute{\varepsilon}_d}{\acute{\varepsilon}_p}\right)^{UA} \acute{\varepsilon}_p \tag{45}$$

where $(\acute{\varepsilon}_d/\acute{\varepsilon}_p)^{UA}$ is the ratio of exergy destruction to exergy of the product for the unavoidable condition. The term $\acute{\varepsilon}_p$ is the product exergy under the real operating conditions for the heat exchanger. The avoidable exergy destruction represents the irreversibilities that can be eliminated in the heat exchanger. The maximum exergy efficiency of the heat exchanger can be calculated from

$$\eta_{\varepsilon\, max} = \frac{\acute{\varepsilon}_p}{\acute{\varepsilon}_f} \tag{46}$$

where, $\acute{\varepsilon}_p$ is the product exergy and $\acute{\varepsilon}_f$ is the fuel exergy. The unavoidable part in the exergy destruction represents the part that cannot be eliminated even with the technological advancements available today. The advanced analysis is performed by analyzing each component separately as if the component is removed from the system. The conditions or the assumptions for the advanced analysis is made considering future enhancements that can be made for the component. For this purpose, decision makers must understand the working of the entire system and rely on conditions that can improve the system. In general, it can be said that the unavoidable conditions that are considered are better than the real working conditions but are not the ideal theoretical working conditions for the component.

The conditions for each component for calculating unavoidable exergy destruction are shown in Table 3.4. For a combustion chamber, Tsatsaronis [31] had suggested considering the highest possible temperature for the products and the reactants in the combustion chamber. In this case, we consider complete combustion of the fuel such that the required temperature is reached. The air entering the chamber is preheated to 50^0C raising the temperature to 323K. A reasonable temperature of 1000K that can be sustained by the ceramic material used in the burner is assumed. For heat exchangers, Vuckovic et al [32] and Tsatsaronis [31] in their studies suggested assuming a minimum temperature difference of 10K exists between the fluids for the unavoidable conditions. Similarly, for both the heat exchangers we consider a minimum temperature difference of 10K between the fluid entering and leaving the system. For the Stirling engine we considered no heat losses from the burner thus helping in maintaining the same temperature for the hot side of the engine. Also, we consider the maximum power output that can be practically achieved by the Stirling engine of 850W.

Table 3.4 Conditions for Unavoidable Exergy Destruction

Component	Ideal Conditions
Burner (Combustion Chamber)	1. Complete combustion of diesel fuel 2. $T_1 = 323K$ 3. $T_2 = 1000K$
Exhaust Heat Exchanger	1. $T_{2a} = 625K$ 2. $T_9 = 323K$
Stirling Engine	1. $T_3 = 660K$ 2. $W_{out} = 850W$
Secondary Heat Exchanger	1. $T_7 - T_6 \geq 10K$ [32]

CHAPTER 4

RESULTS AND DISCUSSIONS

This chapter describes the major results of the analysis and the conclusions that can be formed from them. The chapter presents the results of the first law analysis and the second law analysis. It also provides the results from the advanced exergy analysis.

4.1 First Law Analysis

Table 4.1 provides the results of the analysis, including the thermal and exergy efficiency. The calculated thermal efficiency of 67% was around 4% lower than what was recorded by previous experiments. A direct comparison of these two studies is not possible, as the fuel used by Farra et al [25] and Conroy et al [33] was natural gas. An earlier experiment performed on the same WhisperGen engine had a thermal efficiency of 73% [34]. This difference was investigated, and it was found that the temperature difference between the hot side and the cold side was greater by 50 ^{0}C, which resulted in higher power output and thus the higher efficiency. The Carnot efficiency of the Stirling engine was calculated to be 53%, while the thermal efficiency of the Stirling engine was found to be 29%.

