國立交通大學

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碩士論文

養豬場使用沼氣渦輪發電機發電之實驗研究

Experimental Study for power generation by Turbine Using Biogas in a swine Farm

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養豬場使用沼氣微型渦輪發電機發電之實驗研究

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摘要

本計劃書在小型養豬場使用 30kW 沼氣微型渦輪發電機發電。第一部分，測試引擎在不同負載條件下的效能以及針對目標引擎的能量使用進行效率分析；第二部分，利用所量測的數據計算並分析，包括熱效率、甲烷消耗率等，進而分析及比較微渦輪機與四行程汽缸式引擎的發電效能，並利用本論文的數據估算在台中養豬場規模為 3000、5000 頭豬使用沼氣發電的經濟效益分析。

實驗結果顯示，隨著負載從 15 調整到 30kw 時，渦輪引擎的沼氣供給量從 184.9 上升至 251.8L/min，渦輪發電機的最大發電功率、熱效率和甲烷消耗率分別為 25.23kW、23.12%和 168.7 L/min，而汽缸引擎的最大發電功率、熱效率和甲烷消耗率分別為 26.48kW、26.37%和 155.2 L/min。數據顯示汽缸引擎在高負載的情況下(發電功率 21~30kW)有較高的熱效率相對於渦輪發電機，然而在低負載的情況下渦輪發電機能比汽缸引擎發電機提供較佳及穩定的發電功率，值得注

意的是渦輪發機的額定發電功率最低限制為 15kW.

利用上述兩台發電機的實驗數據來估算在台中養豬場規模為 3000 及 5000 頭豬其使用沼氣發電的經濟效益，其分析內容包括設備初設成本、維護成本、年發電量及投資回收期。在豬隻規模為 3,000 的情況下，估算出使用汽缸引擎發電有較佳的經濟效益，而在豬隻規模為 5,000 的情況下，由於渦輪引擎可 24 小時不間斷的提供電力輸出，相對於汽缸引擎則受構造因素因而不可長時間持續供電(一天至多 20 小時，超過易使引擎損壞)，故 5000 頭豬隻所提供的沼氣量足夠發電機進行持續性的發電時，使用渦輪引擎發電有較佳的經濟效益.

關鍵字：沼氣發電、渦輪發電機、甲烷消耗率、熱效率、經濟效益

Experimental Study for power generation by Turbine Using Biogas in a swine Farm

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ABSTRACT

This proposal carries out the 30kW micro turbine generator experiments on small biogas plant in a swine farm. Firstly, the performance of turbine engine under various operating loads was tested, and the energy analysis for micro turbine engine was studied. Second, in this study, it conducted a series of comparison tests by using piston and turbine engines such as thermal efficiency, CH_4 consumption rate. The economic benefits were also estimated by the data obtained with 3000-head and 5000-head pigs by this research. The experimental results showed that the range of biogas flow rate to the turbine engine is from 184.9 to 251.8L/min with 67% CH_4 of biogas under varying loads from 15 to 30kW, and the maximum power generation, the corresponding thermal efficiency and the CH_4 consumption rate is 25.23kW, 23.12% and 168.7 L/min, respectively. For piston engine, the maximum power generation, the corresponding thermal efficiency and the CH4 consumption rate is 26.48kW, 26.37% and 155.2L/min, respectively. The

threshold efficiency of turbine engine is 2.97% lower. Compared with turbine engine, the fuel efficiency and thermal efficiency for piston engine is higher under the higher load operating range (21 to 30kW). However, the turbine engine can provide higher performance with lower efficiency variation compared to those of piston engine in range of 15~ 21 kW. Remind that the lower load limit is 15kW for this turbine engine, whereas piston engine still can be operated as low as 5.4kW.

As to the economic benefits of using biogas, it estimated that a swine farm with a scale of 3,000 heads and 5,000 heads in Taiwan The economic analysis consists of equipment cost, annual power generation, the revenue of power generation, and the payback period. The results showed that using the piston engine to generate electricity in a scale of 3000 swine farm has an advantage over the turbine engine. However, the maximum operating time for piston engine is 20 hours a day in order to protect generator. The turbine engine can work around the clock. Therefore, if the biogas is enough to keep engine working incessantly, then the net electricity production for turbine engine will be greater than the piston engine in a scale of 5000 swine farm.

Keywords: Biogas generator, Gas Turbine, CH_4 consumption rate, Thermal Efficiency, Economic Benefits

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Chapter 1
Introduction

1.1 Motivation and Background

With the increasing in energy demand, fossil fuel price fluctuation have become a global focus of concern, therefore, finding alternative energy becomes an important issue. Global warming is also a major problem for human beings now. The culprit is greenhouse gases (GHG), which include carbon dioxide, chlorofluorocarbons, nitrous oxide, water vapor, methane, etc..

Although fossil fuels are the most widely used in the world, it is expected to deplete eventually. Recently, people pay a lot of attention to renewable energy, such as geothermal energy, wind energy, hydropower, solar energy and biomass energy. Renewable energy, which has the great potential to substitute for the fossil fuels, not only can avoid the carbon dioxide produced from the fossil fuel, but also has renewability to handle energy shortage crisis.

Bioenergy is one of renewable energies that become more popular due to its contribution to stable supply and environmental protection simultaneously. The source of bioenergy can be derived from plant or animal organic materials. For example, The Florida government has advanced renewable bioenergy programs and policies to decrease the costs of biofuel and compete with fossil fuels. Basically, this state has the great potential to generate forest biomass as a source of renewable power because of its good climate.

Biogas is one of the biomass energy resources. At the end of 1990s numerous biogas plants were built for processing the liquid manure together with other digested co-substrates in the European countries, especially Denmark and Germany. Today, there are at least 3900 biogas plants built in Germany.

The biogas can be taken as a carbon neutral energy resource. Plants produce their own organic compounds by photosynthesis that using solar radiation and carbon dioxide to photosynthesize. Carbon is diverted to animals fed by the plants. Then, a part of carbon goes back to the environment in a form of carbon dioxide by Breathing or by combustion of biogas. Thus, the net of carbon amount is equal to zero in this cycle. Figure 1.1 shows a simple carbon cycle.

Untreated manure of swine can generate methane (CH_4), nitrous dioxide (NO_x) and carbon dioxide (CO_2). However, swine manure can produce biogas subjected to the anaerobic treatment and use it for power generation. The biogas is flammable because its contents mostly is made of methane (CH_4), and the others are carbon dioxide (CO_2), ammonia (NH_3), hydrogen (H_2), nitrogen (N_2) and hydrogen sulfide (H_2S) and with some trace amounts of organic compounds. As a consequence, the biogas can be used for power generation after a hydrogen sulfide removal treatment. In the meantime, the burning of methane, whose global warming potential is 23 times that of carbon dioxide, can also reduce the GHG.

Taiwan’s dependence on imported energy was increasing up to 87.25% in 1983, 98.2% in 1998, and 99.3% in 2011[1]. If Taiwan attempt to use limited resources to self-supported, then biogas is a good candidate to invest. As the biogas is used as fuel for combustion, the exhausted heat can be further utilized, like to heat the water to supply the swine farm that is one of heat recovery means. It can also avoid polluting the environment. The manure of swine has high content of volatile organic compounds, which can worsen the quality of rivers. Last, there are many swine, the main livestock in Taiwan, on farms. According to the report [1], the potential of methane generation in Taiwan is about 400Gg/yr. Apparently, biogas is a renewable and green energy source.

This laboratory has been awarded a four-year research project by National Science and Technology Program for Energy from 2010 to 2013. The project is named as *Development of the technology of agricultural waste bioconversion to biogas for electricity generation and carbon dioxide elimination by microalgae*. Constructing a pilot biogas plant is the ultimate goal of this project, which is divided into four subprojects. The subproject 1 is to upgrade the utilization efficiency of biogas by removing H_2S and CO_2 to ameliorate the biogas generation rate. Biogas from the anaerobic tank contains very high degree of hydrogen sulfide (H2S), around 5000ppm. Such high concentration H_2S will corrode the engine, so a H_2S biological desulfurization system developed by the subproject 1 was installed such that it could effectively reduce H_2S concentration from 5000 to 50ppm. In the subproject 2, the desulfurized biogas of subproject 1 will be utilized to operate the biogas engine to produce electricity under different monitoring parameters. The subproject 3 is to produce biodiesel from high lipid-content algae utilizing waste CO_2 either from the engine flue gas or the biogas itself. The purpose of the subproject 4 is to research the operating conditions which will affect biogas production rates and methane concentrations emission during the anaerobic processes.

This study is originated from the subproject 2 mentioned above. In the first year of the project, Lin [4] used a 30kW generator to build a waste heat recovery system and to analyze the power output and thermal efficiency under different excess air ratio. Secondly, the effect of oxygen-enriched combustion for engine was tested. Followed by Huang [5], she applied a waste heat recovery system to preheat the inlet gas with different flue gas temperatures and investigated the preheating influence on the generator performance.

Wu [6] installed a complete ignition measurement system, consisting of spark plug pressure sensors and rotary encoder, to record the in-cylinder pressure and crank angle of piston cylinder. He found the optimum spark timing that makes the power generation, thermal efficiency and percentage of used CH_4 being the highest. In this year, a completely self-operated biogas plant and a remote control system have been established. Up to 2012, this subproject is prospective to be able to generate 90,000kW-h electricity per year, equivalent to electricity bill saving of 270 thousand NT dollars, in a swine farm of 3000-head pigs. In the past research, the biogas power generation system used a 30kW-reciprocating internal combustion engine in the pilot plant. Now, this study is going to use a 30kW gas turbine engine system, then compare its performance with the internal combustion engine.

1.2 Literature Review

Tsai and Lin [1] analyzed bio-energy management from farm animals manure in Taiwan. With a realistic basis of the entire swine population from the farm scale of more than 1000 heads, this research showed following benefits: emissions of methane decrease 21.5 Gg and electricity generation of 7.2 × 107 kW-h per year, the same as electricity charge saving of USD 7.2 × 106, carbon dioxide easing is of 500 Gg per year.

Su et al. [2] set up GHG production data from anaerobic domestic animal wastewater treatment processes in Taiwan, and expound implications of the difference between the domestic animals wastewater treatment system presented by the IPCC. Study of GHG specimen from in situ anaerobic wastewater treatment systems of farms revealed, in that order, average emissions of 0.714 and 4.200/kgCO_2, 0.768 and 4.898/kgCH_4 per head per year for the period of three temperature periods. Average emissions rates of CH_4 from chosen dairy farms do not exceed the limits imposed by the IPCC, for the reason that animal manure is scattered before being deal with with a solid-liquid separator and an anaerobic wastewater treatment system.

Lee et al. [3] pointed that Biogas can be used in numerous different ways, but the energy application is maximized if it is transformed into electricity, it is easy to use and transfer, biogas mainly consists of methane and carbon dioxide, which has a lower specific heating value than natural gas, for the reason that of its component. Electricity could be generated from livestock excrement via a biogas generator and a Gas engine. They analyzed the peak values of generating efficiency, maximum cylinder pressure, and the emissions of NOx were elevated at an excess air ratio of about 1.2 as the hydrogen concentration was greater than before. CO2 emissions reduced as the excess air ratio and hydrogen consistency increased. These results confirmed that the addition of hydrogen to biogas enabled the efficient generation of electricity by using the gas engine generator during lean-burn combustion.

Lin [4] tested different air-fuel ratios for 30kW generator with 60% methane consistency of biogas in a swine farm in Miaoli. This research also contains the oxygen-enriched combustion and heat recovery The results showed that a higher power output and better thermal efficiency can be achieved by a greater conversion of CH4 in the combustion process. The engine performances are not improved much by 1% oxygen-enriched air, but with 3% oxygen-enriched air, the maximum power generation and thermal efficiency are increased, especially the engine can be operated normally at a lower limiting fuel supply rate. The heat recovery system is used to heat water, leading to an improvement of overall efficiency.

Next year, Huang [5] used 73% methane concentration of biogas to compare with the results of Lin [4] and applied the heat recovery system to heat the inlet gas under different temperature and analyze the preheating influence on the generation performance. The results showed that the power generation with 73% CH4 of biogas are higher than the ones with 60% CH4 of biogas, except the region around $\lambda<0.85$. However, the thermal efficiency increases with the increasing methane concentration just in the region of $\lambda>0.95$. In the case of the increasing inlet gas temperature effect, there is an obvious improvement when the temperature increases from 40℃ to 120℃ for biogas supply rate of 140L/min with λ=1.58.

Wu [6], the same as Lin [4] and Huang [5], used the same biogas generator under similar conditions, but he considered the effects of the water vapor content in biogas and the spark timing on generator. The results showed that at a given biogas supply rate, the biogas with dehumidification provides the higher power generation and thermal efficiency than the one without dehumidification. The power outputs increasing rates under the biogas supply rates of 200, 220 and 240L/min at stoichiometric condition are up to 4.7, 5.9 and 2.7%. The dehumidified biogas offers enthalpy increasing rate up to 0.79%, 1.17% and 1.27% than the biogas without dehumidification. Besides, the optimum spark timing of present engine is located at BTDC13, which can supply larger power output than the other spark timings. At a given biogas supply rate and excess air ratio, the power generation, thermal efficiency and percentage of used CH_4 by operating at the spark timing of BTDC13 are the highest.

Jiang et al. [7] established two kinds of biogas engines, spark-ignition biogas engine generators and biogas-diesel dual fuel engine generator. The purpose of this study was to solve the biogas engine trouble, such as the high exhaust temperature, serious back burning and low burning velocity.

Kang et al. [8] investigated the effect of firing biogas on the efficacy and operating characteristics of gas turbines. They used simple and recuperative cycle gas turbines. The simple cycle engine yielded 6.3 MW with 32.9% efficiency, whereas the recuperative cycle one yielded 4.6 MW at 38.5% efficiency at the standard ambient conditions. The heat recovery steam system consisted of an economizer, a superheater, and an evaporator, to generate superheated steam at a moderately elevated pressure. The design point function of both systems was obtained by employing natural gas as a fuel. At that time, off-design operations of the two systems by switching fuel from natural gas to biogas were simulated. The results showed that firing biogas will enlarge the gas turbine power output for both types of gas turbines by increasing flow rate. In the recuperative cycle, the gas turbine efficiency without including the fuel compression penalty is predicted to be lowered as the CH_4 content of the gas reduces, and the heat recovery increases with decreasing methane content of the fuel in both gas turbines.

Zubaidy [9] developed two kinds of turbine engines. SGT 94.2 and SGT 94.3. This research proposed an empirical relationship between the abilities to generate power when exposed to ambient conditions. For one degree of temperature rise in ambient temperature would result in the 0.1% Gas Turbine loses in terms of the thermal efficiency and 1.47 MW of its useful power output. The results showed that the thermal efficiency and its useful power output varies with temperature. At higher ambient temperatures, the thermal efficiency and useful power output have a tendency to be lowered. Furthermore, the turbine inlet temperature is a restriction as dictated with the air mass flow being reduced and the blade of turbine will change shape at higher temperatures.

Cho et al. [10] made a review for spark ignition of natural gas engines. In order to meet the emission standards and consider the stable combustion of engine, several methods can be used. First, lean burn is an effective way to reduce NOx emissions, but for recovering power output losses, turbocharging technology should be considered. Second, stoichiometric natural gas engine can equip with three-way catalyst to convert CO, HC and NOx, however, air-fuel ratio controller is needed. Third, EGR can improve knocking situation by reducing combustion temperature.

