Application of jatropha oil and biogas in a dual fuel engine for rural electrification.

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Abstract

In this thesis, the technical feasibility of using jatropha oil and biogas for dual fuel generators is investigated. This technology could be used for electricity generation in rural areas in developing countries. The use of jatropha oil and biogas is considered a sustainable energy supply, when both fuels are produced locally. The local production of fuel and generation of electricity could result in economic development and poverty reduction.

In order to investigate the technical feasibility, a parameter study is performed, an experimental set-up is constructed and experiments are carried out. Three performance parameters are investigated: thermal efficiency, because it is a direct measure for fuel efficiency; volumetric efficiency, because it is a measure for power output; and air-excess ratio, because it influences particulate emissions and HC emissions. The parameter study is conducted, to predict the effect of dual fuel operation, by deriving expressions for the performance parameters. The experiments are carried out to assess the effect of dual fuel operation on performance and to find the operation limits (smoke limit and knock limit).

Experiments are carried out on a 12 kW diesel generator set. The jatropha oil that is used is pure oil. Biogas mainly consists of methane and carbon dioxide. Simulated, bottled, biogas of different quality is used (i.e. CH_4/CO_2 ratios). Pure methane is also tested as gaseous fuel. Gas is added to the inlet air with a venturi. The design of the venturi limited the gas flow; consequently the maximum heat release fraction of methane was 80% for pure methane and approximately 70% for biogas. Tests were performed at 6, 8 and 10 kW load.

The engine showed a thermal efficiency characteristic for pure jatropha oil operation, which is expected for a diesel generator. The characteristic for jatropha oil did not deviate from that of diesel. At full load, thermal efficiency is approximately 32%.

Under dual fuel operation, with biogas, at a 10 kW load; thermal efficiency did not deteriorate up to 70% heat release fraction of methane. At 6 kW and 8 kW, thermal efficiency decreases with increasing heat release fraction from methane. The decrease is in the order of 5% to 10% of the initial efficiency. For dual fuel operation with pure methane thermal efficiency even decreased at a 10 kW load.

Volumetric efficiency and air-excess ratio decreased under dual fuel operation as expected. The decrease did not result in a deficiency of oxygen. Enough oxygen was available to combust oil smoke-free.

The amount of carbon dioxide in the biogas did not influence performance parameters; the smoke limit was not reached. The engine runs without problems up to a heat release fraction of 60% methane. Between 60% and 70% irregularities are observed. The irregularities are attributed to light end-gas knock. Therefore, it is advised against, to replace a larger part of the fuel by biogas. A different engine design (i.e. different compression ratio) might be able to operate without problems with more biogas in the fuel mixture. For a better understanding more research is required; heat release should be measured with an in-cylinder pressure measurement; this would give a better insight into changes in thermal efficiency and operation limits.

It is possible to use a diesel engine to generate electricity with jatropha oil and biogas as fuels. The technologies used in the experimental set-up are low-tech and locally available; therefore it is considered an appropriate technology for the use in rural areas in developing countries.

Preface

This master thesis was written to graduate for the master Sustainable Energy Technology at Eindhoven University of Technology. The initial idea for the subject of research originated from a previous study that I performed in Tanzania. The late Prof. Kees Daey Ouwens, from the Fact Foundation, helped with transforming this idea into a research proposal.
In order to perform this research an experimental set-up had to be build. Due to the lack of space and technical assistance at the TU/e, I had to search for another location to perform experiments. Luckily, Edgar van de Laak, from Fontys pth Eindhoven, was generous enough to offer me a space to construct the set-up and perform tests. I would like to thank Edgar and the staff of the pth for the hospitability and for putting up with all the noise and smells that the engine produced.
The other hurdle was the budget, who was going to provide a generator and budget for fuels and tools? Again Prof. Kees Daey Ouwens helped me by promising a budget and supplying a generator from the Fact Foundation. After Prof. Kees Daey Ouwens passed away, his successor Erik Lysen realised the promise of the budget. I would like to thank everyone at the Fact Foundation for all the support they gave me.
During the build up of the experimental set-up Sander Baens helped me a lot, when the engine broke down again. Thanks a lot.
I am glad and thankful that Carlo Luijten was willing to supervise me with enthusiasm. Therefore, I was able to perform a research that appealed to me. Furthermore I would like thank Ies Biemond for supervising me until January; unfortunately he left the TU/e and could not remain my second supervisor. Marc Willekens, advised me during the build up of the experiment, I would like to thank him for helping me. Finally, I would like to thank Prof. Philip de Goey, for being the coordinator of the graduation commission; Johanna Myrzik and Erik Lysen for being a member of the graduation commission.

Evie Kerkhof
Eindhoven, June 2008

Nomenclature

Roman

$\frac{A}{F}$	air-fuel ratio	[-]
B	bore	[m]
CN	cetane number	[-]
CR	compression ratio	[-]
i	current	[A}
LHV	lower heating value	[MJ kg^{-1}]
m	mass	[kg]
M	molar mass	[kg kmol^{-1}]
n	number of mols	[mol]
N	rotational speed	[RPM]
n_R	number of rotations per cycle	[-]
ON	octane number	[-]
P	Power output	[W]
p	pressure	[kPa]
S	stroke	[m]
$\bar{S}_p$	mean piston speed	[m s^{-1}]
T	temperature	[K}
V	volume	[m^3]
V	voltage	[V]
X	molar fraction	[-]

Greek

α	molar fraction of CH_4 in biogas	[-]
β	molar fraction of oil in total fuel	[-]
ε	fraction of heat release from oil	[-]
η	efficiency	[-]
θ	mass fraction of oil in total fuel	[-]
λ	air-excess ratio	[-]
ξ	mass fraction of air in the intake mixture	[-]
σ	mass fraction of CH_4 in biogas	[-]
τ	induction time	[s]
ψ	molar fraction of air in the intake mixture	[-]

Subscript

act	actual
b	biogas
d	displacement
dual	dual fuel
ref	reference
stoich	stoichiometric
TDC	top dead centre
th	thermal
v	volumetric

Table of contents

1 Introduction

This chapter provides the motivation for this thesis work; it gives the research goal, question and the methodology of the research. An introduction into jatropha oil and biogas production is provided. The structure of the thesis is explained.

1.1 Motivation

In most developing countries, many people do not have access to modern energy sources. Energy use in these countries is characterised by a high use of traditional resources like fuel wood and charcoal; access to electricity is very scarce especially in rural areas [1;2]. These areas are usually very remote and very poor, making it highly unlikely that they will ever be connected to the national grid due to financial constraints. Access to electricity does not give a guarantee for economic development and poverty reduction, but providing people with electricity is associated with economic development and improvement in living conditions for the rural poor [2;3]. The other way around, when a region develops economically, the demand for modern energy carriers like electricity will increase [3].

Rural electrification increases local and eventually global energy demand which contributes to global warming, depletion of fossil fuels and can have a negative effect on the local environment. Therefore it is very important that when electricity is introduced it is done in a sustainable and environmentally sound way and it should be economically viable. Making sustainable energy available in rural areas in developing countries could lead to improved living conditions and improvement of the local environment. Therefore, an option of sustainable electricity generation that is appropriate for rural areas in low-income countries is investigated in this thesis. This is the use of vegetable oil and biogas in a dual fuel diesel engine-generator set. Vegetable oil is supposed to be locally produced jatropha oil. Biogas (a mixture of mainly methane and carbon dioxide) will be produced from digesting the press cake, that is a waste product from the jatropha oil extraction process. For the combustion of biogas a gas engine could be used, where no additional fuel is required. It is decided to use a diesel engine, because the use of stationary diesel engines is already common in most rural areas in developing countries. Therefore the engines and spare parts are locally available. Furthermore the diesel engines are robust and require little maintenance.

Previous study showed that this is an economically viable option [4]. Whether the use of bio-fuel is sustainable is debatable, but in this case the use of vegetable oil is considered sustainable, because the oil is produced and processed locally and no artificial fertilisers are used. Biogas is produced from the wastes from jatropha oil production and possibly other agricultural wastes; in this way, the energy content of the press cake is used without destroying the nutrients. The nutrients taken from the soil are still contained in the slurry that is left after digestion. The slurry can be used as organic fertiliser for the jatropha plantation.

An additional benefit of socio-economic nature is that no diesel fuel is required for electricity generation; therefore no dependence on other countries will exist. The equipment that is required for the production of electricity is mostly locally available and easy to operate and maintain, making it an appropriate technology [4]. These two benefits could result in an increase of economic activity due to the rise of local businesses and industry concerning the production of oil, biogas and electricity. This could lead to economic development and improved living conditions.

1.2 Objective and research questions

Studies in the past showed that it is possible to use a gaseous fuel in diesel engines as long as there is a pilot injection of a liquid fuel; hence the name dual fuel diesel engine. These studies use diesel as a liquid fuel and methane or natural gas as a gaseous fuel [5-7]. Two more recent studies showed that biogas which also contains carbon dioxide, can be used in a dual fuel diesel engine with diesel as a liquid fuel [8;9]. No research exists to the use of pure vegetable oil as a liquid fuel in the diesel engine. Therefore the objective of the thesis is:

"To investigate the technical feasibility to use pure jatropha oil and biogas in a dual fuel diesel generator set."

In order to reach this objective, a main question and several sub questions are answered in this thesis work. In order to answer the main research question six sub-questions are formulated.

Main question:

"What is the effect of running a 1 cylinder diesel genset on dual fuel mode with Jatropha oil and biogas on engine performance and what are the limiting factors?"

Sub-questions:

"What is the effect on engine performance for dual fuel operation, when predicted theoretically?"

"What is the effect on engine performance, when changing oil/biogas ratio, compared to using pure oil or diesel?"

"What is the effect on engine performance, when changing CO_2 /CH_4 ratio in the biogas?"

"What is the maximum possible CO_2 content?"

"What is the maximum possible heat release from biogas, as a percentage of total heat release?"

"What are the limiting factors to dual fuel operation?"

1.3 Methodology

In order to answer these research questions a literature study is performed and a model is formulated that predicts the effect of dual fuel operation on engine performance. Engine performance parameters that are considered are: thermal efficiency, volumetric efficiency and air-excess ratio. Thermal efficiency is investigated because it is a direct measure for fuel efficiency. Volumetric efficiency is a measure for power output and air-excess ratio could influence particulate emissions and HC emissions. Another, very practical reason is that these could be tested with the available equipment. Emissions are not measured in this study because there was no possibility of measuring these. An extensive description of the performance parameters is provided in chapter 2.

To validate the predictions of chapter 2 an experimental set-up is built and experiments are carried out. The experimental set-up contains a of small diesel generator set, which can run on jatropha oil, and to which simulated biogas is added with the intake air. In order to answer the research question, fuel consumption, air consumption, and oxygen levels in the exhaust gas are measured at differing oil-gas ratios. This is done for different methane – carbon dioxide ratios inside the biogas and at variable electrical loads.

A reference measurement for pure oil and pure diesel operation will be performed. Tests for dual fuel operation are performed for both biogas and pure methane.

1.4 Jatropha Curcas plants and oil

Jatropha Curcas Linnaeus or physic nut is a shrub that originates from Central and South America and belongs to the euphorbia family; in this report it will be referred to as Jatropha. It is cultivated in Central and South America, South-East Asia, India and Africa. It is a drought resistant plant that can grow in arid and semi arid areas in the tropics. The shrub or small tree can grow up to 6 meters height and the roots can go 7 meters deep to reach ground water. In the dry season the plant sheds its leaves. Depending on the water supply young plants start producing seeds after half a year to three years. The fruits are green and round when fresh and are about 4 cm long; the fruit contains two or three black seeds [10;11]. Figure 1.1 andFigure 1.2 show a jatropha plant and jatropha fruits and seeds.

Figure 1.1 Jatropha shrub[1]

Figure 1.2 Jatropha fruits and seeds

[1] Photo in 1.1 is taken in Tanzania by myself; photo in 1.2 is taken in Tanzania by Janske van Eijck.

Traditionally jatropha is used as a life fence. The plant and the seeds are toxic; therefore wandering cows will not eat it and come inside the fenced area. The toxicity of the plant and seeds is mainly due to the presence of phorbol esters and curcine [12]. The leaves, stems, roots and seeds are used in traditional medicine. The oil is traditionally used in medicine, as lighting fuel and to produce soap [11].
Jatropha can be productive up to fifty years. Yields are difficult to predict since it is still a wild plant, it is not a crop like maize or rice. Seed production is heavily dependent on water supply, it ranges from 0.4 to 12 t/ha/year in plantations and 0.8 to 1 kg of seeds per meter fence [10]. Realistic values for not very favourable conditions are between 3 and 5 tonnes per hectare per year [11]. Appendix K gives an estimation of the required amount of land for the production of oil and biogas to feed a 12 kW generator.
From the seeds, oil can be gained. Depending on the variety, peeled or dehusked seeds contain 43-59% oil [13]. In practice 1 kg of seeds gives 200 to 300 ml of oil depending on the quality of the pressing process. The rest of the weight of the seeds is left as press cake [14].
At this moment the use of bio-fuels is debated because of the competition with food production. For this case it is unlikely that jatropha oil production will compete with food production for two reasons. Jatropha is a non-edible plant that can grow on marginal soils, this means that land could be used that is not very suitable for food production. Also, jatropha is suitable for intercropping. The second reason is that this kind of jatropha oil production is small-scale and for local use only. Therefore it is only able to endanger local food production. To prevent this from happening it should be assured on forehand that local food production is not endangered by implementing this technology. Local food production is not endangered when marginalised soils are used that are not suitable for food production but are suitable for jatropha production. Also the use of intercropping can prevent endangerment of food production; with intercropping food and jatropha is produced on the same land. Finally making live fences with jatropha does not endanger food production.

One of the drawbacks of using pure vegetable oil in an engine is that vegetable oil has a higher viscosity than diesel; this can result in clogging of filters and nozzles especially in cold climates. Jatropha oil can be esterified like any other vegetable oil to give the oil characteristics that are favourable for operation in a diesel engine. In this study pure jatropha oil is used. This is done because the stationary engine that is used is capable of operating on pure vegetable oil even in cold climates. A more important reason is that the esterification of vegetable oil requires a chemical lab and the availability of chemicals like methanol or ethanol and potassium hydroxide. Esterification of the oil will increase the production costs and will make it a more difficult process; this makes it a less appropriate technology for the use in a rural area in a low-income country, compared to using pure vegetable oil. Since it is not necessary to esterify the oil, and it only complicates matters, and increases costs it is decided to use pure vegetable oil. Appendix K gives an estimation of the required jatropha plantation size. Fuel properties of pure jatropha oil are discussed in chapter 3.

1.5 Biogas

Jatropha press cake is a valuable waste product since it contains the nutrients taken from the soil, and it still contains energy. Press cake is optimally used when both energy and nutrients are used. Nutrients need to be preserved in order to be used as organic fertiliser for the jatropha plantation. The use of bio-oil could only be considered sustainable when no artificial fertiliser is used since artificial fertiliser requires huge amounts of energy to produce [15].

During digestion bacteria break down the hydrocarbons and transform this to mainly methane and CO_2, so called biogas. The main advantage of making biogas out of press cake is that both the energy content and the fertilising properties of the press cake are used. The slurry that is left after digesting contains nutrients and can be used as organic fertiliser.

Designs for biogas installations that are most frequently used in developing countries are fixed dome and floating dome. Appendix A gives an explanation of the operation principles of these two types of digesters. Press cake of jatropha seeds is a well suited organic material for digestion. Jatropha press cake has higher methane yields when digested than cow dung, which is usually used to digest, but it has lower yields than other energy crops like rapeseed or sunflower seeds. Properties of biogas are discussed in chapter 3.

1.6 Report lay-out

Chapter 2 gives the theoretical background of the thesis. It provides a description of dual fuel operation and gives a brief literature overview. The reaction equation for dual fuel operation is set up. Expressions for the three dual fuel performance parameters are derived. An overview of the input and output variable is provided. Finally the operation limits, smoke limit and knock limit, are theoretically explored.

Chapter 3 describes the experiments that are conducted. The properties of the engine-generator set and fuels are discussed. An overview of the experimental set-up is provided for both reference tests and dual fuel tests. Also the experimental procedures of both tests are described. How the measurements are conducted and how measurement data is processed into useful data concerning the performance parameters and measurements inaccuracy is described in the final section of chapter 3.

Chapter 4 describes the results from the experiments. First the reference tests for the three performance parameters are provided. The next section provides dual fuel results for the performance parameters. Finally operation limits are discussed.

Chapter 5 consists of conclusion and recommendation.

2 Theory

This chapter describes the theoretical background of the dual fuel engine and will serve guideline to interpret experimental outcomes.
Section 2.1 gives a brief description of dual fuel operation in a compression ignition engine in comparison to pure diesel operation. A schematic overview is given to illustrate how the engine works in dual fuel mode and how the gas is added. A more elaborate overview of the experimental set-up is provided in chapter 3.
In order to find out what the effect of dual fuel operation is on engine performance, a parameter study is performed in section 2.2. Expressions will be derived for important operating parameters and operating limits under dual fuel operation will be investigated. Engine performance parameters that are considered are: thermal efficiency, volumetric efficiency and air-excess ratio. Thermal efficiency is investigated because it is a direct measure for fuel efficiency. Volumetric efficiency is a measure for power output and air-excess ratio could influence particulate emissions and HC emissions. These parameters are most influenced by dual fuel operation. Due to the addition of biogas a certain amount of air is replaced by methane and carbon dioxide; this gives different definitions for relative air-fuel ratio and volumetric efficiency compared to diesel or vegetable oil operation. Expressions for air-excess ratio (λ) and volumetric efficiency under dual fuel operation are derived. Section 2.2.3 provides an overview of the input and output variables. Section 2.2.4 provides a description of smoke limit and two possibilities of change in smoke limit due to dual fuel operation.
In section 2.3 auto-ignition behaviour is investigated since the addition of biogas can increase the likelihood of auto-ignition. Auto-ignition can result in knock; a model for the likelihood of knock will be described.

2.1 Dual fuel diesel engine

A standard direct injection diesel engine is adapted to dual fuel use; this means that a gaseous fuel is premixed with the intake air.
In a standard compression ignition engine or diesel engine liquid fuel (diesel fuel or vegetable oil) is injected into the cylinder and it auto-ignites by the existing in-cylinder temperature and pressure after compression. In a dual fuel diesel engine both gas and liquid fuel are used. Biogas will not auto-ignite at the temperature and pressure that exists inside the cylinder after compression. Therefore, a pilot injection of liquid fuel is required to start the ignition of the gas [5-7;16]. Usually diesel fuel is used as a pilot injection but in this study pure vegetable oil of jatropha is used. Liquid fuel is injected into the cylinder as usual. Due to the high compression ratio in a diesel engine not every gaseous fuel is suitable for dual fuel operation. The high auto-ignition temperature of methane makes it a very suitable fuel for dual fuel diesel engines. A gaseous fuel at four different fractions of methane and carbon dioxide is used, namely 70-30, 60-40, 50-50 and 40-60 in volume percentages. This simulates biogas of different qualities. These qualities are used because 60% to 70 % methane fraction is very common for biogas; 50% is less common and not so good. A fraction of only 40% methane is considered bad quality biogas but it is tested to find out if the large fraction of carbon dioxide will influence performance.

During the inlet stroke both air and gas enter the cylinder and are compressed. At the end of the compression stroke a liquid fuel (diesel or vegetable oil) is injected in order to start ignition. The diesel engine itself regulates the amount of diesel it needs to inject to deliver sufficient energy for the imposed electrical output. It regulates rotational speed and output torque with a closed–loop control. The dual fuel engine combines principles from the compression ignition engine with principles of the spark ignition engine. The injected liquid fuel is ignited due to compression but the other part of the fuel, the gas, is compressed together with air like in a spark ignition engine. The pilot injection of liquid fuel to start ignition can be compared to the spark in a spark ignition engine.

Several studies on dual fuel diesel engines with diesel and methane or natural gas show that methane can replace diesel up to 90% of the heat release. Two studies on dual fuel engines with biogas and diesel show that a pilot injection that covers at least 10% to 20% of the total heat release is required [9;16]; at least 60% of the diesel fuel can be replaced by biogas without knock [16]. Another study shows that optimum conditions exist with 30% pilot injection [17].

Studies performed in the past are not conclusive on the effect of dual fuel operation on engine performance; Karim claims dual fuel operation results in higher output; better specific energy consumption; superior emissions and quieter and smoother operation [5]. While Duc gives a brief overview of several studies that are not conclusive; some report increase, some decrease and some report no difference in engine performance between diesel operation and dual fuel operation; the experiments carried out by Duc show no deterioration in engine performance [8]. All studies agree that under low-load condition engine performance deteriorates compared to normal diesel operation. Under low-load conditions the mixture is too lean, therefore it burns too slow which contributes to a low thermal efficiency; at higher loads the mixture burns fast enough for a complete and sufficiently fast combustion [17].

Selim showed that for the dual fuel engine with diesel and methane, combustion noise is always higher for dual fuel operation than for diesel operation [18]. Combustion noise is measured as pressure rise per degree crank angle. This means that in the dual fuel case pressure rise per degree crank angle is significantly higher than in the diesel case. The addition of more gas results in more combustion noise. Increase in the mass of methane increases the ignition delay of the diesel fuel. An increase in ignition delay means that the diesel will ignite later and will therefore burn the gaseous fuel at a higher rate of pressure rise, resulting in more combustion noise. Nielssen showed that the introduction of 2% methane in the intake air resulted in a doubled ignition delay [19]. In section 2.3 the causes of the increase in ignition delay are discussed.

The use of biogas introduces carbon dioxide which can influence combustion parameters. When it is assumed that carbon dioxide is an inert gas which does not disassociate significantly, burning velocity can be affected by it. This could result in incomplete combustion, which results in unwanted emissions and deteriorated engine performance [20]. Bari showed that up to 40% carbon dioxide in the biogas mixture did not deteriorate engine performance [20]. The effect of carbon dioxide on engine performance will be further investigated experimentally in this study.

A schematic overview of the dual fuel engine that is used is provided in Figure 2.1. The engine used for the experiment is adapted for the use of straight vegetable oil with a two tank system because vegetable oil is more viscous than diesel. The engine starts on diesel; cooling water preheats the vegetable oil, when the vegetable oil has reached a low enough viscosity it switches to vegetable oil. When the engine is running, gas is added to the inlet air through a venturi. A venturi is a constriction in the airway; due to the smaller tube diameter pressure drops inside the constriction. Holes inside the constriction allow the gas to flow to the air that has a lower pressure locally. An adjustable control valve regulates the amount of gas that can enter into the venturi. Mixing gas with a venturi is a slightly old-fashioned method; it is also possible to inject the gas directly into the cylinder or air-flow with an electronic fuel injector. In this study a venturi is used since this is an easy and low-cost technology which is easy to operate and maintain and therefore suitable for use in a rural area in a low-income country. The amount of gas passing through the venturi is regulated with a control valve.

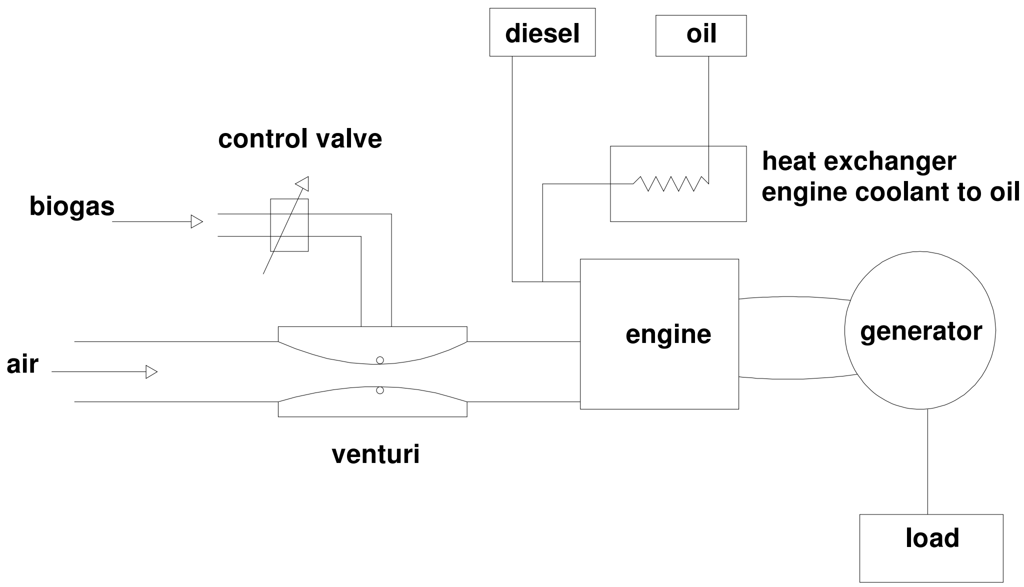

Figure 2.1 Gas addition with venturi

2.2 Parameter study

In order to predict the effect of dual fuel operation on engine performance, a parameter study is performed. Engine performance is characterised in this study with three parameters: (1) thermal efficiency, (2) volumetric efficiency, (3) air-excess ratio. Thermal efficiency is directly important when this system would be used to generate electricity. Air-excess ratio and volumetric efficiency are important parameters that influence thermal efficiency and operation limits, therefore these have indirect influence.

Thermal efficiency is defined as the ratio between power output and power input by means of heat release of the fuels. Volumetric efficiency is the ratio between the volume of air taken into the cylinder and the displacement volume of the cylinder. Air-excess ratio is the ratio between actual air-fuel ratio and the stoichiometric air fuel ratio. A derivation of the expressions for volumetric efficiency and air-excess ratio under dual fuel conditions is provided in this section.

Dual fuel operation influences volumetric efficiency and air-excess ratio and subsequently power output and engine efficiency because biogas replaces part of the air in the inlet mixture.

This section will first provide reaction equations for dual fuel operation. Then, expressions for the engine performance parameters are derived and the effect of dual fuel operation on these parameters is predicted. An expression for air-excess ratio is derived from the reaction equations and standard combustion relations. An expression for dual fuel volumetric efficiency is derived. At standard diesel operation volumetric efficiency is constant at a fixed rotational speed. During dual fuel use, volumetric efficiency changes due to the replacement of air by biogas in the intake mixture. An overview of all described parameters is provided to give a better insight in input and output variables and connections between them. Finally smoke limit for dual fuel combustion is discussed; two hypotheses are postulated concerning dual fuel smoke limit.

2.2.1 Reaction equations

In order to describe the combustion process, separate reaction equations for jatropha oil and biogas and an overall reaction equation are obtained. The overall reaction equation is composed of the separate reaction equations for jatropha oil and biogas. Three ratios are essential to formulate the reaction equations: (1) fraction of methane in the biogas, (2) fraction of oil in the total fuel, (3) fraction of incoming air in the total intake mixture. These ratios and the reaction equations are explained in this section. All three ratios influence the dual fuel combustion process and are therefore included in the overall reaction equations. All ratios are defined in both mass and molar units, which seems confusing but this is done in order to make the accounting easier.

