INVESTIGATION INTO THE THERMODYNAMIC SUITABILITY OF A COMMERCIAL TURBOCHARGER FOR USE IN A MICRO GAS-TURBINE

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Abstract

Micro gas-turbines are expanding to be much more prevalent in the power generating market. They are merely scaled down versions of their larger siblings, gas-turbines powering commercial airplanes on generating megawatts of electrical energy throughout the world. The basic components of a micro gas-turbine and that of a turbocharger unit on internal combustion engines are quite similar. Both have a compressor, a heat source and a turbine. This study investigates the possibility of using a commercial turbocharger, designed for use on internal combustion engines, to function as a micro gas-turbine. The literature study discusses the components that make up a typical gas-turbine. Importantly, the literature explains the principles of operation and thermodynamic and/or mechanical relevance in a gas-turbine. Furthermore, the study shows a hand calculation procedure in order to calculate the excess power available from a turbocharger based on a fixed turbine inlet temperature, calculating the excess power for four different turbocharger Units. After the compressor and turbine characteristics are imported into FLOWNEX (a network solving software package) to recalculate the excess power, the results are compared with the results of the hand calculation. A specific turbocharger is selected to incorporate into a recuperated open cycle gas-turbine simulation. The cycle is firstly calculated by hand after which it is simulated in the Engineering Equation Solver (EES) to facilitate ease of modifications to the input parameters. The results from the cycle simulation are then compared to preferred system parameters. The conclusion is then made that a turbocharger is thermodynamically suitable to function as the core element of a micro-gas-turbine.

Opsomming

Mikro gas-turbines brei uit en neem reeds 'n meer prominente plek binne die kragopwekking industrie in. Hierdie mikro gas-turbines is klein skaal weergawes van die groot gasturbines wat gebruik word om vliegtuie aan te dryf of om elektrisiteit op groot skaal op te wek. Die basiese komponente waaruit 'n mikro gas-turbine bestaan stem nou ooreen met die komponente wat gevind word in 'n binnebrand motor se turboaanjaer. Beide het 'n kompressor, hitte bron en 'n turbine. Hierdie studie ondersoek die moontlikheid of 'n kommersiële turboaanjaer as 'n mikro gas-turbine kan funksioneer. Die literatuurstudie bespreek die basiese komponente wat in 'n gas-turbine voorkom. Verder verduidelik die literatuur die beginsels waarvolgens die komponente funksioneer asook die termodinamiese en meganiese toepaslikheid in 'n gas-turbine. Hierdie studie lê 'n handberekenings metode voor vir die berekening van die ekstra beskikbare krag van 'n turboaanjaer. Die berekening berus op 'n gegewe konstante turbine inlaat-temperatuur. Die prosedure word gebruik om die ekstra krag van vier verskillende turboaanjaer eenhede te bereken waarna dit met FLOWNEX bereken. Die FLOWNEX resultate word dan met die handberekening resultate vergelyk. Die turboaanjaer met die meeste ekstra krag word gebruik in 'n hitte herwinningsiklus simulasie wat eerstens per handberekenings opgelos is en daarna in die Engineering Equation Solver (EES) geprogrammeer word om die inset parameters makliker te kan verander.

Die resultate van die siklus simulasie word met sekere verlangde stelsel parameters vergelyk en die gevolgtrekking word gemaak dat 'n turboaanjaer termodinamies geskik is om te funksioneer as die kern van 'n mikro gas-turbine.

Acknowledgements

I wish to acknowledge all the people who supported me during the course of this study. In particular, my fiancé Tanja, who had to bare the brunt of my frustrations but always supported me with love and compassion. The lessons learnt were sometimes painful but the rewards thereof are priceless. It is a reassuring thought to know that mentors really care about subordinates and want them to grow as intellectuals.

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Nomenclature

Description	Symbol	Unit
Torque	τ	[N/m]
Rotational speed	ω	[rad/s]
Ratio of specific heat at constant pressure	γ	[ ]
Compressor efficiency	η_c	[%]
Turbine efficiency	η_t	[%]
Mechanical efficiency	η_m	[%]
Generator efficiency	η_g	[%]
Density	ρ	[kg/m^3]
Mass flow rate	m	[kg/s]
Radius	r	[m]
Tangential velocity at inlet to axial turbine	V_i	[m/s]
Tangential velocity at exit of axial turbine	V_e	[m/s]
Angular velocity at inlet to axial turbine	U_i	[m/s]
Angular velocity at exit of axial turbine	U_e	[m/s]
Power output	P	[kW]
Specific power output	P_x	[kW/kg]
Temperature	T	[K]
Pressure	p	[bar or Pa]
Specific heat at constant pressure for air	C_{pa}	[kJ/kg K]
Specific heat at constant pressure for gas	C_{pg}	[kJ/kg K]
Universal gas constant	R	[kJ/kg K]

Description	Symbol	Unit
Absolute velocity applicable to a radial turbine	C	[m/s]
Absolute velocity tangential component applicable to a radial turbine.	C_θ	[m/s]
Absolute velocity radial component applicable to a radial turbine.	C_m	[m/s]
Relative velocity applicable to a radial turbine	W	[m/s]
Non dimensional mass flow through a turbine	NDM_t	[ ]
Non dimensional speed of a turbine	NDS_t	[ ]
Pressure ratio over a turbine	PR_t	[ ]
Non dimensional mass flow through a compressor	NDM_c	[ ]
Non dimensional speed of a compressor	NDS_c	[ ]
Pressure ratio over a compressor	PR_c	[ ]

1. Introduction

1.1 History and development of micro turbines

In 1791 Barber wrote of the basic concept of a heat engine for power generation (Horlock, 2003). The efficiency of the components were increased by various designers and in 1939, Brown Boveri produced the first industrial gas-turbine unit. Due to the development of high-temperature materials, coatings and cooling systems together with increased pressure ratios of the compressors in gas-turbines, the efficiency thereof increased considerably over the last few decades (Boyce, 2002).

The greatest impact of the gas-turbine was in the field of aircraft propulsion. The initial development was for military aircraft and the objective was high speed. The first engines had very short existence, high fuel consumption and poor reliability. It took a lot of research and improvement before the first civil aircraft applications appeared in the 1950`s. After the Second World War and the development of more efficient gas-turbines for aviation, gas-turbines were modified for electricity generation and continues to be a discipline of active research (Saravanamuttoo *et al.*, 2001).

In recent times small gas-turbines, defined as micro gas-turbines, are developed for use in model aircraft and for modular power generating units. These power generation units can act as full time power supply or emergency backup units for large buildings.

1.2 Problem statement

Due to increased electricity demand and a lack of sustainable energy supply, there is a need for modular power generating units in the range of 10kW – 100kW throughout the world. According to Liedtke and Schultz (2003) micro gas-turbines in both single cycle and combined cycle applications have become increasingly important to the power market. The low installation costs and fast return of investment are some of the main virtues for power producers.

Numerous research institutes and commercial companies strive to develop small gas-turbines for both power generation and small-scale aviation purposes. Micro-turbines typically provide an electrical efficiency of about 30%, multi-fuel capability, low emission levels, heat recovery potential, and low maintenance. Onovwiona and Ugursal (2004) claim that an overall efficiency of 80% and above can be achieved with cogeneration applications.

A low cost product which generates power by means of a variety of fuels, requires investigation into alternatives options to lower the production cost of a micro gas turbine This study investigated a turbocharger unit as such an alternative to lower production cost. It was recognized that the components found in a turbocharger is similar to that found in a micro gas turbine. The internal combustion engine on which turbochargers is installed can in fact be seen as a combustion chamber powering the turbocharger. The challenge, however, is to establish a suitable thermodynamic turbocharger which has to function as a micro gas turbine.

1.3 Aim of study

The aim of this study is to determine whether a commercial turbocharger is thermodynamically suitable to be used in a micro gas turbine electric generator system.

1.4 Study objectives

The general goal of the research in this study was to determine the thermodynamic suitability of a turbocharger when applied as a micro gas turbine. Therefore, it includes research in all the relevant disciplines to understand the design and operation of a micro gas turbine. Turbochargers which are used in modern internal combustion engines could possibly function as a cost effective alternative compared to custom designed turbo machinery. This study will therefore investigate turbochargers thermodynamically and conclude to which extent the turbochargers can be used in a micro gas turbine.

1.5 Method of investigation

This investigation was done by conducting research on the various topics concerning the design of a micro gas turbine. The most appropriate cycle layout was selected and the various components thereof were considered in detail. Turbochargers were thermodynamically investigated to determine to which extent it can be used to construct a micro gas-turbine. This was done by calculating the excess power available from a turbocharger at conditions well within its specified operating limits of temperature and speed. The turbocharger was used in an open recuperated cycle simulation to calculate the overall net power and cycle efficiency.

1.6 Contributions of this study

This study contributes to innovation as it can be used to develop a viable product with the integration of components originally designed for use in other applications. This study confirms that a turbocharger can function thermodynamically suitable as a micro gas-turbine when connected to a recuperator and a heat source. It further contributes by accumulating knowledge on the design of micro gas-turbines for power generation and therefore it builds on the Faculty's focus in energy systems. This study serves as basic research for further studies on this topic.

2. Literature study

A literature study was done to derive a conceptual design of a micro gas-turbine. Therefore, this study's literature review, specifically focused on the basic theory and principles on which micro gas-turbines function must be considered. This includes a detailed study of all micro gas-turbines' components and each component's fundamental characteristics. Relevant literature examining the different components as well as its limitations assisted in concluding if components originally designed for other purposes can also be successfully used to function in a gas-turbine.

2.1 Micro-gas-turbine cycle configurations

Gas-turbines are mostly found in an open cycle configuration of which the layout of the cycle can vary according to the preference of the designer. Closed cycle gas-turbines exist but are rare due to the fact that the energy source used to power them must not change the characteristics of the working fluid. This is not a problem with open cycles as explained below.

2.1.1 Shaft layout

In addition to the different thermodynamical cycles, there are other mechanical layout differences that can occur with gas-turbines. Gas-turbines can be either of a single shaft layout or consist of a multi-shaft layout. A single shaft configuration fixes the compressor, turbine and generator on the same shaft. Therefore the rotational speed of all the components is equal and this has certain advantages and disadvantages when operational conditions are considered. Any variation of the load requirement will influence the operating speed of the unit. When there is a drop in the load requirement, the gas-turbine unit will tend to speed up, or it will speed down when there is an increase in the load requirement. This will therefore cause the compressor and turbine to operate at conditions other than the designated operating condition. Thus, care in design is important when envisaging a unit which incorporates a control system and accommodates the operating conditions on the respective characteristics without the occurrence of surge or choke. The fluctuation in the operating conditions influences the efficiency of the unit negatively.

When utilizing a multi-shaft layout like shown in Figure 2.1.2.2, this problem is taken care of. The power turbine operates independently of the gas generator. Therefore the gas generator can operate at a fixed point on its respective charts, ensuring optimum fuel efficiency of the unit. Any variation of the load is accommodated by the power turbine and separately controllable without influencing the gas generator. (Saravanamuttoo *et al.*, 2001)

2.1.2 Open cycles

Open cycle gas-turbines are traditionally used for engines producing around 1MW of power and more. In recent years micro turbines producing less than 200kW have been developed using similar open cycle configurations. The three main types of cycles are the Simple-Cycle, the Recuperated Cycle and the Combined Cycle (Alantec, 2005).

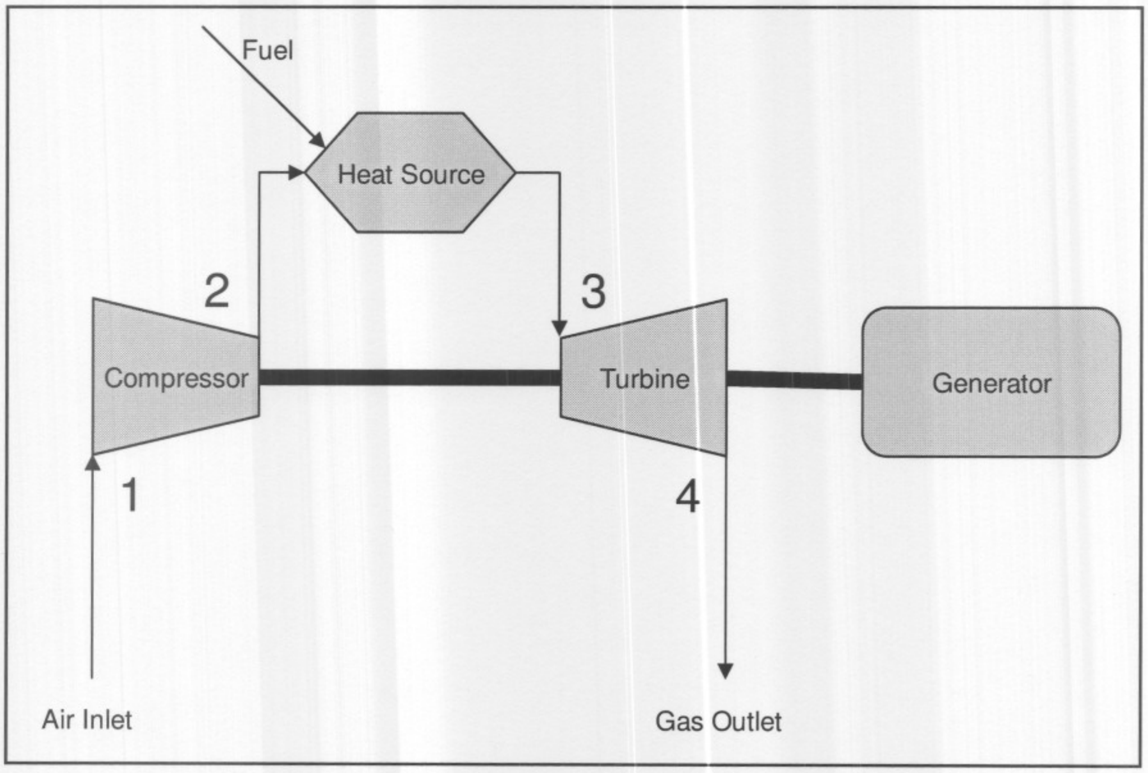

Figure 2.1.2.1: Simple Cycle layout.

Figure 2.1.2.1 is a schematic representation of a Simple-Cycle, single shaft gas-turbine used for power generation. Poullikkas (2004) explains that air entering the compressor at Point 1 is compressed to a higher pressure. No heat is added, however, compression raises the air temperature so that the air at the discharge of the compressor at Point 2 is at higher temperature and pressure. The air then enters the combustion chamber where fuel is injected and combustion occurs. The combustion process occurs in effect at constant pressure. High local temperatures can be reached within the primary combustion zone, but the combustion system is designed to provide mixing, burning and therefore cooling in critical regions. By the time the combustion mixture leaves the combustion system and enters the turbine at Point 3, it is at a mixed average temperature.

In the turbine section of the gas-turbine, the energy of the hot gases is converted into work. This conversion takes place in two steps. In the nozzle section of the turbine, the hot gases are expanded and a portion of the thermal energy is converted into kinetic energy. In the subsequent bucket section of the turbine, a portion of the kinetic energy is transferred to the rotating buckets and converted to work before the gas exits the turbine at Point 4. Some of the work developed by the turbine is used to drive the compressor, and the remainder is available for work at the output of the gas-turbine. Typically, more than 50% of the work developed by the turbine sections is used to power the axial flow compressor.

The simple cycle can be applied in various shaft configurations. Figure 2.1.2.2 shows the simple cycle layout in a twin shaft configuration consisting of a compressor, combustion chamber and turbine that forms a gas generator, and a separate power turbine that drives the load. The air is compressed between Point 1 and Point 2. After the addition of energy in the combustion chamber the hot combustion gas is expanded over the turbine between Point 3 and Point 4. Only enough energy is dissipated to power the compressor and the remainder of the energy is recovered with the free turbine between Point 4 and Point 5.

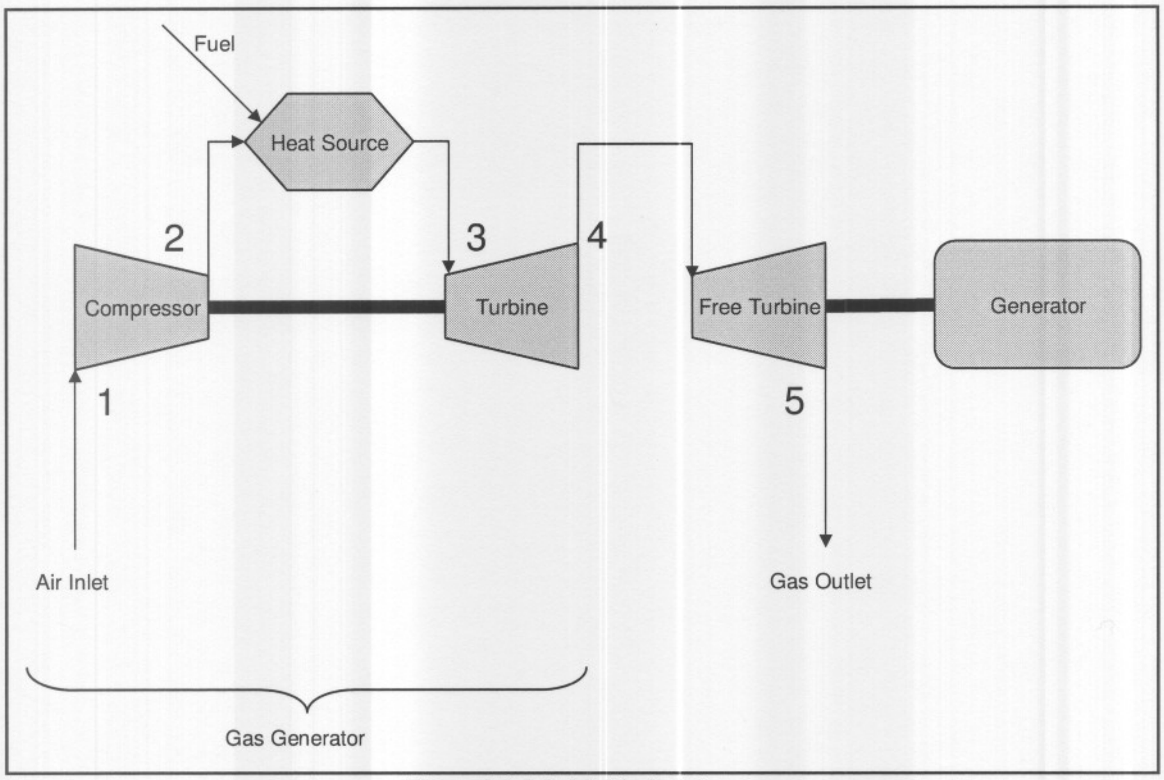

Figure 2.1.2.2: Multi-shaft layout.

The recuperated cycle multi-shaft layout is shown in Figure 2.1.2.3 and it differs from the simple cycle in Figure 2.1.2.3 only by the addition of a specialized heat exchanger called a recuperator. The recuperator captures exhaust thermal energy and heats the compressed air before it enters the combustion chamber. This is done to decrease the amount of fuel that has to be supplied to achieve the preset maximum temperature of the cycle. Therefore it increases the cycle efficiency and the recuperated cycle proves to be more efficient than the simple cycle.

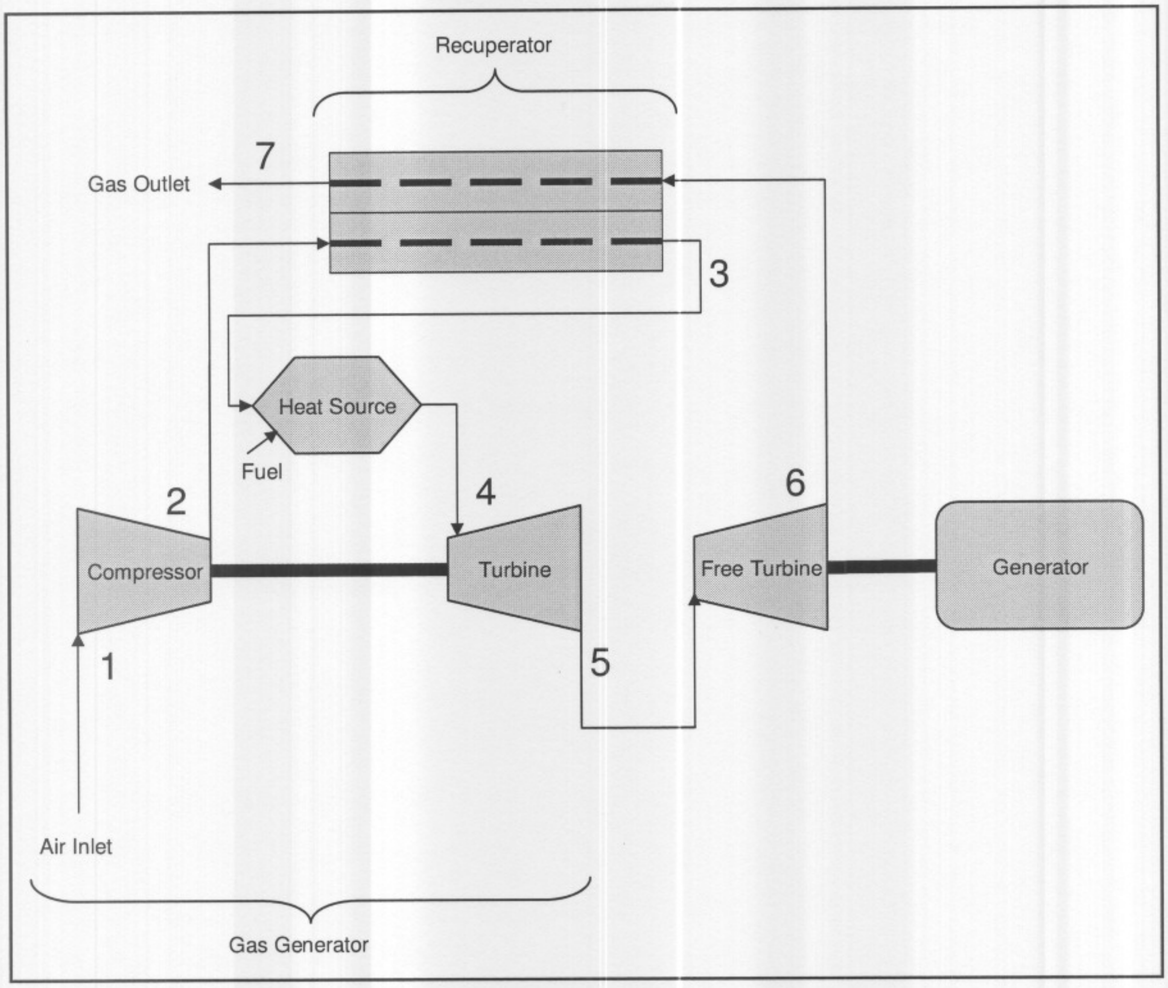

Figure 2.1.2.3: Recuperated cycle multi-shaft layout.

The combined cycle is similar to the simple-cycle, but it captures the exhaust gas from the power turbine and uses it to generate steam. The steam is then used to drive a steam turbine that contribute to driving the load or used directly in processes where steam is needed. The combined-cycle significantly increases power output and efficiency because it utilizes the exhaust energy that would have gone to waste. Figure 2.1.2.4 shows the layout of the combined-cycle.

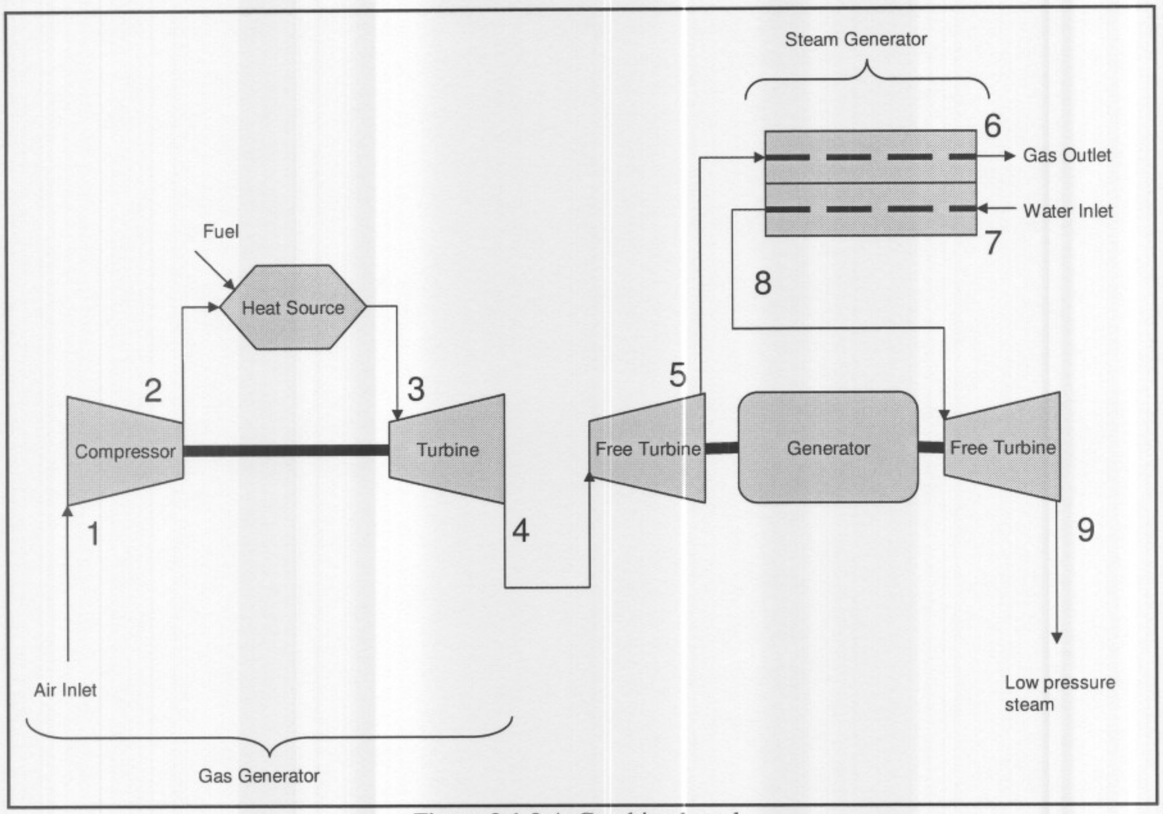

Figure 2.1.2.4: Combined-cycle

The air is compressed between Point 1 and Point 2. Energy is added to the air in the heat source where after it is expanded over the turbine between Point 3 and Point 4. The remainder of the energy is recovered with the free turbine. The hot gas leaving the free turbine at Point 5 is used to generate steam. Water is converted to steam between Point 7 and Point 8 where after it is expanded over a steam turbine and exhausted at Point 9. Both the free turbines are connected to a common generator.