Table 4.1 WhisperGen Engine Efficiencies

WhisperGen Stirling Engine	Power Output (W)	Thermal Output (kW)	Thermal Efficiency (%)	Total Efficiency (%)	Exergy Efficiency (%)
	676	5.54	67	75	30

It can be found from Table 4.1 that there is a big difference between the exergy efficiency and thermal efficiency of the WhisperGen system. The thermal efficiency was lower than it has been found by other researchers. This can be attributed to the low power output from the engine, which was due to the low temperature difference between the hot side and cold side. The energy efficiency is more than twice the exergy efficiency of the engine. The low exergy efficiency indicates that the engine is far away from being ideal and can be improved.

4.2 Exergy Analysis

Table 4.2 shows the exergy destruction and the exergy efficiency of the different components. The combustion chamber had the highest exergy destruction. This is due to the irreversible combustion process in the combustion chamber. The combustion chamber is the least efficient component in terms of exergy with just 24% exergy efficiency. The exhaust heat exchanger had the least exergy destruction. The maximum exergy efficiency was 45%, which was for the exhaust heat exchanger. The two heat exchangers had similar exergy efficiencies of 39% and 45%.

Table 4.2 Exergy Destructions

Components	Exergy Destruction (W)	Exergy Efficiency (%)
Burner	6839	24
Stirling Engine	982.2	41
Secondary Heat Exchanger	240.7	39
Exhaust Heat Exchanger	130.1	45

The avoidable and unavoidable exergy destruction is tabulated in Table 4.3. From the table, it is evident that, for the combustion chamber, which accounts for the most exergy destruction, more than 60% of it is avoidable, meaning that technological advancements should be able to make the burner more efficient. One of the improvements that can be considered for the burner is the use of better insulation materials to reduce heat losses and priority should also be given to improve the combustion process. It should also be noted that the exhaust heat exchanger has the least unavoidable exergy destruction. Additionally, the exhaust heat exchanger is the most exergy efficient, but its efficiency can be increased by more than 50% by making the heat transfer between the fluid more efficient. The heat losses from the heat exchangers during the process might be the reason for the lower efficiency. An interesting result from the analysis is that, although the efficiencies of the secondary heat exchanger and exhaust heat exchanger during the conventional analysis did not differ much, the maximum attainable efficiency of the exhaust heat exchanger is much higher. This is because of the higher unavoidable exergy destruction in the secondary heat exchanger.

Table 4.3 Avoidable and Unavoidable Exergy Destructions

Components	Unavoidable Exergy Destruction ($\acute{\varepsilon}_d^{UA}$) (W)	Avoidable Exergy Destruction ($\acute{\varepsilon}_d^{AV}$) (W)	Maximum Attainable Exergy Efficiency (%)
Burner	2142	4696	55
Stirling Engine	451	531	68
Secondary Heat Exchanger	164	76	49
Exhaust Heat Exchanger	35	95	71

The Stirling engine, which is responsible for the electricity generation, also had a difference in the maximum possible efficiency and the efficiency that was achieved during the operation of the engine. It was noted that the power output of the engine was much lower than the rated power output, which had caused the reduction in the efficiency. If the temperature difference between the hot side and cold side of the engine is higher, then the engine will be much more efficient.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

A thermodynamic analysis of a 1 kW WhisperGen Stirling engine system was conducted. The first and second laws of thermodynamics were used to perform the analysis. The first law efficiency of the engine was around 67%. The overall efficiency, which included the thermal and power output from the engine, was found to be 75%. The exergy efficiency of the engine was 30%. The combustion chamber had the highest amount of exergy destruction rate, contributing 84% of the total exergy destruction in the engine. While the exhaust heat exchanger had the lowest amount of exergy destruction rate. Due to the lower exergy destruction rate, the exhaust heat exchanger had the highest exergy efficiency of 45% among the four components. There is a need to improve the combustion chamber, as most of the exergy destruction occurred in it.