Fraction of methane in biogas

α= molar fraction of CH4 in biogas

σ= mass fraction of CH4 in biogas

The fraction of methane in the biogas is provided as a volumetric percentage by the supplier, therefore α will be the ratio that is given during an experiment.

$$\alpha = \frac{n_{CH4}}{n_{CH4} + n_{CO2}} = \frac{m_{CH4}}{m_{CH4} + \frac{M_{CH4}}{M_{CO2}} m_{CO2}} = \frac{1}{1 + \frac{16}{44}\frac{m_{CO2}}{m_{CH4}}}$$

$$\alpha(\sigma) = \frac{1}{1 + \frac{16}{44}\frac{(1-\sigma)}{\sigma}} = \frac{44 \cdot \sigma}{28\sigma + 16} \qquad \textbf{2.1}$$

$$\sigma = \frac{m_{CH4}}{m_b} = \frac{m_{CH4}}{m_{CH4} + m_{CO2}} = \frac{n_{CH4}}{n_{CH4} + \frac{M_{CO2}}{M_{CH4}} n_{CO2}} = \frac{1}{1 + \frac{44}{16}\frac{n_{CO2}}{n_{CH4}}}$$

$$\sigma(\alpha) = \frac{\frac{16}{44}\alpha}{1 - \frac{28}{44}\alpha} = \frac{\alpha}{2.75 - 1.75\alpha} \qquad \textbf{2.2}$$

$$m_{CH4} = \sigma \cdot m_b \qquad \textbf{2.3}$$

$$m_{CO2} = (1-\sigma) m_b \qquad \textbf{2.4}$$

Fraction of oil in fuel mixture

β is molar fraction of oil in total fuel

θ is mass fraction of oil in total fuel.

For these fractions the average molar mass of biogas is important; this is defined in equation 2.5. The molar mass of jatropha oil is obtained from the fatty acid composition as found in literature. Appendix B describes this derivation.

$$M_b = \alpha M_{CH4} + (1-\alpha) M_{CO2} = \alpha 16 + (1-\alpha) 44 = 44 - 28\alpha \qquad \textbf{2.5}$$

$$\beta(\theta)=\frac{n_{oil}}{n_{oil}+n_b}=\frac{m_{oil}}{m_{oil}+\frac{M_{oil}}{M_b}m_b}=\frac{M_b\theta}{M_b\theta+M_{oil}(1-\theta)}$$
$$=\frac{\theta(44-28\alpha)}{\theta(44-28\alpha)+869(1-\theta)} \qquad \textbf{2.6}$$

$$\theta(\beta)=\frac{m_{oil}}{m_{oil}+m_b}=\frac{n_{oil}}{n_{oil}+\frac{M_b}{M_{oil}}n_b}=\frac{1}{1+\frac{M_b}{M_{oil}}\left(\frac{1}{\beta}-1\right)} \qquad \textbf{2.7}$$

$$m_{oil}=\frac{m_b\cdot\theta}{1-\theta}=\frac{\frac{m_{CH4}}{\sigma}\cdot\theta}{1-\theta} \qquad \textbf{2.8}$$

The mass fraction of oil of total fuel consumption (θ) is now defined. The amount of oil that can be replaced by gas is expressed as ε; this gives the fraction of heat release of the oil from the total heat release of both oil and gas. This is a different ratio since the heating values in MJ/kg of biogas and methane are different.

$$\varepsilon=\frac{E_{oil}}{E_{oil}+E_b}=\frac{m_{oil}\cdot LHV_{oil}}{m_{oil}\cdot LHV_{oil}+m_b\cdot LHV_b}=\frac{m_{oil}\cdot LHV_{oil}}{m_{oil}\cdot LHV_{oil}+m_{CH4}\cdot LHV_{CH4}} \qquad \textbf{2.9}$$

Using the mass of oil as expressed in equation
2.8 results in an expression for ε as function of θ as shown in equation 2.10.

$$\varepsilon(\theta,\sigma)=\frac{\frac{\theta\cdot LHV_{oil}}{1-\theta}}{\frac{\theta\cdot LHV_{oil}}{1-\theta}+\sigma\cdot LHV_{CH4}} \qquad \textbf{2.10}$$

Fraction of air in intake mixture

Ψ is molar fraction of air in the intake gas mixture

ξ is mass fraction of air in the intake gas mixture.

$$\psi(\xi) = \frac{n_{air}}{n_{air} + n_b} = \frac{1}{1 + \frac{M_{air}}{M_b}\frac{(1-\xi)}{\xi}} \qquad \textbf{2.11}$$

$$\xi(\psi) = \frac{m_{air}}{m_{air} + m_b} = \frac{m_{air}}{m_{air} + m_{CH4} + m_{CO2}} = \frac{1}{1 + \frac{M_b}{M_{air}}\frac{(1-\psi)}{\psi}} \qquad \textbf{2.12}$$

$$m_{air,act} = \frac{m_b \cdot \xi}{1-\xi} \qquad \textbf{2.13}$$

Reaction equations

Partial reaction equations of biogas and oil were used to set up an overall reaction equation. It is assumed that 1 mol of CH_4 is combusted; this results in a n_{co2} before reaction (in the biogas), as shown in equation 2.14.

$$n_{CO2} = \frac{n_{CH4}(1-\alpha)}{\alpha} = \frac{1}{\alpha} - 1 \qquad \textbf{2.14}$$

For the ratio between oil and biogas it is again assumed that 1 mol of CH_4 is combusted. This results in:

$$n_{oil} = \frac{n_{CH4}\beta}{\alpha(1-\beta)} = \frac{\beta}{\alpha(1-\beta)} \qquad \textbf{2.15}$$

Molar reaction equation for stoichiometric biogas combustion

$$CH_4 + \left(\frac{1}{\alpha} - 1\right)CO_2 + 2O_2 + 3.76 \cdot 2N_2 \rightarrow \frac{1}{\alpha}CO_2 + 2H_2O + 3.76 \cdot 2N_2 \qquad \textbf{2.16}$$

Mass reaction equation for stoichiometric biogas combustion

$$CH_4 + 2.75\left(\frac{1}{\alpha} - 1\right)CO_2 + 4O_2 + 13.16N_2 \rightarrow \frac{2.75}{\alpha}CO_2 + 2.25H_2O + 13.16N_2 \quad \mathbf{2.17}$$

Molar reaction equation for stoichiometric combustion of Jatropha oil

$$C_{56}H_{101}O_6 + 78.25O_2 + 78.25 \cdot 3.76N_2 \rightarrow 56CO_2 + 50.5H_2O + 78.25 \cdot 3.76 \cdot N_2 \quad \mathbf{2.18}$$

Mass reaction equation for stoichiometric combustion of Jatropha oil

$$C_{56}H_{101}O_6 + 2.88O_2 + 9.48N_2 \rightarrow 2.83CO_2 + 1.05H_2O + 9.48N_2 \quad \mathbf{2.19}$$

Overall molar non-stoichiometic reaction equation

2.20

$$CH_4 + \left(\frac{1}{\alpha} - 1\right)CO_2 + \frac{\beta}{\alpha(1-\beta)}C_{56}H_{101}O_6 + \lambda \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)O_2 + 3.76 \cdot \lambda \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)N_2 \rightarrow$$

$$\left(\frac{1}{\alpha} + 56\left(\frac{\beta}{\alpha(1-\beta)}\right)\right)CO_2 + \left(2 + 50.5 \cdot \frac{\beta}{\alpha(1-\beta)}\right)H_2O + 3.76 \cdot \lambda \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)N_2 + (\lambda - 1)\left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)O_2$$

Overall mass non-stoichiometric reaction equation

2.21

$$CH_4 + 2.75\left(\frac{1}{\alpha} - 1\right)CO_2 + 54.31\frac{\beta}{\alpha(1-\beta)}C_{56}H_{101}O_6 + 2 \cdot \lambda \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)O_2 + 6.58 \cdot \lambda \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)N_2 \rightarrow$$

$$2.75\left(\frac{1}{\alpha} + 56\left(\frac{\beta}{\alpha(1-\beta)}\right)\right)CO_2 + 1.125\left(2 + 50.5 \cdot \frac{\beta}{\alpha(1-\beta)}\right)H_2O + 6.58 \cdot \lambda \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)N_2 + 2 \cdot (\lambda - 1)\left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)}\right) + 2\right)O_2$$

2.2.2 Dual fuel operation parameters

The aim of this section is to predict the effect of dual fuel operation on air-excess ratio, volumetric efficiency and thermal efficiency. Therefore, relations for air-excess fuel ratio λ and for volumetric efficiency are derived. Thermal efficiency is discussed, but no relation is derived to predict thermal efficiency under dual fuel operation. Modelling thermal efficiency would require a complete simulation. This is only useful when it could be compared to in-cylinder pressure measurements.

2.2.2.1 Air-excess ratio (λ)

The expression for overall λ can be derived in several ways. In this paragraph only one possible derivation is discussed. Appendix C shows another approach, this approach gives an identical result as shown here. Equation 2.22 gives the definition for air-excess ratio that is used [21].

$$\lambda = \frac{m_{air}}{m_{oil}\left(\frac{A}{F}\right)_{oil} + m_{CH4}\left(\frac{A}{F}\right)_{CH4}} \qquad [21] \qquad \mathbf{2.22}$$

Stoichiometric air-fuel ratios given in equation 2.23 and 2.24 result from reaction equations 2.17 and 2.19.

$$\left(\frac{A}{F}\right)_{oil} = \frac{2.88 + 9.48}{1} = 12.36 \qquad \mathbf{2.23}$$

$$\left(\frac{A}{F}\right)_{CH4} = \frac{17.16}{1} = 17.16 \qquad \mathbf{2.24}$$

Definitions for m_{air}, m_{oil} and m_{CH4} are used as defined in respectively equation 2.13, 2.8 and 2.3 are used to derive the following expression for the air-excess ratio.

$$\lambda(\xi, \theta, \sigma) = \frac{\frac{\xi}{1-\xi}}{\frac{\theta}{1-\theta} \cdot 12.36 + \sigma \cdot 17.16} \qquad \mathbf{2.25}$$

Equation 2.25 presents λ as a function of ξ, θ and σ. Equation 2.27 expresses λ as a function of ξ, ε and σ.

$$\frac{\theta}{1-\theta} = \frac{\sigma \cdot LHV_{CH4} \cdot \varepsilon}{(1-\varepsilon) \cdot LHV_{oil}} \qquad \textbf{2.26}$$

$$\lambda(\xi,\varepsilon,\sigma) = \frac{\dfrac{\xi}{1-\xi}}{\dfrac{\sigma \cdot LHV_{CH4} \cdot \varepsilon}{(1-\varepsilon) \cdot LHV_{oil}} \cdot 12.36 + \sigma \cdot 17.16} \qquad \textbf{2.27}$$

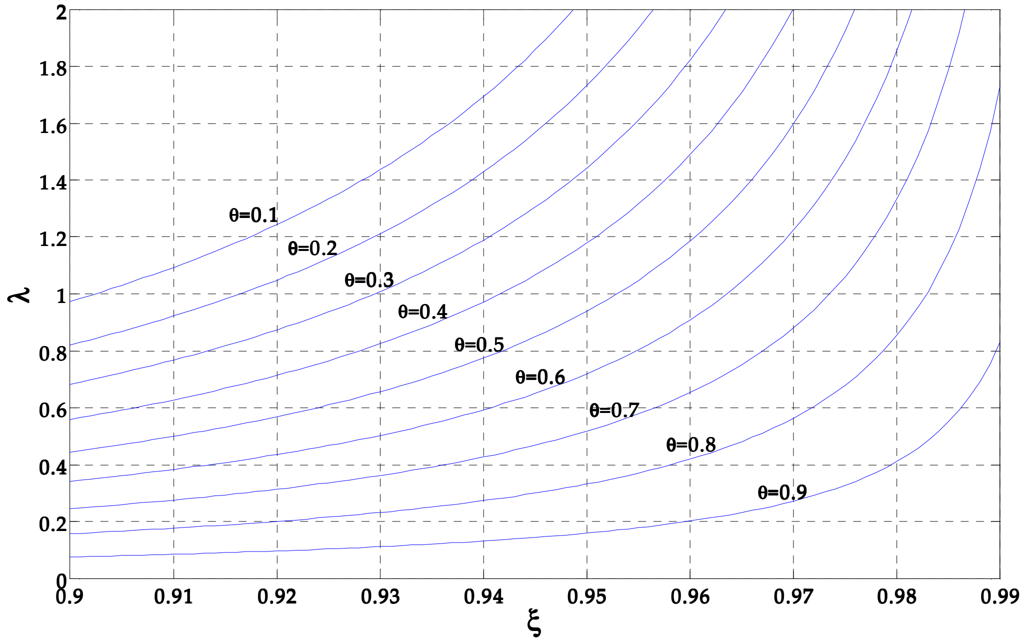

Figure 2.2 Air-excess ratio as a function of fraction of air in intake mixture.

Figure 2.2 presents air-fuel ratio λ as a function of the fraction of oil in the total fuel (θ) and fraction of air in total inlet mixture (ξ); it gives an impression of equation 2.25. The value for methane fraction in biogas is fixed at α=0.7; this means that 70% of the total volume is methane, which corresponds with good quality biogas. Air fraction in the total inlet mixture ξ depends on the volume of the intake mixture taken into the cylinder each inlet stroke. A higher volumetric efficiency results in larger volume flow of the intake mixture. When it is assumed that the fraction of oil in the total fuel mixture (θ) is kept constant, a constant flow of biogas comes into the inlet mixture. Under this assumption a lower volumetric efficiency results in a smaller total volume of the intake mixture. This results in smaller value for ξ, which represents the fraction of air in the total inlet mixture. A line of a constant oil fraction (θ) shows that when ξ decreases, air-excess ratio might become too low to operate smoke-free.

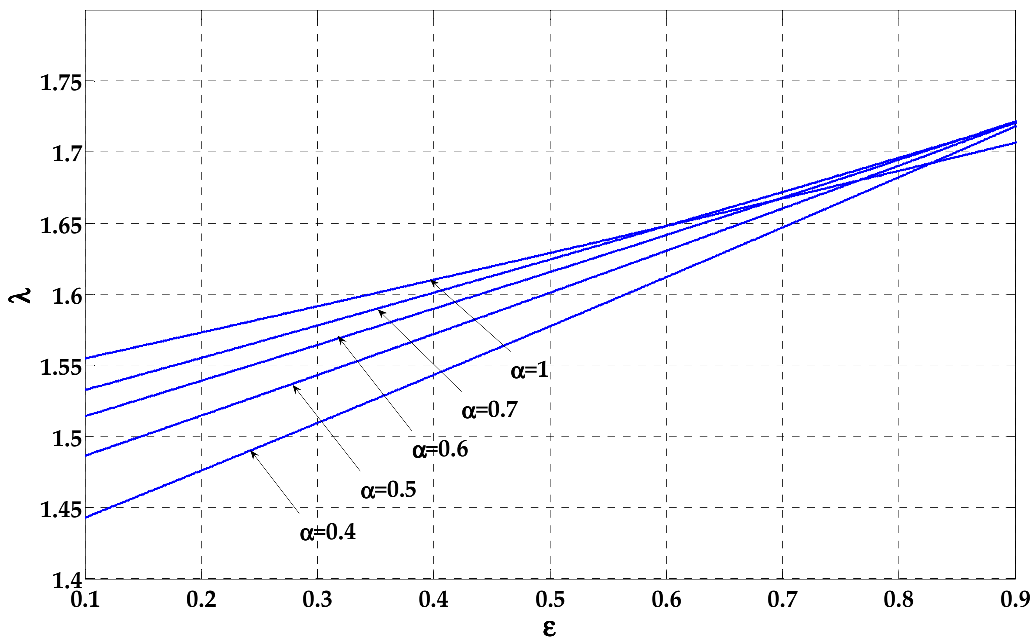

Figure 2.3 Air-excess ratio for different biogas qualities as a function of heat release fraction.

Figure 2.3 shows air-excess ratio as a function of heat release fraction from oil ε for five different fractions of methane in the biogas α. It is assumed that the load is 10 kW and thermal efficiency is 32% and remains 32% and that volumetric efficiency is 89%. This assumption is based on the experimental data for pure oil operation. A decrease in ε means that less of the heat release originates from oil and more from methane. A decrease in ε results in more biogas in the intake mixture; the biogas replaces air, therefore air-excess ratio decreases with decreasing ε. At α=1 only methane is present in the gas; this means that only methane replaces air in the intake mixture, resulting in the smallest decrease in air-excess ratio. At α=0.4 the biogas contains 40% methane and 60% carbon dioxide; this means that more air is replaced by biogas at the same methane fraction, therefore the inclination of the α=0.4 line is higher; air-excess ratio decreases faster[2].
It is expected that the experiments will show a faster decrease of air-excess ratio as a function of ε, for pure methane, than for biogas.

[2] Figure 2.3 shows that the line of α=1 seems to go to another value for λ under pure oil operation at ε.=1. At this point only oil is used; therefore air-excess ratio should be the same. This is not the case, the reason is not found.

2.2.2.2 Volumetric efficiency

Volumetric efficiency is one of the parameters that is considered a performance parameter of the dual fuel engine because it has a direct influence on power output and consequently on thermal efficiency. In general volumetric efficiency is defined as:*"the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the system" [21] (p. 54).* Usually volumetric efficiency is between 80% and 90%. For diesel engines it is on the high end of this range.

The reference density of air is taken at ambient conditions. In dual fuel mode biogas replaces air in the inlet mixture, therefore it is expected that volumetric efficiency decreases with increasing biogas fraction in the intake mixture. The volumetric efficiency of the engine at normal jatropha oil operation can be obtained in an experiment and with the use of equation 2.28 [21]. The reference volumetric efficiency as expressed in equation 2.30 is the volumetric efficiency of normal diesel operation at a certain rotational speed. It is assumed that the total volume of the intake mixture does not change when biogas is added; therefore the reference volumetric efficiency does not change when biogas is added. Appendix D shows why this assumption has been made. Biogas in the intake mixture replaces air and will therefore influence engine performance parameters. An expression for dual fuel volumetric efficiency is derived. This leads to expression 2.33. It is expected that dual fuel volumetric efficiency decreases with increasing biogas fraction since biogas replaces air in the intake mixture.

$$\eta_v = \frac{m_{air}}{\rho_{ref} \cdot V_d} \qquad [21] \qquad \textbf{2.28}$$

V_d = displacement volume of the engine = 1.093 dm^3

$$\rho_{ref} = \frac{p_0}{R_{air} \cdot T_0} \qquad \textbf{2.29}$$

$$\eta_{v,ref} = \left(\frac{(\dot{m}_{air})_{oil}}{\rho_{ref} \cdot V_d \cdot CPS} \right)_{oil} = \frac{(\dot{m}_{air})_{dualfuel} + \dot{m}_b}{\rho_{air+b} \cdot V_d \cdot \frac{N}{n_R}} \qquad \textbf{2.30}$$

N= rotational speed [rounds/sec]

n_R= number of rotations per cycle=2

$$\dot{V}_{ref} = \dot{V}_{dual} \qquad \textbf{2.31}$$

$$\dot{V} = \frac{\dot{m}}{\rho} \qquad \textbf{2.32}$$

Expression 2.30 presents the reference volumetric efficiency, which is the volumetric efficiency under standard jatropha oil operation. It is assumed that volume flow is independent of the composition of the intake mixture. This means that at the same rotational speed the volume floe of the intake mixture is the same for air as for air and biogas together under dual fuel conditions. This is expressed in equation 2.31 and 2.32 The reference density of air at ambient conditions is expressed in equation 2.29.
Dual fuel volumetric efficiency is expressed in equation 2.33. The mass flow of air in this expression is smaller than the mass flow of air for standard jatropha oil operation as expressed in the reference volumetric efficiency. Equation 2.34 and 2.35 give expressions for the molar mass of respectively biogas and air-biogas mixture. These two are used in equation 2.33; this results in the expression for dual fuel volumetric efficiency as provided in equation 2.36. In this expression Ψ is used instead of ξ. Since ξ and Ψ both represent the fraction of air in the intake mixture Ψ can be written as a function of ξ as in equation 2.11. Therefore dual fuel volumetric efficiency is expressed as function of ξ and α (mass methane fraction in biogas). The resulting expression is shown below in equation 2.37; molar mass of air is assumed constant at 28.8 kmol/kg.

$$\eta_{v,dual} = \frac{(\dot{m}_{air})_{dualfuel}}{\rho_{ref} \cdot V_d \cdot CPS} = \underbrace{\frac{\dot{m}_{air} + \dot{m}_b}{\rho_{air+b} \cdot V_d \cdot CPS}}_{\eta_{v,ref}} \cdot \underbrace{\frac{\dot{m}_{air}}{\dot{m}_{air} + \dot{m}_b}}_{\xi} \cdot \frac{\rho_{air+b}}{\rho_{ref}}$$

$$= \eta_{v,ref} \cdot \xi \cdot \frac{\rho_{air+b}}{\rho_{ref}} = \eta_{v,ref} \cdot \xi \cdot \frac{M_{air+b}}{M_{air}} \tag{2.33}$$

$$M_b = 44 - 28\alpha \tag{2.34}$$

$$M_{air+b} = \psi \cdot M_{air} + (1 - \psi)(44 - 28\alpha) \tag{2.35}$$

$$\eta_{v,dual} = \eta_{v,ref} \cdot \xi \cdot \frac{\psi \cdot M_{air} + (1-\psi)(44-28\alpha)}{M_{air}} \tag{2.36}$$

$$\eta_{v,dual} = \eta_{v,ref} \cdot \xi \cdot \frac{\xi\left(\frac{44-28\alpha}{28.8} \cdot \xi - ((44-28\alpha)(1-\xi))\right)}{\xi \cdot (72.8-28\alpha) + 28.8} \tag{2.37}$$

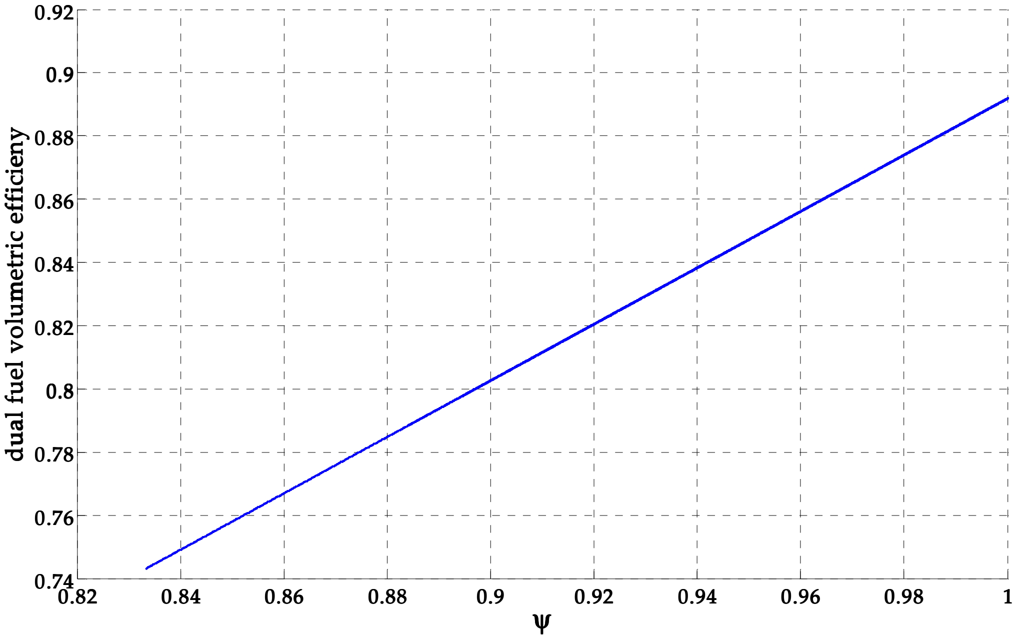

Figure 2.4 Dual fuel volumetric efficiency as a function of ψ

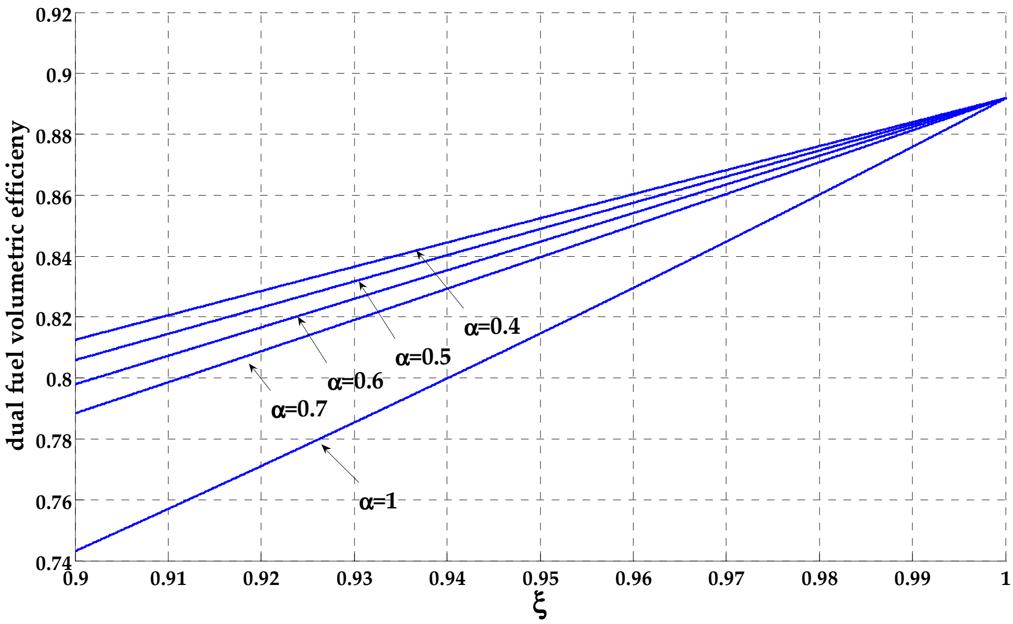

Figure 2.5 Dual fuel volumetric efficiency as a function of ξ

Figure 2.4 shows the dual fuel volumetric efficiency as a function of molar fraction of air in the inlet mixture (ψ). It shows that dual fuel volumetric efficiency decreases with increasing gas fraction in the intake mixture. The line is representative for all methane fractions in biogas (α) The non-compensated value for volumetric efficiency from the reference test for jatropha oil is used, for constructing this plot. A description of this test is found in section 4.2.2. For the ambient conditions a temperature of 20˚C and an ambient pressure of 102 kPa are used that reflect summer weather.

Figure 2.5 shows volumetric efficiency as a function of molar fraction of air in the inlet mixture (ξ) for the same conditions as described before This plot shows that for different methane fraction in the biogas (α) inclination is different. More methane in the biogas results in a higher inclination. An increase in methane concentration in the biogas results in faster decrease in dual fuel volumetric efficiency, because the molar mass of the air-fuel mixture decreases, which results in a decrease in dual fuel volumetric efficiency as expressed in equation 2.33. In other words a higher methane fraction in the biogas results in a lower molar mass and therefore a lower density of the intake mixture. A lower density intake mixture results in a lower volumetric efficiency, when it is plotted in this way.

2.2.2.3 Thermal efficiency

Thermal efficiency is considered the overall system efficiency. It is the ratio between the electrical output power and the input power as expressed in equation 2.38.

$$\eta_{th} = \frac{P_{out}}{P_{in}} = \frac{P_{elec}}{\dot{m}_{oil} \cdot LHV_{oil} + \dot{m}_{CH4} \cdot LHV_{CH4}} \qquad \textbf{2.38}$$

P_{elec}= electrical power output.