2.1.3 Closed cycles

Closed cycles are similar to the open cycles in layout but the difference is that the working fluid does not exit the cycle. The fluid in the closed cycle must be heated by an external heat source, like nuclear reactors or heat exchangers, that will not change its attributes. Therefore closed cycles are not used where the heat source is fossil fuels.

2.1.4 Conclusion

Due to the small scale and application of normal micro-turbines they do not utilize closed cycles. Micro-turbines use open cycles where air is combusted with fuel to create the high temperature gas which is needed throughout the turbine. Although the Recuperated Cycle is more costly, it has the highest efficiency and therefore is most desirable for micro gas-turbines.

2.2 Micro gas-turbine components

Micro gas-turbines are in essence scaled down versions of larger commercial gas-turbines used for power generation or thrust applications in aviation. The components used in a micro gas-turbine are in essence the same as those used in larger gas-turbines. This section addresses the components that are typically found in micro gas-turbines.

2.2.1 Compressor

The compressor is the part of the gas-turbine that delivers the required mass flow for the system and raises the pressure of the gas entering the combustion chamber. There are two types of compressors that are commonly used in gas-turbines. The first is a centrifugal compressor that turns the flow through ninety degrees and whirls it radially outwards. Figure 2.2.1.1 shows a three dimensional view of a centrifugal compressor wheel. The second type is an axial compressor that forces the gas in the axial direction by means of blades rotating on a shaft like shown in Figure 2.2.1.2. The choice of which type of compressor to use depends mostly on the mass flow and the pressure ratio required from a single stage.

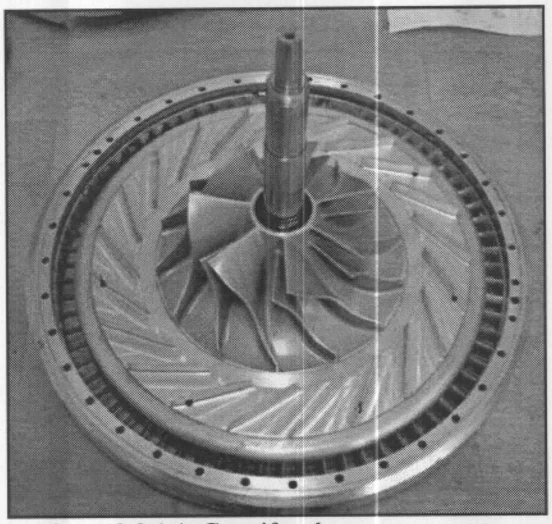

Figure 2.2.1.1: Centrifugal compressor rotor.

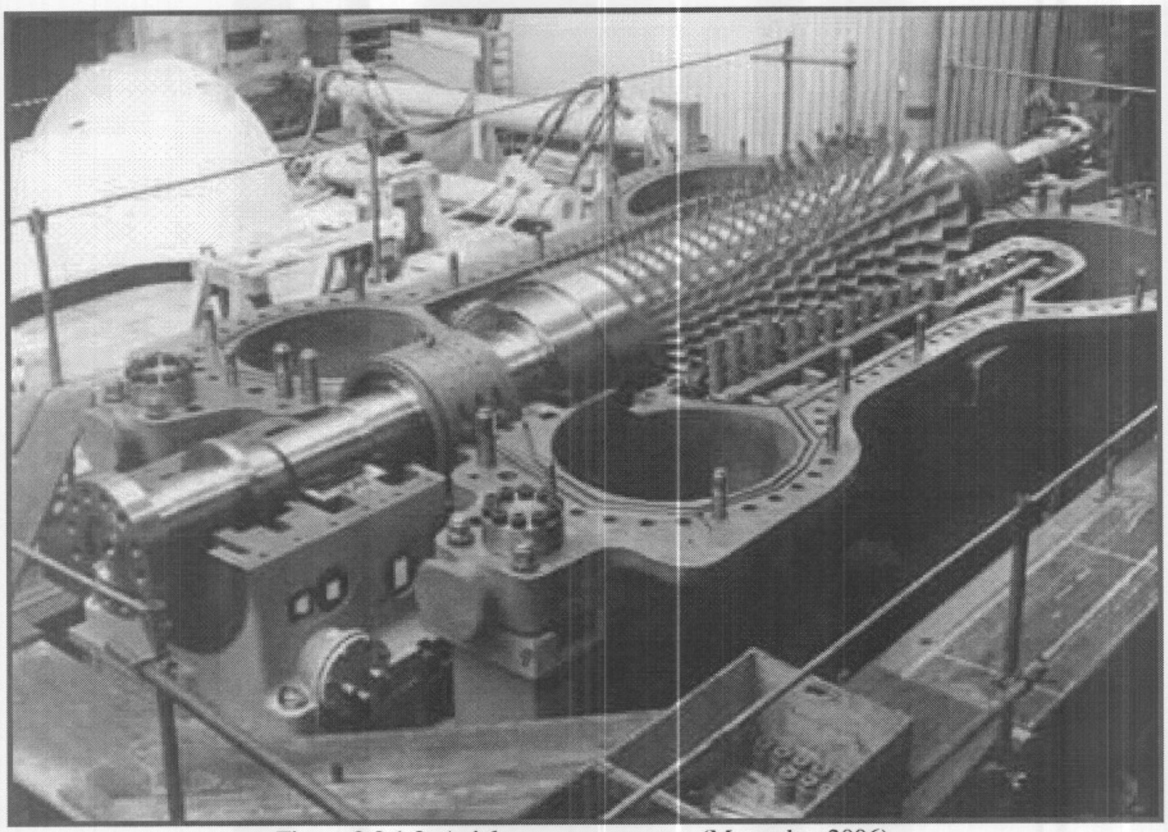

Figure 2.2.1.2: Axial compressor rotor. (Manturbo, 2006)

The most prominent differences between centrifugal compressors and axial compressors are associated with pressure ratio per stage and mass flow. Centrifugal compressors deliver a lower mass flow than axial compressors, but they can sustain higher pressure ratios per stage. Due to the fundamental fluid flow difference, a centrifugal compressor will have a larger frontal area than an axial compressor for a given mass flow. The efficiency of axial compressors is higher than centrifugal compressors for larger mass flows but where the mass flow is too small to be handled efficiently by axial blading, centrifugal compressors prove to be superior (Saravanamuttoo *et al.*, 2001; Rolls-Royce plc., 1996; Kim et *al.*, 2002)

In the conceptual design phase of any gas-turbine development project, the performance of the machine must be evaluated both for the design and the off design conditions. The characteristics of the compressor are mapped by the manufacturer to describe the off-design behavior of the compressor. This characteristic describes the behaviour of the compressor in terms of the pressure ratio, corrected mass flow and the efficiency as a function of the corrected speed. Therefore each point on such a map defines completely the velocity triangles for all blade rows (for axial compressors) and vane rows (for centrifugal compressors) in terms of Mach numbers (Kurzke & Riegler, 2000)

Surge is an operational condition associated with a drop in the pressure ratio, i.e. the delivery pressure, which can lead to pulsations in the mass flow and even reverse it, causing considerable damage to the compressor. A line representing this condition is indicated on the compressor characteristic and the designer must stay clear of it when defining the operational range of the gas-turbine. The other extreme operating condition of the compressor is known as choking. At this point the compressor restricts itself and no increase in flow through it is possible. Both these conditions are associated with low efficiencies (Haugwitz, 2002)

Thus,when an existing compressor is used in the design of a gas-turbine, the compressor characteristic that has been generated either by experimental data or reliable simulations, is an important source of information to use. The accuracy of the cycle simulation of the gas-turbine critically depends on the accuracy of the compressor and turbine maps.

2.2.2 Radial Diffuser

Radial diffusers are an integral part of the centrifugal compressor; it is responsible for almost 50% of the pressure rise in the compressor. Diffusion is the process of decelerating flow in order to create a pressure increase. The flow is decelerated by increasing the flow area, and thus decreasing the flow velocity. It may appear simple to accomplish, but there are many factors playing a role in this process. It is generally known that it is much more difficult to accomplish efficient deceleration of flow than to obtain efficient acceleration of flow. In the diffusion process the fluid has a natural tendency to break away from the walls of the diverging passage. The fluid then tries to flow back in the direction of the pressure gradient. This effect imparts losses and decreases the effective flow area at the inlet of the passage. Diffusion can be carried out in a much shorter flow path if the fluid is controlled. Therefore vaned diffusers are more efficient than vaneless diffusers for a particular operating point (Saravanamuttoo *et al.*, 2001)

Engeda (2003) has done a study on different types of diffusers. He classified the different types of radial diffusers as vaneless diffusers, vaned diffusers, and low-solidity vaned diffusers (LSVDs).

Vaneless diffusers consist of two radial walls that may be parallel, diverging, or converging. Because it offers a wide operating range it is commonly used in automotive turbochargers and process industrial compressors.

After the flow has passed through the compressor and diffuser it is collected with a scroll and fed through some piping arrangement to the next component in the system. This could be anything from a pressure vessel to an internal combustion engine or even a recuperator, which will be discussed next.

2.2.3 Recuperator

One of the critical components in low compression ratio micro-turbines is the recuperator which is responsible for a significant fraction of the overall efficiency of the micro-turbine and it is mandatory to include a recuperator in the micro-turbine layout to achieve efficiencies of 30% and higher. The 25 – 75 kW power generation market is currently dominated by low specific fuel consumption (SFC) diesel generator sets. The performance of these internal combustion diesel engines continues to improve. An efficiency of 42.9% can be obtained with some of these modern diesel engines that utilize turbochargers and inter-cooling.
Gas-turbines have the advantage of a large power-to-weight ratio. When considering that the improvement of compressor and turbine efficiencies are reaching a plateau, and the fact that the maximum inlet temperature are limited by materials and blade cooling, the recuperator is the crucial component that can be improved to enhance the cycle efficiency of gas-turbines (McDonald, 2000)

The recuperator is a heat exchanger that is used to preheat the air exiting the diffuser, before it enters the combustion chamber where the fuel is added. A recuperator has two performance parameters: effectiveness and pressure drop. Higher effectiveness recuperation necessitates large recuperator surface area, resulting in higher pressure drop as well as higher cost (Onovwiona & Ugursal, 2004)

Various recuperator designs exist for a variety of applications. The basic requirement for a recuperator is to separate the fluids to prevent mixing and to achieve the maximum heat transfer between the two fluids. To achieve low cost when manufacturing recuperators, the basic requirements to adhere to is material wastage, automated manufacturing and maximum effective heat transfer area in the Unit. Due to the fact that the primary surface heat exchanger Units are more cost effective (McDonald (2000), a more detailed discussion will follow.).

The simple gas flow paths through the primary surface recuperator are shown in Figure 2.2.3.1. A unique flow path results from the construction since the ends of the folded matrix are sealed. The gas and air streams enter and leave from opposite sides of the core and thus forms a counter flow arrangement. The formed herringbone corrugation in the counter flow section of the recuperator has a sinusoidal curve form. The flow geometry is shown in Figure 2.2.3.2.

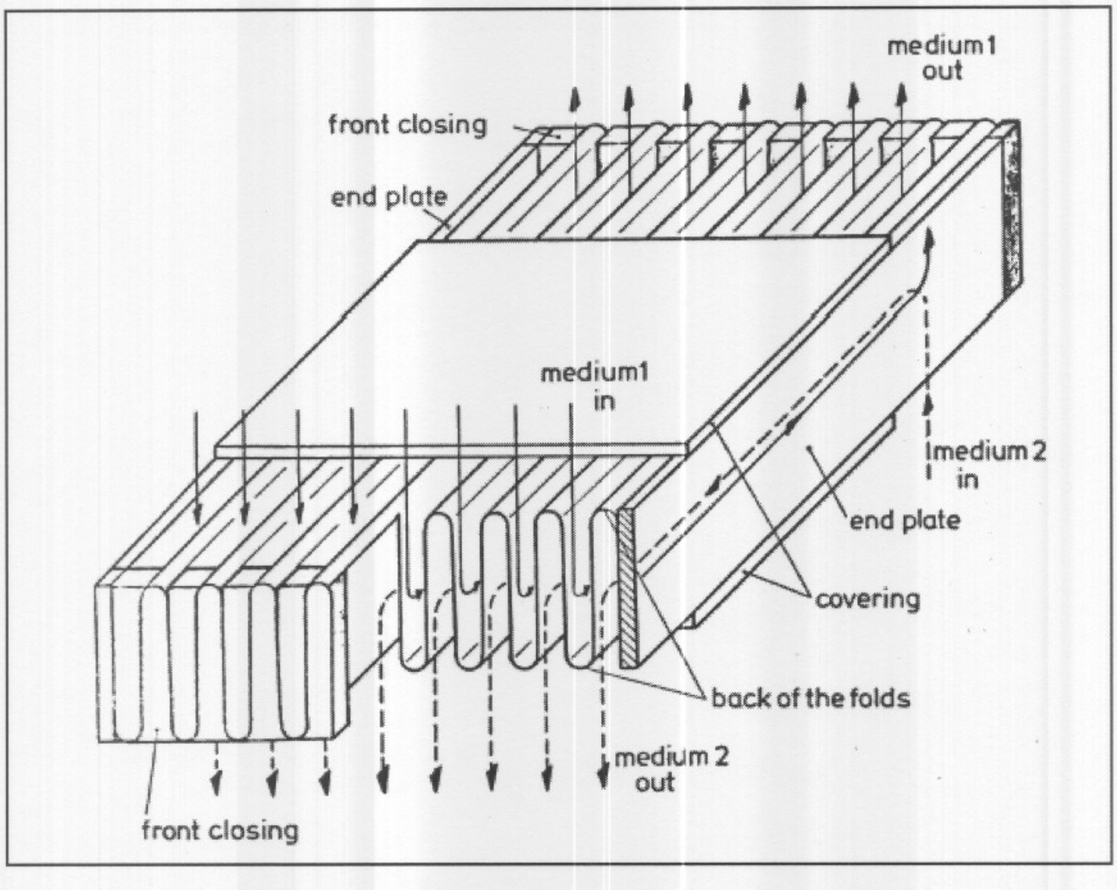

Figure 2.2.3.1: Flow configuration in primary surface recuperator module. (McDonald, 2000)

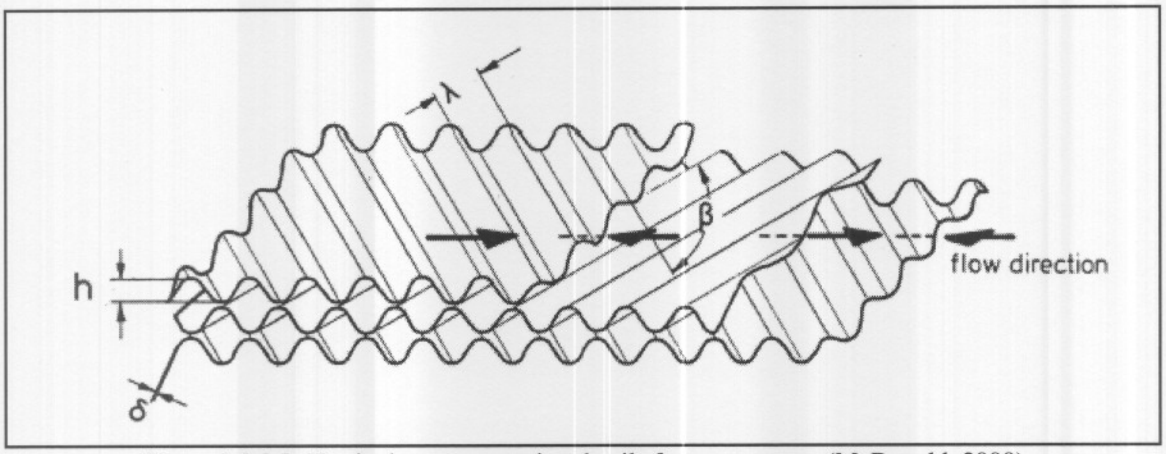

Figure 2.2.3.2: Herringbone corrugation details for recuperator. (McDonald, 2000)

According to McDonald (2000) rig testing of the prototype module revealed turbulent flow at even low Reynolds numbers and good thermal-hydraulic characteristics were obtained. Due to the fact that the primary surface recuperator concept described earlier and researched by McDonald (2000) is manufactured of plate material, it can be easily utilized in a variety of geometries. Figure 2.2.3.3 shows the core of the annular primary surface recuperator that is used in the micro-turbines produced by Capstone Turbine Corp. Figure 2.2.3.4 shows an example of the core section that will typically be used in a rectangular recuperator where the core and the rotating elements (compressor and turbine) will be connected with ducting.

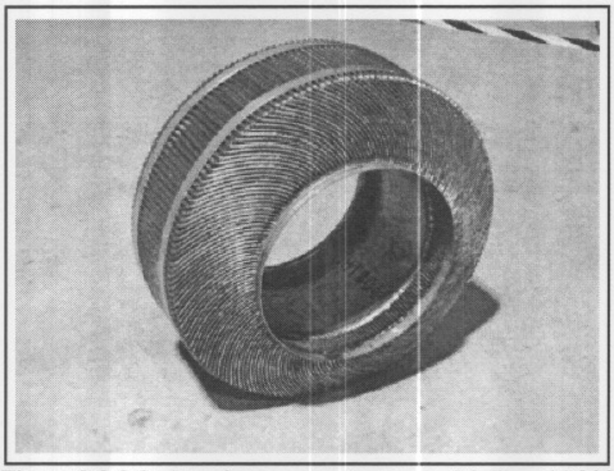
Figure 2.2.3.3: Annular recuperator core. (McDonald 2000)

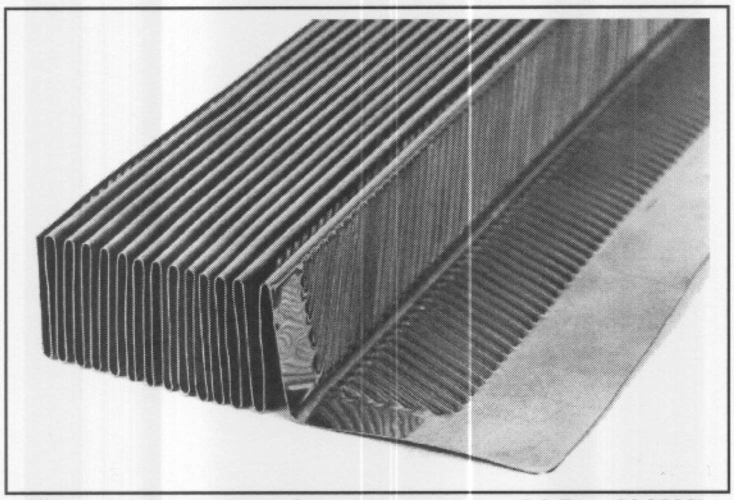

Figure 2.2.3.4: Rectangular recuperator core. (McDonald 2000)

Due to the constraint of manufacturing costs, only metallic recuperators are considered although ceramic recuperators will permit higher inlet temperatures. Most of the modern compact heat exchangers are manufactured using 300-series stainless steels. Type 347 austenitic stainless steel is mostly used for recuperators because of its suitable material properties and relative low cost (McDonald, 2003; Lara-Curzio, 2002). According to McDonald (2003) these heat exchangers have a life expectancy of 40 000h and a temperature limit of 675°C. For higher service temperatures alloys with higher nickel content like Inconel 625 must be used. It is clearly shown by McDonald (2003) that the recuperator cannot be treated as an isolated component and must be included in the overall parametric evaluation of the engine. It is concluded from McDonald (2003) that an average efficiency of a primary surface recuperator manufactured from 347 stainless steel sheet is in the region of 85%.

The primary surface plate recuperator is a fairly low cost solution to increase the efficiency of a low cost gas-turbine.

2.2.4 Heat Source

After the flow passed through the compressor, diffuser and recuperator which were discussed in the previous sections, energy needs to be added to the fluid in order for it to be able to do useful work by expanding it over a turbine.

The heat can be added with a heat exchanger of some sort, or it can be added directly with a combustion process. Combustion is a process where a fossil or bio-fuel is burnt to release the high energy value of the fuel in the form of heat. In gas-turbines the fuel is continuously combusted at an elevated pressure and the energy is used in the form of a high temperature, high energy flow of combusted gas to be expanded over a turbine and therefore do work. (Boyce, 2002)

Background

Essentially combustion chambers can be classified into three major types, the tubular, annular and tubo-annular combustion chambers. There exist three zones in a combustion chamber regardless of its type or design. The first is the recirculation zone where the fuel is evaporated and prepared for combustion in the burning zone. The second zone is the burning zone and at the end of the burning zone all the fuel should be burnt in order for the third zone, which is the dilution zone, to mix the hot combustion gas with the excess air.

The combustion chamber performance is evaluated on the basis of its combustion efficiency, the pressure drop it induces and the temperature uniformity of the outlet gas stream. The pressure drop over the combustion chamber affects both the fuel consumption and the power output. An average pressure drop could be taken as 2 – 8 % of the static compressor delivery. This pressure loss could be therefore be interpreted to be the same as a decrease in compressor efficiency (2 – 8 %). Due to the pressure loss the fuel consumption will increase and the power output of the machine will decrease. If the outlet gas stream does not have a uniform temperature distribution, local damaging to the turbine blades can occur. This is especially true for units that utilize axial turbines.

There are a few factors that affect the satisfactory operation of the combustor. The flame must be self-sustainable. Combustion must be stable over a wide range of fuel-to-air ratios. The temperature must remain within the limits of the material used and no steep temperature gradients must occur as it can cause the combustor to warp or crack. Carbon deposits can influence the flow through the combustor which can decrease the efficiency and increase the pressure loss. Lastly, the gas exiting from the combustor must abide to the emissions regulations. To achieve these requirements from first principles of fluid dynamics is no mean feat and it is strongly recommended that the combustion chamber design be obtained by computational fluid dynamic analysis.

Basic principle of operation

As mentioned there are three types of combustion chambers. The most simple of them is the tubular combustor. In its simplest form it can be described as a straight-walled duct between the compressor and the turbine where fuel is added and burnt. This simple arrangement is impractical because it induces excessive pressure loss. This is because the fundamental pressure loss from combustion is proportional to the air velocity squared. The high-speed in such an arrangement would not permit sustainable combustion and therefore a baffle need to be added. The baffle will create an area of low velocity flow where the flame can be sustained. The problem in a combustor is therefore to create just enough turbulence for mixing the fuel and burning it without inducing an excessive pressure drop. Various ways exist to create flame stability in the primary zone. These methods are more complicated than simple baffles. One method is to create a strong vortex with swirl vanes around the fuel nozzle as shown in Figure 2.2.4.1. Another method is to admit the air through rings of radial impinging jets as shown in Figure 2.2.4.1. This results in upstream flow and forms a recirculation zone that stabilizes the flame. The combustor must be able to operate over a wide range of fuel-to-air ratios. Thus it is designed to operate well below the blowout velocity.

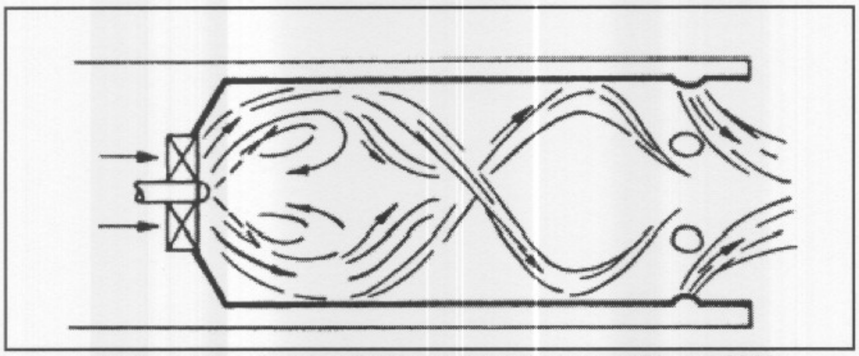

Figure 2.2.4.1: Flame stabilization by swirl vanes. (Boyce, 2002)

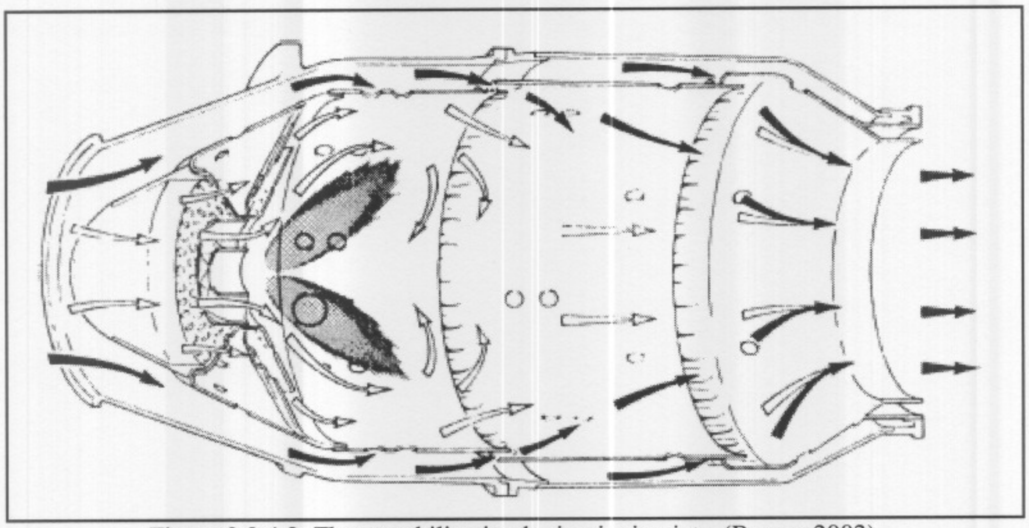

Figure 2.2.4.2: Flame stabilization by impinging jets. (Boyce, 2002)

Combustor fuel injection

In most combustors the fuel is supplied by high-pressure fuel systems that deliver a fine spray of fuel to the primary zone of the combustion chamber where it is then atomized. The spray consists of droplets that have a wide range of diameter. The degree of atomization is expressed in terms of the mean droplet diameter. If the droplets are too small they will not penetrate far enough into the air stream and if it is too big it will not evaporate enough for efficient combustion.