Badr et al. [11] carried out a parametric study on the lean misfiring and knocking limits of gas-fueled spark ignition engine. They tested Ricardo E6 engine by using propane and liquefied petroleum gas (LPG) as fuels. The parameters included compression ratio, engine speed, spark timing, intake temperature, intake pressure, and relative humidity of intake air. The following are the experimental results: Advancing the spark timing leads to the reduction of lean misfire and knocking limit. For low engine speeds, when the intake temperature increases, the lean misfire limit decreases. For high speeds, when the intake air temperature is up to 70℃, the lean misfire limit increases, however, beyond 70℃ the lean misfire decreases. As the relative humidity of the intake air increases, the lean misfire limit increases because the water vapors as a diluents will damp down the reaction rates during compression and combustion processes.

Tsagarakis [12] analyzed the most advantageous number of energy generators for biogas utilization in wastewater treatment installation. The data of this analysis was founded on the first generator for energy production from biogas. The first thing that one notices is if one generator is used, the expense per kWh produced is 0.087 €/kWh covering 15.9% of the facility's requirements. If two generators are used, the average expense for energy production is 0.088€/kWh covering 32.6% of the facility's requirements. Results are remarkably consistent among the estimations of six generators. The cost-effective analysis is calculated by total annual economic fee, which is the sum of the annual operation and maintenance costs and the annuitized construction cost. These results lead to the conclusion that the cost of each kWh produced may increases while the number of generators is increased, the cost decreases while the lifetime of generator increases

Park et al. [13] operated an 8-L, 6-cylinder spark ignition engine fueled by various the proportions of methane and nitrogen. Increasing N2 concentration makes the enhancement of thermal efficiency because with higher N2 dilation makes a decrease of combustion temperature due to less cooling loss to coolant. The experiment also applied different concentrations of H2 in biogas at stoichiometric (λ=1) and lean conditions. The engine operation at λ=1 with more than 5% H2 addition makes the decrease of thermal efficiency caused by increasing cooling loss. The lift of H2 fraction also makes the evaluation of NOx emission due to faster burning speed. The similar situation also happens at lean burning condition. The maximum NOx emission occurs at λ=1.1 for the entire fuel condition.

Tricase and Lombardi [14] evaluated the development of biogas in Europe and Italy. The amount of biogas increases gradually, and the usage of biogas depends on biogas quality. The percentage of biogas for generating electricity is 2/3 of total amount, and for generating heat is 1/3 nowadays. While Great Britain and Germany are the main producing countries so far, France has the highest production potential in the future.

Most of Italian biogas is from landfills, and the biogas from animal waste is only small percentage of total.

Chiu et al. [15] reduced CO_2 by using a high-density culture of Chlorella sp. in a semi-continuous photobioreactor. The result showed that inhibition of microalgal growth cultured in the system with high CO_2 (10-15%) aeration could be overcome via a high-density culture of microalgal inoculum that was adapted to 2% CO_2. Moreover, biological reduction of CO_2 in the established system could be parallel increased using the photobioreactor consisting of multiple units.

Yun [16] analyzed the inertial load and the thermal efficiency of hydraulic free piston engine. He studied the influence of inertial load on the burning performance of internal combustion engines and pointed out that the inertial load is important for cylinder to heated with a constant volume in a fast burning period. In the cylinder, the maximum pressure reaches 6~9MPa, and the heat energy is used fully and the thermal efficiency of engine is high.

Mahdi [17] made a comparison between two inlet air cooling methods, using the evaporative media and a mechanical chiller, respectively, that could to improve performance of a gas turbine. The results demonstrated that the gas turbine inlet air temperature can be reduced in a range of 4~35℃ for almost 9 months, and the corresponding maximum efficiency enhancement approaches 5%.

Popli et al. [18] referred to the fact that the efficiency can be enhanced by reducing gas-turbine compressor inlet air temperature. This can be typically achieved using either evaporative media coolers or electrically-driven mechanical vapor-compression chillers. The results showed that in extreme ambient, such as the summer in the Persian Gulf, the absorption chiller (single-effect H_2O-LiBr) utilizing 17 MW of gas turbine waste heat can provide 12.3 MW of cooling to cool the compressor inlet air to 10℃. Consequently, the evaporative cooling enhances gas turbine’s power output and energy efficiency by 4.2% and 1.6%, respectively, whereas the absorption cooling enhances these parameters by 23.2% and 13%, respectively. The additional electric power generated over a complete year by the waste heat powered ARS and the evaporative cooler is of 5264 MW and 1774 MW.

Farzaneh [19] proposed a method to improve gas turbine efficiency, which cooled the inlet air of the gas turbine by potential cooling capacity of the refinery natural gas pressure drop station. The Khangiran gas refinery had five gas treating units, which refined sour gas and made sweet gas available to transmission pipe line at high pressure. However, the gas pressure had to be reduced before entering the refinery. The pressure was reduced at the pressure drop station. In this study, it utilized a heat exchanger for refrigeration through the natural pressure drop process. The results showed that the gas turbine inlet air temperature can be reduced in range of 15 K and the performance can be improved in range of 4.5%

Martínez [20] showed the effect of excess air on combustion gas temperature at turbine inlet, and how it determined the power and thermal efficiency of a gas turbine at different pressure ratios and excess air. Besides, the humidity impact on excess air calculation was also analyzed and presented. The turbine inlet gas temperature implicitly makes reference to exhaust temperature from combustion chamber. It is known that the temperature during steady combustion in the combustion zone greatly exceeds the maximum allowable temperature of the turbine blades on the first stage, therefore, an increase in excess air is needed to bring the combustion gas temperature down to allowable value at the turbine inlet. The results showed that the thermal efficiency of gas turbine cycle increases as the excess air decreases and the methodology shown here is a useful tool to determine the turbine inlet gas temperature using the excess air and the pressure ratio.

Wu [21] applied a commercial package CFD-ACE+ to simulate the combustion flow field in combustion chamber of a micro gas turbine. The research contents were the applications of CFD simulations on low heating value methane gas fuel for the acceptability of MW54 micro gas turbine. Two parameters were studied. One was the mass fraction of CH_4 in the fuel mixture, consisting of CH_4 and CO_2, and the other was the turbine rotational speed. The simulation results showed that the thrust is diminished again as a result of adding non-fuel substance, CO_2, into pure methane fuel. It also indicated that the total air mass flow rate of primary zone decrease with the reduction of turbine rotational speed. In addition, the flow field was analyzed by selecting the cross-sections, locating at symmetric face. The result showed that the CH4 mass fractions and temperature in primary zone increase with rising methane concentration.

1.3 Scope of Present Study

The scope of this research is shown in Fig. 1.2. Two 30kW generators, one is reciprocating internal combustion engine and the other turbine engine, are used in this research. The piston engine is 8031i06, which was a four-stroke diesel engine originally. In order to use biogas gas as fuel, the spark ignition system was installed to this engine with some other modifications, in other words, it was customization and can do the modification services required by this project. It has been used extensively by Lin [4], Huang [5] and Wu [6] for consecutive three years. On the other hand, the gas turbine engine is Capstone CR30 Micro Turbine, which is provided by the Aerospace Industrial Development Corporation (AIDC). Because the turbine engine is a commercial product, it cannot make any change by the end-user. Therefore, this research work can only fix the load as a parameter and measures the resultant data, such as the biogas flow rate.

In this thesis, it conducted a series of comparison tests by using both engines mentioned above. The first one is the thermal efficiency as a function of load. For piston engine, the biogas and air flow rates can be fixed at demanded quantities, and then the resultant data, such as power output and waste gas concentrations, can be obtained in order to deduce the thermal efficiency. However, this procedure cannot apply to turbine engine. Because the air flow and biogas flow rates are automatically adjusted to comply with the change in engine speed of turbine engine. Therefore, the load (or turbine rotational speed) is fixed in advance, then the corresponding fuel flow rate and power output are recorded to get the thermal efficiency. The loads of turbine engine vary from 15 to 30 kW. Remind that the lower load limit is 15kW for this turbine engine. The comparisons include the amount of CH_4 consumed and the thermal efficiency.

Finally, a set of cost benefit analyses for 3000-head and 5000-head pigs are carried out for both two engines/generators mentioned above. The economic analysis consists of equipment cost, annual power generation, the revenue of power generation, and the payback period, such that it can demonstrate whether the use of turbine engine or piston engine has the advantages under the specific scale of a swine farm.

Chapter 2

Biogas Generation System

2.1 Process of Biogas Production

Figure 2.1 is a flow chart which shows the process of biogas production. The manure of swine after collection goes to wastewater treatment. The three-step piggery wastewater treatment system is based on a typical continuous plug-flow design, which includes solid-liquid separation, anaerobic treatment and aerobic treatment (activated sludge treatment system). The first thing that one notices is solid/liquid separation. Separation of the solid from the wastewater is to reduce the content of solids for subsequent handling and treatment, and to recovered solids can be made use of as fertilizer, etc. This physical process is accomplished by using every kind of filters. The second is Anaerobic treatment, which is conducted after solid/liquid separation, and occurs inside of anaerobic basins enclosed with “red-mud plastic (RMP) cover” (1.2~1.8mm of thickness), made of a kind of PVC material, which is corrosion-resistant and gas-and-water impermeable. There is one further the anaerobic treatment system can also salvage a part of chemical energy content of wastewater by producing methane that we must not ignore.

Biogas from the anaerobic tank contains very high degree of hydrogen sulfide (H_2S), which can erode the power generator, so the desulfurization process is considered necessary in advance. The common method for reduction of hydrogen sulfide is biological desulfurization. In the process, the H_2S is absorbed in water and then its content is mitigated greatly by biological method. Having got this case out of the way, the biogas will store in a red plastic bag.

2.2 Utilization of Biogas

Biogas can be used either for the production of heat only or for the generation of electric power. Normally heat and power are produced in the same time for higher energy efficiency. Such power generators are called combined heat and power (CHP) generation plants, and it is generally used in a four-stroke or a Diesel engine. CHP generation is an widespread way for energy conversion of biogas at small- and large-scale plants of biogas production. The points made so far apply in principle to A Stirling engine or gas turbine, a micro gas turbine, high- and low-temperature fuel cells, or a combination of a high-temperature fuel cell with a gas turbine.

So far, we have seen how that Biogas can also be used by burning it to produce steam, by which can drive an engine in the Organic Rankine Cycle (ORC) or the Cheng Cycle, the steam turbine, the steam piston engine, or the steam screw engine.

The range of capacities for the power generators are shown in Fig. 2.2, which are available on the market for the pilot-plant or industrial scale. The efficiency is defined as the ratio of the electrical power generated to the total energy content in the biogas. Efficiency figures are also provided by different manufacturers. Small-capacity engines generally can result in the lower efficiencies than that of high-capacity engines.

We may note, in passing that the generated current and heat can provide to the bioreactor itself, associated buildings, and neighboring industrial companies or houses. The power can be fed into the public electricity network, and the heat into the network for long-distance heat supply.

2.3 Engines

Table 2.1 lists some engines that can be operated with biogas. These have been improved during the recent years by following the development works inspired by the worldwide boom in biogas plants. The performance by some manufacturers even has already exceeded that of those given in this figures.

Table 2.1 Comparison of Different Power Generators

Feature	Four-stroke engine	Gas-Diesel engine	Stirling engine	Fuel cell	Gas turbine	Micro gas turbine
Capacity(kW)	<100	>150	<150	1-10000	20MW	28-200
Electrical efficiency	30-40%	35-40%	30-40%	40-70%	25-35%	15-25%
Pressure ratio	10:1	20:1	5:1	n.a.	5:1	5:1
Lifetime	Medium	Medium	Long	Very short	Long	Long
Alternative fuel in case of shortage of biogas	Liquid gas (gasoline)	Liquid gas	Any	Natural gas	Natural gas	Natural gas, Fuel oil

2.3.1 Micro Gas Turbine

Micro gas turbines are small high-speed gas turbines with low combustion chamber pressures and temperature, which are designed to generate the electrical powers between 28kW to 200kW. They are operated on a Brayton cycle. This study adopts the micro gas turbine generator, CR30, which comprises a gas compressor, a combustion chamber and an expansion turbine. It can runs on various fuels, such as propane, biogas, or liquid fuel. The rotating components are mounted on a single shaft sustained by patented air bearings, and spin at 84,000 to 96,000 rpm. The permanent magnet generator is cooled by the air flowing into the micro turbine, and it can provide up to 30 kW of electrical power. For normal operation, the compressor sucks in the combustion air. The fuel is normally supplied to meet the combustion air in the combustion chamber. When biogas with a low calorific value is used, it must be adjusted to a flammable mixture of biogas and air before it is supplied into the combustion chamber.

The electrical efficiency of 15~25% for today's micro gas turbines is still unsatisfactorily low. An attempt to increase the efficiency has been made by preheating the combustion air in heat exchange with the hot turbine exhaust gases. But great improvements are still necessary before micro gas turbines can be introduced into the market of industrial biogas plants. More specifically, the coupling of a micro gas turbine with a micro steam turbine to form a micro gas-steam turbine seems already interesting and economical today because of its high electrical efficiency.

2.3.2 Stirling Engine

An alternative to the commonly used four-stroke and the Diesel engines is the Stirling engine. The efficiency of the Stirling process is closest to that of the ideal cycle. The Stirling engine has been recommended for power generation for many years, but is seldom realized on an industrial scale because of technical problems in details. Power generated from biogas in Stirling engines is not known yet in industrial scale installations.

2.3.3 Gas Turbine

Biogas can be converted to current via gas turbines of medium and large capacity (20 MW and more) at a maximum temperature 1200 ℃. The tendency is to go to even higher temperatures and pressures, whereby the electrical capacity and thus the efficiency can be increased. The main parts of a gas turbine are the compressor, combustion chamber, and turbine.

Ambient air is sucked and compressed in the compressor and transmitted to the combustion chamber, where biogas is introduced and burnt with the compressed air. The flue gas that is so formed is passed to a turbine, where it expands and transfers its energy to the turbine. The turbine propels the compressor on the one hand and the power generator on the other hand. The exhaust gas leaves the turbine at a temperature of approximately 400~600 ℃. The waste heat can be recovered by driving a steam turbine downstream for heating purposes or for preheating the air that is sucked in.

2.3.4 Four-stroke Gas Engine and Diesel Engine

Given that the four-stroke biogas engines were originally developed for natural gas, we can know why the engines are well adapted by the special features of biogas. Their electrical efficiencies normally do not exceed 34~ 40%, as the nitrogen oxide (NO_x) output has to be kept below the prescribed values. The capacity of the engines ranges from 100 kW to 1 MW.

Four-stroke biogas engines regularly run in the lean-burn range (ignition window $1.3 < \lambda < 1.6$, λ is air-fuel ratio/stoichiometric air-fuel ratio), where the efficiency is expected to drop. The efficiency of lean-burn engines with turbocharger is 33~ 39%. The NO_x emissions can be reduced, nevertheless, by a factor of 4 in comparison to ignition (by compression) oil Diesel engines, and the limiting values can be met without further measures. Since the engines tend to knock with varying gas qualities, a methane content of at least 45% in biogas should be ensured.

In small agricultural plants, ignition oil Diesel engines are frequently installed. These engines are more economical and have a higher efficiency than four-stroke engines in the lower capacity range. Nevertheless, they produce more NOx emissions. Their lifetimes usually are given as 35,000h of operation.

Most of us would accept that Diesel engines burning gas fuel can be operated by direct injection because pre-chamber engines develop hot places, resulting in uncontrolled spark failures with biogas. Owing to the internal formation of gas mixtures, Diesel engines can be faster controlled. The ignition oil Diesel engine is operated ideally at $\lambda < 1.9$. The efficiency is then up to 15% better than that in a four-stroke engine.