The effect of dual fuel operation on thermal efficiency is difficult to predict, since complex combustion processes define how much of the internal energy of the fuel is converted into useful electrical power. The combustion process (flame development and propagation) taking place during dual fuel operation is different from oil operation. This results in differences in energy conversion between the two operation modes resulting in possible differences in thermal efficiency. Thermal efficiency is measured during the tests. Modelling thermal efficiency would require a complete simulation. This is only useful when it would be able to be compared to experimentally obtained in-cylinder pressure data. Pressure inside the cylinder during combustion can not be measured.
The power output of an engine can be obtained with equation 2.39 from Heywood [21].

$$P_{out} = \frac{m_{air} \cdot N \cdot \eta_{thermal} \cdot LHV_{average} \cdot \left(\frac{F}{A}\right)}{n_R} = \dot{m}_{air} \cdot \eta_{thermal} \cdot LHV_{average} \cdot \left(\frac{F}{A}\right) \qquad \textbf{2.39}$$

N= rotational speed [rounds/sec]
n_e= number of rotations per cycle=2

The power output is imposed by the electrical load and is not expected to change significantly under dual fuel operation. The other variables in equation 2.39 can change under dual fuel operation. The mass of air will change as described in section 2.2.2.1. $LHV_{average}$ and actual fuel air ratio change according to expressions 2.40 and 2.41.

$$LHV_{average} = \varepsilon \cdot LHV_{oil} + (1-\varepsilon) \cdot LHV_b = \varepsilon \cdot LHV_{oil} + (1-\varepsilon) \cdot \sigma \cdot LHV_{CH4} \qquad \textbf{2.40}$$

$$\left(\frac{F}{A}\right)_{act} = \frac{\dot{m}_b + \dot{m}_{oil}}{\dot{m}_{air}} = \frac{1-\xi}{\xi \cdot (1-\theta)} \qquad \textbf{2.41}$$

Thermal efficiency in 2.41 changes due to changes in the other parameters; the change under dual fuel operation can also be a result of changes in combustion processes. Since dual fuel combustion combines principles of spark ignition and compression ignition, it is difficult to predict these changes. Propagation speed and temperature of the flame front could change under dual fuel operation, resulting in a change in thermal efficiency. Air-excess ratio influences flame propagation of the methane flame. A too lean mixture results in slower flame propagation and eventually total extinguishing of the flame. A slower flame propagation results in lower thermal efficiency. This effect is expected to happen at lower end of ε, when most of the heat release comes from methane. This can not predicted exactly because there is no model and simulation of thermal efficiency under dual fuel operation.

2.2.3 Overview

This section provides an overview of the above described parameters. This overview shows which parameters are considered input and which are considered output variables and it shows mutual influences between parameters. Figure 2.6 gives a schematic overview. The input parameters ξ, ψ, θ, β and ε are computational parameters and air-excess ratio λ, thermal efficiency η_{therm} and dual fuel volumetric efficiency η_{vol} are the engine performance parameters, considered in this study. Electrical load, biogas flow, oil use, oxygen content of the exhaust gas, rotational speed, and volume of the intake mixture are actually measured during the experiments.

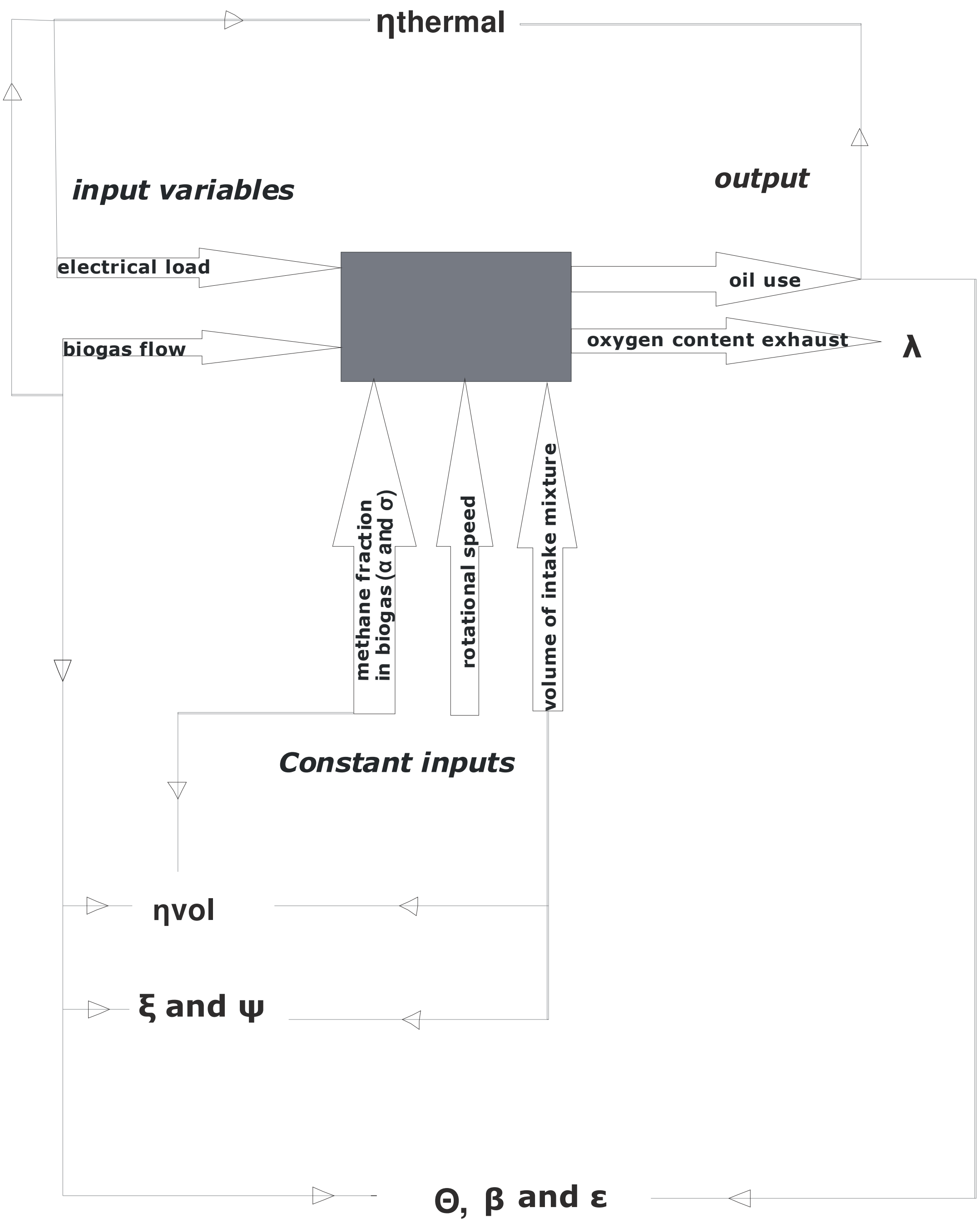

Figure 2.6 Overview input and output parameters

During the experiment the following parameters are measured: rotational speed, volume of intake mixture, electrical load, biogas use, oil use and oxygen content in exhaust gas.

Input

Parameters that can be varied during the experiment are the load connected to the generator and biogas flow and methane fraction in the biogas. Electrical load can be adapted manually with steps of approximately 1 kW. Biogas flow can be varied with a control valve. Fraction of methane in biogas can be altered but only at fixed values in premixed bottles; therefore it cannot be altered during an experiment. For this reason it is grouped with the fixed variables in Figure 2.6.

Fixed input

Rotational speed and volume of intake mixture are considered engine characteristics. The engine has a fixed rotational speed, which is always in the region of 2000 rpm. The engine regulates the amount of fuel that is required automatically to meet the demand of the load that is imposed at this fixed speed. When gas is added to the system, the amount of liquid fuel that is injected, is automatically adapted to a level that sufficient energy is released to meet the demand. The amount of liquid fuel that is injected can not be regulated manually; the engine automatically regulates this; it is imposed by the electrical load. Reference volumetric efficiency is considered constant. This means that the volume of the intake mixture is considered constant. Appendix D shows why this assumption is made.

Output

Oil use and oxygen content of the exhaust gas are considered output variables, since these can not be altered. Power output, oil and biogas use, and methane fraction $\alpha(\sigma)$ are relevant for obtaining a value for thermal efficiency. When, oil and biogas use are known, oil fraction in the total fuel mixture θ and β can be obtained. To obtain a value for volumetric efficiency, volumes of the intake mixture and biogas flow are necessary. When these two are known, air fraction in the intake mixture ξ and Ψ can be obtained. In order to obtain a value for air-excess ratio; oil use; biogas use; volume of intake mixture; oxygen content of the exhaust gas and methane fraction of the biogas are required.

2.2.4 Smoke limit

Smoke limit is the smallest amount of oxygen that is required for smoke-free combustion; this results in a minimum air-excess λ at which the combustion can take place smoke free. It is very difficult to obtain the air-excess ratio at which smoke will occur under standard diesel or standard jatropha oil operation theoretically, because processes are very complex. Typically this value is in the order of 1.3 for diesel[22]. This value is assumed to be in the same order for jatropha oil.

Biogas replaces air in the intake mixture resulting in a decrease in volumetric efficiency and air-excess ratio. This means that the smoke boundary might be reached due to dual fuel combustion.

One of the limits that is investigated is the maximum possible amount of carbon dioxide in the biogas. Too much carbon dioxide in the biogas could result in a too large decrease in air-excess ratio under dual operation. The amount of carbon dioxide in the biogas influences the occurrence of the smoke boundary.

It might be possible that the use of biogas changes the initial smoke boundary for jatropha oil. Whether or not the smoke limit changes under dual fuel operation compared to pure oil operation, depends on which assumptions are made, concerning requirement of oxygen during combustion. These assumptions are described in two hypotheses below.

1. It is assumed that oil requires λ at the normal smoke limit and methane combusts at a relative air-fuel ratio λ of 1. This results in a required overall air-fuel ratio that is smaller than that of pure oil combustion.
2. It is assumed that the overall air-excess ratio λ should at least be equal to the smoke limit of oil, because locally, near the spray of oil, λ needs to be at least at the smoke boundary. It is assumed that methane requires the same air-excess ratio as oil; therefore λ at smoke boundary equals λ_{oil}

Hypothesis 1

Since methane combusts at a relative air-fuel ratio of 1 and oil requires a lean mixture with $\lambda>1$ for smoke-free combustion, the overall relative air-fuel ratio is somewhere in between these two values, depending on the biogas fraction in the intake mixture and methane fraction in the biogas. Equations 2.42 and 2.45 show how the minimum required overall λ can be obtained. In these equations λ_{CH4} is assumed to be 1 and λ_{oil} is the smoke limit for oil combustion, which is supposed to be in the order of 1.3.

$$\lambda_{\min imum\, overall} = \frac{\lambda_{oil} \cdot m_{air,stoich,oil} + \lambda_{CH4} \cdot m_{air,stoich,CH4}}{m_{air,stoich,oil} + m_{air,stoich,CH4}} \qquad \textbf{2.42}$$

It assumed that 1 kg of methane is combusted.

$$m_{air,stoich,CH4} = 16.17 \qquad \textbf{2.43}$$

$$m_{air,stoich,oil} = 671.4 \cdot \frac{\beta}{\alpha(1-\beta)} \qquad \textbf{2.44}$$

$$\lambda_{mimimum\,overall} = \frac{671.4 \cdot \beta \cdot \lambda_{oil} + \alpha \cdot (1-\beta) \cdot 17.16 \cdot \lambda_{CH4}}{671.4 \cdot \beta + \alpha \cdot (1-\beta) \cdot 17.16} \qquad \textbf{2.45}$$

Filling in equation 2.45 with data obtained from the reference measurement results in a decrease in required air-excess ratio. It is assumed that biogas is used with a methane fraction 70% (α=0.7) and 40% of the fuel mass is oil (θ=0.4, β=2.76 10^{-2}). During the reference oil experiment an air-excess ratio of 2.1 is measured for a load of 9.9 kW. When these values are used and it assumed that methane combusts stoichiometric at λ=1; a value of 1.57 is obtained for minimum overall dual fuel air-excess ratio at these conditions.

Hypothesis 2
This hypothesis assumes that the overall relative air-fuel ratio λ should at least be equal to λ_{oil} required for smoke-free combustion. Therefore, the required overall λ for smoke free combustions remains constant in the order of 1.3.
At full load oil combusts with a λ in the order of 1.6; under dual fuel conditions with a high fraction of biogas in the intake mixture expected air-excess ratio λ is not smaller than 1.348. At lower loads or at lower biogas fraction the occurring λ is higher than this value.

Therefore no real problems are expected concerning the smoke limit, it is expected that λ will not decrease too much under dual fuel conditions to reach the smoke limit. During the experiments air-excess ratio is measured; in this way it can be seen if air-excess ratio comes near the smoke limit. Smoke is also detected visually during experiment, when it would occur.

2.3 Ignition delay and knock in a dual fuel diesel engine

The dual fuel diesel engine shows characteristics of both compression ignition (CI) and spark ignition (SI) engine. It takes in gaseous fuel (biogas) with the intake air. This means that biogas is compressed during the compression stroke; at the end of the compression stroke a pilot injection of jatropha oil is added. The pilot injection in a dual fuel diesel engine is required to start ignition, therefore it could also be seen as the spark in a spark ignition engine.

In both spark ignition and compression ignition engines knock can occur but the origin of the knocking behaviour is different for both cases. Both types of knock can occur in a dual fuel diesel engine, both are described. The effect of dual fuel operation on ignition delay and knock is described in section 2.3.3.

2.3.1 Spark-ignition engine

In a spark ignition engine knock is defined in Heywood as follows:

> *"Knock is the name given to the noise which is transmitted through the engine structure when essentially spontaneous ignition of a portion of the end-gas- the fuel-air, residual gas, mixture ahead of propagation flame- occurs. When this abnormal combustion process takes place, there are very high local pressures and the propagation of pressure waves of substantial amplitude across the combustion chamber."   [21]. p.450)*

Models for knock are based on models for auto-ignition of the fuel-air mixture in the end-gas. Auto-ignition of the end-gas usually occurs on hot-spots on the surface of the cylinder walls and it can occur before or after the spark in a spark ignition engine or the pilot injection in the case of a dual fuel engine. The surface ignition takes place before the flame front of the normal combustion reaches the spot where surface ignition took place [21]. The ability of a fuel to resist knock is measured by its octane number.

In order to find out whether or not auto-ignition will occur two types of models can be used: empirical correlation models or chemical mechanisms.

"Induction-time correlations are derived by matching an Arrhenius function to measured data on induction or autoignition times, for given fuel-air mixtures, over the relevant mixture pressure and temperature ranges. It is then assumed that auto-ignition occurs when

$$\int_{t=0}^{t_i} \frac{dt}{\tau} = 1 \qquad \textbf{2.46}$$

Where τ is the induction time at the instantaneous temperature and pressure for the mixture, t is the elapsed time from the start to the end-gas compression process (t=0) and t_i is the time of autoignition."[21] , p.468)

The most frequently tested empirically obtained relation for induction time in a spark ignition engine is that proposed by Douaud and Eyzat [21]:

$$\tau = 17.68\left(\frac{ON}{100}\right)^{3.402} p^{-1.7} \exp\left(\frac{3800}{T}\right) \qquad \textbf{2.47}$$

2.3.2 Diesel engine

Ignition delay is an important parameter for diesel knock. Ignition delay is defined as

> *"the time (or crank angle) interval between the start of injection and the start of combustion. The start of injection is usually taken as the time when the injector needle lifts of its seat."* *[21] (p.540)*

Diesel knock occurs when the cetane rating of a fuel is too low. A too low cetane rating results in a too long ignition delay. Most of the fuel is already injected into the cylinder before combustion starts; this results in a very rapid combustion process resulting in rapid pressure rise and pressure peaks. This process can cause an audible knocking sound [21].

"Studies with fuel injection into constant-temperature and pressure environments have shown that the temperature and pressure of the air are the most important variable for a given fuel composition. Ignition delay data from these experiments have usually been correlated by equations of the form:

$$\tau_{id} = A \cdot p^{-n} \exp\left(\frac{E_A}{\tilde{R}T}\right) \quad \text{"[23]} \qquad (p.543) \qquad \mathbf{2.48}$$

"In general τ_{id} is a function of mixture temperature, pressure, equivalence ratio, and fuel properties"[21].

$$\int_{t_{si}}^{t_{si}+\tau_{id}} \left(\frac{1}{\tau}\right) dt = 1 \qquad \mathbf{2.49}$$

t_{si} is the time of start of injection

An empirical relation for ignition delay in diesel engines is developed by Hardenberg and Hase [21] and is given in equation 2.50.

.

$$\tau_{id} = (0.36 + 0.22\overline{S}_p)\exp\left[E_A\left(\frac{1}{\tilde{R}T} - \frac{1}{17.190}\right)\left(\frac{21.2}{p-12.4}\right)^{0.63}\right] \qquad \mathbf{2.50}$$

$$E_A = \frac{618,840}{CN + 25} \qquad \mathbf{2.51}$$

p and T are pressure and Temperature at TDC as calculated in Appendix E and $\overline{S}_p$ is mean piston speed equal to $\frac{2Sn}{60}$, where S represents the stroke and n the rotational speed.

2.3.3 Dual fuel

In a dual fuel engine both SI and CI knock can occur. Spark ignition knock occurs when the end-gas spontaneously ignites and CI knock occurs when ignition delay is too long resulting in rapid pressure rise and pressure peaks.

The occurrence of SI nock depends on the octane number of the fuel. Methane has a very high octane number of 120, which means that it is highly knock resistant. Still this type of knock can occur in a dual fuel engine, it is called end-gas knock. High intake temperature and high substitution levels result in end gas knock [24]. For SI knock equation 2.47 can be used to obtain induction time for methane.

Appendix E gives values for end-gas pressure and temperature at several air biogas ratios in the intake mixture. A plausible air fraction of 92% gives a temperature of 790K and a pressure of 45.8 bar. This results in an induction time of 6 ms. which corresponds with approximately 72 degrees crank angle. This is long, but probably the estimation is not very accurate. But, there is a risk of end-gas knock.

Diesel knock occurs when the cetane rating of a fuel is too low. Cetane number of pure Jatropha oil is 45 and the cetane rating of diesel ranges from 45 to 55 [25]. Cetane number of Jatropha oil is comparable to that of diesel, at the lower end of the range, which could result in a higher likelihood of the occurrence of diesel-knock when pure jatropha oil is used.

Ignition delay increases with the addition of gaseous fuel in the intake air [26] this could result in excessive rates of pressure rise and creates diesel knock [24]. The intake of methane in the intake air results in an increase of the ignition delay of the diesel fuel or vegetable oil as described in section 2.1 [18]. This is partly because of the decrease in compression temperature due to the change in specific heat of the compressed intake mixture and partly due to the reduced oxygen concentration in the intake mixture and inhibiting effect of the presence of methane [19].

In order to obtain an accurate estimation of ignition delay equation 2.49 should be solved. The injection timing is required to solve this equation; injection timing of the engine used during the experiments is not known therefore equation 2.50 is used to estimate ignition delay. Pressure and temperature are taken at top dead centre as provided in Appendix E with the cetane rating for jatropha oil is taken to be 45. The ignition delay calculated with equation 2.50 is 4.36 CD which corresponds with 0.36 ms. This is a low value, which decreases the possibility of the occurrence of diesel knock.

The experiments will prove whether knock will occur during dual fuel operation and under which condition this will happen.

The occurrence of knock is investigated experimentally by listening to combustion noise, because no measurement equipment to observe knock, like in-cylinder pressure measurement, is available. When heavy knock would occur this is also observed in the rapid decrease in thermal efficiency. Since the occurrence of diesel knock is unlikely, it is likely that when knock occurs it is end-gas knock. The end-gas knock occurs when too much methane is added to system. Therefore the occurrence of knock corresponds with the heat release fraction of methane in the total fuel mixture.

3 Experiments

This chapter describes the engine-generator set that is used and provides properties of the fuels used. The experimental set-up is described as well as the measured parameters and the measurement equipment for both reference measurement and dual fuel measurement. Section 3.4 describes the experimental procedure. Section 3.5 describes the equipment used and how the gathered data is transformed into data for the performance parameters. Measurement accuracy is also described in this section.

3.1 Engine-generator set

The generator set that is used for the experiment consists of a horizontal 1.093 litre one-cylinder, naturally aspirated, direct injection, four stroke diesel engine connected with v-belts to a generator with a 12 kW electrical output. Data from the engine- generator set provided by the manufacturer are presented in Table 3.1.

Table 3.1 Yangke 12GF-SF

No. cylinders	1
No. stroke	4
Bore	110 mm
Stroke	115 mm
$V_{displacement}$	1.093 l
Compression ratio	17
Rated Voltage	400/230 V
Rated output	12 kW
Rated speed	1500-1800 rpm
Rated output/speed	13.24 kW at 2200 rpm
Generator type	synchronous 3 -phase
Cos ϕ	0.8

The engine is of Chinese make, produced under Deutz license. One of the known downsides of the use of straight vegetable oil is the high viscosity, which is supposed to clog filters and nozzles. Therefore the engine was initially adapted to run on straight vegetable oil. The engine is fitted with a two-tank system: one for diesel and one for straight vegetable oil. Diesel is used to start and run until the engine is warm. The vegetable oil flows through a heat exchanger mounted in the cooling system of the engine; the vegetable oil is pre-heated in order to reduce viscosity. When the engine, coolant fluid and vegetable oil is warm enough fuel use is switched to vegetable oil. During the construction of the experimental set-up the heat exchanger started leaking; coolant fluid flowed into the vegetable oil. The mixture of water and oil resulted in an emulsion that completely locked the engine. Furthermore it was discovered that the engine started, stopped and ran perfectly well on vegetable oil. Therefore the two-tank system was not required anymore and it was decided not to use the extra tank and the leaking heat-exchanger. For this study the engine will not run long term therefore no problems are expected due to the use of non pre-heated

vegetable oil. Although experience shows that not pre-heating the fuel that cold starts causes defects in the long run and should usually be avoided [27].
The synchronous 3-phase generator has a power factor of 0.8, which means that the apparent power output is 15 kVA and the active power output is 12 kW. To each phase a load of maximum 4 kW is connected. Figure 3.1 presents a photo of the generator set. Figure 3.2 shows the adapted fuel system.

Figure 3.1 engine-generator set

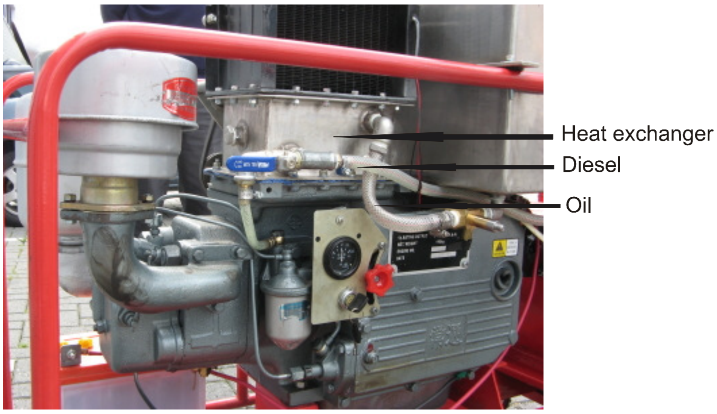

Figure 3.2 two tank system and heat-exchanger

3.2 Fuel

The oil is filtered with a 1 μm filter before use. This specific oil is not tested for its properties therefore properties are obtained from literature. Table 3.2 provides fuel properties relevant for this study. For comparison fuel properties of diesel and sunflower oil are also provided. Fuel properties for methane are also provided, since methane is the component in biogas that is used as fuel. Table 3.2 is constructed with data from [21],[28] and [29]. Appendix B shows how values for molar mass and lower heating value for jatropha and sunflower oil are obtained.

Table 3.2 Fuel properties

	Jatropha oil	Sunflower oil	Diesel	CH_4
Molar mass (kg/kmol)	869	880	170	16
Density (kg/m³) at 0˚C and atm. pressure	917	923[3]	850	0.72
Viscosity (cSt) at 40˚C	36	32.6	2.44-2.7	-
LHV (MJ/kg)	37	37	42.5	50

[3] At 5˚C.

Biogas is a mixture of mainly methane and carbon dioxide but also water, nitrogen, oxygen and traces of hydrogen sulphide can be present. Average composition of biogas is provided in Table 3.3 [30].

Table 3.3 Biogas content

Component	Concentration
CH_4	45-75 %
CO_2	24-45 %
H_2O	2-7 %
H_2S	20-20000 ppm
N_2	<2%
O_2	<2%
H_2	<1%

Biogas produced from jatropha press cake or other agricultural wastes is not available, therefore simulated biogas is used. A bottled premix of methane and carbon dioxide is used since these are the main components in natural biogas. The effect that the other components have on the combustion process and engine performance is expected to be very small; therefore they are neglected. Hydrogen sulphide can be very corrosive and should always be avoided in a fuel, but it is relatively easy to remove it from biogas. Table 3.4 provides the methane- carbon dioxide ratios that are used.

To examine the effect of carbon dioxide in the biogas on engine performance dual fuel experiments are also carried out with pure methane as a reference.

Table 3.4: Simulated biogas content

CH_4 (vol.%)	CO_2 (vol.%)
100	0
70	30
60	40
50	50
40	60

3.3 Experimental set-up

Before the engine is operated in dual fuel mode a reference measurement is performed for jatropha oil and for diesel (without biogas). This measurement gives engine characteristics for thermal efficiency, volumetric efficiency and air-excess ratio. Diesel is also tested to see if the engine performs differently for jatropha oil than for diesel.
For dual fuel operation thermal efficiency and air-excess ratio is obtained; it is assumed that reference volumetric efficiency does not change between oil operation and dual fuel operation. This means that the total volume flow of the intake mixture does not change between operation modes. Appendix D shows why this assumption has been made.
First a schematic overview of the experimental set-up and a brief overview of the input and output variables is provided for both cases.

3.3.1 Reference measurement

Figure 3.3 gives a schematic overview of the experimental set-up used for the reference measurement of jatropha oil operation. A brief overview of the input and output variables for the reference measurements is provided below.

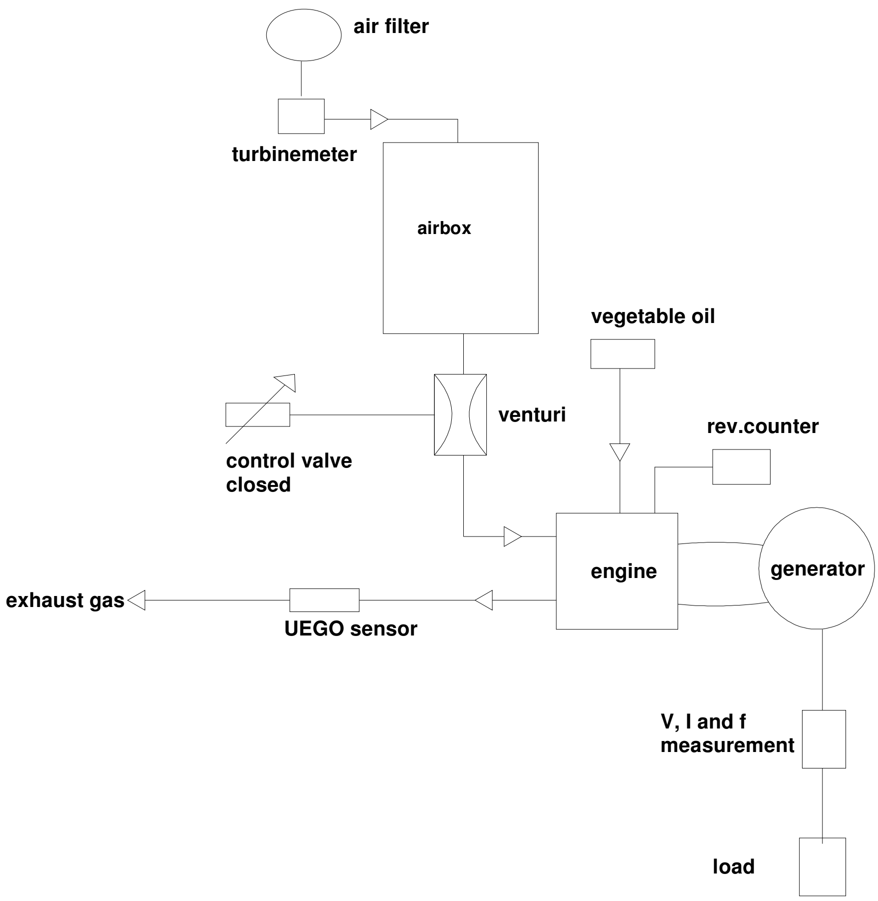

Figure 3.3 Overview of the reference measurement set-up

Input variables

Electrical load

The only parameter that is an input variable for the reference tests is the imposed electrical load.

