Bachelor of Engineering Thesis

Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application

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Course Code: ENGG4011

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A thesis submitted in partial fulfillment of the requirements of the Bachelor of Engineering degree in the specialization Mechanical and Aerospace Engineering

UQ Engineering

Faculty of Engineering, Architecture and Information Technology
30/05/2011

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Dear Sir,

I hereby submit my Thesis titled “Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application” for consideration as partial fulfillment of the Bachelor of Engineering degree.

All the work contained within this Thesis is my original work except where otherwise acknowledged.

I request that this thesis be kept confidential for a minimum period of 5 years from the date of submission. Please refer to the copy of the confidentiality agreement which is enclosed.

Yours sincerely

Adam Joseph Head
Student ID: 41244432
A design report developed in collaboration with Microturbine Technology (MTT), Technische Universiteit Delft (TUD) and Toegepast Natuurwetenschappelijk Onderzoek (TNO)

MTT/ TUDelft/ TNO

The Netherlands

(June, 2011)

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Abstract

The microturbine seems to be a viable option for implementation into Hybrid Electric Vehicle (HEV) systems. A microturbine was constructed in the Gas Turbine Simulation software GSP in order to assess its viability for a low-power range extender. Due to the small magnitude of the turbine, scale effects need to be incorporated into the performance models. A microturbine has various advantages over other heat engines (Wankel, Piston, or Fuel Cell), and the capabilities in terms of range extension of the HEV are potentially superior.

Through the use of the Engineering Design Process, a new microturbine design was developed that allows the system to be implemented into the HEV system. Aspects such as geometric size, weight, cost, availability, and ease of production were used to compare the different concepts and determine their feasibility. Empirical loss models previously researched were adapted and implemented into scheduling components of the microturbine base model. The model was used to simulate the required output data (Mechanical/ Electric Power, \(CO_2\) emissions, Fuel flow rate, Exhaust gas temperature and Exhaust gas mass flow) for a range of predefined design powers (9, 15, 22, 30, 36 kW). Power-to-weight ratios and component dimensions were also calculated and sent for analysis. The data above was generated under two control schemes (fixed and variable speed) and at three power codes; maximum power (100%), part load (60%) and idle (20%). The HEV model used this data to configure and size its own system. Simulations of design and optimization are important as it restricts the size of the HEV. The results suggest that the variable speed control scheme will extend the life of the system and reduce emissions substantially. If the microturbine is operated at or below ISA conditions the scheme offers numerous other advantages. However the control system is far more complex and will cost more to develop.

Recommendations have been highlighted for model configuration improvement and focus on the control system is important for continued programme development.
Acknowledgements

For most I wish to thank Wilfried Visser, my industrial supervisor, for the continuous support and invaluable supervision. I would like to extend my eternal gratitude for the guidance and knowledge offered. The experience mastered will be of great use in the engineering industry. I would also like to offer thanks to Arvind Rao, my TU university supervisor, for the kind assistance and suggestions which greatly improved the quality of my work. I wish to offer thanks to my UQ supervisor David Mee, whose continued constructive comments improved the quality of my report and authority allowed me to conduct this thesis. A brief thanks to Savad Shakariyants and Colin Rodgers for their personal support, suggestions and help with various aspects of the assignment.

I would like to thank the following PhD students for their help with various issues throughout my internship. Thanks to Michel Verbist, Adeel Javed and Mattia Olivero for providing their resources in literature. Iliane Dountchev for his help with the GSP working environment.

I would also like to offer a special thanks to Fatma Cinar (international coordinator) for making my stay here in Delft possible. A brief special thanks goes to Wolbodo, the studentenvereniging and my friends within for providing me with a wonderful experience here in the Netherlands.

I finally wish to thank my parents Benjamin J. Head and Kathryn F. Head for supporting me financially and emotionally throughout my stay in Delft.
Special Notes

This thesis is a master thesis topic and was issued by the TU delft, The Netherlands. It was chosen and completed as a bachelor thesis for qualifications awarded by the University of Queensland, Australia. It was conducted at a company called MTT within the city Eindhoven. Three months of work was conducted prior to the start of the bachelor thesis; Chapter 2.2.3 contains empirical size effect correlations adapted to experimental data which is further elaborated on in Appendix E.6. The author sincerely hopes you enjoy reading through the study. Enjoy!

Key Words: Microturbines, Hybrid Electric Vehicles, Capstone C30, Geometric Size and Weight, Cost, Fuel Analysis, Performance, Carpet Plot, Flat Rating, Power code, Effect Graphs.
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<th>Units</th>
<th>Description</th>
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<tr>
<td>C</td>
<td>m/s</td>
<td>Velocity</td>
</tr>
<tr>
<td>$C_f$</td>
<td>€/kWh</td>
<td>Cost of fuel (per kilowatt hour)</td>
</tr>
<tr>
<td>$c_p$</td>
<td>J/(kg·K)</td>
<td>Specific heat capacity at constant pressure</td>
</tr>
<tr>
<td>D</td>
<td>cm</td>
<td>Rotor Tip Diameter</td>
</tr>
<tr>
<td>$D_s$</td>
<td>-</td>
<td>Specific diameter</td>
</tr>
<tr>
<td>$f$</td>
<td>-</td>
<td>Fuel/Air ratio by weight or Fanning friction factor</td>
</tr>
<tr>
<td>$F_t$</td>
<td>kg/kW</td>
<td>The specific weight of turbomachinery</td>
</tr>
<tr>
<td>h</td>
<td>J/kg</td>
<td>Specific enthalpy</td>
</tr>
<tr>
<td>$\Delta H$</td>
<td>J</td>
<td>Enthalpy of reaction</td>
</tr>
<tr>
<td>H</td>
<td>m</td>
<td>Head</td>
</tr>
<tr>
<td>j</td>
<td></td>
<td>Colburn heat transfer factor</td>
</tr>
<tr>
<td>LHV</td>
<td>J/kg</td>
<td>Lower heating value (net caloric value)</td>
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<tr>
<td>$m$</td>
<td>kg/s</td>
<td>Mass flow</td>
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<tr>
<td>n</td>
<td>-</td>
<td>Polytropic Index</td>
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<tr>
<td>$n$</td>
<td>rpm</td>
<td>Rotational speed</td>
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<tr>
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<tr>
<td>P</td>
<td>Pa</td>
<td>Pressure</td>
</tr>
<tr>
<td>PR</td>
<td>-</td>
<td>Pressure Ratio</td>
</tr>
<tr>
<td>Symbol</td>
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<td>------</td>
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<tr>
<td>$P_s$</td>
<td>[ kW ]</td>
<td>Shaft power required</td>
</tr>
<tr>
<td>$\dot{P}W$</td>
<td>[ J/s or W ]</td>
<td>Power Output</td>
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<tr>
<td>$pw$</td>
<td>[ J/kg ]</td>
<td>Specific work output</td>
</tr>
<tr>
<td>$\dot{p}w$</td>
<td>[ (J/s)-kg ]</td>
<td>Specific power output</td>
</tr>
<tr>
<td>$q$</td>
<td>[ J/kg ]</td>
<td>Specific heat</td>
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<tr>
<td>SFC</td>
<td>Kg/kWh</td>
<td>Specific Fuel consumption</td>
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<tr>
<td>$T$</td>
<td>[ K ]</td>
<td>Absolute temperature</td>
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<td>$TR$</td>
<td>$\left[ \frac{T_s}{T_i} \right]$ or [-]</td>
<td>Temperature ratio</td>
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<tr>
<td>TRQ</td>
<td>[ N·m ]</td>
<td>Torque</td>
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<tr>
<td>$U$</td>
<td>[m/s]</td>
<td>Blade speed</td>
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<tr>
<td>V</td>
<td>[kg]</td>
<td>Recuperator matrix volume, Theoretical Spouting Velocity</td>
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<tr>
<td>$\dot{V}$</td>
<td>[g/s]</td>
<td>Air or gas mass flow rate</td>
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<td>[ Years ]</td>
<td>The years from 1958</td>
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**Greek Symbols**

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<tr>
<td>$\beta$</td>
<td>[ degrees ]</td>
<td>Angle of wave pattern, and total surface compactness</td>
</tr>
<tr>
<td>$\rho$</td>
<td>[ kg/m$^3$ ]</td>
<td>Density</td>
</tr>
<tr>
<td>$\eta$</td>
<td>[-]</td>
<td>Efficiency</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>[-]</td>
<td>Pressure Drop</td>
</tr>
<tr>
<td>$\Pi$</td>
<td>$\left[ \frac{P_2}{P_1} \right]$ or [-]</td>
<td>Pressure ratio</td>
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Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application

\( \gamma \)  
[ - ]  
Ratio of specific heats

\( \omega \)  
[ rad/sec ]  
Rotational velocity

\( \varepsilon \)  
[ - ]  
Recuperator effectiveness

\( \tau \)  
[ Nm ]  
Torque

\( \Delta \)  
[ - ]  
Change in

**Suffixes**

0  
Stagnation value

1, 2, 3…  
Reference planes

air  
Ambient, air

b  
Combustion chamber

c  
Compressor

E  
Electrical

e  
Exit

f  
Fuel

gg  
Gas

\( h_g \)  
Gas side

\( h_a \)  
Air side

h  
Heat-exchanger

i  
Intake, mixture constituent

m  
Model turbomachinery
<table>
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<td>Mechanical</td>
<td></td>
</tr>
<tr>
<td>N</td>
<td>Net</td>
<td></td>
</tr>
<tr>
<td>p</td>
<td>Prototype turbomachinery</td>
<td></td>
</tr>
<tr>
<td>Recup</td>
<td>Recuperator</td>
<td></td>
</tr>
<tr>
<td>ref</td>
<td>Reference</td>
<td></td>
</tr>
<tr>
<td>shaft</td>
<td>Shaft</td>
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</tr>
<tr>
<td>S</td>
<td>Stage</td>
<td></td>
</tr>
<tr>
<td>SC</td>
<td>Compressor specific</td>
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<td>ST</td>
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<td>th</td>
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<tr>
<td>t</td>
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**Abbreviations:**

*APU*  
Auxiliary Power Unit

*CV*  
Conventional Vehicle

*CHP*  
Combined Heat and Power

*DP*  
Design Point

*ETAdes*  
Thermal Efficiency at the design point

*FRT*  
Flat Rated Temperature

*GT*  
Gas Turbine
<table>
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<td>GSP</td>
<td>Gas turbine Simulation Program</td>
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<tr>
<td>HEV</td>
<td>Hybrid Electric Vehicle</td>
</tr>
<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
</tr>
<tr>
<td>ISA</td>
<td>International Standard Atmosphere</td>
</tr>
<tr>
<td>MGT</td>
<td>Micro gas turbine</td>
</tr>
<tr>
<td>MTT</td>
<td>Microturbine Technology</td>
</tr>
<tr>
<td>OD</td>
<td>Off design</td>
</tr>
<tr>
<td>PT</td>
<td>Power turbine</td>
</tr>
<tr>
<td>( PR_c )</td>
<td>Pressure Ratio (compressor)</td>
</tr>
<tr>
<td>PMG</td>
<td>Permanent Magnet Generator</td>
</tr>
<tr>
<td>SAE</td>
<td>Society of Automotive Engineers</td>
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<td>TET or EGT</td>
<td>Turbine Exit Temperature</td>
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<td>TIT</td>
<td>Turbine Inlet Temperature=Combustor Exhaust Temp</td>
</tr>
<tr>
<td>TNO</td>
<td>Toegepast Natuurwetenschappelijk Onderzoek</td>
</tr>
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<td>USD</td>
<td>United States Dollar</td>
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Chapter 1

Project Background and Goals

1.1 Mission Statement

The purpose of this Design and Simulation assignment is to try and find the optimum size and corresponding performance of a microturbine\(^1\) for its application as a range extender in a Hybrid Electric Vehicle (HEV).

1.2 Background – context, problem, scope

1.2.1 Project Description and Formal Statement of Work

The microturbine project, as an electric vehicle range extender, is part of a larger initiative concerning range extender innovations. AgentschapNL, a subsidiary of the Ministry of Economic Affairs, has awarded the REI project (Range Extender Innovations) with a HTAS EVT subsidy, Appendix F. The purpose is to develop three different Range Extender Technologies for three separate market segments. Formed from a conservatorium of institutions including Peec Power (leader), TNO, TU/e, TUDelft, Fontys SME, Micro Turbine Technologie (MTT), All Green Vehicles (AGV) and ProDrive; all hope to improve E-vehicle range and therefore easier consumer acceptance of the electric driving. Within the REI consortium, MTT focuses on the development of a range extender for small electric cars, together with TNO, TU/e, Prodrive and All Green Vehicles BV.

\(^1\) Machines whose power outputs are in the range between 30kW and 250 kW, however the definition is flexible.
As the maximum output power and corresponding mass flow rate is decreased, turbomachinery component sizes diminish resulting in lower component efficiency. A project whereby a series of design sweeps from 9-36 kW in set intervals are made in order to try and find the optimum performance conditions conforming to the HEV constraints. In order to carry this out correctly, cycle and component parameters have to be scaled accordingly. A preliminary design can be drafted simply by scaling from an existing reference using the appropriate non-dimensional parameters. Precise prediction of efficiencies at the initial design stage, or at any other stage, is difficult, and the designer usually relies upon empirical loss models and correlations. The predicted efficiencies are then as good as the underlying empirical loss models.

If the designer really wants to generate a model with significant improvements in efficiency and/or size then a fundamental aerodynamic design must be undertaken of the impeller and other essential parts. Efficiency is maximized when the irreversibilities associated with the flow process are minimized, (Whitfield 1990; Whitfield and Baines 1990). This is of course costly and impractical for preliminary design analysis.

To achieve preliminary design goals in this project it is necessary to look at previous designs and take them as a reference point during further study. Proven technology such as the 30 kW Capstone turbogenerator (Capstone C30) is used as a reference point for the project’s preliminary design.

*Engine temperature* (or maybe time since last shut down) for thermal effects and *engine conditions* (deterioration) may also be taken into account for model improvement. However due to the time constraints of the project this may not be possible. The technology level and related costs will need to be varied and are related by relations of component efficiencies which are a function of technology/cost level parameters, (Walsh and Fletcher 2004).
1.2.2 MTT (Microturbine Technology)

Microturbine technology (MTT) develops microturbine systems with predominant focus on two businesses:

- Micro-cogeneration (micro CHP); and
- Combined Auxiliary Power Unit (APU)/parking heater (CAP)

Their microturbine configurations follow an unconventional approach to try to limit issues related to scale effects. The complete turbine package includes a centrifugal compressor and a reaction turbine mounted on a single monolithic rotor. A rotating combustion chamber, and finally a recuperator is integrated to increase efficiency and reduce the specific fuel consumption (SFC). For the 3 kW unit a 60-80mm diameter rotor is used, which, depending on the rotational speed and other factors the rotor could be reduced.

![Figure 1-1 Capstone C30 Microturbine-taken from (Capstone 2009)](image)

The current configuration and design of MTT’s unconventional microturbine is not suitable for the application as a range extender in a HEV. Thermal efficiencies close to those of conventional
engines and power outputs around 15-30 kW are required and so a different design configuration is necessary.

The microturbine system includes the compressor, turbine, shaft, bearings, housing, seals, fuel canisters, fuel delivery, choked flow nozzle and all hardware required for the housing to remain closed. To a limited degree, it also includes the controls sensors, power converters, signal modifiers, and the generator. Figure 1-1 shows primary component locations.

1.2.2.1 Gas Turbine Simulation Program (GSP)

The Gas Turbine Simulation Program (GSP), Figure 1-2, is a tool for gas turbine engine performance analysis, which has been developed by the NLR the Netherlands. It is a component based modeling environment and is NLR’s primary tool for gas turbine engine performance analysis. Its component-by-component approach enables modeling of virtually any type of system. This program enables both steady state and transient simulations for any kind of gas turbine configuration. GSP gives the possibility to perform off-design simulations, which allows analyzing the performance of a “physically-based” component. The simulation is based on one-dimensional modeling of the processes in the different gas turbine components with thermodynamic relations and steady-state characteristics (component maps). GSP can be used to calculate gas temperatures, pressures, velocities, and composition at relevant engine stations from measured engine data, (Visser and Broomhead 2000).

A gas turbine model is created within GSP by arranging different predefined components (inlet, compressors, turbines, combustors, exhaust nozzles and recuperators). The GSP output is used
for further processing by the CFD and MARC finite element models, (Visser, Kogenhop et al., 2011). To aid this study, various performance models of the microturbine have been designed using the NLR/Delft University Gas turbine Simulation Program (GSP). This software has been used because of its capability to model heat transfer among gas turbine components and the environment. The GSP model has been used to generate performance data of a microturbine with power outputs of 9-36 kW.

1.2.3 Scope

1.2.3.1 Goals

The designer is assigned to complete the following goals:

- Adaption and implementation of empirical loss models for the reference Capstone C30. (The research on correlations associated with sizing effects in decreasing turbomachinery was done in an earlier study, Appendix E). The influences associated with variation in geometric size, weight, cost and efficiency are closely evaluated.
- Conceptual/ Cycle optimization study using GSP- keeping in mind vehicle design constraints.
- Definition of some sort of performance representation format so data can be transferred to the vehicle model.
- Off-design (steady-state or transient if necessary) simulations to provide data to the vehicle model; and
- Analysis of possible different design cases.

1.2.3.2 Key Business Goals

A successful project is defined by the identification of the optimum point of the microturbine in accordance with the specifications of the HEV. The model must be appropriately sized conforming to weight and size restrictions. The results of this project serve as a foundation for further development in this field and extending the range extender innovations project. The success of this project will bring future research a step closer to replacing/providing a power efficient substitute for the power source in HEVs.
1.2.3.3 Boundaries and interactions

The microturbine range extender project does not directly influence or interact with the other range extender options; at least within the defined lifecycle of the thesis. It is plausible that the results can be used as a comparative measure between the other options during a later period. Data from all systems are assessed and choices made on what options warrant a continuation and use of government resources.

1.2.3.4 Limitations

During the design and optimization process it was noted that cost and component selection would become a constraint. MTT takes components off the shelf from automotive turbochargers, adapts the configurations and tries to optimize the performance. Since everything is on the micro scale, the complete package will be light and compact. Research within the microturbine area will aid any electrical required applications limited by space and weight, (Gong, Driscoll et al. 2006).