Of the total exergy destruction taking place in the combustion chamber, more than 60% can be avoided with technological improvements. As the combustion chamber contributes 84% of the total exergy destruction in the engine, improving it would result in an overall increase in the exergy efficiency. There is just a 6% difference in the maximum exergy efficiency that can be attained and the exergy efficiency under real conditions of the secondary heat exchanger. The secondary heat exchanger does not contribute a lot towards the exergy destruction and can be the last component that needs attention. This analysis of the WhisperGen Stirling engine has helped to identify the components contributing towards the irreversibly in the engine, thus providing a path towards improvements in the future. If the avoidable exergy destructions can be avoided in the engine, the exergy efficiency of the engine can be improved by at least 15%. This increase in the efficiency would in turn help in much more efficient performance of the engine. The scope of this study was limited to the avoidable and unavoidable exergy destruction analysis. To identify the exact locations of the irreversibility in each component and to understand how a component affects the performance of the other, an endogenous and exogenous exergy analysis of the engine is suggested.

REFERENCES

[1] "World Energy Balances – Analysis – IEA," 2018. Accessed on: April 2020. [Online]. Available: https://www.iea.org.reports/world-energy-balances-overview.

[2] "Renewable Energy Statistics 2019", *International Renewable Energy Agency* (IRENA) 2019. Accessedon: June2020. [Online] Available:https://www.irena .org/publications/2019/Jul/Renewable-energy-statistics-2019.

[3] H. Ritchie and M. Roser, "Access to Energy," *Our World in Data,* 2019. Accessed on June 2020. [Online]. Available: https://ourworldindata.org/energy-access.

[4] D. Akinyele, L. Olatomiva, D. E. Ighravwe, M. O. Babatunde, C. Monyei, and D. Aikhuele, "Optimal planning and electricity sharing strategy of hybrid energy system for remote communities in Nigeria," *Scientific African*, Vol 10, 2020.

[5] V. Balaji and H. Gurgenci, "Search for optimum renewable mix for Australian off-grid power generation," *Energy*. Volume 175, pp. 1234-1245, May 2019.

[6] J Knowles, "Overview of small and micro combined heat and power (CHP) systems," *Small and Micro Combined Heat and Power (CHP) Systems*, pp. 3-16, 2011.

[7] P. Breeze, "Combined Heat and Power*," Power Generation Technologies,* pp. 111-127, 2014.

[8] R. O. Lamidi, L. Jiang, Y. Wang, P. B. Pathare, and A Roskilly, "Techno-economic analysis of a biogas driven poly-generation system for postharvest loss reduction in a Sub-Saharan African rural community." *Energy Conversion and Management*, vol. 196, pp. 591-604, 2019.

[9] P. Atănăsoae, "Technical and economic assessment of micro-cogeneration systems for residential applications." *Sustainability*, vol 12, no. 3, p. 1074, 2020.

[10] O. Mahian, M. R. Mirzaie, A. Kasaeian, and S. H. Mousavi, "Exergy analysis in combined heat and power systems: A review," *Energy Conversion and Management*, vol. 226, 2020.

[11] J. Harrison and E. On, "Stirling engine systems for small and micro combined heat and power (CHP) applications," *Small and Micro Combined Heat and Power (CHP) Systems*, pp. 179-205, April 2011.

[12] D. G. Thombarse, "Stirling Engine: Micro-CHP System for Residential Application," *Encyclopedia of Materials: Science and Technology*, pp. 1-8, December 2008.

[13] F. E. Heffner, "Highlights From 6500 Hr of Stirling Engine Operation." *SAE Transactions*, pp. 33-54. 1965.

[14] B. Kongtragool and S. Wongwises, "A review of solar-powered Stirling engines and low temperature differential Stirling engines," *Renewable and Sustainable Reviews* vol. 7, pp. 131-154, 2003.

[15] N. Parlak, A. Wagner, M. Elsner, and H. S. Sohyan, "Thermodynamic analysis of a gamma type Stirling engine in non-ideal adiabatic conditions," *Renewable Energy* vol. 34, pp. 266-273, 2009.

[16] G. Walker, *Stirling Engines*, Oxford: Clarendon Press, 1980.

[17] T. Finkelstein and A. J. Organ, *Air Engines: the history, science and reality of the perfect engine*, ASME Press, 2001.

