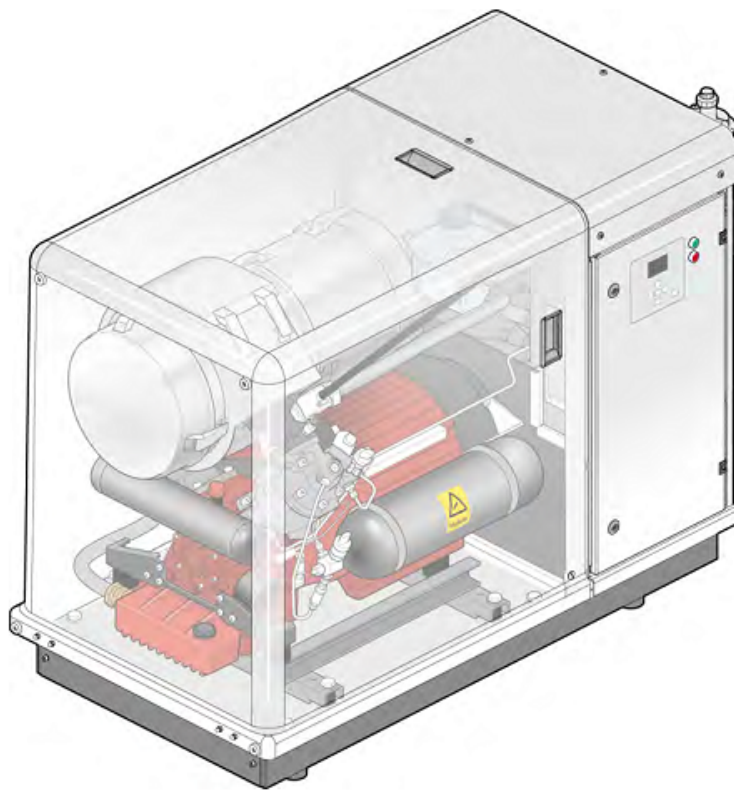


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An Experimental Evaluation of Low Calorific Value Gaseous Fuels as a Heat Source for Stirling Engines

Master's thesis in Sustainable Energy Systems

JOHAN WESTERBERG

Department of Applied Mechanics

Division of Combustion

CHALMERS UNIVERSITY OF TECHNOLOGY

Göteborg, Sweden 2014

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Cleanergy's C9G Stirling engine.

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ABSTRACT

This report, An Experimental Evaluation of Low Calorific Value Gaseous Fuels as a Heat Source for Stirling Engines, was written by Johan Westerberg as a Master's thesis at Chalmers University of Technology. It covers the topic of MILD combustion of liquid propane gas, G25 natural gas, synthesis gas and landfill gas, of which the latter two represents low calorific gaseous fuels. The investigated engine was a C9G Stirling engine from Cleanergy.

The primary incentive to this study was to determine if steady and reliable operation of the C9G Stirling engine could be maintained while using low calorific gaseous fuels. Another incentive to this project was to achieve further understanding of the combustion device, which purpose is to supply the Stirling engine with heat. Thus, an understanding of the changed characteristics caused by varying the air to fuel ratio was targeted.

Engine operation by use of landfill gas, made up by 20 percent methane in carbon dioxide and nitrogen, was successful at the lowest load setting allowed by the C9G engine, generating 1.5 kW of shaft power. For liquid propane gas and G25 natural gas, up to 10.9 kW of shaft power was generated. Combustion of synthesis gas was not attempted due to the hazard of carbon monoxide poisoning, caused by insufficient ventilation in the test cell.

Apparent changes in energy flows and temperatures within the combustion device at varied air to fuel ratios suggest that incomplete combustion prior to the heat sink of the Stirling engine was an issue, especially at low to moderate air to fuel ratios. Further, a reversed flow direction through the flue gas recirculation cavities was observed when high flow rates of fuel, which primarily occurs for high load settings and low air to fuel ratios, was supplied to the engine. Consequently, a higher air to fuel ratio resulted in a higher system efficiency for all tested fuels and load settings. Thus, the system efficiency was concluded to be heavily impacted by the air to fuel ratio.

Keywords: Stirling, biogas, landfill gas, synthesis gas, combustion, MILD, FLOX, C9G

PREFACE

The reader of this report should have basic knowledge in thermodynamics, especially the fundamentals of energy balances. Further, some understanding of combustion kinetics and the various modes of heat transfer is also beneficial.

If the reader is already familiar with the principles of the Stirling engine, Chapter 2 may be omitted. Further, Chapter 3 may be superseded if the characteristics of low calorific fuels and the principles of MILD combustion is known to the reader. However, the author suggest that the reader studies the definition of the gaseous fuels, Chapter 3.1, before commencing to later Chapters in this report.

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Anders Mattsson and Patrik Wåhlin, technical support staff of the department of Applied mechanics at Chalmers University of Technology, made extensive contributions in solving technical issues during the installation of several mechanical systems in the test cell. Both had prior experience in setting up the C9G Stirling engine for operation. Calibration of the various measurement systems was aided by Daniel Härensten, also a member of the technical support staff. Alf Magnusson coordinated the support staff to ensure proper time management.

From within the industry, Gustav Bos and Thomas Kröll, research engineers at Cleanergy, gave support in solving technical issues with the engine. Martin Nilsson and Pontus Wettrell, also from Cleanergy, contributed by supplying the author with calculations and information on expected output values.

Finally, Professor Ingemar Denbratt was the supervisor of this project. Together with Martin Nilsson, Ingemar drew the outlines of the study and served as an official link between Chalmers University of Technology and Cleanergy.

NOMENCLATURE

ϵ	Effectiveness of the preheater	
η_C	Efficiency of the ideal Carnot engine	
η_c	Efficiency of the combustion device	
η_{c+s}	Combined efficiency of the combustion device and the Stirling engine	
η_s	Efficiency of the Stirling engine	
λ	Air to fuel ratio	
ρ_c	Density of the coolant	[kg m ⁻³]
ρ_f	Density of the fuel	[kg m ⁻³]
τ	Torque	[N m]
ω	Angular velocity	[rad s ⁻¹]
$c_{p,air}$	Specific heat of the air	[J kg ⁻¹ K ⁻¹]
$c_{p,c}$	Specific heat of the coolant	[J kg ⁻¹ K ⁻¹]
$c_{p,fg}$	Specific heat of the flue gases	[J kg ⁻¹ K ⁻¹]
G_S	Specific density of the fuel	[]
$\dot{H}_p$	Enthalpy flow of the product gases	[W]
$\dot{H}_r$	Enthalpy flow of the reactant gases	[W]
$h_{air,amb}$	Enthalpy of the air at ambient conditions	[J kg ⁻¹]
$h_{air,ph}$	Enthalpy of the air at the preheater's exit	[J kg ⁻¹]
$h_{c,i}$	Enthalpy of the coolant the engine's inlet	[J kg ⁻¹]
$h_{c,o}$	Enthalpy of the coolant the engine's outlet	[J kg ⁻¹]
$h_{f,p}$	Heat of formation of the product gases	[J mol ⁻¹]
$h_{f,r}$	Heat of formation of the reactant gases	[J mol ⁻¹]
$h_{fg,ahh}$	Enthalpy of the flue gases after the heater head	[J kg ⁻¹]
$h_{fg,amb}$	Enthalpy of the flue gases at ambient conditions	[J kg ⁻¹]
$h_{fg,e}$	Enthalpy of the flue gases at the exhaust pipe	[J kg ⁻¹]
$h_{fg,fg}$	Enthalpy of the flue gases at the preheater's inlet/flue gas recirculation's inlet	[J kg ⁻¹]
$h_{fg,phh}$	Enthalpy of the flue gases prior to the heater head	[J kg ⁻¹]
$h_{fg,ref}$	Enthalpy of the flue gases at the reference state	[J kg ⁻¹]
$I_{W,HHV}$	Wobbe index based on the higher heating value of the fuel	[J m ⁻³]
$I_{W,LHV}$	Wobbe index based on the lower heating value of the fuel	[J m ⁻³]
$\dot{m}_{air}$	Mass flow rate of the air	[kg s ⁻¹]
$\dot{m}_c$	Mass flow rate of the coolant	[kg s ⁻¹]
$\dot{m}_{fg}$	Mass flow rate of the flue gases	[kg s ⁻¹]
$\dot{n}_p$	Molar flow rate of the product gases	[mol s ⁻¹]
$\dot{n}_r$	Molar flow rate of the reactant gases	[mol s ⁻¹]
n_{rev}	Revolutions per minute	[rpm]
P_i	Indicated power	[W]
P_s	Shaft power	[W]
p	Cylinder pressure	[Pa]
$T_{air,amb}$	Air temperature at ambient conditions	[K]
$T_{air,ph}$	Air temperature at the preheater's exit	[K]
T_C	Heat sink temperature	[K]
$T_{c,i}$	Coolant temperature at the engine's inlet	[K]
$T_{c,o}$	Coolant temperature at the engine's outlet	[K]
$T_{c,e,i}$	External coolant temperature at the heat exchanger's inlet	[K]
$T_{c,e,o}$	External coolant temperature at the heat exchanger's outlet	[K]
$T_{fg,ahh}$	Flue gas temperature after the heater head	[K]
$T_{fg,e}$	Flue gas temperature at the exhaust pipe	[K]
$T_{fg,fg}$	Flue gas temperature at the preheater's inlet/flue gas recirculation's inlet	[K]
$T_{fg,phh}$	Flue gas temperature prior to the heater head	[K]
T_H	Heat source temperature	[K]
T_{ref}	Reference temperature	[K]
V	Cylinder volume	[m ³]
V_C	Calorific value of the fuel	[J m ⁻³]

$\dot{V}_c$	Volume flow rate of the coolant	$[\text{m}^3 \text{s}^{-1}]$
$\dot{V}_f$	Volume flow rate of the fuel	$[\text{m}^3 \text{s}^{-1}]$
$\dot{V}_{fg}$	Volume flow rate of the flue gases	$[\text{m}^3 \text{s}^{-1}]$
$\dot{W}_{comp}$	Compression work	$[\text{J}]$
$\dot{W}_i$	Indicated work	$[\text{J}]$
$\dot{W}_{exp}$	Expansion work	$[\text{J}]$
$\dot{Q}_c$	Heat rejected by the internal cooling system	$[\text{W}]$
$\dot{Q}_e$	Heat rejected by the exiting exhaust gases	$[\text{W}]$
$\dot{Q}_f$	Heat rejected by friction	$[\text{W}]$
$\dot{Q}_i$	Heat added by the fuel	$[\text{W}]$
$\dot{Q}_{o,c}$	Heat rejected by other sources in the combustion device	$[\text{W}]$
$\dot{Q}_{o,s}$	Heat rejected by other sources in the Stirling engine	$[\text{W}]$
$\dot{Q}_{p,air}$	Heat accepted by the air in the preheater	$[\text{W}]$
$\dot{Q}_{p,fg}$	Heat rejected by the flue gases in the preheater	$[\text{W}]$
$\dot{Q}_{p,max}$	Maximum heat transfer through the preheater	$[\text{W}]$
$\dot{Q}_{comb,prompt}$	Combustion heat released prior to the heater head	$[\text{W}]$
$\dot{Q}_u$	Useful heat accepted by the heater head	$[\text{W}]$

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1 Introduction

In the year 1816, Robert Stirling registered a patent for the Stirling engine [6]. It was intended to be an alternative to the steam engine, not only providing a higher efficiency but also eliminating the hazard of a boiler explosion. Following the development of the combustion engine in the late 19th century, the commercial interest in the Stirling engine disappeared [26]. Even though research and development was resumed in the late 1930s, the technology has only been applied in a few engineering applications. Current applications include propulsion of submarines of the Royal Swedish Navy and electricity generation from solar energy, but the interest in this newborn heat engine is growing [19]. Cleanergy, founded in 2008, is a company providing energy solutions based on Stirling engine technology.

Stirling engines have the advantage of being supplied by an external heat source. Hence, Cleanergy is able to utilise Stirling engine technology in both solar and combined heat and power production, where the latter is based on combustion of biogas or natural gas.