1.2.4 Stakeholders

The primary stakeholders of the range extender innovations project are the conservatorium of companies previously listed. MTT and TNO are the most important primary stakeholders in the scope of the project. Secondary stakeholders include the outside venders sought for the manufacture of intricate parts (Turbocharger suppliers). Stakeholders extend to members/researchers of current and future design teams at MTT/TNO and other institutions involved in micro turbine research in this application.
1.3 Chapter Summaries

This section is an outline of the work that was conducted in each chapter.

Chapter 1 Project Background and Goals

This contains the introduction to the design assignment. The background material needed to define the scope and outline the problem at hand is discussed.

Chapter 2 Literature Review

This chapter provides an overview on microturbine technology and an introduction to scaling effects and geometric size weight and volume of small radial turbomachinery. The focus is directed towards HEV application and NLR’s GSP will be frequently used as a medium of explanation.

Part 1 briefly describes the history, configuration, economics and design of microturbines in today’s industry. This will be specifically related HEV application and Capstones C30 microturbine will be used as a reference due to its success as a marketed unit in both CHP and HEV areas.

Part 2 focuses on the size, weight and cost effects encountered with turbomachinery. The concept of scaling machinery is discussed first then the specific effects on turbine components. Correlations are proposed which were developed in Appendix E. The complete defined empirical relations will be adapted and formulated in the context of the problem. An overview of the design method relating specific speed and specific diameter is given in Appendix B.6.

Chapter 3 Conceptual/Cycle Optimization Study

This chapter involves the initial design point study analysis and construction of the basic model structure that GSP will use to numerically calculate the points that is used for off-design analysis. Carpet plots and a fuel analysis will define the appropriate inputs for fuel choice, pressure ratio, components efficiencies and other parameters needed for each scale.

Chapter 4 Off-design: Steady-state simulations

This chapter moves on to the off-design analysis of the configuration. A definition of the microturbine performance data format or data link method will be defined in order to supply the data to the Hybrid Electric Vehicle model. Off-design phase used to provide ($\dot{w}_f$, SPF, Power etc) by changing power code and ambient operating conditions.
Chapter 5 Conclusions
This chapter provides the reader with a summary of the findings.

1.4 Appendices

Appendix A
Appendix A contains the design conditions and published performance data of the Capstone C30.

Appendix B
Contains the analytical development and explanation of the equations GSP utilizes to generate the performance data. Equations from this section will be commonly used within GSP. It also contains a design method (Balje 1981) for turbomachinery involving specific diameter, specific speed and efficiency.

Appendix C
The raw off-design data for the 30kW is placed in this section for the required parameters.

Appendix D
The Aerospace Standard 755 by the SAE (Society of Automotive Engineers) for gas turbine engine station designation numbers.

Appendix E
Appendix E addresses the issue of applying the empirical correlations and fitting them to experimental data. This provides a more accurate representation of the scaling effects/penalties experienced with the technology level in the last 30 years. They will be adapted to allow for future tech improvement. The author researched and developed a series of empirical relationships that compensated for the scale effects (if present) during the decreasing scale. These empirical relationships need to be adapted and implemented into the GSP software environment.

Appendix F
The Range Extender Innovations project outlines the conservatorium and the purpose of the microturbine for low weight vehicles. A brief overview of current microturbine use in the automotive sector is described.

Appendix G
A Paper submission to the ASME Turbo Expo of 2012 resulting from the work done during the thesis.
Chapter 2

Literature Review

Chapter two is divided into two sections; Part 1 focuses on the history and development conducted by others in the field and Part 2 focuses on work done by the author. This complete chapter is meant to give the reader a basic introduction to microturbine technology and an introduction to scaling effects, geometric size, weight and cost concerning small turbomachinery. The focus will be directed towards HEV application and NLR’s GSP will be frequently used as a medium of explanation. Capstones C30 will be used as a reference due to its success as a marketed unit in both CHP and HEV areas.

2.1 PART 1: Microturbine Development and History in Industry

2.1.1 Introduction

Sustainable development is crucial to preserving the environment and prolonging current resources, so that the needs of future generations can be satisfied. Current patterns of most industrial development are still unsustainable. Industrial processes play a major role in the degradation of the global environment and they continue to severely abuse resources. The utilization of cleaner fuels and development of more efficient or alternative energy systems is desperately needed. Simple or complicated configurations of gas turbine systems are promising candidates for further focus and development. Gas turbines are well known for their high power to weight ratio, extreme reliability and low maintenance. Their primary application is focused around the aircraft industry and are currently being used for propulsion and onboard APUs for generating electricity. They have also found application in the industrial sector and to a lesser
extent in the marine industry, (Kolanowski 2004). They pose great potential for distributed power generation in applications where heat and power generation can be combined. However, another application with great potential has risen over the last two decades.

The petroleum-based transportation infrastructure has severely impacted the environment and so there has been continued interest in developing an electric transportation infrastructure. Many automobile companies developed prototype gas turbines for use in the transport sector during the nineteen sixties. Rover being the first, Chrysler, General Motors and Allison built and tested various models. Ford, Volkswagen, BMW, Toyota, Nissan and many others have experimented with gas turbine technology in the transport sector. All with primary focus being on large heavy duty vehicles such as trucks and busses. All were unable to produce a commercially viable and reliable unit that could penetrate the market. NoMac Energy was the first company with the primary goal of developing and producing small gas turbines for their use in the automotive industry. The company was formed in 1988 and evolved into Capstone Turbine Company, commercially marketing microturbine units in late 1998 with success, (Kolanowski 2004). The technology used in microturbines is derived from aircraft APUs, diesel engine turbochargers and small jet engines, (Energy 2010).

Current target customers include financial services, data processing, telecommunications, restaurant, lodging, retail, office building, and other commercial sectors. Microturbines are currently operating in resource recovery operations at oil and gas production fields, wellheads, coal mines, and landfill operations, where byproduct gases serve as essentially free fuel, (TechPro 2002). Small production quantities mean high prices compared to the conventional reciprocating engine. This has limited market penetration. Current microturbines are less efficient than reciprocating engines with an efficiency of around 30% vs. 40% respectively. The long-term goal to achieve an efficiency of 40% can only be realized based on the utilization of ceramic hot end components2, (McDonald 2003).

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2 Hot end components refer to the parts of the microturbine system that are subjected to the highest temperatures in the cycle. Eg. turbine, combustor and recuperator (hot gas side).
2.1.2 Primary Configurations of Microturbines

Turbines and microturbines are both classified by the physical arrangement of the component parts. Whereby either may utilize single or dual shaft systems, simple or recuperated cycles, with inter-cooled and/or reheat additions. The ‘single shaft recuperated microturbine’ generally rotates with speeds of 90,000 to 120,000 and is generally the more common design class. The second class of microturbine; the single shaft simple cycle, is also a successfully marketed unit. The single-shaft simple cycle can be an attractive design because it is simpler and less expensive to build. Simple and recuperated units together with their current manufactures will be discussed below, (Kolanowski 2004).

The micro-generators will be divided into two general classes:

- **Simple cycle microturbines (no recuperator):** These units are a less common class of microturbine due to their low thermal efficiencies compared to recuperated units. These microturbines operate on the simple Brayton cycle as with normal gas turbines. Outside air is compressed, mixed and burnt with fuel under constant pressure conditions (ideally). The turbine extracts work from the hot exhaust gases, thereby generating shaft power. Their typical efficiencies are around 15%, however they have lower capital costs, more heat available for cogeneration applications and higher reliability compared to recuperated units. They also have the possibility of matching the recuperated microturbines in efficiency if a large temperature and pressure ratio (TR & PR) could be obtained, (Capehart 2010).

- **Recuperated Microturbines:** These units are the current successfully marketed class and Capstone has released two different units which are currently used in HEV applications today. The C-30 and C-65 units (Appendix A) are used in the automotive industry and are succeeding in their goal to reduce emissions and savings in fuel. In a recuperated microturbine, a radial compressor compresses the inlet air that is then preheated in a sheet metal heat exchanger, which recovers heat from the turbine exhaust that would otherwise be wasted energy. This heat energy is then used to raise the temperature, velocity and volume of the gas of the incoming air stream into the combustor. The shaft power
generated from expanding the flow of hot gases through the turbine is used to drive the
compressor and rotate the electric generator. The electric generator converts this
mechanical power to electric power with an obvious small conversion loss. The frequency
of this electric power can be modified as per the requirements using power electronics
systems. The remaining wasted heat energy from the exhaust can be passed through a
waste-heat recovery device for cogeneration applications, eg. Space heating or water
heating in the HEV, (Energy 2010). The mechanical efficiencies are in the range of 28-
32% and save 30 to 40% in fuel use. Future ambitions are directed towards increasing the
efficiency to 40%, (Capehart 2010).

The low emissions, high efficiency and the capital costs are the important factors in the design of
these units. In microturbines, the rotor generally turns at high rotational speed, about 96,000 rpm
in the case of a 30 kW machine and about 85,000 rpm in a 65 kW machine. The specific turbine
and compressor design characteristics influence the physical size of the components and
consequently the rotational speed. For every design, as the power rating decreases, the shaft
speed increases, hence the high shaft speed of the small microturbines, (Capehart 2010).

2.1.3 C30 Microturbine Configuration

Automobile, truck, and other small reciprocating engine turbochargers are quite similar to small
turbines available today. Superchargers and turbochargers have been used for almost 80 years to
increase the power of reciprocating engines by compressing the inlet air to the engine. Current
small gas turbines, of the size and power rating of microturbines, serve as auxiliary power
systems on airplanes. In fact both engines come from the same design family of compressors and
turbines. The decades of experience with these applications provide the basis for the engineering
and manufacturing technology of microturbine components. Microturbines are more complex
than conventional simple-cycle gas turbines because of their dimensions. The current
successfully marketed microturbine engine meets demanding cost goals due to its simplicity. It
includes the following: (1) single stage radial compressor, (2) single stage uncooled metallic
radial inflow turbine, (3) high speed rotor supported on hydrodynamic air bearings, (4) direct-
drive high speed air-cooled generator, (5) multi-fuel annular combustor (conventional or
catalytic), (6) a simple control system, and (7) a compact stainless steel wrap-around annular primary surface recuperator.

*Figure 2-1 Capstone Microturbine C30 taken from ([Capstone 2009](#))*

The unit can be scaled up (perhaps down) depending on the actual power rating required. The Capstone C30 unit is shown in Figure 2-1. The Capstone C-30 microturbine and the positioning of all the essential components within are shown in Figure 2-2 and Figure 2-3.

*Figure 2-2 General cycle process, capstone microturbines adapted from ([Capehart, 2010](#))*

The single rotor is supported by two air foil bearings which are not required to have a liquid lubrication system. Microturbines operate on either oil-lubricated or air bearings, which support
the shaft(s). Uses of Oil bearings or Air Bearings have different attributes concerning safety, cost, maintenance and efficiency. All Capstone’s microturbines for HEV applications use air bearings, (TechPro 2002). All the dynamic parts of the microturbine are mounted on the rotor and include the permanent magnetic generator on the left, the (centrifugal) Rotary Flow Compressor wheel in the center and the turbine wheel on the right, shown in Figure 2-3 and Figure 2-4. The single moving part of the one-shaft design has the potential for reducing maintenance needs and enhancing overall reliability. Two-shaft designs also exist, in which the turbine on the first shaft directly drives the compressor while a power turbine (free turbine) on a second shaft drives a gearbox and a conventional electrical generator producing 50-60 Hz power. The two-shaft design features more moving parts but does not require complicated power electronics to convert high frequency AC power output to 50-60 Hz. A majority of microturbines are single-shaft units and use a high speed permanent magnet generator because of its simplicity. The generator produces variable voltage, variable frequency AC power and so an inverter is employed to produce 50 or 60 Hz AC power, (Kolanowski 2004).

The fuel is injected into the combustor via an injector and mixes with the high-pressure air (air supplied by the compressor). Combustion takes place and the temperature of the gases is increased, (Saravanamutto, Rogers et al. 1996). Annular and cannular combustion chambers are designs of chambers that are appropriate for microturbine engines because of their compactness
and simplicity of design, (Kolanowski 2004). The recuperator is a *heat exchanger* that preheats the compressor discharge air by using waste heat from the engine exhaust. The heat exchanger is the biggest physical component of the microturbine system and is subject to varying thermal stresses that reduces performance and respectively shorten its lifespan. The recuperated microturbine improves overall efficiency but influences the unit’s durability, size and cost. This represents about 30 percent of the overall engine cost. Significant advancements in recuperator technology have been made in recent years and annular primary surface types of units that are amenable to high volume production have been demonstrated and have seen service in various microturbines, (McDonald 2003; McDonald and Rodgers 2007). Capstone incorporates the basic counter flow design. It is also essential that a recuperator be *compact*, especially if space/weight restrictions, such as with automobiles are of a concern, (Shah 2005).

Figure 2-4 and Figure 2-5 show the location of the *generator* within the microturbine. The generator is the component that produces the electrical power needed for use in a HEV. The generator is either attached to a single shaft turbo-compressor or is on a separate shaft turning with a power turbine.

Digital *power controllers* are essential on single-shaft microturbines. They are used to convert the high frequency AC power produced by the generator (due to the high rotational speeds) into usable electricity.

Microturbine technological development is a complicated area since one cannot just simply scale down high performance large gas turbines. There are problems associated with large changes in Reynolds number, large heat transfer between hot and cold components, and geometrical restrictions and complications related to material and manufacturing of militarized components, (RTO/NATO 2005).
Most microturbine units are currently designed for continuous-duty operation and are recuperated to obtain higher thermal efficiencies. The microturbine challenge is to minimize the losses that become noticeable due to decreasing the physical size. These losses arise due to: increased heat transfer, the Reynolds number effect and manufacturing tolerances. Heat transfer results in more work needed by the compressor and less work delivered by the turbine. The Reynolds number effect and manufacturing tolerance reduce the component efficiencies, (TechPro 2002).

A simplistic exploded view of the turbogenerator is shown in Figure 2-5. Detailed studies by manufacturing engineers are required to determine the optimum type of fabrication (ie. castings, forgings, sheet metal pressings, extrusions, rolled and welded structures etc.) for each of the
components. A compressor impeller with a diameter of 117.5 mm (4.7”) would likely be cast from an aluminum alloy assuming the PR to be less than 4. The cast, non-internally cooled turbine impeller would be made from a high temperature alloy and is welded to the shaft. The compressor impeller would be shrunk fit to the steel drive shaft, (Rodgers and McDonald 2007). The components such as the electric generator, power conditioning systems, bearings, nozzles, fuel filtration, metering and injection systems have not been analysed in this study. It is possible to add more components to the microturbine but depends on the application at hand. A more detailed elaboration on the above microturbine components together with scale effects can be found in Appendix E.

2.1.4 Advantages/Disadvantages

The advantages and disadvantages of microturbines are basically the same as their larger counterparts but with a few more specific concerns due to the physical size, (CHRYSLER 1979).

2.1.4.1 Advantages of Microturbines

- Very high power-to-weight ratio, compared to reciprocating engines.
- Microturbines are smaller than their reciprocating counterparts of the same power rating.
- Far less noise and vibration than a reciprocating engine.
- Single or very few moving parts compared to reciprocating engines.
- Reduced Maintenance.
- High operation speeds.
- Low lubricating oil cost and consumption; and
- Potential use for a large number of applications.

2.1.4.2 Disadvantages of Microturbines

- High initial (capital) costs due to small production quantities.
- Thermal efficiency is below that of reciprocating engines, 28-32% vs 40%.
- Longer startup than reciprocating engines. (Not an issue when incorporated with HEV Technology); and
• Less responsive to changes in power demand compared to reciprocating engines.

2.1.5 Microturbine Parameters and Issues

The key problems/issues that are associated with microturbines are:
• Large heat fluxes, due to the extreme temperature differences and small dimensions which has an unfavourable impact on flow and thermodynamic performance.
• Air bearings are needed because they can operate at very high RPMs and withstand the thermal stresses encountered in microturbine systems.
• The compact heat exchangers, for maximizing the cycle (thermal) efficiency and minimizing the exhaust temperature.
• The special small combustion chamber designs and the use of non-conventional fuels, together with gasification technology are required to cope with the short residence time.
• The unconventional materials and appropriate manufacturing techniques, that can resist the very high metal temperatures (1200 to 1500 K) resulting from the absence of any turbine cooling due to the small dimensions; and
• Newly developed electrical generators and motors capable of operating at the very high RPM and the related power control.

The above key issues need to be addressed and progressed on before portable gas turbines become a competitive technology in the transport market, (RTO/NATO 2005).

The maximum cycle temperature is limited by metallurgical considerations. Efficiency improvements can be obtained in larger gas turbines due to the use of ceramic coatings or integral cooling of turbine blades, which allow the increase of the maximum operating temperature, (Saravanamuttoo, Rogers et al. 1996). These advanced design techniques however, cannot currently be used for microturbine technology (at least haven’t been successfully marketed). The very small blades which are utilized in the microturbines today cannot be cooled for reasons of manufacturing complexity and cost, (Energy 2010). The focus of improving gas turbine technology is centered on combinations of higher temperatures and finding the corresponding optimum compressor pressure ratios. Equations B-41 and B-42 give an analytical representation of the efficiency and power. Figure 3-5 provides a graphical representation, which was created
numerically using GSP. Higher values of TIT, $PR_c^3$ and effectiveness can be seen to increase the thermal efficiency and power. The optimum pressure ratio increases as technological advances permit higher inlet temperatures, which are desired developments due to resulting higher efficiencies and specific power, refer to Figure 3-5. Current microturbine inlet temperatures are generally quite low to enable the use of relatively inexpensive materials for the turbine wheel, and to maintain pressure ratios at 3.5 to 4.0, (Kolanowski 2004).

2.1.6 Market for Microturbines

Due to the compact size, low operational and maintenance costs, automatic electronic control microturbines are the ideal engine for hybrid vehicle applications. To date their high capital costs and low comparative thermal efficiency to reciprocating engines have limited their use to:

- Public service vehicles (buses).
- Heavy commercial trucks (10 tonne payload).
- Light rail systems where public utility supply is unreliable; and
- Shunting locomotives.

(Limited 2000)

2.1.7 Emissions and Pollution

Air pollution is a global issue that has an enormous influence on global environmental problems and heath issues. Emissions are strictly regulated and guidelines are set as to the amount that can be released. This differs in every state and country but is a very important matter in regards to environmental welfare. The transportation sector contributes to at least 35% of air pollution today and so alternatives to reduce the amount of toxins need to be implemented, (Limited 2000).