The goal is to produce an approximately stoichiometric mixture of air and fuel that is uniformly distributed across the primary zone, and to sustain it over the range of flow from idling to full-load conditions. The addition of the fuel to the combustor will be discussed in more detail later in this section (Saravanamuttoo *et al.*,2001)

Emissions

Although the combustion equation assumes complete combustion of the carbon to CO_2, incomplete combustion can produce small amounts of carbon monoxide and unburned hydrocarbons. Excess air is also present in the exhaust and therefore the pollutants present in the exhaust will include oxides of nitrogen, carbon monoxide and unburned hydrocarbons. When sulphur is present in the fuel, sulphur oxides will also be present. Three major methods of minimizing emissions are commonly employed: (1) Water or steam injection into the combustor. (2) Selective catalytic reduction (SCR) and (3) dry low NO_x systems. A detail discussion of these methods is beyond the scope of this study and it is recommended that future combustor designers conduct thorough research on the subject of emissions control.

Combustion chamber development for micro turbines

Various research institutes and industrial groups are developing micro gas-turbines for power generation. A lean-burning combustor with fuel film evaporation has been developed by Liedtke and Schulz (2003). It is essentially a reverse flow tubular combustor with an improved method to evaporate the fuel. They intend to use conventional turbocharger components and therefore the cycle parameters are moderate. Figure 2.2.4.3 shows the layout of the combustor. Fuel is injected by the main pressure atomizer with a hollow cone angle of 120° onto the inner surface of the premix tube. The droplets form a fuel film and the swirler containing the main nozzle introduces a vortex to the primary airflow. The centrifugal force stabilizes the film flow and prevents the formation of droplets from the film surface. At the end of the premix tube the fuel-air mixture enters the flame tube. The strong recirculation of the hot exhaust gases in the combustor liner re-ignites the fresh mixture. The cold air inlet prohibits a flame flashback under operating conditions. The evaporation is accomplished by the flow of the hot combustion gas over the outside of the premix tube. Dilution air is then added downstream to obtain the correct exhaust temperature for the inlet to the turbo-charger turbine. The combustion liner consists of three layers including monolithic ceramic forming the inner wall, Insulating fiber (Al_2O_3) or silicone carbide and a metallic outer casing. The liner produces almost adiabatic conditions in the combustor.

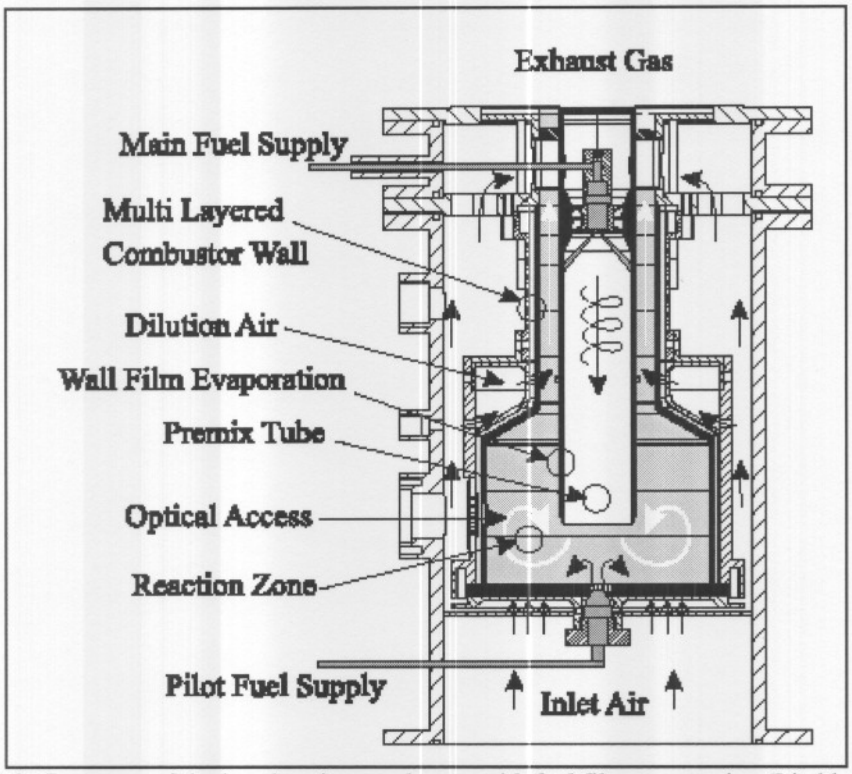

Figure 2.2.4.3: Geometry of the lean burning combustor with fuel film evaporation (Liedtke & Schultz, 2003)

Liedtke and Schulz (2003) concluded that their combustor functioned at various operating conditions. Lean burning conditions in the primary zone of the flame tube were achieved and the potential for reduced emissions were demonstrated. The introduction of silicone carbide as a wall material of the flame tube leads to almost adiabatic combustion at low temperatures and therefore low emissions and complete fuel burn out.

Other types of combustors are used in other micro-turbine systems like the annular reverse flow dry premix combustion systems utilized by Capstone Turbine Corp. shown in Figure 2.2.4.4. The design of the combustion chamber will depend on the mass flow, maximum temperature, fuel to be used and other parameters depending on the specific needs of the design.

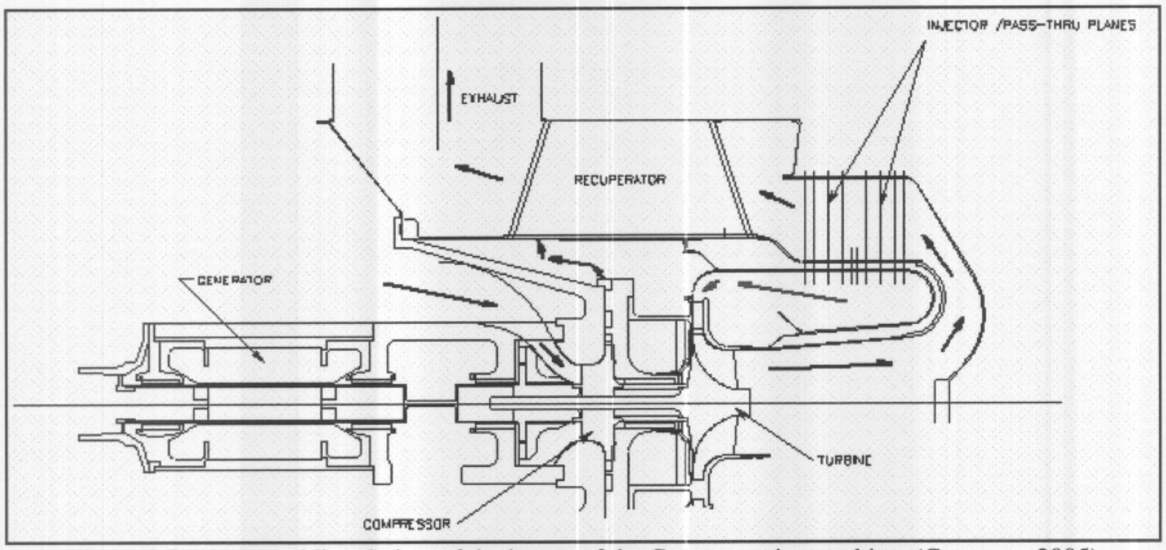

Figure 2.2.4.4: A meridional view of the layout of the Capstone micro-turbine. (Capstone, 2005)

2.2.5 Turbine

The choice of turbine for the use in gas-turbines is limited to a radial turbine and an axial turbine. The radial turbine is similar to the centrifugal compressor in appearance and the axial turbine is similar to the axial compressor. The choice of which type of turbine to use, depends on the specific requirements. In small gas-turbines (25kW and less), radial inflow turbines are usually used. In gas-turbines in the range of 30kW to 300kW, radial inflow turbines can still be used, but the inertia of the large hubs required in this range poses a host of problems to be solved. Therefore the normal practice is to use an axial flow turbine in larger gas-turbines (Wilson, & Korakiantis, 1998).

The axial turbine

An axial turbine stage normally includes a row of stator blades that is followed by a row of moving rotor blades. The gas is expanded and swirled round the axis of the turbine in the stator row and it leaves the stator with a high velocity. The flow then enters the rotor blades and is expanded further. This causes the blades to rotate about the turbine axis and generates work. Depending on the application, one single stage can be insufficient to generate the required power and more than one stage must be used. In gas-turbine engines the separate stages can be mounted on separate shafts to drive different compressors or fans and it may be termed high pressure, intermediate- or low pressure turbines. The Euler turbo machine equation links the changes in flow velocity with the work output. Thus, a particle of fluid moving through a turbine blade row will have components of velocity V_x in the axial direction, V_r in the radial direction, and V_θ in the tangential direction. The axial and radial components do not contribute to the energy transfer in the turbine. They are the components responsible for the mass flow rate, which is equal to the product of the annulus area, the velocity normal to the area, and the local density. The change in axial velocity across the blade row creates an axial force on the shaft which must be absorbed by a thrust bearing. Figure 2.2.5.1 shows the orientation of the velocity components. The energy transfer is determined by the change in the tangential component of velocity. Newton's Second Law of Motion in the angular frame of reference states that the torque which developed is equal to the rate of change of angular momentum across any blade row. This is obtained with Equation 2.2.5.1.

$$\tau = m(r_2 V_e - r_1 V_i) \qquad (2.2.5.1)$$

The power output is given by equation 2.2.5.2:

$$P = \tau\omega = m(U_e V_e - U_i V_i) \qquad (2.2.5.2)$$

Where ω is the angular speed of the rotor and the blade speed $U=\omega r$.

Equation 2.2.5.3 gives the specific power output and is called the Euler turbo machine equation.

$$W_x = U_e V_e - U_i V_i \qquad (2.2.5.3)$$

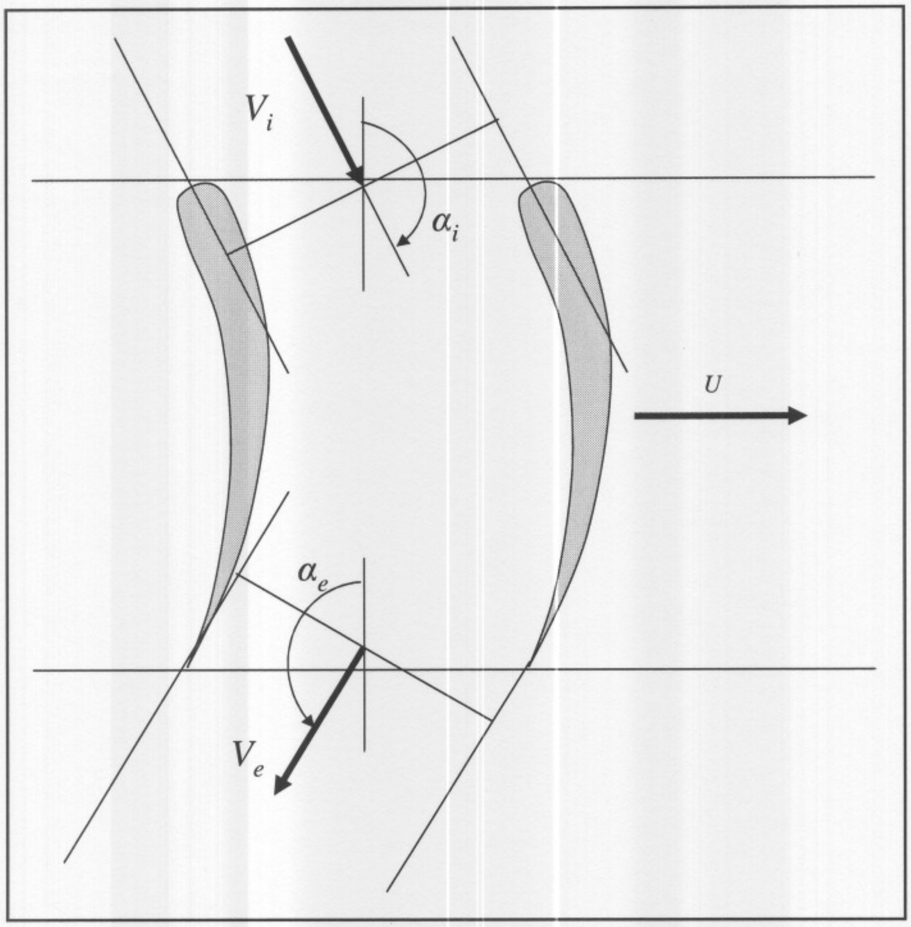

Figure 2.2.5.1: Velocity components of the flow through an axial turbine rotor.

There are many additional issues to be taken in consideration when designing axial turbines. However, for the purpose of this study, the principle of operation given above is sufficient.

The Radial turbines

In a radial flow turbine, gas flow with a high tangential velocity is directed inwards and leaves the rotor with the smallest, but most practical, whirl velocity near the axis of rotation. The result is that the turbine looks very similar to the centrifugal compressor, but with nozzle vanes replacing the diffuser vanes. The nozzle vanes direct the flow onto the rotor with a specific flow angle. This angle is important because it dictates the power

output from the turbine and the efficiency of it. The nozzle vanes can be replaced with a volute that will impart a flow angle on the gas and therefore onto the rotor.
At the exit of the turbine, there normally is a diffuser to reduce the exhaust velocity to a negligible value (Baines & Nicholas, 2003; Saravanamuttoo *et al.*, 2001)

A radial turbine stage can deliver a greater specific power (power per unit mass flow rate of gas) than an equivalent axial stage, thus giving the same power, but with less space needed. The reason for this can be seen from the Euler turbo machinery equation and the velocity triangle; such as illustrated in Figure 2.2.5.2. The former equation is: (Station 4 is the rotor inlet and 6 the rotor exit)

$$P_x = U_4 C_{\theta 4} - U_6 C_{\theta 6} \qquad (2.2.5.5)$$

The geometry of the velocity triangles gives the following relation:

$$P^2 = U^2 + C^2 - 2UC \sin \alpha \qquad (2.2.5.6)$$

Which can be combined with Eq. (2.2.5.5) to give:

$$P_x = 0.5\left[\left(U_4^{\ 2} - U_6^{\ 2}\right) - \left(W_4^{\ 2} - W_6^{\ 2}\right) + \left(C_4^{\ 2} - C_6^{\ 2}\right)\right] \qquad (2.2.5.7)$$

From equation 2.2.5.7 one can clearly see the contribution made to the work output by the change in blade speed $\left(U_4^{\ 2} - U_6^{\ 2}\right)$, and hence the radius, in the radial turbine. U is approximately constant in an axial stage and there is no significant contribution. Several other designs that are necessary in order to achieve a high specific power output P_x, is also shown by this equation. The relative velocity term $\left(W_4^{\ 2} - W_6^{\ 2}\right)$ is subtracted and therefore it must be arranged that $W_6 > W_4$ in order that this term makes a net positive contribution to the work output. The absolute velocity term $\left(C_4^{\ 2} - C_6^{\ 2}\right)$ is then added. In order to maximize the stator exit velocity and hence the rotor inlet velocity C_4, the stator have to be designed to accelerate the flow at inlet. The exit velocity triangle should be arranged in order to minimize the absolute velocity at exit C_6. The exit velocity triangle is shown in Figure 2.2.5.3.

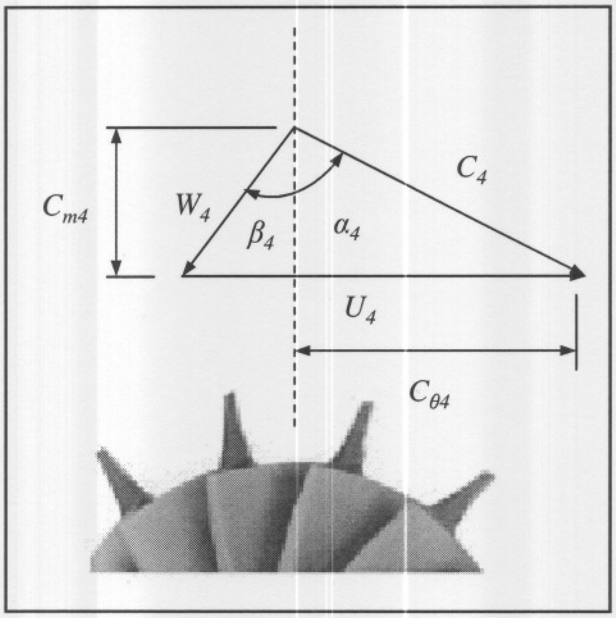

Figure 2.2.5.2: Inlet velocity triangle of the turbine rotor. (Baines, 2003)

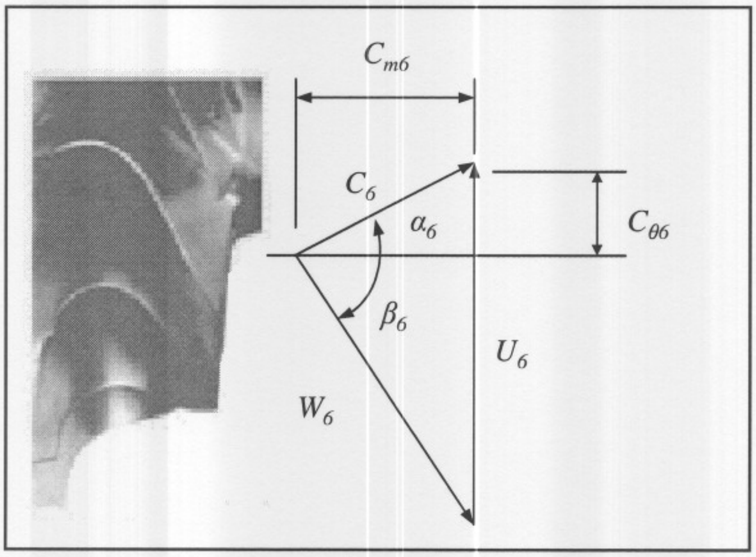

Figure 2.2.5.3: Outlet velocity triangle of the turbine rotor. (Baines, 2003)

Once again this discussion is only introductory and there are more detailed effects that must be considered when a radial turbine is designed. The nature of this study does not focus on the detail design, but rather on general characteristics to enable the author to conclude on the best configuration for this specific application.

The most common application of radial turbines is in automotive turbochargers. These turbines are characterized by performance maps (performance characteristics) as shown in Figure 2.2.5.4. It is set up experimentally to relate the characteristics of the pressure ratio and non dimensional flow through the turbine. The maps are used to predict the performance of the turbine at various operating conditions.

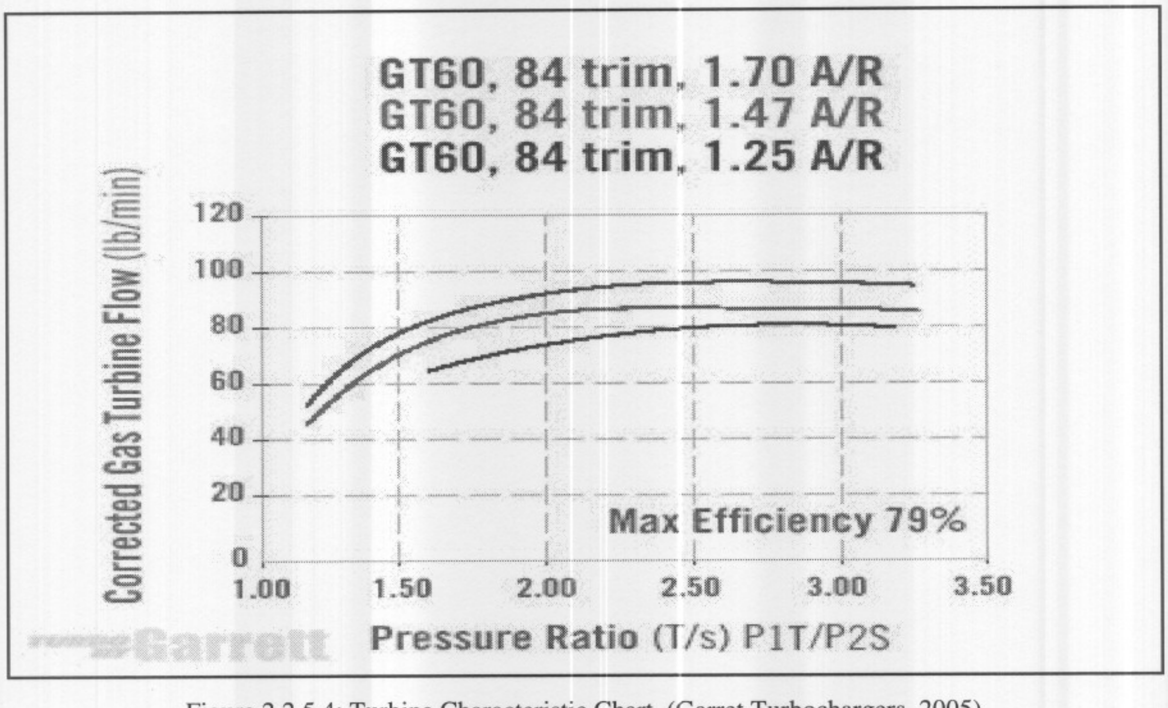

Figure 2.2.5.4: Turbine Characteristic Chart. (Garret Turbochargers, 2005)

It is concluded from the literature that the radial inflow turbine is the preferred choice of turbine for micro-turbine applications. An example is the Capstone C30 micro-turbine which is the market leader in the micro turbine industry. It gives a larger specific power output per stage than an axial turbine and it is more efficient at low mass flow rates. The radial turbine is produced in masses for the turbocharger industry, resulting in relatively cheap cost compared to custom manufactured axial turbines as well as easy availability.

2.2.6 Bearings and seals

The main function of bearings in gas-turbines is to provide support and positioning for the rotating elements of the engine which is usually the compressor and turbine. Rolling element bearings and oil film bearings are the two conventional bearing types used in turbo machinery, while magnetic bearings and foil (or air film) bearings are more recently developed technology.

The conventional types of bearings have generally been associated with lower speed turbo machinery with the exception of turbochargers which generally employ them at rotating speeds in the order of 100 000 min^{-1}. Sufficient lubrication at these speeds is crucial for satisfactory operation.

The majority of the newly developed commercial micro gas-turbines utilize the new technology of magnetic bearings or air film bearings. Turbocharger manufacturers have also indicated that they might try to incorporate the technology in their products. A discussion of these types of bearings is therefore a nothing less than appropriate.

Magnetic bearings

Magnetic bearings are non-contacting bearings which mean that the need for a lubrication system is eliminated and there is negligible friction loss and no wear on the components. The magnetic bearing consists of three basic technologies: Bearings and Sensors, a control system and control algorithms. Magnetic bearings provide electromagnetic suspension of the shaft (rotor) by applying an electric current to ferromagnetic materials used in both the stationary and rotating elements. Thus a flux path is created that includes both the rotating and stationary elements and an air gap separating them. As the gap between the elements decrease, the attractive force between the elements increase. A control system is needed to regulate the current and to stabilize the magnetic forces, thus assuring stable positioning of the rotor. The control process begins with measuring the position of the rotor with some kind of positioning sensor. This measured position is compared with the desired position by the control electronics. The difference in the position is used to calculate the force necessary to pull the rotor back to the desired position. The current of the specific stator section is increased which increases the magnetic flux and thus the attracting force between the rotor and stator. This process is repeated thousands of times per second and thus enables precise control of the rotor at speeds in the excess of 100 000 min^{-1}. Magnetic thrust bearings provide axial support and positioning that is based on the same principle with a rotor disc situated between two stators and an axial sensor that monitor the axial position of the shaft. (SKF, 2005)

Some magnetic bearings are capable of radial forces ranging from 50N on a 9mm shaft, to 25 000 N on a 230mm shaft and thrust bearings that are capable of thrust ranging from 13N on a 9mm diameter disc, to 24500N on a 130 mm diameter disc (SKF, 2005)

Foil bearings

Foil bearings are self-acting fluid film bearings that use air as the working fluid and lubricant. These bearings do not need external pressurization. The fluid film is formed between the rotating shaft surface and a flexible sheet metal foil that is supported by spring foils. These foils are coated with high temperature solid lubricants that allow operation during startup and shutdown when there is not sufficient speed to create the air film and they can operate at temperatures up to 650 ^{o}C.

Three generations of foil bearings have been developed since the 1950's. The Generation I bearings as shown in Figure 2.2.6.1 and Figure 2.2.6.2 were not able to carry large loads and were commercialized in small turbo machinery.