At present, biological energy sources provides 10 percent of the global primary energy supply [35]. However, this figure is significantly higher in rural areas, in which biomass is primarily used for cooking and heating purposes. The use of primitive equipment results in low conversion efficiencies, thus resulting in enhanced deforestation.

Due to the possibility to reduce greenhouse gas emissions, the International Energy Agency identified bioenergy as an important tool for dealing with the climate crisis [18]. One identified pathway for bioenergy utilisation is the conversion of biomass into biogas, which is then either combusted directly in a combined heat and power application or upgraded to a methane concentration equivalent to natural gas.

Current applications to burn biogases include various types of engines adapted to the Otto principle. However, these engines requires a methane concentration in excess of 45 percent [31]. In order to facilitate further development within this area, Cleanergy intends to gather additional knowledge in operating Stirling engines with biogases containing even less combustibles.

1.1 Purpose

This thesis was intended to investigate the running conditions of a C9G Stirling engine using four different fuels. The fuels of interest were liquid propane gas, G25 natural gas, a methane based low grade gas from landfill sites and synthesis gas produced by biomass gasification. The two former fuels served as a reference for testing the latter two. Hence, the main ambition was to investigate the operational conditions of the engine while fuelled by landfill gas or synthesis gas. The engine system was set up in order to determine some key operational characteristics;

- The maximum possible work output of the Stirling engine when operated by low calorific gaseous fuels and while maintaining a shaft speed of 1500 rpm
- A quantification of important energy flows
- The efficiency of the combustion device and the Stirling engine
- The effectiveness of the air preheater
- The influence on engine performance caused by the air to fuel ratio
- The overall stability of operation

1.2 Limitations

The main focus was directed towards investigating the operational characteristics of the Stirling engine and the combustion device using four gaseous fuels. Hence, only fuel properties related to combustion heat release is thoroughly discussed. No regard was taken to other fuel properties, including;

- Availability, cost and security of supply
- Corrosiveness and foulness to the combustion equipment

- Emissions forming and potential flue gas cleaning
- Legislation or prohibitions

The three main Stirling engine configurations are briefly presented. However, since the C9G engine is of alpha configuration, the principal point was to investigate the mechanical function and calculation methods applicable to this engine type.

This thesis, proposed by Cleanergy, was performed in one of the on site testing cells located at Chalmers University of Technology, department of Applied Mechanics, division of Combustion. It should be stressed that the amount of fuel available for engine testing was limited not only by price, but also due to the extensive volume of fuel required for low calorific fuel combustion. Further, the amount of hours available in the test cell was also limited, thus forming additional constraints to the extent of the test schedule.

2 The Stirling engine

In order to provide an understanding of the fundamental principle of the Stirling engine, an explanation of the thermodynamic process of this heat engine is presented, as well as its similarities with the Carnot cycle. Later sections of this chapter present the different mechanical configurations of the Stirling engine as well as some of its key characteristics.

2.1 Thermodynamic cycles

The thermodynamic cycle undergone by a heat engine often implies a system using a fluid as the working substance [8]. The system can be either open or closed, and the working fluid is carried through four different states. In a closed system, the working fluid will return to the initial state when the cycle is completed. A Stirling engine operates as a closed system. An opened system holds the same basic states, but the working fluid is disposed as the last state of the cycle is reached. Additional working fluid is then added at the initial state, forming the characteristics of an open cycle. While carried through the different states, the working fluid in a heat engine may change phases.

2.1.1 The Carnot cycle

The Carnot cycle is a reversible thermodynamic cycle, proposed by Nicolas Léonard Sadi Carnot in 1824 to demonstrate the theoretically most efficient energy conversion cycle between two heat reservoirs [8]. Consequently, the Carnot efficiency is the maximum energy conversion efficiency to be obtained between a heat source and a heat sink at finite temperatures. Hence, it is used to evaluate the efficiency of a real process. To properly describe the Carnot cycle, four different steps are defined, Figure 2.1.

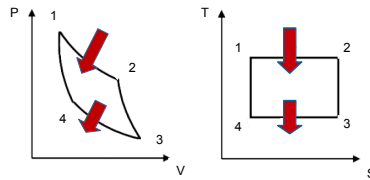


Figure 2.1: *The P-V and T-S diagram of the Carnot cycle.*

1-2) Reversible, isothermal expansion of the working fluid at high temperature. The working fluid is expanded, allowing heat to be accepted from the high temperature reservoir at constant temperature.

2-3) Isentropic (reversible and adiabatic) expansion of the working fluid. During this step, in which energy is extracted from the system through boundary work, the piston and the cylinder is assumed to be insulated, preventing any heat exchange with the environment. The expansion is carried out until the working fluid has reached the temperature of the low temperature reservoir.

3-4) Reversible, isothermal compression of the working fluid at low temperature. The working fluid is compressed, allowing heat to be discharged to the low temperature reservoir at constant temperature.

4-1) Isentropic (reversible and adiabatic) compression of the working fluid. Again, the piston and the cylinder is assumed to be insulated. The compression is carried out until the working fluid has reached the temperature of the high temperature reservoir.

In reality, it is impossible to achieve complete Carnot efficiency. However, trying to construct a heat engine that closely resembles the Carnot cycle will result in a more efficient engine. The factors that rule out the possibilities of complete Carnot efficiency can be directly linked to the different steps of the process.

Difficulties associated with accepting and discharging heat at a constant temperature during step (1-2) and (3-4) exists. This would require a heat transfer process that is rapid enough to ensure that the temperature of the working fluid does not decrease or increase during the expansion and compression respectively. If the temperature of the working fluid deviates from the temperature of the reservoir, heat transfer over a finite temperature difference follows as a result, thus allowing losses to occur due to increased entropy. The quality of the insulation in terms of preventing heat transfer to the environment during step (2-3) and (4-1) also influences the efficiency of the cycle.

Furthermore, the Carnot cycle does not account for any friction losses within the heat engine and additional losses may also occur following non-quasi-steady expansion or compression at high speed of revolution. The above described factors are irreversibilities, directly corresponding to efficiency losses as part of the energy associated with the process become non-retrievable [37].

2.1.2 The Stirling cycle

The fundamental principle of the Stirling cycle is based upon a constant quantity of gas being contained by a warm and a cold compartment. In a closed cycle, the gas is heated and cooled in turns, giving rise to volume changes and boundary work. The fact that the Stirling cycle resembles the Carnot cycle provides the Stirling engine with the possibility to reach high efficiencies compared to other types of engines [40].

The P-V and T-S characteristics of the Stirling cycle and Carnot cycle are almost similar. The differences are due to deviations from the Carnot cycle, leaving the real Stirling cycle with a smoother P-V and T-S appearance as it fails to complete the different steps of the Carnot cycle in a completely reversible manner. In other words, an ideal Stirling engine entirely resembles the Carnot cycle. Further, in the Stirling cycle, step (1-4) and (2-3) are isochoric processes.

In order to explain the fundamental principle further, two important components need to be defined; the piston that converts boundary work into shaft work and seals the working fluid from the surrounding environment, and the displacer, whose main purpose is to transport the fluid. The piston and the displacer may be operating in the same cylinder, or in separate cylinders, depending on the considered type of Stirling engine. Since a part of the work output of the Stirling engine is used to compress the working fluid, the net work is defined as the difference between the boundary work gained when the working fluid is expanded and the boundary work consumed when the gas is compressed.

Stirling engines often use a regenerator in order to store a part of the rejected heat as the working fluid is transported from the warmer side of the engine to the colder side. When transported in the opposite direction, the working fluid will accept heat, partly from the high temperature reservoir and partly from the regenerator, thus decreasing the demand for heat addition from the reservoir.

2.2 Mechanical concepts

Stirling engines are typically divided into three different basic concepts or types; alpha, beta and gamma [6]. The thermodynamic principle is the same for each type of engine. These three basic concepts are forming a basis, from which other configurations of the Stirling engine have been developed.

One example of these sub-configurations is the double-acting alpha Stirling engine, where several pistons and cylinders co-operate by using the upper surface of every piston for expansion and the lower surface for compression [22]. Another example is the beta configured rhombic drive Stirling engine, in which two gears are forced to rotate in opposite directions [11]. As a result, the piston and the displacer is completing the cycle in a perfectly coaxial manner.

2.2.1 The Alpha engine

The alpha engine consists of two pistons in separate cylinders, connected by a pipe allowing the working fluid to flow between the cylinders [40], Figure 2.2. There is no dedicated displacer in the alpha engine as both pistons are contributing to the movement of the working fluid. A phase shift between the two pistons results in the working fluid being compressed and expanded in turns and forced to flow between the cylinders. The expansion occurs while accepting heat from the high temperature reservoir (red), whereas compression occurs while heat is rejected at the low temperature reservoir (blue).

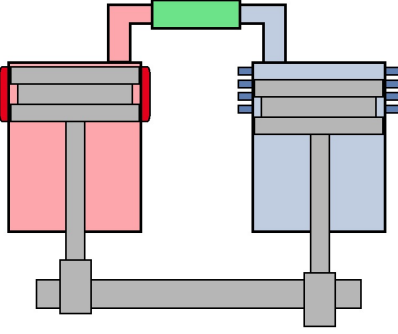


Figure 2.2: *The schematics of an alpha engine. Components include a crankshaft, two pistons and two cylinders. A Heater and a cooler is attached to the warm and cold cylinder respectively. A regenerator is used to conserve some of the heat within the system as the working fluid is passing through the channel between the cylinders.*

1. The piston in the cold cylinder is at its upper dead state and the piston in the warm cylinder is descending. Most of the working fluid is located in the warm cylinder, except the amount located in the regenerator (green) and the dead volume of the cold cylinder. Heat addition causes the working fluid to expand, generating boundary work as the gas forces the piston in the warm cylinder to descend.
2. The piston in the warm cylinder is at its lower dead state and the piston in the cold cylinder is descending. The working fluid is expanded and will shortly reach maximum volume.
3. The piston in the cold cylinder is at its lower dead state and the piston in the warm cylinder is ascending. The majority of the working fluid is transported from the warm to the cold cylinder where it is compressed during heat rejection to the low temperature reservoir. Work is consumed during the compression, aided by the momentum stored in the flywheel.
4. The piston in the warm cylinder is at its upper dead state and the piston in the cold cylinder is ascending. The working fluid is compressed and will shortly reach minimum volume.

The basic components in the alpha engine do not differentiate principally from those in an internal combustion engine as the design of the engine block, the crankshaft, the cylinders and the pistons are similar. The major disadvantage of the alpha engine compared to the other configurations of the Stirling engine is the difficulty associated with sealing the warm cylinder, especially in high temperature applications [40]. High temperatures enhance the challenge of designing well sealed piston rings, since a gas at high temperature has more internal energy and therefore contains atoms at greater velocities than a gas at lower temperature. Added to this, the working fluid often consists of highly diffusive gases such as helium or hydrogen.

2.2.2 The Beta engine

The most significant difference between the beta engine and the alpha engine is the design of the pistons and the fact that they operate within the same cylinder [40], Figure 2.3. This results in the possibility of designing an engine with a lower total volume.

In contrast to the alpha engine, in which both pistons contribute to the transportation of the working fluid, the beta engine uses a displacer for this purpose. The displacer transports the working fluid between the cold and the warm end of the cylinder through a pipe, often located in the cylinder wall, and is thereby not increasing the pressure of working fluid. The piston's location on the crankshaft is shifted to the displacer's. Throughout the cycle, the same piston is collecting the boundary work from the working fluid during the expansion as well as performing boundary work on the working fluid during the compression.

In the following description of the beta cycle, using steps of crank angle, the working fluid refers to the fluid above the piston and the crankshaft is rotating counter-clockwise.

1. The displacer is at its lower dead state and the piston is descending. The working fluid, except the amount of fluid located in the regenerator (green), is located in the warm end of the cylinder. Heat is accepted from the high temperature reservoir and causes the working fluid to expand, pushing the piston down to generate boundary work.