Microturbines have the potential for extremely low emissions. All microturbines operating on gaseous fuels feature lean premixed combustor technology, which was developed relatively

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3 Increasing the $PR_c$ and holding TIT constant will in fact increase both the thermal efficiency and power up until the optimum $PR_c$ is found. From this point thermal efficiency will decrease, Figure 3-5. Follow the green Pressure Ratio line for optimal conditions.

4 Small railroad locomotive intended not for moving trains over long distances but rather for assembling trains ready for a road locomotive to take over, also known in the US as switching (UK and Australia: shunting).
Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application

recently in the history of gas turbines and is not universally featured on larger gas turbines. Microturbines are able to meet emissions requirements with this built-in technology and as a result post-combustion emission control techniques are not needed.

The primary pollutants from microturbines are oxides of nitrogen (NO\textsubscript{x}), carbon monoxide (CO), and unburned hydrocarbons. Microturbines are designed to limit emissions at full load; depending on the particular pollutant emissions are often higher when operating at part load (or closer to idle\textsuperscript{5} mode). The rate of thermal NO\textsubscript{x} formation increases rapidly with flame temperature and is a mixture of mostly NO and NO\textsubscript{2} in variable composition. CO and unburned hydrocarbons both result from incomplete combustion which tends to be higher at idle. CO emissions are also heavily dependent on operating load. For example, a unit operating under low loads will tend to have incomplete combustion, which will increase the formation of CO.

Emissions of carbon dioxide (CO\textsubscript{2}) are of concern due to its contribution to global warming. The amount of (CO\textsubscript{2}) emitted is a function of both fuel carbon content and system efficiency, (TechPro 2002).

2.1.8 Economics of Microturbines

Microturbine capital costs in 2002 ranged from $700-$1,100/kW USD. These costs include all hardware, associated manuals, software, and initial training. Adding heat recovery increases the cost by $75-$350/kW. Installation costs vary significantly by location but generally add 30-50% to the total installed cost, (DER 2002). It is assumed that costs would have decreased due to technological development and Capstone’s increasing unit production volume in the market. Microturbine manufactures hope the units can provide higher reliability than conventional reciprocating generating technologies because of fewer moving parts. Initial units will naturally require more unexpected visits due to the prematurity of the technology. Capstone ensures that a once-a-year maintenance schedule should be sufficient. Most manufacturers are targeting maintenance intervals of 5,000-8,000 hours. Due to limited market penetration, maintenance costs for microturbine units are still based on forecasts with minimal real-life situations. Current

\textsuperscript{5} Idle (idling) is a term generally referring to “in operation” but not producing any work. So in the case of a turbine, the shaft is spinning only to maintain operation not producing any work. This is unwanted because certain emissions are at max output.
estimates range from $0.005-$0.016 per kWh, which would be comparable to that for small reciprocating engine systems, (DER 2002).

Table 2-1 Microturbine cost overview

<table>
<thead>
<tr>
<th>Microturbine Cost</th>
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<tr>
<td>Capital Cost</td>
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<td>$700-$1,100/kW</td>
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<tr>
<td>O&amp;M Cost</td>
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<tr>
<td>$0.005-0.016/kW</td>
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<tr>
<td>Maintenance Interval</td>
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<td>5,000-8,000 hrs</td>
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Microturbine manufacturers are targeting a future costs to fall below $650/kW. This appears to be feasible if the market expands and sales volumes increase, (DER 2002).

Continued debate continues over the design and layout of the engine; single or multi-shaft design, simple or recuperated cycles and the optimum utilization and balance of performance parameters. One must conduct thermodynamic design point studies in the initial designing stages. Known component efficiencies, variable fluid properties, fuel variants, air-bleeds, and pressure losses of the components need to be calculated over a range of pressure ratios. Using computational assistance, values for the power output and associated fuel consumption will be calculated for the above cycle parameters.

2.1.9 Hybrid Electric Vehicle

2.1.9.1 Introduction

Certain drive trains and lines are needed within the HEV when a microturbine is implemented as the range extender. The Conventional Vehicle (CV) has been adapted, modified and incorporated with many more newly introduced components which are all necessary to satisfy the development of a vehicle with an electric system. An electric vehicle (EV) uses one or more electrical motors and its corresponding electrical system for propulsion. The broad category of EVs include;

- Electric cars (from lightweight to heavy duty trucks).
- Electric trains.
- Electric lorries.
- Electric boats.
• Electric motorcycles and scooters; and
• Electric Spacecraft.

The area of focus will be directed towards Lightweight Series Hybrid Electric Cars, although much of the discussion is relevant to automobiles in general. An interesting area of development is the investigation of Plug-In Hybrid Electric Vehicles. A schematical view of the components in a HEV is shown in Figure 2-6.

![Figure 2-6 Complete Series Hybrid Electric System taken from (DER, 2002)](image)

2.1.9.2 Advantages/Disadvantages

EVs are compared to assess their potential viability in the market.

2.1.9.2.1 Advantages of EVs

• Increased fuel efficiency.
• Emit fewer pollutants, use alternative fuels and less amounts compared to conventional vehicles.
• Potential for lower maintenance costs, especially in remote applications (depending on the technology).
• Potential use of waste heat.
• Regenerative braking.
• Suspension.
• Reduce dependence on foreign oil.
• Cheaper to run; and
• Quieter than conventional vehicles.

2.1.9.2.2 Disadvantages of EVs

• High capital costs.
• Expensive batteries that wear out too fast.
• Create hazardous waste (used batteries may not be completely recyclable); and
• Electric Vehicles have small limited range, Hybrids do not. (DER 2002)

2.1.9.3 EV Components

There are several areas of focus concerning the issue of EV technology, if it is to become economically feasible then attention must be directed to

• Fuel Converters.
• Drives.
• Auxiliaries; and
• Energy storage.

The following contents will focus on drive and energy storage, which are important topics associated to microturbine technology, (Passier, Conte et al. 2007).

2.1.9.4 Drives

Drive technology for electric vehicles—including drivetrain/ driveline configurations and the corresponding electrical devices that make up the electrical machine is an important issue for discussion.

2.1.9.4.1 Hybrid drive trains

Electric vehicles (EVs) are categorized into three areas:

• **Battery electric vehicles (BEVs or EVs),** are only powered by large rechargeable batteries. The battery powers an electric motor and this drives the vehicle. The battery is recharged as if it were a house hold appliance via the mains electricity supply.

• **Hybrid electric vehicles (HEVs),** are powered by a combination of electricity stored in a battery and a heat engine, the heat engine itself is referred to as a ‘range extender’. A
hybrid vehicle does not need to be plugged in to recharge its battery, as this is recharged automatically as the vehicle is being driven.

- **Plug-in hybrid electric vehicles (PHEVs)** work similarly to conventional hybrids. PHEVs have much larger batteries than conventional HEVs and so can also be charged from the mains when not in use. The term ‘plug-in’ has been adopted and this means the vehicle has another option for energy sources.

EV designs have a limited driving range because of the low energy density of the batteries. They also have long recharge times compared to refueling a fuel tank and are why HEVs are more popular on the market, (SEI 2007).

2.1.9.4.2 Hybrid Electric Vehicle drivelines

Different drivelines exist for HEVs and the three configurations are series, parallel and powersplit. Each one has its disadvantages and advantages especially with the choice of Internal Combustible Engine (ICE) for HEV application.

![Figure 2-7 Schematic of a series hybrid powertrain taken from (Passier, Conte et al. 2007)](image)

**Series Hybrids**, which is shown in Figure 2-7, the ICE (microturbine) turns a generator, which sends power to the batteries then to the motor which drives the wheels. While the microturbine is the ICE it will not directly power the vehicle. Capstone Microturbines are well suited for use in series hybrid electric drive systems, (SEI 2007).

**Parallel Hybrids** have the ability to transmit power to their wheels from two sources via an ICE and a battery powered electric drive. Due to microturbine as the ICE and the parallel
configurations complexity it is not used but this drivetrain may be attractive for use with other configurations.

**Series-Parallel Hybrids** can operate in either series or parallel mode. Can be referred to as “series-parallel with power split” which is used by Ford, Lexus, Nissan and Toyota, whereby series and parallel mode operate at the same time. The plug-in powertrains commonly but not always use this configuration.

Figure 2-8 shows how a series PHEV’s internal system works. The battery storage system powers the motor through an electric drive system. This system controls the speed and power levels of the motor. The electric motor drives the wheels where there may be regenerative braking coupled to the axial system. The battery storage system draws its energy from different sources.

- Regenerative braking is where the electric drive motor acts as a generator to take power from the wheels back to the battery a reason why hybrid systems achieve such good efficiency. This can be used to reduce the wear on brake systems (and consequent brake pad dust) and reduce the total energy requirement of a trip. Regenerative braking is especially effective for start-and-stop city use.

- The battery system can be recharged from electric plug-in stations either at home or at public recharging stations. The only problem with this idea is the infrastructure. It simply isn’t developed enough. The battery is also utilised for starting the gas turbine engine during the driving cycle.

- The most important source, is the microturbine (heat engine), it generates electricity to recharge the batteries even while the vehicle is in use. This significantly increases the range, (hence the coined term ‘range extender’), that the vehicle can travel compared to only relying on the battery energy storage.

![Figure 2-8 Schematic of a Plug-in Series Hybrid Electric Drive System complements (Capstone Corporation)](image-url)
2.1.9.5 Energy storage

In addition to the battery in the series driveline, the electricity storage device may be supplemented by supercapacitor packs, a flywheel, or combinations of all three. While not yet generally used in series HEVs, they offer many potential areas for improvement. Another form of chemical to electrical conversion is fuel cells, which are used and are projected for future use.

2.1.9.5.1 Fuel Cells and Batteries

Fuel Cells and Batteries are often confused to mean the same thing. A fuel cell is similar to a battery in that an electro-chemical reaction is used to create electric current. A battery has all of its chemicals stored inside, a limited supply of internal fuel, and it converts those chemicals into electricity. A battery eventually “runs flat” and you then discard or recharge it. A fuel cell has similar chemical reactions however chemicals constantly flow into the cell and combine to form a catalytic reaction. If there is a flow of chemicals into the fuel-cell, electricity will flow out of the cell. Most fuel cells in use today use hydrogen and oxygen as the chemicals. *Currently* it is Li-Iron Batteries used as the energy storage device in series hybrid electric cars with microturbine technology. The fuel cell has great potential to work in conjunction with batteries or by itself in a different driveline configuration, *(Boles 2006)*. Fuel cells have been proven to be very clean and efficient power-generation technologies. However, the high capital costs and operational problems associated with fuel cells have limited commercial and industrial applications. Lithium-iron batteries significantly improve fuel economy and have a higher energy-to-weight ratio; they are an ideal choice for the energy storage package on a series HEV, however they do cost more to produce and raise safety concerns due to the high operating temperatures, *(Passier, Conte et al. 2007)*.

2.1.9.6 Current Series Hybrid Electric Vehicle

The HEV has space and durability requirements and the original microturbine was designed to meet these specifications. There are a variety of power units for hybrid electric vehicles (HEV) and each one has its advantages. The selection criteria will depend on a variety of factors including capital costs, emissions, total efficiency, configuration of the hybrid system and the model of vehicle being used. The intelligent combination of the battery (Energy Storage Unit)
Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application

Microturbine technology (Hybrid Power Unit) and advanced power train design seem to be a promising solution for HEVs, (Limited 2000).

2.1.9.6.1 Microturbine/Li-Ion Battery System

There are many benefits of hybrid microturbine battery systems, and is what has recently driven the technology toward potentially more practical and cost-effective solutions that will help to develop new markets. A microturbine has a smaller volume and weight but also a lower efficiency (about 30%) than a "normal" gas turbine. Therefore, a microturbine working on its own in a vehicle is not so beneficial, not to mention it would require a massive gear reduction. A battery provides clean energy storage/supply and has a considerably higher and constant efficiency even at different operating temperatures, but its volume is still extremely large, (Bohn 2005). The combined efficiency of battery, microturbine and HEV can be as high as 65 per cent, (Watts 1998). The microturbine acts as an on-board battery charger in a series driveline configuration, and is known as the “range extender”.

![Capstone microturbine complete engine package](Calmotors 2010)

It is known that turbines tend to use more fuel when they are idling, and they prefer a constant rather than a fluctuating load. Which has been a major problem faced with placing one under the hood of a car, (Kolanowski 2004). This design utilizes a unique combination of battery units,
microturbine technology and powertrain to achieve a highly efficient cycle in a small, compact market-driven size. Figure 2-9 outlines the components of Capstone’s Microturbine package minus the energy storage device because of the wide selection discussed. Appendix A.6 outlines a breakdown of weights and dimensions for the package. Figure 2-10 and Figure 2-11 show the reader the positioning of the microturbine package in a lightweight vehicle and a truck.

![Microturbine and battery schematic](image1)

**Figure 2-10** Microturbine and battery schematic; the positioning of a microturbine fuel hybrid system taken from ([Calmotors 2010](#))

![Hybrid Electric Truck Schematic](image2)

**Figure 2-11** Hybrid Electric Truck Schematic taken from ([Calmotors 2010](#))

For hybrid electric vehicles, the very small gas turbine has the attributes of multi-fuel capability, low emissions, and a compact and light weight package. However, in the past two years or so, it seems to have lost ground as a premium prime-mover in the emerging hybrid vehicle market.
place. In the near-term, small high efficiency direct injected turbocharged and intercooled diesel and gasoline engines are viewed as the most attractive engine option. For the longer term, fuel cells are currently receiving intense focus and, in particular, efforts to reduce their cost. For the microturbine to be competitive, with efficiency values of over 40%, it would have to utilize ceramic components in the complete hot end section of the engine. This would include the combustor, turbine, scrolls and heat exchanger. It is recognized that ceramic technology is still in its infancy and today could not support an advanced 45 to 50% efficiency microturbine with the level of reliability needed for automotive service.

2.1.9.7 Space Heating

Heating the interior of the car cabin requires the utilization of waste heat, and in the case of the proposed configuration this can be taken from the microturbine and battery. Heating the interior of EVs in cold climates requires about 3 – 5 kW of power (standard vehicle size class) and this is not easily attainable from general\(^6\) EV configurations due to their low waste heat generation, (Technology 2009). This generates another user use from the microturbine waste exhaust which is currently 275 °C, Table A-2. The generic configuration of the microturbine system is shown in Figure 2-12.

![Microturbine cycle diagram](image)

Figure 2-12 Microturbine cycle diagram (use of waste heat) taken from (Bristow 2010)

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\(^6\) General is defined as any other configuration not utilising a microturbine.
2.1.9.8 Current “Range Extended” HEV and Companies

*Capstone Turbine Corporation* has recently partnered with *CalMotors*, manufacturer of traction drive systems. The two companies combined will be able to provide a complete line of microturbine-powered Hybrid electric drive solutions. Their produce will include small to mid-sized automobiles and heavy duty applications such as trucks and marine motive machines. The combined partnership will give Capstone an entry point into the growing hybrid electric vehicle market, *(Bristow 2010)*. The Capstone Drive Solution provides a great opportunity for vehicle manufacturers to integrate microturbines into a series hybrid electric drivetrains.

*DesignLine* and *EcoPower Technology (EPT)* have worked with Capstone since it went public in 2000 and have been able to penetrate the transport market using Capstone Microturbines coupled with their novel bus designs, *(Capstone 2009)*. Capstone’s 30-kilowatt microturbines have since been installed in hybrid electric buses, trolleys and transit shuttles around the world. There are hybrid buses operating today in U.S. cities like New York, Baltimore, Charlotte, London, Tokyo, Paris, Rome and Auckland using Capstone technology.

Major US truck manufacturers will utilize Capstone’s 65kW microturbine as a range extender in a series hybrid electric drive system. Class 5 to class 8 trucks will take advantage of the complete Capstone Drive Solution, *(2010)*.

*Tomoe Mine Railcar* and “*Spinner*” are private companies which have taken advantage of the unique characteristics of the Capstone microturbine for specialty applications.

Recently, Capstone has seen increased interest in using its microturbines for passenger vehicles and due to the recent partnership with *CalMotors* it seems likely that marketable units that will compete with conventional combustible vehicles will appear soon.

The EcoLogic Hybrid SUV called “*Whisper*” Is a series Plug-in Hybrid Electric Vehicle, the first Hybrid vehicle of its size to use Capstones C30 with a plug-in configuration. Fuel economy has been measured at 34 kilometers per liter – and is much better than current production hybrids. This makes it different from marketable hybrids available, such as the Lexus and Toyota which use conventional 4 stroke engines to provide both vehicle drive and battery charging (Power-split driveline). C-30 liquid-fueled microturbine was successfully integrated into the “*Whisper*” and is registered and operating on public roads in the UK, *(Capstone 2009)*.
One of Capstone’s customers also has integrated a Capstone C-30 into a high performance hybrid electric vehicle. This has challenged many as hybrid vehicles are generally associated with reduced emissions and high efficiency. The super performance CMT-380 microturbine Supercar has it all. This performance hybrid uses high energy density lithium-polymer batteries with an expected range of 130km. Energy from the batteries is supplemented by a 30kW microturbine operating on diesel or biodiesel fuel, with enough fuel storage to extend the vehicle range to an estimated 804km. This car is obviously not a worldwide marketable unit but has been successful in increasing people’s awareness and challenging expectations of this technology, (Capstone 2009). Detailed descriptions of the above applications can be further read in Appendix F.

Timing is important when introducing new technologies and HEVs are gaining interest. Many people are taking a serious look at microturbines as ideal electric vehicle battery chargers and it appears that it is the right time to supply the market with more advanced and developed models.

2.1.10 Conclusion

Microturbines are not miniaturized copies of large ones and the major problems associated with microturbines are the decrease of overall efficiency with decreasing dimensions. Problems also arise with the integration of the power electronics system with the Hybrid Electric Vehicle (HEV). The Capstone C30 is a successfully marketed unit and is a perfect starting point for conducting this study. The microturbine as the heat engine for lightweight vehicles will be further discussed in following chapters.
2.2 PART 2: Turbomachinery; Design and Operation

Part 2 focuses on the geometric size, weight and cost effects encountered with turbomachinery. The physics affecting scaling is discussed first and then the specific effects on turbine components together with the correlations are proposed, which were developed in Appendix E. The complete defined empirical relations are adapted and formulated in the context of the current problem. An overview of the design method relating specific speed and specific diameter is given.