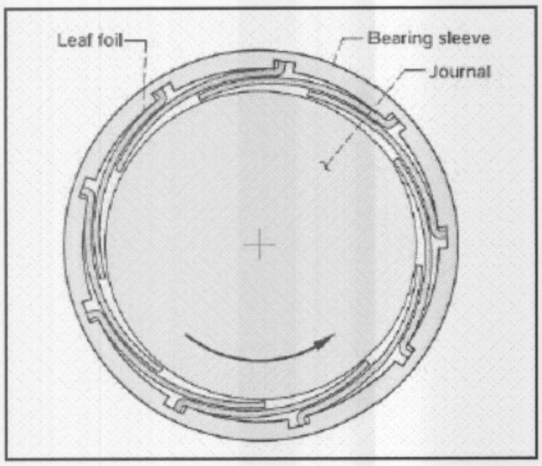

Figure 2.2.6.1: Leaf type foil bearing
(Dellacorte & VALCO, 2003)

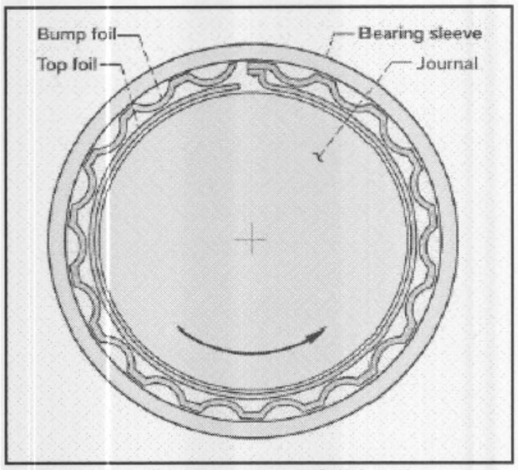

Figure 2.2.6.2: Bump type foil bearing
(Dellacorte & VALCO, 2003)

Generation II bearings emerged in the 1980's and incorporated more complex spring support systems. Figure 2.2.6.3 shows a Generation II bearing that has nearly double the load capacity of Generation I bearings. Although the Generation II bearings were superior to the Generation I bearings, they were still insufficient for use in gas-turbines and many attempts were unsuccessful.

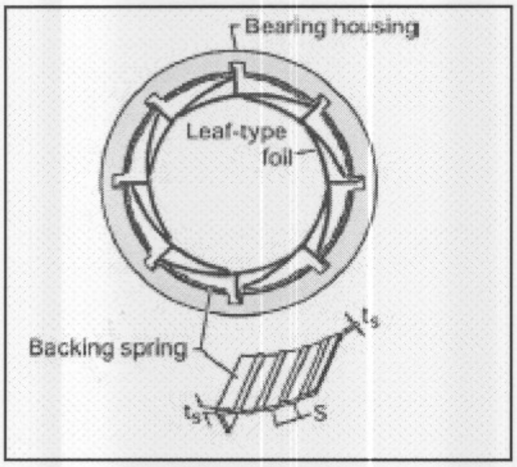

Figure 2.2.6.3: Generation II foil bearing
(Dellacorte & VALCO, 2003)

The Generation III foil bearing was developed in the 1990's. These Generation III bearings have double the load capacity of the Generation II bearings and therefore they are suitable for use in gas-turbines. Figure 2.2.6.4 shows a typical Generation III foil bearing (Dellacorte & VALCO, 2003).

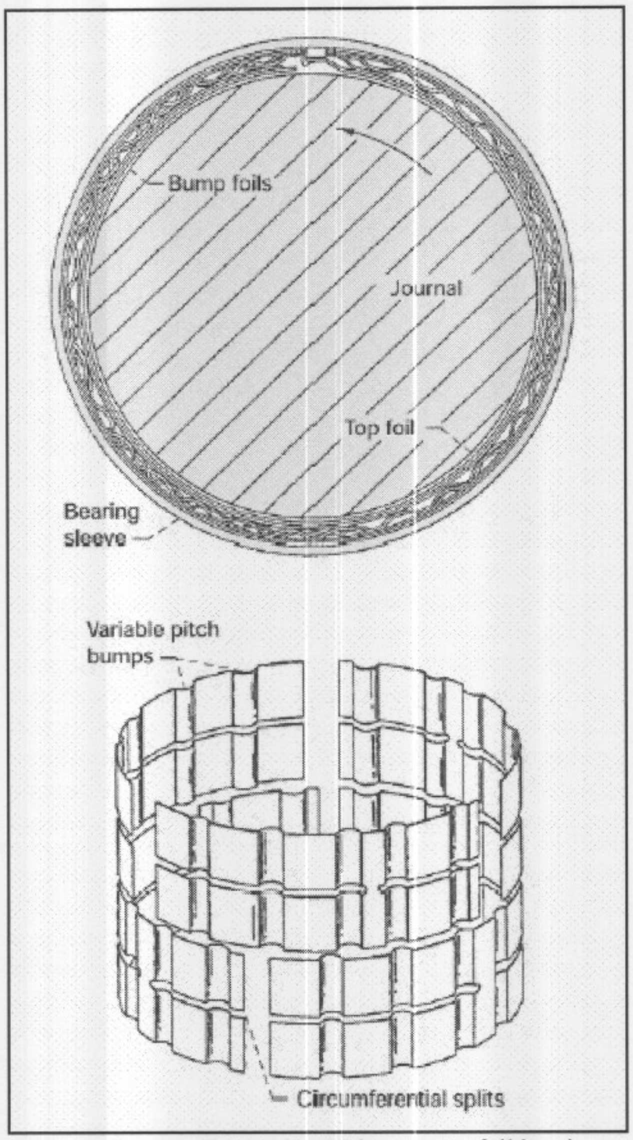

Figure 2.2.6.4: Generation III bump type foil bearing
(Dellacorte & VALCO, 2003)

As mentioned earlier, during the startup or coast down of the rotating element, the surface velocities are too low to generate the air film needed for frictionless rotation and solid lubricants are needed to prevent wear on the rotating element. PS 304 is a plasma spray deposited high temperature coating specifically developed by NASA for foil bearings. It is a composite made from a nickel-chromium binder and chromium oxide hardener with silver, barium fluoride eutectic as solid lubricants. Various other lubricants have been developed by other institutes for commercial use in their products. One such company is Capstone Turbine Corp. that introduced the world's first commercial gas-turbine to use a oil free foil bearing. The Capstone micro-turbine shown in Figure 2.2.6.5 is a 30kW Unit that employs patented foil air bearings (Dellacorte & VALCO, 2003)

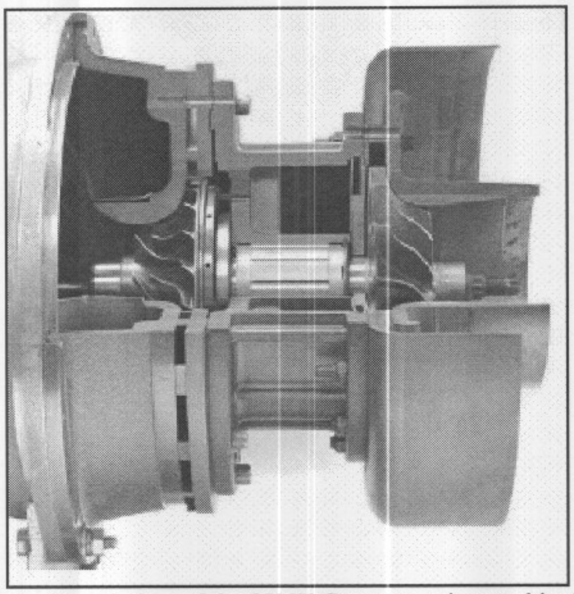

Figure 2.2.6.5: Sectioned view of the 30kW Capstone micro turbine's foil bearing. (Dellacorte & VALCO, 2003)

Foil bearings are still an active field of research and improvements of the solid lubricants and support systems are continually made. Hou *et al.* (2004) presents results of a comparison between two recent developed foil bearings. In depth discussions on al the different foil bearing variations are irrelevant in this study. It can be concluded as sufficient to note that foil bearings seems like the future solution to oil free high-speed turbo machinery.

Seals

The seals used in gas-turbines are an integral part of the design because it can affect the dynamic operating characteristics of the machine. There are two main categories of seals, the first is non-contacting seals and the second is face seals. Non-contacting seals used in high-speed turbo machinery are labyrinth seals and ring seals. A brief discussion thereof should be in order because it is relatively old technology.

The labyrinth seal consists of a series of circumferential strips of metal extending from a shaft or the housing around a shaft. They are usually used where a small loss in seal efficiency can be tolerated. Some advantages of labyrinth seals include simplicity, reliability, adaptability, low shaft power consumption, material flexibility, minimal effect on rotor dynamics and tolerance to thermal variations. There are however disadvantages like high leakage, loss of machine efficiency and tolerance to ingestion of particles that can damage other items.

The leakage of a labyrinth seal can be reduced by minimizing the clearance between seal lands and the seal sleave, providing sharp edges on the lands to reduce the flow discharge coefficient and steps in the flow path to reduce dynamic head over carry from stage to stage.

The wind-back seal is a labyrinth seal but the seal lands form a thread around the shaft and thus it acts as a screw pump.

2.2.7 Fuel supply system and engine control

The fuel system must provide the engine with fuel in a suitable form for combustion and control the flow for easy starting, acceleration and stable operation. This is accomplished with a fuel pump that delivers the fuel to the spray nozzles. The spray nozzles inject the fuel into the combustion chamber in the form of an atomized spray. The fuel supply is usually automatically controlled by the engine control system (Rolls-Royce plc., 1996)

2.2.7.1 Fuels

Micro-turbines can operate on a wide variety of fuels, including natural gas, sour gasses, low-Btu gases like landfill gas and digester gas, bio-fuels and liquid fuels such as gasoline, kerosene and diesel. The quality and composition of fuel burned in a gas-turbine impact the life of the gas-turbine, particularly the combustion chamber and turbine section. The type of fuel that is used determines the design of the fuel injectors. The fuel injectors must be designed according to the combustion chamber in order for the combustion process to be satisfactory. If multiple fuels are used, the combustor, fuel injectors and control system must be designed to operate sufficiently with all types of fuel (Kurz, 2005)

2.2.7.2 Ignition system

Modern gas-turbine engines are all started with some kind of spark device. The more common form of igniter is a high-energy (4 – 20 Joules) surface-discharge spark plug. This is usually placed near the primary zone of the combustion chamber. Normally two igniters are used to create a flame that spreads to the rest of the combustion chamber. The igniters are energized by the control system just before fuel is added and switched off after stable operating point is reached (Wilson & Korakiantis, 1998).

2.2.7.3 Fuel control system

Fuel controls can be divided into two basic groups: hydro-mechanical and electronic. The hydro-mechanical fuel control consists of the following main components:

- Fuel pump to pressurize the fuel.
- Governors to control the rotational speed.
- Pressure relief valves.
- Manual override control systems.
- Fuel shutoff valve.
- Nozzles to atomize the fuel in the primary zone.

The basic principle of operation of the hydro-mechanical control system is that the engine speed is kept constant by a mechanical governor that increases or decreases the fuel flow in relation to the reaction of the engine to the load. When the load increases the speed of the engine decreases and this in turn causes the governor to react. More fuel is added when the speed decrease and less fuel is added when the speed increases. This method is mostly used on engines that produce thrust, for example in aircraft applications. In applications where the speed must vary with the load, namely when a high-speed generator is driven by a gas-turbine, the mechanical governor will not be able to control the engine. Therefore electronic control systems should be utilized to follow the load required from the engine (U.S. ARMY, 2005)

2.2.7.4 Micro gas-turbine engine control

The performance of the gas-turbine generally degrades with a reduction in power. It is therefore important to enhance the part-load performance of the gas-turbine to improve the fuel efficiency because scenarios may exist where the gas-turbine will run at part load for extended periods. Various methods of part-load control exist for different configurations of gas-turbines but due to the scope of this study only the single shaft engine will be considered.

The methods used for controlling the single shaft engine at part-load (and full-load) conditions include:

- Simple operation (fuel only control)
- Variable mass flow by variable shaft speed (VS)
- Variable mass flow by variable inlet guide vane control (VIGV)

As mentioned earlier, the speed of larger turbines driving electric generators should be kept constant to produce the electricity at a constant frequency. For micro gas-turbines the output frequency can be kept constant by a digital power controller, and therefore the shaft speed can vary according to the load.

According to (Kim & Hwang, 2005) the purpose of controls is to maintain the turbine exit temperature (TET) as high as possible for a maximum recuperation effect. For the variable speed operation it is assumed that the TET can be maintained constant until zero loads. The most distinct feature of the recuperated cycle is the utilization of high temperature exhaust gas. Therefore a rapid drop of the TET reduces the recuperation effect and results in a reduction in efficiency. In order to maintain the design TET, the shaft speed must be reduced. It should be noticed that the variable speed operation leads to greater differences between the TET and the exhaust gas temperature (EGT) at a given power. This means that the heat recovery at the recuperator is greater and thus the efficiency is enhanced up to a certain limit. Control by the VIGV method has a similar effect, but it can be seen from Figure 2.2.7.1 that the VS method exhibits the highest efficiency.

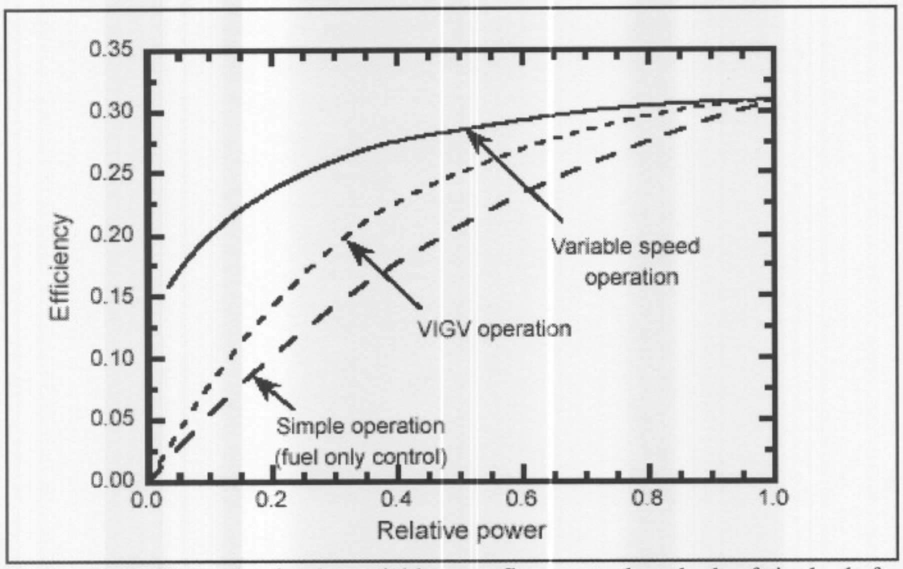

Figure 2.2.7.1:Part load efficiencies for the variable mass flow control methods of single shaft gas-turbine. (Kim & Hwang, 2005)

The VIGV method gives a lower efficiency than the VS method because it experiences a greater reduction in the compressor efficiency due to the effect of the flow angles imparted by the VIGV's. Therefore it can be concluded that the VS method should be the first choice for controlling a single shaft micro gas-turbine. Furthermore, the method depends on the technology of the digital power controller (Kim & Hwang, 2005)

Typically a controller for a gas-turbine will consist of computer-based systems that control the engine with the aid of programmable logic controllers (PLC's) and sensors and actuators or valves. These systems are prone to unexpected failure and backup control methods must be installed and maintained. (Bologna *et al.*, 2002)

2.2.8 High-speed generators

Vaidya and Gregory (2004) comment that there are three different types of electric generators most suited for high-speed operation. These are the permanent magnet, the induction and the switched reluctance generators. In their study they have compared these three types of generators and are of the opinion that the induction generator is the best application in the 100kW to 500kW range. Permanent magnets in machines of such size cause that the magnets are expensive and difficult to retain at high-speeds due to the inertia forces. However for the size of generator applicable in this study, the permanent magnet generator is better suited and will thus be discussed in more detail.

Electrical machines consist of two sets of windings in which one set rotates with respect to the other. The annular space between the stator and the rotor is known as the air gap and is kept very small in order to produce a large magnetic field for a given current. The field winding (stationary windings) produces the flux density and the armature winding (rotating winding) in which the working emf (electro magnetic flux) is induced. Permanent magnets can be used to provide the flux, but it is more expensive than wound field coils around a laminated iron core.

These electrical machines can either be used as motors where the electro-mechanical torque drives the machine against a mechanical torque load, or as generators where the mechanical torque is used to drive an electro-mechanical torque and the generated emf drives the current out of the windings against the terminal voltage.

The stator is usually made from laminated low-loss electrical steel and the winding made from Litz-wire. The rotor consists of a magnetic steel body with surface mounted permanent magnets. The permanent magnets are made by using high-energy rare earth materials such as Neodymium Iron Boron or Samarium Cobalt. Retention of the permanent magnets on the shaft is provided by high strength metallic or composite containment rings. An important issue to keep in mind with PM synchronous generators is to to prevent overheating of the rotor and the de-magnetization of the magnets. The two main factors that cause the rotor to heat up is air friction in the air gap between the rotor and stator and asynchronous mmf-waves caused by stator current time harmonics. Cooling air must be provided in the air gap and additional cooling of the stator is also important. Figure 2.2.9.2 shows a commercial high-speed permanent magnet machine (Magnetic Power Drive 50) produced and sold by CALNETIX (Aglen, 2000; ANON, 2000; Staunton & Ozpimeci, 2003; Sarma, 1996; S2M, 2005; Takahashi *et al.*, 2005)

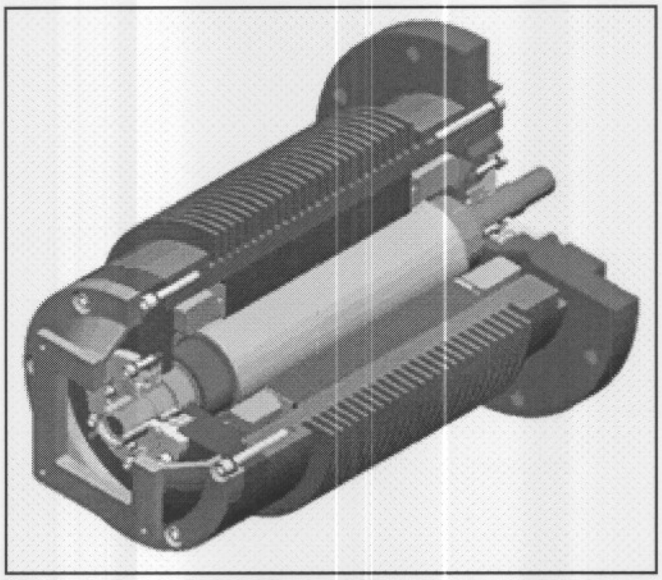

Figure 2.2.8.1: The Magnetic Power Drive 50, a multifunctional machine that can be utilized either as a motor or a generator. (Courtesy CALNETIX)

An advantage of the high-speed generator is that the dimensions of the generator decrease in almost direct proportion to the increase in speed. Therefore a very small Unit can be used with a high-speed micro turbine. The power generated is rectified to Direct Current (DC) and then converted to three phase Alternating Current (AC) before it is connected to the grid. The electrical system can be used in reverse to start the turbine. The control system can use the rectifier as an inverter to accelerate the turbine to the 30 000min^{-1} region from where it becomes self sustainable and can accelerate itself to produce the desired amount of net power (Aglen, 2000; ANON, 2000; Staunton & Ozpimeci, 2003; Sarma, 1996; S2M, 2005; Takahashi *et al.*, 2005)

2.2.9 Commercial status of micro gas-turbines

Existing micro gas-turbines systems range in size from 25kW to 150kW. In Europe various companies are developing micro gas-turbines for various applications but mostly for power generation and smaller units for model aircraft applications. TURBEC, a joint venture between ABB and Volvo was formed in 1998 and develop a 100kW Unit for cogeneration applications.

Micro-turbo in France is developing micro gas-turbines in the range from 200 to 350kW. Their main objective is to study and test the critical components to increase the overall efficiency and to reduce the emissions and the cost of the Units.

In the United States of America, three manufacturers entered the market. Capstone has a 30kW product, Elliott has 45kW and 80kW units and Northern Research Engineering Company will develop several units in the 30kW to 250kW size range. Other companies like Allison Engine Company, Williams International and Teledyne Continental Motors are also interested in developing micro-turbines. The potential market for micro-turbines could increase substantially if the cost, efficiency, durability, reliability and emissions of existing Units are improved (Pilavachi, 2002)

2.3 Turbochargers

Turbochargers are commonly used to improve the performance of internal combustion engines by increasing the intake pressure to the combustion chambers of the engine (Baines, 2004). It ranges in size of application from automotive engines to huge internal combustion engines used for marine applications.

Figure 2.2.10.1 shows a sectioned view of a typical turbocharger used in the automotive industry. It is connected via flanges to the engine and can be easily added or removed if necessary.

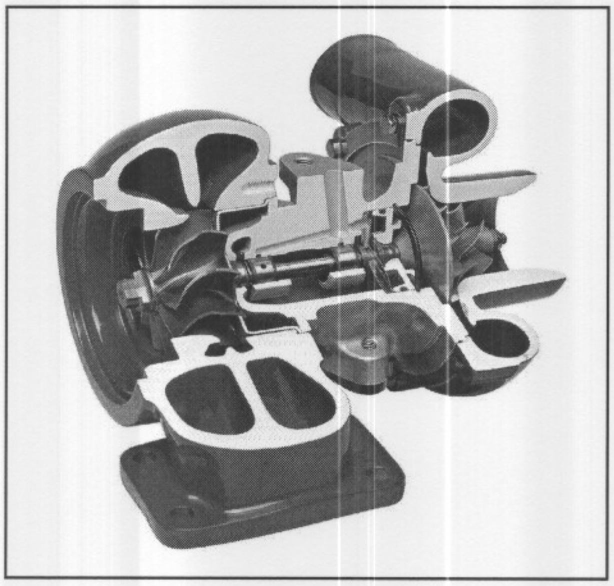

Figure 2.2.10.1: Sectioned view of typical automotive turbocharger. (Author unknown)

A turbocharger consists of a compressor section, a turbine section and bearings, to support the rotor connecting the compressor and turbine. Commonly turbochargers utilize centrifugal compressors and radial turbines because of the associated low mass flow of internal combustion engines. The compactness of these types of compressor end turbine is an added benefit and by being able to manufacture it with single castings, it is relatively cheap compared to their axial counterparts.

The turbine of the turbocharger unit is driven by the hot exhaust gas emitted from the engine. It is commonly known that 30% - 40% of the energy produced from the combustion process is blown down the exhaust. The turbine which is coupled to the compressor by a single shaft, drives the compressor. The compressor outlet is connected to the inlet manifold of the engine and thus increases the pressure to the engine. Therefore, the engine could be seen as non-ideal combustion chamber that could be compared to combustion chambers of gas-turbine engines.
Turbochargers are engineered and manufactured to the standards set by the internal combustion engines. Typical maximum operating temperatures for diesel engines are in the range of 1100K and can rise to 1300K for gasoline engines. The turbocharger is specifically designed for a specific engine size which relates to specific operating limitations (Baines, 2004)

Bearing technology associated with turbochargers are reasonably reliable, as turbochargers are relatively former technology. Traditionally, oil film bearings were employed and many modern turbochargers still utilize this technology. Rolling element bearings are currently emerging in the market and Garret Turbo, to name just one, offer several of their products with either oil film bearings or rolling element bearings.

Performance characteristics (maps) that characterize the performance of the compressor and turbines of a turbocharger Unit are obtainable from the manufacturer and used to predict the performance of the components at various operating conditions.

The rotor assemblies of turbochargers are designed to operate stable at a speed ranging from a few thousand revolutions per minute to a speed well in the excess of one hundred thousand revolutions per minute. Any additions to the shaft of an existing turbocharger will therefore have an influence on the vibration characteristics of the rotating element and should be properly investigated to ensure that the rotor will still give satisfactory performance.

2.4 *Conclusion from literature study*

This chapter presented a literature study on the subject of micro-gas-turbines. The various systems and components of micro-gas-turbines were discussed in detail with the aim of explaining the dynamics and characteristics of each. These characteristics vitally influence the choice of component or system for a particular application and design criterion.

One of the aims of this study is to determine to which extent turbochargers can be used in the construction of micro-turbines. The conclusion drawn from the literature study is that most of the components found in a turbocharger can be used in a micro-gas-turbine if it is controlled to operate within the thermal limits of the turbo charger's design.

The centrifugal compressor and radial turbine found in turbochargers would function properly when utilized in a gas-turbine provided that the inlet temperature to the turbine is restricted according to the limits of the turbine material. An internal combustion engine functions as a heat source similar to a combustion chamber found in gas-turbines while the control and lubrication of the turbocharger could be easily supplied with electronic components and components used in the automotive industry. The turbocharger could be integrated with a recuperator necessary to increase the performance and the turbine section thereof can be mated to a combustion chamber. Therefore even though a turbo charger may be less efficient than a custom designed micro gas-turbine, it is less expensive to use the turbocharger than to manufacture custom designed components.

The next chapter examines the conceptual layout of a micro gas-turbine based on using a standard turbocharger. It discusses the system requirements cycle layout as well as the proposed components of the micro gas-turbine.

3. System thermodynamic requirements

In order to be able to assess the thermodynamic suitability of a turbocharger for use in a micro gas-turbine, the thermodynamic performance of a turbocharger must be evaluated against certain requirements.

3.1 System requirements

The conclusion made from the literature on existing micro turbines is that the power output can vary between 25 and 150 kW. Typical values of their efficiency are in the range of 15% for simple cycles and 30% for recuperated cycles. Combined cycles yield higher efficiencies but for the purpose of this study the recuperated cycle will be used.

The most basic thermodynamic requirements are laid down to provide some guideline to evaluate the suitability of a turbocharger for use in a micro gas-turbine:

- A turbocharger unit must be able to generate power in the range of 20kW to 30kW. This will allow it to compete in the market with similar products like the Capstone C3 micro-turbine which generates 30kW.
- When a turbocharger is used in a micro gas-turbine, the system must have efficiencies higher than 20% to be considered a viable option apposed to diesel internal combustion engines.
- The range where the turbocharger unit produces excess power must be adequate to allow for reasonable load following.
- The turbocharger must be integrated with the recuperator and the combustion chamber.
- The operating temperature of the turbocharger must be within the working limits while producing desired levels of excess power.