2.3.5 Fuel Cell

It is worth noting that the fuel cell converts the chemical energy of hydrogen and oxygen directly into current and heat. Water is formed as the reaction product. Fuel cells may differ in a number of ways from batteries since they demand a steady source of fuel and oxygen to operate. However, they can produce electricity continually as long as these inputs are kept supplying.

Simply stated, a fuel cell works with a liquid or solid electrolyte held between two porous electrodes–anode and cathode. The electrolyte lets ions pass only and allow no free electrons from the anode to the cathode side. The electrolyte is thus “electrically non-conductive.” It separates the reaction partners and thereby prevents direct chemical reaction. For some fuel cells, the electrolyte is also permeable to oxygen molecules. In this case the reaction occurs on the anode side. The electrodes are connected by an electrical wire.

Both reaction partners are continuously fed to the two electrodes, respectively. The molecules of the reactants are converted into ions by the catalytic effect of the electrodes. The ions pass through the electrolyte, while the electrons flow through the electric circuit from the anode to the cathode. Taking into account all losses, the voltage per single cell is 0.6 ~ 0.9 V. The desired voltage can be reached by arranging several single cells in series, a so-called stack. In a stack, the voltages of the single cells are added.

The data summarized indicate that the biogas has to be purified to remove CO and H2S especially before feeding into the fuel cell. Only a small number of fuel cell plants, mostly pilot plants, are in operation for the generation of electricity from biogas.

Chapter 3

Experimental Apparatus and Procedures

3.1 Experiment layout

The Experiment layout is shown in Figure 3.1. When the turbine engine starts, the air and the biogas are sucked into the engine. The flow meters, marked by F1 and F2, measure the air and the biogas flow rates, which are automatically adjusted according to the change in engine speed. It is important that not all of the air for combustion, part of the total air is used for cooling the hot gas exhausted from combustor outlet, which prevent turbine blade from heat damage.

First of all, the compressor would increase the pressure and temperature of biogas by reducing its volume, the compressor outlet temperature is about 40℃, and the pressure of biogas is about 5.6kgf/cm^2. In the second place, the water vapor of biogas is removed by Freeze dryer, and then the biogas will be stored in the biogas tank. Finally the fuel is mixed with air and ignited in the chamber. The waste gas analyzer sets at the engine outlet to measure the compositions of waste gases, and the waste gas temperature is measured by a k-type thermocouple.

3.1.1 Engine

Figure 3.2 is a flow chart which shows the schematic diagram of gas turbine. For gas turbine engine, initially, gases are accelerated and pressurized in a compressor, and then they are slowed by using a diverging nozzle. For these processes, the temperature is increased. In an ideal system, this is an isentropic process. After that, gases go to a combustion chamber. Ideally, the combustion is isobaric process. Furthermore, the combustion product gases are expanded and accelerated by nozzle guide vanes before the energy is extracted by a turbine. In an ideal system, these gases are expanded isentropically and leave the turbine at their original pressure, mentioned previously. This ideal power cycle is called Brayton cycle. Figure 3.3 shows the engine and its detailed data, which can be referred in the following table.

Table 3.1 Engine Technical Data

<table>
<tr><td colspan="4">Capstone Turbine Engine Technical Data</td></tr>
<tr><td>Engine model</td><td colspan="3">CR30</td></tr>
<tr><td>Electrical Power Output</td><td colspan="3">30kW</td></tr>
<tr><td>Voltage</td><td colspan="3">400 to 480 VAC</td></tr>
<tr><td>Electrical Service</td><td colspan="3">3-Phase, 4 wire</td></tr>
<tr><td>Engine speed</td><td colspan="3">84000 to 96000 rpm</td></tr>
<tr><td>Rated Efficiency</td><td>Compressor
0.81</td><td>Combustion chamber
0.9</td><td>Turbine
0.84</td></tr>
<tr><td>Dimensions</td><td colspan="3">762 x 1524 x 1956 mm (30 x 60 x 77 in)</td></tr>
<tr><td>Compression ratio</td><td colspan="3">5.6:1</td></tr>
<tr><td>Digester / Landfill Gas HHV</td><td colspan="3">13.0 - 32.6 MJ/m3 (350 – 875 BTU/scf)</td></tr>
<tr><td>Frequency</td><td colspan="3">50/60 Hz</td></tr>
<tr><td>Maximum Output Current</td><td colspan="3">46A, grid connect operation</td></tr>
<tr><td>Digester/Landill Gas HHV</td><td colspan="3">350 to 1,275 BTU/scf</td></tr>
<tr><td>Dry weight</td><td colspan="3">~ 159 kg</td></tr>
<tr><td>Power-to-weight (specific power)</td><td colspan="3">0.188 kW/kg(0.115 hp/lb)</td></tr>
</table>

3.1.2 Air Flow Meter (VA-400)

The flow meter at air inlet is insertion CS flow sensor type VA-400 flow sensor, whose range varies with the installed pipe diameter. In order to maintain the accuracy stipulated in the data sheets, the sensor must be inserted in the center of a straight pipe section with an undisturbed flow progression. An undisturbed flow progression is achieved if the sections in front of the sensor and behind the sensor are sufficiently long, absolutely straight and without any obstructions such as edges, seams, curves etc. The minimum length ahead the sensor along the pipe should be 10 times of pipe diameter and 5 times behind sensor for the fully developed turbulent flow profile, so the measured flow rate can be accurate enough. Figures 3.4a and 3.4b show the flow meter and its detailed data.

3.1.3 Biogas Flow Meter (P-050)

The flow meter at biogas inlet is TOKYO KEISO P-050 purgemeter. Figures 3.5a and 3.5b show the picture and its technical data. The principle is to use the float, which moves up and down within the tapered pipe, to measure the amount of volume rate of fluid. As the flow rate increases, it will go up. On the contrary, the float will be lowered if the flow rate is decreased. The float can rotate that is why it is also called the rotameter.

3.1.4 Dehumidifier (RD20)

Figure 3.6 shows the dehumidifier, GTT RD20, used for removing the water vapor of biogas. The maximum inlet biogas flow rate is 44 L/sec. It is pre-cooled as biogas leaves from the evaporator. The coolant in the dehumidifier is R-134a.

3.1.5 Thermocouple

A thermocouple is a sensor for measuring temperature. It consists of two dissimilar metals joined together at one end, which can produce a small unique voltage at a given temperature. This voltage is measured and interpreted by a thermometer. Thermocouples are available in different combinations of metals or calibrations. The four most common calibrations are J, K, T and E. Each calibration has a different temperature range and environment.

Type K (Chromel–Alumel) is the most commonly used thermocouple with a sensitivity approximately 41 μV/°C. The voltage of Chromel is positive relative to the one of alumel. It is inexpensive and its temperature is wide, ranging from −200 °C to +1350 °C.

In this research, four K-type thermocouples is used for measuring waste gas temperature and inlet gas temperature. Figure 3.7 shows the picture of K-type thermocouple.

3.1.6Gas Analyzer (ECA450)

Figure 3.8 is the gas analyzer, BACHARACH ECA 450, used for measuring waste gas component data, which include the concentrations of oxygen, NOx and carbon dioxide. The measured data and calculated data are shown in the following table 3.2 and table 3.3.

Table 3.2 The measured data of gas analyzer ECA450

Measured Data	
Oxygen	0.1 to 20.9%
Carbon Monoxide (hydrogen compensated)	0 to 4,000 ppm
Carbon Monoxide High	4,001 to 80,000 ppm
Nitric Oxide	0 to 3,500 ppm
Nitrogen Dioxide	0 to 500 ppm
Sulfur Dioxide	0 to 4,000 ppm
Combustibles	0 to 5% (application dependent)
Stack Temp.	-4 to 2400℉ (-20 to 1315℃)
Primary/Ambient Temp.	-4 to 999℉ (-20 to 999℃)
Pressure/Draft	-27.7 to 27.7 inches of Water

Table 3.3 The calculated data of gas analyzer ECA450

Calculated Data	
Combustion Efficiency	0.1 to 100.0%
Excess Air	1.0 to 250%
Carbon Dioxide(dry basis)	0 to fuel dependent maximum
NOx	0 to 4,000 ppm
NOx (ref. to %O2)	0 to 17,000 ppm
CO (ref. to %O2)	0 to 99,9999 ppm
NO (ref. to %O2)	0 to 14,900 ppm
NO2 (ref. to %O2)	0 to 2100 ppm
SO2 (ref. to %O2)	0 to 17,000 ppm

3.1.7 Methane Concentration Analyzer (GuardCH4)

Figure 3.9 is guardian plus infra-red gas monitor GuardCH4, which is used for measuring the methane concentration of the inlet biogas.

3.1.8 Temperature with Humidity Transmitter (JHTD3010-N)

Such transmitter is shown in Fig. 3.10, whose humidity accuracy covers the full range from 0 to 100% RH, allowing precise measurement of the humidity over the operating temperature from -40 to 80 ℃. It is used for measuring the temperatures and humidities of biogas that with and without dehumidification.

3.1.9 Humidity Temperature Meter (Center 311)

The Center 311 humidity temperature meter is shown in Figure 3.11. It is used to measure the humidity and temperatures of the environment and biogas.

3.2 The Theoretical Calculation

The following calculations include the excess air ratio, thermal efficiency, theoretical mole fraction of CO2 in waste gas, theoretical percentage of consumed CH4, the percentage of water vapor removed from biogas and combustion stability. These data will be used in the analyses of the following experiments.

3.2.1 Excess Air Ratio

The air-fuel ratio (AF) is defined as a ratio of the mole of air to the one of fuel in the combustion process. The composition of biogas in this study contains air, leaking from the atomosphere to the storage tank when the water line of anaerobic fermentation pool is too low. Hence, the stoichiometric reaction for combustion of biogas with standard air is given as:

$$CH_4 + xCO_2 + y(O_2 + 3.76N_2) + zH_2O + (2-y)(O_2 + 3.76N_2) \rightarrow$$
$$(1+x)CO_2 + (z+2)H_2O + 7.52N_2 \tag{3.1}$$

where x, y and z are the moles of CO2, air and water vapor in the biogas, respectively. Both x and y can be measured by instruments, and z can be obtained from the absoulate humidity(ω) of biogas. Since the water vapor is considered as an ideal gas, the percentage of vapor from biogas can be calculated as follows:

$$\text{Mole Fraction of } H_2O \text{ in Biogas}(\%) = \frac{18}{16\alpha+44\beta+28.8\gamma}\frac{P_v}{P_{biogas}-P_v} \tag{3.2}$$

where α, β and γ stand for the percentages of CH4, CO2 in biogas and air in biogas, respectively. P_{biogas} is the pressure of biogas and P_v is the vapor pressure in biogas, which is obtained from:

$$P_v = \Phi P_g \tag{3.3}$$

where Φ is the relative humidity, measured by instrument, and P_g the saturation pressure of vapor at the same temperature.

The stoichiometric air-fuel ratio, AFstoich, is:

$$AF_{stoich} = \frac{\text{mole of air}}{\text{mole of } CH_4 + \text{mole of } CO_2 + \text{mole of air in biogas} + \text{mole of } H_2O}$$
$$= \frac{(2-y)\times 4.76\text{mole}}{(1+x+y\times 4.76+z)\text{mole}} \tag{3.4}$$

On the other hand, AFact is the air-fuel ratio of the actual mole of the air to the summation of moles of the methane, CO2 and air in biogas into the engine. Because the mole ratio is equal to the volume flow rate ratio, and the summation of the methane, CO2, air and water vapor in biogas flow rate is equal to the biogas flow rate. AFact can be also expressed as:

$$
\begin{aligned}
AF_{act} &= \frac{(\text{mole of air })_{act}}{(\text{mole of } CH_4 + \text{ mole of } CO_2 + \text{mole of air in biogas} + \text{mole of } H_2O)_{act}} \\
&= \frac{\text{Air flow rate}}{CH_4 \text{ flow rate} + CO_2 \text{flow rate} + \text{air flow rate in biogas} + H_2O \text{ flow rate}} \\
&= \frac{\text{Air flow rate}}{\text{Biogas flow rate}} \qquad (3.5)
\end{aligned}
$$

The air flow rate can be measured by air flow meter directly, whereas the methane flow rate is obtained by the measured biogas flow rate multiplied by the mole fraction of methane (both flow meters were demonstrated in sections 3.1.2 and 3.1.3).

The Excess Air Ratio (λ) is the ratio of the actual mole of air used to the stoichiometric mole of air, defined as:

$$
\lambda = \frac{(\text{mole of air})_{act}}{(\text{mole of air})_{stoich}} = \frac{(\frac{\text{mole of air}}{\text{mole of fuel}})_{act}}{(\frac{\text{mole of air}}{\text{mole of fuel}})_{stoich}} = \frac{AF_{act}}{AF_{stoich}} \qquad (3.6)
$$

Note that the actual mole of fuel is equal to stoichiometric mole of fuel because in the engine experiments the fuel supply rate is fixed, whereas the air volume flow rate is changed. As a consequence, the excess air ratio is equal to ratio of AFact to AFstoich. Also remind that λ is reciprocal of equivalence ratio.

3.2.2 Thermal Efficiency

The thermal efficiency is calculated for how much energy converting into electric power, its formulation is as following :

$$Thermal\ Efficiency = \frac{Actual\ Power\ Generation}{Energy\ Input} \tag{3.7}$$

The actual power generation of this study is the net output of turbine generator.The energy input is calculated from the lower heating value (LHV) of methane, whose value is 50020kJ/kg, in the biogas. It is expressed as:

$$Energy\ Input = \dot{m}_{CH_4} \times LHV\ \text{of}\ CH_4 \tag{3.8}$$

Where $\dot{m}_{CH_4}$ is the methane mass flow rate in biogas, and it is calculated by:

$$\dot{m}_{CH_4} = biogas\ flow\ rate \times CH_4\% \times \rho_{CH_4} \tag{3.9}$$

where ρ_{CH_4} is the density of methane

3.3 The Effect of Varying loads

With the increase in operating load, engine speed is also rise. To study the operating load and the thermal efficiency of gas turbine engine, the design rated powers are set from 15 to 30kW, and the increment of power output is assigned as 1 kW for each test under the same environmental conditions. Variation of operating load will make impact on the generator performance. When the output power is adjusted to the 30kW, then the engine speed reaches 96,000 rpm. At a higher operating load, the heat energy is used fully, resulting in higher thermal efficiency [16]. As mentioned above, this study used the hydraulic free piston engine, which is different from the turbine engine.

One of the experimental parameters is methane concentration of biogas. Before experiment, the methane concentration of biogas is measured. The collected data in experiment include compositions of biogas, engine speed, air flow rate, biogas flow rate, air-fuel ratio and actual power generation. The measurements start as the engine is operating continuously until all conditions are ensured to be steady. Then, all measurements are taken twice and make an average. The experimental procedure is as follows:

1. Measure the methane concentration of biogas.
2. Operate the engine at least 10 minutes so it would be steady.
3. Collect all of the measured data.
4. Adjust the output power at demanded quantity.
5. Repeat the procedure for different output power.
6. Repeat the above procedure for different methane concentration.

3.4 Uncertainty Analysis

The accuracy of the experiment data should be confirmed before the analyses of experimental results are carried out because the exactitude of the data may not be very good. Error analysis is a procedure used to quantify data validity and accuracy. Experimental measuring results are always in errors. Experimental errors can be classified into fixed error and random error respectively. Fixed error is the same for each reading and can be removed by proper calibration and correction. Random error is different for every reading and hence cannot be removed. The objective of uncertainty analysis is to estimate the probable random error in experimental results.