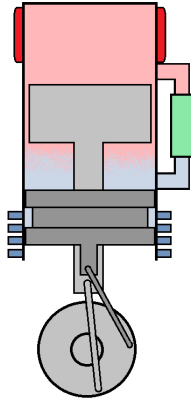


Figure 2.3: *The schematics of a beta engine. Components include a crankshaft, a cylinder and two pistons. A heater and a cooler is attached to the warm and cold ends of the cylinder respectively. A regenerator is used to conserve some of the heat within the system as the working fluid is passing through the channel between the two ends of the cylinder. The upper piston, called displacer, is used for transporting the working fluid between the compartments.*

2. The piston is at its lower dead state and the displacer is ascending. The working fluid has been completely expanded and has reached its maximum volume.
3. The displacer is at its upper dead state and the piston is ascending. The working fluid is compressed as it flows from the warm to the cold end of the cylinder, rejecting heat to the cold temperature reservoir simultaneously. Work is consumed during the compression, aided by the momentum stored in the flywheel.
4. The piston is at its upper dead state and the displacer is descending. The working fluid has been fully compressed and has reached its minimum volume.

The beta engine has the advantage of having only one piston, thus reducing the total contact surface between pistons and cylinders. Further, the difficulties associated with designing well sealed piston rings are aided by the fact that the engine is sealed by the piston at the cold part of the engine only.

2.2.3 The Gamma engine

The gamma engine can be described as a mixture of the alpha engine and the beta engine, Figure 2.4. Like the beta engine, the gamma engine uses a displacer and a piston. However, they are located in separate cylinders, as is the case in the alpha engine. The cylinders are linked with a pipe to allow the working fluid to flow between the cylinders. Another pipe is used to lead the working fluid around the displacer, from the cold to the warm end of the cylinder. The latter pipe, a feature shared with the beta engine, holds the regenerator. The four step definition of the cycle is similar to the beta engine's.

One disadvantage of the gamma engine compared to the beta engine is the increased volume associated with linking the two cylinders together. The added volume is not contributing to the compression of the working fluid, resulting in a decreased compression ratio and thereby a lower thermal efficiency [40]. Hence, the gamma engine becomes a suitable alternative when the advantages of using two separate cylinders surpass the request for a high efficiency.

2.3 Aspects of power generation

Some of the qualities of Stirling engines make them particularly appropriate for electricity generation. One positive feature is the possibility to use the same type of engine in different applications, without major modifications. The external heat source may vary from solar and geothermal energy extraction to combustion heat release.

Another positive attribute that comes with an external heat source is that no combustion occurs inside the cylinders. Hence, soot and other combustion products are not mixed with the working fluid. As a result, Stirling engines generally have longer maintenance intervals than internal combustion engines, in which soot

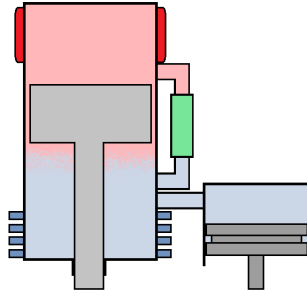


Figure 2.4: *The schematics of a gamma engine. Components include a crankshaft, two pistons and two cylinders. A heater and a cooler is attached to the warm and cold ends of the primary cylinder respectively. A regenerator is used to conserve some of the heat within the system as the working fluid is passing through the channel between the two ends of the primary cylinder. The piston in the main cylinder, called displacer, is used for transporting the working fluid between the compartments. Work is extracted by the piston in the smaller secondary cylinder.*

is accumulated in the cylinders [40]. Further, vibrations inside Stirling engines can be almost completely eliminated due to the use of an external heat source.

Stirling engines obey similar mechanical principles as internal combustion engines, allowing the designer to use components currently in use by the automotive industry [20]. If Stirling engines are used to generate electricity from solar energy, they also have the advantage of not using any scarce materials when manufactured, as is the case for most photovoltaic devices [28].

The fact that Stirling engines use an external heat source is also contributing to one of their most significant disadvantages. Apart from internal combustion engines, in which fuel is combusted inside the engine, Stirling engines reacts slowly to changes in work load. Consequently, Stirling engines are often considered inappropriate for use in the automotive industry [36].

Last, a major disadvantage of Stirling engines is that they require a high gauge pressure in order to obtain a sufficient rated power. Added to this, diffusive gases such as helium or hydrogen are often used as the working fluid. High pressures are always associated with high demands on piston rings and other components being used to seal the engine from the environment [36].

3 Combustion of gaseous fuels

This chapter will serve as an introduction to combustion of gaseous fuels. First, the four fuels used in this investigation will be defined. Special attention is directed towards the complications and benefits provided by using fuels having a low calorific value. Further, a brief summary of previous research on low calorific fuel as a heat source will be provided. Last, a description of MILD combustion technology is presented.

3.1 Definition of the selected fuels

Four different gaseous fuels were chosen to be a part of this study, Table 3.1.

Table 3.1: Gas composition of selected fuels [molar%]

	Liquid propane gas	Natural gas	Synthesis gas	Landfill gas
CH ₄	-	86.0	1.5	20.0
C ₂ H ₆	0.7	-	-	-
C ₃ H ₈	98.1	-	-	-
C ₄ H ₁₀ ≤	1.2	-	-	-
H ₂	-	-	18.0	-
CO	-	-	23.5	-
CO ₂	-	-	9.0	20.0
N ₂	-	14.0	48.0	60.0

One representation of a fuel with high calorific value was provided in liquid propane gas, LPG in short. However, small quantities of other carbon chained molecules also occurred in this compound. Another second representation of a high calorific fuel was made by G25 natural gas. By definition, G25 natural gas, hereby denoted natural gas, has a methane concentration of 86 molar percent in nitrogen [4].

Wood producer gas served as a representation of a fuel with a low calorific value. Being primarily comprised of hydrogen, carbon monoxide and carbon dioxide it may also be referred to as a synthesis gas [27]. Further, this gas included some traces of methane and a large share of nitrogen.

A second representation of a fuel with low calorific value was provided in landfill gas, which was comprised of methane and carbon dioxide in nitrogen. It should be stressed that the amount of carbon dioxide in the studied gas was lower than that of conventional landfill gas. This modification was made due to problems with gas condensation in gases with high content of carbon dioxide at elevated pressures. However, the calorific value of the gas remained representative for landfill gas.

3.2 Low calorific gaseous fuels

Wood producer gas is derived from biomass in a process called pyrolysis, in which part of the biomass is combusted in an environment with a restricted oxygen supply [1]. Pyrolysis products include gases, vapours, liquids, oils, solid char and ash. The process output depends on the temperature, the oxygen environment and the characteristics of the input material. In cases where a combustible gas is the desired output, one may instead refer to gasification, which is essentially pyrolysis adapted for optimal secondary gas production. The efficiency of the process is calculated by dividing the heat of combustion of the secondary fuel by the heat of combustion of the primary fuel. The efficiency of producer gas generation from biomass may be as high as 80 percent, since a significant part of the initial energy from biomass is in the form of combustible hydrogen and carbon monoxide [35].

Anaerobic digestion is the process that governs the production of landfill gas. In an oxygen depleted environment, micro-organisms meet their need for energy by reacting with carbonaceous compounds to form methane and carbon dioxide [23]. The efficiency of biogas generation from landfill is ranging from 60 to 90 percent [35]. Still, one has to keep in mind that landfill or sewage gas is generally obtained from slurries with a significant water content. Thus, no energy could be conveniently extracted without anaerobic digestion. In other words, the volume of waste requiring disposal is reduced while transforming waste material into a flexible energy carrier [31].

3.2.1 Characteristics

Fuels having a low calorific value generally have similar features. Even though they are made up of different compounds, they all share the feature of having a low percentage of combustibles.

Problems may arise when low calorific fuels are used in conventional combustion devices. A low heating value directly translates into a low flame temperature, thus resulting in slower combustion kinetics within the combustion device [25]. Further, gases with a high nitrogen content generate combustion products with a high nitrogen content, thus producing a product gas with lower thermal emissivity [12]. Consequently, a smaller amount of the released heat is accepted by the combustion device. Last, combustion of low calorific fuels result in a higher volume of combustion products. Thus, insufficient residence time may not only lead to incomplete combustion, but also decreased heat transfer due to a decreased residence time adjacent to the heat accepting areas in the combustion device.

During combustion of fuels with low calorific value, supplementary firing may be required should the temperature drop lower than what is necessary to sustain combustion [3]. If the flame temperature of the fuel is sufficient in order to sustain combustion, a supplementary fuel source may still be required initially. Because of the low temperature in the combustion device before thermal equilibrium has been reached, unwanted flame quenching may result if the heat supply is insufficient.

Another lineament is the variability of fuel constituents. Variations in the processes producing these gases occur, namely variations in biomass composition supplied to the gasifier or varying conditions at the landfill site. Further, the fact that wood producer gas is made up by five main substances of different characteristics makes it inconvenient to compress [35]. This is also true for landfill gas due to the high content of carbon dioxide.

Apart from the primary constituents of a low calorific gaseous fuel such as wood producer gas or landfill gas, there are several trace compounds to consider [2, 14]. Monitoring trace compounds is important not only because of the existence of legislative restrictions on emissions of these compounds, but also because of possible threats to the integrity of the combustion device.

3.2.2 Applications

Information on the application of low calorific gaseous fuels as a heat source for Stirling engines is scarce. Unsurprisingly, the application of low calorific fuels is primarily restricted to more commercially mature technologies on the energy market. For example, successful operation has been accomplished while using low calorific fuels in gas turbines and internal combustion engines [2, 9, 21, 24, 25, 32, 34].

3.3 MILD combustion

The technology used for fuel oxidation in these experiments is a combustion technology referred to as moderate or intense low-oxygen dilution, MILD. Other denotations for this technology is flameless oxidation, FLOX [41], or high temperature air combustion, HTAC [43]. A. Cavaliere and M. de Joannon defined MILD combustion as a process in which both the inlet temperature of the reactant mixture and the maximum allowable temperature increase during combustion do not exceed the self ignition temperature of the mixture [7]. The self-ignition temperature is defined as the inlet temperature at which any additional temperature increase makes the chemical process self-sustaining. This occurs since a higher branch in the S-shaped curve governing ignition characteristics is reached [13].

In order to obtain MILD combustion, flue gases are recycled, intending to dilute the supplied oxidiser. The oxygen concentration of the oxidiser is decreased as combustion products are mixed into the oxidiser, thus resulting in slower combustion kinetics, which in turns limit the rapid characteristics of conventional flame propagation [3]. Further, the recycled flue gases add heat capacity to the compound, thus limiting the maximum temperature increase.

One key difference between MILD combustion and conventional combustion, which is associated with a significant heat release under chaotic conditions, is that the former requires preheating of the reactants in order to sustain combustion. Further, apart from conventional combustion being characterised by flame propagation, a combustion system operating at MILD conditions lacks a distinct shift between unoxidised and oxidised conditions [7].

Advantages of MILD combustion include an enlarged reaction zone as combustion is spread over a larger volume, thus resulting in lower peaks in flame temperature. Reduced flame temperature translates to reduced formation of nitric oxides and nitrogen dioxide, whereas a larger volume is equivalent to more complete

combustion, thus reducing the formation of carbon monoxide [38]. A combustion device operating at MILD conditions, which lacks peaks in flame temperatures due to a larger combustion zone, has a more uniform temperature profile. Thus, fluxes in radiative heat transfer are more uniform, which simplifies the prediction of heat transfer performance [39]. Further, the larger volume of combustion enables greater mixing and thus higher thermal efficiency [5, 42, 44]

Despite the obvious advantages connected to MILD combustion, one substantial problem is present. Given the fact that the reactants are mixed at an elevated temperature, there is a potential risk of spontaneous combustion of the reactant mixture prior to reaching the primary combustion zone [5]. The result may be a flashback and flame stabilisation at the injector, thus deviating from the desired performance due to the occurrence of a flame. A flashback may also result in hazardous physical damage to the combustion device [3]. The gas composition is a decisive factor in determining the temperature at which spontaneous combustion occur. Thus, information on auto-ignition temperatures for the selected fuels is essential in order to guarantee a process with MILD combustion characteristics.