3.2 *Cycle layout for thermodynamic evaluation*

A single shaft, recuperated open cycle layout shown in Figure 3.2.1 is used to calculate the thermodynamic performance of a turbocharger. Air is drawn in by a centrifugal compressor which raises the pressure and therefore determines the maximum pressure of the cycle. The air is then passed through a recuperator to increase the temperature of the air before it reaches the combustion chamber where fuel is added and the mixture burned to increase the inlet temperature to the turbine. A radial inflow turbine then reduces the pressure as well as the temperature and extracts power from the hot gas flowing through. This power is used to drive the compressor and external loads. The hot exhaust gas is passed through the recuperator before it exits the cycle.

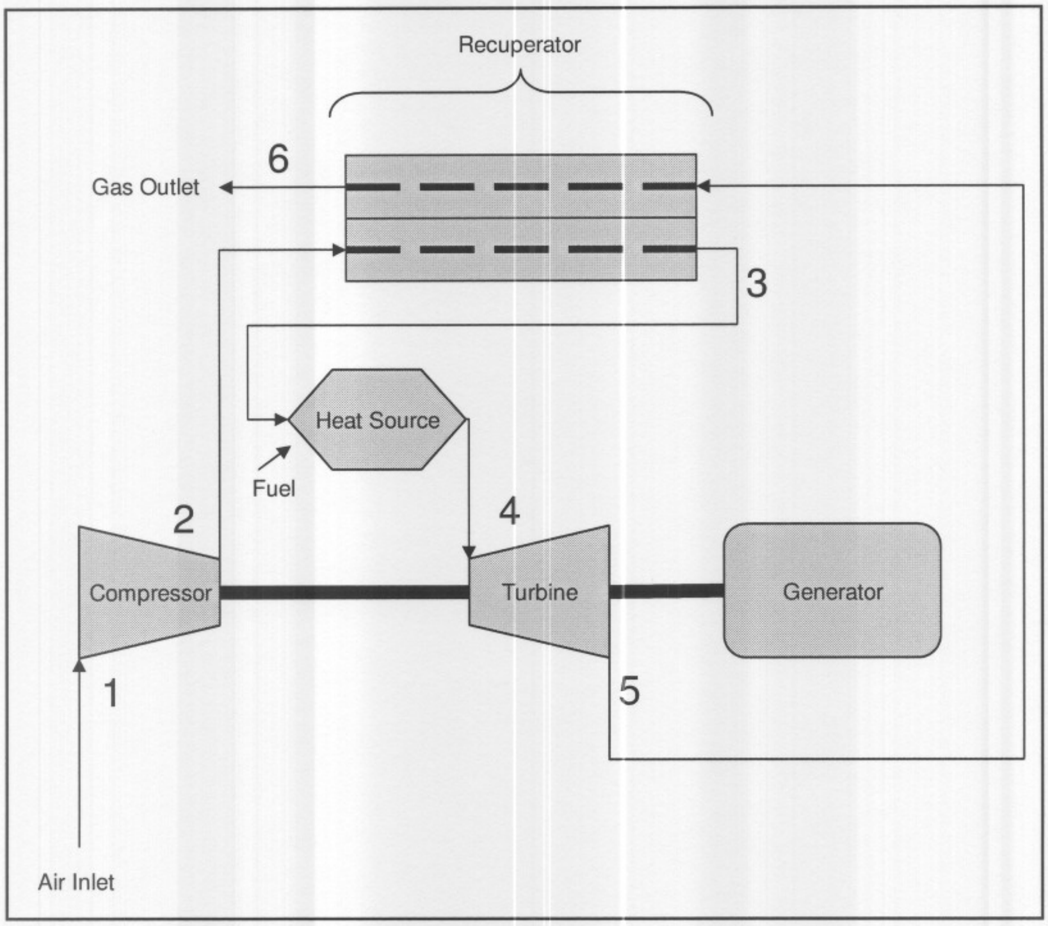

Figure 3.2.1: Single shaft recuperated open cycle used to calculate the thermodynamic performance of an automotive turbocharger when used as a gas turbine.

4. Thermodynamic performance of a turbocharger

In order to conclude on the thermodynamic suitability of a turbocharger for use in a micro gas-turbine, calculations is needed to compare the performance of a turbocharger against the thermodynamic system requirements specified in Chapter 3.

4.1 Calculating available excess power

4.1.1 Obtaining matched operating points on a compressor chart

This section explains the procedure to find match-points between a certain compressor and a certain turbine when connected by means of a single shaft. The procedure is based on the methods given by Saravanamuttoo *et al.* (2001). The method described by Saravanamuttoo *et al.* (2001) assumes that full expansion takes place over the turbine for each point on the NDS (Non-Dimensional Speed) line and the turbine inlet temperature is then calculated for that point to satisfy the compatibility equations.

The method used in this particular matching procedure varies from the method described in Saravanamuttoo *et al.* (2001), in that it is carried out by using a fixed turbine inlet temperature resulting from material limitations of the turbine. Therefore the maximum power output obtainable on each constant NDS (Non-Dimensional Speed) line is calculated by following an iterative procedure that calculates the equilibrium running point for the specific inlet temperature. The procedure is explained in detail in Appendix A. Figure 4.1.1.1 shows schematically how the procedure is carried out.

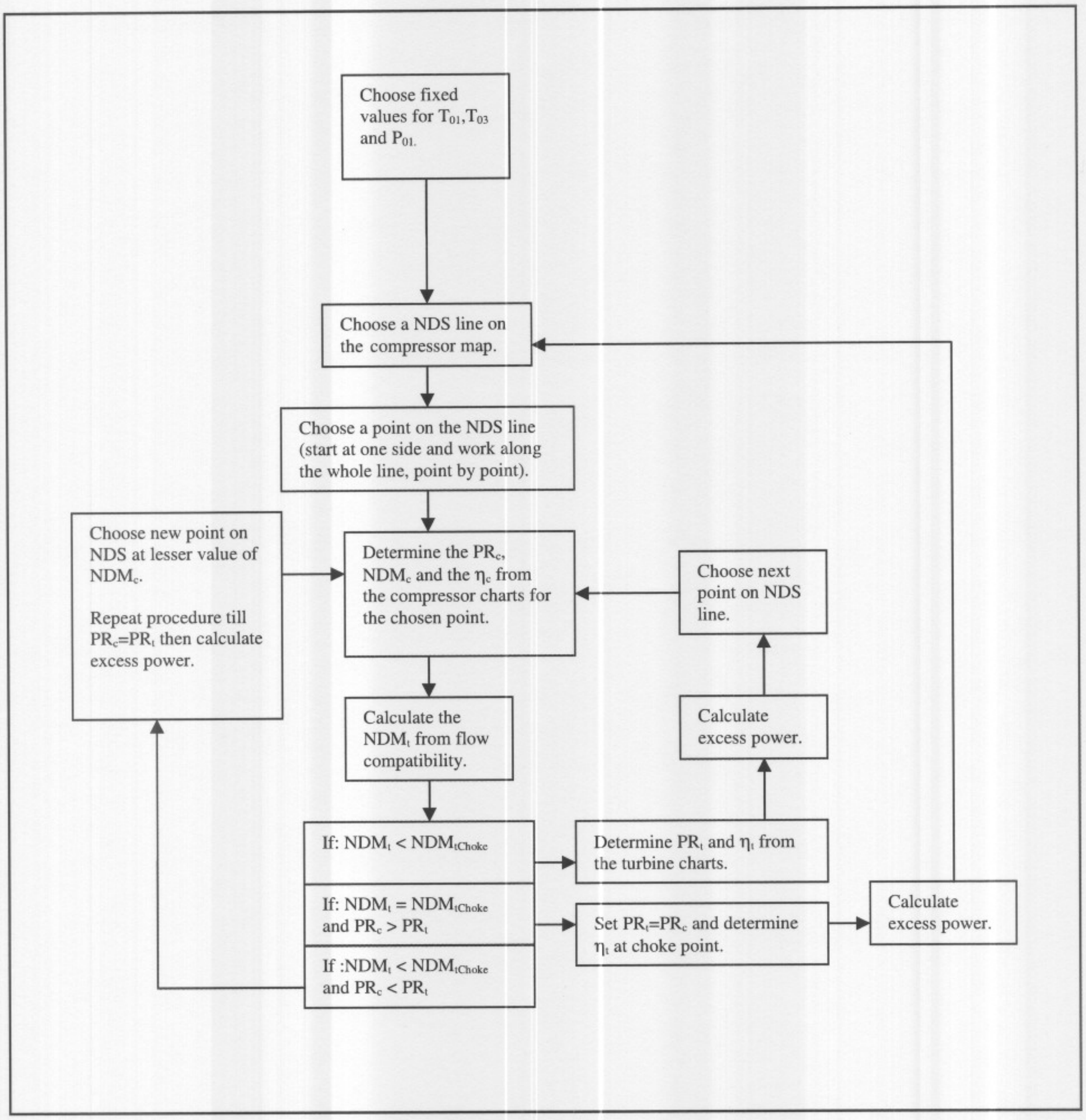

Figure 4.1.1.1: Matching procedure of a single shaft gas-turbine with a fixed turbine inlet temperature to obtain the maximum power output.

4.1.2 Excess power produced by a turbocharger

The goal of this exercise was to verify the matching procedure of certain compressor and turbine pairs. Matching was originally done by using the principal laws of flow compatibility and work compatibility. Furthermore it was used to utilize the characteristics (maps) of the components to obtain the values of the pressure ratios and efficiencies at certain non-dimensional mass flow points on specific NDS (non-dimensional speed) lines. This procedure was carried out graphically and by hand calculation as shown in Appendix A. The calculations were then done in an Excel spreadsheet to assist with the repetitive nature of the procedure, necessary to obtain the match point on each NDS line.

The maps (compressor and turbine characteristics) were then imported into FLOWNEX and a similar simple open cycle was set up. The matching was automatically done by FLOWNEX for each constant NDS line used in the Excel calculations.

The inlet conditions to the compressor and the inlet temperature to the turbine was held at constant values for both the Excel spreadsheet and the FLOWNEX simulation. The inlet temperature to the compressor was taken as T_{01}=298K. The inlet pressure to the compressor was assumed to be p_{01}=100kPa and the inlet temperature to the turbine was held at T_{03}=900K resulting from the limitations of the turbine material. No pressure losses between the compressor and turbine were brought into consideration.

The units used for matching were all turbocharger units used on commercial internal combustion engines. The compressor characteristics and turbine characteristics of these turbochargers were made available to the North West University under special agreement from the manufacturer. The make and model of the turbochargers may therefore not be disclosed.

The excess power produced with the inlet conditions stated above was calculated for each method and compared. Figures 4.2.1.1 to 4.2.1.4 show the results of the excess power plotted against NDS for the four turbocharger units compared. Unit 1 produced excess power of 16kW while rotating at 90000 min^{-1}. Unit 2 produced 17kW at 70000 min^{-1}. Unit 3 produced 23kW at 65000 min^{-1} and Unit 4 produced 36kW at 62000 min^{-1}.

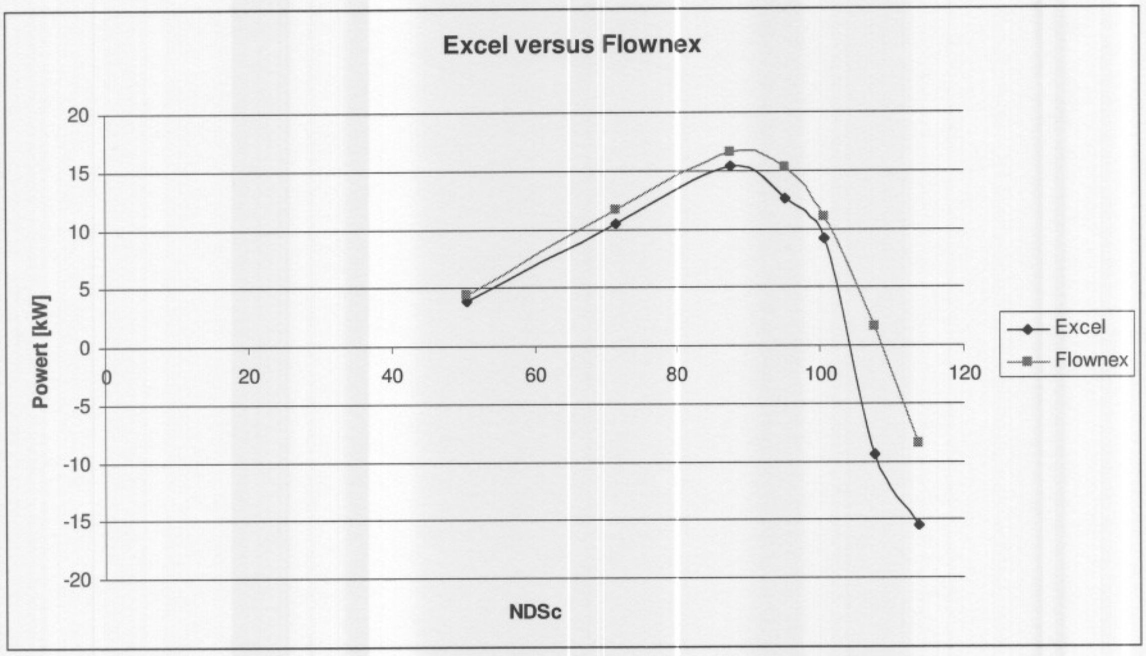

Figure 4.2.1.1: Unit1 Excess Power versus Non-Dimensional Speed.

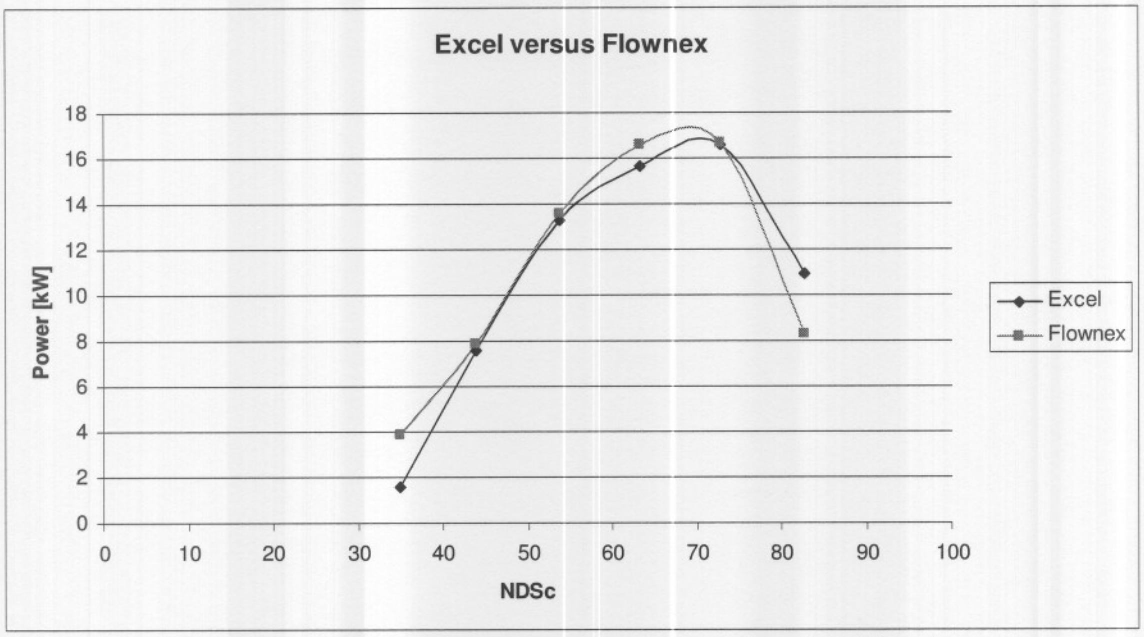

Figure 4.2.1.2: Unit 2 Excess Power versus Non-Dimensional Speed.

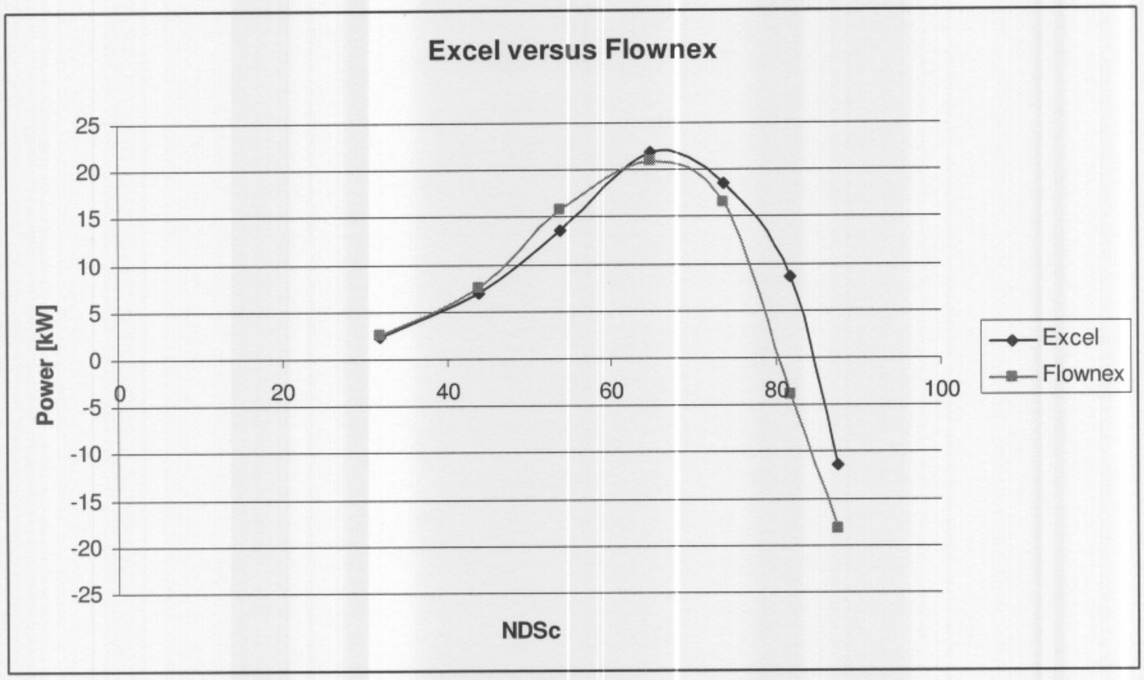

Figure 4.2.1.3: Unit 3 Excess Power versus Non-Dimensional Speed.

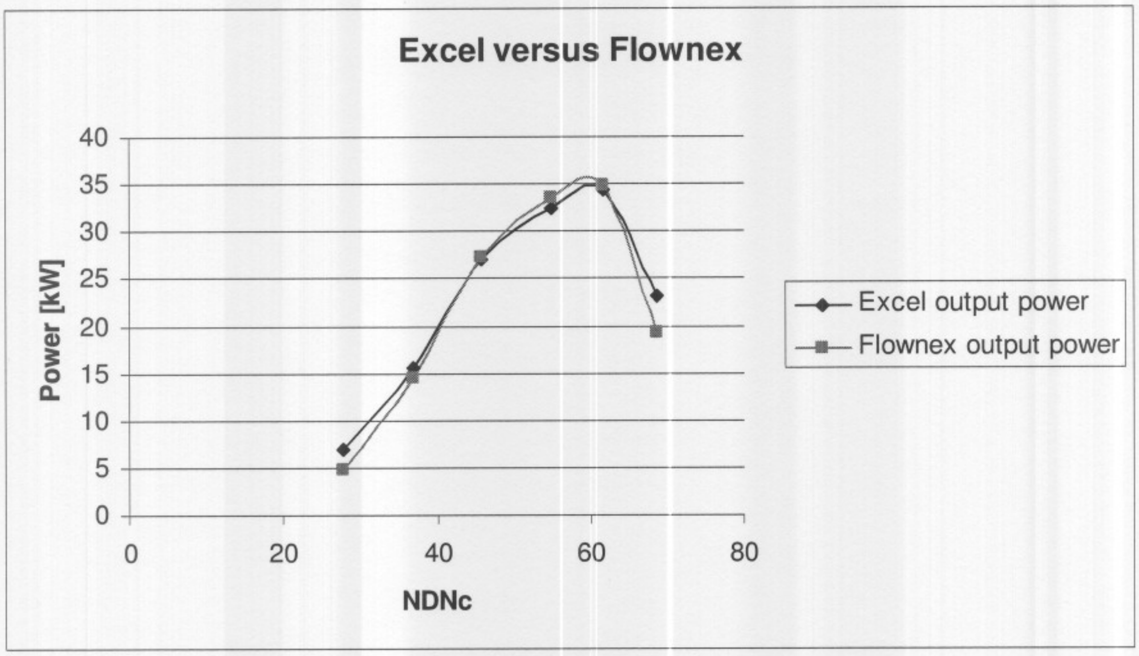

Figure 4.2.1.4: Unit 4 Excess Power versus Non-Dimensional Speed.

The trend of the excess power is similar for all four turbocharger units under consideration. The excess power increases with an increase in non-dimensional speed. This only occurs up to a certain non-dimensional speed where the optimum pressure ratio is reached. Thereafter it decreases. The increase in non-dimensional speed causes an increase in non-dimensional mass flow. The compressor work increases due to a reduction in efficiency and the higher mass flow and pressure ratio. The inlet temperature to the turbine is kept constant and thus the excess power reduces.

Unit 4 has the highest excess power output equal to 36kW. The excess power calculated for Unit 4 compares well with the requirements of Chapter 3 and is used in the cycle simulation.

4.2 Recuperated open cycle simulation

After calculating the excess power available from a turbocharger, the thermodynamic performance of a turbocharger integrated in a recuperated open cycle needs to be calculated. This is done to assess the system performance. This section uses the thermodynamic parameters obtained in section 4.1 to calculate the cycle performance when a turbocharger is used.

4.2.1 Assumptions used in the simulation

To simulate the cycle consisting of the compressor, turbine, heat source (combustion chamber) and recuperator, basic thermodynamics, fluid-dynamics and the turbo machinery equations are used. The maximum inlet temperature to the turbine is fixed by the limit of the turbine materials. For this cycle simulation a maximum value of 900K is used. The efficiency of the recuperator was conservatively taken as 85% and the efficiency of the combustion chamber was taken as 95% due to the fact that both items must still be developed. Pressure loss through the recuperator and pipes that connect components is ignored but a 2% pressure loss was assumed over the combustion chamber. The corresponding pressure ratios obtained from the matching procedure described previously is used to obtain the power output and the efficiency from the cycle. Therefore the principle of operation will be to operate at variable speed at the points of maximum efficiency with variation in load. The formatted equations of the EES (Engineering Equation Solver) simulation program are included in Appendix A together with a discussion of the results.

4.2.2 Applying matched operating conditions in a cycle simulation

Although the simulation is based on assumed values for the combustion pressure loss combustion efficiency and recuperator efficiency, it gives a good approximation to the possibilities of the proposed cycle. This study focuses on the thermodynamic suitability of turbochargers in micro gas-turbines and not on selecting the best make or model. Of the four turbocharger models, Unit 4 is selected as the best model to test in a cycle simulation because the excess power available from it is the closest match to the specifications required.

A certain non-dimensional speed was chosen to calculate the cycle parameters when turbocharger Unit 4 is connected in an open recuperated micro gas-turbine cycle. The pressure ratio and efficiency of the compressor and turbine were read from the compressor and turbine characteristics at the equilibrium operating point of the chosen non dimensional speed. The detailed hand calculations are shown in Appendix A4.1. The set of equations were then programmed in EES (Engineering Equation Solver) and the exact same values used in the hand calculation were used to calculate the cycle parameters. The results obtained from the detailed hand calculations and the EES calculation is compared in table 4.2.2.1.

Table 4.2.2.1: Comparison of thermodynamic cycle parameters as calculated with hand and EES.

Parameter	Hand Calculation	EES calculation
Compressor inlet temperature [K]	298	298
Compressor outlet temperature [K]	385.6	385.6
Compressor pressure ratio	2.06	2.06
Compressor efficiency [%]	78	78
Recuperator high pressure inlet temperature [K]	385.6	385.6
Recuperator high pressure outlet temperature [K]	777.5	777.5
Recuperator low pressure inlet temperature [K]	789.2	789.2
Recuperator low pressure outlet temperature [K]	446.2	446.2
Recuperator efficiency [%]	85	85
Combustion chamber inlet temperature [K]	777.5	777.5
Combustion chamber outlet temperature [K]	900	900
Combustion chamber efficiency [%]	95	95
Turbine inlet temperature [K]	900	900
Turbine outlet temperature [K]	789.2	789.2
Turbine pressure ratio	2.06	2.06
Turbine efficiency [%]	75	75
Compressor power absorbed [kW]	60.7	60.7
Turbine power generated [kW]	87.7	87.7
Excess power available [kW]	27	27
Duty recovered in recuperator [kW]	271.5	271.5
Duty required from heat source [kW]	89.3	89.3
Cycle efficiency [%]	30.2	30.2

The values obtained with the hand calculation validate the solution obtained with EES. It is therefore concluded that the EES calculations for all the other non-dimensional speed lines of turbocharger Unit 4 is correct. Therefore it is a realistic representation of the behavior of the turbocharger when used as a micro gas-turbine.

The results are shown in detail in Appendix A4.2 but Table 4.2.2.1 is a summary of the most important data.

Table 4.2.2.2: Results for the recuperated cycle simulation.

1..6	NDS_c	NDM_c	PR_c	η_c	PR_t	η_t	m_{flow}	η_{cycle}	W_{gen}
Run 1	68.57	20.8	3.59	0.69	3.59	0.75	1.205	0.09585	23.22
Run 2	61.77	18.4	3.16	0.728	3.16	0.75	1.066	0.174	34.46
Run 3	54.9	15.35	2.65	0.74	2.65	0.75	0.8892	0.2233	32.45
Run 4	45.67	11.9	2.06	0.78	2.06	0.75	0.6893	0.302	26.98
Run 5	36.89	8.8	1.62	0.78	1.62	0.75	0.5098	0.3229	15.64
Run 6	27.79	6.2	1.33	0.76	1.33	0.75	0.3592	0.3005	6.998

The excess power output from the cycle is graphically shown in Figure 4.2.2.1 and the system efficiency is shown in Figure 4.2.2.2. The peak power of 34.46kW is realized at a mass flow of 1.066kg/s while the peak efficiency of 32% is realized at a mass flow of 0.5098kg/s.