Output variables

Thermal efficiency

Jatropha oil use is measured by placing the oil tank on a measuring scale; it is measured over time. Fuel use and electrical power output are used to obtain a value for thermal efficiency.

Volumetric efficiency

In order to obtain volumetric efficiency, airflow into the cylinder is measured with a turbine meter. Before the air reaches the inlet valve it passes through an air filter, turbine meter, and air-box. An air box is used as a buffer of air to remove fluctuations in the incoming air. In a one-cylinder engine fluctuations in pressure and velocity of the incoming air are quite severe; the turbine meter is sensitive to these fluctuations therefore the air box is used as a buffer. The turbine meter and air box are only used during the reference measurement. A description of the turbine meter is provided in the next section.

Air-excess ratio

Air-excess ratio is obtained in two ways: from the oxygen in exhaust gas and from the turbine meter measurement. The UEGO (Universal Exhaust Gas Oxygen) sensor is used to measure the oxygen concentration in the exhaust gas; a value for air-excess ratio λ is obtained. The next section will show how air-excess ratio is obtained from a value for oxygen concentration in the exhaust gas. A description of the UEGO sensor is provided in the next section. The amount of air that is measured with the turbine meter and the amount of fuel measured are also used to obtain a value for air-excess ratio. This is compared to the UEGO result.

The photo in Figure 3.4 gives an overview of the experimental set-up as used for the reference measurements.

Figure 3.4 Experimental set-up for reference measurement

3.3.2 Dual fuel measurement

Figure 3.5 provides a schematic overview of the experimental set-up that is used for dual fuel measurement. A brief overview of the measured input and output variables is provided below.

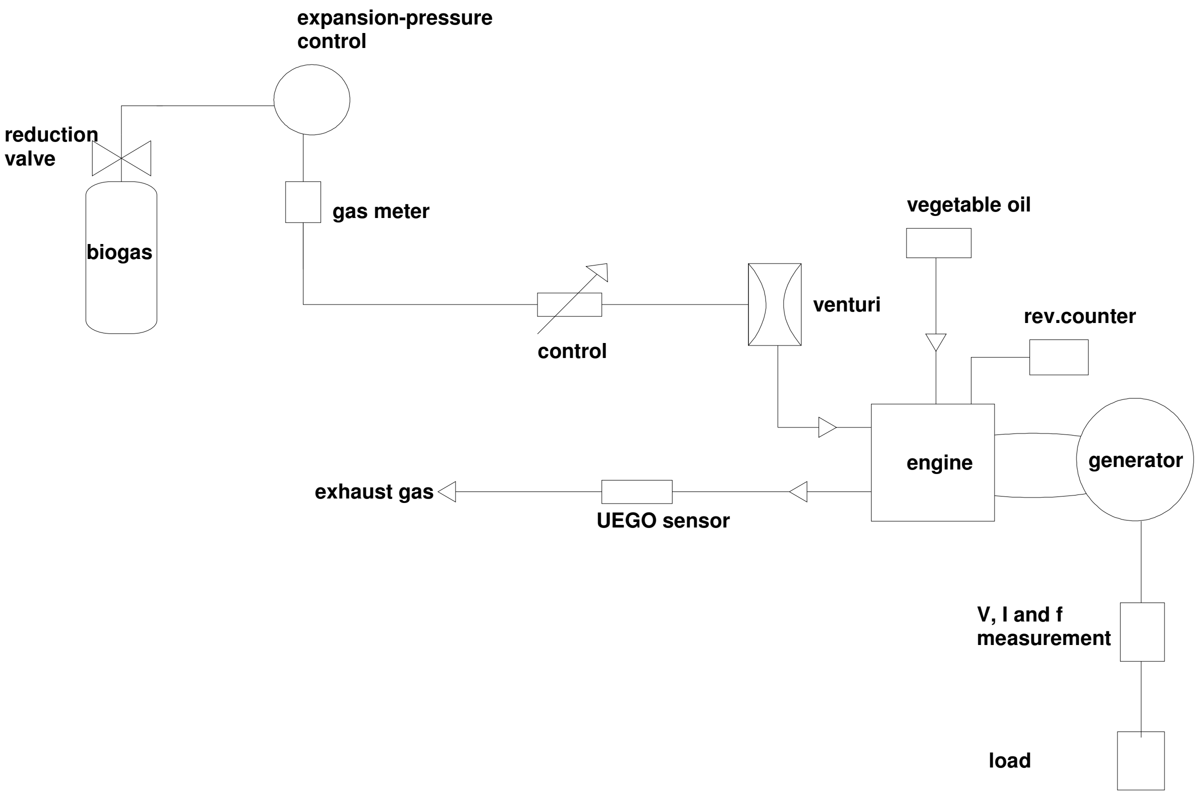

Figure 3.5 Experimental set up for measurements with oil and gas

For the dual fuel measurements the air box and turbine meter are removed. The air-box is removed for safety reasons. When a fault occurs it might be possible that the methane containing biogas flows into the air box. When the air-box fills up with biogas a very dangerous situation exists since methane is highly explosive and it is invisible and odourless. Without the air box the turbine meter does not work properly due to excessive pressure fluctuations.

A mixture of air and biogas enters the cylinder through the inlet valve. It is assumed that the volume of this mixture is equal to the volume of the inlet air in the reference measurement. The biogas cylinder is at high pressure therefore a reduction valve is used to decrease pressure and an expansion-pressure control is used to release the gas when the inlet valve is opened; it is a similar technology as used for LPG systems in cars. A venturi is connected to the air inlet of the engine which allows the biogas to flow into the cylinder together with the air as explained in chapter 2. A gas meter is placed between the expansion pressure control and the control valve.

Input variables

Fraction of methane in biogas α

Methane concentration in biogas is considered an input variable although it cannot be altered during an experiment since premixed bottles are used as described before.

Electrical load

Electrical load is an input variable that is imposed onto the system with a resistive load.

Biogas flow

The volume flow of biogas can be altered by a control valve between the gas meter and the venturi.

Output variables

Thermal efficiency

Thermal efficiency is obtained from electrical load, gas use and jatropha oil use are measured.

Volumetric efficiency

Dual fuel volumetric efficiency is expected to be lower than reference volumetric efficiency. With reference volumetric efficiency and gas use the dual fuel volumetric efficiency can be obtained. The total volume flow of the intake mixture is supposed to be constant and gas flow is measured, with these two values the air flow under dual fuel operation is obtained, dual fuel volumetric efficiency is now computed.

Air-excess ratio

Air-excess ratio is obtained from the UEGO sensor measurement for oxygen content in the exhaust gas. The next section explains how fraction of oxygen in the exhaust gas is transferred to air-excess ratio. Expression 2.25 for air-excess ratio that is derived in chapter 2 is also used to compute a value for air-excess ratio. These two are compared.

3.4 Experimental procedure

This section describes how the tests are conducted for both the reference case and the dual fuel case.

3.4.1 Reference measurement

For the reference measurement the air box and turbine meter are connected to the air inlet. In the reference measurement the electrical load is the only input variable; electric heaters are used as load. There are six heaters that can be switched between 1 and 2 kW. In this way a measurements series can be made from 1 to 12 kW with steps of 1 kW. For each load fuel use is measured over time; each measurement takes approximately five minutes. With measuring the total range of the generator output a characteristic for thermal efficiency is obtained. Volumetric efficiency is measured with the turbine meter and air-excess ratio with the UEGO sensor. Volumetric efficiency is only measured during the reference measurement. Jatropha oil and diesel are tested as fuel, this is done to see what the differences are between jatropha operation and diesel operation.

3.4.2 Dual fuel measurements

For the dual fuel measurement the heaters are not used as a load because the outside temperature did not allow that, instead an adjustable 3-phase resistive load is used. As discussed in section 2.1 the dual fuel engine performance deteriorates fast compared to oil operation under low load condition because the mixture is too lean for methane. Furthermore thermal efficiency decreases very fast under low load conditions. For these two reasons it is decided not to test dual operation under low load and idle conditions. The amount of gas and oil that is available is limited; therefore measurement series are only conducted for 3 different loads. Initially it would have been 6, 9 and 12 kW. The cooling system of the engine that was used appeared not to be sufficient to operate on full load for dual fuel operation. Therefore, measurements were carried out for 6, 8 and 10 kW, except for the measurements with biogas that only contained 40% methane. In this case there was not enough biogas to carry out all three measurements; tests were only performed at 6 and 10 kW.

The amount of gas that is added to the inlet air is also an input variable. The gas is added as described in section 3.3.2; with a control valve the amount of gas is regulated. The gas flow cannot be controlled very accurately; the design of the control valve does not allow for this. The number and size of holes in the venturi dictate the flow rate. The venturi used does not allow for the complete spectrum of gas flows to be available. With more holes in the venturi the smallest possible gas flow that is set with the control valve is relatively high. The other way around, when less holes are available the largest possible air flow, which exists when the control valve is completely open, is not large enough. Therefore a trade off has to be made between fewer holes and less flow or more holes and more flow. It is decided to open up more holes so a larger flow is possible; with a larger flow the maximum heat release of the biogas might be obtained.

Dual fuel volumetric efficiency is not measured directly. It is obtained with the gas flow and the assumption that the volume of intake mixture is constant. The oxygen concentration in the exhaust gas is measured with an UEGO sensor. It is measured to obtain a value for air-excess ratio.

3.5 Measurements and data processing

This section describes how input and output variables are measured and what equipment is used. It also describes how measurement data is transferred to output variables. Finally measurement inaccuracy is discussed.

Electrical power

An active load is connected to each phase; voltage and current are measured in order to calculate electrical power. A synchronous generator has a fixed power factor of 0.8. This means that 80% of the apparent power is active power and 20% is reactive power; a 12 kW output means that the apparent power is 15 kVA. The voltage is measured with a multimeter, frequency is measured with an oscilloscope. Current is measured with a current probe, the fluke 80i-110s AC/DC current probe is used with a specified accuracy better than 3%. The accuracy of the multimeter is not known therefore it is estimated that the approximate inaccuracy of the power measurement is 3 %.

Biogas content

Simulated biogas is used; it consists of bottled premixed methane and carbon dioxide. For each bottle the content is analysed with a relative measurement inaccuracy of 2%. Results from the analyses are shown in Table 3.5.

Table 3.5 Analyses results simulated biogas

CH_4 (%)	CO_2 (%)	α
100	0	1
70.0	30.0	0.7
59.9	40.1	0.599
49.7	50.3	0.497
39.9	60.1	0.399

Fuel use

Fuel use is measured to find out what the thermal efficiency is and to see what fraction of the total heat release comes from the oil and which comes from the gas (ε).

Vegetable oil use is measured by placing the oil tank on a measurement scale; oil use is measured over time. Since oil use is measured as a difference the systematic error approaches zero. Therefore only a random error and an error in the time measurement can occur. Error in time measurement is very small. The inaccuracy of the oil flow measurement is estimated to be 0.5%.

The gas flow is measured with a natural gas meter that is usually used in homes. Fuel use is measured in cubic meters over a certain amount of time. Pressure drop and temperature are required since the mass of the consumed gas has to be established from the volume flow.

The pressure drop inside the gas meter is established on forehand, as a function a gas flow. Appendix F shows the measurement data for the pressure drop measurement. Temperature of the gas cannot be measured; it is near 0˚C since the reduction valve on the bottle cools down to (near) freezing. The overall accuracy of the gas flow meter is not known but is estimated to be 1.5%. The systematic error in the flow measurement approaches zero; it is assumed that the inaccuracy of the flow meter is similar to that of the measurement scales, it is 0.5%. The temperature of the gas has an estimated inaccuracy of 1%. This results in a total inaccuracy of the gas flow measurement of 1.5%.

Rotational speed
Rotational speed is measured with a revolution counter. A reflecting sticker is attached to the flywheel, by pointing the light of the meter on the flywheel rotational speed is measured. Inaccuracy is estimated to be 1%.

Volumetric efficiency from turbine meter
The air flow is measured with a turbine meter. In a turbine meter a fluid passes through a rotor; the angular velocity of the rotor is proportional to the speed of the flow. An instromet turbine gas meter Q75 with a diameter of 100mm and maximum flow of 650 m^3/h is used. The engine runs at approximately 2000 rpm and takes 1 litre of air each cycle; 60 m^3/h air passes through the turbine meter. This is less than 10% of the maximum flow; therefore the measurement error is 2% [31]. In order to transfer the volume of intake air to a mass of intake air ambient pressure, pressure drop and temperature of the air are measured. These also give measurement inaccuracy estimated at 1%. Total inaccuracy of the air-flow measurement is 3%.
The four stroke single cylinder engine takes in air only 25% of the time, resulting in large fluctuations in the incoming airflow. The rotor inside the turbine meter continues to rotate due to inertia during the 75% of the time when the intake valve is closed. This results in large deviations in the airflow measurement. In order to smooth these fluctuations an air box is placed between the turbine meter and the engine. Not much literature is available on air box dimensions; one, rather old, article provides a way to calculate air box dimension. According to this method the air box needs to be 1.6 m3 when an orifice plate is used; this calculation can be found in Appendix G. Since a turbine meter is less sensitive for fluctuations than an orifice plate, it is expected that a smaller air box can be used. An air box of approximately 1 m3 was available and it proved sufficient, because air-excess ratio obtained from these measurements and air-excess ratio derived from the fraction of oxygen in the exhaust gas corresponded within the boundaries of the estimated errors. The measurements performed with the UEGO sensor as described in the next paragraph gave similar results for air-excess ratio.
Volumetric efficiency depends on the temperature of the air; an increase in air temperature results in a decrease of the density. A decreased air density results in a decrease of mass of air taken into the system with the same volume. The reference volumetric efficiency is obtained with equation 3.1.

$$\eta_v = \frac{m_{air}}{\rho_{ref} \cdot V_d} = \frac{V_N}{V_d} \qquad \textbf{3.1}$$

V_N is the volume taken in normal cubic meters; this is the actual air taken into the system compensated for ambient temperature and pressure as presented in equation 3.2.

$$V_N = \frac{V_{act} \cdot p_{amb} \cdot T_0}{T_{amb} \cdot p_0} \qquad \textbf{3.2}$$

T_0=273 K

p_0=101.325 kPa
In this equation V_{act} is the actual volume of air taken in not compensated for with ambient temperature and pressure as measured with the turbine meter. The ambient temperature and pressure are also measured during the experiment.
Since reference volumetric efficiency depends on the weather it is decided to calculate a non-compensated reference volumetric efficiency. This is calculated using equation 3.3.

$$\left(\eta_{v,ref}\right)_{nc} = \frac{V_{act}}{V_d} \qquad \textbf{3.3}$$

This gives approximately equal values for each measurement since it is assumed that the engine takes in a fixed volume each inlet stroke.

Air-excess ratio from turbine meter
The volumetric efficiency for the reference measurement as described above results in a value for volume of the intake mixture. It is assumed that this is a constant value. Under dual fuel conditions the total volume of the intake mixture, air and biogas, is the same as that in the reference measurement. The volume of intake air is not measured during the dual fuel tests but the volume of gas is measured and the total volume of the intake mixture is known. The volume of the intake air results from the difference between these two values.
Air-excess ratio is obtained with expression 3.4, which is derived section 2.2.2.1. ξ is the fraction of air in the intake mixture, this is calculated with the measured values for biogas flow and total intake mixture flow. θ is the mass fraction of oil in the total fuel mixture, which is obtained from the values for biogas flow and oil flow during dual fuel operation. σ is the mass fraction of methane in the biogas, this is a fixed value per bottle of simulated biogas that is used. Air-excess ratio is obtained with these values. In order to verify the derived expression for dual fuel air-excess ratio, the value obtained for air-excess ratio in this way should be equal to that obtained with the UEGO sensor, within the boundaries of the experimental accuracy, at least for reference measurements. The results are discussed in section 4.2.3.

$$\lambda(\xi,\theta,\sigma) = \frac{\dfrac{\xi}{1-\xi}}{\dfrac{\theta}{1-\theta}\cdot 12.36+\sigma\cdot 17.16} \qquad \textbf{3.4}$$

Air-excess ratio from oxygen concentration in exhaust gas

The oxygen concentration in the exhaust flow is measured with an UEGO sensor (Universal Exhaust Gas Oxygen Sensor) placed in the exhaust gas pipe. The UEGO sensor gives a voltage as an output signal which is proportional to the partial pressure of the oxygen in the exhaust gas. A signal conditioning system is used to transfer the voltage output to a readable volume fraction of oxygen. The UEGO sensor is capable of measuring oxygen concentration in the exhaust gases of combustion processes for both lean and rich mixtures. The exact relative error is not known, it is estimated to be similar to that of the turbine meter when a higher load is connected.

The fraction of oxygen in the exhaust gas measured by the UEGO sensor is not a direct measure for the amount of oxygen in the air from the intake mixture that is used for combustion. Part of the oxygen that is used during combustion comes from the oil because vegetable oil contains oxygen. This section describes how air-excess ratio is derived from the UEGO measurement and how fuel oxygen is taken into account for both reference and dual fuel measurement.

The UEGO sensor measures the fraction of oxygen in the exhaust gas $X_{o2,exh}$. An expression for $X_{o2,exh}$ is obtained with the non stoichiometric molar reaction equations for pure oil and dual fuel operation as presented in equation 3.5 and reaction equation 2.20.

$$C_{56}H_{101}O_6 + 78.25 \cdot \lambda_{ref} \cdot O_2 + 78.25 \cdot 3.76 \cdot \lambda_{ref} \cdot N_2 \rightarrow \\ 56CO_2 + 50.5H_2O + 78.25 \cdot 3.76 \cdot \lambda_{ref} \cdot N_2 + (\lambda_{ref} - 1) \cdot 78.25 \cdot O_2 \qquad \textbf{3.5}$$

For pure oil operation the fraction of oxygen in the exhaust gas is expressed in equation 3.6. Air-excess ratio for pure oil operation is obtained with this equation; it is provided in equation

$$X_{O2,exh} = \frac{78.25 \cdot \lambda_{ref} - 78.25}{28.25 + 372.47 \cdot \lambda_{ref}} \qquad \textbf{3.6}$$

$$\lambda_{ref} = \frac{-78.25 - 28.25 \cdot X_{O2,exh}}{372.47 \cdot X_{O2,exh} - 78.25} \qquad \textbf{3.7}$$

A reference measurement is also performed for diesel fuel. Therefore air-excess ratio is also derived from diesel fuel from oxygen fraction in the exhaust gas. A typical average composition for diesel fuel of $C_{12}H_{23}$ is used. Equation 3.8 provides the molar non-stoichiometric reaction equation for combustion of diesel. Air-excess ratio is derived in a similar way in equation 3.9 and 3.10.

$$C_{12}H_{23} + 17.75 \cdot \lambda_{diesel} \cdot O_2 + 17.75 \cdot 3.76 \cdot \lambda_{diesel} \cdot N_2 \rightarrow \\ 12CO_2 + 11.5H_2O + 17.75 \cdot 3.76 \cdot \lambda_{diesel} \cdot N_2 + (\lambda_{diesel} - 1) \cdot 17.75 \cdot O_2 \qquad \textbf{3.8}$$

$$X_{O2,exh} = \frac{17.75 \cdot \lambda_{diesel} - 17.75}{2.75 + 84.49 \cdot \lambda_{diesel}} \qquad \textbf{3.9}$$

$$\lambda_{diesel} = \frac{-17.75 - 5.75 \cdot X_{O2,exh}}{87.94 \cdot X_{O2,exh} - 17.75} \qquad \textbf{3.10}$$

Air-excess ratio for dual fuel operation is derived in a similar fashion as for diesel and pure oil operation. Equation 3.11 provides an expression for oxygen fraction in the exhaust gasses, which is obtained from the reaction equation, the expression for variable S is provided in equation 3.12. Equation 3.13 provides air-excess ratio for dual fuel operation.

$$X_{O2,exh} = \frac{78.25 \cdot S \cdot \lambda_{dual} + 2 \cdot \lambda_{dual} - 78.25 \cdot S - 2}{\frac{1}{\alpha} + 28.25 \cdot S + 372.47 \cdot S \cdot \lambda_{dual} + 9.52 \cdot \lambda_{dual}} \qquad \textbf{3.11}$$

$$S = \frac{\beta}{\alpha \cdot (1 - \beta)} \qquad \textbf{3.12}$$

$$\lambda_{dual} = \frac{-78.25 \cdot S - 2 - \frac{X_{O2,exh}}{\alpha} - 28.25 \cdot S \cdot X_{O2,exh}}{372.47 \cdot S \cdot X_{O2,exh} + 9.52 \cdot X_{O2,exh} - 78.25 \cdot S - 2} \qquad \textbf{3.13}$$

Jatropha oil contains oxygen, which is also used for combustion together with the oxygen from the air. The air-excess ratio that is defined in equation 3.7 and 3.13 does not take this into account. Therefore, air-excess ratio will be adapted so that fuel oxygen is taken into account. Equation 3.14 gives a definition for air-excess ratio that does take fuel oxygen into account. Equations 3.15 and 3.16 give expressions for real air-excess ratio, where fuel oxygen is taken into account, for both reference and dual fuel operation.

$$\Lambda = \frac{n_{02,available}}{n_{o2,required}} \qquad \textbf{3.14}$$

$$\Lambda_{ref} = \frac{78.25 \cdot \lambda_{ref} + 3}{78.25 + 3} \qquad \textbf{3.15}$$

$$\Lambda_{dual} = \frac{78.25 \cdot S \cdot \lambda_{dual} + 2 \cdot \lambda_{dual} + 3 \cdot S}{78.25 \cdot S + 2 + 3 \cdot S} \qquad \textbf{3.16}$$

The value for air-excess ratio as derived in expression 2.22 should give the same result as the one derived here in expression 3.16. In both expressions fuel oxygen of the jatropha oil is taken into account. In the expression for λ in 2.22, fuel oxygen is taken into account in the value for stoichiometric air-fuel ratio for jatropha oil combustion (12.36).

3.5.1 Measurement accuracy

The (estimated) relative errors per measured parameter are summarised in Table 3.6. For each performance parameter the overall relative measurement error is established. Table 3.7 gives which measurement errors occur for the three output variables for both the reference and the dual fuel case. For some measurements measurement error is obtained from the manual from the equipment and for some it is estimation. Therefore it is difficult to say what the exact measurement order is. It can be said that is more than 3% and less than 10%.
The reference measurements without addition of biogas have better measurement accuracy than the dual fuel measurements, because the addition of biogas results in the use of more measurements equipment which introduces more inaccuracies.

Table 3.6 Relative errors

Measurement	inaccuracy (relative error)
power	3%
methane fraction	2%
oil use	0.5%
gas flow	1.5%
rev. counter	1%
turbine meter	3%
UEGO sensor	3%

Table 3.7 Measurement inaccuracies

reference	dual fuel
thermal efficiency:	**thermal efficiency:**
- oil use (0.5%)	- oil use (0.5%)
- power (3%)	- power (3%)
	- gas use (1.5%)
	methane fraction (2%)
volumetric efficiency (turbine)	**volumetric efficiency (turbine)**
- turbine(3%)	- turbine (3%)
- rev. counter (1%)	- rev. counter (1%)
-oil use (0.5%)	- oil use (0.5%)
	- gas use (1.5%)
	- methane fraction (2%)
air-excess ratio (turbine)	**air-excess ratio (turbine)**
- turbine (3%)	- turbine (3%)
- rev.counter (1%)	- rev.counter (1%)
- oil use (0,5%)	- oil use (0,5%)
	- gas use (1.5%)
	- methane fraction (2%)
air-excess ratio (UEGO)	**air-excess ratio (UEGO)**
-UEGO (3%)	-UEGO (3%)
- rev. counter (1%)	- rev. counter (1%)
-oil use (0.5%)	-oil use (0.5%)
	- gas use (1.5%)
	- methane fraction (2%)

4 Results and analyses

This chapter describes the results of the experiments conducted on the diesel engine-generator set. The results of the three output variables, thermal efficiency, volumetric efficiency and air-excess ratio are discussed. Results of the reference measurements for pure diesel operation and pure jatropha oil operation are discussed before results of the dual fuel tests. Due to the limited availability of simulated biogas and jatropha oil the test series could only be performed once for each load. This means that the result are not reproduced, which means that it is not certain if the experiments would give the same results when they are executed again.

4.1 Reference measurement

For the reference measurement jatropha oil and diesel are both tested in the generator set for thermal efficiency, volumetric efficiency and air-excess ratio.

4.1.1 Thermal efficiency

During these tests electrical heaters are used as a load; an engine characteristic for thermal efficiency is obtained over the complete output range of the generator. Table 4.1 gives the results of the thermal efficiency for both diesel and jatropha oil. The relative measurement error is 3.2%.

Table 4.1 Thermal efficiency reference measurement

Diesel		Jatropha	
P (kW)	η	P (kW)	η
12.89	32.4%	13.26	32.9%
11.97	32.6%	12.18	32.8%
10.92	32.8%	11.01	32.3%
9.89	32.5%	9.98	32.2%
8.76	31.4%	8.79	30.9%
7.61	30.7%	7.69	29.9%
6.46	28.4%	6.55	28.2%
5.27	25.9%	5.42	25.9%
4.13	22.7%	4.29	23.0%
3.03	18.6%	3.08	18.8%
2.01	14.4%	2.03	14.4%
1.00	8.4%	0.99	8.3%

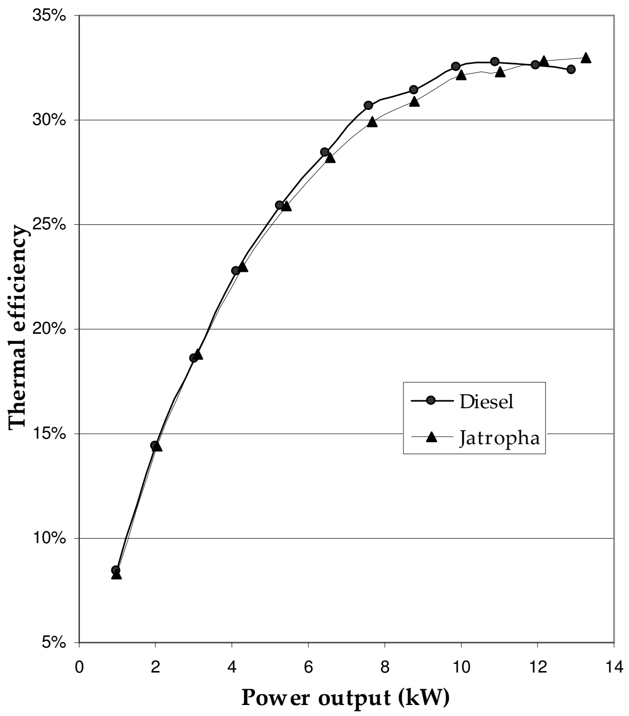

Figure 4.1 Thermal efficiency reference measurement

Figure 4.1 shows thermal efficiency as a function of power output; it shows that thermal efficiency decreases drastically under half load conditions. The inaccuracy for thermal efficiency for the reference measurement is 3.5%. Thermal efficiency characteristics for both fuels do not deviate significantly; both characteristics are the same within the expected error. This means that the use of pure jatropha oil that is not preheated, instead of diesel, does not deteriorate thermal efficiency.