2.2.1 Introduction

Microturbine technology continues to receive more attention due to the multi-application potential, low emission expulsion and large energy density. They are for these reasons attractive for use in distributed power generation (CHP) and applications in the transportation industry. There is a strong need for high performance in both areas and so has generated a wide interest in microturbine development. It is known that microturbines, typically below 250kW in size, use centrifugal (radial-outflow) compressors and radial-inflow turbines, while larger sized gas turbines, typically over 2 MW, use axial-flow compressors and turbines. These designs are most efficient at their respective sizes for a number of technical and economic reasons.

Large gas turbines have higher performance, are well developed, easily scalable and are concreted in a solid economical industry. The reverse is true for very small gas turbines with a radial flow compressor and turbines, having significantly lower aerodynamic efficiencies, (Balje 1981).

The engine can be modeled as a system by means of thermodynamic analysis and this must be the starting point. This analysis, denoted cycle analysis, can be used to predict the performance of the engine in terms of efficiency and power output for a given mass flow rate. The thermodynamic performances of open Brayton recuperated cycle at design point conditions are predominantly a function of four parameters:

- Peak cycle temperature, (TIT).
- Recuperated Inlet Temperature (EGT)
- Compressor pressure ratio ($P_{Rc}$); and
- Component efficiencies and size effects.
The TIT is essentially determined by the turbine rotor alloy stress rupture and low cycle fatigue strengths, duty cycle, and rotor cooling options. Likewise the Recuperator inlet temperature, equal to turbine exhaust temperature (EGT), is also determined by the recuperator matrix material life limitations. Capstone Corporation informs us that the Capstone C30 EGT is currently 650 °C, the technology level on all scales will be a reasonable 680°C during the preliminary design for future development. The pressure ratio is dictated by the compressor type and material and values higher than 3.0 are desirable to minimize recuperator and combustor volume. The turbo machinery component efficiencies are related to the compressor and turbine specific speeds $N_{sc}$ and $N_{st}$, (Rodgers 1997). The first three parameters (TIT, EGT and $P_{Rc}$) are easily variable within the GSP environment. Component efficiencies and size effects are not easy to model. GSP component schedulers and empirical relations must be used.

The component efficiencies are influenced by:

- The rotor/impellor sizes,
- Aerodynamic excellence; and
- Clearance gaps.

Blade thickness and throat area tolerances concern both efficiency, casting producibility, plus blade erosion. The aerodynamic excellence of both radial and axial turbines is dominated by two parameters:

- Velocity ratio $= \frac{U_t}{V_0}$, ratio of rotor tip speed to theoretical spouting velocity; and
- Exit Flow coefficient $= \frac{c_m}{U_t}$, ratio of exit meridional velocity to tip speed.

The velocity ratio is direct measure of the blade loading. The exit flow coefficient is an indirect measure of the specific speed. Radial turbine design for example is dictated by criteria like specific speed and/or velocity ratios. For smaller turbines the size of the turbine wheel needs to be reduced and thus the rotational speed increased in order to reach a high efficiency.

Microturbine thermodynamic performance is therefore, size (output power) dependent, in contrast to larger industrial gas turbines, (Rodgers 1997). The rotor/impellor sizes and clearance gaps will be addressed in greater detail.

Creating a microturbine is not a simple matter of scaling down their large industrial counterparts. Miniaturization of turbines cause large changes in Reynolds number, the geometrical restrictions related to material and manufacturing of miniaturized components.
become a greater concern and the heat transfer between the hot and cold components increase, which is negligible in large machines, (Kolanowski 2004).

2.2.2 Assumptions

Specific assumptions must be made in order to create a viable model design. A few important ones will be noted below;

- The generator doesn’t vary significantly with size and so a constant efficiency of 92% will be assumed throughout all the scales. This value includes the losses through the power electronics and the generator losses.

- The temperature limits of the technology (recuperator inlet temperature), is 680 degrees Celsius, 30 degrees higher than the Capstone C30. This is a reasonable assumption for future technological development.

- The design compressor pressures will scale appropriately with the change in efficiency at each power level.

- The design spool speed at every scale will be the same, even though it would increase as power level decreases. The spool speed is assumed to be the same maximum value for all scales and will vary in a similar way when the speed control is used (See Chapter 4).

- The turbine and compressor component efficiencies will scale with the relationships mentioned and were developed to represent future technology level; and

- The Recuperator will maintain a constant effectiveness throughout the size variation. It will vary in weight and volume only due to material selection and air mass flow.

2.2.3 Empirical Size Effect Correlations adapted to Experimental Data

In a previous study conducted by the author empirical relations were found and correlated against experimental data. The background required to understand the equations is provided in Appendix E.1 and a more detailed analysis of the equations can be found in Appendix E.5.

The equation for the turbine for size effects takes the form
The equation for the compressor for size effects takes the form

\[
\frac{1 - \eta}{1 - \eta_{\text{ref}}} = \left( \frac{D_{\text{ref}}}{D} \right)^{0.31}
\]

2-1

where the reference is the original turbomachinery design and the power index (0.31 and 0.15 but formally ‘n’) is usually in the range from 0.1 to 0.5, which is dependent on the size and design of the turbomachinery, Appendix E.5.

The equation for the compressor for PR effects takes the form

\[
\Delta \eta_c = \frac{0.1}{\sqrt{\gamma_c R}} \left( \frac{P_2}{P_1} - 2.0 \right)
\]

2-3

The value of the constant ‘0.1’ is for current small compressor designs, with zero inlet air pre-rotation and is usually of the order 0.1-0.13. Small radial turbines are not prone to significant Mach number and pressure ratio penalties providing pressure ratios are lower than approximately 5.0, (Rodgers 1969). R is the gas constant in ft lb/lb deg R and is gas dependent, therefore 53.3 was used for air.

The relation for tip clearance effects

\[
\Delta \eta_t = 0.20 \left( \frac{C_c}{h_c} - 0.02 \right) \ll 0
\]

2-4

\[
\Delta \eta_t = 0.1 \frac{C_t}{h_t}
\]

2-5

The losses resulting from clearance gaps are basically related to the ratio of the effective clearance gap to blade height. Due to a lack of blade height and clearance gap data on different
sizes of turbines it is unlikely that a relationship can be established for tip clearance effects. They will therefore not be included in this study. This will obviously induce some error in predicting the full size effect efficiency loss due to size decrease. This loss can be as much as 1% in both turbine and compressor components, Appendix E.5.1.

2.2.4 Implementation in GSP

2.2.5 Design and Off-design Performance Computation:

The gas turbine performance simulation can be sharply divided into two categories: design point analysis and off design modeling. The first category mainly involves the engine designer because it consists in selecting the best thermodynamic cycle in order to achieve a performance goal: For example delivered shaft power for turboshift engines, or temperature conditions for APU starts. This analysis is led at a single working point, the so called design point, which is supposed to be representative of typical customer use. The design analysis allows optimization of the cycle and preliminary design of the engine components usually by selection of compressor pressure ratios and turbine inlet temperatures.

It is only after this analysis that a first engine geometry is defined. At this step, the engine performances are known only at design point. In order to estimate the performances under various ambient air conditions and power or thrust settings, it is necessary to create an off design model which has the ability to describe the behavior of the engine components in conditions other than those at design point.

2.2.6 Size, Weight and Cost

The size of turbomachinery is not only related to component efficiency but driven by material selection and also other important performance parameters. The type, geometry and dimensions depend primarily on:

- Specific speed, \( N_s \): determines the type & basic shape of the runner & other parts of the unit; and
- Cost

2.2.6.1 Specific Speed
Similarity Relations and Design Criteria of Turbines:
Low specific speeds are associated with high heads and high specific speeds are associated with low heads. Moreover, there is a wide range of specific speeds which may be suitable for a given head. Higher specific speeds for a given head results in smaller turbine/generator dimensions and higher speed generators. Since turbine capital cost decreases with an increase in speed, a balance must be found with the efficiency, (Benguedouar 1988).

Turbine Setting
The negative aspect of high specific speeds is they requiring a deeper setting to avoid cavitation and must also be included in the assessment.

Parameters that contain the rotational speed and rotor diameter are useful when describing the characteristics of turbomachines. These values are provided in the form of specific speed \( N_s \) and specific diameter \( D_s \).

Using the \( N_s - D_s \) diagrams another factor called specific diameter can be obtained and from which the size and the approximate weight of the turbine are calculated. These diagrams are found in Appendix B.6. The specific diameter is defined as the diameter of a turbine which handles a volume flow \( Q \) of unity at exit and expands a head of unity. \( N_s \) and \( D_s \) are derived in Appendix B.5 and are stated below:

\[
N_s = \frac{NQ^2}{(gH)^{3/4}} \quad 2-6
\]

where,

\( N_s \) is specific speed (unitless)
\( N \) is pump rotational speed (revolutions per seconds)
\( Q \) is flowrate (m³/s) at the point of best efficiency
\( H \) is total head (m) per stage at the point of best efficiency
\( g \) is acceleration due to gravity (m/s²)

\[
D_s = \frac{DH^{1/2}}{Q^{1/2}} \quad 2-7
\]

With
\[ \omega = \pi \frac{N}{30} \]

Ns is called the specific speed. Ns is directly proportional to N. In the case of a turbine the \textit{power specific speed} \( N_{ps} \) is more useful and is defined by

\[ N_{ps} = \frac{\hat{P}_1^{\frac{1}{2}}}{\psi_1^{\frac{5}{4}}} = \frac{N(P/\rho)^{\frac{1}{2}}}{(gH)^{\frac{5}{4}}} \]

N is based on the values of N, P, and H used at the design point. i.e. At maximum efficiency. Both Ns equations are dimensionless, (Dixon 1998). Specific speed shows that when selecting a turbomachine for a given head H and capacity Q, the highest possible value of Ns should be chosen because of the resulting reduction in size, weight and cost. Keeping mind the constraints with efficiency a turbomachine could be made extremely small were it not for the corresponding increase in the fluid velocities. “For machines handling liquids the lower limit of size is dictated by the phenomenon of cavitation”, (Dixon 1998). Since a higher specific speed implies a smaller machine, for reasons of economy, it is desirable to select the highest possible specific speed consistent with good efficiency, (Dixon 1998). A design method in Appendix B.6 allows the reader to estimate the tip diameter, efficiency and specific speed (shape number) with Equations 2-6, 2-7/“power specific speed” and Figure B-7 to B-9.

2.2.6.2 Head in Turbomachines

The specific work can, on the basis of the energy equation, be expressed with the head as:

\[ pw = g * H \]

2.2.6.3 Turbine Size and Weight

The weight and the efficiency of a recuperated microturbine generation system are very important parameters. It can be found that the turbine and the compressor weight are very small compared to that of the whole system. Therefore the efficiency alone can be a fine selection criteria when choosing the type/size of the turbine or the compressor, (Benguedouar 1988).

\textit{Turbine Size:}

For fixed specific speed and cycle conditions, the scalar relationships can be applied.
It should also be pointed out that when scaling the component dimensions the associated power output and mass flow rates will not be the same as theoretical predictions. This is due to issues such as surface finish on airfoils and tolerances which do not scale. Recurring manufacturing cost is one of the other many factors that do not scale. The rotor tip diameter can now be linked with power output and mass flow. Figure 2-13 and 2-14 show a range of tip diameters and their related predicted efficiencies and mass flow rates. Cost usually must be independently evaluated because it is difficult to predict. A usual trend is observed however that when scaling up the cost per unit, power should go down. It is simply more cost effective to use more expensive technologies for application in bigger engines first. Refer to Appendix E.1 for the particular scale effects and associated issues.

![Figure 2-13 Turbine size, efficiency and mass flow](image)

Figure 2-13 and 2-14 show variation in tip diameter with the respective predicted efficiency and mass flow rate variation. This variation in size was formulated from Equation 2-1, 2-2, and 2-8. The highlighted yellow row is the reference data obtained from the Capstone C30.
Turbine Weight:
The weight may still be an important parameter for other generation systems. An empirical model was formulated. This model gives the specific weight, which is the weight of the engine per kilowatt, (Berner 1985; Benguedouar 1988). This weight estimation was determined to include the scroll weight and it generally doubles the weight of the turbine system.

This empirical expression is given as follows:

\[ F_t = 10.944(P_s)^{-0.292}(0.985)^{\Delta Y} \]

Where:

- \( F_t \): is the specific weight in (kg/kw)
- \( P_s \): is the shaft power required (kw)
- \( \Delta Y \): are the years from 1958.

Figure 2-14 Compressor size, efficiency and mass flow
Since the empirical correlation was correlated in 1958 and is generally meant for larger turbine systems for space power generation care must be taken on estimation on a micro scale. Turbines using this relation are generally around the 1000kW mark, (Berner 1985). It still may be applicable if the constants can be adapted appropriately with turbocharger data. Differences in materials and difference in application used may also skew data. Three different microturbines (Capstone C30/C65 and MTT 3kW) were placed on a plot of total microturbine generator weight vs. output shaft power. The MTT 3kW microturbine is suited for comparison in this application because it is assumed the recuperator uses an optimized plate thickness between 0.15-0.20mm giving an estimated recuperator weight of 10kg. Considering that the 3kW microturbine system runs on oil bearings a 15% reduction in total weight is assumed, (Capstone 2009). The turbine-compressor-alternator weights were then calculated by removing the recuperator weights from the total system, shown in Figure 2-15. The green data points are a representation of the empirical expression with values altered. It does not appear to give an accurate representation of the trends. Therefore two trend lines plotted for the microturbines give a better representation of power to weight variation. The microturbine for HEVs would lie somewhere within the above range. The

![Turbomachinery Weight Variation](image)

**Figure 2-15 Turbomachinery Weight Variation with Power**
Capstone Drive Solution package is represented by Figure 2.9 and the design data with weight and dimension breakdown for the C30 and C65 are listed in Appendix A.5.

2.2.6.4 Recuperator Size

The size of a prime surface metallic recuperator is impacted by several parameters, the recuperator matrix volume (and cost) can be considered approximately proportional to the following group of parameters, that includes an engine power related function, heat exchanger parameters, and surface geometry characteristics:

\[
V \propto \frac{\dot{m}_a}{\sqrt{P R_c}} \times \left( \frac{\varepsilon}{1 - \varepsilon} \frac{1}{\Delta \rho \sqrt{P}} \right) \times \left( \frac{T}{f} \frac{1}{\beta} \right)
\]

Where:

\[
\frac{\Delta \rho}{P}
\]

is the heat exchanger pressure loss

\[f\]

is the fanning friction factor

If the heat exchanger mass flow rates were kept constant, the impact of increasing the effectiveness from say 0.83 to 0.92 would result in an increase in recuperator size by a factor of about 2.3, (McDonald 2000). Recuperator effectiveness values exceeding 0.90, therefore, have very large weight and high cost. These can only be justified when thermal efficiency and the value of power produced dominates operating costs.

It is seen that the recuperator volume is increased by raising the effectiveness (to yield higher engine thermal efficiency), the size of the recuperator is also affected by the reduction in airflow (the term \(\dot{m}_a\) in the above equation). The matrix frontal area and length can be calculated with the relations in Appendix E.10 assuming a fixed surface geometry. The Capstone C30 recuperator core weight is assumed approximately 29.5 Kg and is 45.5cm in diameter, (Treece, Vessa et al. 2002). McDonald provides a compact analysis for a 50kW unit relating cost, weight and materials in his paper “Low-cost compact primary surface recuperator concept for microturbines”. The Recuperator volume will decrease with varying turbine size (mass air flow
decrease). The other parameters will be assumed constant, eg. effectiveness throughout the sizes in this assignment, *(McDonald 2000)*.

2.2.6.5 Generator

High Permanent Magnet Generator (PMG) tip speeds may be preferred to reduce rotor length with a stiffer shaft, but as a result generator efficiency decreases due to higher windage losses. PMG power capability \( P \) is linked to rotor speed and volume by the following relationship:

\[
P = \eta N L D \frac{ESS}{Constant}
\]

Where:

- \( ESS \) is the electromagnetic shear stress,
- \( \eta \) is the efficiency, and
- \( N \) is the rotational speed.

The terms \( L \) and \( D \) are shown in Figure E.8, and the equation has been used to prepare the plot shown for a rotor length to diameter ratios of 1.0, 2.0 and 3.0.

Appendix E.7 provides graphs of reference, *(Rodgers, Watts et al. 2001)*. Typical microturbine system cost percentages are of the order;

- Powerhead 25%,
- Recuperator 30%,
- Electronics 25%,
- Generator 5%,
- Accessories 5%,
- Package 10%

*(Rodgers, Watts et al. 2001)*

2.2.6.6 Combustor

A combustor sizing relation is suggested in Appendix E.9, however for reasons of complexity combustors are considered not easily scalable. A breakdown of the weights for the Capstone commercialized units (capstone drive solution) is given in Appendix A.5. Refer to Figure 2.9 for a graphical representation of the individual components.
2.2.7 Conclusion

Turbine thermal efficiency to a large extent depends on losses resulting from flow leakage, thermal losses and friction. These losses become more dominant when a turbine is down-scaled in terms of size and power, due to blade tip clearance and volume-surface ratio scale effects. Manufacturing geometrical tolerance limitations inhibit solutions to reduce tip leakage losses for example. Moreover, when decreasing size (which leads to lower Reynolds number) viscous friction losses become larger in conventional turbo machinery. As a result, there is a fundamental limitation to efficiency of microturbines with a conventional configuration. Within the current turbine design the size effects will not include tip clearance influences and are not incorporated due to lack of C30 design data. The physical size of turbomachinery, recuperator and the proposed empirical relationships will serve the purpose of providing a solid preliminary design.
Chapter 3

Conceptual/Cycle Optimization Study

This chapter involves the initial design point study analysis and construction of the basic model that GSP will use to numerically calculate the points that will be used for off-design analysis.

3.1 Introduction

The Capstone C30 will be taken as a reference during this study. It was therefore necessary to obtain as much information about the microturbine as possible. The design information is given in Appendix A. This model together with its adaptions is to be simulated numerically in a Gas Turbine simulation program called GSP. All configurations are based on a single spool unit and so the free turbine is also the gas generator turbine, a configuration adopted by the C30 and C65 Capstone Microturbines, (Kolanowski 2004).

3.1.1 GSP Engine Design Model and Tree

Figure 3-1 The Core C30 Engine and Scale Tree
The model is depicted schematically in Figure 3-1 and shows what the recuperated turboshaft microturbine looks like in the GSP model environment. GSP was used to construct the model and consists of the recuperator (component 20), compressor (component 19), turbine (component 23) and the link bars to make the connections. The scale tree is represented on the left and has design and steady state series sections. Each scale represents a different engine size and all components have separate windows that become available upon selection.