Further, the auto-ignition time for a fuel in a MILD combustion device is significantly dependent on the concentration of carbon dioxide in the reactants. Carbon dioxide inhibits the chemical reaction rate due to the decreased amount of the free radicals O, H and OH, thus decreasing the oxidation rate of methane and other combustibles [5]. Landfill gas and synthesis gas are both comprised of considerable amounts of carbon dioxide. Thus, complications to obtain complete combustion may occur if the flue gas recirculation is not compensated properly.

The oxidisation of biogas using MILD combustion technology has been experimentally tested in a lab-scale furnace by S. Hosseini and M. Wahid. Conclusions include reduction in pollutant formation compared to conventional combustion and increased durability of exposed equipment due to decreased thermal stress [16]. M. Sánchez, F. Cadavid and A. Amell concluded that flameless characteristics can be obtained for methane combustion at enriched oxygen concentrations, thus enabling oxy-fuel combustion [30]. The effect of hydrogen blending to increase the calorific value of the fuel has been investigated by S. Chen and C. Zheng, concluding that only minor fuel alterations appear necessary for a counterflow diffusion flame operating at MILD conditions [10]. A. Colorado, B. Herrera and A. Amell discovered that efficiencies were similar for combustion of natural gas and biogas in a furnace operated at MILD conditions. Thus, the interchangeability of these fuels appeared to be eased when operating at MILD conditions [12]. However, great differences in heat transfer patterns were noticed due to differences in flue gas composition.

4 Method

This chapter presents the methodology used for testing the four gaseous fuels. First, the thermodynamic relationships required for calculating the performance of the combustion device and the Stirling engine are presented. Second, the interchangeability of the four fuels is discussed, coupled with a description of the constraints to engine operation. Last, information about the test installation is laid out.

4.1 Performance evaluation of the combustion device

A combustion device operating at MILD conditions requires preheating of the oxidiser, Chapter 3.3. Thus, in a C9G Stirling engine, the supplied air is heat exchanged with the flue gas stream before the flue gases are released to the exhaust pipe. The heated air stream is then mixed with the fuel as it is ejected into the mixing tube. Here, the gas mixture is diluted with recirculated flue gases in order to decrease the oxygen concentration of the mixture, as well as to increase the mixture's heat capacity. As a result, the pattern of chaotic temperature increase following combustion is reduced. After oxidation, useful heat is extracted from the combustion device as the gases pass through a heat exchanger, which is the heater head of the Stirling engine. A schematic description of the combustion device is presented in Figure 4.1.

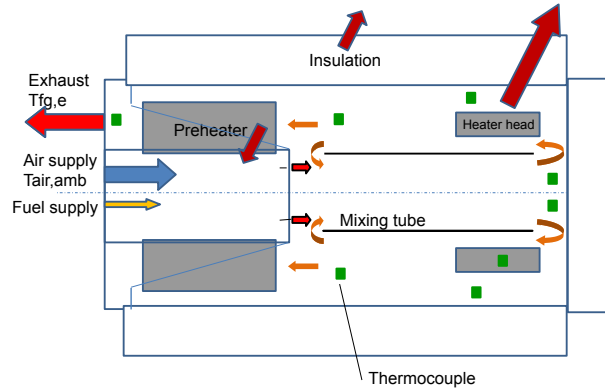


Figure 4.1: *The schematics of the combustion device and the position of the installed equipment for temperature measurements.*

The control volume for evaluating engine performance was drawn around the entire combustion device. From a system perspective, energy is entering the system in the form of thermal energy in the fuel and air streams, as well as chemical bond energy in the fuel stream. The exiting energy flows are associated with the useful heat extracted by the heater head, the heat losses through the insulation of the combustion device and the exhaust gas losses. Further, all reactants enter the system at the ambient conditions of the test cell. Thus, the heat flow associated with the thermal energy of the fuel and air streams was eliminated by using ambient conditions as the reference state. Heat transfer through the preheater was considered to be occurring within the system boundaries.

4.1.1 Heat addition

Heat addition was calculated based on the energy released by combustion, $\dot{Q}_i$. Under the assumption of complete combustion, Equation (4.1), the mass flow of combustion products, $\dot{m}_{fg}$, can be determined by measuring the volume flow rate of fuel, $\dot{V}_f$. Then, by considering the heat of formation, h_f , this allows for calculating the heat release, Equation (4.2). This calculation was carried out for all species of combustion product, n_p . In order to verify the accuracy of the calculated flue gas composition, the calculated heat release was compared to the expected heat release based on the lower heating value of the fuel. Additional verification

was obtained by determining conservation of mass.

$$C_a H_b O_c N_d + (a + \frac{b}{4} - \frac{c}{2})\lambda(O_2 + 3.773N_2) \Rightarrow aCO_2 + \frac{b}{2}H_2O + 3.773\lambda(a + \frac{b}{4} + \frac{d}{2})N_2 + (\lambda - 1)(a + \frac{b}{4} - \frac{c}{2})O_2 + \dot{Q}_i \quad (4.1)$$

$$\dot{Q}_i = \sum_{j=1}^n (\dot{H}_{r,j} - \dot{H}_{p,j}) = \sum_{j=1}^n \dot{n}_{r,j} h_{f,r,j} - \sum_{j=1}^n \dot{n}_{p,j} h_{f,p,j} \text{ [W]} \quad (4.2)$$

4.1.2 Heat rejection

Determining the molar fractions of the combustion products further allows for calculating the heat rejected by the exhaust gases leaving the combustion device, $\dot{Q}_e$. Proper evaluation of the specific heat, $c_{p,fg}$, coupled with measurements of the exhaust gas temperature, $T_{fg,e}$, and the reference temperature, T_{ref} , were used to determine the heat rejection through the exhaust pipe, Equation (4.3).

$$\dot{Q}_e = \sum_{j=1}^n \dot{m}_{fg,j} (h_{fg,e,j} - h_{fg,ref,j}) = \sum_{j=1}^n \dot{m}_{fg,j} c_{p,fg,j} (T_{fg,e} - T_{ref}) \text{ [W]} \quad (4.3)$$

In a similar manner, the useful heat recovery was calculated from an energy balance over the heater head of the Stirling engine, Equation (4.4). To facilitate this, the flue gas temperature was measured prior to, $T_{fg,phh}$, and after, $T_{fg,ahh}$, passing through the heater head of the Stirling engine.

$$\dot{Q}_u = \sum_{j=1}^n \dot{m}_{fg,j} (h_{fg,phh,j} - h_{fg,ahh,j}) = \sum_{j=1}^n \dot{m}_{fg,j} c_{p,fg,j} (T_{fg,phh} - T_{fg,ahh}) \text{ [W]} \quad (4.4)$$

4.1.3 Other losses

A summation of all energy flows, Figure 4.1, did not completely equal the total heat input. The remainder of the expected heat supply which remained unaccounted for was due to heat losses through the insulation of the combustion device and possible losses due to incomplete combustion of the fuel. These additional heat losses, $\dot{Q}_{o,c}$, were determined by subtracting the useful heat and the heat rejected by the exhaust gases from the heat input, Equation (4.5).

$$\dot{Q}_{o,c} = \dot{Q}_i - \dot{Q}_u - \dot{Q}_e \text{ [W]} \quad (4.5)$$

4.1.4 Combustion efficiency

A principal step in evaluating the performance of the combustion device is to determine the combustion efficiency, being defined as the ratio of the useful heat extraction, $\dot{Q}_u$, and the total potential heat release based on the chemically bond energy available in the fuel, $\dot{Q}_i$, Equation (4.6), [3].

$$\eta_c = \frac{\dot{Q}_u}{\dot{Q}_i} \quad (4.6)$$

4.1.5 Internal heat balance

The indirect effect of preheating in MILD combustion will result in a higher combustion efficiency, Chapter 3.3. Thus, in the C9G Stirling engine, heat is transferred from the flue gas stream to the air supply stream within the system boundaries of the combustion device. The amount of heat transferred from the flue gas stream through the preheater, $\dot{Q}_{p,fg}$, was calculated by setting up an energy balance over the preheater, Equation (4.7), thus applying the same procedure as for calculating the heat rejection, Chapter 4.1.2. In order to evaluate

this, the flue gas temperature was measured prior to the preheater, $T_{fg,fg}$, as well as after, $T_{fg,e}$. The latter temperature is the exhaust gas temperature of the combustion device.

$$\dot{Q}_{p,fg} = \sum_{j=1}^n \dot{m}_{fg,j} (h_{fg,fg,j} - h_{fg,e,j}) = \sum_{j=1}^n \dot{m}_{fg,j} c_{p,fg,j} (T_{fg,fg} - T_{fg,e}) \text{ [W]} \quad (4.7)$$

Further, conservation of energy concludes that the air supply will accept any heat being rejected by the flue gases, Equation (4.8).

$$\dot{Q}_{p,fg} = \dot{Q}_{p,air} \quad (4.8)$$

Thus, the outlet temperature of the preheated air stream, $T_{air,ph}$, can be calculated by interpolation, Equation (4.9). The mass flow of air, $\dot{m}_{air}$, was calculated using the assumption of complete combustion coupled with a measurement of the air to fuel ratio, Equation (4.1). The enthalpy of the air supplied to the preheater was denoted as the ambient temperature, $T_{air,amb}$.

$$\dot{Q}_{p,air} = \sum_{j=1}^n \dot{m}_{air,j} (h_{air,ph,j} - h_{air,amb,j}) = \sum_{j=1}^n \dot{m}_{air,j} c_{p,air,j} (T_{air,ph} - T_{air,amb}) \text{ [W]} \quad (4.9)$$

The fluid having the lowest heat capacity will have the highest temperature increase or decrease. Hence, the maximum amount of heat transfer possible between the two streams was calculated by letting the fluid with the lowest heat capacity operate between the inlet temperatures of both streams, Equation (4.10).

$$\dot{Q}_{p,max} = \sum_{j=1}^n (\dot{m}_j c_{p,j})_{min} (T_{inlet,fg} - T_{inlet,air}) = \sum_{j=1}^n (\dot{m}_j c_{p,j})_{min} (T_{fg,fg} - T_{air,amb}) \text{ [W]} \quad (4.10)$$

The preheater performance can then be fully assessed by calculating the effectiveness of the heat transfer, Equation (4.11) [15]. The effectiveness is the ratio of the actual heat transfer, $\dot{Q}_{p,air}$, and the maximum heat transfer possible between the two streams, $\dot{Q}_{p,max}$.

$$\epsilon = \frac{\dot{Q}_{p,air}}{\dot{Q}_{p,max}} \quad (4.11)$$

Finally, in order to verify the accuracy of these calculations, a separate calculation of the effectiveness was carried out using the NTU-method [17, 15].

4.1.6 Additional remarks

In addition to the above, mapping the temperature patterns of the entire combustion process would prove interesting. If a uniform temperature profile throughout the entire primary combustion zone was discovered, this would confirm that the combustion device operates at MILD conditions. However, the thermocouples available for engine testing were considered insufficient for this task due to the expected heat release.

4.2 Performance evaluation of the Stirling engine

In evaluating a Stirling engine, four energy flows of primary interest were identified. These flows are related to the four steps in the Stirling cycle, namely heat addition, heat rejection, expansion work and compression work, Figure 4.2. The heat addition is the useful heat extracted from the combustion device, $\dot{Q}_u$, Chapter 4.1.2.