4.1.2 Volumetric efficiency

For the reference tests the non-compensated volumetric efficiency is obtained, as described in section 3.5. The non-compensated volumetric efficiency should give similar results for each measurement, since it is assumed that the engine takes in a fixed volume each inlet stroke. Table 4.2 shows results from tests performed on two different days, with different weather and consequently different ambient pressure and temperature. It shows that the $\eta_{v,ref}$, which is compensated for ambient pressure and temperature gives different values on two different days. The non-compensated $(\eta_{v,ref})_{NC}$ gives a value that is almost identical; deviation is less than 1%. This shows that the non-compensated volume of intake air does not change between different days and different weather conditions.

Table 4.2 Volumetric efficiency

P(kW)	V_{air} (m^3/cycle)	$\eta_{v,ref}$	$(\eta_{v,ref})_{NC}$	Vair (m^3/cycle)	$\eta_{v,ref}$	$(\eta_{v,ref})_{NC}$
12.99	1.036E-03	89.49%	94.83%	1.030E-03	90.27%	94.22%
11.94	1.035E-03	89.30%	94.67%	1.036E-03	91.06%	94.75%
10.97	1.039E-03	89.14%	95.06%	1.033E-03	91.34%	94.48%
9.98	1.046E-03	89.21%	95.67%	1.034E-03	92.05%	94.62%
8.74	1.041E-03	89.45%	95.23%	1.042E-03	90.87%	95.30%
7.62	1.039E-03	89.56%	95.07%	1.049E-03	91.62%	96.02%
6.52	1.044E-03	89.84%	95.50%	1.045E-03	91.89%	95.65%
	Average	89.43%	95.15%	Average	91.30%	95.01%

Table 4.3 gives the non-compensated volumetric efficiency for both jatropha oil and diesel operation. It shows that the engine has slightly higher volumetric efficiency when diesel is used as a fuel. This is a 0.6% difference, which cannot be considered a significant difference since the measurement inaccuracy is 3.2% for these tests.

Table 4.3

Fuel	$\eta_{v,ref\ NC}$
Jatropha oil	95.1%
Diesel	95.7%

4.1.3 Air-excess ratio

Air excess ratio is measured with the UEGO sensor as described in section 3.5. It is also derived from volumetric efficiency measured with the turbine meter.

UEGO sensor

With the UEGO sensor the fraction of oxygen in the exhaust is measured. Air-excess ratio is derived from this measurement as explained in section 3.5.

Turbine meter

The mass of air that is taken in per second is measured with the turbine meter; the mass of oil that is used per second is measured. These two values give the actual air-fuel ratio and the stoichiometric air-fuel ratios for both fuels are known. Equation 4.1 gives air-excess ratio.

$$\lambda = \frac{\left(\frac{A}{F}\right)_{act}}{\left(\frac{A}{F}\right)_{stoich}} \qquad \textbf{4.1}$$

Table 4.4 gives air-excess ratio for both diesel and oil measurements, measured with UEGO sensor and turbine meter. For jatropha oil the difference between UEGO and turbine is larger than for diesel. For jatropha differences are in the order of 10% for half load to full load operation, at low load and idle operation it becomes less. Reference measurements for low load and idle running are performed for completeness; dual fuel tests will be performed between half load and full load condition.

Table 4.4 Air-excess ratio reference measurement

Jatropha oil			Diesel		
P (kW)	λ UEGO	λ turbine	P(kW)	λ UEGO	λ turbine
12.99	1.40	1.58	13.07	1.43	1.50
11.94	1.52	1.70	11.94	1.57	1.64
10.97	1.70	1.87	11.04	1.74	1.81
9.98	1.87	2.08	9.99	1.96	2.03
8.74	2.07	2.27	8.76	2.14	2.23
7.62	2.30	2.54	7.64	2.39	2.51
6.52	2.58	2.81	6.57	2.67	2.76
5.44	2.90	3.19	5.38	2.99	3.10
4.33	3.35	3.54	4.18	3.43	3.50
3.07	3.88	4.11	3.09	3.93	3.89
2.07	4.60	4.79	2.07	4.66	4.59
0.99	5.56	5.68	1.00	5.76	5.63

Figure 4.2 shows air-excess ratios as a function of power output for both UEGO measurements and for turbine meter measurements for pure jatropha oil operation. Figure 4.3 shows the output of the same test, but than for diesel operation. Dual fuel tests will only be performed between half load and full load, therefore only these values are important. For completeness the complete range is provided. Air-excess ratio obtained with the turbine meter is consistently higher than that obtained from UEGO measurement. Figure 4.3 shows that for diesel operation air-excess ratio measured with the UEGO sensor and turbine meter are almost identical. For jatropha oil operation the two measurement methods deviate more, however the deviation falls within the boundaries of the measurement inaccuracy of 3.2%.

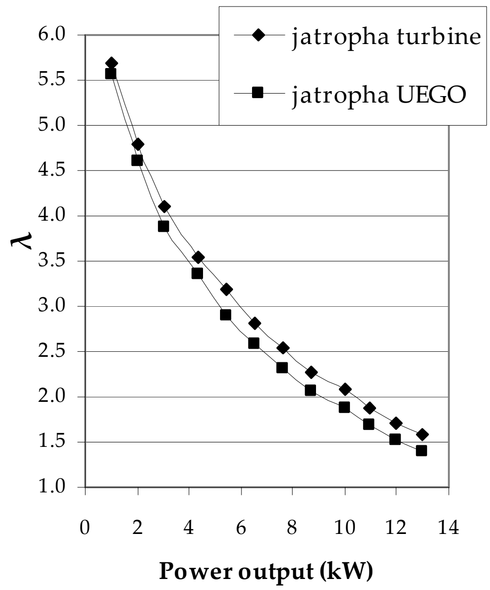

Figure 4.2 Air-excess ratio jatropha oil

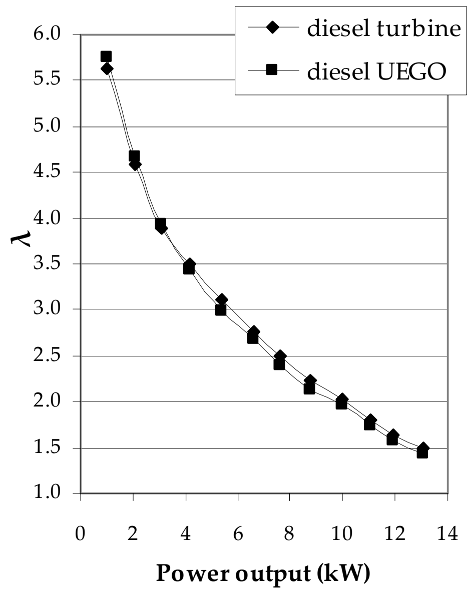

Figure 4.3 Air-excess ratio diesel

4.2 Dual fuel measurements

This section describes the results of the dual fuel tests. The goal of the experiments is to assess the effect of dual fuel operation on three performance parameters, namely thermal efficiency, volumetric efficiency and air-excess ratio, but also to find the limits of dual fuel operation. First a general introduction into the experiences with dual fuel operation is provided, then the results for the three performance parameters are described; this section will finish with an explanation of the experienced limits of dual fuel operation and the occurrence of knock.

For the dual fuel experiments 4 different qualities of biogas and pure methane are tested as a gaseous fuel. The input variables are fraction of methane in the gaseous fuel used, electrical load and fraction of gas in the inlet mixture. The fraction of gas in the inlet mixture is controlled with a control valve. The electrical loads are imposed on the system with a resistive load. For pure methane experiments are carried out at 6, 7, 8, 9, and 10 kW. Because less biogas is available measurements for biogas cannot be conducted at all these loads. For biogas measurements are conducted at 6, 8, and 10 kW. The simulated biogas that contains 40% and 60% carbon dioxide was only tested at 6 and 10 kW, because not enough biogas was available to perform tests at all three loads.
The coolant temperature was not measured during the experiments but it was observed that the engine had a higher temperature during dual fuel operation compared to pure oil operation. This is probably due to a higher combustion temperature of methane. The cooling system of the used engine proved not sufficient, resulting in an overheated engine. The cooling fluid started boiling and coolant vapour came out of the cooling system. For this reason it was decided not to perform tests at full load for dual fuel operation. This does not mean that dual fuel operation results in an overheated engine in each engine. The cooling system of this engine was adapted as discussed in chapter 3, and the cooling fluid is not pumped through the system but has to reach the radiator by convection. The combination of these two probably resulted in overheating; another engine with a better cooling system will probably not overheat under dual fuel operation.

The engine that is used during the experiment has a fixed rotational speed around 2000 rpm. Under dual fuel operation the rotational speed of the engine increases slightly; the quantity of the increase depends on the amount of methane that is added and on the electrical load. Higher methane fraction in the total fuel mixture results in larger increase in rotational speed. A higher electrical load also results in a larger increase of the rotational speed. The rotational speed increased with a maximum of 2.5%. Consequently output frequency increases with a similar percentage. Initially output voltage increases with the increase in rotational speed but when too much methane is added and the combustion process runs less smoothly the output voltage decreases slightly again. The addition of too much methane results in a sort of knocking behaviour. This mainly happens when pure methane or biogas with 70% methane is added. The maximum increase in output voltage is approximately 2% of the initial voltage. Table 4.5 provides measurement data for rotational speed and output voltage at 10 kW load for biogas with α=0.7 and of biogas with α=0.5. Oil fraction of the total fuel mixture ε is also provided to show how much of the heat release originated from oil and how much from methane. The increase in output voltage results in similar increase in output power; the current remains constant since the electrical load demands a constant current. at the end of this section there is an elaboration on knocking behaviour. At the end of this section there is an elaboration on knocking behaviour.

Table 4.5 Rotational speed dual fuel operation

	α=0.7			α=0.5	
RPM	**V(V)**	**ε**	**RPM**	**V(V)**	**ε**
1946	233.3	100.0%	1938	230.8	100.0%
1956	233.4	72.7%	1952	230.8	54.7%
1956	233.5	68.2%	1959	231.8	45.7%
1956	233.5	65.2%	1961	232.6	41.2%
1966	233.8	58.2%	1963	233.3	39.3%
1956	233.8	54.8%	1964	234.5	36.6%
1961	234.3	55.5%	1967	233.7	36.6%
1966	234.6	49.9%	1967	234.4	36.1%
1969	235.6	47.2%	1971	234.4	34.8%
1969	235.0	43.6%	1971	234.9	36.8%
1967	234.7	41.3%	1971	235.4	35.3%
1970	234.5	39.7%			
1970	234.4	38.4%			

The increase in power output due to the addition of methane is probably due to the fact that methane combusts faster and at higher temperature resulting in a faster heat release. Faster combustion results a higher power output. Figure 4.4 gives rotational speed as function of heat release fraction of oil (ε). Figure 4.5 shows the increase in output voltage as function of heat release fraction from oil (ε).

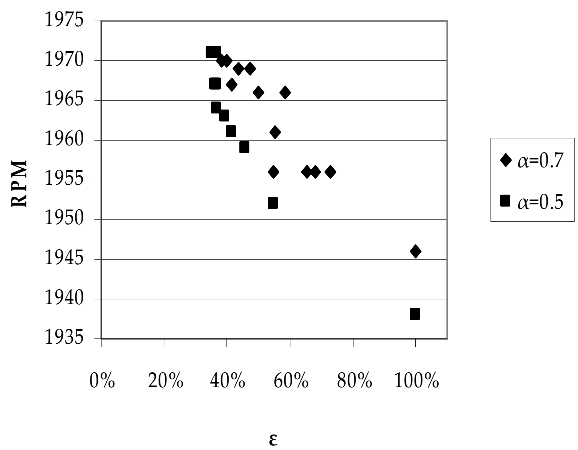

Figure 4.4 Change in rotational speed under dual fuel

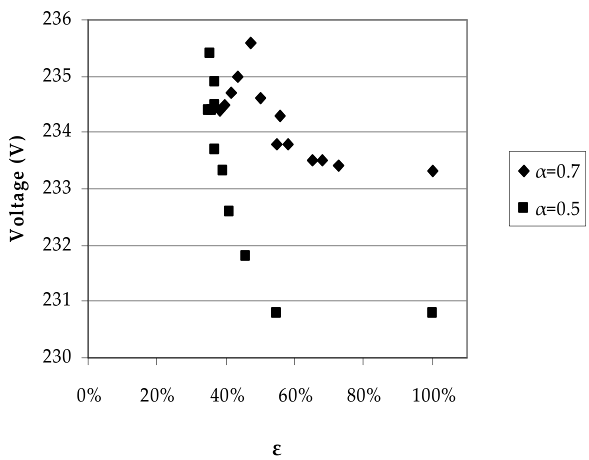

Figure 4.5 Change in output voltage under dual fuel

4.2.1 Thermal efficiency

Thermal efficiency is obtained from the test results for fuel use (both gas and oil) and power output. The reference measurements showed that thermal efficiency decreases with decreasing load. Figure 4.6 shows the thermal efficiency for dual fuel operation at 6, 8 and 10 kW[4] load as a function of the heat release fraction from oil (ε); the smaller ε the more heat originates from methane. Figure 4.6 also shows that there is a large interval between the reference measurement at ε=1 and the first at dual fuel conditions. This is the result of the design of the venturi and control valve. This combination does not allow less biogas to be mixed into the inlet air.

The biogas that is used contains 59.9 % methane, the rest is carbon dioxide. It shows that at a relatively high load, of 10 kW, thermal efficiency does not decrease with increasing biogas concentration. At a lower load thermal efficiency decreases with increasing heat release from methane in biogas. The decrease of efficiency at lower load under dual fuel conditions is probably due to the surplus of oxygen at low load; the mixture becomes very lean. Methane does not combust properly under lean conditions; combustion is too slow which can result in a slower heat release and finally into incomplete combustion resulting in deteriorated thermal efficiency.

The other three qualities of biogas that were tested show similar results, which are presented in Appendix H. The use of pure methane as a gaseous fuel also results in a decrease in thermal efficiency when a higher fraction of the heat release originated from methane (smaller ε). Figure 4.7 gives the results for thermal efficiency of dual fuel operation with pure methane as a function of heat release fraction from oil ε at several loads. In this case thermal efficiency decreased even under higher load conditions as 9 or 10 kW, though the decrease is larger at lower load conditions as 6 or 7 kW. It is difficult to explain this phenomena but it might be due to the presence of carbon dioxide in the biogas. The addition of carbon dioxide results in more oxygen being replaced in the intake mixture at an equal ε, compared to pure methane where no carbon dioxide is added to the intake mixture. Therefore, the use of biogas results in less oxygen in the intake mixture creating more favourable conditions for methane combustion. In other words, for dual fuel combustion with pure methane too much oxygen is available resulting in too slow combustion process of methane, consequently thermal efficiency drops even at 10 kW load.

For dual fuel operation with biogas thermal efficiency remains unchanged for the higher load of 10 kW. At lower load operation thermal efficiency decreases with increasing heat release from methane. The decrease is in the order of 10%. Heat release of methane combustion decreases when too much oxygen is available. Too much oxygen results in a slower heat release, consequently thermal efficiency drops.

[4] The loads of 6, 8 and 10 kW are the loads as set on the resistive load. Because the resistive load also feeds a ventilator the actual load is slightly higher. 6 kW becomes 6.4 kW; 8 kW becomes 8.8 kW; and 10 kW becomes 10.8 kW. These are approximate values because power output increased a little under dual fuel operation.

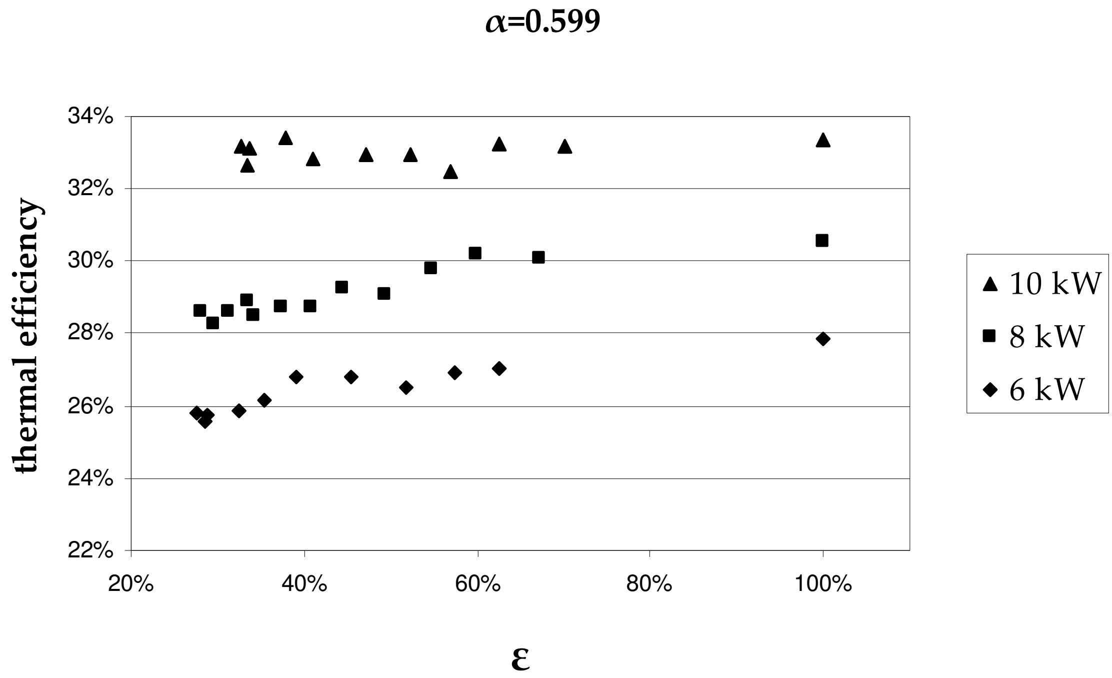

Figure 4.6 Dual fuel thermal efficiency for α=0.599

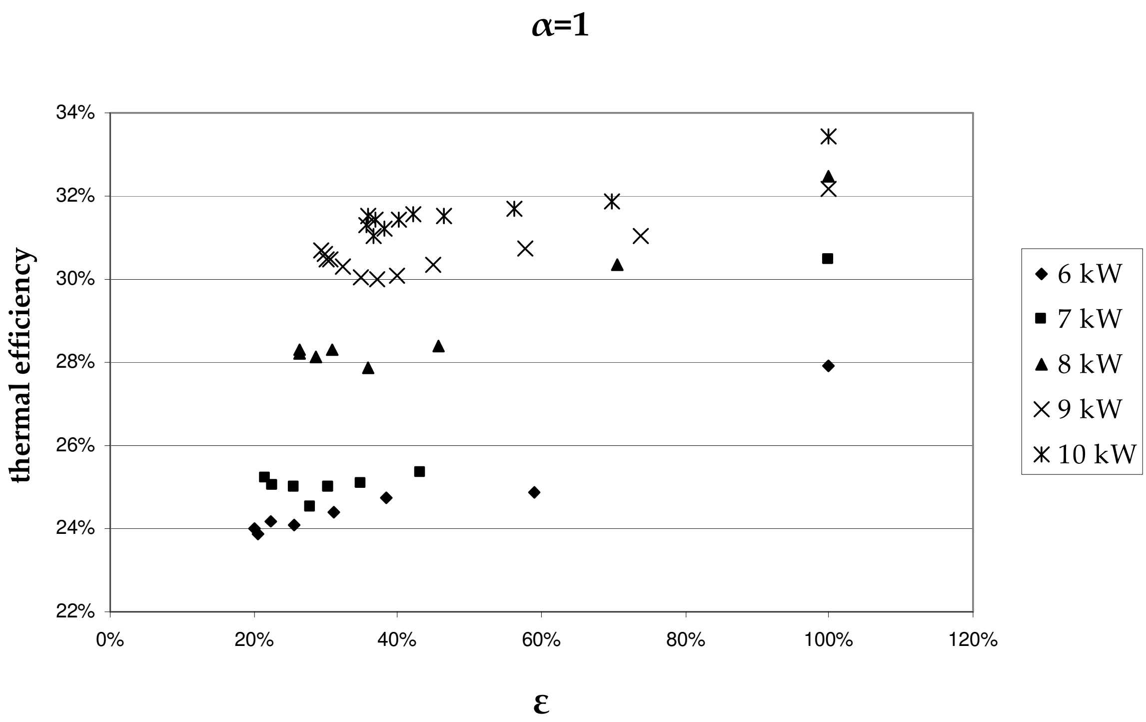

Figure 4.7 Dual fuel thermal efficiency for α=1

4.2.2 Dual fuel volumetric efficiency

For the dual fuel measurements the air box and turbine meter are removed for safety as explained in section 3.3.2. Therefore, volumetric efficiency can only be obtained with the assumption that the total volume of the intake mixture does not change between oil-only operation and dual fuel operation. Therefore, total volume of the intake mixture is known and volume of the biogas is measured, together this results in the volume (and mass) of the intake air. Dual fuel volumetric efficiency is computed.

Dual fuel volumetric efficiency decreases with increasing biogas fraction in the intake air, because air is replaced by biogas. Figure 4.8 gives the test results for volumetric efficiency for biogas with 59.9% methane at 6, 8, and 10 kW. Dual fuel volumetric efficiency is plotted as a function of heat release fraction from oil ε. A decrease in ε means an increase in methane in the total fuel mixture. It shows that increase in methane fraction results in a decrease in volumetric efficiency. ε is a heat release fraction, which means that at a higher load a similar value for ε gives more biogas in the intake mixture. This implies that at a higher load the inclination of the plot should be larger. Figure 4.8 shows that this is the case; the results for 10 kW load give a larger inclination than those for 8 and 6 kW. The initial values for volumetric efficiency at ε=1 are different for the three different loads due to differing weather conditions. For example a higher ambient temperature of the air results in a lower mass of air taken into the system, consequently decreasing volumetric efficiency. For this reason the lines of 10 kW load and 6 kW load cross. If they would have had the same initial volumetric efficiency they would not have crossed. Appendix I provides results of tests for the other three qualities of biogas, these results are similar.

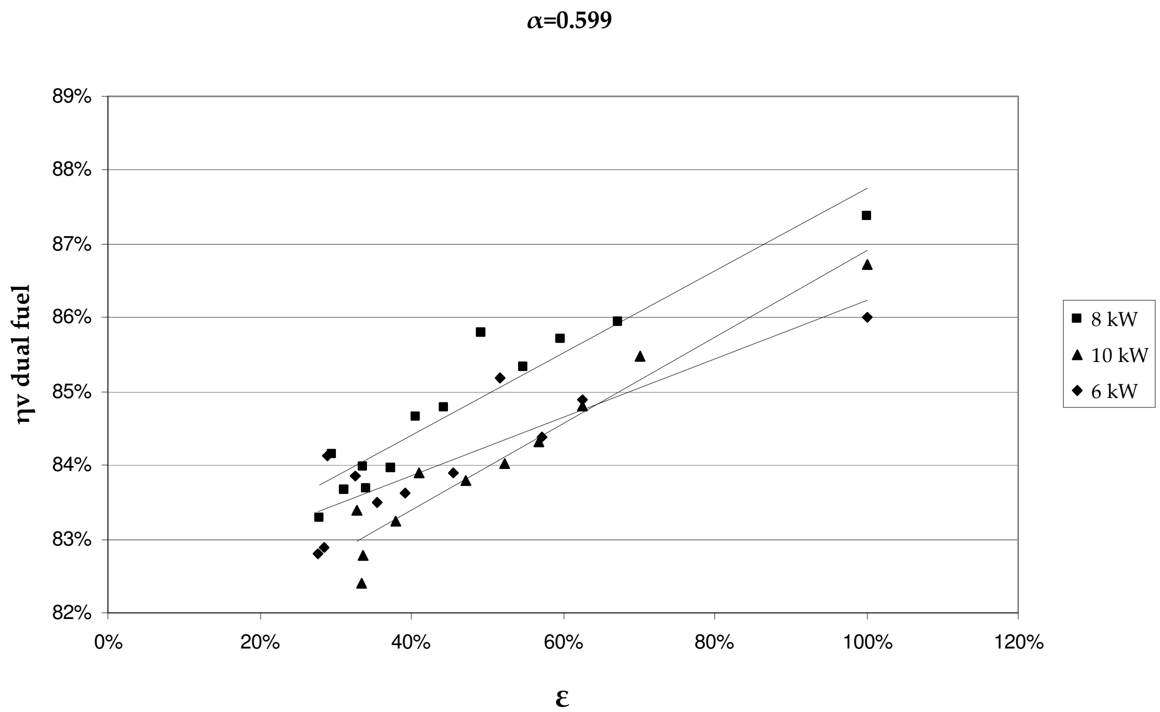

Figure 4.8 Dual fuel volumetric efficiency at α=0.599.

In chapter 2 an expression for dual fuel volumetric efficiency is derived. Figure 2.4 provides a plot that gives a prediction of volumetric efficiency as a function of air fraction in the intake mixture (ξ,). This showed that when volumetric efficiency is plotted in this manner that a higher methane fraction results in a larger inclination. Figure 4.9 shows the results from the experiments in a similar fashion. For this plot volumetric efficiency is calculated with equation 2.36.
Volumetric efficiency is obtained for four different qualities of biogas at a 10 kW load. In the prediction it was assumed that the reference volumetric efficiency is constant. For the tests this cannot be constant since it depends on the ambient pressure and temperature and therefore on the weather condition on the day of the test. Therefore the volumetric efficiency at ξ=1 (only oil) differs between the 5 tests but the inclination of the plots shows similar results as the prediction in chapter 2. Pure methane results in the highest inclination, when biogas is used inclination is lower. Table 4.6 shows the inclination for each line; it shows that the inclination is approximately twice as high for pure methane as for biogas. Between the four qualities of biogas the differences are not significant except biogas with 50% methane shows a larger deviation with the others, which cannot be explained. It is probably due to measurement errors in this test series or in that with biogas with 40% methane.

Table 4.6 Inclination

α	inclination
1	1.8248
0.7	1.10115
0.6	0.9035
0.5	0.7372
0.4	0.9089

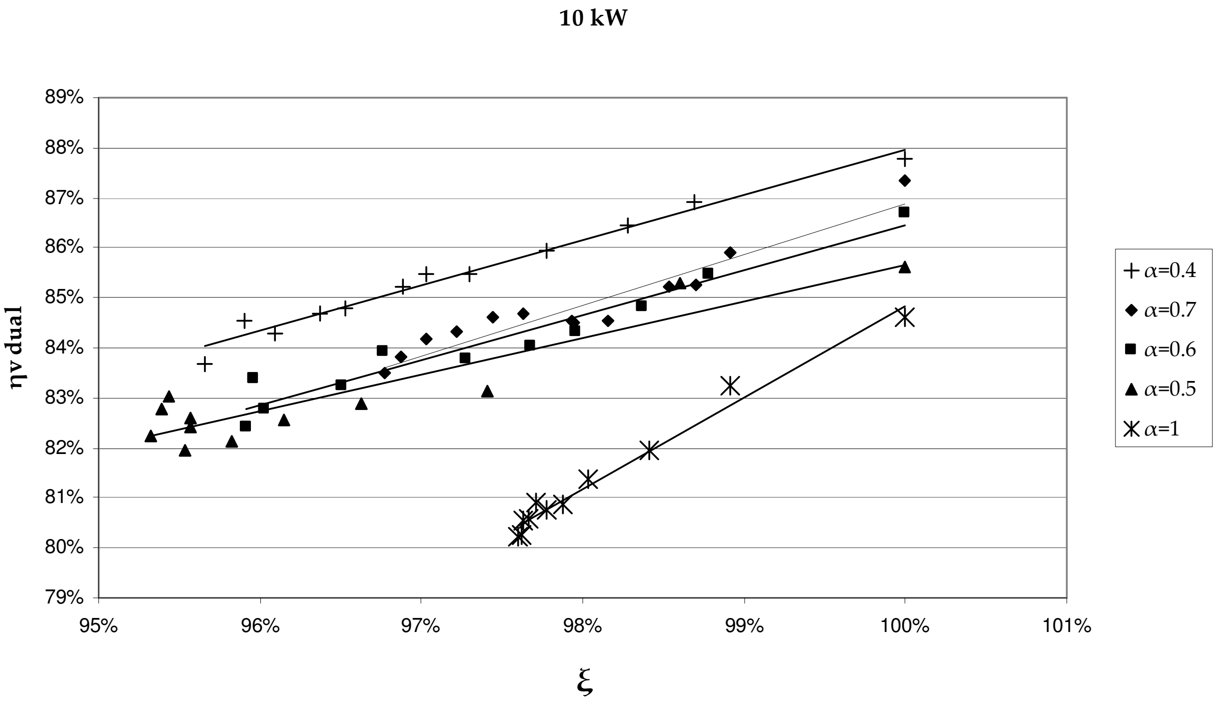

Figure 4.9 Dual fuel volumetric efficiency at 10 kW for several biogas qualities

4.2.3 Air-excess ratio

Two values for air-excess ratio are obtained, one with the UEGO sensor and one with the turbine meter measurement from the reference case. For the latter it is assumed that intake mixture volume (as measured with the turbine meter) remains constant as described in section 3.5. The procedure of transferring the UEGO output to air-excess ratio is explained in section 3.5.