### 3.2 Assumptions for GSP Simulations

GSP is software program and it has its limitations. That is why some assumptions were necessary before conducting the calculations, and so they will be described in this subchapter. It was previously stated that the Capstone C30 was to be taken as a reference during the performance analysis and was also used to establish values in the scaling relationships. The amount of data available on the C30 unit was limited. Values were used from engine performance specifications where possible, which was sourced from official manufacturing data or other reliable technical sources. Other data was obtained from marketing brochures as distributed on exhibitions and cross referenced with data available in published journals.

#### 3.2.1 Factors Affecting Performance

A material limit had to be defined and corresponded to a T5 (EGT) temperature of 680 degrees Celsius. This was ultimately determined from the material temperature limits of the recuperator and the Capstone C30 has a current limit of 650 degrees Celsius, ([Capstone 2009](#)).

### 3.3 GSP simulations

#### 3.3.1 Design Point

Gas turbine performance means *Design Point Performance, Off-Design Performance* and *Transient Performance*. Design Point Performance is central to the engine concept design process. Off-Design Performance is the steady state performance of gas turbine as its operational
condition is changed. Transient Performance deals with the operating regime, where engine performance parameters are changing with time.

The design point is described as running at the particular speed, pressure ratio and mass flow for which the components were designed. The problem then remains to find the variation of performance of the gas turbine over the complete operating range of speed and power output, which is normally referred to as off-design performance, (Saravanamuttoo, Rogers et al. 1996).

In any simulation session in GSP, the first step in determining the engine performance is the design point calculation. This step is used to size the model components which results in a defined geometry. The results serve as a reference point for off-design calculations. With the user-defined ambient conditions, the design point performance is determined by calculating the component performances from the intake towards to the exhaust component.

The cycle design mode (DP) performance calculations require engine rated performance data. Input parameters are typically the compressor pressure ratios, the component efficiencies, burner exit temperature, total mass flow and bypass ratio. Together with assumptions about the secondary air system and parasitic losses; the turbine flow capacities, pressure ratios and nozzle areas may be derived. No component maps are needed for these cycle design mode calculations, (Kurzke 2005; Verbist 2010).

Study of the thermodynamic cycle with GSP

Variables that were controlled during different series of simulations by the case control; Turbine Inlet Temperature (TIT) and compressor pressure ratio (PRc), turbomachinery component efficiencies $\eta_{te,c}$ and air mass flow ($\dot{m}_a$). The generator efficiency most likely will vary slightly with scale but was assumed constant with limited error. The variables that were kept constant were combustion pressure drop and heat exchanger hot/cold side pressure loss.

3.4 Description and Presentation of the Simulations

The focus of this report has been put on a study of the thermal efficiency and specific work output (power, if flow rates are used). The reason for these specific parameter investigations is that dimensionless specific work output is used as a sizing parameter and the efficiency is

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7 Operational conditions are defined as the input parameters power setting, ambient conditions, engine condition etc.
important because it tells us how much useful work we can extract. The dependence of thermal efficiency on different configurations of pressure ratios and temperatures was researched. Generally the thermodynamically favorable conditions are (but are solely dependent on configuration):

1. **High pressure and temperature at the turbine inlet;**
2. **Minimal losses during compression and expansion.**

To achieve the above conditions high turbine and compressor efficiency together with low pressure losses must be employed. High component efficiency comes at a price; reduced durability of the gas path components.

### 3.5 Implementation of Scaling Relationships

The method to implement the equations into GSP involves defining a constant which will be directly used to control the scale of the model. Then the scaling equations will be adapted and implemented into the Scheduling windows for the respective components that need to be scaled.

#### 3.5.1 Constant Expression

In the input fields of GSP you can enter expressions of constraints and constraint identifiers which are definable in the constant expression component. **Constant expressions** (GSP component) provide the user with a means to represent constant values with parameter identifier names. The modeler can define constants or constant expressions for an identifier that in turn can be used in any component model input field expression that is parsed during simulation. As an example, one could define a 'Scale' constant parameter that is multiplied with other values in expressions such as the inlet mass flow to represent different engine scales. With changing the scale in the constant expression component, the actual inlet mass flow will be scaled accordingly, which is convenient for scale effect studies. A scale constant “C30” was defined in the constant component within GSP. This constant will be given a reference of “1” at the Capstone C30 microturbine. Variation of this value will be intended to indicate an increase or decrease in the scale of the microturbine. Equation 2.8 was adapted for this purpose.

\[
C30 = \frac{P}{P_{ref(C30)}} \propto \frac{\dot{m}}{\dot{m}_{ref air}}
\]

Within GSP
• Used to scale the mass flow rate.
• Used to scale the compressor and turbine efficiencies

3.5.2 Equation Scheduling

To add relations among input parameters, we use scheduling components such as the equation or (special for DP) DP equation components, here you can assign a component input parameter as a function of other parameters (have to be in the output); you can select from a list of available parameters. Refer to Figure 3.1 or 4.1 for a graphical representation of the constructed model in the GSP working environment. Parameters that are scaled:

3.5.2.1 Turbine efficiency

If we rearrange Equation 2-1

\[ \eta = 1 - (1 - \eta_{\text{ref}}) \left( \frac{D_{\text{ref}}}{D} \right)^{0.31} \]

Diameter scales with \( \sqrt{C_{30}} \) roughly. Where the C30 is the ratio of output shaft powers of the chosen and reference microturbines.

\[ \eta_{\text{turbine}} = ET A_{\text{des}} = \left[ 1 - (1 - \eta_{\text{ref}}) \left( \frac{1}{\sqrt{C_{30}}} \right)^{0.31} \right] \]

3.5.2.2 Compressor Efficiency

The same arrangement is made for the compressor efficiency but in addition we also have the Pressure Ratio effects on efficiency due to variation with size. Equation 2-2 and 2-3 are used to form:

\[ \eta_{\text{comp}} = ET A_{\text{des}} = \left[ 1 - (1 - \eta_{\text{ref}}) \left( \frac{1}{\sqrt{C_{30}}} \right)^{0.15} \right] + \left[ 18137 - \left( \frac{0.1}{\sqrt{R}} \right) \right] \]

With \( PR_{c} = 3.5 \) and C30=1 the expression above falls down to 75% which is the reference efficiency of the Capstone C30 compressor.
3.5.2.3 Inlet mass flow rate

The inlet mass flow rate is simply scaled with the C30 factor.

\[ \dot{m}_{\text{a}} = C30 \times \dot{m}_{\text{ref}} \]  

\( \dot{m}_{\text{ref}} \) is the inlet mass flow rate of the Capstone C30. This relation may induce some error in the calculation when reducing scale. At small scales the proportionality relationship between mass flow, power and tip diameter defer as previously mentioned in Chapter 2.2.6.

3.6 Fuel Cost Consumption Analysis

A concise feasibility study was conducted to analyze a selection of fuels and their associated costs. This is important to the project because the fuel selection is an important design parameter which differs with available resources. It is to be stressed that the indirect cost of the associated infrastructure is also important to take into account. In reality all these factors will influence the selection criteria of the fuel. The thermal efficiency and specific power output of a microturbine stays the same regardless of what fuel the user selects. The difference will be the amount of fuel mass flow needed to sustain the output shaft power. The mass fuel rates will differ (hence SFC) and so the costs associated with the specific fuel chosen.

We know that the microturbine has a low efficiency in power generation applications, especially when compared to that of conventional vehicles. The configuration within the vehicle will affect all other items needed for the microturbines installation and use. In relation to fuel this includes natural gas piping (if natural gas is the fuel). Microturbines using fuels such as diesel, which require large storage tanks, will need other types of fuel delivery and system configurations that may be more expensive than the configuration for gas. Eg. Maintenance costs, cost of fuel, repairs and taxes. Keep this in mind when the fuel cost analysis is done, (Soares 2007).

3.6.1 SFC Capstone C30

The Capstone C30 can be investigated via its SFC \( \frac{l}{kWh} \). Observe the SFC for the cycles below for a PR range of 2-6 and a TIT range of 1000 -1400 K. The recuperated cycle experiences reduced fuel consumption at higher TR. This will enable the most suitable fuel to be chosen for off-design analysis.
3.6.2 Various Fuels

Three fuels are analyzed in terms of cost and using Equation B-2 and so three graphs are generated. The fuels analyzed are Methane (CNG), Propane (LPG) and Diesel. The fuels have been selected based on their differing LHV and H/O ratios. Diesel is currently used in Capstones C30 Microturbines with the future possibility to utilize CNG. Bio-diesel is briefly mentioned by means of a comparison to diesel as it is an oxygenated fuel. This improves the combustion of fossil diesel and reduces the particulate emissions form un-burnt carbon, (Faiz, Weaver et al., 1996). It is also known that Capstone uses biodiesel in some of its C30 models even though fuel costs are more than regular diesel fuel. Biodiesel has slightly higher specific fuel consumption but has reduced emissions due to the absence of sulfur, aromatic hydrocarbons, metals or crude oil residues which makes it less corrosive and reduces harmful emissions. In regards to diesel/biodiesel and its comparison to CNG: “The Capstone C30 Liquid Fuel microturbine is the only diesel-fueled range extender that has been certified to the CARB Urban Bus and Urban Bus Hybrid standard for operation on diesel fuel, according to the company. Capstone recently completed third party testing and validation of its compressed natural gas (CNG) product and anticipates filing for CARB certification in the coming weeks.”(Congrass 2010). The purpose for use of CNG is based virtually on its nonexistent emissions, significantly less than diesel and so would conform to any countries regulations on pollutant emissions, (Faiz, Weaver et al. 1996).

3.6.3 Fuel Costs

The following will analyse the fuel costs for the marketed Capstone C30 microturbine (recuperated) under three different liquid fuels. The fuel property data for the analysis is taken from Table B.1. Fuel prices are taken from the period of May 2011 and are therefore valid for comparative purposes, all prices are noted from, (Neutral 2010). The Fuel price is country specific. CNG was noted to be 0.78 €/l, Diesel 1.384 €/l and LPG 0.82 €/l. The study was conducted in South Holland, Netherlands.

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8 It contains a reduced amount of carbon and higher hydrogen and oxygen content, in this instance less than fossil diesel.
3.6.3.1 Diesel

![Diesel Fuel Cost Variation](image)

**Figure 3-2 Diesel Fuel cost variation**

3.6.3.2 Propane (LPG)

![LPG Fuel Cost Variation](image)

**Figure 3-3 LPG Fuel cost variation**
3.6.3.3 Methane (CNG)

The fuel cost variations have been plotted for three different fuels in Figure 3-2, Figure 3-3 and Figure 3-4 for the recuperated cycle. The turbine inlet temperature curves (TITs) are separated in increments of 50 degrees Celsius and have been plotted for a PR<sub>c</sub> range of 2-6. Fuel costs are reduced as the TR is increased but there is a material limit associated with increasing TR, and currently it is quite low for microturbine technology. The dips which are marked by the green line are the optimum points which correspond to the optimum pressure ratio for that temperature. If we look the different fuels at a TIT of 1125 K and the associated optimum point, which is close to 3.5, the fuel costs for the Diesel, LPG and CNG respectively are 0.505, 0.42, 0.57 €/kWh. We conclude from this that LPG is the cheapest fuel to use per kWh followed by diesel then CNG. A careful choice must be made on the selection of fuel; a decision based on cost and associated emissions which differ according to the laws and resources of the country at hand.

![CNG Fuel Cost Variation](image)
3.7 Capstone C30 Design

Figure 3-5 is a design series plot of the Capstone C30 with LPG as the fuel. The range of PR\(_c\) chosen was 3-4 because, as discussed in Appendix E, high values of PR for radial turbomachinery incur losses and the scaling relationships are only valid for PR values of less than 5. We see that the design point of the C30 is a red dot and corresponds reasonably with all design data. It also lies along the optimum pressure ratio line. The Capstone C30 was built to run at optimum conditions as it is the most efficient point to operate within thermodynamical limits.

The graph states that the mechanical efficiency is 26.5 which appears to be a bit on the low side. Expected efficiencies are around 28-30%. Discrepancies could be an overestimation of system losses or an underestimation of component efficiencies. It is clear to see the potential efficiency and power increase with higher material limits.

3.8 Scale Performance Plots; Design Sweep (36, 30, 22, 15, 9, 3kW)

The performance plots were created through the dimensionless form of Equation B-42. Equation B-42 and B-43 gives an idea of what the shaft power and thermal efficiency are a function of analytically and fundamentally important for understanding numerically. Figure 3.6 was created
to indicate the range of interest and to point out the danger zone where the order of inaccuracy increases.

Figure 3-6 Design Sweep analysis

Custom implemented equations allow relationships to be developed among the data generated in GSP. In order to generate a carpet plot one must use equations from Appendix B.3, Equation B-34. A Design sweep was run with the scaling effects and pressure ratio influence implemented. The carpet plots above represent 6 power settings (scales- 3, 9, 15, 22, 30, 36kW) for TITs ranging with 1000-1400 and a PR 3-4. The PR varies in intervals of 0.25 from top to bottom in each plot. The blue carpet plot is the Capstone C30 and the rest are appropriately labeled above. The above design sweep of various scales provides a brief overview of the power settings to be investigated and outlines comparative behavior and trends.

It is noted that the tip diameters of the rotors associated with a power of 8.5kW for turbines and compressors generally fall below the 12.7cm mark. The sharp decrease in efficiency with power is quite notable below 8.5kW. Scaling effects, PR influence along with tip clearance become quite sensitive with scales in this range. This highly sensitive zone (1-8.5kW) would essentially need an adaption of Equations 3.1 and 3.2 and a reference turbine chosen closer to that of the scaled one. This is not a particularly worrying issue because the region of investigation is be focused around 9-36 kW. The validity of the above equations will hold.
3.9 **Optimization based on expected technology level in 5 years**

Essentially the optimization process should be conducted on expected technology level in 5 years rather than existing / previous tech level. E.g. estimation of better turbine efficiency, compressor efficiency, and recuperator effectiveness or less sensitive scale effects. This adaption was made to the reference Capstone C30 after comparisons with original data. Increased component efficiencies leading to overall thermal efficiencies of greater than 40% and reduction of toxic emissions are needed to show the microturbines benefit as a combustible engine in HEV technology, *(Kolanowski 2004)*. All changes made to the reference microturbine propagate through every scale.

3.10 **Conclusion**

A preliminary design model was constructed to create design sweeps of the various scales which in effect sized the turbines for the input data specified. The input data was obtained from the Capstone C30 model sized for the Capstone Drive Solution. Scaling relationships were also used in the scheduling components in order to get a more accurate model. A fuel cost analysis was conducted to determine the most suitable fuel to be used for further simulations. LPG was used in a design sweep of all scales to graphically show the region of interest. In order to make microturbine technology more feasible, improvement towards cheaper materials for hot-end components such as the recuperator and turbine are necessary.
Chapter 4

Off-design: Steady-state Simulations

This chapter moves on to the off-design model construction and analysis of the configuration. A definition of the microturbine performance data format or data link method is defined in order to create a way to supply the data to the Hybrid Electric Vehicle model.

4.1 Off-design Performance

In order to calculate the steady state and transient performance of a gas turbine engine over a range of operating powers and ambient conditions, it is necessary to consider the interaction of all the components in an engine. The off-design or part-load performance calculations are necessary to ensure that the engine is capable of operating throughout its flight envelope and power range in a safe, stable and efficient manner. It has already been pointed out that the Design Point (DP) is the starting point for engine design and is usually max/high power. The data is first used to create a thermodynamic model and is subsequently extended with a control suite.

The Off-design performance (OD) is concerned with

- How does the operating point change with deviating operating condition?
- Steady-state equilibrium; and
- Transient (change in time, accel/decel of rpm’s).

A more refined off-design model can be created using compressor and turbine maps/ recuperator maps to predict off-design corrected mass flows, pressure ratios, efficiencies, relative shaft speeds, etc. A further refinement is to allow the component off-design pressure losses to vary with corrected mass flow, or Mach number, etc. With the design point serving as a reference
point, off-design engine performance can be simulated. In GSP, off-design performance is an iterative process towards a solution for a steady-state or transient (quasi steady-state) operating point. Steady state off-design calculations start with a user defined change in operating conditions. For the C30 models, this can be a change in ambient conditions, or a change in the specified fuel flow using the fuel flow control component.

Steady-state OD simulation requires additional information with respect to the DP simulation. This information is included in the model by the component maps that contain the OD performance of a component model. It is via power codes⁹ that we can solve specified values of key engine parameters and to restrict any solutions that would exceed input limiting values.

Transient simulations include time responses, thereby capturing the dynamic effects of an engine model subjected to a change in operating condition. For the transient OD simulation of gas turbine models additional information is required with respect to a steady-state OD simulation, in order to describe its time dependent behavior. It will not be necessary to incorporate transient simulations in the model. Reasons for this are due to the configuration of the heat engine in a series hybrid power train as discussed in Chapter 2.1.9. The battery is able to use its stored energy to drive the vehicle for the required amount of time so that transient behavior can be accurately dismissed.

### 4.1.1 Operating Conditions

The operating conditions are defined by

- Power setting;
  eg. PLA (with control system model), power lever angle, fuel flow, torque, combustor exit temperature, variable nozzle area or power turbine load (power/speed/torque).

- Fuel properties;
  e.g.: jet fuel, diesel, LNG, bio-gas, hydrogen, etc.

- Inlet/Ambient conditions;

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⁹ The power code is NOT necessarily a representation of a percentage of the shaft power. It is user defined and in this particular case represents scheduling of the shaft spool speed and TET/EGT. It just so happens that the 100% PC corresponds to max power.
eg. temperature, pressure (air speed) or Mach number and inlet air composition (humidity)

- Engine condition;

eg. component condition (efficiencies, erosion, corrosion, fouling, failures, damage) on component maps and control system

- Installation losses

e.g. inlet pressure loss and compressor bleeds (bleed air, PTO)

Certain variables above are selected and varied in the GSP Simulation Input. The GSP Simulation Output displays Design point performance and Steady-state off-design performance tables ready for graphical interpretation. Design point data is usually available and is used to calculate missing data. The Off-design performance data is concerned with validation. Building a GSP model for off-design simulations usually requires component characteristic maps which are often hard to obtain or scale from available standard/similar maps. The selected operating conditions to be varied are the ambient conditions and power setting.