3.4.1 Uncertainty Analysis of Volume flow Meter

The apparatuses must be corrected by other standard instrument to make sure that they can normally operate and let the inaccuracy of the experimental results reduce to minimum. In this study, the major sensor in the experiment was the volume flow meter. The measurement range of P-050 Flow Meter adopted in this study was 45~450L/min±5%.

3.4.2 The Experimental Repeatability

To verify experimental accuracy, perform one test under the loads varying from 15 to 30 kW two times to ensure experimental repeatability. The evaluation included three measurements for volume flow rate. Standard deviation is defined as the absolute difference among the three volume flow rates. Table 3.4, 3.5 and Fig. 3.12, 3.13 show the coefficients of variation (CV) and experimental error bars. The CV is defined as the ratio of standard deviation S to mean $\overline{X}$, where S is derived by

$$S=\sqrt{\frac{1}{N}\sum_{i=1}^{N}(X_i-\overline{X})^2}$$

The CV is a dimensionless number that can be used to specify the variation of data points in a data series around the mean. As the experiments were conducted outdoors, environmental conditions were difficult to control, for safety reason. As a consequence, the errors (＜4) in these experiments were expected to be higher than general experiment errors, but they should be acceptable.

Table 3.4 Experimental Repeatability for Thermal Efficiency for Piston Engine

Outpu Power (kW)	Biogas supply (L/min)	CV(%)
5.4	180.6	3.63
9.4	191.5	3.11
14.97	211.4	3.62
16.10	221.1	2.83
17.03	217.7	3.82
18.12	223.8	2.55
18.99	225.5	2.21
20.14	212.8	3.21
20.85	209.5	3.45
21.97	212.5	3.28
22.99	215.8	3.33
23.94	219.4	2.25
24.79	221.1	2.40
25.75	223.4	2.31

Table 3.5 Experimental Repeatability for Thermal Efficiency for Turbine Engine

Power generation(kW)	Biogas Flow Rate(L/min)	CV(%)
14.83	184.9	3.41
15.89	189.7	3.22
16.94	193.2	3.42
18.05	200.1	3.82
18.97	207.0	3.61
19.92	213.9	2.63
20.89	220.8	3.01
21.98	227.7	2.22
22.91	234.6	2.13
24.04	241.5	1.81
24.84	248.4	2.42
25.17	248.4	2.14
25.23	251.8	2.10

Chapter 4

Results and Discussion

The experimental study, a continuous effort of Wu [6], is carried out with a 30kW gas turbine, whose operation, as mentioned previously, is quite different from the 30kW piston engine used previously by Lin [4], Huang [5] and Wu [6]. Both engines operate in different way because of the inherent design pholosophies.

4.1 Power Generation by Turbine Engine

The biogas used in this research is supplied from the anaerobic tank made of red plastic bag. The original biogas from the tank contains high concentration of H_2S, around 5000ppm. H_2S would corrode the engine severely without proper treatment. Therefore, an H_2S removal system is built up by using biological process, which is favorable to environment protection and cost friendly. The removal rate of screened microorganism could remove at most 99% of H_2S from the biogas. In other words, the H_2S concentration in the biogas is effectively reduced from 5000ppm to 50ppm.

The desulfurized biogas passes a methane concentration analyzer and gas analyzer, which can measure the concentrations of oxygen, carbon dioxide and NO_x. The resultant measurements indicates that the biogas comprises 67% of CH_4, 9.2% of CO_2 and 5.6% of O_2. It is unlikely that the biogas contains O_2 after the anaerobic process, indicating an existense of air leakage from atmosphere to storage tank. According to the concentration of O_2 in the biogas, the corresponding concentration of Air is deduced as 19.38%. Besides, the content of water vapor in biogas is

1.6% at 30°C derived from Eq. (3.15) in Section 3.2.4, while the relative humidity of biogas is considered to be 100%. These data are summarized in Table 4.1.

Table 4.1 Compositions of Biogas flown into Turbine Engine

CH_4	CO_2	Air	H_2O	Residues
67%	9.6%	19.38%	1.61%	2.41%

Table 4.2 shows the measured and derived data as a function of specific power demand of CR30 gas turbine. The rated power output, actual power generation and engie rotational speed are directly read from the engine itself. Biogas flow rate is obtained from measurement and both CH_4 consmption rate and thermal efficiency are derived from the reading data. It can be seen that the the rated power output is greater than actual power generation because part of the turbine work generated is needed to drive the compressor. The corresponding data are illustrated in Fig. 1.

The maximum power output is only 25.23kW under the rated power output of 30kW, since the rotational speeds behind 25kW can no longer increase propotionally but approach a limit value, 96000rpm approximately, as shown in Fig. 4.2. Therefore, it may conclude that the upper operation limit for this type engine is 25kW. On the other hand, the lower limit is 15kW, below which the engine cannot be drived.

As expected, the required biogas gas flow rate increases with an increase of power generation. As shown in Fig. 4.3, the their relatioship appears as a nearly postive linearity as the power output less than 25kW.

The CH_4 consumption rate is calculated directly by multiplying the biogass flow rate with 67% given in Table 4.1 under an assumption of fuel-lean combustion. There are two main reasons for turbine engine not able to calculate the consumption ratio of CH_4 as that for piston engine used by Lin [4], Huang [5] and Wu [6]. First, the air flow rates, measured by the flow meter at the engine inlet, are listed in Table 4.2. It is obvious that the amount of inlet air of turbine engine is much much larger than the one for piston engine given in the Table 4.4. It is because that most of inlet air is used to cool the turnine liner and downstream combustion product gas to avoid the heat damage to the turbine blades, and only small portion, whose quantity cannot be justified, is used for combustion. Therefore, it is impossible to evaluate λ. Figure 4.4 shows a schematic diagram of a turbine combustion chamber with its main components, burning and cooling zones. Furthermore, the concentration of CH_4 in flue gas is diluted so greatly by the secondary cooling air, illustrated in Fig. 4.3, that it is not expected to be measurable exactly. In the meantime, the percentage of O_2 consumed in combustion cannot be measured either, because the inlet λ is not known in advance. As consequence, the real consumption ratio of CH_4 cannot be calculated for turbine engine.

The calculation of thermal efficiency in Table 4.2 is obtained by Eqs. 3.7, 3.8 and 3.9 of Sec. 3.2.2. As mentioned previously, the input energy is assumed to be complete combustion of CH_4 in the biogas. It is found that the thermal efficiency increases with an increase of power generation. Figure 4.3 shows that the relationship is almost linear between 17 and 25kW of power generation.

Table 4.2 The measured and derived data as a function of specific rated power of CR30 gas turbine at 30℃

The rated power output(kW)	Actual Power generation (kW)	Engine Speed(rpm)	Biogas Flow Rate(L/min)	Air Flow Rate (L/min)	CH_4 Consumption Rate (L/min)	Thermal efficiency (%)
15	14.83	84510	184.9	10161.6	123.91	18.51
16	15.89	85704	189.7	10383.8	127.09	19.33
17	16.94	87054.	193.2	10652.7	129.44	20.24
18	18.05	88252	200.1	10904.1	134.06	20.82
19	18.97	89140	207.0	11038.6	138.69	21.15
20	19.92	90620	213.9	11377.7	143.31	21.49
21	20.89	91670	220.8	11629.1	147.93	21.84
22	21.98	92650	227.7	11833.8	152.55	22.28
23	22.91	93888	234.6	11991.6	157.18	22.54
24	24.04	95078	241.5	12284.2	161.80	22.97
25	24.84	96006	248.4	12465.2	166.42	23.08
26	25.17	96258	248.4	12400.9	166.42	23.39
30	25.23	96334	251.8	12459.4	168.70	23.12

4.2 Comparison with piston engine

Now a comparison is made with the piston engine, whose data are from the work of Wu [6]. Table 4.3 shows the details of compositions of biogas which is used by Wu [6]. With 69% CH_4 of biogas, the maximum power output for piston engine is 26.5kW and the corresponding biogas supply is 225 L/min. The minimum power output is 5.4kW at biogas supply of 181 L/min. Apparently, the power generation range is wider for the 30kW piston engine, which can be seen in Fig. 4.5 as well (The effective range is from 15 to 25kW for turbine engine).

Table 4.3 Compositions of Biogas flown into Piston Engine; Wu [6]

CH_4	CO_2	Air	H_2O	Residues
69%	13.3%	12.38%	1.99%	3.33%

The corresponding experimental results are given in next table.

Table 4.4 The experimental data of Piston Engine under various output power with 69% CH_4; Wu [6]

Actual Power generation (kW)	Air Flow Rate (L/min)	Biogas supply (L/min)	CH_4 Consumption Rate (L/min)	Thermal efficiency (%)
5.4	1146	180.6	124.64	6.7
9.4	1208	191.5	132.16	11.11
14.97	1357	211.4	145.88	15.87
16.10	1424	221.1	152.57	16.32
17.03	1454	217.7	150.24	17.53
18.12	1428	223.8	154.48	18.14
18.99	1528	225.5	155.64	18.87
20.04	1506	212.8	146.88	21.10
20.90	1550	209.5	144.62	22.35
21.97	1750	212.5	146.64	23.17
22.99	1733	215.8	148.95	23.87
23.94	1777	219.4	151.43	24.45
24.79	1717	221.1	152.56	25.13
25.75	1816	223.4	154.17	25.83
26.48	1981	224.9	155.244	26.37

Figure 4.3 shows the biogas volume flow rate under different output powers for these two engines. Remind that for piston engine, the obtained power generation is resulted from a specific biogas flow rate, whereas for piston engine, the biogas flow rate is obtained under a specific rated

power. It can be seen that the power generation increases as the biogas flow rate increases for piston engine.

The detailed performance comparions for both engines are summarized in Table 5, which is illustrated graphically in Fig. 5.

Table 4.5 Thermal Efficiencies and CH_4 Consumption Rates of Turbine and Piston Engines under Various Output Power

Piston Engine			Turbine Engine		
Actual Power generation (kW)	CH_4 Consumption Rate (L/min)	Thermal efficiency (%)	Actual Power generation (kW)	CH_4 Consumption Rate (L/min)	Thermal efficiency (%)
5.4	124.64	6.7	-	-	-
9.4	132.16	11.11	-	-	-
14.97	145.88	15.87	14.83	123.91	18.51
16.10	152.57	16.32	15.89	127.09	19.33
17.03	150.24	17.53	16.94	129.44	20.24
18.12	154.48	18.14	18.05	134.06	20.82
18.99	155.64	18.87	18.97	138.69	21.15
20.04	146.88	21.10	19.92	143.31	21.49
20.90	144.62	22.35	20.89	147.93	21.84
21.97	146.64	23.17	21.98	152.55	22.28
22.99	148.95	23.87	22.91	157.18	22.54
23.94	151.43	24.45	24.04	161.80	22.97
24.79	152.56	25.13	24.84	166.42	23.08
25.75	154.17	25.83	25.17	166.42	23.39
26.48	155.24	26.37	25.23	168.70	23.12

Figure 4.5 shows the thermal efficiency and CH_4 consumption rate for both turbine and piston engines as a function of load. It indicates that for turbine engine, the thermal efficiency increases with increasing power generation and reaches its maximum value around 23.39%. For the piston

engine [6], the maximum thermal efficiency can be reached is 26.37% at the CH_4 consumption rate of 155.2 L/min. The maximum efficency of turbine engine is 2.97% lower. For gas turbine, most of the air is used for cooling, the hot gas exhausted from combustor outlet is cooled down by jet flow discharged from dilution holes, which prevent turbine blade from heat damage [21]. Because of lots of the air is used for cooling the exhaust from the combustor outlet, the partial energy is absorbed by air flow during cooling process. The result leads to lower thermal efficiency for turbine comparing to that for the piston engine under the higher load operating range (21 to 30kW). However, it can be seen from this figure that under the low load operating condition, the turbine engine can provide higher performance with lower efficiency variation compared to those of piston engine. It is because that under the lower loading, the fuel supply rate becomes less that leads to a lower volumetric efficiency with fixed cylinder volume. Furthermore, the lower volumetric efficiency will result in an decrease of in-cylinder pressure and make the mixture to become not richer. Therefore, the combustion efficiency is lower than in the high load operating condition. To sum up, the results show that the operation of turbine engine is more stable than that of piston one. Besides, the lower load limit is 15kW for turbine engine, whereas piston engine still can be operated as low as 5.4kW. Table 4.2 also reveals that the thermal efficiency of turbine engine increases negligibly when the power demand changes from 25 to 30kW, revealing a consistent trend with the engine speed. Therefore, the halt to the thermal efficiency indicates that the engine has reached its threshold speed.

4.2.1 Waste Gas Analysis

The waste gas concentrations, including O_2, NO_x and CO_2, the data are presented in Table 4.6.

Table 4.6 The Measurements of the Waste Gas Constitutes and their Concentrations for Turbine

Actual Power generation (kW)	14.83	18.05	20.89	24.04	25.23
Waste gas temperature (℃)	577.8	577.9	576.6	576.7	576.1
O_2 (%)	16.9	16.7	16.5	16.3	16.2
NO_x (ppm)	-	-	-	-	-
Estimation values					
CO_2 (%)	2.10	2.18	2.27	2.37	2.41

The actual power generation at 25.23kW leaves less O_2 concentration in the waste gas than the others. Apparently, under the higher loading that the combustion becomes more completed, so the exhaust gas will contain lower levels of O_2.

For gas turbine, most of the air is used for cooling (Approximately 80 percent) , mention above, It is because that the waste gas contains large amounts of air, the O_2 concentration in waste gas is too large. BACHARACH ECA 450, used for measuring waste gas component data, which include the concentrations of oxygen, NO_x and carbon dioxide, in order To calculate NO_x, it referenced to a user defined Oxygen level of between 0 and 15%, however, we measure the concentrations of oxygen in the waste gas exceeds 15% that the analyzer can not calculate the concentrations of NO_x.

This study applies the measured O_2 data in the waste gas to estimate the corresponding CO_2 concentration and to calculate the mole number of waste gas compositions during calculation of combustion efficiency. The estimated CO_2 is derived by using Eq. (4.5). The percentages of CO_2 in waste gas in the combustion process are calculated as follows:

The balanced reaction is:

$$CH_4 + xCO_2 + y(O_2 + 3.76N_2) + zH_2O + \lambda(2 - y)(O_2 + 3.76N_2) \rightarrow aO_2 + bCH_4 + (2 + z - 2b)H_2O + (1 + x - b)CO_2 + 3.76(y - \lambda y + 2\lambda)N_2 \tag{4.1}$$

x, y and z are the moles of CO_2, moles of air and mloes of water vapor in the biogas, respectively. a and b are the moles of O_2 and CH_4, respectively, in waste gas. The CH_4 concentration in waste gas can be neglected (less than 0.1%), therefore, the combustion process can be regarded as complete combustion that b equals to zero.