[18] C. M. Hargreaves, *The Philips Stirling Engine*, Amsterdam: Elsevier Science Publishers, 1991.

[19] E. Rogdakis, G. Antonakos, and I. P. Koronaki, "Influence of a Regenerator on Stirling Engine Performance," *Journal of Energy Engineering*, vol. 142, no. 2, June 2016.

[20] D. M. Clucas, "Development of a Stirling Engine Battery Charger based on a Low Cost Wobble Mechanism," PhD Dissertation, University of Canterbury, 1993.

[21] L. Crema, F. Alberti, A. Bertaso, and A. Bozzoli, "Development of a pellet boiler with Stirling engine for m-CHP domestic application." *Energy, Sustainability and Society*, vol. 1, no. 1, 2011.

[22] M. Sheykhi, M. Chahartaghi, and S. M. Hashemian, "Performance Evaluation of a Combined Heat and Power System with Stirling Engine for Residential Applications," *Iranian Journal of Science and Technology, Transactions of Mechanical Engineering*, vol. 44, no. 4, pp. 975-984, 2019.

[23] G. T. Reader and C. Hooper, *Stirling Engines*. E and F Spon, London, 1983.

[24] T. S. Oros, I. Vadan, R. Pop, and C. Brad, "Linear generator for a free piston Stirling engine." *Journal of Electrical and Electronics Engineering*. vol. 7, no. 1, p. 111, 2014.

[25] N. Farra, T. Tzanetakis, and M. J. Thomson, "Experimental Determination of the Efficiency and Emissions of a Residential Micro Cogeneration System Based on a Stirling Engine and Fueled by Diesel and Ethanol*,"* *Energy & Fuels* vol. 26, no. 2, pp. 889-900, 2012.

[26] A. Alvarez-Aguirre, E. Garcia-Canseco, and J. M. Scherpen, "Linear dynamics and control of a kinematic wobble-yoke Stirling Engine" *49th IEEE Conference on Decision and Control*, December 15-17, 2010.

[27] “WP-5012-001-01 DC Marine User's Manual,” Whisper Tech Limited, Christchurch, New Zealand, 2005.

[28] M. Rosen and C. A. Bulucea, “Using Exergy to Understand and Improve the Efficiency of Electrical Power Technologies,” *Entropy*, vol. 11, no. 4, pp. 820-835, 2009.

[29] I. Dincer, “The Role of Exergy in Energy Policy Making,” *Energy Policy*, vol. 30, pp. 137-149, 2002.

[30] J. Szargut, M.R. David, R. Frank. “Exergy analysis of thermal, chemical, and metallurgical processes.” 1987.

[31] G. Tsatsaronis and M. H. Park, “On avoidable and unavoidable exergy destructions and investment costs in thermal systems,” *Energy Conversion and Management*, vol. 43, pp. 1259-1270, 2002.

[32] G. D. Vuckovic, M. V. Vukic, M. M. Stojiljkovic, D. D. Vuckovic, “Avoidable and Unavoidable Exergy Destruction and Exergoeconomic Evaluation of the Thermal processes in a Real industrial Plant,” *Thermal Science*, vol. 16, pp. S433-S446, 2012.

[33] G. Conroy, A. Duffy, and L. M. Ayompe, “Economic, energy and GHG emissions performance evaluation of a WhisperGen Mk IV Stirling engine μ-CHP unit in a domestic dwelling,” *Energy Conversion and Management*, vol 81, pp. 465-474, 2014.

[34] P. P. Kaliram, "Commissioning and Performance Analysis of WhisperGen Stirling Engine," M.A.Sc thesis, University of Windsor, 2016. *5661*.

VITA AUCTORIS

NAME:	Akash Mathew
PLACE OF BIRTH:	Kerala, India
YEAR OF BIRTH:	1995
EDUCATION:	Don Bosco Institute of Technology, BE (Mechanical Engineering), Maharashtra, India, 2017