Figure 4.10 shows the results for air-excess ratio as a function of heat release ratio ε for both the UEGO sensor data and the computations from the turbine meter data for biogas with a methane fraction of 0.599. Appendix J shows the results for the other biogas qualities. For dual fuel tests, air-excess ratio obtained from turbine meter measurements is consistently higher than those for UEGO measurement. This is the same as in the reference tests.

An increase in the biogas fraction in the total fuel mixture results in a decrease in air-excess ratio, because biogas replaces air in the intake mixture. A decrease in air fraction in the intake mixture results in a decreased value for actual air-fuel ratio, consequently air-excess ratio decreases.

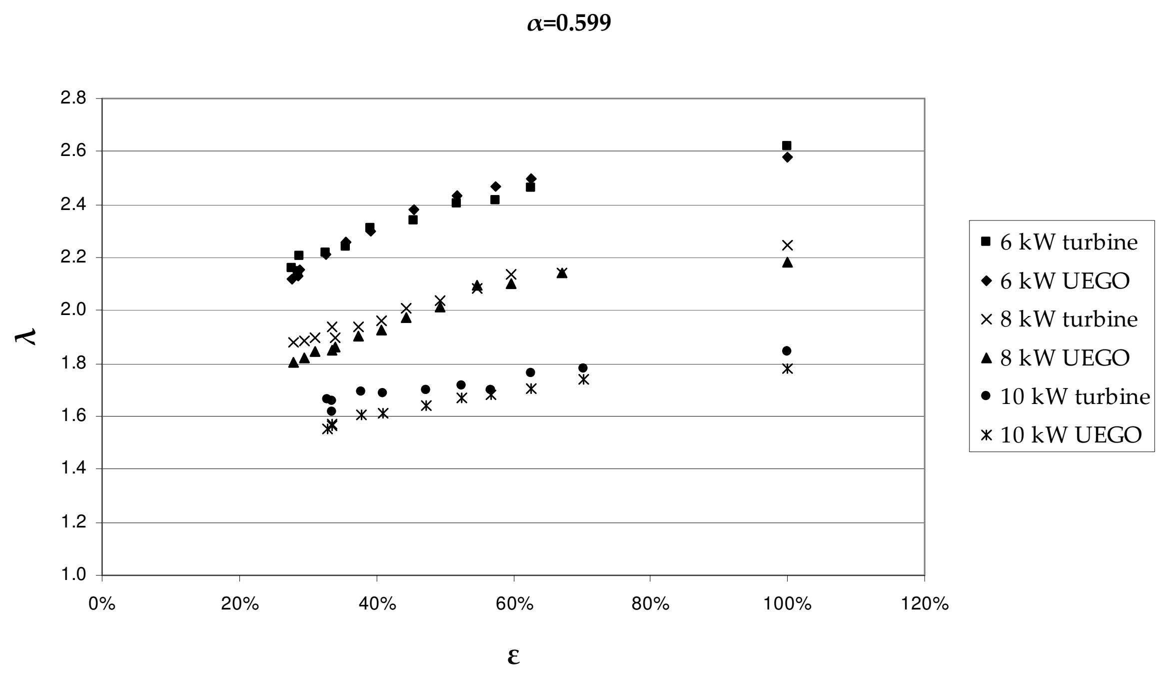

Figure 4.10 Air-excess ratio for biogas with α=0.6.

Figure 4.11 gives air-excess ratio obtained with the UEGO measurement as a function of ε, at a 10 kW load for the three different qualities of biogas and pure methane. Biogas with 70% methane is not presented here because during this test the UEGO sensor was blocked and did not give a good output. Figure 2.3 in section 2.2.2.1 gives a prediction of λ as a function of ε for several fraction of methane in the biogas. It is expected that more CO2 in the biogas results in a larger decrease in air-excess ratio. Figure 4.9 in the previous section showed that more CO2 resulted in a larger decrease in volumetric efficiency. A decrease in volumetric efficiency means that less air is taken into the system. An increase in CO2 in the biogas should therefore result, at the same ε, in a larger decrease in air-excess ratio. In other words the data at α=0.4 should show the highest inclination and at α=1 the lowest. Figure 4.11 does not show this difference clearly, this cannot be explained. A more accurate measurement is required to show whether or not this relation does exist as it is predicted.

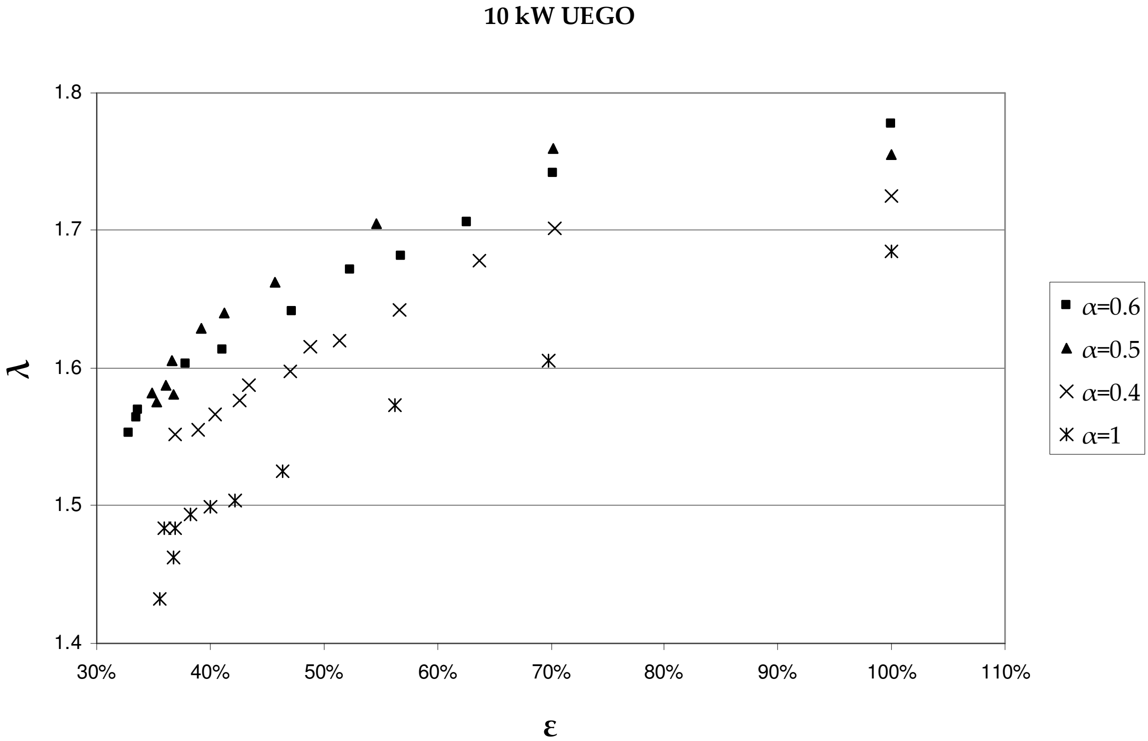

Figure 4.11 Air-excess ratio determined with UEGO sensor at 10 kW load.

4.3 Limits

One of the objectives was to find out what the limits of dual fuel operation are. A limit for dual fuel operation is the knock limit, which is reached when the maximum possible heat release from methane is reached. This limit was not reached (completely), since the venturi that was used did not allow enough biogas to pass through to reach this limit. Depending on the quality of the biogas not more than 80% of the heat release originated from methane in biogas. For all qualities of biogas and pure methane the engine runs smoothly until 60% to 70% of the heat release originated from methane. Above this, the engine starts running irregularly, this results in an audible combustion noise that is different to the normal combustion noise. Combustion irregularities are not observed in any other way than by listening. The change in noise that is noticed is not the result of intense knock, because the occurrence of knock would result in major decrease in thermal efficiency and this does not happen. Though some sort of knock occurs when these irregularities are observed.

In general, the engine operates more smoothly with biogas than with pure methane. Pure methane as a gaseous fuel resulted in more combustion irregularities than biogas. For pure methane the engine started running irregular very soon, but at 70% heat release from methane the irregularities result in (excessive) audible combustion noise. For biogas the combustion noise is less excessive but it still occurs. Although, both are not intense knock it can probably be characterised as light knock. This occurs late in the burning process and the amplitude of pressure fluctuations remains small [21]. Irregularities in the combustion should be observed by measuring in-cylinder pressure and temperature over time to say anything about whether or not it is knock that is heard and which kind of knock it is.

At a higher load the engine starts running irregularly at a lower heat release fraction from methane than at a lower load. At a high load the total quantity of methane in the cylinder is higher at the same heat release fraction, therefore the engine starts running irregularly earlier.

The other limit that could be reached is the smoke limit, when air-excess ratio decreases too much. The exact air-excess ratio at which smoke would occur for this engine and fuel is not known. For diesel engines it is usually in the order of 1.3. Figure 4.11 shows that at 10 kW air-excess ratios do not reach below 1.4. Together with the fact that no smoke was visually detected it is concluded that the smoke limit is not reached under the test conditions.

The quality of biogas could be a limiting factor for dual fuel operation. It was expected that too much carbon dioxide results in deteriorated performance parameters because too much oxygen is replaced by carbon dioxide. The qualities of biogas that were tested ranged from good quality biogas (α=0.7) to bad quality biogas (α=0.4). Even with bad quality biogas engine performance did not deteriorate compared to the other qualities of biogas. This means that for the type of engine and test conditions used in this study biogas with up to 60% carbon dioxide can be used without deteriorating engine performance.

5 Conclusion

The objective of this thesis work is to investigate the technical feasibility of the use of jatropha oil and biogas in a dual fuel diesel generator set. This is executed by assessing three performance parameters and operation limits. The three performance parameters are: thermal efficiency, volumetric efficiency and air-excess ratio. Thermal efficiency is directly important when this system would be used to generate electricity. Air-excess ratio and volumetric efficiency are important parameters that influence thermal efficiency and operation limits, therefore these have indirect influence. Operating limits are the smoke limit and knock limit, which correspond with maximum possible carbon dioxide fraction in biogas and maximum heat release fraction from biogas. The effect of dual fuel performance is predicted theoretically and then verified with experiments. In a reference test, engine performance for pure oil operation and pure diesel operation is obtained. Diesel is tested to observe the difference between jatropha oil and diesel for the three engine performance parameters. Dual fuel tests are performed with four qualities of biogas and pure methane as a gaseous fuel, at 6, 8, and 10 kW loads for a series of fractions of methane in the total fuel mixture. The design of the venturi limited the gas flow; consequently the maximum heat release fraction of methane was 80% for pure methane and a little less for biogas. Due to the limited availability of simulated biogas and jatropha oil the test series could only be performed once for each load.

The effect of dual fuel operation on thermal efficiency is difficult to predict, since complex combustion processes are involved. The reference tests show that thermal efficiency for jatropha oil does not deviate from diesel, at least not within the estimated error. For reference operation both fuels show an expected efficiency characteristic for a diesel generator. At full load thermal efficiency is approximately 32%.
Under dual fuel operation with biogas thermal efficiency remains unchanged compared to the reference measurement at 10 kW. For pure methane thermal efficiency even drops with increasing heat release from methane at 10 kW load. At lower loads, of 6 kW and 8 kW, thermal efficiency decreases with increasing heat release fraction from methane. The decrease was in the order of 5% to 10% relative to the initial thermal efficiency. This difference might have something to do with the carbon dioxide in the biogas. The carbon dioxide in biogas replaces oxygen in the intake mixture. At a 10 kW load a higher surplus of oxygen exists for pure methane than for biogas, at equal heat release fraction from methane. Consequently, the surplus of oxygen does not become too high for biogas and it does for methane. Therefore, thermal efficiency decreases at 10 kW when methane, but used as a gaseous fuel and it does not when biogas is used.

Volumetric efficiency is expected to decrease under dual fuel operation because air is replaced by biogas. In theory, dual fuel volumetric efficiency should decrease faster at a lower methane fraction in the biogas. At a lower methane fraction in biogas more carbon dioxide is present in the intake mixture at the same heat release fraction of methane than at a higher methane fraction in the biogas. Therefore, more air is replaced by biogas. Consequently it is expected that volumetric efficiency decreases faster at a low methane fraction in biogas than at a higher methane fraction or pure methane.

Volumetric efficiency is only directly measured during reference tests. The reference tests show a slightly higher volumetric efficiency for diesel than for jatropha oil. This difference is within the boundaries of the measurement error. Volumetric efficiency, which is not compensated for ambient pressure and temperature, is 95.1% for jatropha oil and 95.7% for diesel.
Dual fuel volumetric efficiency decreases with increasing biogas fraction in the intake mixture, because air is replaced by biogas. As predicted, volumetric efficiency decreases faster, when biogas contains more carbon dioxide. Tests with equal quality of biogas showed that volumetric efficiency decreases faster at a higher load when it is plotted as a function of heat release fraction of methane. The total heat release is higher at a higher load; therefore the absolute amount of methane in the intake mixture is higher at equal heat release fraction of methane.

Air-excess ratio is theoretically derived from the reaction equation for non-stoichiometric dual fuel combustion. It is predicted that air-excess ratio decreases under dual fuel operation for the same reason as volumetric efficiency decreases under dual fuel operation: biogas replaces air in the intake mixture. The methane fraction in biogas is expected to have a similar effect on air-excess ratio than on volumetric efficiency.
Air-excess ratio is obtained in two different fashions during the tests. From the reference volumetric efficiency the total intake volume is known and gas volume is measured. These two result in an intake volume of air. When the amount of air and mass of fuel is known air-excess ratio is obtained with the expression derived in section 2.2.2.1. Air-excess ratio is also derived from the oxygen fraction in the exhaust gas. The results from these two measurements deviate within the boundaries of measurement accuracy. Turbine meter results for air-excess ratio are consistently higher than those obtained for oxygen fraction in the exhaust gas. The predicted decrease in air-excess ratio with increasing heat release from methane is observed in the dual fuel experiments for all qualities of biogas and pure methane. The expectation, that air-excess ratio decreases faster for lower methane fraction biogas, could not be validated. Measurement data did not show the expected relation. More accurate measurements methods are required to be able to verify this prediction.

Dual fuel operation is only technically feasible when the operation limits are not reached. The operation limits, investigated in this study, are the smoke limit and the knock limit. Smoke limit occurs when air-excess ratio decreases too much, resulting in smoke production from the oil combustion. Knock limit is reached when either SI knock or CI knock occurs during dual fuel combustion.
The smoke limit is not reached. The smallest measured air-excess ratio is approximately 1.4. Generally, smoke limit is reached in the order of λ=1.3 for diesel fuel; for jatropha oil it is not known but it is assumed to be similar. In the type of engine that is used volumetric efficiency is relatively high, resulting in a high reference air-excess ratio. The addition of biogas does not result in a decrease in air-excess ratio, which is sufficient to reach smoke limit, even with the worst quality biogas with 60% carbon dioxide. This means that the maximum possible limit for carbon dioxide fraction in biogas was not established for this engine and experimental set-up. The occurrence of smoke was also not detected visually.
The knock limit could only be detected by listening. No heavy knock was detected but the engine started running irregularly when 60% to 70% of the heat release originated from methane. This could be considered the start of knocking behaviour, which does not become

intense. It probably is end-gas knock, which occurs at the end of combustion when auto-ignition occurs in the air-methane mixture. Therefore it is concluded that up to 60% heat release fraction from methane can be added without problems. For engines with a different design irregularities can occur at a different methane fraction.

The technical feasibility of jatropha oil and biogas as fuels for dual fuel diesel engines is proven in this study. It is recommended to operate the engine set at, at least half load. Below half load thermal efficiency is very low for both pure oil operation and dual fuel operation. Furthermore, literature showed that dual fuel operation is difficult under low load and idle conditions. Additionally, for dual fuel operation it is best to operate in the higher load region, since then thermal efficiency does not decrease with the addition of biogas. It is sensible not to operate the engine above 60% heat release from methane, irrespective of the quality of biogas. Up to 60% heat release of methane the engine operates without problems. Furthermore the quality of the biogas does not influence thermal efficiency. The use of bad-quality biogas does not result in decreases in volumetric efficiency and air-excess ratio that cause problems. Therefore the quality of the biogas is not of significant importance, for the engine used in this experiment. Another design engine, with for example a higher compression ratio, could experience problems with the qualities of biogas that were used for these experiments.
In order to determine operation limits more precisely, and to get a better view on heat release during combustion, an in-cylinder pressure measurement is required. It is also recommended to measure coolant temperature in order to prevent the engine from overheating. By investigating these two parameters a more detailed picture of dual fuel combustion could be obtained. For further research it is also advised to measure emissions like, CO_2, CO, NOx and HC and particulate emissions. Emissions are related the combustion processes. For instance an increase in carbon monoxide would indicate incomplete combustion and NO_x emission are related to combustion temperature. Furthermore additional research is required into the effects of pure oil use.

This technology is considered an appropriate technology for the use in rural areas in developing countries, because the technologies used are low-tech and mostly locally available. Furthermore there are some issues that need to be considered before applying this technology. A generator set like the generator set that was used for the experiments could, in principle, be used for rural electricity generation. But, the engine that was used is a (cheap) Chinese reproduction that is sensitive for failure. For example, the cooling system was not sufficient, mainly because no coolant pump was available. It is better to use a more robust engine, when it is used in rural areas where maintenance is more difficult and spare parts are harder to obtain. Concerning maintenance and spare parts, it is sensible to use an engine that is locally available. Besides, the local economy would profit, when the engine is purchased locally.

The study shows that there are no technical obstacles to obstruct the use of this technology, as long as the recommendations are followed. Furthermore, it is important that the production and use of jatropha oil and biogas is sustainable and environmentally sound. The use of the technology should be economically viable. Attention has to be paid to adoption and implementation.

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Appendix A Digester design

The two most popular designs for digesters in developing countries: fixed dome and floating dome [32;33] are discussed.

Fixed dome digester

Several designs are possible for this type of system; the different fixed dome designs differ a little but the main principle is the same. It consists of an underground fermentation chamber that is build from brick and cement walls. The sludge (feeding material mixed with water) enters on one side in the mixing pit. It digests inside the chamber and the gas is collected above the sludge in the same chamber, in the so-called gasholder. Gas pressure pushes out the digested sludge into the displacement pit. Most of the time a ditch is made where the digested slurry can flow through and from where it can be collected and used as fertiliser. A schematic overview of the basic principle behind a fixed dome digester is given in Figure 1.

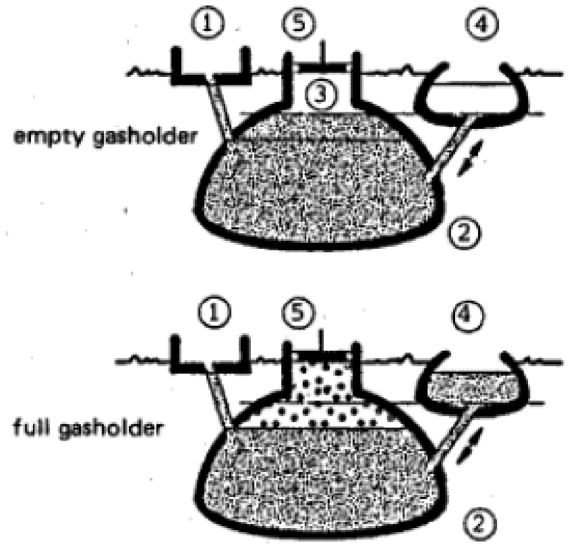

Figure 1 Basic function of fixed dome biogas plant: 1, mixing pit, 2 digester, 3 gasholder, 4 displacement pit, 5 gas pipe

The advantages of using a fixed dome digester are: relatively low costs, they hardly require any maintenance, there are no moving parts that can wear. For fixed dome no metal plates are used so no welding is required and also no corrosion can occur. Fixed dome is a permanent structure that is under ground, which protects the dome and saves space. It also protects from low temperatures during nights and rainy seasons; on the other hand it takes longer to heat it up during dry season. The construction of this plant is labour intensive and difficult; making expert guidance necessary. The success of this design in developing countries can also be seen in the huge adoption in Asian countries like Nepal, Vietnam and India [34].

Floating dome

In a floating dome design the fermentation chamber and the gas chamber are two separate constructions: the digestion chamber and a plate steel gasholder. The gasholder floats directly on the slurry or on a water jacket and will move up and down with the gas content; the gasholder is kept in place by a guide frame. This gives a constant gas pressure. This model is mainly used in India and according to FAO became obsolete after the invention of the fixed dome design because of competitively high investment and maintenance costs [33]. Figure 2 shows a water jacket biogas plant design.

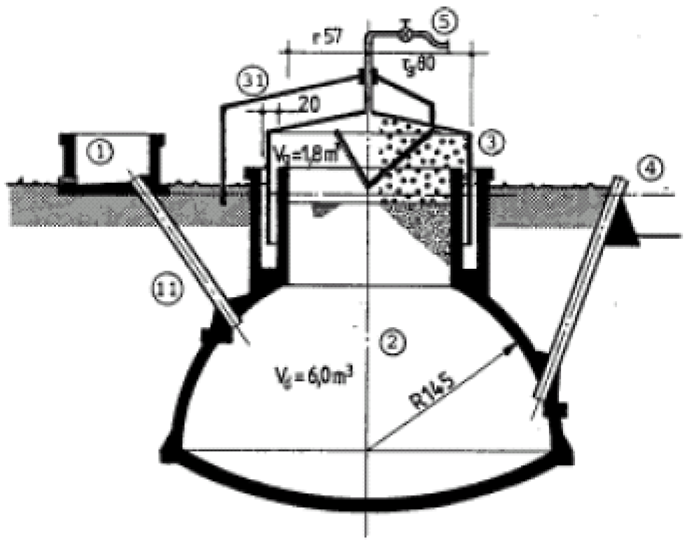

Figure 2 Water jacket plant with external guide frame: 1. mixing pit, 11. fill pipe, 2 digester, 3 gasholder, 31 guide frame, 4 slurry store, 5 gas pipe

The main advantage of a floating drum design is the constant pressure, which makes it easier to use the biogas to feed an engine. It is easy to understand and operate and the users can immediately see how much gas is still contained inside. Gas tightness should be no problem if the gasholder is welded well, derusted and painted regularly; this can also be considered a disadvantage since it can be very hard to find good materials and a good welder in a developing country. Also a floating drum design has moving parts that require relatively a lot of maintenance just as the regular painting does. The lifetime of the drum is between 5 and 15 years, which is short, compared to the lifetime of the complete construction. The gasholder shows a tendency to get stuck if fibrous substrates are used like Jatropha press cake.

Appendix B Molar mass and heating value Jatropha oil

Vegetable oil mainly consists of triglycerides, which are fatty acids bound together by a glycerol molecule. In the study of Foidle the fatty acid content of Jatropha oil is given [35]. From the fatty acid composition the average composition of a Jatropha oil molecule is calculated. This is done using the reaction equation for making biodiesel, which is provided below.

triglyceride + 3*methanol → 3* methyl ester + glycerol

A methyl ester consists of a fatty acid and a CH2 group.

Fatty acid composition

Table 1

Jatropha oil caboverde

	C	H	percentage	C average	Haverage
C14H28O2	14	28	0.001	0.014	0.028
C16H32O2	16	32	0.15	2.4	4.8
C18H36O2	18	36	0.071	1.278	2.556
C22H44O2	22	44	0.002	0.044	0.088
C16H30O2	16	30	0.009	0.144	0.27
C18H34O2	18	34	0.447	8.046	15.198
C18H32O2	18	32	0.314	5.652	10.048
C18H30O2	18	30	0.002	0.036	0.06
C22H42O2	22	42	0.002	0.044	0.084
				17.7	33.1

jatropha oil Nicuragua

	C	H	percentage	C average	H average
C14H28O2	14	28	0.001	0.014	0.028
C16H32O2	16	32	0.14	2.24	4.48
C18H36O2	18	36	0.074	1.332	2.664
C22H44O2	22	44	0	0	0
C16H30O2	16	30	0.008	0.128	0.24
C18H34O2	18	34	0.343	6.174	11.662
C18H32O2	18	32	0.432	7.776	13.824
C18H30O2	18	30	0.002	0.036	0.06
C22H42O2	22	42	0.003	0.066	0.126
				17.8	33.1

Table 2 Average molar mass

average composition C17.8H33.1O2 per fatty acid
triglyceride+3*methanol=3*methyl ester+glycerol
glycerol is C3H5(OH)3

methanol CH3OH
methyl ester =fatty acid +CH2
triglyceride=3*methylester+glycerol-3*methanol

average molecule Jatropha oil			Molar mass
C	56	12	
H	101	1	
O	6	16	
Molar mass	869		

Heating value

The fraction of carbon, hydrogen and oxygen in the oil is known. With this the higher and lower heating value can be obtained with equation 1 and 2and [36].

1

$$HHV = 0.3491 \cdot Y_C + 1.1783 \cdot Y_H + 0.1005 \cdot Y_s - 0.0151 \cdot Y_N - 0.1034 \cdot Y_o - 0.0211 \cdot Y_{ash}$$

2

$$LHV = HHV \cdot \left(1 - \frac{w}{100}\right) - 2.447 \cdot \frac{w}{100} - 2.447 \cdot Y_H \cdot 9.01 \cdot \left(1 - \frac{w}{100}\right)$$

w= moisture content

Table 3

	jatropha oil	sunflower oil
moisture content	0	0
C	77.3	77.73
H	11.62	11.36
S	0	0
N	0	0
O	11.05	10.91
ash	0	0
HHV (MJ/kg d.b.)	39.53	39.39
LHV (MJ/kg w.b.)	36.97	36.89

Appendix C Air-excess ratio

Definition for overall relative air fuel ratio λ

The expression for overall relative air-fuel ratio λ can be defined in several ways. The two definitions that are used to derive air-fuel ratio are given in equation 3 and 4. The relative air fuel ratio is determined using both these equations; it is shown that the results are identical.

3

$$\lambda_1 = \frac{m_{air}}{m_{oil}\left(\frac{A}{F}\right)_{oil} + m_{CH4}\left(\frac{A}{F}\right)_{CH4}}$$

4

$$\lambda_2 = \frac{m_{air}}{m_{airoil} + m_{airch4}} = \frac{m_{air}}{m_{air,stoich}}$$

Method 1

The air-fuel ratio's given in equation 5 and 6 result from the reaction equations 16 and 14.

5

$$\left(\frac{A}{F}\right)_{oil} = \frac{2.88 + 9.48}{1} = 12.36$$

6

$$\left(\frac{A}{F}\right)_{CH4} = \frac{17.16}{1} = 17.16$$

When the definition for m_{air}, m_{oil} and m_{CH4} are used as defined in respectively equation 2.3, 2.8 and 2.13 results in the following expression for the relative air fuel ratio.