From the collected data (Table 4.1), the reaction for combustion of methane becomes:

$$CH_4 + 0.143CO_2 + 0.061(O_2 + 3.76N_2) + 0.024H_2O + 1.939\lambda(O_2 + 3.76N_2) \rightarrow aO_2 + (2.024 - 2b)H_2O + (1.143 - b)CO_2 + 3.76(0.061 - 0.061\lambda + 2\lambda)N_2 \tag{4.2}$$

a can be calculated from the percent of O_2 in waste gas as follow:

$$Mole\ Fraction\ of\ O_2(\%) = \frac{a}{1 + x + 4.76(y - \lambda y + 2\lambda) + z} \tag{4.3}$$

Where $1 + x + 4.76(y - \lambda y + 2\lambda) + z$ is the total moles in waste gas, b is obtained from the atom balance as:

$$b = 1 - \lambda + \frac{a-y+\lambda y}{2} \tag{4.4}$$

The percent of CO_2 in waste gas can be calculated by:

$$Mole\ Fraction\ \ of\ CO_2(\%) = \frac{1+x-b}{1+x+4.76(y-\lambda y+2\lambda)+z} \tag{4.5}$$

To take two examples from the Table 4.6, the reaction becomes:

- **16.2% O_2 in waste gas (25.23kW)**

$$CH_4 + 0.143CO_2 + 0.061(O_2 + 3.76N_2) + 0.024H_2O\ + 9.454(O_2 + 3.76N_2)$$
$$\rightarrow 7.538O_2 + 2.024H_2O + 1.143CO_2 + 35.863N_2 \tag{4.6}$$

The percent of CO_2 in waste gas is 2.4%

- **16.9% O_2 in waste gas (14.83kW)**

$$CH_4 + 0.143CO_2 + 0.061(O_2 + 3.76N_2) + 0.024H_2O\ + 11.091(O_2 + 3.76N_2)$$
$$\rightarrow 9.174O_2 + 2.024H_2O + 1.143CO_2 + 42.014N_2 \tag{4.7}$$

The percent of CO_2 in waste gas is 2.1%

Apparently, the higher load operating provides higher CO_2 concentration than the lower ones. The reason is the same as before that the combustion becomes more completed, therefore, more CO_2 is generated.

For pistone engine, the waste gas concentrations, including O_2, NO_x and CO_2, are shown in Table 4.7.

Table 4.7 The Measurements of the Waste Gas Constitutes and their Concentrations for Piston Engine; Wu [6]

Actual Power generation (kW)	14.97	18.12	20.90	23.94	25.75
Excess air ratio	0.944	0.940	1.089	1.197	1.190
Waste gas temperature (℃)	498.3	502.4	504	506	506.5
O_2 (%)	2.3	2.1	3.7	5.6	5.2
NO_x (ppm)	586	860	1045	1479	1945
Estimation values					
CO_2 (%)	13.5	13.6	11.2	9.2	9.4

It can be found that the concentrations of oxygen and carbon as a function of excess air ratio. CO_2 and O_2 concentration decreases with excess air ratio increases. It also can be seen that lower concentration of carbon in waste gas for turbine comparing to that for the piston engine, it is because that the exhaust gas contain large amount of air for turbine, therefore, the ratio of carbon dioxide emissions is relatively lower in comparison to piston engine.

4.3 Theoretical Thermal Efficiency for CR30 Micro Turbine

A picture of the experimental measurements of the temperature data is depicted in Fig. 4.6. η_C is the isentropic efficiency of the compressor, and η_T is the isentropic efficiency of the turbine. It is expressed as:

$$\eta_C = \frac{W_{C,isen}}{W_C} \quad (4.8)$$

$$\eta_T = \frac{W_{T,isen}}{W_T} \quad (4.9)$$

Where $W_{out,net}$ is the net work output by turbine.It is expressed as:

$$W_{out,net} = W_T - W_C \quad (4.10)$$

Table 4.8 Nomenclature and List of Recorded Values of Experimental Measurements for Tubine

Symbol	Description	Value
η_{th}	Theoretical thermal efficiency	Estimation values
η_c	Isentropic efficiency of the compressor	0.81
$\dot{m}_{air}$	Air mass flow rate	16.125 kg/min
$\dot{m}_{CH_4}$	CH_4 mass flow rate	0.1743 kg/min
$C_{P,air}$	Specific heat capacity at constant pressure	1.011 kJ/kg
C_{P,CH_4}		2.304 kJ/kg
T_1	Compressor inlet temperature	303 K
P_1	Compressor inlet pressure	14.7 psi
P_2	Compressor outlet pressure	70 psi
T_3	Heat exchanger outlet temperature	711 K
T_4	Turbine inlet temperature	1182 K
T_5	Turbine outlet temperature	828 K
T_6	Exhaust gas temperature	576 K

Theoretical thermal efficiency is the ratio between the useful output of a device and the input, its formulation is as following :

$$Theoretical\ thermal\ efficiency = \frac{\mathrm{W_{out,net}}}{\mathrm{Q_H}} \tag{4.11}$$

As shown in Figure 4.6:

$$\dot{m}_{gas}=\dot{m}_{air}+\dot{m}_{CH_4} \tag{4.12}$$

$\dot{m}_{CH_4} = CH_4\ flow\ rate\ \times\ \rho_{CH_4}$ =0.1743 kg/min

$\dot{m}_{air} = Air\ \ flow\ rate\ \times\ \rho_{Air}$ =16.125 kg/min

$$Q_H=q_H\times\dot{m}_{gas} \tag{4.13}$$

$C_{p,air}$ =1.011 kJ/ kg

C_{P,CH_4}= 2.304 kJ/ kg

$$C_{p,gas} = \frac{C_{P,air}\times\dot{m}_{air}+C_{P,CH_4}\times\dot{m}_{CH_4}}{\dot{m}_{air}+\dot{m}_{CH_4}} = 1.024\ kJ/kg$$

$T_{2S}= T_1\times(\frac{p_2}{p_1})^{k-1/k}$ =472k

$\eta_c = 0.81 = \frac{T_{2S}-T_1}{T_2-T_1}$ T_2=511.6k

$q_H = h_4-h_3=C_{p,gas}\times(T_4- T_3)$

$W_C = \dot{m}_{air}\times C_{p,air}\times (T_2 - T_1)$

$W_T = \dot{m}_{gas}\times C_{p,gas}\times (T_4 - T_5)$

$\eta_{th} = \frac{W_T-W_C}{Q_H} = 0.306$

$$\eta_T = \frac{W_{a,T}}{W_{S,T}} = \frac{h_4 - h_5}{h_4 - h_{5,s}} = \frac{C_{p,gas} \times (T_4 - T_5)}{C_{p,gas} \times (T_4 - T_{5,s})}$$

$T_{5,s} = T_4 \times (\frac{p_5}{p_4})^{k-1/k} = T_4 \times (\frac{p_1}{p_2})^{k-1/k} = 756K$

$\eta_T = \frac{(T_4 - T_5)}{(T_4 - T_{5,s})} = 0.834$

$$\text{Combustion efficiency} = \frac{\dot{m}_{gas} \times C_{p,gas} \times (T_4 - T_3)}{\dot{m}_{CH_4} \times LHV\ of\ CH_4} = 0.893 \quad (4.14)$$

4.4 Energy Analysis for CR30 Micro Turbine

The engine's power output comes from the chemical energy released by combustion, but not all the resultant energy is used for turbine output. A partial energy loss is due to waste heat and friction. The mechanical work generated by the turbine blades has four different paths. The first one is the power output that is also the actual output of the engine. The second path is the loss of heat transfer, and the third path is a friction loss, including nozzle loss and vane loss. Finally, the fourth path is for the attachments horsepower, including a compressor and dryer. Not all the fuel energy supplied to the engine is released by the combustion process since the combustion usually is incomplete. Therefore, it is necessary to cosider the combustion efficiency for evaluating the thermal efficiency. The average of combustion efficiency of the engine is about 0.85~0.90. In other words, about 10~15% of energy is lost by the form of heat during combustion process. It shall comcentrate on the remaining energy. Figure 4.7 indicates that about 60% of energy is lost in the form of heat, 15% of energy is lost by the friction, and only about 25% of energy is is usedful.

4.5 Economic Benefits

In this section, the cost analysis is carried out for the piston and turbine engines. To evaluate the cost effectiveness of these two engines in order to know whose economic benefit is higher. In this part, the additional revenues are calculated as a result of additional electricity production in kWh. We can have a vision to know whether it is worth to replace piston engine by turbine engine in a swine farm.

The Benefit due to additional electricity generation in term of NT$ can be calculated as follows:

$$\text{Benefit} = \Delta W(\text{kWh}) \times 2.7\text{NT\$ per kWh}$$

From the study of Lin [4], the average biogas produced is around 0.078 m^3 per head pig per day, and the resultant erergy by using piston engine is 1.7 kWh per m^3 biogas. For turbine engine, the resultant energy is 1.55kWh per m^3 biogas according to the data in Table 4.1. Based on the current test data, the economy benefits in a scale of 3,000, 5,000 and 10,000-head swine farm can be estimated and summarized in the following table.

Table 4.9 Economy Benefits for 3,000 Scale Swine Farm per year

	Piston Engine	**Turbine Engine**
Biogas Production	85,410 m^3 / year	
Electricity Generation	145,000kWh / year	132,000 kWh / year
Electricity Charge saving	392,000 NT$ / year	356,000 NT$ / year
CO_2 Emission Reduction	3,000 tons	

Table 4.10 Captial Costs for 3,000 Scale Swine Farm per year

	Piston Engine	**Turbine Engine**
Equipment cost (Power generation system)	3,500,000 NT$	4,300,000 NT$
Electricity bill	24,000 NT$ / year	32,000 NT$ / year
Maintenance cost	100,000 NT$ / year	54,000 NT$ / year
personnel expenses	110,000 NT$ / year	110,000 NT$ / year
depreciation cost	380,000 NT$ / year	467,000 NT$ / year
Payback Period (Year)	21.8	26.8
Cost of electricity per kilowatt	4.23 NT$	4.92 NT$

Table 410 shows equipment costs for the turbine engine and piston engine in a scale of 3000-head swine farm. The equipment cost of piston engine is 18% lower than that of turbine engine.

Table 4.9 shows the electric energy in electricity production for the turbine and piston engines per year. For the piston engine, the electric energy can reach about 145,000kWh per year, and it can save 392,000NT$ of electricity charge per year, providing that the present electricity purchase charge is 2.7NT$ per kWh. On the other hand, the electricity generated by turbine engine can reach about 132,000kWh per year, and it can save 356,000NT$ of electricity charge per year. It estimated that a swine farm with a scale of 3,000 heads in Taiwan can decrease 3,000 tons of CO_2 per year.

The payback ratio may be calculated as follows:

$$\text{Payback period} = \frac{\text{Capital Costs}}{\text{Benefit}}$$

Table 4.10 shows the payback period for the different engines in a scale of 3000 swine farm. The times of recovery cost are 21.8 and 26.8 year, respectively. The payback period is lower for the piston engine. Besides, the investment costs of the turbine engine are higher and the maximum value of thermal efficiency is lower, leading to higher cost of electricity per kilowatt for turbine engine. So based on a cost benefit analysis, using the piston engine to generate electricity in a scale of 3000 swine farm has an advantage over the turbine engine.

If it increases from 3,000-head pigs to 5,000-head pigs in a swine farm, then it has different result and CO_2 reduction is 5,000 tons. Table 4.11 indicates that the net electricity production for piston engine is around 182,000 kWh/ year, and for turbine one is around 219,000 kWh/ year. The generated electrical energy of piston engine is 16.8% lower than the one of turbine engine. The times of recovery cost are 14.1 and 10.9 year, respectively; it can be observed that the disparity is less in this case. Besides, the cost of electricity per kilowatt for turbine engine is lesser than that of piston one. It is because the maximum operating time for piston engine is 20 hours a day in order to protect generator. The turbine engine can work around the clock. Therefore, if the biogas is enough to keep engine working incessantly, then the net electricity production for turbine engine will be greater than the piston engine. In a cost benefit analysis, the results show that the economic benefits of turbine engine is higher than that of piston engine in a scale of 5000 swine farm.

Table 4.11 Economy Benefits for 5,000 Scale Swine Farm per year

	Piston Engine	**Turbine Engine**
Biogas Production	142,350 m^3 / year	
Electricity Generation	182,000kWh / year	219,000 kWh / year
Electricity Charge saving	492,000 NT$ / year	591,000 NT$ / year
CO_2 Emission Reduction	5,000 tons	

Table 4.12 Captial Costs for 5,000 Scale Swine Farm per year

	Piston Engine	**Turbine Engine**
Equipment cost (Power generation system)	3,500,000 NT$	4,300,000 NT$
Electricity bill	30,000 NT$ / year	40,000 NT$ / year
Maintenance cost	100,000 NT$ / year	54,000 NT$ / year
personnel expenses	110,000 NT$ / year	110,000 NT$ / year
depreciation cost	380,000 NT$ / year	467,000 NT$ / year
Payback Period (Year)	14.1	10.9
Cost of electricity per kilowatt	3.4 NT$	3.0 NT$

Table 4.13 presents the results of the test from 10,000-head pigs in a swine farm. This test carried out two 30kW-generators and made the comparison between piston engine and turbine. The net electricity production for piston engine is around 350,000 kWh/ year (which is lower) and for turbine is 420,000 kWh/ year. Based on the data in this table, it can be found that the cost of electricity per kilowatt for piston engine and turbine is lower than the present electricity purchase charge (2.7NT$ per kWh).

Table 4.13 Economy Benefits for 10,000 Scale Swine Farm per year

	Piston Engine	**Turbine Engine**
Biogas Production	284,700 m^3 / year	
Electricity Generation	350,000kWh / year	420,000 kWh / year
Electricity Charge saving	945,000 NT$ / year	1,135,000 NT$ / year
CO_2 Emission Reduction	10,000 tons	

Table 4.14 Captial Costs for 10,000 Scale Swine Farm per year

	Piston Engine	**Turbine Engine**
Equipment cost (Power generation system)	5,000,000 NT$	6,500,000 NT$
Electricity bill	60,000 NT$ / year	80,000 NT$ / year
Maintenance cost	200,000 NT$ / year	108,000 NT$ / year
personnel expenses	110,000 NT$ / year	110,000 NT$ / year
depreciation cost	542,000 NT$ / year	700,000 NT$ / year
Payback Period (Year)	8.7	7.6
Cost of electricity per kilowatt	2.6 NT$	2.3 NT$

Chapter 5

Conclusions and Recommendations

5.1 Conclusions

This study carries out the 30kW turbine engine/generators experiments in a swine farm, and the dissertation analyze the performance of power generation on gas turbine through experimental evaluations. When the turbine engine starts, the air and the biogas are sucked into the engine. The flow meters, measure the air and the biogas flow rates, which are automatically adjusted according to the change in engine speed. It is important that not all of the air for combustion, part of the total air is used for cooling the hot gas exhausted from combustor outlet, which prevent turbine blade from heat damage. The sensors which include the biogas flow meter and manometer for the turbine were established to form our test stand. First of all, the compressor would increase the pressure and temperature of biogas by reducing its volume, the compressor outlet temperature is about 40℃, and the pressure of biogas is about 5.6kgf/cm^2. In the second place, the water vapor of biogas is removed by Freeze dryer, the dryer outlet temperature is about 36℃, and then the biogas will be stored in the biogas tank. Finally the fuel is mixed with air and ignited in the chamber. One of the parameters is the turbine operating loads. We collect the data, such as power output, waste gas concentrations, fuel flow rate under various operating loads from15 to 30kW. There are three parts in this research. In the first part, experiments were completed to evaluate the thermal efficiency. Secondly, the cost benefit analysis is carried out for the turbine engine. The revenues are calculated as a result of generating capacity in kWh per year, the revenue calculation is based on the present electricity purchase charge which is 2.7 NT$/kWh, and then the economic benefits were estimated by the data obtained by this research. Finally, a comparison with Wu's [6] results was made. It conducted a series of comparison tests by using turbine engine and piston engine.