4.1.1.1 Ambient Conditions

The ambient conditions and load have an effect on the operating characteristics of gas turbines. The power output and efficiency are quite susceptible to varying ambient conditions and at elevated inlet air temperatures both will decrease. Ambient conditions affect a gas turbine at both the compressor inlet and turbine outlet. At the compressor inlet, the higher the temperature and the lower the pressure, decreased airflow mass rate\textsuperscript{10} causes power drops. The efficiency decreases because the compressor requires more power to compress air of higher temperatures, (Kolanowski 2004). If the environment is overly humid then this also induces an effect on the operational characteristics. A value of 60\% was specified where appropriate in the models, which was the value specified at the design point of the Capstone C30. Higher specific humidity increases the specific volume of the inlet airflow, so that the mass flow through the turbine is reduced, resulting in increased heat rate. This may also require a higher fuel mass flow rate to sustain the same power output and hence will result in a less efficient unit. It is not possible to design any engine for optimum performance at all flight conditions, since a conventional engine

\textsuperscript{10} As temperature increases density of air declines, therefore air mass flow rate decreases.
with a fixed design point will give optimum performance at that point only and not for the entire flight spectrum. The practice has been to design the engine for optimum performance at particular operating conditions (ISA\textsuperscript{11}), and to select the engine parameters based on this operating condition. Thus the ISA condition was the chosen design point, (Arun Prasad and Sundararajan 1985).

4.1.1.2 Limiters

“The Limiter control component limits a user specified GSP output parameter to a predefined limit schedule during an OD simulation”, (Visser, Michiel et al. 2010).

**Maximum and Minimum Limiters**

The limiters govern the amount of power able to be delivered by an engine and are meant to represent the physical quality of the technology available. The turbine engine may be restricted by maximum spool speed, maximum temperature and/or maximum pressure. More than one limit may be active but it depends on the flight condition, the amount of power off take and bleed air off take, (Kurzke 2007). Various limiters dictate the roofs of the carpet plots when conducting the off design steady state study. The Torque, Load, TET or $PR_c$ can be limited and in this study a maximum is set on the EGT while scheduling the torque in order to create the maps required for each power code. These limiters can be graphically seen in Figure 4-1. The maximum power which may be used will be when the first of the 3 limits are reached. When the engine reaches its limits of temperature or rpm it is at its thermodynamic limit. The Flat Rated Temperature (FRT) will be used when discussing the effect graphs. The difference between the low and high power rating is called flat rating, or de-rating.

\textsuperscript{11} Corresponds to a compressor inlet total temperature of 288 K and inlet total pressure of 101,325 Pa.
4.1.1.3 Maps

**Simulation with Standard Component Maps**

All the component maps are physically sound representations of real turbo machines. They may be scaled within certain limits to represent the behavior of similar engine designs.

**Turbine Maps**

Component performance characteristics are represented via ‘maps’ in the form of a graph. The maps may be of different form and depends on the component type. Compressors, turbines, fans and combustors have standard predefined graphical layouts that are used by various programs and the gas turbine community. For other components simple 1-D and 2-D maps are used for which simple X-Y plots can give a simple graphical representation of the variables. These are used such as in scheduler-, control- and case control components, (Visser, Michiel et al. 2010). The turbine map was utilized from a similar design in a previous study on the Capstone C30.

**Compressor Maps**

Component map selection should be selected from machines of similar design or at least in the same area of application. For instance an axial compressor is not suitable for describing the performance of a radial compressor. If the compressor design pressure ratio deviates from the original map then the speed-flow relationship will be represented incorrectly, (Kurzke 2005). The compressor map was utilized from a similar design in a previous study on the Capstone C30.

**Recuperator Maps**

No viable recuperator map was found and verified to the confidence of being incorporated in the model. A constant effectiveness value of 0.83 was used for each scale. Since the recuperator performance is not being scaled throughout the different sizes the results are still comparable.

**4.2 Simulating the Control System**

A control system to drive the model is required. In an engine the operating point is influenced by modulating the fuel flow. The Proportional-Integral-Differential controller is a simple representation of an engine control system. GSP is able to model sophisticated engine control systems and does so by testing known thermodynamic limits. These thermodynamic limits are
translated to control schedules and the power controller is a suitable component for this task, (Visser, Michiel et al. 2010).

4.2.1 Implementing the Model

The GSP engine performance model featuring the components described above is shown in Figure 4.1. The primary engine/gas path is represented by components 17-20 and 22-25. Components 5,13,14,16 are the thermodynamic limiters and rotor speed scheduler used during the rating code assignment. The generic power controller (component #15) uses maps created from the defined ratings (created via the limiters and schedulers). The function of the maps has already been discussed but generally depends on the application. “When a deteriorated engine or transient performance is being studied, the function should be the engine control parameter”, (Visser, Shakariyants et al. 2007; SHAKARIYANTS 2008).

![Figure 4-1 Microturbine engine model in GSP](image)

Components 1, 9-12 are responsible for the scale effects and engine size and are for incorporating the influence of downscaling the turbomachinery components. Components 3,4,6,8 are responsible for the heat transfer effects that become noticeable below 9 kW. Component #2 is
responsible for looping the parameters power code and operating conditions in off design simulation. The remaining components are located in the control suit group. Component 21 in Figure 4-1 selects the fuel flow to maintain the value of the parameter scheduled by the engine controller. Component 7 is responsible for handling bleeds, (Visser, Shakariyants et al. 2007).

4.2.2 Rated Performance and Control

Turbines are usually flat rated on a net thrust (power) up to a ‘kink-point’ climate. Ratings define allowable power levels versus ambient temperature. Rating levels are designed to ensure satisfactory engine life, while achieving the power or thrust required by the application at key operating conditions. The power output of a turbine engine is limited by internal temperature and the rpm of its rotating components. If temperature, pressure or rpm exceed material limits then there can be permanent damage to the engine. As previously mentioned the power potential of the gas producing section of the engine is totally dependent on the density of the air it is operating in. If the environment can supply a dry, cool ISA standard day then the compressor can feed its maximum charge of air while using only low rpm and relatively low PR. If air becomes less dense, for instance air temperature is above ISA then the compressor works harder to supply the same air charge into the burner. The compressor must spin faster to do its work. At some point the density of the air available to the compressor will not be enough for it to deliver the full charge of air into the burner before reaching rpm, or temperature limits, or both, (Visser and Broomhead 2000; SHAKARIYANTS 2008).

4.2.2.1 Power Controller

Within the power controller component (15 in Figure 4-1) a power mode and power code value is specified for both design and off-design condition. The design power code value and power mode are used to define the scaling point. The scheduled parameter (T5, Torque etc.) can be directly given values with associated power codes or obtained from maps (created via limiters). Maps can be used for each power code and X, Y input parameters must be selected for the map. Maps are useful and can be user customized and created by GSP. Assuming the same temperature limit for all scales (engine sizes- 9, 15, 22, 30, 36 kW), maps were custom created for each power code (20, 60, 100%) by scheduling the torque and using a temperature limiter. This appropriately amended the “dead band” issue, creating maps of T5 at different power codes ensured values of
Tsa and Psa could be correctly interpolated at the defined code. Fixed values of T5 (EGT) assumed values from max power. The maximum temperature limit set on the tech level was 680 °C (limit for T5/EGT) and was used for final data generation to TNO. This represents a significantly higher metallurgical limit, which is well over the current Capstones C30 (650 °C max). The idea of using control schedules in the simulations is that it gives a better approximation of real engine control systems. “With a control schedule you can make maximum limiters dependent from the flight condition”, (Kurzke 2007).

4.2.2.2 Rating Code Assignments

The rating command control is a common industry practice. A standard is set by the SAE to create uniformity in engine performance modeling and presentation of results and so predefined power code (PC) and rating code (RC) assignments are defined, (SAE 1974). These standards are meant to define the thrust level and other performance parameters for engine ratings. The SAE power code represents the actual throttle position—the power lever angle. The rating code “permits selection of specified ratings that may require different power lever angles as flight or atmospheric conditions vary”, (SAE 1974).

4.2.3 Maximum (Power/thrust), Part-load and Idle Performance

The maximum power, part-load and idle performance, can be represented by a power code within the GSP GUI. 100% for max power and 20% for idle with percentages in-between representing part load. Part-load performance can sometimes be referred by variation in SFC with reduction in power. Vehicular gas turbines make frequent use of part load power setting and the poor SFC at part load is probably the biggest disadvantage of the gas turbine for vehicular use. When anything less than max power is required from a microturbine, the output is reduced by a combination of mass flow reduction (achieved by decreasing compressor speed) and EGT/TIT reduction. In addition to reducing power, this change in operating conditions also reduces efficiency. Gas turbine performance depends on the turbine inlet temperature (TIT) and typically the EGT. However, the actual limit depends on other factors also: ambient temperature and pressure, internal pressure level limit and shaft torque limit to name a few. In general, a 'flat rated' control concept is used, maintaining max torque and combustor pressure below the 'flat rated
temperature' and this can only be realized with decreasing max TIT/EGT limit. Therefore graphs were prepared to represent the flat rating schedule graphically for different ambient pressures.

![Figure 4-2 Flat Rated Temperature](image)

The situation whereby the engine is below its thermodynamic limits\(^\text{12}\) is the region of ambient temperature where the engine is "Flat Rated\(^\text{13}\)", a rise or fall in ambient temperature will have negligible effect upon thrust/power, it is "Flat", with the thrust/power available being quite constant, Figure 4-2. As much of the marketing is aimed at operations above ISA, the performance is generally rated to ISA+15 °C although early in the engine development this can be lower. The temperature below which thrust is flat rated varies significantly from engine design and the point ISA was used in the model. The efficiency and power for the Capstone C30 over a range of ambient temperatures can be seen in Figure A-1. Figure 4-3 demonstrates the net maximum power for the model vs. ambient temperature at sea level. As ambient temperature is increased further, the EGT limit will be reached and at this point (corner point), fuel flow and thrust/power must be decreased to remain within these engine temperature limits. The engine is

\(^{12}\) Thermodynamic limits include pressure limit (P3), EGT limit, spool limit (N1) etc.

\(^{13}\) A flat rated engine is one that is restricted to a maximum power even though it is capable of producing more due to lower ambient temperatures than ISA, higher density, etc. Therefore, at temperatures below the temperature at which the engine is flat rated to, the thrust is not affected by temperature.
now in a declining thrust/power situation with thrust being governed by the EGT limit. This is known as "Full Rating" as observed in Figure 4-2. The temperature limit to flat rated thrust is defined to be at ISA conditions. Below that temperature the engine is Flat Rated, above that temperature, it is Full Rated. Operating the engine right at the ambient temperature limit for the flat rating (corner point), stress is at its maximum, as the engine is simultaneously at its pressure and temperature limit. Figure 4-2 gives the “Flat Rated Temperature” of the Capstone C30 for an EGT of 923K, (Arun Prasad and Sundararjan 1985). The differences in magnitude of power decrease in simulated Figure 4-3 and Capstone C30 Figure A-1 could be because of an underestimation of pressure losses in the recuperator.

Figure 4-3 C30 Net Power vs Ambient Temperature at Sea Level

Methods of improving part-load performance
The part-load performance of gas turbines intended for vehicular or naval use is of great importance because of the considerable portion of the running time spent at low power. Some way of raising the TIT at low powers must be found if part-load efficiency of gas turbines is to be improved. A free turbine could be used in the majority of applications where good part-load economy is required. If a free turbine engine is used, the turbine inlet temperature at part-load can be increased by using variable-area power-turbine stators, (Saravanamutto, Rogers et al. 1996). Use of such a configuration has its pitfalls and was briefly discussed earlier in the report.
4.3 Data to Vehicle Model

Performance tables are required for different power settings or respectively different sizes of microturbines. Single tables with all parameters in columns will be used as a medium to supply the required data to the HEV model. Each scale has its own performance table and a number of tables will be generated for these series (e.g. 9, 15, 22, 30, 36 kW) for example. The Capstone C30 will be taken as a reference at 30kW. The format of the tables is defined below:

For the following combinations of inputs:

<table>
<thead>
<tr>
<th>Power Code [%]</th>
<th>Ambient Pressure [Pa],</th>
<th>Ambient Temperature [K]</th>
</tr>
</thead>
</table>

For the output parameters:

- Exhaust Gas Temperature [K]
- Exhaust Gas Mass flow [g/s]
- CO2 emissions [g/s]
- Fuel mass flow [g/s]
- Mechanical Output power [W]
- Electric Output power [W]

The range of ambient pressures and temperatures to be investigated are 248.15-323.15 K and 0.91325-1.1133 kPa. From here the HEV model uses the output data to formulate and define the framework of component selections, for which a low-power range extended electric vehicle is to be evaluated. TNO proposed a selection of component scenarios for the range of powers under investigation in the microturbine in Appendix F.2. The electric power is calculated via equation B-45 and the rest through iterating thermodynamical equations; some of which are listed in Appendix B. All the raw off-design data has been placed in tables in Appendix C.

4.4 Capstone C30 Effect graphs

4.4.1 Fixed Spool Speed

We are interested in variable ambient conditions at sea level and so this implies a calculation of steady state points at a series of different ambient temperatures, pressures and different power codes. We can also look at the advantages of a variable spool speed instead of just running at fixed speed. A few effect graphs for the output parameters requested are shown in Figure 4-4. SFC and CO2 are a function of $T_{amb}$ (Xaxis) and $P_{amb}$ (different curves within plot) and are shown below for the Capstone C30. The operating conditions varied are power code, ambient pressure and temperature. The results for fixed speed are represented in Figure 4-4 drawn in red.
Two output parameters (CO2, SFC) are shown for three power codes (PC) and three ambient pressures. The results in Figure 4-4 show the trends used for specifying performance at temperatures above the Flat Rated Temperature (FRT). Then, an ambient temperature ‘parameter sweep’ from 248.15 K up to 323.15 K is performed, (Visser and Broomhead 2000).

![Figure 4-4 Fixed Spool Speed](image-url)
4.4.2 Variable Spool Speed and Comparison

The Control Scheme can be altered to implement a variable load speed (instead of a fixed). The rotor speed can be adapted as function of Power Code (PC) using a 1D table scheduler. A schedule is defined such that; reducing N from a PC of 100 down to 60 % and then at that fixed speed reduce T5 (EGT) at a PC of 60 down to 20%. Figure 4-5 contains two graphs of both schemes. The region of interest for SFC has been blown up for comparison.

![Figure 4-5 Variable/Fixed Spool Speed](image-url)
The advantages associated with being able to vary speed upon decreasing to part load are numerous; however it depends on the environment. The SFC at idle load (20% power code) is very inefficient, as shown by Figure 4-5 in both speed schemes and is why turbines are not operated at this level. At part load (60% power code) the SFC with variable control scheme (blue graph) begins to increase past the fixed control (blue graph) as it crosses the corner point. This is noted in Figure 4-5. The $CO_2$ output is a lot lower at part and idle load for the variable speed control scheme. This is however not true for all emissions as $O_2$ and $N_2$ are higher at part load (both schemes). $CO_2$ decreases as the engine passes into the ‘full rated’ region of the ambient temperature. However, the emissions are not the sole selection criteria when designing the control system. Varying the shaft speed also increases the complexity of the power controller and substantially increases the cost of generator. While the fixed speed was a switch on/off configuration which reduces the life of a microturbine unit, the variable speed configuration has the advantage of being modulated. This would provide numerous advantages within the HEV configuration, including decreasing emission output at part load. It seems that at ambient temperatures above ISA, the fixed scheme would be selected on the basis of higher efficiency, power and SFC. At part load a variable control scheme is important if emissions are of concern and if it is operated in climates at or below ISA conditions.

4.5 Conclusion

Using refined relations and engine limitations a realistic representation of micro turbine/generator performance with state-of-the-art technology was developed. The completed model can be adapted quickly to represent more advanced or more moderate technology levels (with related implications on efficiency, weight, volume, cost, life etc.). A data method was developed to supply TNO with the generated data which is supplied in Appendix C. Maximum and variable speed were used and compared to understand the potential benefits for developing a variable speed configuration. If the power electronics become too complicated due to a variable speed adaption then one may be able to use a free turbine configuration. This would reduce some of the complexity involved electronically but as a consequence raise mechanical complexity. Heat transfer effects and bleed control was not incorporated in the study but may be adapted in the future for model refinement. These components may be needed to investigate possible losses in heat, if any, that exist throughout the components. If further study is to be undertaken for detailed
analyse one may investigate many of the various operational conditions for the current configuration and one important consideration is the engine condition.

### 4.6 Deterioration Effects and Emissions

The components within GSP include deterioration tab-sheets for specification of map modifiers which represent the effect of deterioration. To quantify deterioration effects literature is used. Deterioration effects are analyzed with steady state calculations for parameter sweeps with varying deterioration map modifiers, (*Visser, Shakariyants et al. 2007*).

Performance deterioration in gas turbines is due to:

- Fouling, due to minute dust particles, pollen, salt spray and insects, which are deposited on blade surfaces;
- Changes in blade surfaces due to erosion or fouling, and the effect on the blade aerodynamics both caused by particulate ingestion;
- Tip clearance increase of blade tips and seal geometries caused by particulate ingestion;
- Water ingestion during rain; and
- Changes in the combustion system (e.g. which result in different pattern factors).

It was determined that deterioration will be evaluated in future projects associated with this issue.

### 4.7 Improvements

#### 4.7.1 Further Study

More complex turbine designs eg. reheat and intercooling could have been explored to provide better overall cycle efficiencies and power output. Even though it doesn’t seem feasible due to the complexity involved, numerical investigations of intercooling and reheating might have been beneficial for the same conditions, to see the possible improvements in efficiency and specific power output.

Incorporation of steam injection by further utilizing the cycle’s waste heat could have been analysed. The exact increase in efficiency could have been compared to that of direct water
injection into the combustor. The injection of steam or water into the combustor boosts power output and/or controls the NOx emission levels, because it increases the fuel mass flow rate, but as a consequence raises the SFC and hence the price per kWh, (Kolanowski 2004).

The microturbine could have been adapted to incorporate a pneumatic system concept whereby a high and low pressure cycle is used to pressurize a vessel. The pressurized gas could then be used for mechanical motion. Heat transfer effects are likely to be an issue with this configuration.

If the variable control is to be implemented perhaps changing the configuration of the microturbine to incorporate a free turbine together with a power split hybrid configuration. The free turbine would eliminate the need for complicated power electronics therefore making shaft speed modulation easier due to the separate shaft. This may also allow for a power split configuration but a more complete investigation would have to be undertaken.