4.2.1 Heat rejection

Heat rejection from the internal cooling system, $\dot{Q}_c$, was calculated based on the difference in enthalpy flow of the inlet and the outlet stream, Equation (4.12). The inlet and outlet temperatures, $T_{c,i}$, $T_{c,o}$, of the heat exchanger were continuously measured. In addition to this, the volume flow rate of the cooling fluid, $\dot{V}_c$, the specific heat, $c_{p,c}$, and density, ρ_c , of the cooling fluid was determined.

$$\dot{Q}_c = \dot{m}_c (h_{c,o} - h_{c,i}) = \dot{m}_c c_{p,c} (T_{c,o} - T_{c,i}) = \rho_c \dot{V}_c c_{p,c} (T_{c,o} - T_{c,i}) \text{ [W]} \quad (4.12)$$

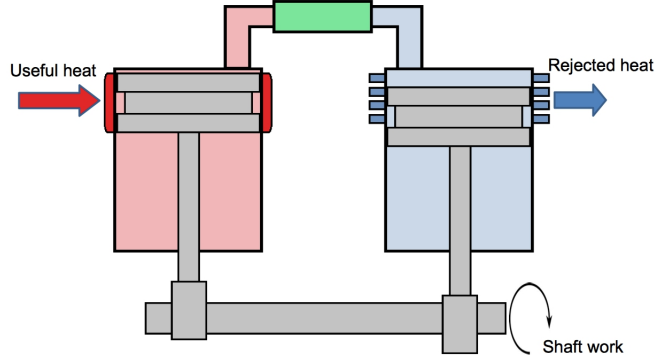


Figure 4.2: *The energy flows of a Stirling engine.*

4.2.2 Expansion and compression work

The time derivative of the shaft work, P_s , which is the net difference of the time derivatives of the expansion and the compression work, $\dot{W}_{exp}$, $\dot{W}_{comp}$, was evaluated by measuring the break moment, or torque, τ , and rotational speed, n_{rev} , Equation (4.13).

$$P_s = \dot{W}_{exp} - \dot{W}_{comp} = \omega\tau = \frac{2\pi n_{rev}\tau}{60} [\text{W}] \quad (4.13)$$

The indicated work was calculated by combining the work of the buffer gas channel and the cylinder volumes. In order to achieve this, pressure data, p , was extracted from locations within the engine and converted into a cycle average spanning the complete angular revolution. However, due to the low impact of the buffer gas channel on the indicated work, found in previous test series on the engine, pressure data was not extracted for this volume. Hence, pressure data was only obtained from the two cylinders. Further, the geometry of the engine coupled with the sinusoidal appearance of revolution determines the volume of each of these locations, V , at each point of revolution. With this obtained, the volumetric work per unit time can be calculated for the two pistons, Equation (4.14).

$$P_i = \sum_{j=1}^n \dot{W}_{i,j} = \sum_{j=1}^n \frac{\int_{V_0}^V p_j dV_j}{dt} [\text{W}] \quad (4.14)$$

The heat rejected through friction losses, $\dot{Q}_f$, can be determined by calculating the difference between the indicated power and the shaft power, Equation (4.15). However, most of the heat losses by friction was included in the heat rejected by the internal cooling system, $\dot{Q}_c$.

$$\dot{Q}_f = P_i - P_s [\text{W}] \quad (4.15)$$

4.2.3 Other losses

As was the case with the combustion device, a summation of all energy flows, Figure 4.2, will not completely equal the heat input. In a similar manner, all energy flows were subtracted from the useful heat extracted from the combustion device in order to determine the additional heat losses, $\dot{Q}_{o,s}$, Equation (4.16).

$$\dot{Q}_{o,s} = \dot{Q}_u - \dot{Q}_c - P_s [\text{W}] \quad (4.16)$$

4.2.4 Thermal efficiency

Thermal efficiency was calculated based on two perspectives. First, the system boundary was restricted to that of the Stirling engine, Figure 4.2. In this case, the thermal efficiency of the Stirling engine, η_s , was defined as the ratio of useful power generated, P_s , over the supplied heat, $\dot{Q}_u$, Equation (4.17).

$$\eta_s = \frac{P_s}{\dot{Q}_u} \quad (4.17)$$

Another interpretation is to consider the global system, including both Stirling engine and combustion device. Thus, the thermal efficiency of the entire system, η_{c+s} , was defined by ratio of the useful power generated, P_s , over the chemically bond energy in the fuel, $\dot{Q}_i$, Equation (4.18).

$$\eta_{c+s} = \frac{P_s}{\dot{Q}_i} \quad (4.18)$$

The Carnot efficiency is the maximum efficiency obtainable between two temperature reservoirs of finite temperature, Equation (4.19) [8]. The finite temperature of the heat source was approximated by the average temperature over the heater head. Similarly, the finite temperature of the heat sink was approximated by the average temperature over the external cooling stream.

$$\eta_C = 1 - \frac{T_C}{T_H} = 1 - \frac{(T_{c,e,i}, T_{c,e,o})_{mean}}{(T_{fg,phh}, T_{fg,ahh})_{mean}} \quad (4.19)$$

4.3 Interchangeability of the selected fuels

The Wobbe index, I_W , compares the calorific value, V_C , of a fuel with respect to the specific density, G_S , Equation (4.20) [29]. For a combustion device operating at the same pressure and valve setting, only small variations in heat output can be expected for two fuels having similar Wobbe indices [33]. Additional fuel properties required for calculation of the Wobbe Index is presented in Table 4.1.

$$I_W = \frac{V_C}{\sqrt{G_S}} \text{ [MJ m}^{-3}\text{]} \quad (4.20)$$

Table 4.1: Characteristics of selected fuels at atmospheric pressure, 298.15 [K].

	Liquid propane gas	Natural gas	Synthesis gas	Landfill gas	
LHV	46.34	38.94	4.96	5.57	[MJ kg ⁻¹]
	91.26	30.78	5.45	7.16	[MJ m ⁻³]
HHV	50.32	43.22	5.34	6.18	[MJ kg ⁻¹]
	99.13	34.16	5.86	7.95	[MJ m ⁻³]
ρ_f	1.80	0.72	1.01	1.18	[kg m ⁻³]
G_S	1.52	0.61	0.85	0.99	[]
$I_{W,LHV}$	73.93	39.36	5.91	7.18	[MJ m ⁻³]
$I_{W,HHV}$	80.30	43.68	6.36	7.97	[MJ m ⁻³]

The synthesis gas and the landfill gas had significantly lower Wobbe indices than the primarily propane and methane based fuels. Thus, a significant increase in fuel input was required, when using low calorific fuels, to produce the same thermal input compared to high calorific fuels. Hence, the fuel flow rate was adjusted accordingly in order to ensure that suitable temperatures were obtained within the Stirling engine. However, the combustion device sets an upper limit for both volume flow through the fuel nozzles, $\dot{V}_f$, and total volume flow through the combustion device, $\dot{V}_{fg}$.

Indirectly, even the latter constraint sets the limit of how much fuel the engine can accept. A large volume flow rate of fuel would result in a large volume flow of combustion products passing through the engine. Further, this constraints also affects the maximum allowable air to fuel ratio. However, the air to fuel ratio was also limited due to the fact that a high oxygen concentration will put the engine outside the MILD combustion region. Flue gas flows in the region of 60 m³ h⁻¹ have been recorded during a previous test schedule. Hence,

this constraint was regarded as less critical to engine operation. An additional constraint to engine operation was set by the minimum and maximum allowable engine charge pressure of 30 bar and 150 bar respectively.

Given the above constraints, a lower maximum thermal input was expected for the two low calorific fuels. Consequently, the engine charge pressure had to be reduced to make sure that the heat extraction from the heat accepting surface did not exceed the heat released by the fuel. The maximum heat input for the fuels with high calorific value was limited by the maximum allowable charge pressure, 150 bar, whereas the governing constraint for low calorific fuels was the limitation of the fuel flow through the preheater nozzles, as well as the minimum allowable charge pressure of 30 bar.

4.4 The engine installation

During engine testing, the gases were tested at different charge pressures. Further, the impact of changing the air to fuel ratio on engine performance was measured for all fuels at these charge pressures, thus resulting in additional data points. The complete test schedule is presented in Appendix A. No variations in engine speed were made since the engine should operate at 1500 rpm in order to facilitate electricity generation at 50 Hz. The shaft of the Stirling engine was mounted onto a break, which controlled the rotational speed of the engine. The fuel system was supplied by liquid propane gas during the warm-up period of the engine. Then, when the engine had reached sufficient temperature, a manual valve was used to switch fuels.

4.4.1 The air supply

A thermocouple adjacent to the air filter was used to determine the temperature at the air inlet. This temperature was also assumed for the fuel inlet. Further, this measurement also served as the thermodynamic reference temperature.

4.4.2 The combustion system

The fuel flow of liquid propane gas was measured by a flow rate monitor supplying 10 L of fuel per pulse. However, the fuel flow rate was expected to be higher than the maximum allowable flow rate for this monitor device during operation with low calorific fuels. Hence, a pulse monitor supplying 100 L of fuel per pulse was used for all fuels other than liquid propane gas. Further, a substantial pressure regulation was provided to reduce the pressure in the fuel bottles to pressures between 15 mbar to 200 mbar, where the former is associated with high calorific fuels and the latter with low calorific fuels respectively.

Three categories of temperature measurements were carried out in the flue gas chain of the combustion device. First, continuous temperature measurements were carried out prior to and after the heater head of the Stirling engine in order to determine the combustion efficiency. Second, the temperature difference over the flue gas side of the preheater was determined in order to calculate the effectiveness of the preheater. Last, the temperature of the exiting flue gas stream was used to determine the heat losses to the environment.

4.4.3 The cooling system

The volume flow rate of the cooling fluid was not measured continuously. This decision was based on the fact that for this particular engine, the flow rate of the coolant remains constant, regardless of heat input. However, it is important to note that the composition of ethylene glycol and water affects the fluid properties of the coolant, which in turn affects the volume flow rate. By using the same concentration as was used during previous tests, an equal volume concentration of water and ethylene glycol, the flow rate of the coolant was determined to be $1.5 \text{ m}^3 \text{ h}^{-1}$. Further, by converting the volume concentration into mass fraction, a density of 1061 kg m^{-3} and a specific heat of $3241 \text{ J kg}^{-1} \text{ K}^{-1}$ was determined. At last, thermocouples were mounted to the inlet and the outlet pipes of the heat exchanger of the internal cooling system for continuous temperature measurements.

4.4.4 The external cooling system

The constraint set on the external cooling system was based on the volume flow rate and the temperature of the process water. Throughout all test series, $5 \text{ m}^3 \text{ h}^{-1}$ of process water, at a temperature of 17°C , was continuously supplied to the heat exchanger. This decision was based on the fact that a high flow rate of

process water ensured a relatively constant temperature of the coolant in the internal cooling system, regardless of engine charge pressure. Thus, no major changes in Carnot efficiency occurred. As a result, a fair comparison of the system efficiencies at various engine charge pressure could be made.

4.4.5 Verification of thermal equilibrium

The thermocouple located in the exhaust pipe was used to ensure that the engine reached thermal equilibrium before any data was extracted. This decision was based on the fact that flue gas temperature in the exhaust pipe proved to have a slow reaction to system changes. Thermal equilibrium was confirmed when the arithmetic mean of the collected temperature data deviated less than one degree Kelvin over five minutes.

4.4.6 Limitations

The thermocouples used for temperature measurements was not rated for temperatures exceeding 1200 °C. Since temperature measurements immediately after combustion was considered impossible, measurements were instead attempted further downstream in the flow direction of the combustion device, at the point after which the flue gases has been diverted by the back wall. Still, the uncertainty of the integrity of the thermocouples located at this position was high, since the flue gases had not yet been allowed any substantial temperature reduction.

4.5 Equipment for data acquisition

Temperature measurements were obtained using type K thermocouples, connected to a cDAQ-9188 data acquisition unit by National Instruments. The thermocouples were tested in glass filled with a mixture of ice and water at 0 °C to confirm their accuracy. The same data acquisition unit was also used to collect continuous measurements of the break moment and the flow rate of fuel.

The cylinder pressure was obtained by pressure sensors of two different brands. A Kistler Model 701A Quartz pressure sensor was used in the cold cylinder, where temperatures are moderate. In the warm cylinder, where temperatures are elevated, a Kulite Model WCT-312M-210BARA pressure transducer was used. The Kulite pressure transducer and the Kistler pressure sensor was calibrated with a HBM RM4220 amplifier and a Kistler charge amplifier type 5011 respectively. Finally, these signals were acquired via IndiCom through an IndiMaster.

The engine was controlled by a MODBUS protocol, which extracts further pressure and temperature data. This protocol also acquires the speed of revolution and the air to fuel ratio of the engine. A list of all data to be collected as well as the data acquisition system used is presented in Appendix B.

5 Results

This chapter presents the results obtained from this study. First, notes are made about the completion of the test schedule. Second, the energy flows of the combustion device and the Stirling engine are characterised and quantified. In later sections, the air to fuel ratio's impact on the energy flows is presented. Last, comments about the overall stability of operation are made.