According to above experiment results, this study can obtain the following conclusions:

1. The experimental results showed that the range of biogas flow rate to the turbine engine is from 184.9 to 251.8L/min with 67% CH4 of biogas under varying loads from 15 to 30kW, and the maximum power generation, the corresponding thermal efficiency and the CH4 consumption rate is 25.23kW, 23.12% and 168.7 L/min, respectively. For piston engine, the maximum power generation, the corresponding thermal efficiency and the CH4 consumption rate is 26.48kW, 26.37% and 155.2L/min, respectively. The threshold efficency of turbine engine is 2.97% lower. Compared with turbine engine, the fuel efficiency and thermal efficiency for piston engine is higher under the higher load operating range (21 to 30kW). However, the turbine engine can provide higher performance with lower efficiency variation compared to those of piston engine in range of 15~ 21 kW. Remind that the lower load limit is 15kW for this turbine engine,

whereas piston engine still can be operated as low as 5.4kW.

2. For turbine engine, the maximum power output is only 25.23kW, under the rated power output of 30kW, mention above, since the rotational speeds behind 25kW can no longer increase propotionally but approach a limit value, 96000rpm approximately, it may conclude that the upper operation limit for this type engine is 25kW. On the other hand, the lower limit is 15kW, below which the engine cannot be drived. The thermal efficiency increases with increasing power generation. The probable reason is that when the turbine at high loads with inlet guide vanes in open position, the gas turbine internal friction losses relatively small [9] compared to that when under the low load operating.
3. The average of combustion efficiency of the engine is about 0.85~0.90. In other words, about 10~15% of energy is lost by the form of heat during combustion process. It shall concentrate on the remaining energy. It indicates that about 60% of energy is lost in the form of heat, 15% of energy is lost by the friction, and only about 25% of energy is useful.

4. The average biogas produced is around 0.078 m^3 per head pig per day, and the resultant erergy by using piston engine is 1.7 kWh per m^3 biogas. For turbine engine, the resultant energy is 1.55kWh per m^3 biogas. The estimated overall economic benefits in Taiwan by using biogas for the swine farms with a scale of 3,000 and 5,000 heads can be reached as following:

- **For the swine farms with a scale of 3,000 heads**.

Turbine Engine:

Electricity generation: 132,000 kWh per year

Electricity charge saved: 356,000 NT$/ year.

Payback period: 26.8 years.

Cost of electricity per kilowatt: 4.9 NT$

Piston Engine:

Electricity generation: 145,000 kWh per year

Electricity charge saved: 392,000 NT$/ year.

Payback period: 21.8 years.

Cost of electricity per kilowatt: 4.2 NT$

Based on a cost benefit analysis, using the piston engine to generate electricity in a scale of 3000 swine farm has an advantage over the turbine engine.

- **For the swine farms with a scale of 5,000 heads**

Turbine Engine:

Electricity generation: 219,000 kWh per year

Electricity charge saved: 591,000 NT$/ year.

Payback period: 10.9years.

Cost of electricity per kilowatt: 3.0 NT$

Piston Engine:

Electricity generation: 182,000 kWh per year

Electricity charge saved: 492,000 NT$/ year.

Payback period: 14.1 years.

Cost of electricity per kilowatt: 3.4 NT$

The results show that the economic benefits of turbine engine is higher than that of piston engine in a scale of 5000 swine farm.

- **For the swine farms with a scale of 10,000 heads**

Turbine Engine:

Electricity generation: 420,000 kWh per year

Electricity charge saved: 1,135,000 NT$/ year.

Payback period: 7.6 years.

Cost of electricity per kilowatt: 2.3 NT$

Piston Engine:

Electricity generation: 350,000 kWh per year

Electricity charge saved: 945,000 NT$/ year.

Payback period: 8.7 years.

Cost of electricity per kilowatt: 2.6 NT$

5.2 Recommendations

1. Continue the turbine engine tests at varying ambient temperature.
2. Installing an inlet air cooling system in order to decrease the temperature of air entering the engine. With the lower temperature, the density of air will be larger.

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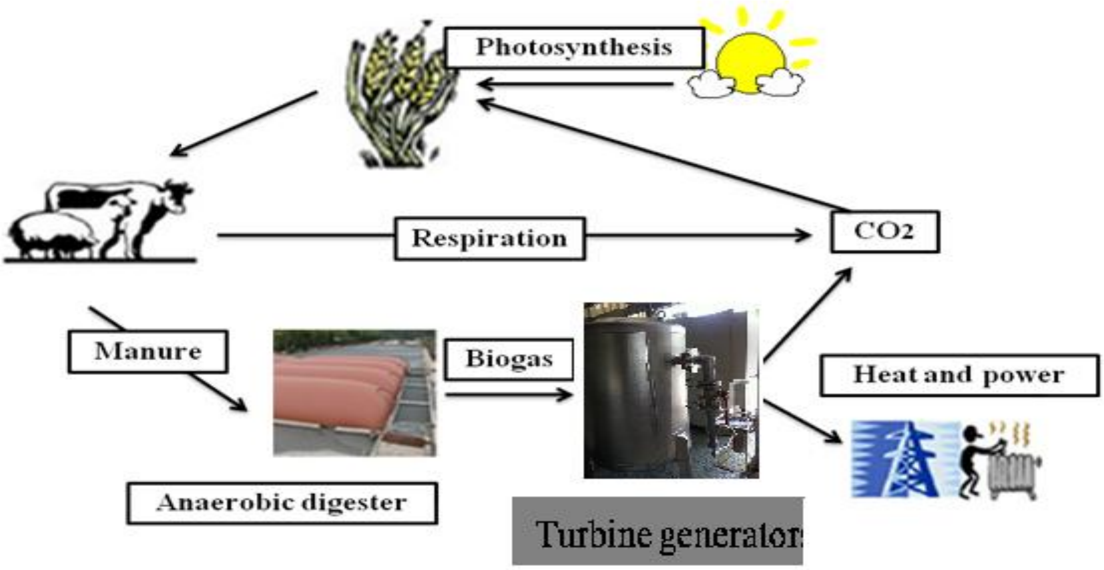

Figure 1.1 Simple Carbon Cycle for Biogas

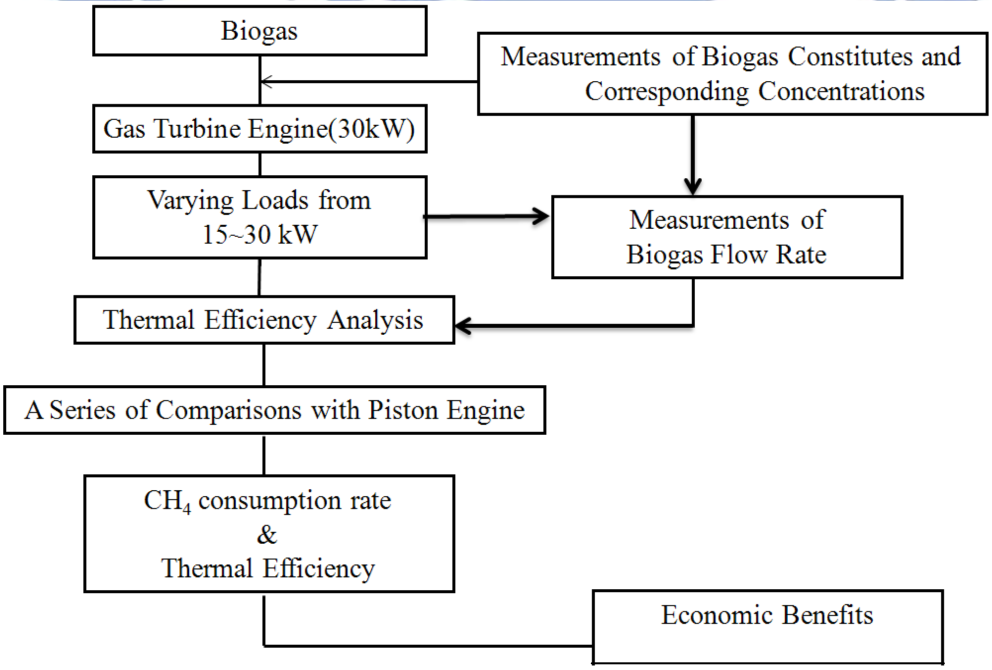

Figure 1.2 Scope of this Research

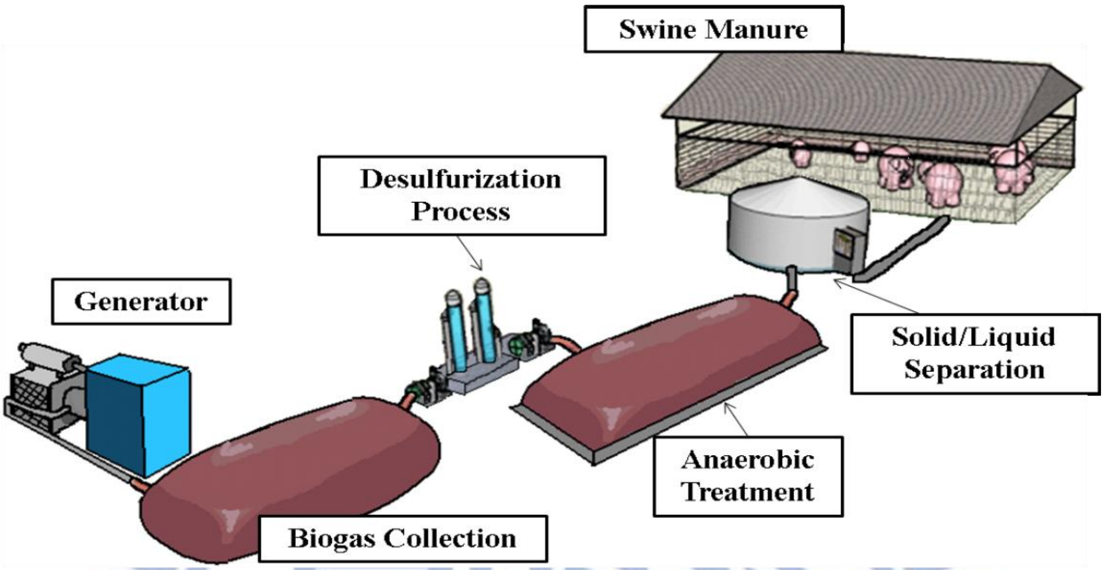

Figure 2.1 Process of Biogas Production

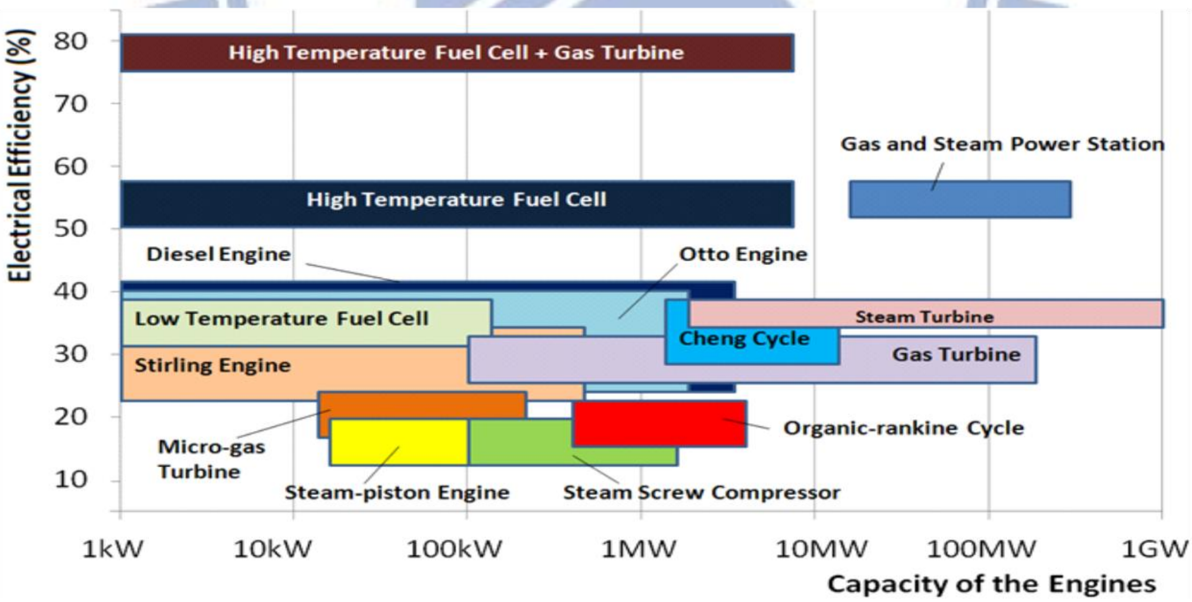

Figure 2.2 Range of Capacities for the Power Generators

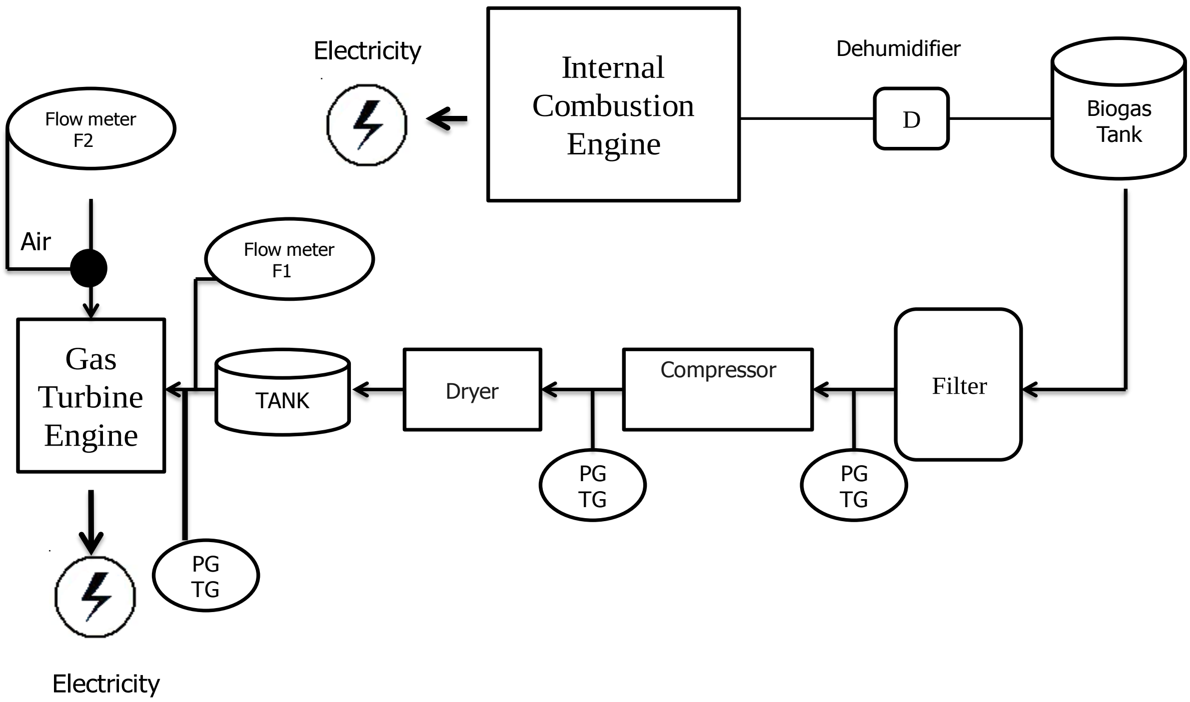

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Fig. 3.1 Experiment Layout

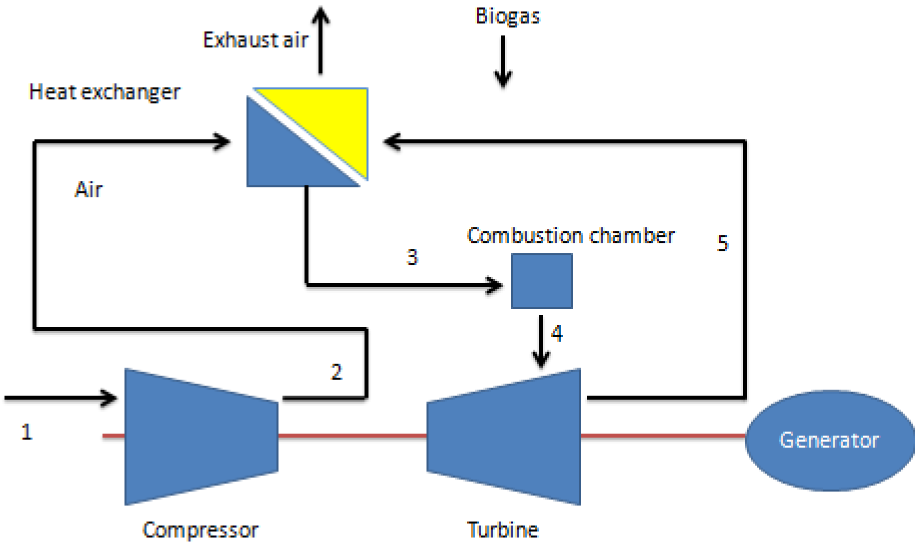

Fig. 3.2 Micro Gas Turbine: Scheme

Fig. 3.3 CR30 Micro Turbine Engine

Fig. 3.4a VA-400 Flow Sensor

Technical data VA 400

Measured unit	m³/h, m³/min, l/min, cfm
Accuracy	± (3% of measured value + 0.3% full scale)
Medium	Air, gas, non explosive
Operating temperature	-30 ~ 140 °C probe tube -30 ~ 70 °C casing
Operating pressure	Up to 50 bar
Analogue output	Signal: 4 ~ 20 mA Scaling: 0 ~ max range
Pulse output	1 pulse per m³
Power supply	12 ~ 30 VDC, 100 mA