7

$$\lambda_1 = \frac{\frac{\xi}{1-\xi}}{\frac{\theta}{1-\theta} \cdot 12.36 + \sigma \cdot 17.16}$$

Method 2

m_{air} is the actual amount of air that is taken in. m_{airoil} and m_{airch4} are the stoichiometric amounts of air required for combustion. With the reaction equations the required amount of air for stoichiometric combustion can be derived. It is assumed that 1 kg of methane is combusted. This results in an required mass of air for stoichiometric combustion as expressed in equation 8 with β as defined in equation 6.

8

$$m_{air,stoich} = 2 \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)} \right) + 2 \right) + 6.58 \cdot \left(78.25 \cdot \left(\frac{\beta}{\alpha(1-\beta)} \right) + 2 \right)$$

$$= \frac{671.4 \cdot \beta}{\alpha(1-\beta)} + 17.16$$

The actual mass of air that is taken in can be derived from the air fraction ξ. Assuming 1 kg of CH_4 is combusted results in a mass of biogas of 1/σ.

9

$$m_{air} = \frac{m_b \xi}{1-\xi} = \frac{\xi}{\sigma(1-\xi)}$$

This results in the following expression for air-fuel ratio λ_2.

10

$$\lambda_2 = \frac{\dfrac{\xi}{\sigma(1-\xi)}}{\dfrac{671.4 \cdot \beta}{\alpha(1-\beta)} + 17.16}$$

Both method 1 and 2 should give identical result. The equations for λ_1 and λ_2 look different but the comparison below shows that λ_1 and λ_2 are actually identical.

Comparison of λ_1 and λ_2

This derivation shows that λ_1 and λ_2 are identical.

11

$$\lambda_1 = \lambda_2$$

12

$$\lambda_1 = \frac{\dfrac{\xi}{1-\xi}}{\dfrac{\theta}{1-\theta} \cdot 12.36 + \sigma \cdot 17.16}$$

13

$$\lambda_2 = \frac{\dfrac{\xi}{\sigma(1-\xi)}}{\dfrac{671.4 \cdot \beta}{\alpha(1-\beta)} + 17.16}$$

14

$$\frac{\dfrac{\xi}{1-\xi}}{\dfrac{\theta}{1-\theta} \cdot 12.36 + 17.16 \cdot \sigma} = \frac{\dfrac{\xi}{\sigma(1-\xi)}}{\dfrac{671.4\beta}{\alpha(1-\beta)} + 17.16} = \frac{\dfrac{\xi}{(1-\xi)}}{\dfrac{671.4\beta \cdot \sigma}{\alpha(1-\beta)} + 17.16 \cdot \sigma}$$

From this it can be concluded that equation 15 needs to be valid in order to let λ_1 and λ_2 be identical.

15

$$\frac{\theta}{1-\theta} \cdot 12.36 = \frac{671.4 \cdot \sigma}{\alpha(1-\beta)}$$

Filling in the following equations for θ and σ results in identical expressions, which means that λ_1 and λ_2 are identical.

16

$$\theta = \frac{1}{1 + \dfrac{M_b}{M_{oil}}\left(\dfrac{1}{\beta} - 1\right)} = \frac{869\beta}{825\beta - 28\alpha + 28\alpha\beta + 44}$$

17

$$\sigma = \frac{16\alpha}{44 - 28\alpha}$$

18

$$\frac{16 \cdot \beta}{-44\beta - 28\alpha + 28\alpha\beta + 44} = \frac{\beta\sigma}{\alpha(1-\beta)} = \frac{16 \cdot \alpha \cdot \beta}{44\alpha - 44\alpha\beta - 28\alpha^2 + 28\alpha^2\beta}$$

$$= \frac{16 \cdot \beta}{-44\beta - 28\alpha + 28\alpha\beta + 44}$$

Appendix D Ideal gas

The intake mixture can be considered an ideal gas under cetain circumstances. The velocity of the intake mixture may not exceed half the velocity of sound in that mixture. The velocity of sound (C) can be obtained with 19.

19

$$C = \sqrt{k \cdot R \cdot T}$$

With

k= specific heat ratio = 1.4

R= gasconstant = 0.286 KJ/Kmol K

T=temperature = 293 K

This gives a velocity of sound of 342.5 m/s.

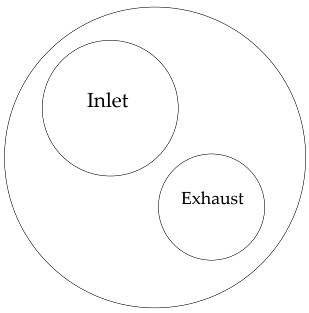

Figure 3 Schematic cylinder head

As shown in Figure 3 the cylinder has two valves, one inlet valve and one outlet valve. The exact measures of the valves are not known but it assumed that the area of the inlet valve is approximately one fourth of the total area.

The bore of the engine is 110 mm, this results in an area of the inlet valve of 2.38 10^{-3} m^2

Per cycle approximately 1 litre of air is taken in. With a rotational speed of 2000 rpm an inlet stroke takes about 0.015 seconds. This means that the volume flow of inlet air during the inlet stroke is 66.67 l/s; this equals 0.06667 m^3/s

This gives a velocity of the air through the inlet valve of 28 m/s.

The velocity of air in the valve could not exceed the half of the velocity of sound, which is 171 m/s. Because the velocity of air does not exceed this it is save to assume that the air behaves as ideal gas. In other words the intake velocity does not depend on the kind of molecules.

When the engine is operated in dual fuel mode biogas is added to the system. The velocity of the intake mixture does not change. The velocity of sound in the intake mixture changes a little because methane and carbon dioxide have different values for R and k. Only a small part of the intake mixture consists of biogas (<10%), therefore it is assumed that the velocity of sound is similar in the intake mixture with biogas than in pure air. Therefore it is safe to assume that the intake mixture behaves like ideal gas which means that the volume of intake mixture taken into the cylinder does not change between oil alone operation and dual fuel operation.

Appendix E Isentropic expansion factor

Table 4: constants

[37]	c_p(kJ kg^{-1} K^{-1})	c_v (kJ kg^{-1} K^{-1})	γ =(c_p/c_v)	R =(c_p-c_v)
Methane	2.21	1.687	1.31	0.523
CO2	0.82	0.626	1.31	0.194
air	1	0.714	1.4	0.286

In order to find the end gas temperature and pressure, the isentropic expansion factor (γ) of the mixture should be known. γ is defined as the ratio between c_p and c_v.. For a mixture the definition is as follows:

20

$$\gamma = \frac{c_p}{c_v} = \frac{\sum Y_i c_{p,i}}{\sum Y_i c_{v,i}} = \frac{Y_{air} c_{p,air} + Y_{CH4} c_{p,CH4} + Y_{CO2} c_{p,CO2}}{Y_{air} c_{v,air} + Y_{CH4} c_{v,CH4} + Y_{CO2} c_{v,CO2}}$$

With Y_i the mass fraction per species. From this equation a definition for γ can be derived that only consists of ξ and σ. The derivation is shown below:

21

$$Y_{air} = \frac{m_{air}}{m_{air} + m_{CH4} + m_{CO2}} = \xi$$

22

$$m_{air} + m_{CH4} + m_{CO2} = \frac{m_{air}}{\xi}$$

23

$$Y_{CH4} = \frac{m_{CH4}}{m_{air} + m_{CH4} + m_{CO2}} = \frac{m_{CH4}\xi}{m_{air}}$$

24

$$Y_{CO2} = \frac{m_{CO2}}{m_{air} + m_{CH4} + m_{CO2}} = \frac{m_{CO2}\xi}{m_{air}}$$

All masses are converted as function of m_b, in order to be able to define γ a function of ξ and σ and values for c_p and c_v are used as given in Table 4.

25

$$\gamma = \frac{\left(\frac{\xi}{1-\xi}\right) + 2.21 \cdot \sigma + 0.82 \cdot (1-\sigma)}{0.714 \cdot \left(\frac{\xi}{1-\xi}\right) + 1.687 \cdot \sigma + 0.626 \cdot (1-\sigma)}$$

The value for α is taken to be 0.7 and for ξ needs to be between 0.9 and 0.99. This results in values for gamma between 1.387 and 1.399 as shown in fFigure 4.

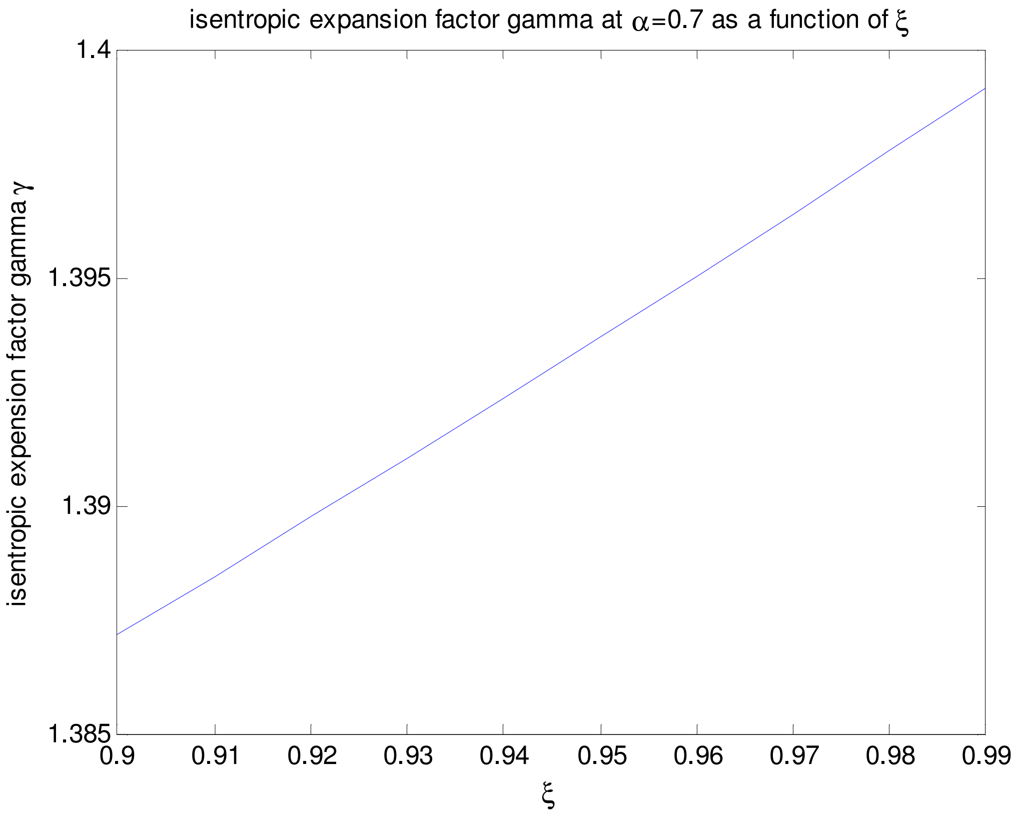

Figure 4: Isentropic expansion under factor of dual fuel

Temperature and pressure at the end of the compression stroke at TDC are important parameters for ignition delay and knock characteristics. These can be determined equation 26 and 27.

26

$$T_{TDC} = T_0 \cdot (CR)^{\gamma-1}$$

27

$$p_{TDC} = p_0 \cdot (CR)^{\gamma}$$

Table 5 shows the values for γ at given values for ξ and the corresponding temperature and pressure at TDC.

Table 5 Isentropic expansion factor, Pressure and Temperature at TDC

ξ	γ	T_{TDC} (K)	P_{TDC} (atm.)
0.9	1.387	878	50.9
0.91	1.389	881	51.1
0.92	1.390	884	51.3
0.93	1.391	887	51.5
0.94	1.392	891	51.7
0.95	1.394	894	51.9
0.96	1.395	897	52.1
0.97	1.396	901	52.3
0.98	1.398	904	52.5
0.99	1.399	907	52.7

The isentropic expansion factor that is used here is assumed to be at room temperature. For iar the degrees of freedom increase with increasing temperature, therefore the expansion factors used in Table 5 are not entirely correct. When a weighted average of 1.35 is used it gives values for pressure and temperature at TDC as provided in .

γ	T_{TDC} (K)	P_{TDC} (atm.)
1.35	790	45.8

Appendix F Pressure drop gas meter

Pressure drop inside the gas meter is measured before the tests. The results are shown in Table 6 and Figure 5.

Table 6 results pressure drop measurement gas meter

V0	Vt	V (m3)	t	Vdot (m3/s)	Δp(mbar)
291.898	291.95	5.20E-02	92.3	5.63E-04	6.4
291.97	292.03	6.00E-02	107.5	5.58E-04	6.3
292.055	292.12	6.50E-02	117	5.56E-04	6.25
292.14	292.21	7.00E-02	127.5	5.49E-04	6.1
292.225	292.3	7.50E-02	140.5	5.34E-04	5.8
292.52	292.6	8.00E-02	173.8	4.60E-04	4.6
292.45	292.5	5.00E-02	114.5	4.37E-04	4.3
292.615	292.67	5.50E-02	161	3.42E-04	2.9
291.71	291.82	1.10E-01	214	5.14E-04	5.4

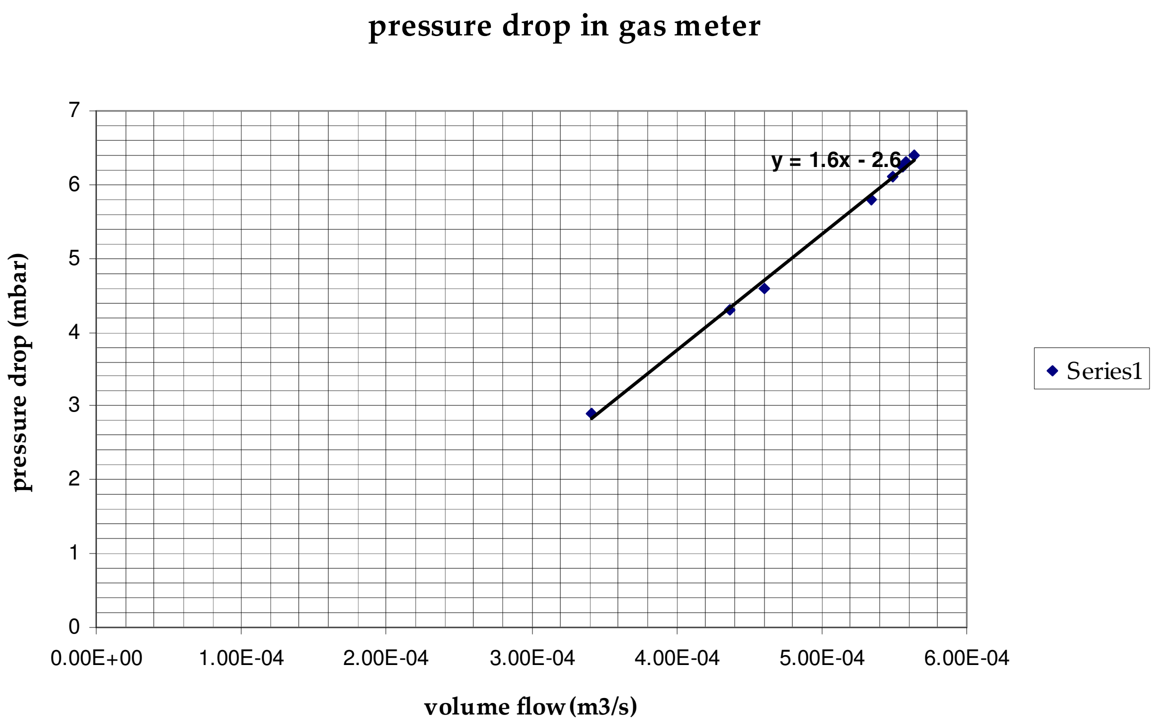

Figure 5 Results pressure drop measurement gas meter

Appendix G Air box

Not much literature is available on how to dimension an airbox for air measurements for an internal combustion engine, therefore a rather old article is used [38]. This article gives a dimensionless criterion for determination of air box meter dimensions for a measurement orifice. Since a turbine meter is used in this study it is expected that a smaller volume of the air box than is calculated here is required. This calculation is done to get an idea of measuremenst of an air box for a single cylinder engine.
First an overview of the equations used is provided and then the dimensions of the required air box are calculated.

U in equation 28 is a dimensionless number; when it is large the error in the measurements will be small. It should at least be 2.5.

28

$$U = \frac{C \cdot I}{p \cdot V_s}$$

C= airbox volume
I= pressure drop across measurement orifice
V_s = effective displacement volume of the engine= V_d* η_v = 1.093 10^3*0.89
p= atmospheric pressure.

Pressure drop over the orifice is calculated with equation 29.
29

$$I = \frac{8 \cdot W^2 \cdot R \cdot T}{g \cdot C_D^2 \cdot \Pi^2 \cdot p \cdot d^4}$$

W= mass flow per unit time = 0.02 kg/s
R= gas constant = 0.286 kJ/(kg K)
T= ambient temperature = 293 K
g= gravity = 9.81 m/s^2
C_D= coefficient of discharge or friction factor
d= is orifice plate diameter = 15 cm

With these values the pressure drop is calculated to be 0.152 kPa.

When the smallest possible value for U of 2.5 is taken and equation 29 is used this results in a volume of the air box of 1.6 m^3.

Appendix H Results thermal efficiency

In this appendix the results of thermal efficiency are presented as a function of heat release fraction ε.