5.1 Completion of the test schedule

Tests were successfully completed for liquid propane gas and natural gas. A fuel pressure in the region of 15 mbar and 45 mbar was supplied to the engine while operated by liquid propane gas and natural gas respectively. The maximum charge pressure of the engine, while operated by natural gas, was set to 130 bar due to vibrations caused by the measurement equipment attached to the shaft. Further, the flow meter used to measure the fuel flow rate of liquid propane gas delivered inconsistent data at low flow rates. Thus, in order to ensure consistent data extraction, the lowest engine charge pressure was set to 55 bar while the engine was operated by liquid propane gas.

Fuel switching from liquid propane gas to a low calorific gaseous fuel during engine operation was attempted for an engine charge pressure of 30 bar, which is the lowest set point allowed by the engine. Engine operation was successfully maintained while using landfill gas as the fuel source. However, operation failed when the air to fuel ratio was taken below 1.93. At lower air to fuel ratios, the exhaust gas temperature rose to a level at which an error code was generated by the engine. This error code activation triggered an automatic procedure to shut down the engine.

The increased exhaust gas temperature was the result of the increased fuel flow rate associated with combustion of low calorific gaseous fuels. The high flow rate through the fuel nozzles, $5.5 \text{ m}^3 \text{ h}^{-1}$, generated a fuel stream of high velocity, which was not completely affiliated by the air stream. As a result, part of the fuel was transported backwards through the flue gas recirculation cavities and combusted just adjacent to the inlet of the preheater. Since the flow rate of the air supply remained virtually constant, the increased amount of energy prior to the preheater could not be transferred to the inlet air at much greater extent. Thus, the exhaust temperature increased, resulting in increased flue gas losses and eventually an error code activation.

Combustion of the synthesis gas was not attempted because of the hazard of carbon monoxide poisoning. It was determined that testing of this gaseous fuel could not be made since leakage of liquid propane gas from the system had been observed during earlier test series. This leakage was primarily noticed during the start up of the engine, thus presumably originating from the outflow of non combusted fuel in the exhaust pipe due to inadequate suction pressure.

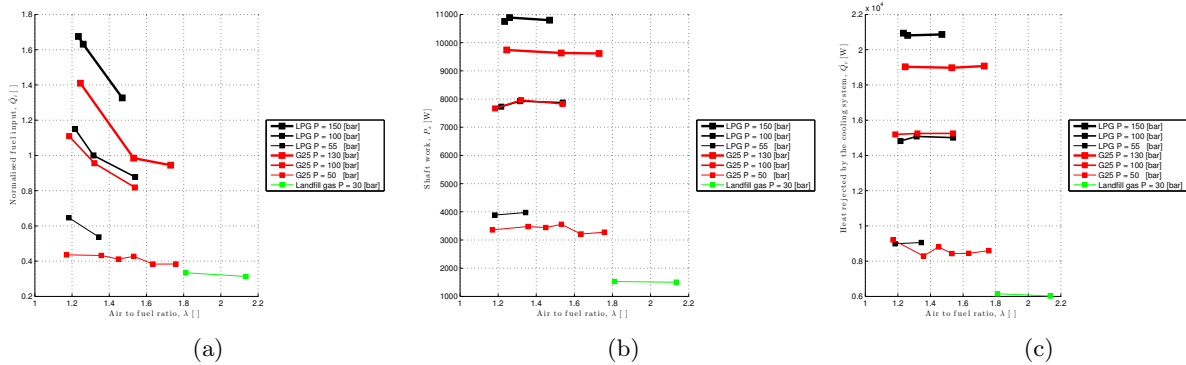


Figure 5.1: Fuel input(a), shaft power(b) and heat rejected by the cooling system(c) as a function of air to fuel ratio. Each curve represents a specific fuel and pressure setting. (a) is normalised with respect to the property obtained while operating the engine by liquid propane gas at a charge pressure of 100 bar and an air to fuel ratio of 1.32.

5.2 Energy flows

The heat input from the fuel, $\dot{Q}_i$, the heat rejected by the exhaust gases, $\dot{Q}_e$, the useful heat accepted by the heater head, $\dot{Q}_u$, the heat transfer through the preheater, $\dot{Q}_p$, the heat rejected by the cooling system, $\dot{Q}_c$, and the shaft power, P_s , were determined from the data extracted by the acquisition systems. All but one energy flow, the useful heat accepted by the heater head, $\dot{Q}_u$, where within expected ranges. The useful heat accepted by the heater head was much smaller than anticipated, indicating a value of only half the combined energy flows of the cooling system and the shaft of the Stirling engine. Clearly, more heat was transferred through the heater head compared to what was estimated by use of Equation (4.4). Hence, $\dot{Q}_u$, is from now on referred to as the heat transfer through the preheater originating from combustion occurring prior to the heater head only, or prompt combustion, $\dot{Q}_{comb,prompt}$. An explanation of the remaining part of the heat transfer through the heater head is attempted in Chapter 6. Consequently, given the uncertainties regarding the heat extracted by the heater head, the combustion efficiency, Equation (4.6), could not be determined.

The energy flows of the combustion device and the Stirling engine is presented by Sankey charts. The energy flows of the combustion device for all successful test points are presented in Appendices C, D and E.

5.3 Influence of the air to fuel ratio

There was a visible trend of reduced system efficiency for reduced air to fuel ratios. The reduced fuel input as a function of air to fuel ratio for all tested fuels is presented in Figure 5.1a. A decreasing fuel input at higher air to fuel ratios appeared for all fuels and pressure settings. However, the effect appeared to be less predominant at low charge pressures. At the same time as the air to fuel ratio was reduced, only small changes in shaft power were noted, Figure 5.1b. Further, these variations were not following any distinct pattern. In consistency with the low correlation between air to fuel ratio and shaft power, changes in the heat rejected by the cooling system are slight, Figure 5.1c.

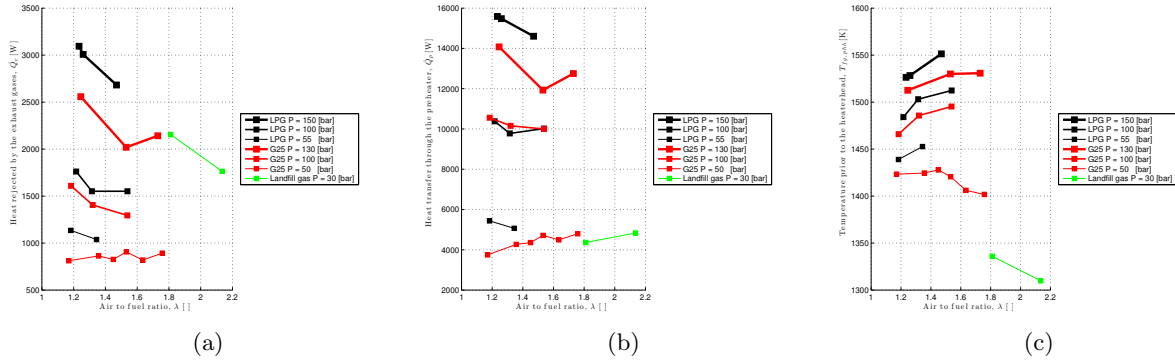


Figure 5.2: *Heat rejected by the exhaust gases(a), heat transferred through the preheater(b) and temperature prior to the heater head(c) as a function of air to fuel ratio. Each curve represents a specific fuel and pressure setting.*

The shaft power and the heat rejected by the cooling system remained virtually unchanged. Thus, the increased fuel input at low air to fuel ratios only contributed to increasing losses. The losses within the Stirling engine itself were assumed to be mostly unchanged, given that the Stirling engine remained at a constant speed and almost at the same power output, regardless of fuel input. Hence, most of the difference in overall performance caused by a varied air to fuel ratio was presumed to be due to changes in the performance of the combustion device, rather than changes in the Stirling engine.

In the combustion device, the heat rejected by the exhaust gases showed a decreasing trend for increasing air to fuel ratios at high engine pressures, Figure 5.2a. However, no consistent behaviour was discovered. Further, the heat transfer through the preheater, occurring within the system boundaries of the combustion device, seems to be unaffected by the air to fuel ratio, Figure 5.2b.

The change in temperature prior to the heater head of the Stirling engine with respect to air to fuel ratio is presented in Figure 5.2c. Most of the regulation of air to fuel ratio in a C9G Stirling engine is due to changes in the amount of fuel being supplied to the engine, rather than changes in the air supply. Hence, an increased

combustion temperature was discovered at the same time as the amount of fuel was decreased. However, when the engine was operated by natural gas at a low engine charge pressure, 50 bar, a decrease in the temperature prior to the heater head was noted at high air to fuel ratios.

The fraction of fuel combusted prior to the heater head increased at high air to fuel ratios, although the amount of fuel combusted prior to the heater head remained virtually unchanged. This characteristic held for all tested fuels and engine charge pressures, Figure 5.3a. The part of the fuel that was not combusted prior to the heater head is either combusted during its residence time adjacent to the heater head or after leaving the heater head. In the latter case, the released heat was either recirculated to the heater head, transferred to the combustion air through the preheater, transferred to the surroundings through the insulation or discharged through the exhaust pipe.

The impact of the air to fuel ratio on the system efficiency is demonstrated in Figure 5.3b. The trend of increasing system efficiency at high air to fuel ratio held for both liquid propane gas and natural gas. However, this correlation appeared to be smaller when the engine was operated by natural gas at low charge pressures.

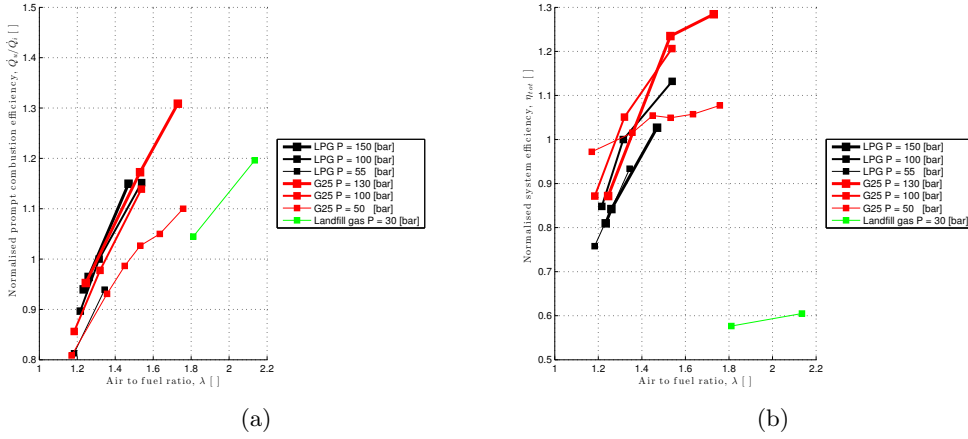


Figure 5.3: Ratio of the useful heat combusted prior to the heater head versus fuel input(a) and system efficiency(b) as a function of air to fuel ratio. Each curve represents a specific fuel and pressure setting. (a) and (b) are normalised with respect to the property obtained while operating the engine by liquid propane gas at a charge pressure of 100 bar and an air to fuel ratio of 1.32.

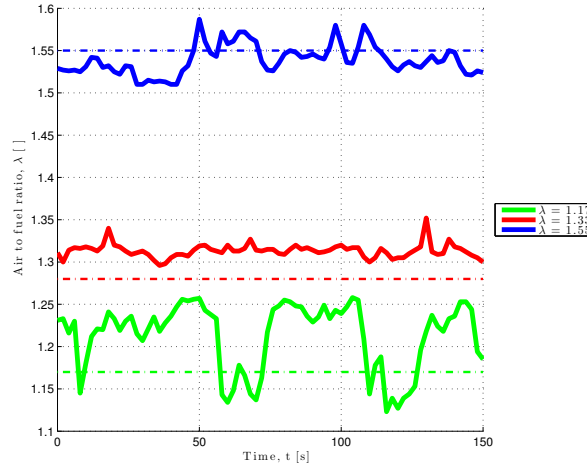


Figure 5.4: Observed air to fuel ratio while operated by liquid propane gas at a charge pressure of 100 bar at differing air to fuel ratios.