Fig. 3.4b VA-400 Flow Sensor Data

Fig. 3.5a P-050 Flow Meter

Measuring object	Gas(33℃ 5.6kgf/cm2G)
Range ability	10:1
Accuracy	±5%
Max Temp	120℃
Max Press	0.8 MPa
Scale range	45~450L/min
Mass	2.0 kg
Material	SUS316

Fig. 3.5b P-050 Flow Meter Data

Figure 3.6 Dehumidifier (RD20)

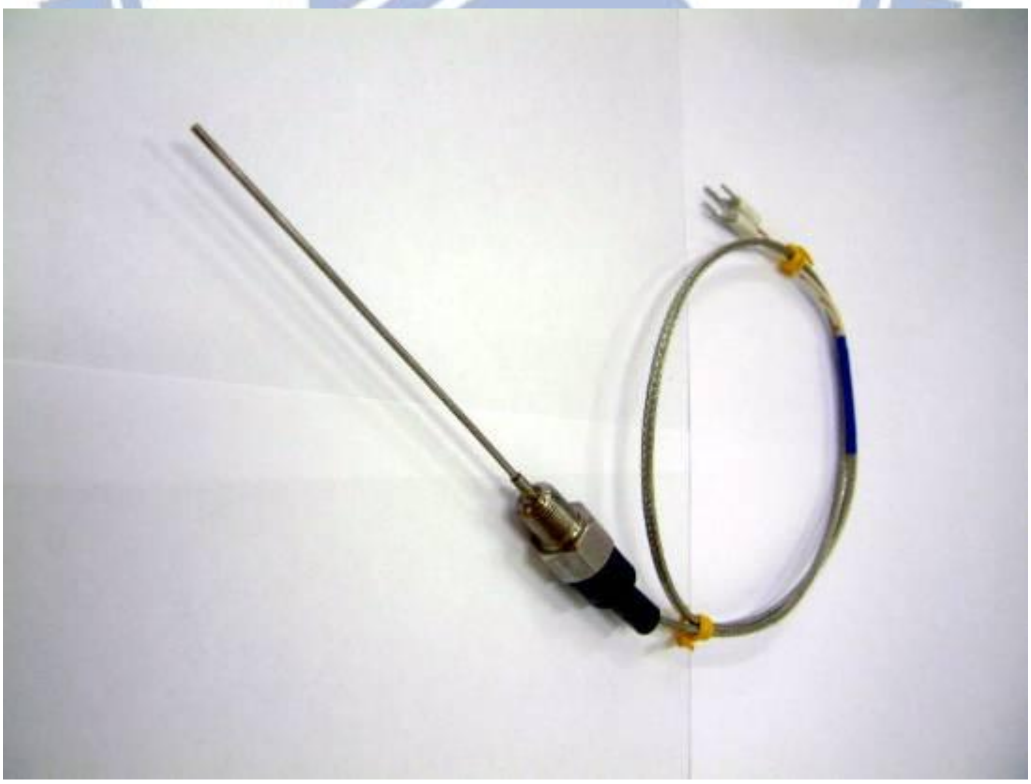

Fig. 3.7 K-Type Thermocouple

Figure 3.8 ECA450 Gas Analyzer

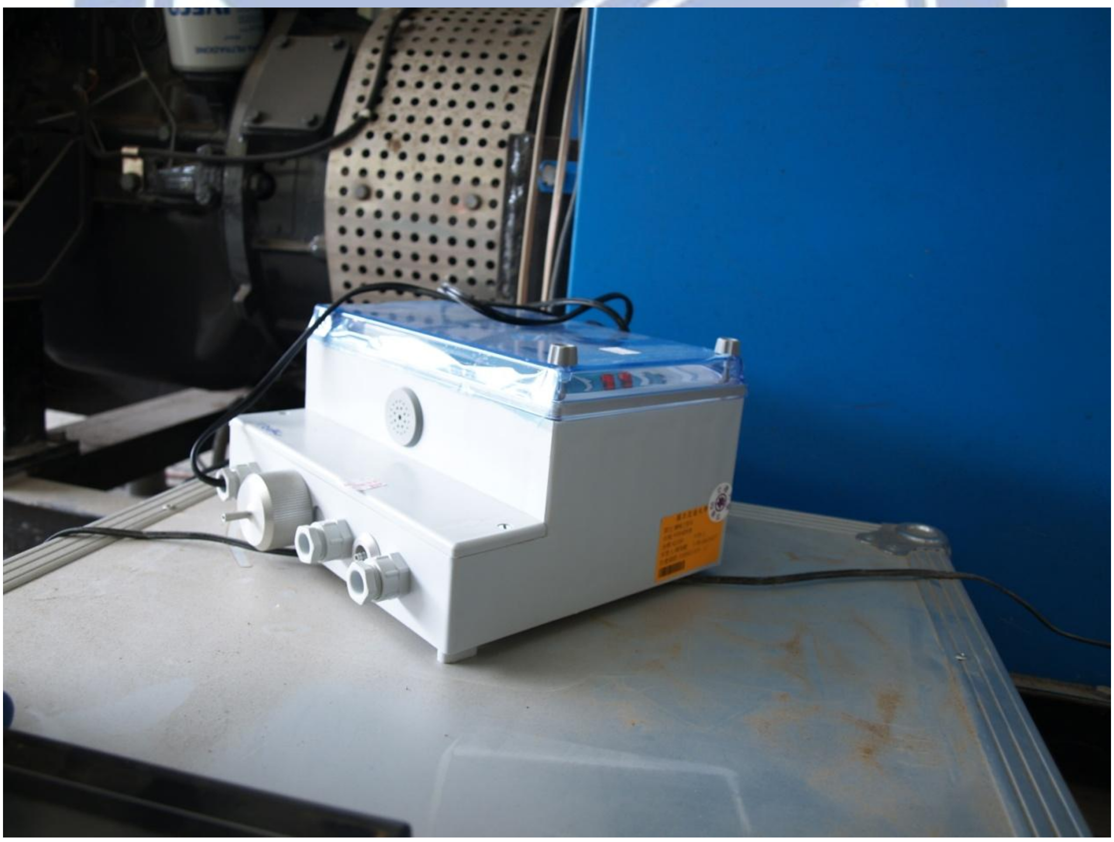

Fig.3.9 Guardian Plus Infra-Red Gas Monitor

Figure 3.10 JHTD3010-N Temperature with Humidity Transmitter

Fig.3.11 Center 311 Humidity Temperature Meter

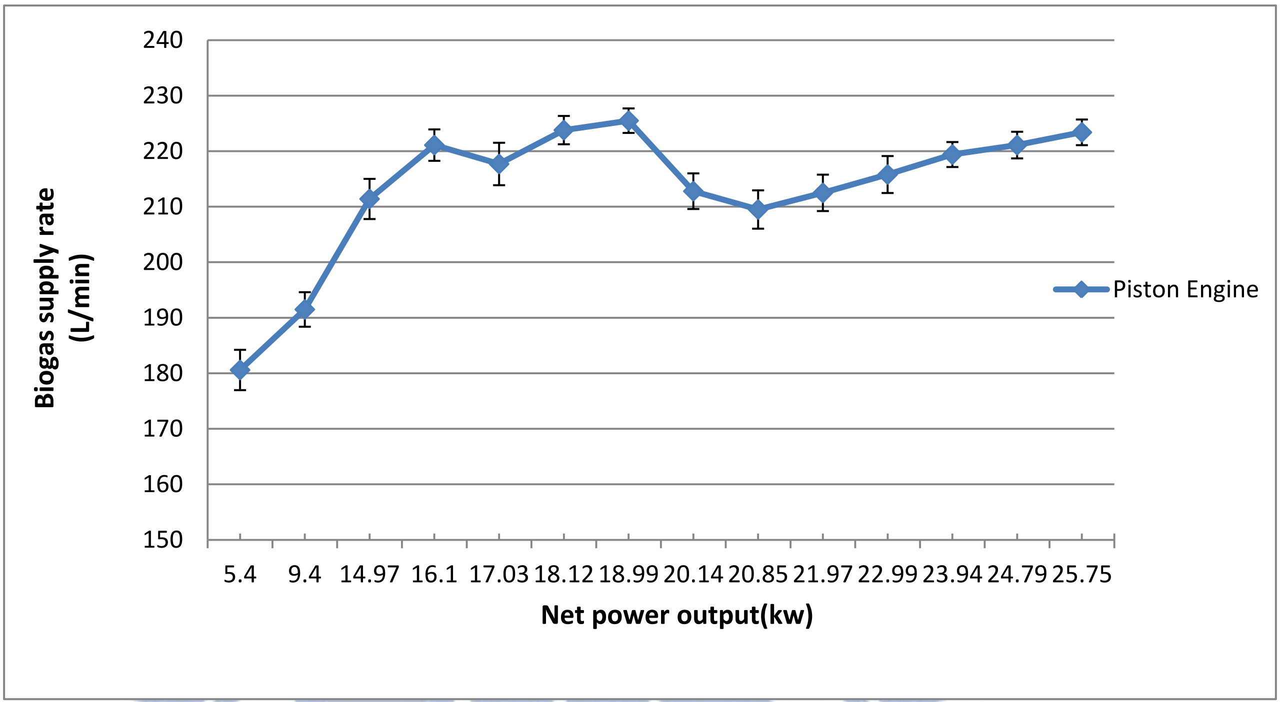

Figure 3.12 Experimental error bars for biogas supply rate for piston engine

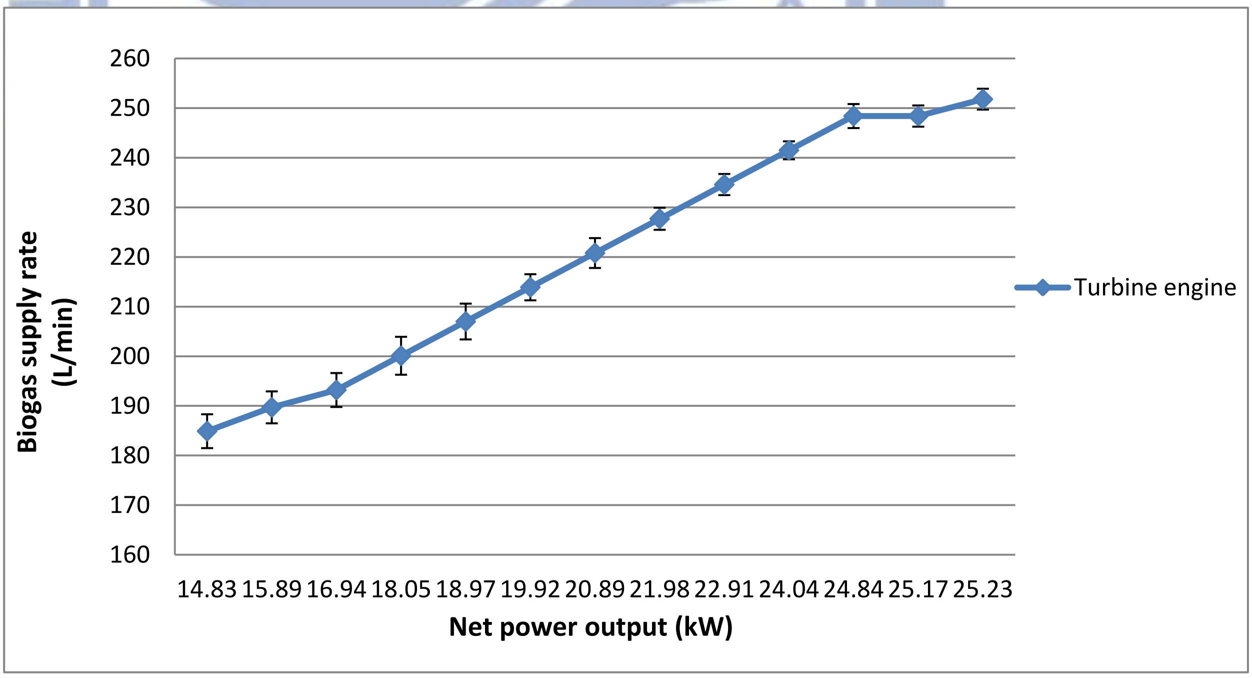