4.7.2 Improvements

Preliminary or Detailed Design could be taken a step further by utilizing software packages such as COMPAL®, RITAL™ and AxCent® whereby geometry design and fluid dynamic analysis could have been undertaken. Simulating experimental data with a unit on a test bench would have generated data that could be used for accuracy and further verification. The simulation results could have been compared to measured test data to validate the model. It is likely that the company MTT will investigate future developments with this technology in the future.
Chapter 5

Conclusion

The REI project (Range Extender Innovations) was initiated in the Netherlands in 2010 and is composed of a conservatorium of institutions. Toegepast Natuurwetenschappelijk Onderzoek (TNO) and Microturbine Technology (MTT) were two companies participating in the project and directly related to the thesis work. Microturbines seem to be a viable option for use as a range extender in Hybrid Electric Vehicles (HEVs). Capstone Corporation is the current world leader in microturbine research and development. The Capstone 30kW Drive Solution Microturbine (C30) was used as the reference of research and focus throughout the work conducted. The thesis work was concerned about finding the optimum size and corresponding performance of a microturbine, design and simulation are therefore essential for preliminary stages of analysis. Gas Turbine Simulation Software (GSP) was used to build a numerically component based microturbine model whereby sizes in the power range of 9, 15, 22, 30, 36 kW were investigated. Output variables such as shaft/electric power, carbon dioxide output, fuel mass flow, exhaust mass flow and exhaust gas temperature were supplied to TNO in data table format. The associated geometric size, weight and cost of the different sizes of the microturbine systems gave a potential breakdown of the physical dimensions. Turbine characteristics such as the diameter of the compressor and turbine impeller with associated efficiency are shown in Figures 2-13, 2-14, together with the design method in Appendix B.6 the shape number can be calculated for every power level. The turbomachinery weight and its variation with shaft output power is shown in Figure 2-15. A proposed weight for the whole microturbine at each power level was supplied to TNO for incorporation into their system. The weights for the microturbine system were 35 – 110 kilograms for the defined power range. The data supplied by the microturbine model allowed the
HEV system and its respective components to be sized and optimized appropriately. Microturbine capital costs in 2002 are between $775-$1,450/kW and so the complete costs (lower/upper limits) for the units in the interested power range would be 6,975- 52,200 USD, Table 2-1. This therefore is only feasible if the market expands and sales volumes increase therefore decreasing capital costs. The lowest allowable power level conforming to the HEV constraints, in the power range investigated is clearly desired.

Performance data (especially efficiency) is strongly affected by turbine cycle and configuration, Turbine Inlet Temperature (TIT), Compressor Pressure Ratio (PRc), Multi/single spool, Component performance efficiencies and Scale effects. During the initial design phase optimizing the architecture within the microturbine was explored at each scale to investigate the influence on thermal efficiency and power output. Varying the TIT for a selection of compressor pressure ratios at each power level allowed the construction of carpet plots. The carpet map corresponding to Capstones C30 design values gave a thermal efficiency of 26.5% at optimum conditions (EGT at 650 °C and PRc 3.5) which corresponds close to published data. There is room improvement regarding TIT as Figure 3-5 suggests. Focus towards developing the hot end components of a microturbine system is the most important issue when optimizing microturbine architecture. The Specific Fuel Consumption (SFC) is important and differs in fuel selection. Microturbines have the ability to utilize a variety of fuels and so fuel cost analysis was investigated. For the Capstone C30 design conditions three fuels were tested and graphed in Figure 3-2 to 3-4. LPG was the cheapest fuel to use per kWh (0.42 €/kWh) followed by diesel then CNG for the Netherlands. A careful choice must be made on the selection of fuel; a decision based on cost and associated emissions.

The optimum size was found by design point calculations for each power level but the performance data at the defined operational conditions (Ambient temperature, pressure and power code) was also needed. The performance at maximum, part and idle load together with fixed and variable shaft load control speed schemes were important considerations for the HEV simulations. Extension of the core design model and development of control system provided the tools needed to investigate off-design data generation. The data for fixed and variable speed schemes are depicted in Figures 4-4 and 4-5. In comparison fixed speed is less expensive and requires less complex power electronics. The microturbine may also have a shorter lifespan due
to the higher number of shutdowns. It has a lower SFC at part load and is more efficient above ISA conditions. *Variable speed* schemes cost more and are more complex however exhibit considerably lower emissions. If the turbine can be operated at or below ISA conditions the variable speed scheme has the complete advantage in respect to efficiency, SFC and shaft power.

The final configuration of the microturbine is dependent on many factors and governmental future prospects. The variable speed control scheme was chosen in the microturbine configuration and data was simulated and sent to TNO. As a result the company was successfully able to size and simulate data for three vehicle segments (Appendix F.2.4) which allowed characteristics on the drive power, battery size and range extender power to be derived. However, manufacturing and material selection, constrains of HEV system, goals for higher efficiency and lower emissions, machinery life span, large production volume costs and their benefits need to be closely evaluated for future development.
Glossary of Terms

**Air–fuel ratio AF**  Quantifies the amount of fuel and air usually on a mass basis.

**Closed system**  A system with a fixed amount of mass (control mass), and no mass can cross its boundary. However energy, in the form of heat or work, can cross the boundary.

**Compressor**  A device used for increasing the pressure and density of gas.

**Combustion**  The rapid oxidation of fuel gases accompanied by flame and the production of heat, or heat and light.

**Efficiency**  A measure (usually a ratio) of the useful energy provided by a dynamic system versus the total energy supplied to it during a specific period of operation.

**Exhaust heat**  Waste heat produced by a mechanical, chemical, or electrochemical process.

**Exhaust heat recovery**  The use of by-product heat as a source of energy.

**Fuel cell**  An electrochemical device that can continuously convert the chemical energy of a fuel and an oxidant to electrical energy. The fuel and oxidant are typically stored outside of the cell and transferred into the cell as the reactants are consumed.

**Flat Rated Temperature**  Temperature limit to flat rated thrust/power is at ISA conditions (15 degrees Celsius). Below that temperature the engine is Flat Rated, above that temperature, it is Full Rated. If you happen to be operating the engine right at the ambient...
temperature limit for the flat rating, stress is at its maximum, as the engine is simultaneously at its pressure and temperature limit. This is commonly referred to as the corner point of the engine.

**Flat Rating**  Relatively constant power or thrust is offered over a range of ambient temperature.

**Heat exchanger**  A vessel in which heat is transferred from one medium to another.

**Incomplete combustion**  is a combustion process in which the combustion products contain any unburned fuel or components such as C, H₂, CO, or OH.

**Ideal cycle**  Is an actual cycle stripped of all the internal irreversibilities and complexities. The ideal cycle resembles the actual cycle closely but is made up totally of internally reversible processes.

**Intercooler**  A heat exchanger for cooling gas between stages of a multistage compressor with a consequent saving in power.

**Mechanical efficiency**  of a device or process is the ratio of the mechanical energy output to the mechanical energy input.

**Microturbine**  Machines whose power outputs are in the range between 30 kW and 250 kW.

**Miniturbine**  Machines whose power outputs are above 250 kW – 500 kW.

**Process heat**  is required energy input in the form of heat for many industrial processes. The process heat is often obtained as heat transfer from high-pressure, high-temperature working fluid.
Shaft work is energy transmitted by a rotating shaft and is the related to the torque applied to the shaft and the number of revolutions of the shaft per unit time.

Specific gas power Is the gas power per unit of mass flow.

Stage One rotor and one stator make up a stage in a compressor or turbine.

Stoichiometric combustion (theoretical combustion) is the ideal combustion process during which a fuel is burned completely with theoretical air.

Thrust-to-weight ratio Is a dimensionless quantity and is an indicator of the performance of the engine or vehicle.

Turboshift In a Turboshift nearly all the energy in the exhaust is extracted to spin the rotating shaft.

Turbine A rotary engine whereby it extracts energy from a continuous flow of combustion gases.

Thermal efficiency of a power plant For shaft power engines is defined as the ratio of the engine power output of the turbine to the rate of fuel energy input.

Waste heat Is energy that must be dissipated to the atmosphere from a process.

Carpet Map A performance plot.

Maximum Power Maximum ratings correspond to a maximum allowable value of some limiting parameter for engine integrity.
Part Power A part load rating is generally set to be some fraction of the maximum rating, broadly to achieve a set percentage of defined engine capability.

Idle This is the minimum feasible level of power or thrust, produced when the ideal requirement of the application is actually zero.

Heat Sink component (GSP) To represent the heat transfer effects through various components.

Component Map A majority of GSP components use “maps” to represent multi-dimensional non-linear component characteristics. The maps are stored in a text file format and can be used by a wide variety of programs. They are read into the component’s memory upon model initialization. All maps will be scaled before the off-design calculation commences, and will be consistent with the cycle design point, (Kurzke 2007).

Scheduler Map 2-D Map Schedule Control: A 2-dimensional map is used by this component to schedule the selected output parameter or component property. The map is read from a file (GasTurb/MTU map format similar to the compressor maps etc.). Select input parameters X and Y from the drop down lists on the Map tab sheet. The value for the output parameter or component property will be inter/extrapolated from the map, (Visser, Michiel et al. 2010).

Power controller The General and Design tab sheets specify the off-design and design conditions for the power controller respectively. The power modes are defined on the Schedule tab sheet.

“The Schedule tab sheet allows the specification of the schedules. The Scheduled Parameter dropdown list allows the selection of the parameter that is scheduled (e.g. a rotor speed or
EGT parameter). In the table, columns are defined for Power code and one or more power/operating modes”, (Visser, Michiel et al. 2010).

The power control component is able to set a certain power setting as function of map scheduled parameters. “The scheduled parameter can also be obtained from a map”, (Visser, Michiel et al. 2010)
Cited Works


GCC (2010) "Electric (battery)." Green Car Congress.


Bibliography

http://www.globalsecurity.org/military/library/policy/army/accp/ac0993/le1.htm


Appendix A

This appendix contains all the design data found for the Capstone C30 and some information about various other microturbines.

A.1 GSP Model Initial Conditions

Table A-1: Design Conditions

<table>
<thead>
<tr>
<th>Static/Total Ambient Conditions</th>
<th>Combustor</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{s,t} = 101,325 \text{Pa}$</td>
<td>$Fuel \text{ flow } (W_f) = \text{Variable}$</td>
</tr>
<tr>
<td>$T_{s,t} = 288.15 \text{K}$</td>
<td>$Exit \text{ temperature } (T_3) = \text{Variable}$</td>
</tr>
<tr>
<td>Relative humidity = 60%</td>
<td>$Design \text{ combustion efficiency } = 0.999 \text{ [\text{\textendash}]}$</td>
</tr>
<tr>
<td><strong>Load</strong></td>
<td>$Design \text{ point rel. pressure loss } = 0.0200 \text{ [\textendash]}$</td>
</tr>
<tr>
<td>Design Power = 30kW</td>
<td><strong>Turbine</strong></td>
</tr>
<tr>
<td><strong>Inlet</strong></td>
<td>$Design \text{ rotor speed } = 96,164.5 \text{ [rpm]}$</td>
</tr>
<tr>
<td>Design mass flow = 0.313302 $\frac{Kg}{s}$</td>
<td>$Design \text{ efficiency } = 82%$</td>
</tr>
<tr>
<td>Pressure ratio = 1 [-]</td>
<td>$Expansion \text{ heat loss fraction } = 0.50 \text{ [-]}$</td>
</tr>
<tr>
<td><strong>Compressor</strong></td>
<td>Tip diameter: 16.95cm</td>
</tr>
<tr>
<td>Design Rotor Speed = 96,164.5 [rpm]</td>
<td>Nsc = 0.707</td>
</tr>
<tr>
<td>Design Pressure Ratio = Variable</td>
<td>Spool</td>
</tr>
<tr>
<td>Design Efficiency = 75 %</td>
<td>Spool inertial moment = 0.7578 [kgm^2]</td>
</tr>
<tr>
<td>Heat transfer fraction = 0.50 [-]</td>
<td>Spool mechanical efficiency = 0.980 [-]</td>
</tr>
<tr>
<td>Tip Diameter: 14.7cm</td>
<td><strong>Exhaust</strong></td>
</tr>
<tr>
<td>Recuperator</td>
<td>$Velocity \text{ coefficient } CV = 1.0 \text{ [-]}$</td>
</tr>
<tr>
<td>Effectiveness = 0.83 [-]</td>
<td>$Thrust \text{ coefficient } CX = 1.0 \text{ [-]}$</td>
</tr>
<tr>
<td>Rel.total pressure loss:</td>
<td>$Throat \text{ CD } = 1.0 \text{ [-]}$</td>
</tr>
<tr>
<td>Flow 1 = 0.015 [-]</td>
<td><strong>Duct</strong></td>
</tr>
<tr>
<td>Flow 2 = 0.040 [-]</td>
<td>$Heat \text{ flux (input)} = 3.14995 \text{ [kW]}$</td>
</tr>
<tr>
<td>Relative total pressure loss = 0 [-]</td>
<td>Relative total pressure loss = 0 [-]</td>
</tr>
</tbody>
</table>

(Capstone 2009) and Colin Rodgers
The Design conditions used for the models will be noted in Table A-1: Design Conditions above. All components are on the same shaft and values are taken from Capstones C30 microturbine to simulate the model with the most accurate conditions.

**A.2 Performance Parameters**

The following performance parameters are the official published performance data for Capstones C30 microturbine seen in Table A-2: Published Performance output data for the Capstone C-30. It is stated here purely as a comparative medium for the data which was simulated by the models. A majority of the values, especially the thermal efficiency, are rated slightly higher than the experimental values calculated. This is possibly due to more accurate results obtained by test rig analysis. From this table and my graphs it is possible to see that the data corresponds fairly accurately and that the C30 recuperated microturbine is still only touching the beginning of the performance curves.

**Table A-2: Published Performance output data for the Capstone C-30**

<table>
<thead>
<tr>
<th>Capstone C30</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output power (kW)</td>
</tr>
<tr>
<td>Electrical efficient (%)</td>
</tr>
<tr>
<td>Mass flow rate (kg/s)</td>
</tr>
<tr>
<td>Pressure ratio</td>
</tr>
<tr>
<td>Axial speed (rpm)</td>
</tr>
<tr>
<td>TIT (°C)</td>
</tr>
<tr>
<td>Exhaust Temperature (°C)</td>
</tr>
<tr>
<td>Fuel</td>
</tr>
<tr>
<td>27</td>
</tr>
<tr>
<td>26.1 (± 2)</td>
</tr>
<tr>
<td>0.31</td>
</tr>
<tr>
<td>3.5</td>
</tr>
<tr>
<td>96,164.5</td>
</tr>
<tr>
<td>840</td>
</tr>
<tr>
<td>275</td>
</tr>
<tr>
<td>Gaseous Propane/ Natural Gas/Diesel</td>
</tr>
</tbody>
</table>

(2003) and (Capstone 2009)
A.3 Various Microturbines from Literature

Table A-3 Microturbines representing today’s tech level

| Cycle Parameters: Sea level 27°C. Inlet heating 3°C. EGT680°C. Recuperator ε=85%. Pressure loss 8% leakage 2%, generator efficiency 85% |
|---|---|---|---|
| Output Kwe | 25 | 10 | 5 |
| Pressure Ratio | 3.5 | 3.5 | 3 |
| Compressor Effy % | 77.5 | 76.5 | 75.3 |
| Mechanical Effy % | 96 | 94 | 87 |
| Turbine Effy % | 85.2 | 85 | 83.8 |
| Speed krpm | 100 | 130 | 150 |
| $ET_{AH}$ % | 27 | 26 | 22.5 |
| Airflow Kg/s | 0.22 | 0.15 | 0.07 |
| Compressor Diameter mm | 150 | 114 | 92 |

(Rodgers 2000)

A.4 Capstone Drive Solution Component Data

C30 Microturbine; Light-Duty Class 4/5 Application

<table>
<thead>
<tr>
<th>Electrical Power Output</th>
<th>30kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage</td>
<td>550-650 VCD</td>
</tr>
<tr>
<td>Electrical Efficiency</td>
<td>26%</td>
</tr>
<tr>
<td>Fuel Options</td>
<td>Natural Gas, (Bio)Diesel</td>
</tr>
<tr>
<td>Exhaust Gas Flow</td>
<td>0.3 kg/s</td>
</tr>
</tbody>
</table>

| Turbo-generator |
|---|---|
| Dimensions | 57.2 x 72.9 x 83.6 cm |
| Weight | 91 kg |

| Microturbine Inverter |
|---|---|
| Dimensions | 28.5 x 15.5 x 55.6 cm |
| Weight | 23 kg |

| Traction Motor |
Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Continuous Power</td>
<td>85kW</td>
</tr>
<tr>
<td>Maximum Torque</td>
<td>425 Nm</td>
</tr>
<tr>
<td>Efficiency</td>
<td>97%</td>
</tr>
<tr>
<td>Dimensions (WxHxD)</td>
<td>26.7 x 26.7 x 38.1 cm</td>
</tr>
<tr>
<td>Weight</td>
<td>115 kg</td>
</tr>
</tbody>
</table>

Motor Drive Inverter

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak Output Current</td>
<td>1000A - 450A</td>
</tr>
<tr>
<td>Operating Voltage</td>
<td>550 – 650 VCD</td>
</tr>
<tr>
<td>Switching Frequency</td>
<td>4 – 8 kHz</td>
</tr>
<tr>
<td>Dimensions (WxHxD)</td>
<td>28.5 x 15.5 x 55.6 cm</td>
</tr>
<tr>
<td>Weight</td>
<td>13 kg</td>
</tr>
</tbody>
</table>

(Corporation 2011)

C65 Microturbine; Heavy-Duty Class 8 Application

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical Power Output</td>
<td>65kW</td>
</tr>
<tr>
<td>Voltage</td>
<td>550 – 650 VCD</td>
</tr>
<tr>
<td>Electrical Efficiency</td>
<td>29%</td>
</tr>
<tr>
<td>Fuel Options</td>
<td>Natural Gas, (Bio)Diesel</td>
</tr>
<tr>
<td>Exhaust Gas Flow</td>
<td>0.49kg/s</td>
</tr>
<tr>
<td>Turbo-generator</td>
<td></td>
</tr>
<tr>
<td>Dimensions</td>
<td>66.6 x 75.9 x 83.3cm</td>
</tr>
<tr>
<td>Weight</td>
<td>135kg</td>
</tr>
</tbody>
</table>

Microturbine Inverter

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Dimensions</td>
<td>28.5 x 15.5 x 55.6 cm</td>
</tr>
<tr>
<td>Weight</td>
<td>23 kg</td>
</tr>
</tbody>
</table>

Traction Motor

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Continuous Power</td>
<td>200kW</td>
</tr>
</tbody>
</table>
### Maximum Torque

<table>
<thead>
<tr>
<th>Maximum Torque</th>
<th>670 Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency</td>
<td>97%</td>
</tr>
<tr>
<td>Dimensions (WxHxD)</td>
<td>26.7 x 26.7 x 48.3 cm</td>
</tr>
<tr>
<td>Weight</td>
<td>143 kg</td>
</tr>
</tbody>
</table>

### Motor Drive Inverter

<table>
<thead>
<tr>
<th>Peak Output Current</th>
<th>1000A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating Voltage</td>
<td>550 – 650 VCD</td>
</tr>
<tr>
<td>Switching Frequency</td>
<td>4 – 8 kHz</td>
</tr>
<tr>
<td>Dimensions (WxHxD)</td>
<td>43.3 x 52.7 x 51.5cm</td>
</tr>
<tr>
<td>Weight</td>
<td>36kg</td>
</tr>
</tbody>
</table>

(Corporation 2011)

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**Figure A-1 C30 and C65 Power and Efficiency**
Appendix B

This appendix provides a detailed outline of the various equations used in the thermodynamics within GSP. An analytical derivation of the specific power and efficiency gives the reader an idea of what the dependent variables are within the relations.