5.4 Stability of operation

The engine had difficulties to maintain a steady air to fuel ratio when forced to operate outside the predetermined air to fuel ratio schedule. The measured air to fuel ratio while operated at a charge pressure of 100 bar is presented in Figure 5.4. The air to fuel ratio set point of the smoothest curve is 1.33, which is also the predetermined air to fuel ratio at a charge pressure of 100 bar.

Considerable fluctuations in air to fuel ratio was observed when the air to fuel ratio set point was changed from its predetermined set point value. For example, the fluctuations increased when the engine was operated at a high air to fuel ratio, but the air to fuel ratio was primarily fluctuating around the set point value. However, at a low air to fuel ratio, the fluctuations were even more significant and the engine was primarily operating at a higher air to fuel ratio than the set point value.

The effectiveness of the preheater was very high for all test points, regardless of fuel type, charge pressure setting or air to fuel ratio. Calculations based on the temperature and the flow rate measurements suggest that the heat transfer through the preheater was well above 95 percent of the maximum possible heat transfer. This suggest a high robustness of the air preheating.

The Carnot efficiency exceeded 80 percent during engine operation by liquid propane gas and natural gas, Equation (4.19). However, due to the reduced combustion temperature, the Carnot efficiency was reduced by a few percent while operated by landfill gas. Further, the actual power produced by the engine shaft was in the region of 65 to 70 percent of the indicated power, Equation (4.14).

6 Discussion

This chapter attempts to answer and discuss the results obtained in this study. Topics include issues regarding the consistency of the extracted measurement data and comments about the influence of the air to fuel ratio on engine operation. Last, some issues regarding the stability of operation is targeted.

6.1 Consistency of the measurement data

The relationship between the prompt combustion, $\dot{Q}_{comb,prompt}$ and the total heat transfer through the heater head could not be fully established by this study. However, from combustion theory, it is suggested that some of the heat transfer through the heater head could have originated from combustion occurring adjacent to the heat transfer surfaces of the heater head. Thus, this amount of heat transfer could not be detected by the temperature measurement placed prior to the heater head, $T_{fg,phh}$. Additional heat transfer may have been caused by surface radiation from surfaces having a configuration factor of significant importance with respect to the heater head.

Additional contingencies were added as a result of the uncertainty of the various modes of heat transfer within the combustion device. Since the amount of fuel combusted prior to, and after, the heater head was unknown, the mass fractions, $\dot{m}_{fg}$, required for calculating the heat transfer through the heater head, Equation (4.4), remained undecided. In other words, since the assumption of complete combustion could not be verified, no assertive conclusions could be drawn from this data alone.

The thermocouples used in the combustion device may also have added to the misrepresentation of the heat transfer through the heater head. At the position prior to the heater head, the temperature was expected to be greater than the rated maximum temperature of the thermocouples. As a result, the temperature measurement at this position may have been flawed.

Added to this, inconsistency was observed regarding the energy flows of the entire system for some combinations of charge pressure and air to fuel ratio while the engine was operated by natural gas and landfill gas. The summation of the outgoing energy flows, namely the shaft power, the heat rejected by the cooling system and the heat rejected by the exhaust gases exceeded the fuel input by a minor fraction. Hence, the first law of thermodynamics appeared to be violated in these calculations.

One of the flow rate monitors may have been the reason for this inconsistency. There were two fuel flow rate monitors installed in the system, in which the first monitored the flow rate of liquid propane gas whereas the second monitored the flow rate of natural gas and landfill gas. If losses in the same ranges as for liquid propane gas source were to be expected, the fuel flow rate of natural gas and landfill gas should be about ten percent higher than the measured value. Hence, for natural gas, the true system efficiency may have been up to five percent lower than what is presented in Figure 5.3b. If this was the case, the system efficiencies were almost identical for liquid propane gas and natural gas at the same engine charge pressure. On the other hand, in the case in which landfill gas was used as the fuel source, the true system efficiency may have been just about one percent lower than what is presented in Figure 5.3b.

Another possible source of error could be due to failure to achieve thermal equilibrium in the engine prior to data extraction. If large quantities of energy was stored in the engine block during the start-up schedule fuelled by liquid propane gas, that energy would have been observed as heat rejection through the cooling system for a long period of time. Further, failure to reach thermal equilibrium would result in a slightly non representative exhaust gas temperature. In other words, if energy was stored in the system during the start-up schedule, this would cancel any calculative violations of the first law of thermodynamics.

Finally, the cause of the slight changes in shaft power at varied air to fuel ratios, Figure 5.1b, were mostly likely linked to a disability to maintain a fixed charge pressure level within the C9G Stirling engine. This pressure fluctuation originated from small sources of leakage from the mounting adapters of the installed pressure sensors, rather than changes in the air to fuel ratio. Pressure deviations up to six percent were noted before the engine automatically refilled the engine to the set point value.

6.2 Influence of the air to fuel ratio

The increased losses while operating the engine at low air to fuel ratio were possibly the result of incomplete combustion prior to and during the residence time at the heater head of the Stirling engine. This theory was

supported by the fact no changes in power output was observed despite the increased fuel input, Figure 5.1a-b. Further, the temperature prior to the heater head of the Stirling engine was increased following a reduction in fuel input, Figure 5.2c. This suggests an increased reaction rate, which would be a direct response to an increased oxygen concentration.

The increased reaction rate should allow for a larger percentage of the energy content in the fuel to be released prior to the heater head, which is in conjunction with what is presented in Figure 5.3a. Further, a strong correlation exists, suggesting that a larger heat release prior to the heater head supports a higher system efficiency, Figure 5.3b. The trend of increasing system efficiency at high air to fuel ratio held for both liquid propane gas and natural gas. However, this correlation appeared to be smaller when the engine was operated by natural gas at low charge pressures.

The same line of reasoning can be used to explain the decreasing temperature prior to the heater head for air to fuel ratios above 1.45 for an engine charge pressure of 50 bar, Figure 5.2c. If the temperature decreased, at a measurement point soon after the mixing of the oxidiser and the fuel, following the addition of more excess air, this suggest that less additional heat was released per unit of increase in excess air. In other words, further increasing the air to fuel ratio did no longer significantly aid combustion. Thus, the derivative of the heat release prior to the heater head as a function air to fuel ratio was decreasing at high air to fuel ratios, Figure 5.3a. Consequently, there was no significant gain in system efficiency at air to fuel ratios above 1.45, Figure 5.3b.

Last, soot was observed to have accumulated on surfaces located soon after the flue gases has passed through the heater head. As soot is a direct sign of flame quenching, this may also be proving the occurrence of incomplete combustion. However, the soot may also have been accumulated during the start-up period of the engine, when surface temperatures within the combustion device were far below the temperatures of normal operation.

Another reason for the decreased system efficiency at low air to fuel ratios, apart from incomplete combustion, was that a part of the fuel stream took a reversed flow direction through the flue gas recirculation cavities at high flow rates of fuel, Chapter 5.1. Since the correlation of decreased system efficiency held for all fuels and charge pressures, a reversed flow direction may have been observed even at relatively low flow rates, for example those required while the engine was operated with liquid propane gas as the fuel source. As a higher air to fuel ratio directly correlates to a lower flow rate of fuel, which translates into a lower flow velocity of fuel at the nozzle, a larger part of the fuel stream may have been engulfed by the air stream at high air to fuel ratios. Hence, the pattern of reduced flow direction may have been reduced at high flow rates of fuel.

6.3 Stability of operation

Caution should be taken while operating the engine by natural gas at high charge pressures. Increased charge pressures coincides with increased fuel flow rates. Thus, there is a risk of an error code activation for high exhaust gas temperature as the fuel flow rate approaches the region around $5.5 \text{ m}^3 \text{ h}^{-1}$. At this flow rate, during operation with landfill gas as the fuel source, the flow direction was reversed through the flue gas recirculation cavities, thus increasing the risk of severely increased exhaust gas temperatures. However, an error code for high exhaust gas temperature was not generated at a charge pressure of 130 bar during operation with natural gas as the fuel source.

The system barely managed to operate with landfill gas as the fuel source. By increasing the engine charge pressure from 30 bar to 32 bar, the air to fuel ratio rose above 2.3 which after a short period of time triggered an error code and an automatic termination of operation. Decreasing the air to fuel ratio below 1.93 also resulted in the generation of an error code due to a heavily increased exhaust gas temperature. Thus, successful operation was only maintained for an air to fuel ratio ranging between 1.93 to 2.3. Thus, at such an elevated air to fuel ratio, the system could not be considered as stable for operation. Again, it should be stressed that the content of carbon dioxide and nitrogen was shifted in the researched gas in order to facilitate compression to high pressures, Table 3.1.

In order to operate the engine by landfill gas, the lowest possible engine charge pressure, which translates into the lowest possible shaft power, as well as a high air to fuel ratio, was required. Even though synthesis gas was not tested as a fuel sources, it was determined to be very unlikely for the system to operate successfully with this fuel, which energy content was just two thirds of the energy content of landfill gas. Thus, in order to increase the fuel flow into the engine, an alternative placement of the fuel nozzles is needed in order to avoid a reversed flow direction through the flue gas recirculation cavities. Alternatively, a changed nozzle diameter

would be required to facilitate higher flow rates of fuel at the same time as the flow velocity is reduced.

It should also be mentioned that a higher charge pressure setting have been previously obtained while using landfill gas. However, these tests were set up by research engineers at Cleanergy. Thus, differences in the characteristics of the surrounding systems may be of great impact to engine operation. For example, a higher flow rate of the external cooling fluid was used at Chalmers, thus allowing the engine to reject more heat. Further, the exhaust gas suction pressure was inadequate, thus resulting in more resistance while injecting air and fuel into the engine. At last, different piping structures in the fuel supply chain may have resulted in a lower fuel injection pressure.

It appears that the engine operates at a higher system efficiency at an increased air to fuel ratio, Figure 5.3b. However, an increased air to fuel ratio results in higher temperatures within the combustion device, Figure 5.2, which in turns may contradict MILD combustion characteristics, Chapter 3.3.

7 Conclusions

To summarize, the following set of conclusion were drawn regarding operation of the C9G Stirling engine.

- Landfill gas with a methane content of 20 percent was successfully combusted in the C9G Stirling engine. However, operation could only be maintained at an engine charge pressure of 30 bar for air to fuel ratios between 1.9 and 2.3. At this setting, 1.5 kW of shaft power was generated. Further, the highest percentage of flue gas losses was recorded for this type of fuel.
- Liquid propane gas was successfully tested up to a charge pressure of 150 bar, generating 10.9 kW of shaft power. At this set point, the predetermined air to fuel ratio of 1.26 was used. The highest recorded system efficiency, while not varying the air to fuel ratio, was recorded at a charge pressure of 100 bar, while generating 7.9 kW of shaft power.
- Natural gas was successfully tested up to a charge pressure of 130 bar, generating 9.7 kW of shaft power. At this set point, the predetermined air to fuel ratio of 1.26 was used. The highest recorded system efficiency, while not varying the air to fuel ratio, was recorded at a charge pressure of 100 bar, generating 7.9 kW of shaft power.
- For all fuels and engine charge pressures, the system efficiency of the C9G Stirling engine increased at high air to fuel ratios.
- Part of the fuel stream took a reversed flow direction through the flue gas recirculation cavities at high flow rates of fuel. This part of the fuel was then combusted and directed through the preheater, thus resulting in an increased exhaust gas temperature. This, in a possible conjunction with incomplete combustion prior to the heat accepting surfaces of the Stirling engine, explained the decreased system efficiency at low air to fuel ratios.
- Fluctuations in air to fuel ratio were more intense when the engine was forced to operate outside the predetermined air to fuel ratio schedule. However, these irregularities may be reduced by changing the parameters governing the control system.