Figure 3.13 Experimental error bars for biogas supply rate for turbine engine

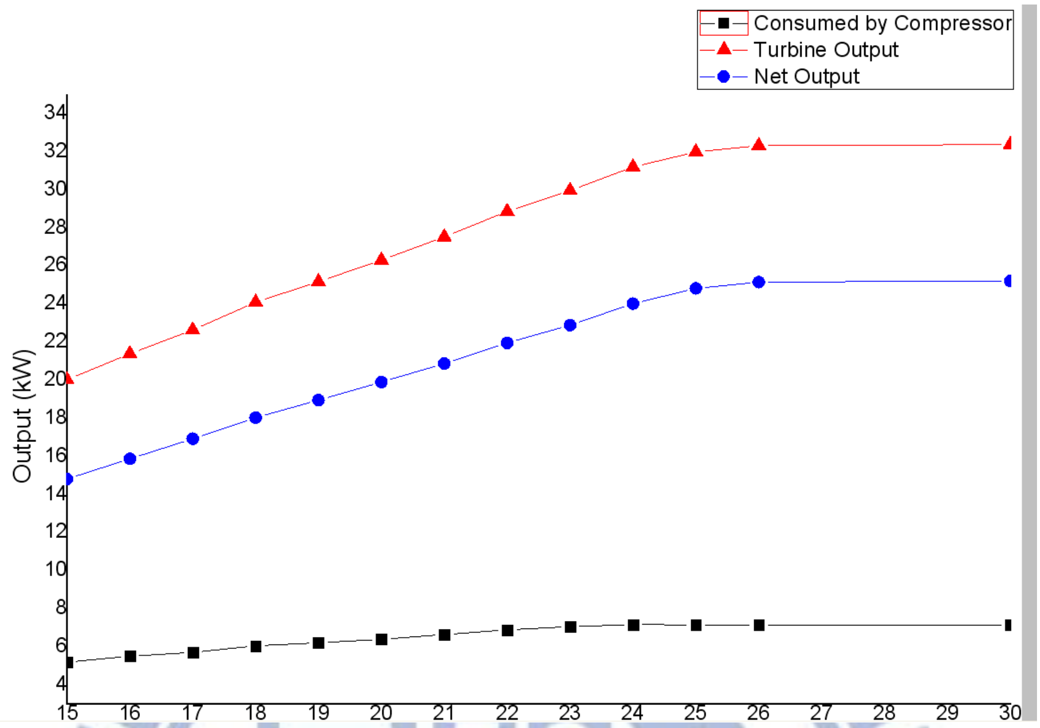

Fig. 4.1 The power consumption of the compressor

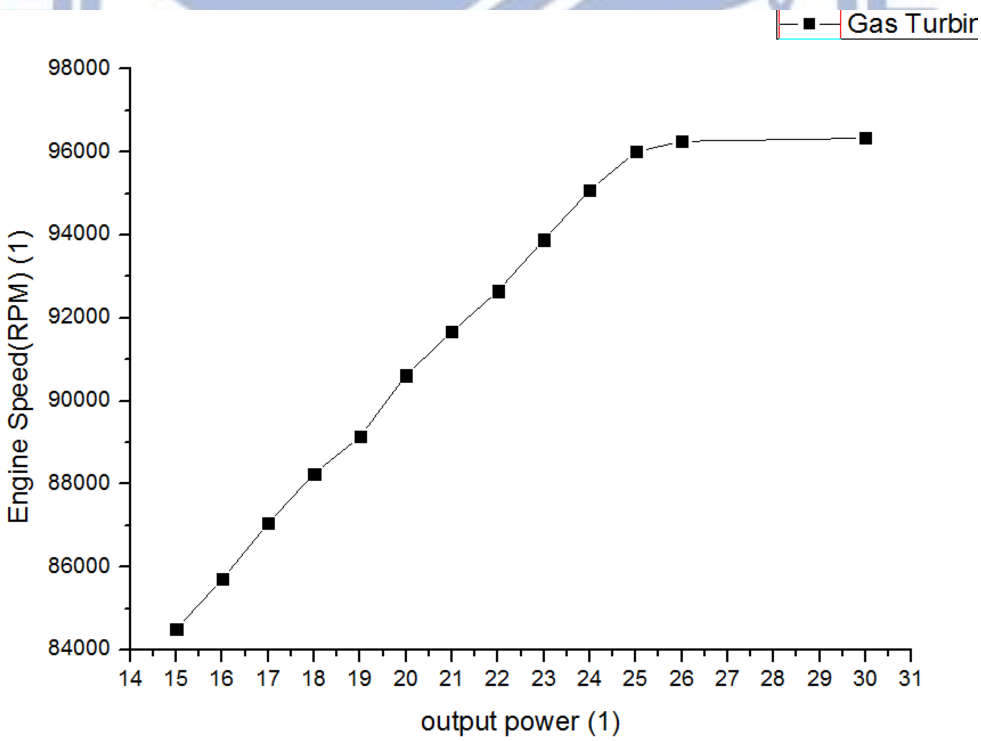

Fig.4.2 Engine speed v.s. output power

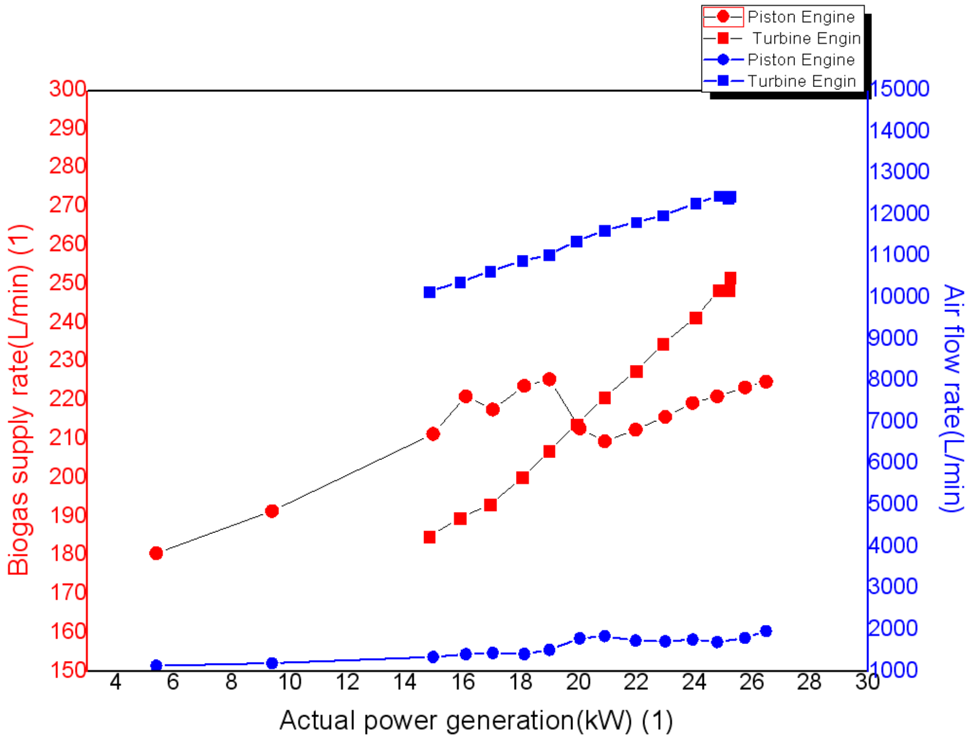

Figure 4.3Biogas supply rates and air flow rates v.s. actual power generation

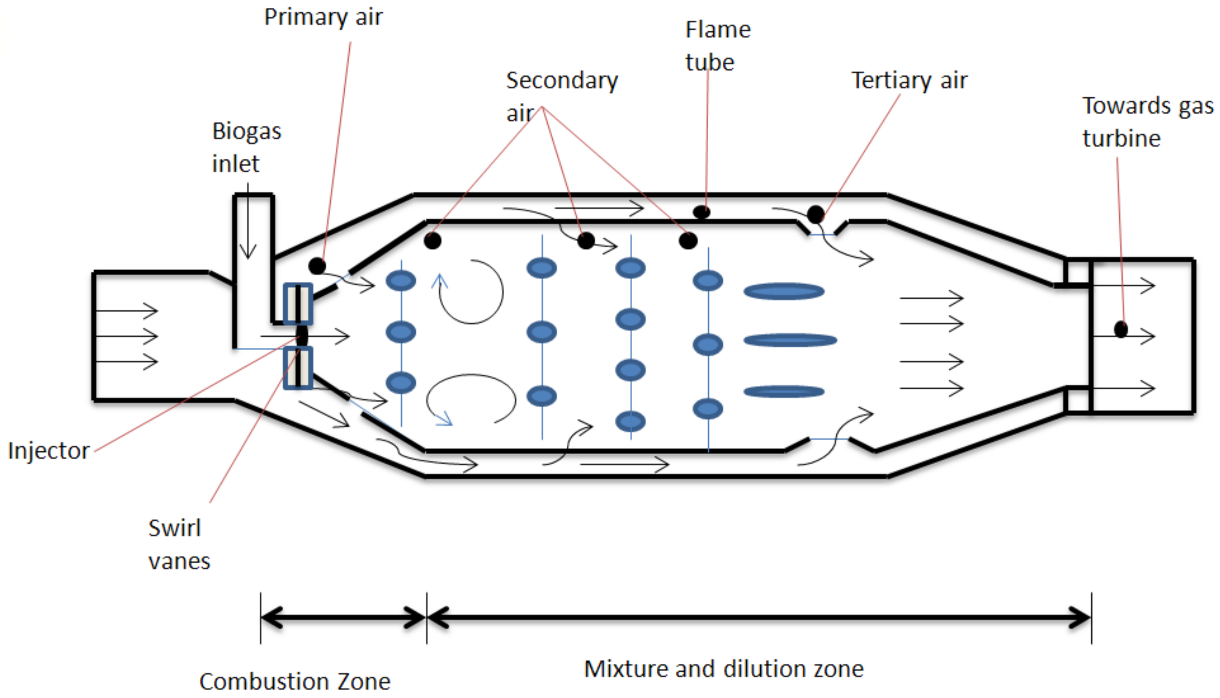

Figure 4.4 Schematic diagram of a turbine combustion chamber

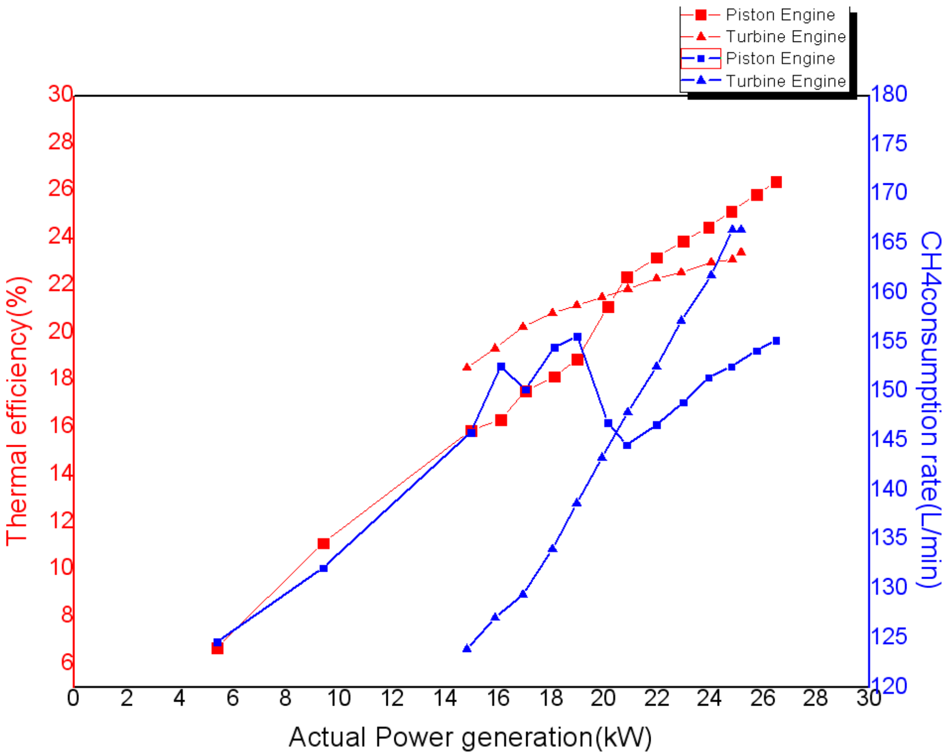

Figure 4.5 Thermal efficiency and CH4 consumption rate v.s. actual power generation

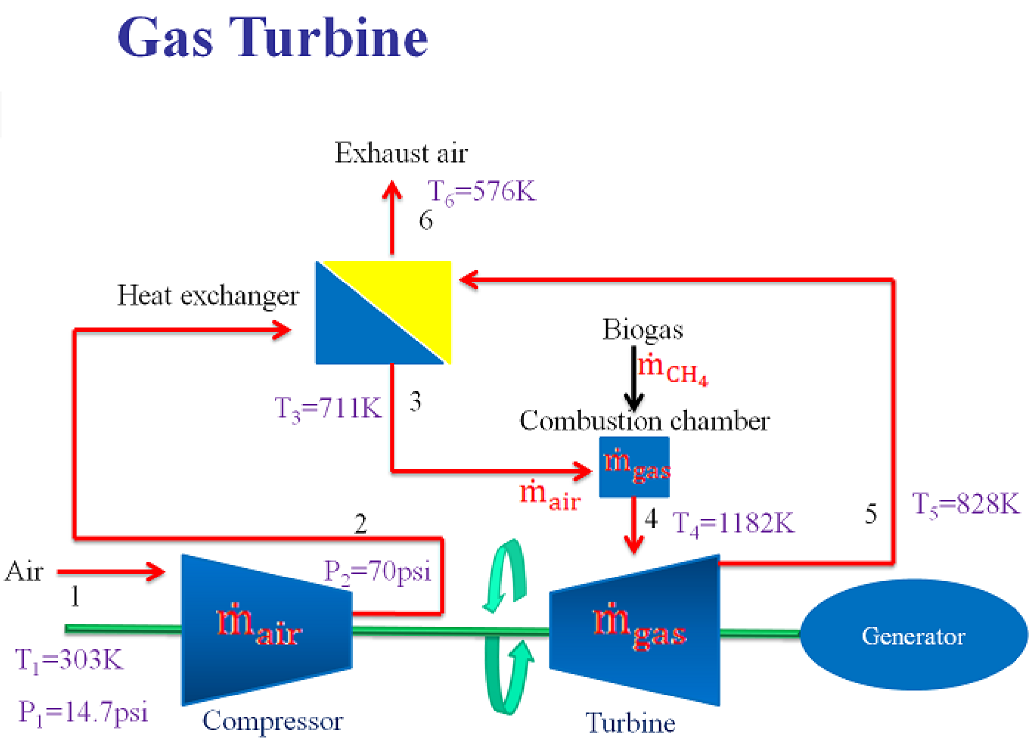

Figure 4.6The experimental measurements of the temperature

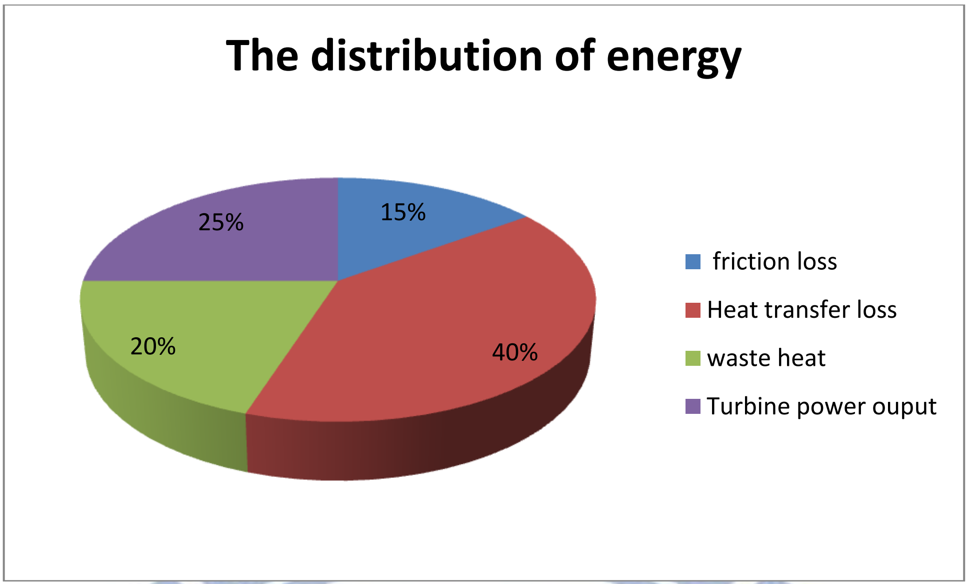

Figure 4.7 The energy distribution