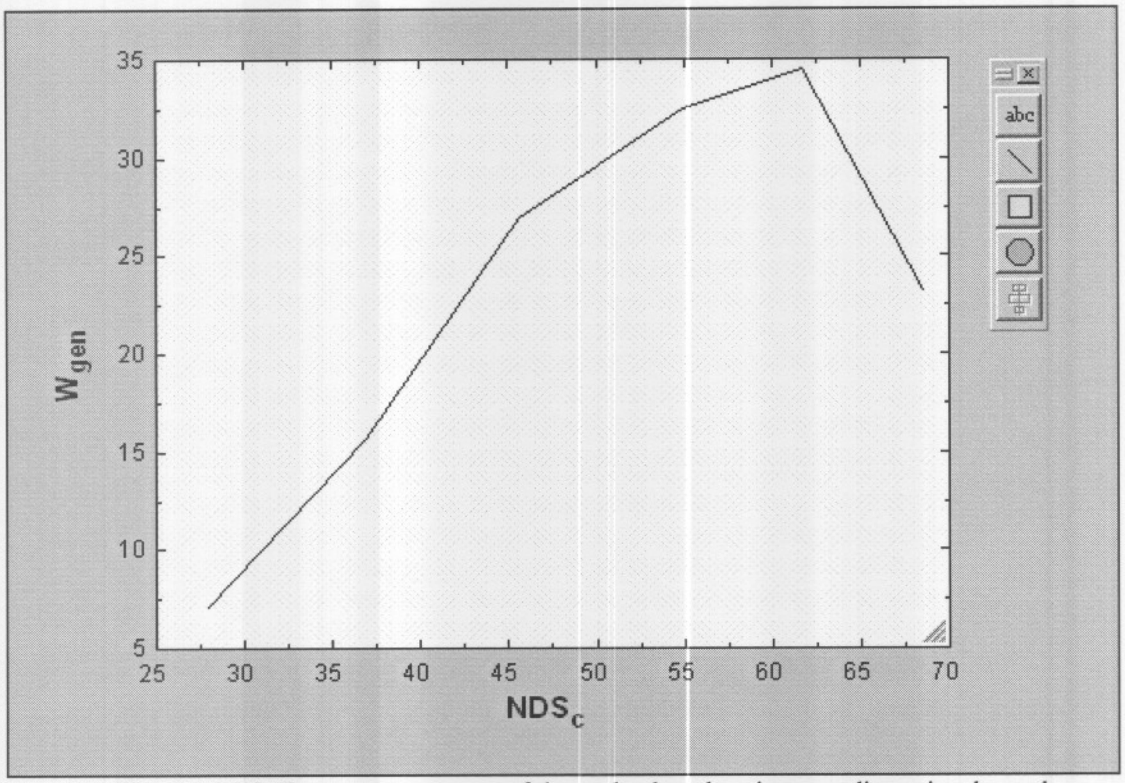

Figure 4.2.2.3: Excess power output of the cycle plotted against non-dimensional speed.

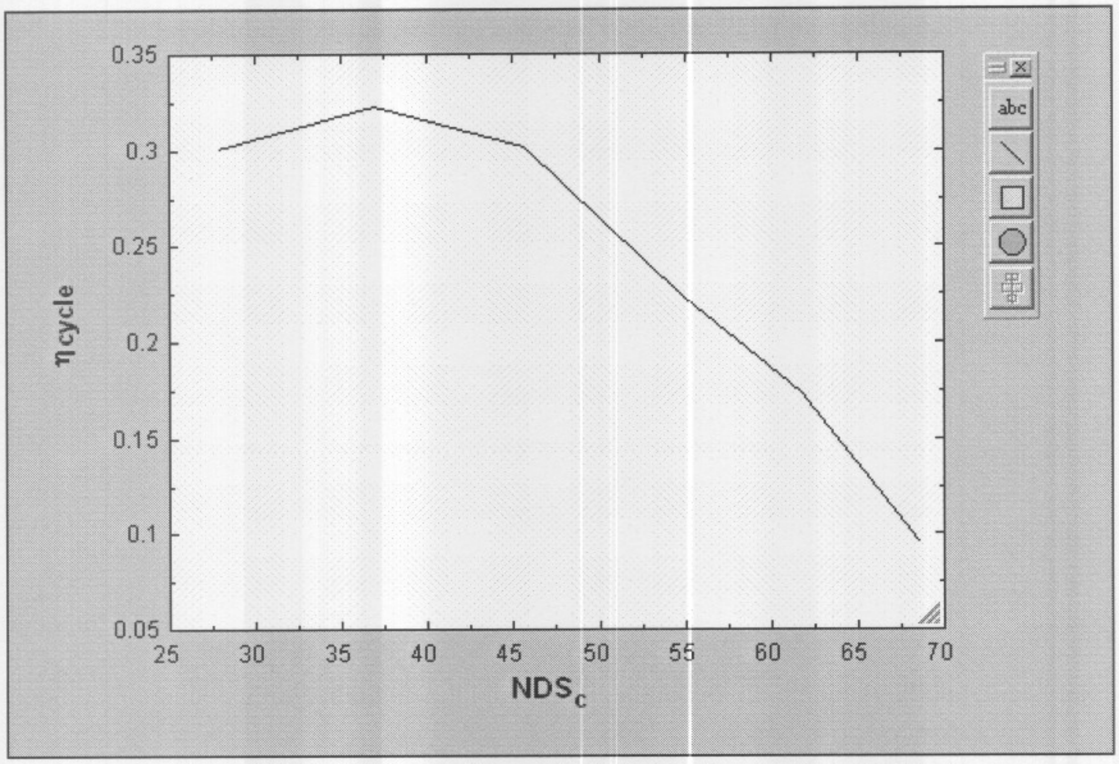

Figure 4.2.2.2: Efficiency of the cycle plotted against non-dimensional speed.

The power output and efficiency compare well to the other commercial micro gas-turbines. The efficiency is however somewhat lower compared to other micro turbines when a recuperator efficiency of 85% is assumed. If the recuperator efficiency is increased to 90%, the cycle efficiency increases to a maximum of 58% at a mass flow of 0.359kg/s. At this point however the power output is only 6kW but when the cycle efficiency is calculated at a mass flow of 0.689kg/s (the previous highest efficiency with a recuperator efficiency of 70%), it is found to be 37.19%. This is a value highly comparable with the commercial micro turbines of equal size. Therefore the recuperator is a component with a high influence on the overall success of the micro turbine cycle. It is therefore advisable to obtain the highest recuperator efficiency possible.

4.3 *Thermodynamic parameters at typical operating conditions*

The cycle simulation done for turbocharger Unit 4 in EES yielded the parameters as indicated in Table 4.3.1. This first order analysis ultimately has the goal to determine if a turbocharger is thermodynamically suitable for use in a micro gas-turbine when connected to other crucial components in an open recuperate cycle.

Table 4.3.1: Thermodynamic cycle parameters.

Compressor	
Inlet temperature [K]	298
Outlet temperature [K]	385.6
Pressure ratio	2.06
Efficiency [%]	78
Recuperator	
High-pressure inlet temperature [K]	385.6
High-pressure outlet temperature [K]	777.5
Low-pressure inlet temperature [K]	789.2
Low-pressure outlet temperature [K]	446.2
Efficiency [%]	85
Combustion chamber	
Combustion chamber inlet temperature [K]	777.5
Combustion chamber outlet temperature [K]	900
Efficiency [%]	80
Turbine	
Inlet temperature [K]	900
Outlet temperature [K]	789.2
Pressure ratio	2.06
Efficiency [%]	75
Calculated power/duty	
Compressor power [kW]	60.71
Turbine power [kW]	87.69
Excess power [kW]	26.98
Recuperator duty [kW]	271.5
Cycle efficiency [%]	30.2

The compressor and turbine are inherently specified through the matching and selection procedure and the fact that they are integrated in a turbocharger Unit. However, the requirements for the combustion chamber and the recuperator are dictated by the parameters of the cycle simulation. The combustion chamber must be able to constantly aad 89.36kW of heat to the fluid stream. The recuperator is required to give 271.5kW of heat transfer at a mass flow of 0.6893kg/s. The pressure drop should be kept as low as possible and its efficiency should be at least 85%. Any increase in recuperator efficiency will result in an increase in the system efficiency and therefore benefit the design.

4.4 Conclusion

This section discussed the results obtained by the matching procedure in Excel and FLOWNEX. It was illustrated that for a certain turbine inlet temperature there exist certain points on the turbocharger compressor characteristic where the compressor and turbine rotor can operate self sustainable and even produce excess shaft power. The flow conditions where the excess power is generated by the turbocharger were then used in an open recuperated cycle simulation. This simulation was done in EES and a hand calculation was done to verify the EES solution. The solution for turbocharger Unit 4 satisfies the thermodynamic system requirements stated in Chapter 3. The unit produces $27kW$ while having a cycle efficiency of 30.2%. The operating turbine temperature of $900K$ is within safe limits of the turbocharger unit as specified by the manufacturer.

5. Conceptual Mechanical Integration

The conceptual mechanical integration of a turbocharger unit with other equipment to function as a micro gas-turbine is discussed in this section. The issues that will need to be addressed when designing a micro gas-turbine by using a complete turbocharger are identified and conceptual solutions are presented to serve as a take off for a possible design phase.

5.1 Mechanical layout

In order to utilize a standard commercial turbocharger, the other components in the rercuperated open cycle must be designed to fit the turbocharger. A recuperator designed for the specific output determined in the cycle simulation, is connected to the outlet of the compressor and to the inlet of a tubular combustion chamber. The combustion chamber is connected to the inlet of the turbine. The hot gas exiting from the turbine is connected to the recuperator to recover the waste heat. The excess power developed by the micro turbine is recovered by a high-speed generator which converts the shaft power to electric power.

A conceptual representation of how a turbocharger could be integrated with the components mentioned above is illustrated in Figure 5.1.1. This layout illustrates the versatility of connecting the various components to the turbocharger. The dimensions of all the components should be easy to optimize which could improve the overall efficiency of the micro gas-turbine and minimise the production cost.

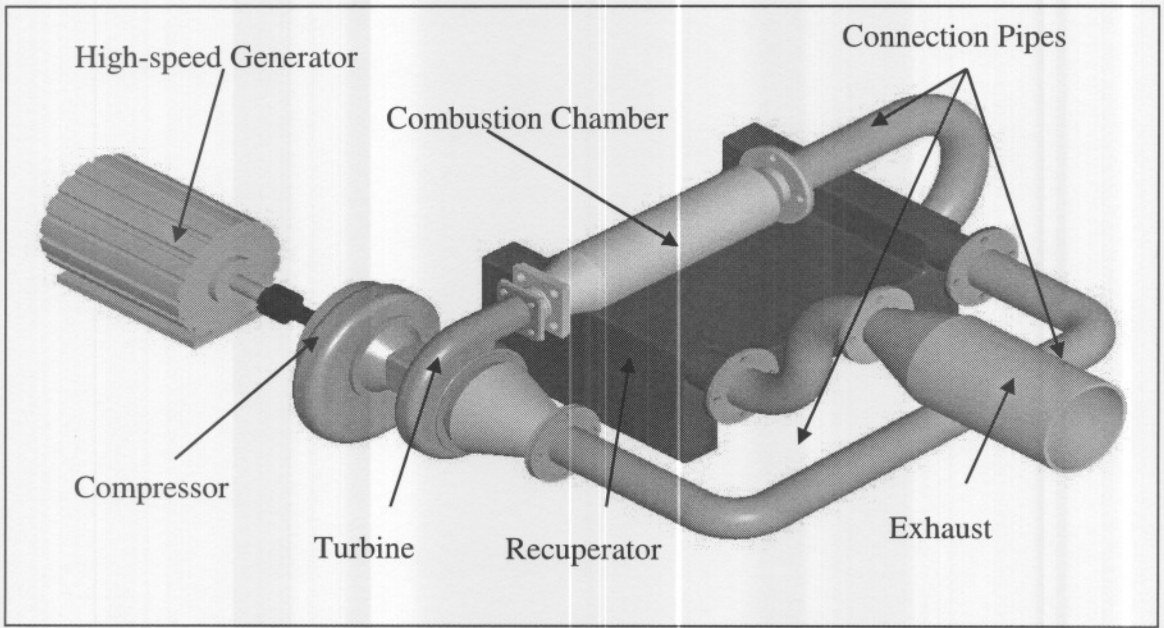

Figure 5.1.1: Conceptual mechanical integration of turbocharger with other components to function as a micro gas-turbine.

5.1.1 Turbocharger modifications

A commercial turbocharger is fitted with a waste gate and a pressure relieve valve for purposes of mechanical control of the turbocharger. In order to use the turbocharger to the extent determined in Chapter 4, these control devises must be removed and replaced with sensors and control actuators where applicable. Because the waste gate is already designed to control the flow through the turbine of the turbocharger, rather than to control the inlet flow to the compressor, it makes more sense to connect it to an active control actuator.

To connect the turbocharger to a generator, a coupling must be designed to screw onto the main shaft of the turbocharger at the compressor wheel. The coupling must be designed to fit to a high-speed generator and need to be as light as possible and perfectly balanced. Provision must be made to allow some misalignment between the turbocharger and the high-speed generator by means of a flexible connection.

5.1.2 Recuperator

The literature study on recuperators discusses a particular core design that is simple to manufacture and has an acceptable high efficiency. To fit this core to a turbocharger the simplest design is a rectangular core section to which the other components is connected by means of headers and pipes as shown in Figure 5.1.2.1. This design is very versatile because the length and width thereof can be increased or decreased to achieve the desired heat transfer and pressure loss (within practical limits). The pipes connecting the other components to the recuperator can be adjusted to refit to the changed recuperator dimensions, allowing for the layout to be very versatile in terms of late design changes.

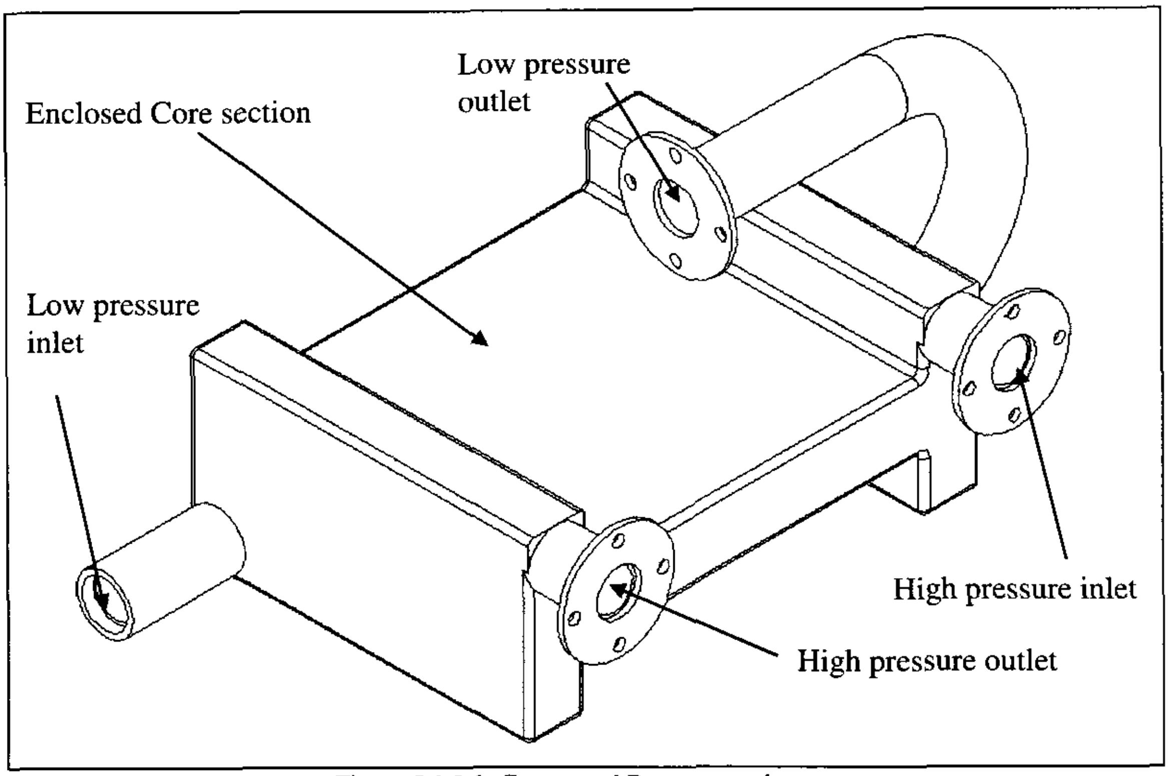

Figure 5.1.2.1: Conceptual Recuperator layout.

The connecting pipes can be designed to accommodate different types of connections on the components. To connect the turbocharger turbine to the recuperator, a reduction piece can be fitted onto the clamp connection at the turbine side and connect to the recuperator with a standard pipe and flange arrangement. The other components are similarly accommodated as shown in Figure 5.1.2.1 and Figure 5.1.1. Expandable metal bellows should be placed between the turbocharger connections and the corresponding components to accommodate thermal expansion.

The material to be used for the recuperator and the pipes could be 347 Austenitic Stainless Steel. This grade of steel was successfully tested in a similar application as discussed in the literature on recuperators.

5.1.3 Combustion chamber

It was concluded from the literature study that the combustion chamber should be of a tubular type. The basic function of the combustor is simple: To burn fuel and therefore supply hot gas to the turbine. Achieving this is however not a simple task. Complete combustion of the fuel must occur to prevent pollutants and maximize the efficiency. The outlet temperature must be stable and uniformly distributed across the outlet. Thermal expansion must be controlled to prevent the combustor from distorting and the pressure loss through the combustor must be kept to a minimum.

5.1.4 Auxiliary systems

The auxiliary systems include the fuel supply system and the lubrication system. The fuel supply system is dependant on the type of fuel for which the combustion chamber is designed. If a liquid fuel is used, it will consist of a fuel tank, a fuel pump, some sort of fuel regulating valve system and a fuel injector inserted into the combustion chamber. When a gas fuel is used, the gas supply might be stored in high-pressure vessels fed directly to the gas injector in the combustion chamber and controlled with a regulating valve.

Because a standard automotive turbocharger is used, the lubrication thereof can be supplied by an electric driven automotive oil pump and standard high-pressure hydraulic piping and filters used on automotive internal combustion engines.

5.1.5 High-speed electric generator

High-speed electric generators were discussed in the literature study. This technology is currently very expensive and in order to remain with the original design specifications of a low-cost product, a low-cost high-speed electric generator should be used.

5.2 Control system

The control system must enable the turbine to follow the load required by the generator. When the load on the high-speed electric generator is followed with variable speed operation of the micro gas-turbine, a reasonable wide operating range can be obtained. As discussed in the literature (2.2.7.4), a control system must be designed to control the mass flow through the gas-turbine with guide vanes or bypass valves and the turbine inlet temperature by regulating the fuel supply to the combustion chamber. Pressure and temperature sensors must be installed at strategic locations on the micro gas-turbine and connected to the control hardware to provide real time monitoring of the micro gas-turbine.

The control system must further function as the primary device to start the micro gas-turbine. During start-up the electric generator must function as a high-speed electric motor to speed up the compressor and turbine. The control system must ad the correct amount of fuel and ignite the fuel at an appropriate time to enable self sustainability of the micro gas-turbine and ultimately to produce excess power.

6. Conclusion and Recommendations

6.1 Conclusion

This study investigated the thermodynamic suitability of a turbocharger for use in a micro gas-turbine. Therefore the literature study investigated the components that make up a micro gas-turbine and specifically focused on the thermodynamic characteristics of these components. Turbochargers were investigated thermodynamically in order to relate the components found in a turbocharger to those found in micro gas-turbines.

A mathematical procedure was taken from recognized literature on gas-turbines and applied in a different manner to enable to the calculation of the excess power available from a certain turbocharger with a fixed turbine inlet temperature. This calculation was done for four different turbochargers with a hand calculation procedure and thereafter compared with calculations done in FLOWNEX (Validated network simulation software package recognized by the industry). The results of the FLOWNEX simulation compared very well to the hand calculations as discussed in chapter 4 and the overall conclusion is the certain turbochargers could produce 30kW excess power. The results from the excess power calculations were used in an EES open recuperated cycle simulation and produced 27kW at an efficiency of 30.2%. When this is compared to the thermodynamic system requirements it is evident that the requirements of 25kW and an efficiency of 20% are met quite realistically. These outputs were obtained with a turbine inlet temperature of 900K (627°C) which is well within the safe operating limits of most turbochargers.

Certain issues with the integration of a turbocharger with other components in the open recuperated cycle were discussed and possible solutions to some of these problems were given. The intent thereof was merely to highlight possible areas of concern foreseen by the author.

Therefore this study proved that a turbocharger is thermodynamically suitable for use in a micro gas-turbine.

6.2 Recommendations

This study proved that a turbocharger is thermodynamically suitable for use in a micro gas-turbine. However, further research is necessary to determine whether a turbocharger is mechanically suitable for use as a micro gas-turbine. The following topics are recommended to be investigated further.

6.2.1 Turbocharger selection: The choice of the turbocharger made in this study cannot be claimed as the best choice, the reason being that no optimization was done during the selection. This study did not aim on selection, but to illustrate that the concept of using a turbocharger is viable from a thermodynamic point of view. Therefore a detailed study must be made of all the commercially available turbocharger units. Factors like the initial cost of the unit, the maintenance cost, the useable output power, the reliability amongst other non obvious factors must be taken into account and a selection must be made accordingly. This unit must then serve as the heart of the micro-turbine and the other components must be designed to accommodate it.

6.2.2 Low-cost recuperator: Another study to be made is the design and testing of a low cost recuperator for the micro gas-turbine. It was concluded from cycle simulation, as done in this study, that the recuperator efficiency is directly responsible for the overall cycle efficiency. Therefore the recuperator efficiency must be as high as possible without creating an excessive pressure drop (above it). The literature on recuperators and the conceptual recuperator design presented in this study can serve as a good starting point for further studies.

6.2.3 Combustion chamber: The development of a combustion chamber is a further study necessary to ultimately produce an efficient micro-turbine. This study concluded that a tubular combustion chamber will be the most appropriate choice due to the inlet characteristics of the turbocharger turbine. It is necessary to conduct detail Computational Fluid Dynamics (CFD) simulations to ensure stable temperatures to the turbine and to produce the least amount of pollutants possible. The design must however be rugged, simple and reliable.

6.2.4 High-speed generator: Developing a low cost high-speed generator for the micro-turbine will contribute to its success in the market. Current high-speed generator technology is expensive and more cost effective designs are needed to reduce the overall cost of the micro gas-turbine. At this stage, a high-speed permanent magnet synchronous generator is recommended because of its simplicity. Should new technology be discovered that poses more advantages, it should be investigated.

6.2.5 Control and Auxiliaries: In addition to the hardware of the micro-turbine the software, which control the micro-turbine, ought to be researched and further developed. Apart from the control sequence and software, actuators and sensors necessary for the control must also be selected or developed. The control system must be sophisticated and reliable but it must also be cost effective and easily maintained. It is the brain of the machine and thorough development thereof is critical to the success of the micro gas-turbine in the market. After final selection of a turbocharger the lubrication requirements for the specific turbocharger must be obtained from the manufacturer or determined by tests. A suitable lubrication system then need to be designed for the turbocharger. It could typically consist of a oil sump, a positive displacement gear pump and high pressure oil pipes with relevant fittings.

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Appendixes A: Thermodynamic calculations

A1. Obtaining curves of maximum matched power output

This section has the aim of explaining the procedure used to find match-points between a specific compressor and a specific turbine when connected with a single shaft. The procedure is based on the methods given by Saravanamuttoo *et al.* (2001). The method described by Saravanamuttoo *et al.* (2001) assumes that full expansion takes place over the turbine for each point on the NDS line and the turbine inlet temperature is then calculated for that point to satisfy the compatibility equations.

The method used in this particular matching procedure varies from the method described in Saravanamuttoo *et al.* (2001), as it is carried out by using a fixed turbine inlet temperature resulting from material limitations of the turbine. Therefore the maximum power output obtainable on each constant NDS (non-dimensional speed) line is calculated by following an iterative procedure that calculates the equilibrium running point for the specific inlet temperature.

The method can be summarized as follows:

- Choose values for the inlet conditions to the compressor (T_{01} and p_{01}). These are usually the ambient conditions where the gas-turbine will operate. The value of the turbine inlet temperature must be chosen and it is useful to choose it equal to the maximum operating temperature of the turbine as stated by the manufacturers. The power calculated will then be the maximum obtainable power from the gas-turbine.
- Choose a constant NDS line on the compressor chart. It is advised to start at the maximum or minimum NSD line and to work ones way through the chart.
- Choose a point on the NDS line, once again it is advised to start at the minimum value of NDM (non dimensional mass flow) on the line.
- Determine the PR, NDM and efficiency of the compressor at the specific point from the compressor charts.

- Now the NDM of the turbine must be calculated from the flow compatibility equation:

$$\frac{m\sqrt{T_{03}}}{p_{03}} = \frac{m\sqrt{T_{01}}}{p_{01}}\left(\frac{p_{01}}{p_{02}}\right)\left(\frac{p_{02}}{p_{03}}\right)\left(\sqrt{\frac{T_{03}}{T_{01}}}\right) \quad \text{(A1.1.1)}$$

The designations 01 and 03 refer to the inlet of the compressor and the turbine respectively as indicated in Figure A1. The equation can then be rewritten to make it better understandable with the use of the compressor and turbine charts. It becomes:

$$NDM_{turbine} = NDM_{compressor}\left(\frac{p_{01}}{p_{02}}\right)\left(\frac{p_{02}}{p_{03}}\right)\left(\sqrt{\frac{T_{03}}{T_{01}}}\right) \quad \text{(A1.1.2)}$$

p_{01} is known and p_{03} is taken as the outlet pressure of the compressor when zero pressure loss is assumed through the combustion chamber. The inlet temperature to the compressor T_{01} is also known and the inlet temperature to the turbine T_{03} is fixed by the maximum operating temperature of the turbine.