References

- [1] P. Basu. *Biomass Gasification and Pyrolysis: Practical Design and Theory*. Elsevier Science, 2010. ISBN: 9780080961620.
- [2] M. Berg, E. M. Johansson, and S. G. Järås. Catalytic combustion of low heating value gas mixtures: comparison between laboratory and pilot scale tests. *Catalysis Today* 59.1–2 (2000), 117–130. ISSN: 0920-5861.
- [3] G. Borman and K. Ragland. *Combustion engineering*. Mechanical engineering series. McGraw-Hill, 1998. ISBN: 9780070065673.
- [4] British Standards Institute Staff. *Tests Gases - Test Pressures - Appliance Categories EN 437:2003*. ICS 27.060.20; 91.140.40. B S I Standards, 2003. ISBN: 9780580679896.
- [5] Z. Cao and T. Zhu. Effects of CO₂ Dilution on Methane Ignition in Moderate or Intense Low-oxygen Dilution (MILD) Combustion: A Numerical Study. *Chinese Journal of Chemical Engineering* 20.4 (2012), 701–709. ISSN: 1004-9541.
- [6] S. Carlqvist. *Motorer med kontinuerlig förbränning*. Chalmers tekniska högskola, 1984.
- [7] A. Cavaliere and M. de Joannon. Mild Combustion. *Progress in Energy and Combustion Science* 30.4 (2004), 329–366. ISSN: 0360-1285.
- [8] Y. Çengel and M. Boles. *Thermodynamics: An Engineering Approach, SI Version*. McGraw-Hill Series in Mechanical Engineering. McGraw-Hill Higher Education, 2007. ISBN: 9780071257718.
- [9] R. Chandra et al. Performance evaluation of a constant speed {IC} engine on CNG, methane enriched biogas and biogas. *Applied Energy* 88.11 (2011), 3969–3977. ISSN: 0306-2619.
- [10] S. Chen and C. Zheng. Counterflow diffusion flame of hydrogen-enriched biogas under {MILD} oxy-fuel condition. *International Journal of Hydrogen Energy* 36.23 (2011), 15403–15413. ISSN: 0360-3199.
- [11] C.-H. Cheng and Y.-J. Yu. Combining dynamic and thermodynamic models for dynamic simulation of a beta-type Stirling engine with rhombic-drive mechanism. *Renewable Energy* 37.1 (2012), 161–173. ISSN: 0960-1481.
- [12] A. Colorado, B. Herrera, and A. Amell. Performance of a Flameless combustion furnace using biogas and natural gas. *Bioresource Technology* 101.7 (2010), 2443–2449. ISSN: 0960-8524.
- [13] H. Fogler. *Elements of chemical reaction engineering*. Prentice-Hall international series in the physical and chemical engineering sciences. Prentice-Hall, 1986. ISBN: 9780132634762.
- [14] Great Britain Environment Agency. *Guidance for Monitoring Trace Components in Landfill Gas*. Environment Agency, 2010.
- [15] J. Holman. *Heat Transfer*. McGraw-Hill series in mechanical engineering. McGraw Hill Higher Education, 2010. ISBN: 9780071267694.
- [16] S. E. Hosseini and M. A. Wahid. Biogas utilization: Experimental investigation on biogas flameless combustion in lab-scale furnace. *Energy Conversion and Management* 74 (2013), 426–432. ISSN: 0196-8904.
- [17] F. Incropera. *Fundamentals of heat and mass transfer*. Fundamentals of Heat and Mass Transfer v. 1. John Wiley, 2007. ISBN: 9780471457282.
- [18] International Energy Agency. *Technology Roadmap Bioenergy for Heat and Power*. Corlet, 2012.
- [19] L. Karlberg. Genombrott för Stirlingmotorn. *Ny Teknik* (2008).
- [20] L. Karlberg. Stirlingmotorn åter till Sverige. *Ny Teknik* (2008).
- [21] J. D. Martínez et al. Syngas production in downdraft biomass gasifiers and its application using internal combustion engines. *Renewable Energy* 38.1 (2012), 1–9. ISSN: 0960-1481.
- [22] W. Martini et al. *Stirling Engine Design Manual*. NASA contractor report. The Office, 1983. ISBN: 9781410216045.
- [23] A. Mudhoo. *Biogas Production: Pretreatment Methods in Anaerobic Digestion*. Wiley, 2012. ISBN: 9781118062852.
- [24] M. Muñoz et al. Low heating value gas on spark ignition engines. *Biomass and Bioenergy* 18.5 (2000), 431–439. ISSN: 0961-9534.
- [25] S. Purwono, P. Silveston, and R. Hudgins. “Catalytic Combustion of Low-Heating-Value Gases Using Flow Reversal”. *Progress in Catalysis Proceedings of the 12th Canadian Symposium on Catalysis*. Ed. by K. J. Smith and E. C. Sanford. Vol. 73. Studies in Surface Science and Catalysis. Elsevier, 1992, pp. 227–229.
- [26] A. Ross. *Stirling cycle engines*. Solar Engines, 1977. ISBN: 9990353905.

- [27] J. Rostrup-Nielsen and L. Christiansen. *Concepts in Syngas Manufacture*. Catalytic science series. Imperial College Press, 2011. ISBN: 9781848165687.
- [28] Royal Swedish Academy of Engineering Sciences. *El och värme från solen*. Tech. rep. 2003.
- [29] J. Russell and R. Cohn. *Wobbe Index*. Book on Demand, 2012. ISBN: 9785511207155.
- [30] M. Sánchez, F. Cadavid, and A. Amell. Experimental evaluation of a 20 kW oxygen enhanced self-regenerative burner operated in flameless combustion mode. *Applied Energy* 111 (2013), 240–246. ISSN: 0306-2619.
- [31] T. Seadi et al. *Biogas Handbook*. University of Southern Denmark Esbjerg, 2008. ISBN: 9788799296200.
- [32] G. Sridhar, P. Paul, and H. Mukunda. Biomass derived producer gas as a reciprocating engine fuel—an experimental analysis. *Biomass and Bioenergy* 21.1 (2001), 61–72. ISSN: 0961-9534.
- [33] L. Surhone, M. Timpledon, and S. Marseken. *Wobbe Index*. Betascript Publishing, 2010. ISBN: 9786130371678.
- [34] F. V. Tinaut et al. Method for predicting the performance of an internal combustion engine fuelled by producer gas and other low heating value gases. *Fuel Processing Technology* 87.2 (2006). [jce:title;International Congress on Energy and Environment Engineering and Management;ce:title](#), 135–142. ISSN: 0378-3820.
- [35] J. Twidell and A. Weir. *Renewable Energy Resources*. Taylor & Francis, 2006. ISBN: 9780419253303.
- [36] G. Walker. *Stirling engines*. Oxford science publications. Clarendon Press, 1980. ISBN: 9780198562092.
- [37] K. Wark. *Advanced Thermodynamics for Engineers*. Mac Graw-Hill series in mechanical engineering. McGraw-Hill Higher Education, 1995. ISBN: 9780070682924.
- [38] J. Warnatz, U. Maas, and R. Dibble. *Combustion: Physical and Chemical Fundamentals, Modeling and Simulation, Experiments, Pollutant Formation*. Springer, 2006. ISBN: 9783540259923.
- [39] R. Weber, J. P. Smart, and W. vd Kamp. On the (MILD) combustion of gaseous, liquid, and solid fuels in high temperature preheated air. *Proceedings of the Combustion Institute* 30.2 (2005), 2623–2629. ISSN: 1540-7489.
- [40] C. West. *Principles and applications of Stirling engines*. Van Nostrand Reinhold, 1986. ISBN: 9780442292737.
- [41] J. Wünning and J. Wünning. Flameless oxidation to reduce thermal no-formation. *Progress in Energy and Combustion Science* 23.1 (1997). cited By (since 1996)335, 81–94.
- [42] W. Yang and W. Blasiak. Numerical simulation of properties of a LPG flame with high-temperature air. *International Journal of Thermal Sciences* 44.10 (2005). cited By (since 1996)34, 973–985.
- [43] W. Yang and W. Blasiak. Numerical study of fuel temperature influence on single gas jet combustion in highly preheated and oxygen deficient air. *Energy* 30.2-4 SPEC. ISS. (2005). cited By (since 1996)42, 385–398.
- [44] W. Yang and W. Blasiak. Flame Entrainments Induced by a Turbulent Reacting Jet Using High-Temperature and Oxygen-Deficient Oxidizers. *Energy & Fuels* 19.4 (2005), 1473–1483.

A The test schedule

Table A.1: Liquid propane gas

	Test period	Pressure set. [bar]	Speed set. [rpm]	λ	Duration [min]	Data acquisition
	Warm-up	50	1500	-	45	None
1.1	Test	150	1500	1.23	15	Full transient
1.2	Test	150	1500	1.26	7	Full transient
1.3	Test	150	1500	1.47	7	Full transient
1.4	Test	100	1500	1.22	15	Full transient
1.5	Test	100	1500	1.32	7	Full transient
1.6	Test	100	1500	1.54	7	Full transient
1.7	Test	50	1500	1.18	15	Full transient
1.8	Test	50	1500	1.34	7	Full transient
1.9	Test	50	1500	1.38	7	Full transient

Table A.2: Natural gas

	Test period	Pressure set. [bar]	Speed set. [rpm]	λ	Duration [min]	Data acquisition
	Warm-up	50	1500	-	45	None
2.1	Test	130	1500	1.16	15	Full transient
2.2	Test	130	1500	1.25	7	Full transient
2.3	Test	130	1500	1.53	7	Full transient
2.10	Test	130	1500	1.73	7	Full transient
2.4	Test	100	1500	1.18	15	Full transient
2.5	Test	100	1500	1.32	7	Full transient
2.6	Test	100	1500	1.54	7	Full transient
2.7	Test	50	1500	1.17	15	Full transient
2.8	Test	50	1500	1.36	7	Full transient
2.14	Test	50	1500	1.45	7	Full transient
2.9	Test	50	1500	1.53	7	Full transient
2.13	Test	50	1500	1.64	7	Full transient
2.11	Test	50	1500	1.68	7	Full transient
2.12	Test	50	1500	1.76	7	Full transient

Table A.3: Synthesis gas

	Test period	Pressure set. [bar]	Speed set. [rpm]	λ	Duration [min]	Data acquisition
	Warm-up	50	1500	-	Not Attempted	None

Table A.4: Landfill gas

	Test period	Pressure set. [bar]	Speed set. [rpm]	λ	Duration [min]	Data acquisition
	Warm-up	50	1500	-	45	None
3.1	Test	30	1500	2.10	15	No equilibrium reached
3.2	Test	30	1500	1.93	15	No equilibrium reached
3.3	Test	30	1500	1.81	15	Error code
3.4	Test	30	1500	2.14	15	Full transient

B A list of measurement data and equipment

Table B.1: Measurement, unit, instrument and data acquisition system

Measurement	Unit	Data acquisition system
Coolant temperature at the engine inlet	°C	LabView
Coolant temperature at the engine outlet	°C	LabView
Flue gas temperature at the exhaust pipe	°C	LabView
Air temperature at ambient conditions	°C	LabView
Flue gas temperature prior to the heater head 1	°C	LabView
Flue gas temperature prior to the heater head 2	°C	LabView
Flue gas temperature after the heater head 1	°C	LabView
Flue gas temperature after the heater head 2	°C	LabView
Flue gas temperature at the preheater's inlet/flue gas recirculation's inlet 1	°C	LabView
Flue gas temperature at the preheater's inlet/flue gas recirculation's inlet 2	°C	LabView
Surface temperature of the heater head	°C	LabView
Working gas temperature at the regenerator	°C	LabView
Break moment	N m	LabView
Fuel flow rate	m ³ h ⁻¹	LabView
Shaft speed	min ⁻¹	MODBUS Protocol
Air to fuel ratio		MODBUS Protocol
Mean average working pressure	kPa	MODBUS Protocol
Working gas temperature at the front end of the heater head	°C	MODBUS Protocol
Working gas temperature at the center of the heater head	°C	MODBUS Protocol
Working gas temperature at the back end of the heater head	°C	MODBUS Protocol
Flue gas temperature FLOX 1	°C	MODBUS Protocol
Flue gas temperature FLOX 2	°C	MODBUS Protocol
Working gas temperature at cylinder top	°C	MODBUS Protocol
Pressure in heated cylinder	bar	IndiCom
Pressure in cooled cylinder	bar	IndiCom