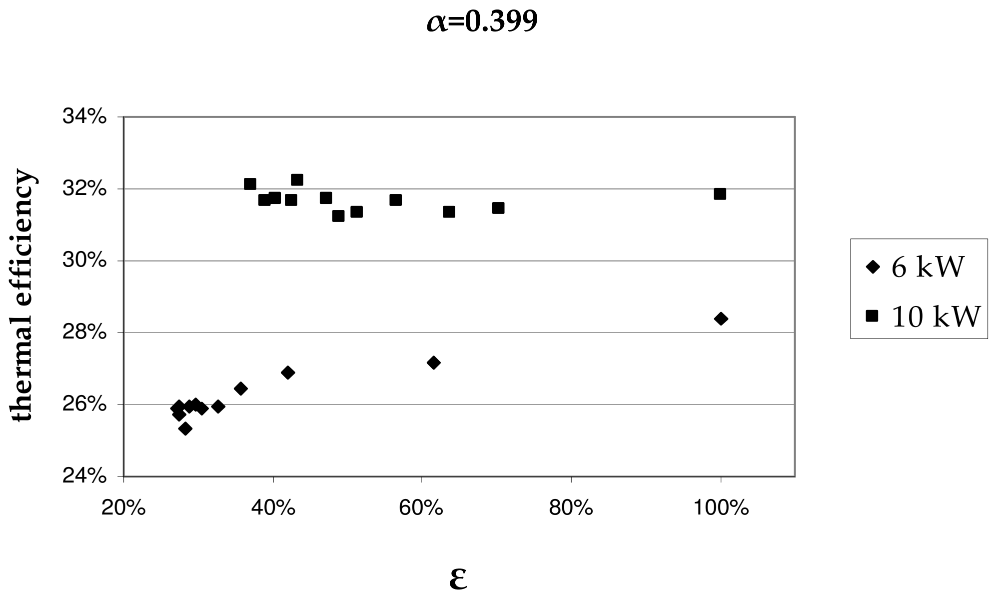

Figure 6 Thermal efficiency for biogas with α=0.399

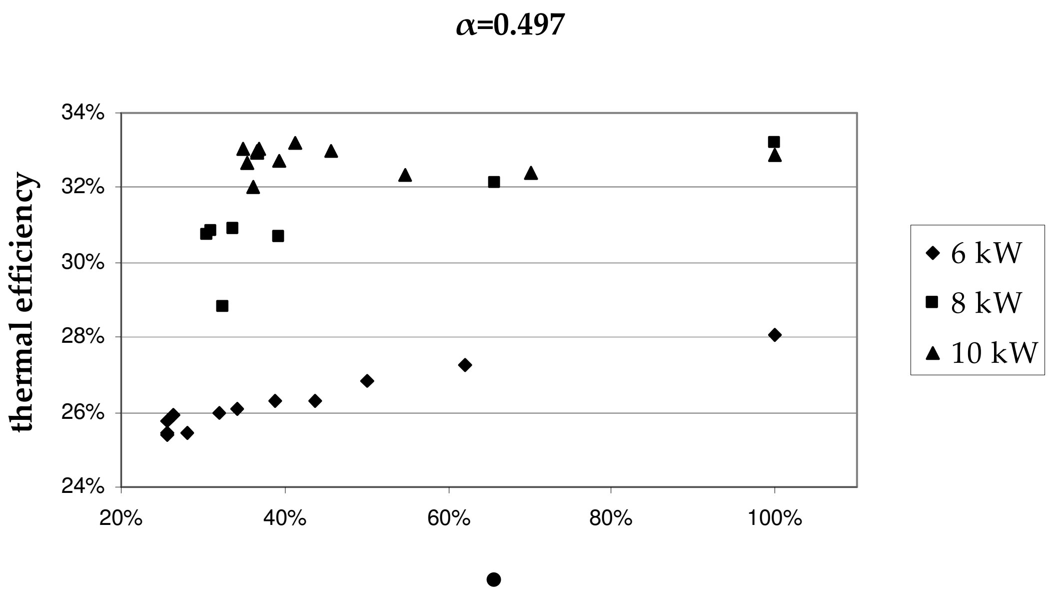

Figure 7 Thermal efficiency for biogas with α=0.497

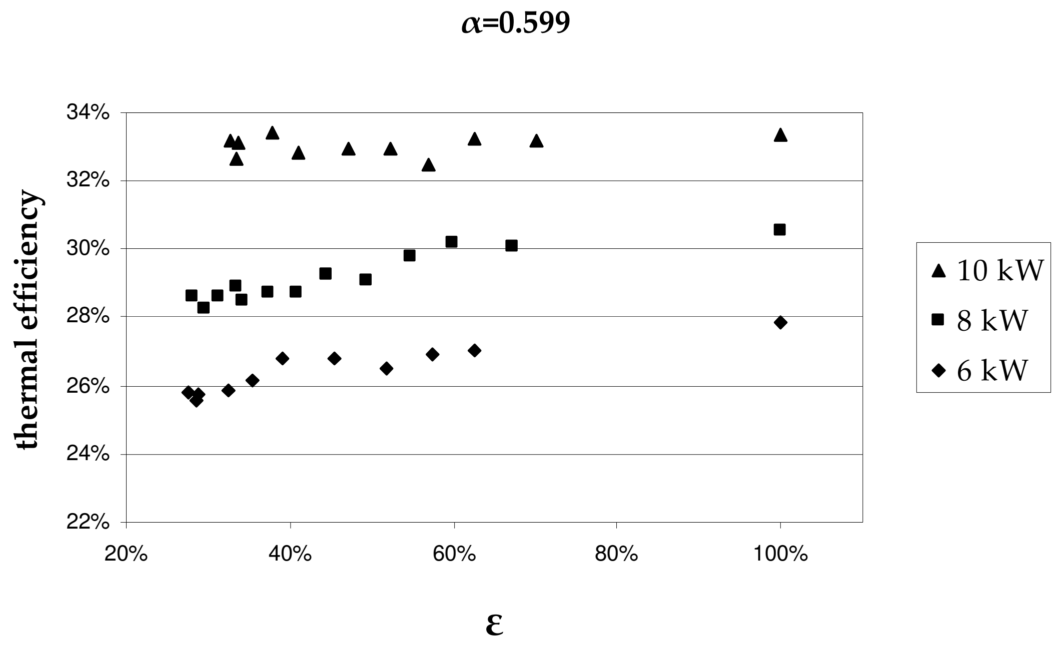

Figure 8 Thermal efficiency for biogas with α=0.599

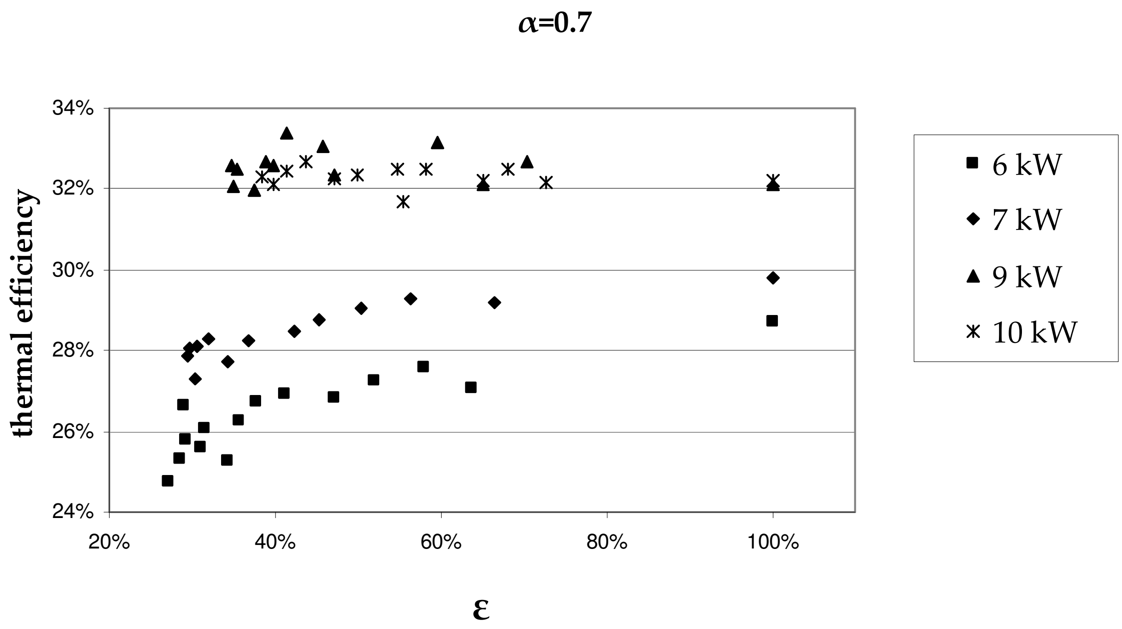

Figure 9 Thermal efficiency for biogas with α=7

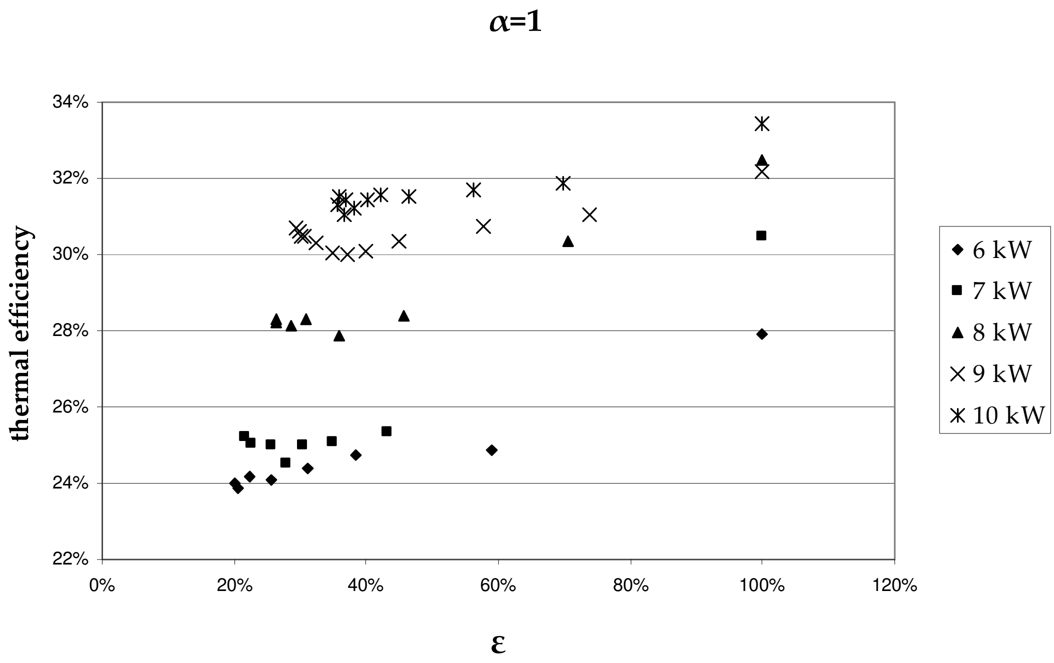

Figure 10 Thermal efficiency for biogas with α=1

Appendix I Results volumetric efficiency

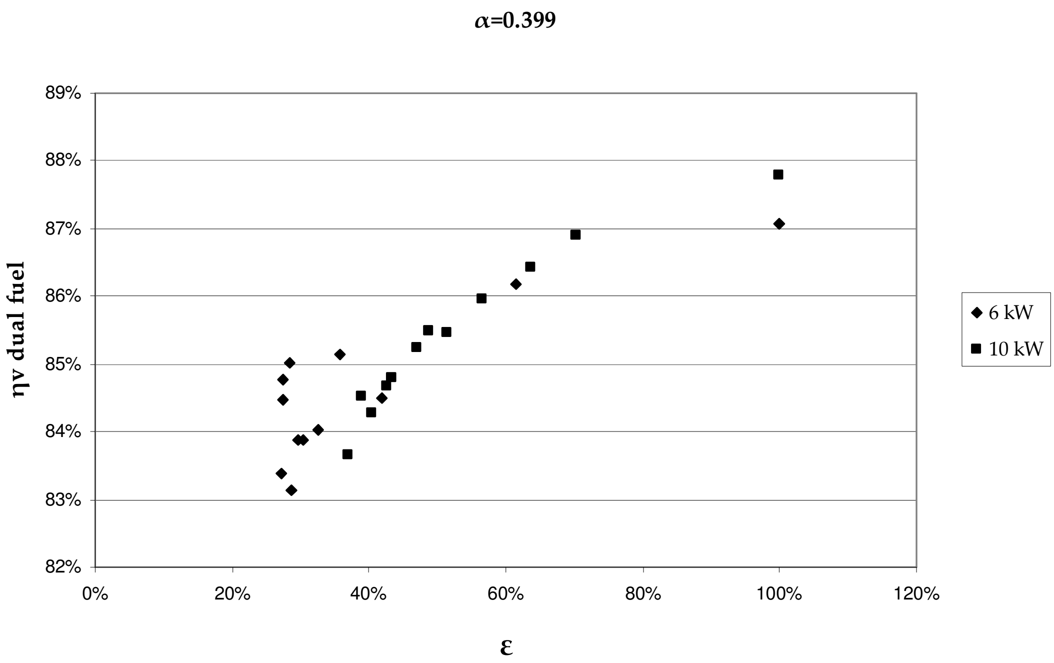

Figure 11 Volumetric efficiency for biogas with α=0.399

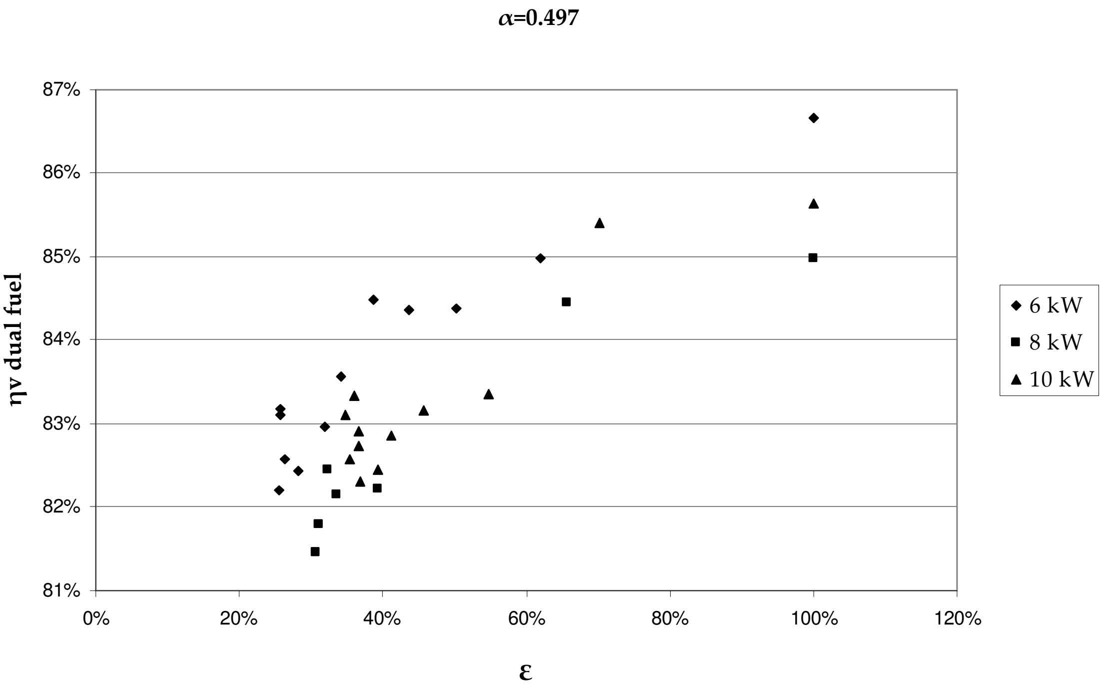

Figure 12 Volumetric efficiency for biogas with α=0.497

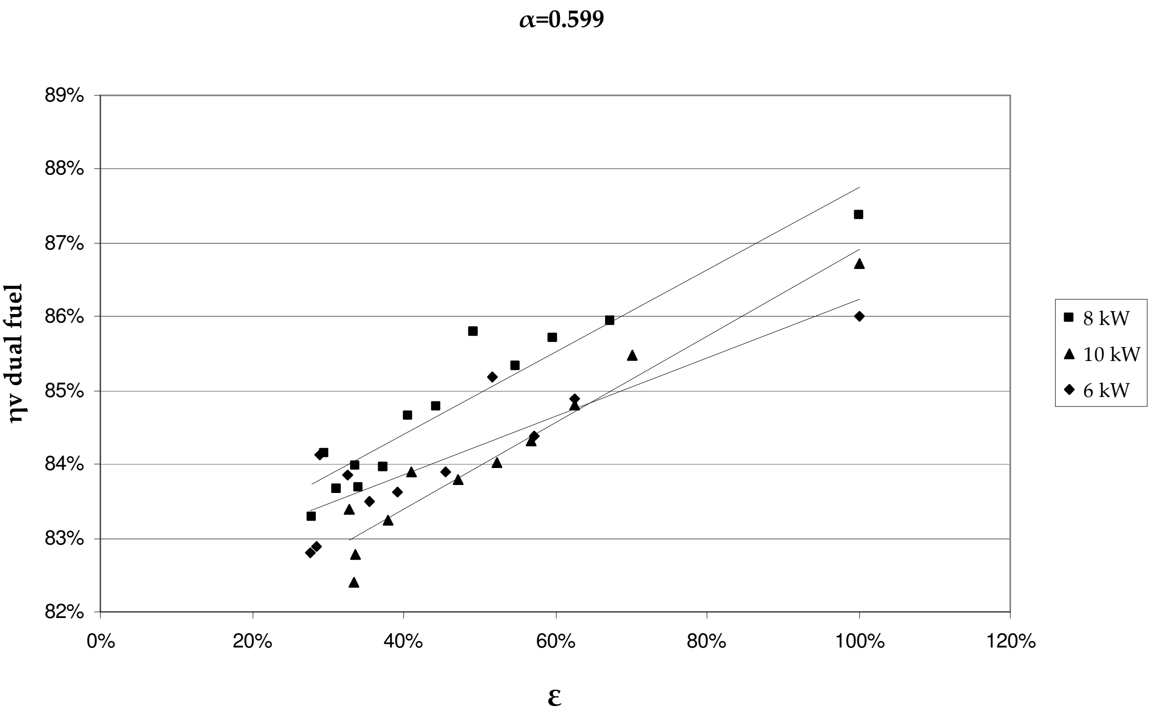

Figure 13 Volumetric efficiency for biogas with α=0.599

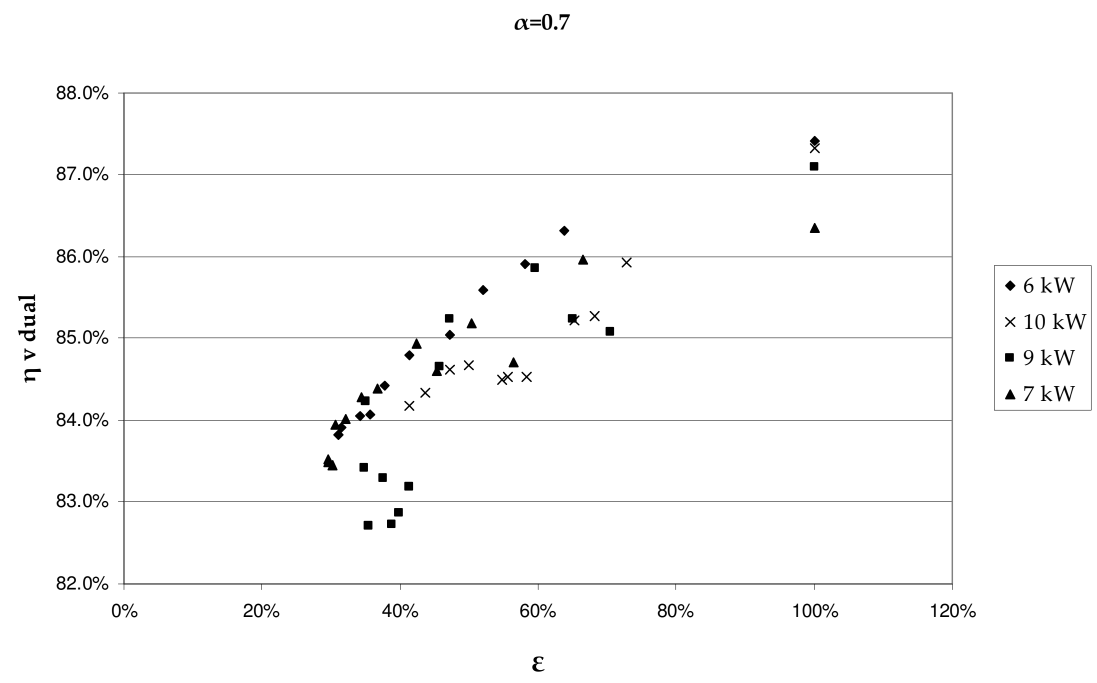

Figure 14 Volumetric efficiency for biogas with α=0.7

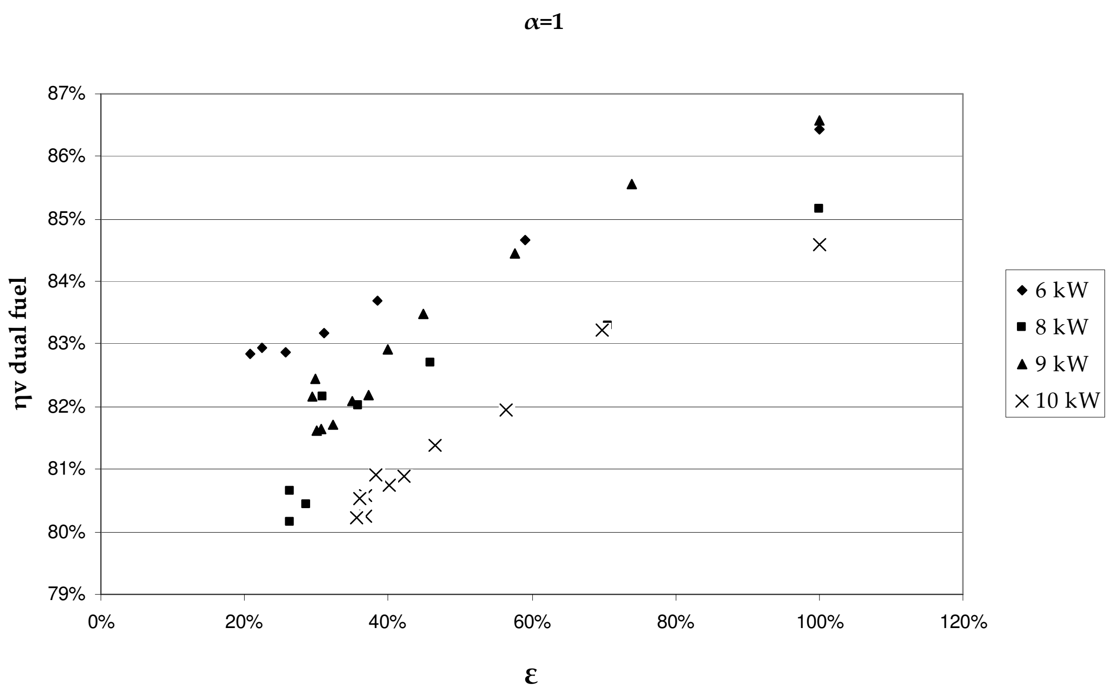

Figure 15 Volumetric efficiency for biogas with α=1

Appendix J Air-excess ratio

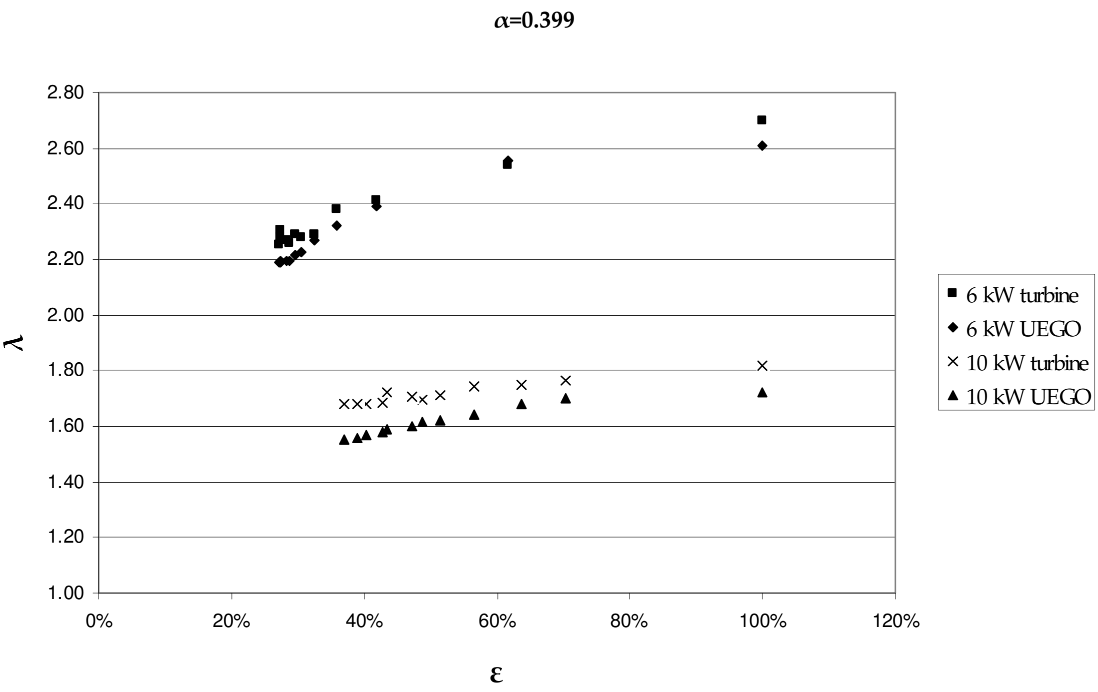

Figure 16 Air-excess ratio for biogas with α=0.399

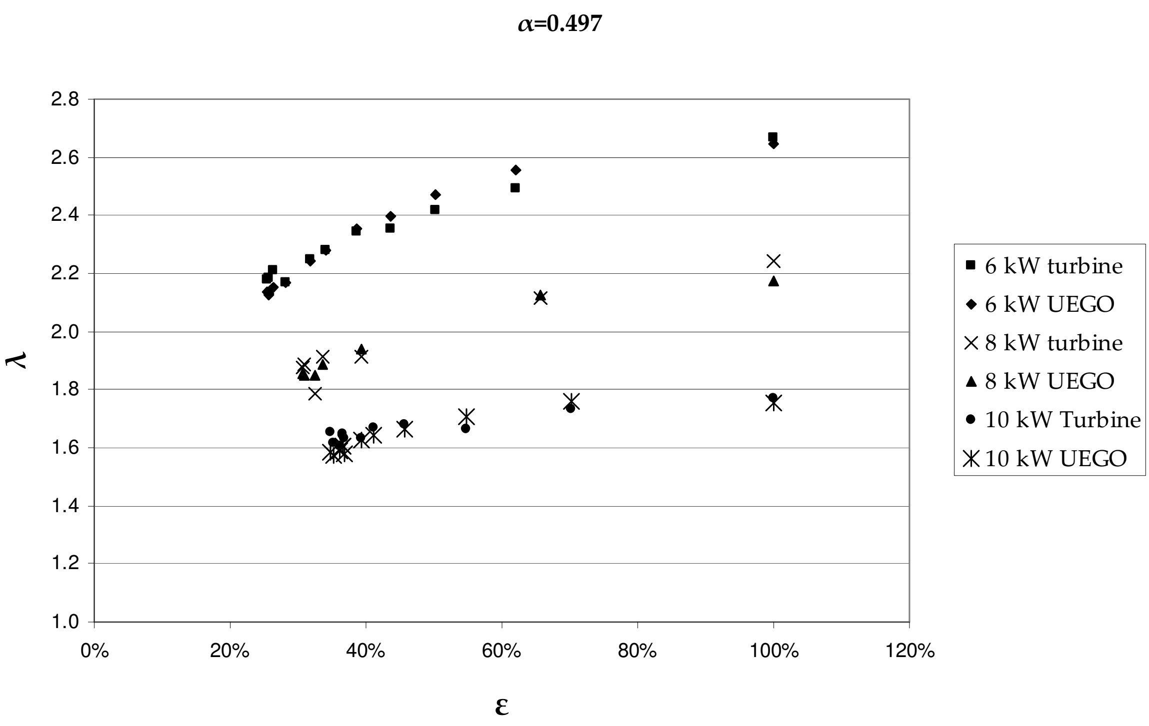

Figure 17 Air-excess ratio for biogas with α=0.497

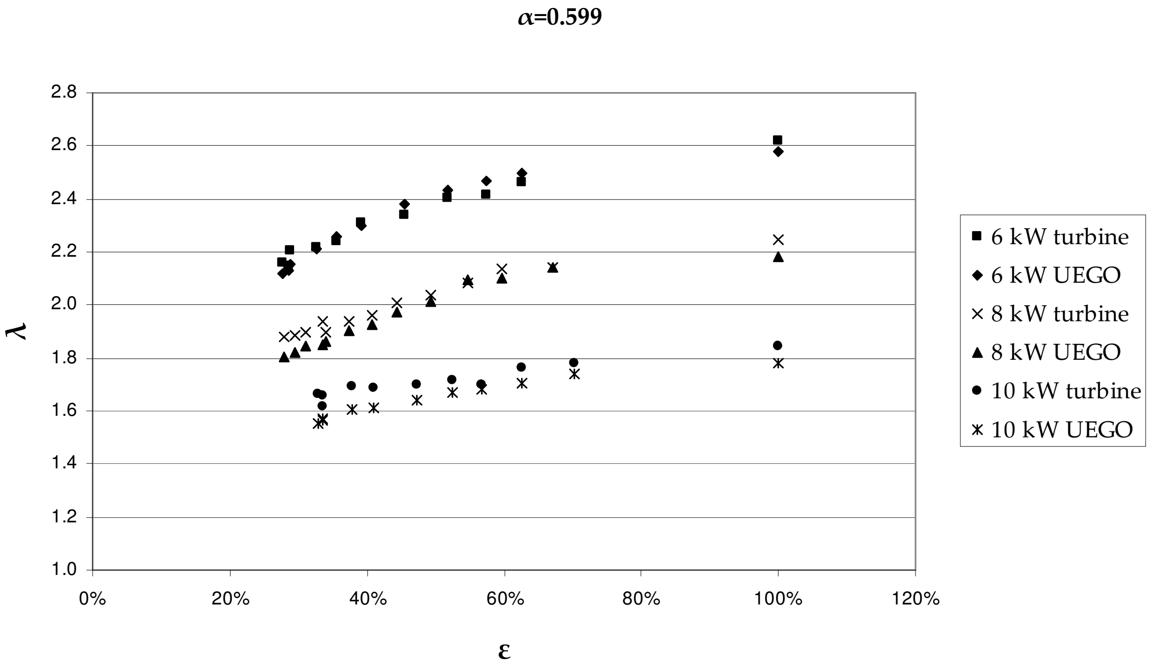

Figure 18 Air-excess ratio for biogas with α=0.599

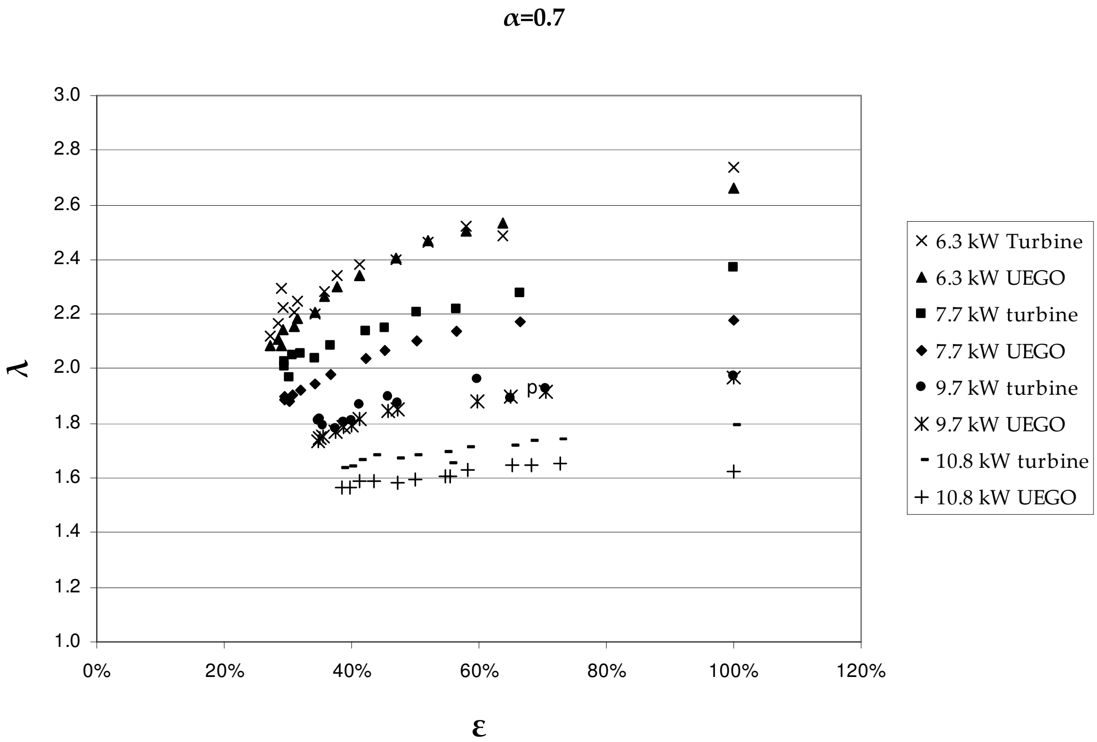

Figure 19 Air-excess ratio for biogas with α=0.7

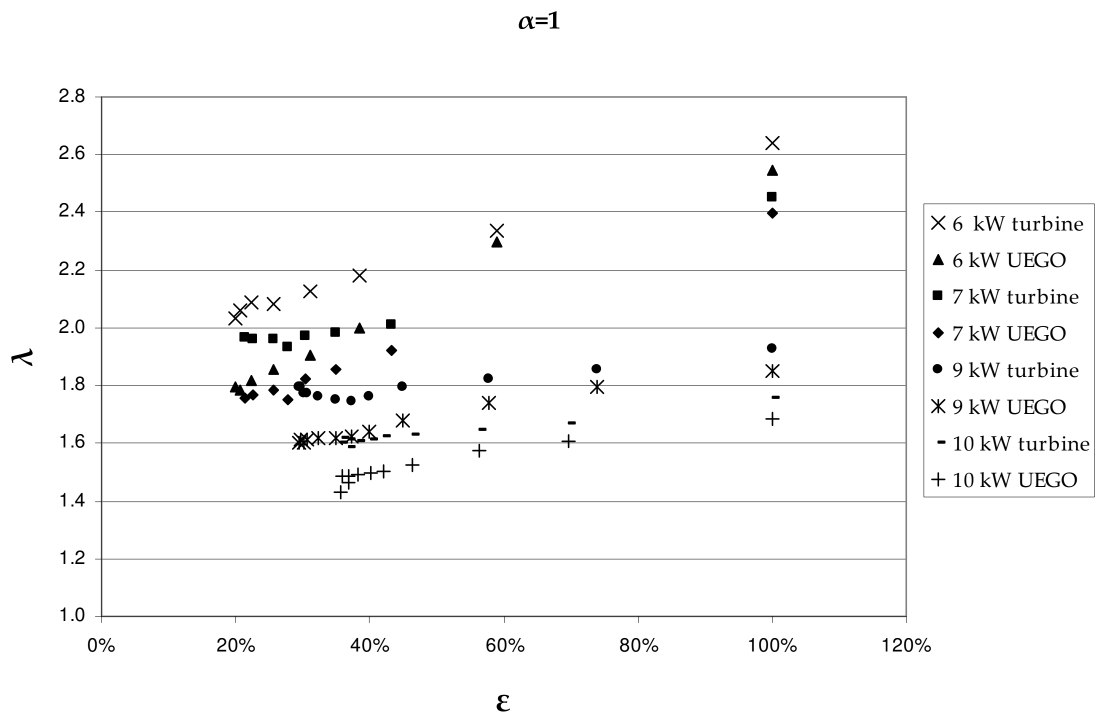

Figure 20 Air-excess ratio for biogas with α=1

Appendix K Estimation of required plantation size.

This gives an estimation of required hectares of jatropha oil plantation to feed a 12 kw dual fuel diesel generator for 24 hours a day with an average load of 70%. A 12 kW generator could supply 60 to 120 households depending on the appliances used by consumers and on the lay-out and construction of the grid.
It is assumed that the jatropha plantation delivers 3 tonnes of seeds per year per hectare and that the press cake delivers 0.18 kg methane per kg COD. When the engine is operated with a heat release fraction methane of 605 (ε=0.4) the oil that is requires 13 hectares. The gas requires 24.5 hectares. At this heat release fraction 24.5 hectares would be sufficient to feed the engine; the left over oil can be sold on the market.

Table 7 requirement of hectares for electricity 24 hours a day

ε	hectares oil	hectares gas
0.1	3.25	36.79
0.2	6.51	32.71
0.3	9.76	28.62
0.4	13.01	24.53
0.5	16.27	20.44
0.6	19.52	16.35
0.7	22.77	12.26
0.8	26.03	8.18
0.9	29.28	4.09

The table below shows how much hectares are required for only 5 hours a day, only during the eveening. Then, only about 4 hectares are required.

Table 8 requirement of hectares for 5 hours a day

ε	hectares oil	hectares gas
0.1	0.68	7.67
0.2	1.36	6.81
0.3	2.03	5.96
0.4	2.71	5.11
0.5	3.39	4.26
0.6	4.07	3.41
0.7	4.74	2.56
0.8	5.42	1.70
0.9	6.10	0.85