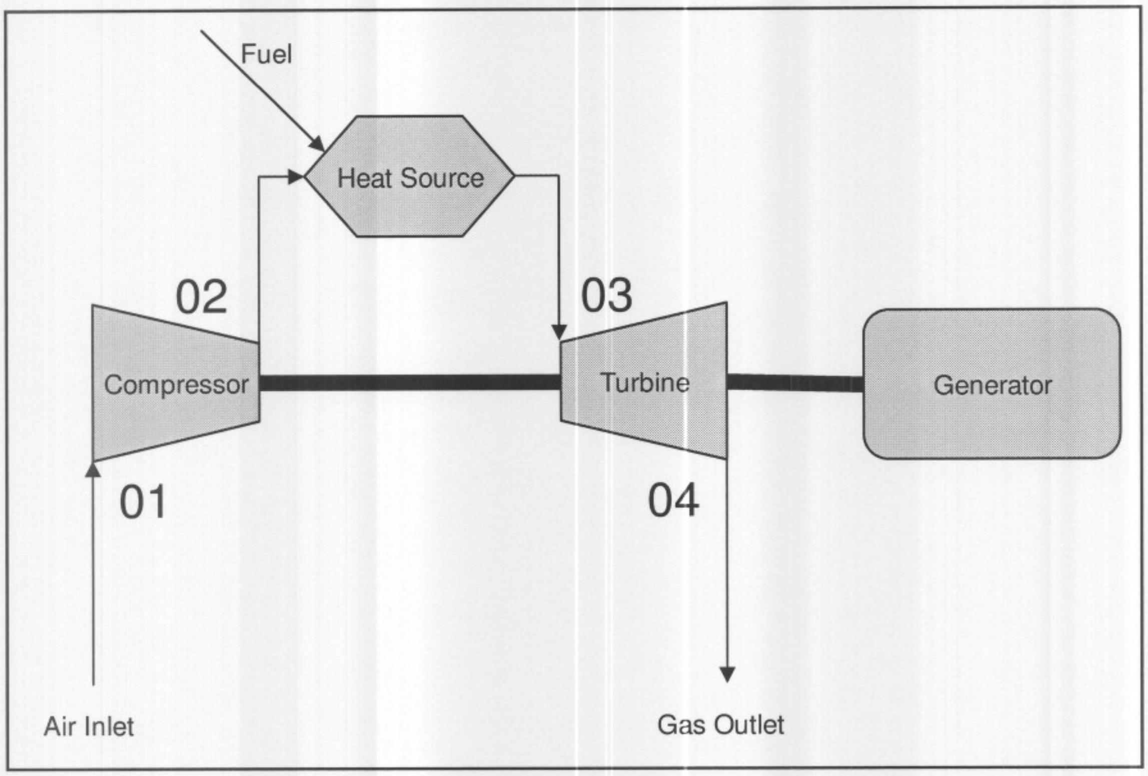


Figure A1: Cycle layout.

- The NDM_t is then used to determine the pressure ratio on the turbine chart. However there are three cases to be assessed with their own sub-procedure to follow.

    - If the NDM_t is less that the value of the NDM_t where choking of the turbine occurs, the value of the turbine PR_t and efficiency must be determined from the turbine charts. The excess power can be calculated for this point and the procedure can be carried out for the next point on the NDS line.
    - If the NDM_t is equal to the NDM_t where choking of the turbine occurs and the $PR_c > PR_t$ at the choke point, the PR_t must be set equal to the PR_c. It therefore implies that the PR_t jumps to the equilibrium value when the choke point is reached on the turbine characteristic. The excess power can

then be calculated with the PR_t and efficiency obtained at the chokepoint. Thus the equilibrium point on the NDS line has is determined and the next line can be considered.

- If the NDM_t is less than the NDM_t where choking of the turbine occurs and the $PR_c < PR_t$ at the calculated NDM_t, the equilibrium point has been overshot and a point to the left of the current point on the NDS line must be chosen. A new value is calculated for the NDMt and the procedure is repeated until the $PR_c = PR_t$. The excess power can then be calculated for this point and it will also be the equilibrium point on the NDS line. The next line can then be considered.

When the equilibrium point on the NDS line has been obtained, the pressure ratios, efficiencies and the mass flow of the compressor and turbine can be used to calculate the excess power for the point. The temperature rise through the compressor and the temperature drop through the turbine are calculated with Equation A1.1.3 and Equation A1.1.4 respectively.

$$\Delta T_{012} = \frac{T_{01}}{\eta_t}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right] \tag{A1.1.3}$$

$$\Delta T_{034} = \eta_t T_{03}\left[1-\left(\frac{1}{p_{03}/p_{04}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right] \tag{A1.1.4}$$

The excess work is the calculated with Equation A3.1.5

$$Excess\ Power = mCp_g \Delta T_{034} - \frac{1}{\eta_m} mCp_a \Delta T_{012} \tag{A1.1.5}$$

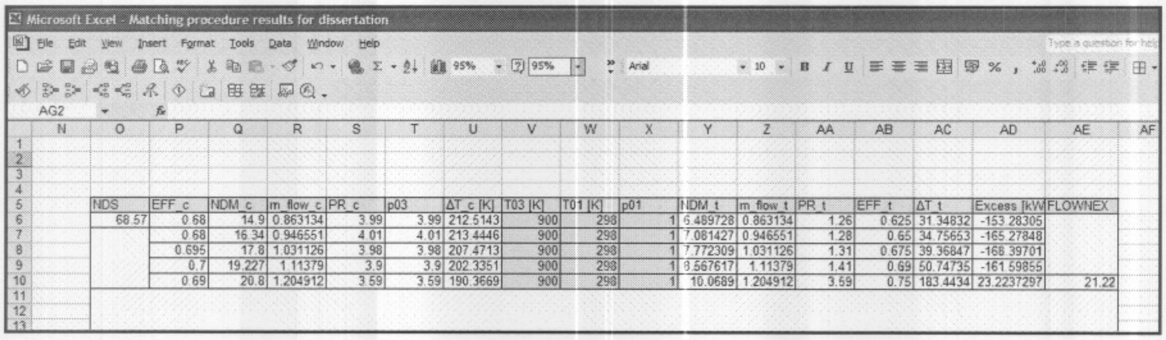
The procedure can be schematically represented like shown in Figure 4.1.1.2. in Chapter 4. It is an iterative procedure and specifically determines the match-points for a specific turbine inlet temperature. The procedure is carried out for each constant NDS line on the compressor chart. Thus a series of points is obtained where the combination of compressor and turbine will operate with the fixed turbine inlet temperature.

A2. Obtaining Equilibrium Operating Points with Excel Spreadsheets

The mathematical procedure discussed in section A1 was set up in Microsoft Excel to assist with the calculation of the equilibrium operating points. This section will attempt to explain the Excel spread sheet and then present the calculated spread sheets for all four turbocharger Units.

A2.1 Spreadsheet set up

The Excel spreadsheet is only an extension of the hand calculation procedure. The calculations were done in a spreadsheet to enable easy data management and to generate the trends discussed throughout Chapter 4. Figure A2.1.1 shows a screen shot taken from the Excel spread sheet. It only shows the calculation of one NDS (Non Dimensional Speed) line, but this procedure is exactly the same for all NDS lines of all the turbocharger Units analyzed. The Microsoft Excel spreadsheet file is attached to this document by means of a CD.


Microsoft Excel - Matching procedure results for dissertation

NDS	EFF_c	NDM_c	m_flow_c	PR_c	p03	ΔT_c [K]	T03 [K]	T01 [K]	p01	NDM_t	m_flow_t	PR_t	EFF_t	ΔT_t	Excess [kW	FLOWNEX
68.57	0.68	14.9	0.863134	3.99	3.99	212.5143	900	298	1	6.489728	0.863134	1.26	0.625	31.34832	-153.28305	
	0.68	16.34	0.946551	4.01	4.01	213.4446	900	298	1	7.081427	0.946551	1.28	0.65	34.75653	-165.27848	
	0.695	17.8	1.031126	3.98	3.98	207.4713	900	298	1	7.772309	1.031126	1.31	0.675	39.36847	-168.39701	
	0.7	19.227	1.11379	3.9	3.9	202.3351	900	298	1	8.567617	1.11379	1.41	0.69	50.74735	-161.59855	
	0.69	20.8	1.204912	3.59	3.59	190.3669	900	298	1	10.0689	1.204912	3.59	0.75	183.4434	23.2237297	21.22

Figure A 2.1.1: Screenshot of the Excel spreadsheet set up for turbocharger Unit 4

In order to explain the calculations done in the spreadsheet, the origin or calculation of the value of each cell is explained in table A2.1.1. However, this is only done for row 6 because it is similar for all the other rows. Cell O6 contains the value of the NDS line analyzed on the compressor characteristic. This value is not used in calculations and therefore only serves as a reference of the NDS line analyzed.

All the values in line 6 relates to the most left hand point on the specific NDS line shown in cell O6.

Table A2.1.1: Explanation of the procedure as used in the excel spreadsheet.

Cell reference	Description	Origin
O6	Non-Dimensional Speed	Compressor characteristic
P6	Compressor efficiency	Compressor efficiency characteristic
Q6	Compressor Non-Dimensional Mass Flow	Compressor characteristic
R6	Mass flow through compressor	$R6 = Q6 \times \left(\frac{X6}{\sqrt{W6}}\right)$
S6	Compressor pressure ratio	Compressor characteristic
T6	Turbine inlet pressure	$T6 = S6 \times X6$
U6	Temperature increase over the compressor	$U6 = \frac{W6}{P6}\left[(S6)^{\left(\frac{1.4-1}{1.4}\right)} - 1\right]$
V6	Turbine inlet temperature	Specified as 900K
W6	Compressor inlet temperature	Specified as 298K
X6	Compressor inlet pressure	Specified as 1bar
Y6	Turbine Non-Dimensional Mass flow	$Y6 = Q6\left(\frac{X6}{T6}\right)\left(\sqrt{\frac{V6}{W6}}\right)$

Z6	Mass flow through turbine	$Z6 = Y6 \times \left(\frac{T6}{\sqrt{V6}} \right)$
AA6	Turbine pressure ratio	Turbine characteristic
AB6	Turbine efficiency	Turbine efficiency characteristic
AC6	Temperature decrease over the turbine	$AC6 = AB6 \times V6 \times \left[1 - \left(\frac{1}{AA6} \right)^{\left(\frac{1.33-1}{1.33} \right)} \right]$
AD	Excess power available	$AD6 = (Z6 \times 1.148 \times AC6) - (Z6 \times 1.005 \times U6)$
AE	Excess power calculated with FLOWNEX	FLOWNEX simulation

In Table A2.1.1 it is indicated that some of the values is obtained from the compressor and turbine characteristics. Figure A.2.1.2 shows the compressor characteristic that relates the pressure ratio over the compressor to the non-dimensional mass flow through the compressor.

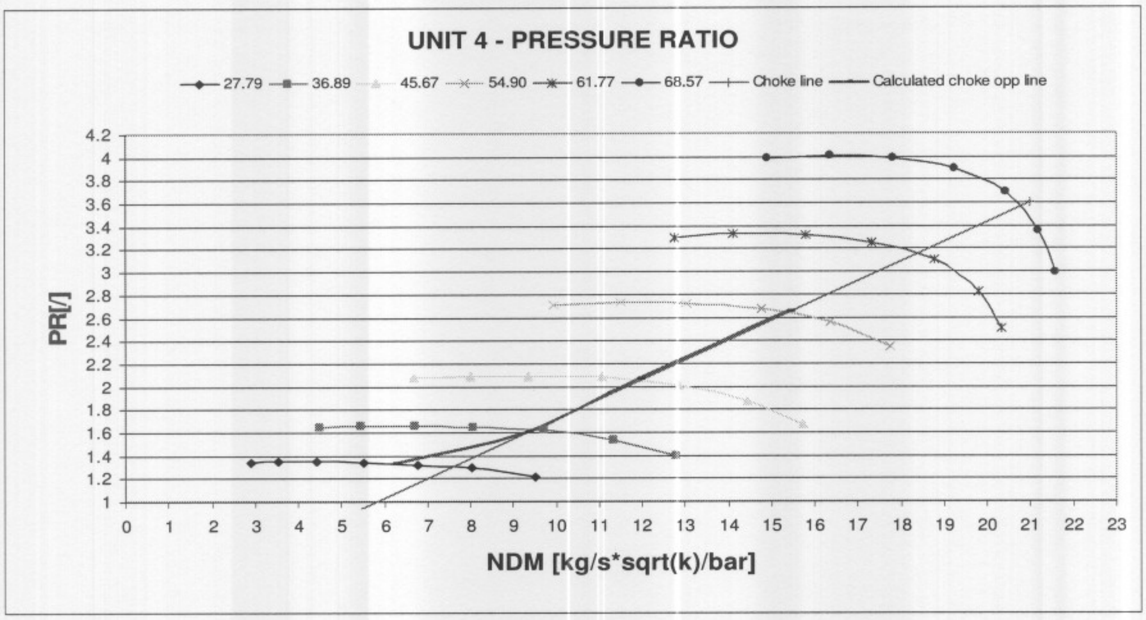


Figure A2.1.2: Compressor characteristic with pressure ratio plotted against non-dimensional mass flow.

The efficiency of the compressor related to the non dimensional mass flow is shown in figure A2.1.3.

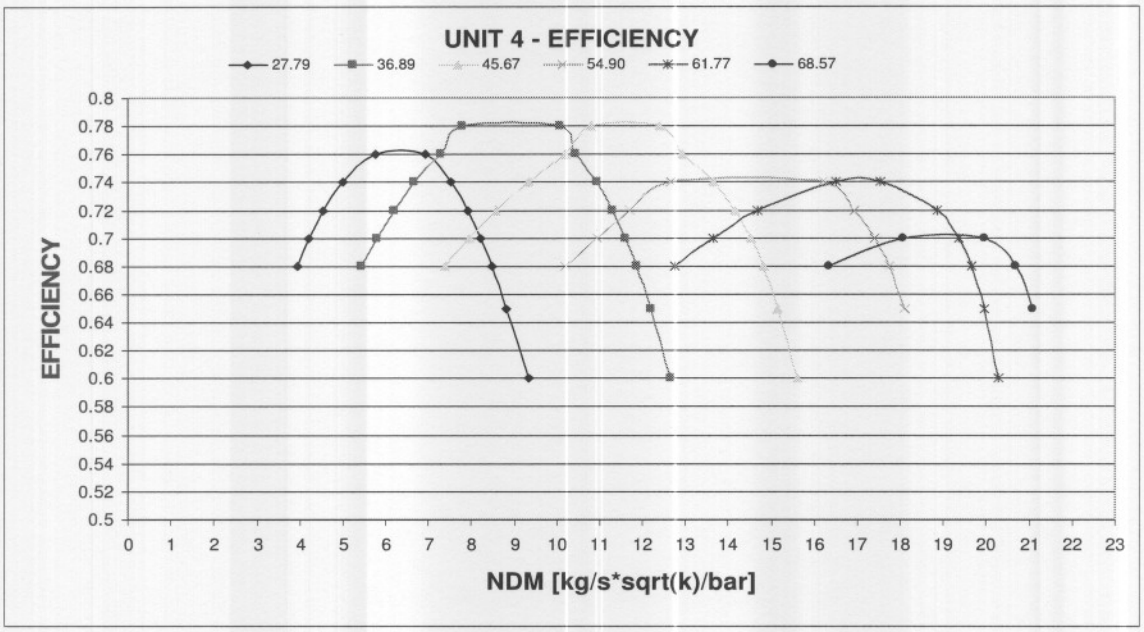


Figure A 2.1.3: Compressor characteristic with efficiency plotted against non dimensional mass flow.

Figure A2.1.4 shows the turbine characteristic with pressure ratio plotted against non dimensional mass flow while figure A2.1.5 shows the turbine characteristic for efficiency plotted against non dimensional mass flow.

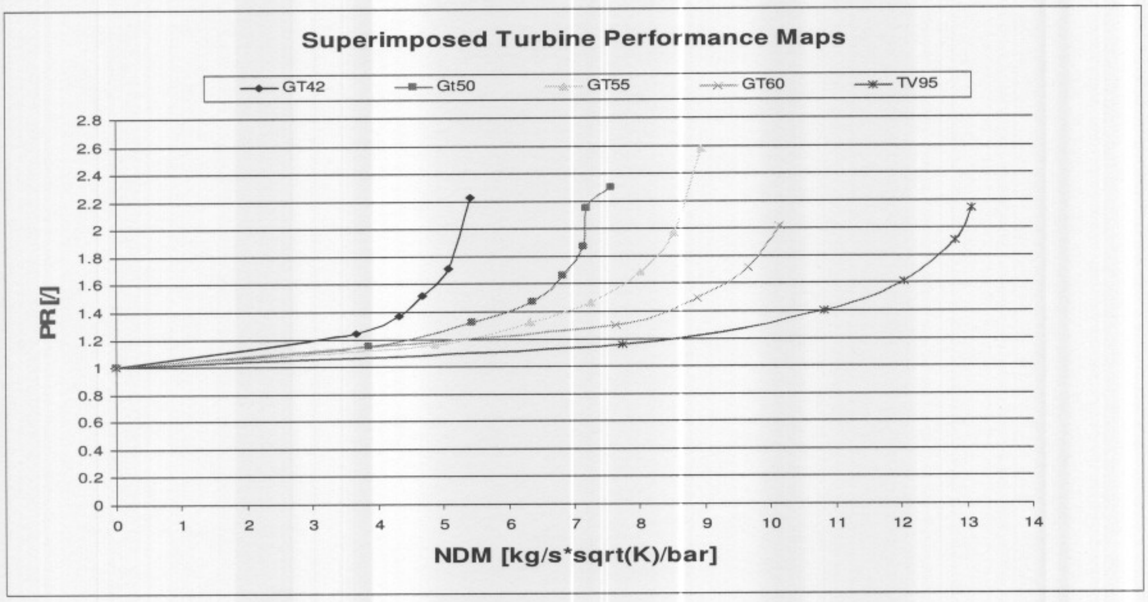


Figure A 2.1.4: Turbine characteristic with pressure ratio plotted against non dimensional mass flow.

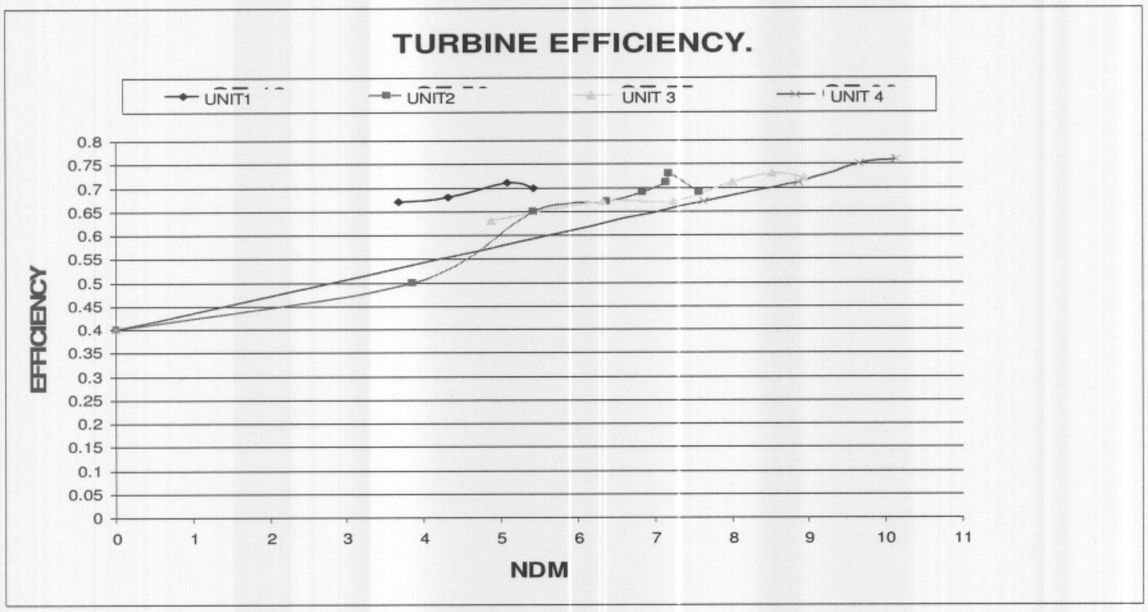


Figure A 2.1.5: Compressor characteristic with efficiency plotted against non dimensional mass flow.

A2.2 Spreadsheet results

The method of calculating the equilibrium operating point with the Excel spreadsheets was discussed in detail in section A 2.1. Different points on each constant speed line were analyzed and the procedure was terminated once the pressure ratios of the compressor and turbine were equal. As a result, it implies a self restrained condition for the given turbine inlet temperature of 900K.

A3 FLOWNEX model

A model was set up in FLOWNEX to calculate the equilibrium operating point for each non-dimensional speed line on the compressor characteristic. Figure A3.1 shows a screenshot taken from the FLOWNEX program. The FLOWNEX model ignores losses through the ducting, recuperator and the combustion chamber. The maps of the compressor and turbine was imported into FLOWNEX and assigned to the compressor and turbine. The shaft speed, ambient inlet conditions and turbine inlet temperature were set to be the same as the values used in the Excel spreadsheet. The value of the excess power was calculated for each constant speed line on the compressor chart and this value is printed next to the equilibrium operating point in each Excel spreadsheet.

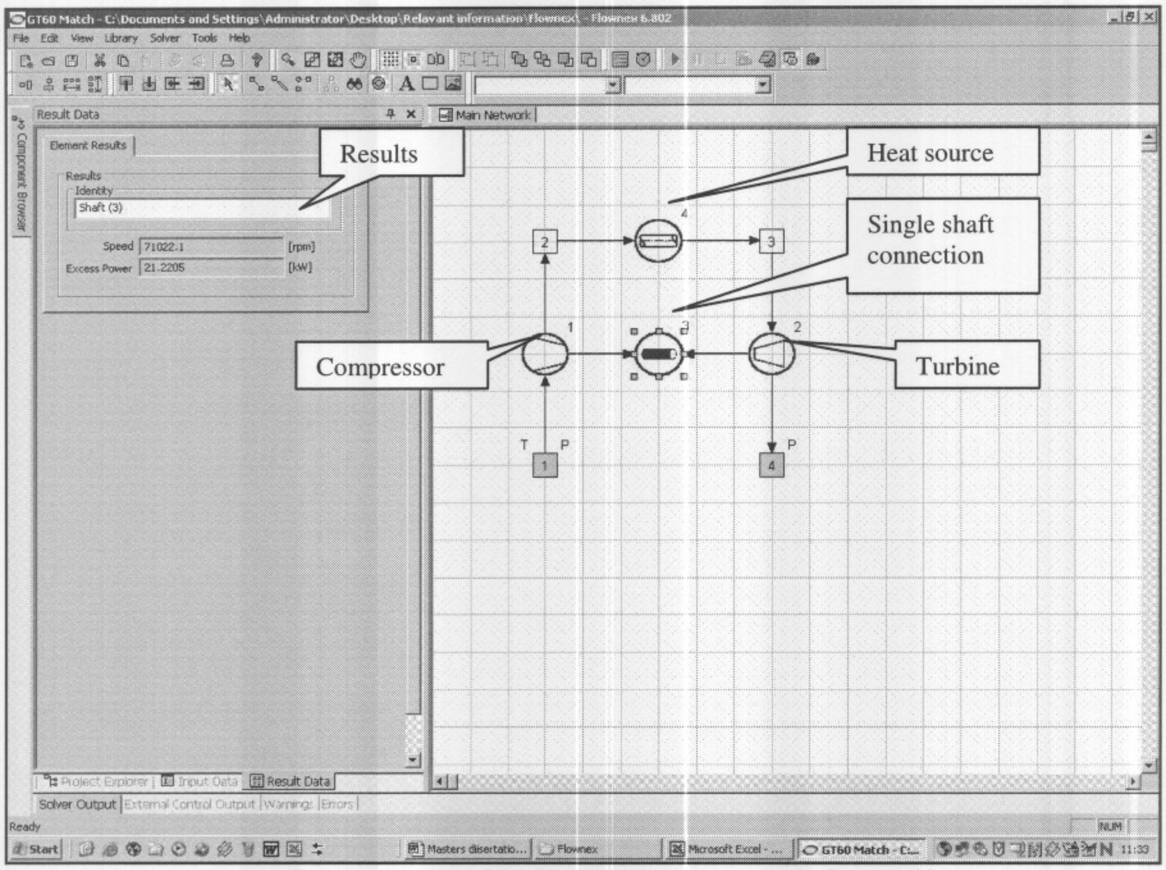


Figure A3.1: Screenshot of the FLOWNEX program used for matching.

The results obtained from both the Excel spreadsheet and the FLOWNEX model was plotted on the same chart for comparison. Figures A3.2 to A3.5 shows comparisons of the excess power as obtained by Excel and FLOWNEX.

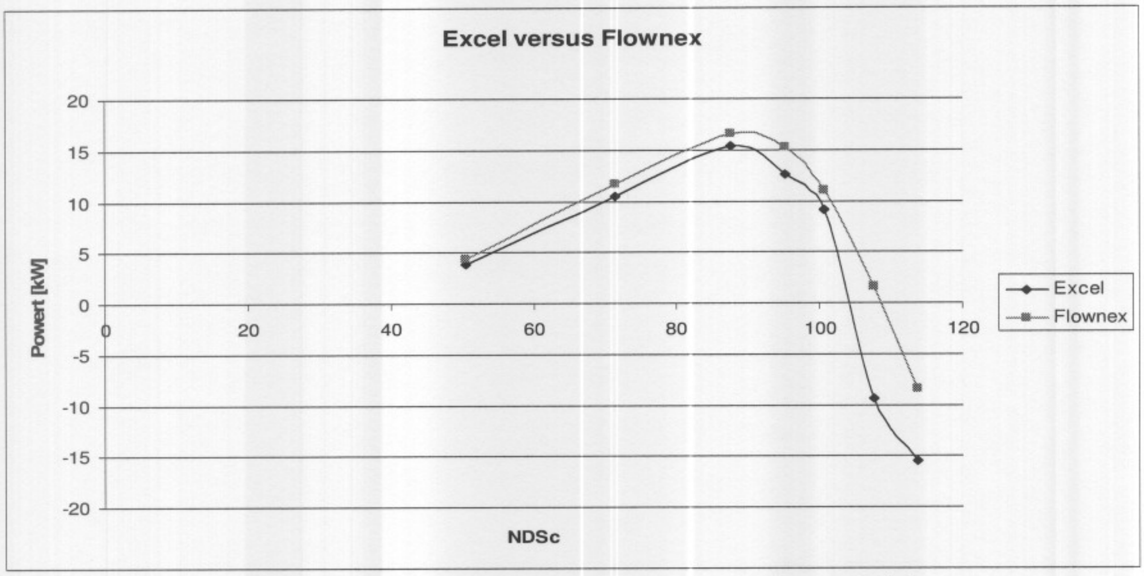


Figure A3.2: Excess power obtained from turbocharger Unit 1.

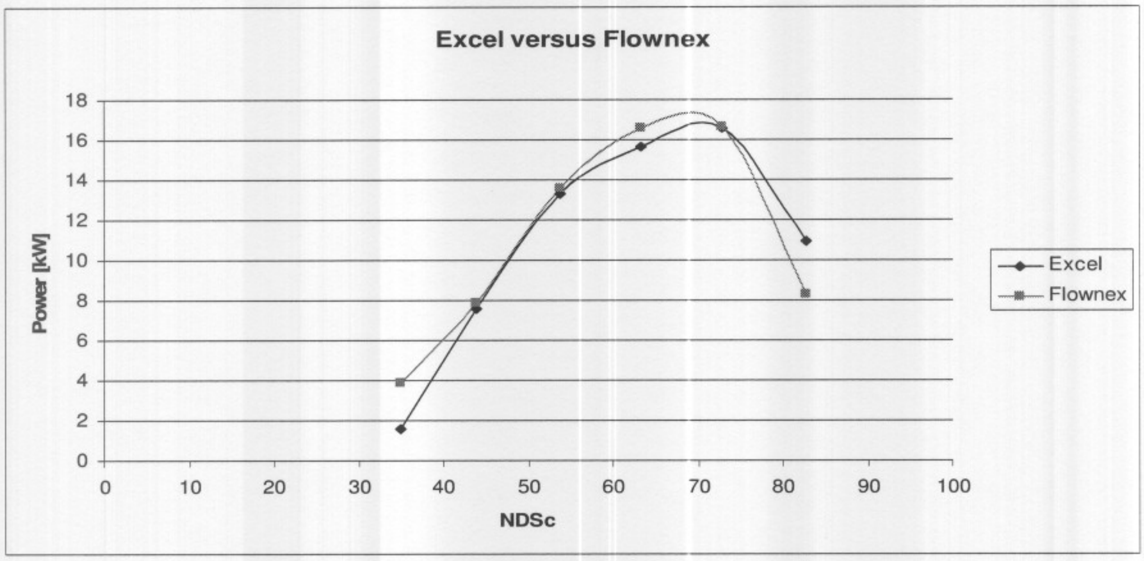


Figure A3.3: Excess power obtained from turbocharger Unit 2.

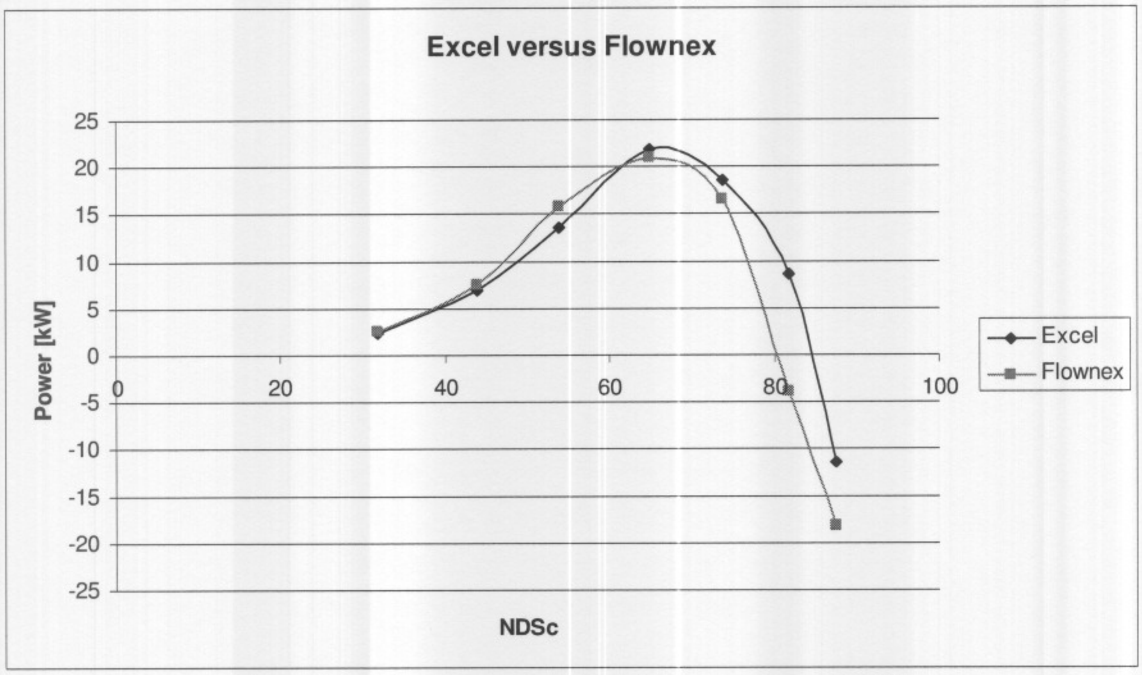


Figure A3.4: Excess power obtained from turbocharger Unit 3.

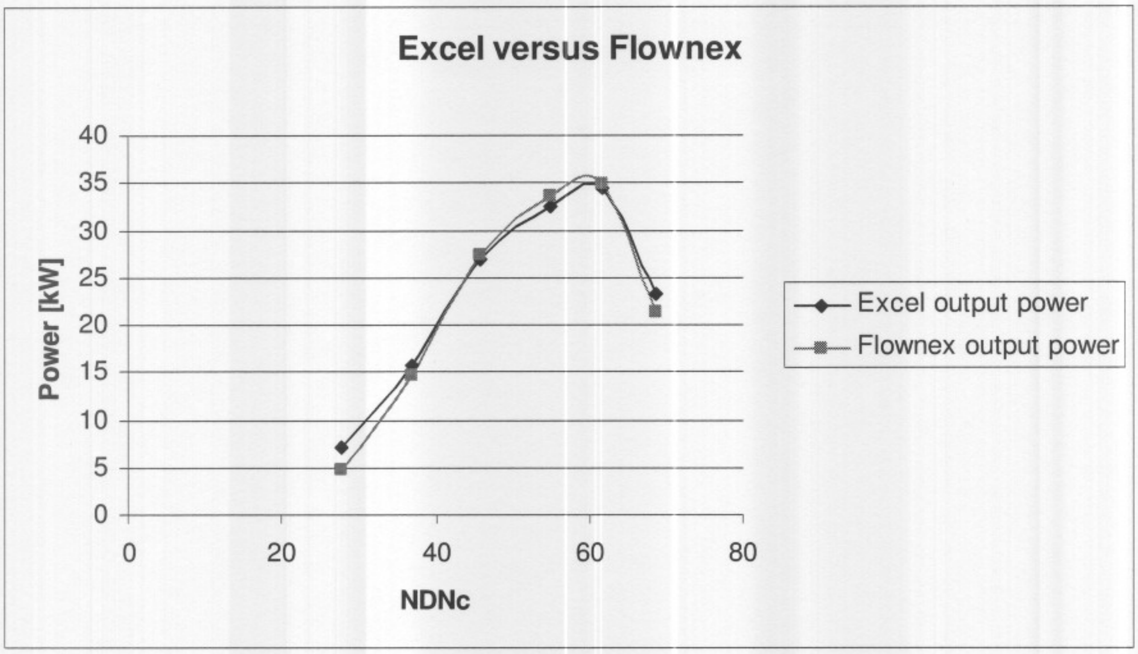


Figure A3.5: Excess power obtained from turbocharger Unit 4.

A4 Recuperated open cycle simulation

The cycle simulation was done by calculating the cycle parameters with a set of hand calculations for a chosen non dimensional speed. The cycle was then simulated by programming the set of calculations in EES (Engineering Equation Solver). The results of the values according to the EES calculation were evaluated against the results obtained from the hand calculation in order to validate the EES program. Once the EES program was validated, it was used to calculate the cycle parameters for the other non dimensional speeds of the turbocharger Unit.

A4.1 Hand calculated cycle parameters

A certain non-dimensional speed was chosen to calculate the cycle parameters when turbocharger Unit 4 is connected in an open recuperated micro gas-turbine cycle. The pressure ratio and efficiency of the compressor and turbine were read from the compressor and turbine characteristics at the equilibrium operating point of the chosen non-dimensional speed.

The following set of calculations is done for turbocharger Unit 4.

Non-dimensional speed (NDS)	=	45.67
Non-dimensional mass flow (NDM)	=	11.9
Corresponding compressor pressure ratio (PR_c)	=	2.06
Corresponding turbine pressure ratio (PR_t)	=	2.06
Compressor inlet pressure (p_1)	=	1 [bar]
Compressor inlet temperature (T_1)	=	298 [K]

Calculate the mass flow:

$$m = \frac{NDM_c \times p_1}{\sqrt{T_1}}$$

$$m = \frac{11.9 \times 1}{\sqrt{298}} = 0.689$$

Calculate compressor outlet pressure:

$$p_2 = p_1 \times PR_c$$

$$p_2 = 1[bar] \times 2.06 = 2.06[bar]$$

Calculate temperature increase over compressor:

$$\Delta T_c = \frac{T_1}{\eta_c}\left(PR_c^{\left(\frac{\gamma-1}{\gamma}\right)} - 1 \right)$$

$$\Delta T_c = \frac{298}{0.78}\left(2.06^{\left(\frac{1.4-1}{1.4}\right)} - 1 \right) = 87.62[K]$$

Calculate temperature after compressor:

$$T_2 = T_1 + \Delta T_c$$

$$T_2 = 298 + 87.62 = 385.6[K]$$

Assume no pressure loss over the recuparator:

$$p_3 = p_2$$

Assume 2% pressure drop over heat source:

$$p_4 = p_3 \times 0.98$$

$$p_4 = 2.06 \times 0.98 = 2.018[bar]$$

The turbine inlet temperature is specified:

$$T_4 = 900K$$

Calculate pressure after turbine:

$$p_5 = \frac{p_4}{PR_t}$$

$$p_5 = \frac{2.018}{2.06} = 0.979[bar]$$

Calculate the temperature decrease over the turbine:

$$\Delta T_t = \eta_t \times T_4 \left(1 - \left(\frac{1}{PR_t} \right)^{\frac{\gamma-1}{\gamma}} \right)$$

$$\Delta T_t = 0.75 \times 900\left(1 - \left(\frac{1}{2.06}\right)^{\frac{1.33-1}{1.33}}\right) = 110.8[K]$$

Calculate the turbine exit temperature:

$$T_5 = T_4 - \Delta T_c$$

$$T_5 = 900 - 110.8 = 789.2[K]$$

Calculate maximum temperature difference between the two fluid streams in the ecuperator:

$$\Delta T_{\max} = T_5 - T_2$$

$$\Delta T_{\max} = 789.2 - 385.6 = 403.6$$

Calculate the theoretical maximum heat transfer in the recuperator:

$$Q_{\max} = m \times Cp_{gas} \times \Delta T_{\max}$$

$$Q_{\max} = 0.6893 \times 1.148 \times 403.6 = 319.4[kW]$$

Calculate the real heat transfer in the recuperator:

$$Q_{real} = Q_{\max} \times \eta_{recuperator}$$

$$Q_{real} = 319.4 \times 0.85 = 271.5[kW]$$

Calculate the exit temperature at the high pressure side of the recuperator just before it enters the combustion chamber:

$$T_3 = \frac{Q_{real} - (m \times Cp_{air} \times T_2)}{m \times Cp_{air}}$$

$$T_3 = \frac{271.5 + (0.6893 \times 1.005 \times 385.6)}{0.6893 \times 1.005} = 777.5[K]$$

Calculate the exit temperature at the low pressure side of the recuperator where it is exhaust to the atmosphere:

$$T_6 = \frac{(m \times Cp_{gas} \times T_5) - Q_{real}}{m \times Cp_{gas}}$$

$$T_6 = \frac{(0.6893 \times 1.148 \times 789.2) - 271.5}{0.6893 \times 1.148} = 446.2[K]$$

Calculate the power absorbed by the compressor:

$$W_c = m \times Cp_{air} \times (T_2 - T_1)$$

$$W_c = 0.6893 \times 1.005 \times (385.6 - 298) = 60.7[kW]$$

Calculate the power generated by the turbine:

$$W_t = m \times Cp_{gas} \times (T_4 - T_5)$$

$$W_t = 0.6893 \times 1.148 \times (900 - 789.2) = 87.7[kW]$$

Calculate the excess power:

$$W_{excess} = W_t - W_c$$

$$W_{excess} = 87.7 - 60.7 = 27[kW]$$

Calculate the size of the heat source:

$$Q_{heat} = \frac{m \times Cp_{air} \times (T_4 - T_3)}{\eta_{hs}}$$

$$Q_{heat} = \frac{0.6893 \times 1.005 \times (900 - 777.5)}{0.95} = 89.3[kW]$$

Calculate the cycle efficiency:

$$\eta_c = \frac{W_{excess}}{Q_{heat}}$$

$$\eta_c = \frac{27}{89.3} = 0.302 \cong 30.2\%$$

The parameters obtained from the hand calculation is summarised in table A4.1.1 and it is clear that turbocharger Unit 4 functions adequately when connected to the other components in the open recuperated cycle.

Table A4.1.1: Thermodynamic cycle parameters.

Compressor	
Inlet temperature [K]	298
Outlet temperature [K]	385.6
Pressure ratio	2.06
Efficiency [%]	78
Recuperator	
High pressure inlet temperature [K]	385.6
High pressure outlet temperature [K]	777.5
Low pressure inlet temperature [K]	789.2
Low pressure outlet temperature [K]	446.2
Efficiency [%]	85
Combustion chamber	
Combustion chamber inlet temperature [K]	777.5
Combustion chamber outlet temperature [K]	900
Efficiency [%]	95
Turbine	
Inlet temperature [K]	900
Outlet temperature [K]	789.2
Pressure ratio	2.06
Efficiency [%]	75
Calculated power/duty	
Compressor power [kW]	60.7
Turbine power [kW]	87.7
Excess power [kW]	27
Recuperator duty [kW]	271.5
Heat source duty [kW]	89.3
Cycle efficiency [%]	30.2

A4.2 Program code as programmed in EES

The set of formulas which was used in the hand calculation of the cycle, was programmed in EES to simulate the open recuperated cycle for various non dimensional speed values. The following is the code as programmed in EES:

```
"Simulation for purpose of M.Ing 2005"
"Cycle simulation of a recuperated open brayton cycle"

"Data required for input variables is obtained from mathing procedure of a compressor and
turbine"

"GT60 turbine and compressor pair"

"============================================================================="
"Supplied Parameters"
T[1] = 298                                           "Minimum Temperature [K]"
T[4] = 900                                           "Maximum Temperature[K]"
P[1] = 1                                             "Minimum pressure [kPa]"
{NDM_c = 10.8                  "Non-dimensional mass flow from compressor chart"
PR_c = 2.1                                           "Compressor pressure ratio"
eta_c = 0.72                                 "Compressor isentropic efficiency"
eta_t = 0.72                                 "Turbine isentropic efficiency"
PR_t = 2.57                                  "Turbine pressure ratio"
NDS_c = 45.67}           "Non-dimensional rotational speed from compressor chart"
"============================================================================="
"Fluid properties"
Cp_a = 1.005                          "Cp-value for air entering the compressor"
Cp_g = 1.148                  "Cp-value for gas exiting the combustion chamber"
gamma_c = 1.4                 "Gamma value corresponding to compressor"
gamma_t = 1.33                "Gamma value corresponding to the turbine"
R = 0.287                     "Gas constant"
"============================================================================="
```

```
"Efficiencies"
eta_m = 1                                          "Mechanical efficiency"
eta_hs = 0.95                                      "Heat source efficiency"
eta_gen = 0.95                                     "Generator efficiency"
eta_recup = 0.9                                    "Recuperator efficiency"
"=============================================================================="
"Mass flow [kg/s]"
m_flow = (NDM_c*P[1])/(SQRT(T[1]))   "Calculated from the non-dimensional mass flow from
                                      the charts"
"=============================================================================="
"Compressor "
P[2] = P[1] * PR_c
(T[2]-T[1]) = (1/eta_c)*(T[1])*(((PR_c)^((gamma_c-1)/gamma_c))-1)    "Compressor equation
                                                                      used to calculate the
                                                                      temperature increase"
DELTAT_comp = (T[2]-T[1])
"=============================================================================="
"Heat Source"
P[3] = P[2]                                        "Assuming no loss in the pipe"
P[4] = 0.98*P[3]      "Pressure loss through combustor assumed 2% of  compressor pressure"
"=============================================================================="
"Turbine"
PR_t = P[4]/P[5]
(T[4]-T[5]) = (eta_t)*(T[4])*(1-((1/(PR_t))^((gamma_t-1)/gamma_t)))    "Turbine equation used
                                                                        to calculate the
                                                                        temperature decrease"
DELTAT_turb = (T[4]-T[5])

"=============================================================================="

"Compressor work"
W_c = (m_flow * Cp_a * (T[2] - T[1]))          "The work consumed by the compressor"
"=============================================================================="
"Work supplied by turbine"
W_t = m_flow * Cp_g * (T[4] - T[5])            "The power supplied by the turbine"
"=============================================================================="
```

```
"Power output  of the generator"
W_gen = ((W_t * eta_m)  - W_c)          "Generator power develloped assuming 100% efficient
                                        generator"
"================================================================================"
"Size of heat source"
Q_hs = (m_flow * Cp_a * (T[4] - T[3]) )/eta_hs          "The heat that must be supplied by the
                                                        heat source (Combuster)"
"================================================================================"
"Overall cycle eff."
eta_cycle = W_gen/Q_hs                                          "The calculated cycle
efficiency"
"================================================================================"
"Recuperator"
DELTAT_max = T[5] -T[2]
Q_max = m_flow*Cp_g*(DELTAT_max)            "The max. amount of heat that can be tranfered
                                            in a 100% eff recuperator"
Q_real = Q_max *eta_recup                   "The actual heat tranferd due to inefficiencies"
Q_real = Q_HP
Q_HP = Q_LP
Q_HP = m_flow*Cp_a * (T[3] - T[2])          "Calculation of temperature, dependant on fluid
                                            charateristic and flow"

Q_LP = m_flow * Cp_g*(T[5]-T[6])            "Calculation of temperature, dependant on fluid
                                            charateristic and flow"
"================================================================================"
"Entropy"
Duplicate i =1,6
        S[i] = ENTROPY(Air,T=T[i],P=P[i])
end
P[6] = P[1]
```

The simulation was done by taking the values of the pressure ratios and the efficiencies for the compressor and turbine from the matching procedure. Therefore the line of maximum output was simulated with a parametric table in EES. Figure A4.2.1 shows the parametric table while Figure A4.2.2 shows the results obtained from the simulation.

1..6	NDS_c	NDM_c	PR_c	η_c	PR_t	η_t	m_{flow}	η_{cycle}	W_{gen}
Run 1	68.57	20.8	3.59	0.69	3.59	0.75	1.205	0.09585	23.22
Run 2	61.77	18.4	3.16	0.728	3.16	0.75	1.066	0.174	34.46
Run 3	54.9	15.35	2.65	0.74	2.65	0.75	0.8892	0.2233	32.45
Run 4	45.67	11.9	2.06	0.78	2.06	0.75	0.6893	0.302	26.98
Run 5	36.89	8.8	1.62	0.78	1.62	0.75	0.5098	0.3229	15.64
Run 6	27.79	6.2	1.33	0.76	1.33	0.75	0.3592	0.3005	6.998

Figure A4.2.1: Parametric table used to simulate the line of maximum output.

Unit Settings: [kJ]/[K]/[bar]/[kg]/[degrees]

Cp_a = 1.005	Cp_g = 1.148	ΔT_{comp} = 87.62	ΔT_{max} = 403.6	ΔT_{turb} = 110.8	η_c = 0.78	η_{cycle} = 0.302
η_{gen} = 0.95	η_{hs} = 0.95	η_m = 1	η_{recup} = 0.85	η_t = 0.75	γ_c = 1.4	γ_t = 1.33
m_{flow} = 0.6893	NDM_c = 11.9	NDS_c = 45.67	PR_c = 2.06	PR_t = 2.06	Q_{HP} = 271.5	Q_{hs} = 89.36
Q_{LP} = 271.5	Q_{max} = 319.4	$Q_{werklik}$ = 271.5	R = 0.287	W_c = 60.71	W_{gen} = 26.98	W_t = 87.69

Calculation time = .0 sec

Array variables are in the Arrays window

Figure A4.2.2: Results obtained from EES.

Figure A4.2.3 shows the cycle power output obtained and Figure A4.2.4 shows the Cycle efficiencies obtained.

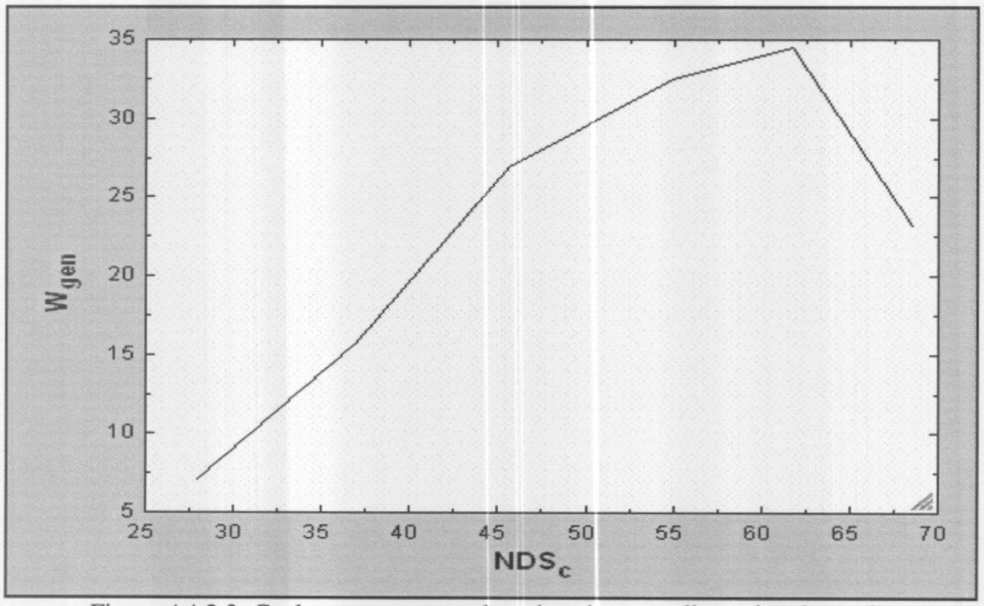


Figure A4.2.3: Cycle power output plotted against non dimensional speed.

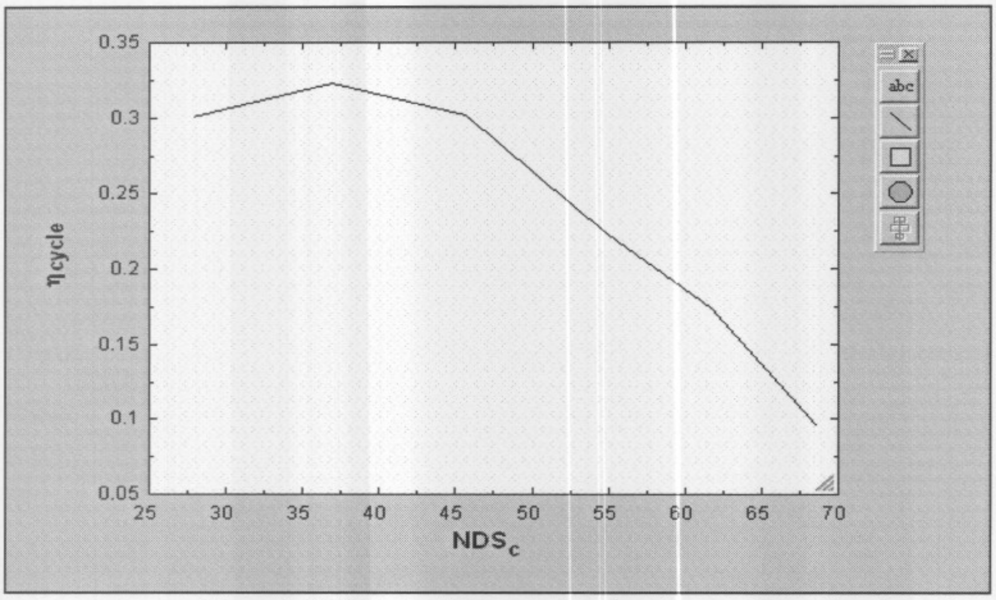


Figure A4.2.4: Cycle efficiency plotted against non dimensional speed.