B.1 Fuels

Microturbines use gaseous or liquid fuels and it is the type and degree of combustion that determines the amount of emissions released. Fuel choice is based on the heating value, purity, viscosity and moisture content particularly in liquid fuels, (Kolanowski 2004). Compressed natural gas (CNG) is a possible candidate but currently capstone’s microturbines currently use LPG, diesel or biodiesel. Some of the other possible choices for fuel are Natural gas, LNG, Renewable landfill and digester gases, kerosene, JP8, coalbed methane and flare gases (sweet and sour-- unprocessed natural gas; comes directly from some gas wells). The following will note some more important performance parameters concerning fuel/air ratios, Specific fuel consumption, fuel cost and associated emissions.

In engine testing, both the air mass flow rate, $\dot{m}_a$, and fuel mass flow rate, $\dot{m}_f$, are normally measured. The ratio of these flow rates is useful in defining engine operating conditions:

fuel/air ratio

$$\left(\frac{F}{A}\right) = \frac{\dot{m}_f}{\dot{m}_a}$$

and air/fuel ratio

$$\left(\frac{A}{F}\right) = \frac{\dot{m}_a}{\dot{m}_f}$$

The stoichiometric fuel/air ratio and is a measure of how far combustion is complete. Different fuels have various stoichiometric combustion ratios for example, gasoline’s stoichiometric
fuel/air ratio is 0.0685 (air/fuel ratio: 14.6). However it is more likely that fuel/air equivalent ratio, $\varphi$, and air/fuel equivalent ratio, $\lambda$, would be used to specify when a stoichiometric mixture has been reached, (Ehsani, Gao et al. 2010). It is therefore easy to determine what the ideal mass fuel flow rate for a particular fuel is for its stoichiometric combustion. This is important because we want combustion complete as possible, so to minimize emissions.

The shaft power Specific Fuel Consumption (SFC) is the fuel flow rate per useful power output. It measures how efficiently an engine is using the fuel supplied to produce work.

$$\text{SFC} = \frac{\dot{m}_f}{P_{shaft}} = \frac{1}{\eta_{th} LHV_f}$$

With

$$\eta_{th} = \frac{P_{shaft}}{LHV_f \dot{m}_f}$$

It is important to quote the calorific value of the fuel and whether it is the higher or lower heating value$^{14}$. SFC is commonly expressed in the units of $[\frac{kg}{kWh}]$ and lower values are clearly desirable.

We can see from equation that if the cycle efficiency is held constant, increases in the LHV (better fuel selection in terms of heating value) reduces the fuel consumption of the microturbine.

Fuel prices are expressed in Euros per liter $[\frac{\text{€}}{l}]$ and so the specific fuel consumption needs to be divided by the liquid density of the fuel used for combustion and multiplied by 3600 to adapt the units in terms of hours.

$$\text{SFC} = \frac{\dot{m}_f 3600}{\rho_f P_{shaft}} \text{ or } \frac{3600}{\rho_f \eta_{thermal} LHV_{fuel}}$$

B-1

Which will give the units of $[\frac{l}{kWh}]$.

Now if we define $C_f$ and the cost of the fuel per liter then the price of the fuel in $[\frac{\text{€}}{kWh}]$ is

$^{14}$ Fuel should be compared on a net calorific value (LHV) since most burning gases cannot utilize the heat content of the water vapor. The LHV is simply the HHV minus the energy from condensing the water vapor that results from combustion.
An investigation on different fuels was briefly presented in chapter 3.

**B.1.1 Fuel**

Within the GSP environment the combustor component allows specification of fuel choice and properties/composition. This enables analysis on effects of alternative fuels on gas turbine performance and emissions. GSP has a flexible user interface for specifying fuel properties, in terms of either

- specification of hydrogen/carbon (H/C) ratio and heating value, or,
- explicit specification of composition.

Quite a number of fuels have many different species and specification of all specie concentrations is not recommended. The H/C ratio option is used with the heating value specified, however if this isn’t known then one can always explicitly specify the fuel in terms of its composition. The H/O option is used in this thesis because it allows for easy specification of fuels. The resulting combustion gas composition is calculated using the H/C ratio.

**B.1.2 Fuel Property Data**

The fuel property data for the analysis is taken from Table B.1. Fuel prices are taken from the period of May 2011 and are therefore valid for comparative purposes, all prices are noted from, *(Neutral 2010).*
Gas Turbine Cycles—Analytical Analysis

The Gas Turbine can be separated into three primary components which are diagrammatically shown below in Figure B-1. The compressor, combustor and the turbine creates the basic configuration which makes up the simple Brayton-Joule cycle, (Saravanamuttoo, Rogers et al., 1996). A thermodynamic analysis for ideal and actual turbines concerning the unrecuperated (simple) and recuperated cycles is presented throughout the following sections.

Ideal analyses and actual analyses differs with regards to accounting for component efficiencies, pressure losses and irreversibilities alike. The ideal cycle consists of two isobaric processes and two isentropic processes. It is used for gas turbines, which operate on an open cycle. The open gas-turbine cycle can be modeled as a closed cycle by utilizing the air-standard assumptions. We will use the closed cycle as an ideal model, a comparative tool for real gas turbines characteristics, where trends represented behave in the same manner, but with a lower efficiency due to various losses. These subjects will be discussed in the following sections. The combustion process can be seen as a constant-pressure heat-addition process from an external source, and the exhaust process replaced by a constant-pressure heat-rejection process to the ambient air. The

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Table B-1 Properties of conventional and alternative fuels

<table>
<thead>
<tr>
<th>Property</th>
<th>Gasoline</th>
<th>Diesel</th>
<th>Methanol</th>
<th>Ethanol</th>
<th>Propane (LPG)</th>
<th>Methane (CNG)</th>
<th>RME</th>
</tr>
</thead>
<tbody>
<tr>
<td>H/C ratio</td>
<td>1.9</td>
<td>1.88</td>
<td>4.0</td>
<td>3.0</td>
<td>2.7</td>
<td>4.0</td>
<td>n.a.</td>
</tr>
<tr>
<td>Energy content (LHV) (MJ/kg)</td>
<td>44.0</td>
<td>42.5</td>
<td>20.0</td>
<td>26.9</td>
<td>46.4</td>
<td>50.0</td>
<td>36.8</td>
</tr>
<tr>
<td>Liquid density (kg/l)</td>
<td>0.72-0.78</td>
<td>0.84-0.88</td>
<td>0.792</td>
<td>0.785</td>
<td>0.51</td>
<td>0.422</td>
<td>0.86-0.90</td>
</tr>
<tr>
<td>Liquid energy density (MJ/l)</td>
<td>33.00</td>
<td>36.55</td>
<td>15.84</td>
<td>21.12</td>
<td>23.66³</td>
<td>21.13b</td>
<td>32.4-33.1</td>
</tr>
<tr>
<td>Boiling point (°C)</td>
<td>37-205</td>
<td>140-360</td>
<td>65</td>
<td>79</td>
<td>-42.15</td>
<td>-161.6</td>
<td>n.a.</td>
</tr>
<tr>
<td>Research Octane Numbers</td>
<td>92-98</td>
<td>-25</td>
<td>106</td>
<td>107</td>
<td>112</td>
<td>120</td>
<td>n.a.</td>
</tr>
<tr>
<td>Motor Octane Numbers</td>
<td>80-90</td>
<td>-</td>
<td>92</td>
<td>89</td>
<td>97</td>
<td>120</td>
<td>n.a.</td>
</tr>
<tr>
<td>Cetane Numbers</td>
<td>0-5</td>
<td>45-55</td>
<td>5</td>
<td>5</td>
<td>-2</td>
<td>0</td>
<td>45-59</td>
</tr>
<tr>
<td>Stoichiometric air-fuel ratio</td>
<td>14.7</td>
<td>14.6</td>
<td>6.5</td>
<td>9.0</td>
<td>15.7</td>
<td>17.2</td>
<td>13.0</td>
</tr>
<tr>
<td>Reid Vapor Pressure (psf)</td>
<td>8-15</td>
<td>0.2</td>
<td>4.6</td>
<td>2.3</td>
<td>208</td>
<td>2,400</td>
<td>0.5</td>
</tr>
</tbody>
</table>

n.a. = Not available

Notes: LHV = lower heating value; LPG = liquefied petroleum gas; RME = rapeseed methyl ether; CNG = compressed natural gas.
a. Energy density of propane at standard temperature and pressure: 0.093MJ/l.
b. Energy density of methane at standard temperature and pressure: 0.036MJ/l; at 200 bar pressure: 70.4MJ/l.

**B.2 Gas Turbine Cycles—Analytical Analysis**

15 Exhaust gases leaving the turbine in the open cycle are not re-circulated.
The ideal Brayton cycle is made up of four internally reversible processes: The system for numbering the cycle differs slightly to the standard mentioned in Appendix D.

1-2 Isentropic compression (in a compressor),
2-3 Constant pressure heat addition,
3-4 Isentropic expansion (in a turbine),
4-1 Constant pressure heat rejection.

**Figure B-1 Simple Idealised Brayton Cycle** (Boles 2006)

**Air-standard assumptions** reduce the analysis of gas power cycles to a manageable level by utilizing the following approximations:

1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal perfect gas (density and molecular weight of fluid is constant).
2. All the processes that make up the cycle are adiabatic and internally reversible.
3. There are no losses in any of the components including no leakages.

---

16 The fresh ambient air has already been processed through other turbine components such as the inlet, ducts and generator.
17 Isentropic: Processes held at constant volume or constant pressure.
18 Internally reversible processes: Thermodynamics states that, for given temperature limits, a completely reversible cycle has the highest possible efficiency and specific work output, reversibility being both mechanical and thermal. Mechanical reversibility is a succession of states in mechanical equilibrium, i.e. fluid motion without friction, turbulence, or free expansion. Thermal reversibility is a consequence of the Second Law of thermodynamics, which states that heat must be added only at the maximum temperature of the cycle and rejected at the minimum temperature.
4. The specific heat capacities and heat capacity ratio are at constant pressure and constant in value throughout cycle
5. The combustion process is replaced by a perfect heat-addition process from an external source (complete combustion); and
6. The exhaust process is replaced by a heat rejection process that restores the working fluid to its initial state.

The simple cycle can be altered by the addition of other components which will increase the net power output and improve the efficiency. The configuration will depend on the application at hand. A heat exchanger is used to make use of the exhaust gases to preheat the air before it enters the combustor, which effectively reduces the fuel needed for the same output.

- Higher-pressure ratios and turbine inlet temperatures improve efficiencies on the simple-cycle gas turbine.
- Finally co-generation utilizes waste heat from the turbine and converts it into process heat and usable power.

**B.2.1 First Law Analysis of the Brayton Cycle**

\[
\frac{dE_{cv}}{dt} = \dot{Q}_{in} - \dot{W}_{out} + \sum_{in} \dot{m}(h + \frac{v^2}{2} + gz)_{in} - \sum_{out} \dot{m}(h + \frac{v^2}{2} + gz)_{out} \tag{B-3}
\]

Assumptions will be made to obtain the appropriate equations. The Brayton cycle for turbines can be modeled as a steady flow process, so the change in energy is zero \( \frac{dE_{cv}}{dt} = 0 \). The elevation difference is negligible due to the dimensions of the microturbine \( \Delta z \approx 0 \). The change in kinetic energy is negligible, even if the fuel mass flow is added into the working fluid \( \Delta v \approx 0 \).

\[
0 = \dot{Q}_{in} - \dot{W}_{out} + \dot{m}(h_{in} - h_{out})
\]

\[
\dot{Q}_{in} - \dot{W}_{out} = \dot{m}_{air, gas}(h_{out} - h_{in})
\]

\[
q_{in} - w_{shaft} = (h_{out} - h_{in}) \tag{B-4}
\]

with
The rate of heat $q_{in}$ is the heat supply to the system. The rate of work done $w_{out}$ (is shaft work, in this case) by the working fluid. The mass flow rate $\dot{m}$ of the working fluid\(^{19}\) and $h$ is the enthalpy.

We know that if the working fluid is in equilibrium then $\Delta h = c_p \Delta T$ and so this will be used in the following equations.

It is necessary to define the mechanical power and heat exchange of the components in the ideal Brayton cycle and if equation B-4 is applied to the various processes throughout the cycle with reference to Figure B-1 we obtain:

The mechanical power output of the Turbine:

$$W_{3-g} = W_T = \dot{m}_{gas} c_{p,\text{gas}} (T_3 - T_g)$$

And so the specific mechanical work output of the Turbine:

$$w_{3-g} = c_{p,\text{gas}} (T_3 - T_g) \quad \text{B-5}$$

The gas power (shaft power)

$$W_{s,gg} = W_{shaft} = W_{g-4} = \dot{m}_{gas} c_{p,\text{gas}} (T_g - T_4)$$

And so the specific gas work output of the shaft

$$w_{g-4} = c_{p,\text{gas}} (T_g - T_4) \quad \text{B-6}$$

The mechanical power input of the Compressor:

$$W_{1-2} = W_c = \dot{m}_{air} c_{p,\text{air}} (T_1 - T_2)$$

And so the specific mechanical work input of the compressor:

$$w_{1-2} = c_{p,\text{air}} (T_1 - T_2) \quad \text{B-7}$$

\(^{19}\) Specific work ($w$) and heat ($q$) quantities used in the calculations have been expressed as energy divided by unit mass flow of air/gas. A distinction will be made and noted, when appropriate between the air mass flow and the gas mass flow within the system, as the fuel will vary as turbine inlet temperature $T_3$ is increased. Hence it is noted that $m_a + m_f = m_g$. (Assuming no leakages)
The combustor (heat input rate):\
\[ \dot{Q}_{2-3} = \dot{m}_f c_p (T_3 - T_2) \]

Or specific heat input

\[ q_{2-3} = c_p (T_3 - T_2) \]  \hspace{1cm} B-8

Waste Heat:

\[ \dot{Q}_{4-1} = \dot{m}_{gas} c_{p,\text{gas}} (T_4 - T_1) \]

Or specific waste heat

\[ q_{4-1} = c_{p,\text{gas}} (T_4 - T_1) \]  \hspace{1cm} B-9

The ideal (isentropic) gas equation can be used as we assume that the processes 1-2 and 3-4 are isentropic:

\[
\frac{p_2}{p_1} = \left( \frac{T_2}{T_1} \right)^{\frac{r}{r-1}} \quad \text{or} \quad \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{r-1}{r}} \quad \text{and} \quad \frac{T_3}{T_4} = \left( \frac{p_3}{p_4} \right)^{\frac{r-1}{r}}
\]

In the ideal cycle pressure losses are negligible

\[
\frac{p_2}{p_1} = \frac{p_3}{p_4} = \text{constant} = \Pi
\]

\[
T_2 = T_1 (\Pi)^{\frac{r-1}{r}} \quad \text{and} \quad T_3 = T_4 (\Pi)^{\frac{r-1}{r}}
\]  \hspace{1cm} B-10

We note that for the Ideal Brayton cycle the net work output \( w_{\text{net}} \) of the system is (where \( c_{p,\text{gas}} = c_{p,\text{air}} = c_p \)):

\[
w_{s,gg} = c_p (T_g - T_4) = w_t - w_c = c_p (T_3 - T_4) - c_p (T_1 - T_2)
\]  \hspace{1cm} B-11

If equations B-8 and B-11 are substituted into the definition of cycle thermodynamic efficiency then we get
\[ \eta_{th, ideal} = \frac{\text{net work output}}{\text{heat supplied}} = \frac{w_{net}}{q_{in}} = \frac{c_p(T_3 - T_4) - c_p(T_1 - T_2)}{c_p(T_3 - T_2)} \]

\[ \eta_{th, ideal} = \frac{(T_3 - T_4) - (T_1 - T_2)}{(T_3 - T_2)} = 1 - \frac{1}{\Pi^{\gamma-1}} \quad \text{B-12} \]

Under the cold-air-standard assumptions the thermal efficiency of an ideal Brayton cycle only depends on the specific heat ratio of the working fluid and the pressure ratio of the gas turbine. This is shown in equation B-12. As it can be seen, the thermodynamic efficiency of the ideal cycle increases dramatically as the pressure ratio is increased.

It should be noted however that the increase in pressure ratio increases the overall efficiency at a set firing temperature. If one increases the pressure ratio beyond a certain optimum value at any set firing temperature then the overall cycle efficiency can be decreased, (Boyce 2002). Another way to achieve higher efficiencies is by using heat exchangers.

The Specific Work Output decides the size of a gas turbine power plant for a given power. The specific work output is given by modifying equation B-6 for net work output into a non-dimensional form as follows

\[ \frac{w_{sgg}}{c_p T_1} = \left( \frac{T_3}{T_1} - \frac{T_4}{T_1} \right) + \left( 1 - \frac{T_2}{T_1} \right) \quad \text{B-13} \]

We want to get this equation in terms of a reference temperature, the turbine inlet temperature and the pressure ratio and so we use equation B-10 into equation B-13

\[ \frac{w_{net}}{c_p T_1} = \left( \frac{T_3}{T_1} - \frac{1}{T_1} \cdot \frac{T_3}{T_1} \cdot \frac{T_3^{\gamma-1}}{\Pi^{\gamma-1}} \right) + \left( 1 - \frac{T_1 \cdot \Pi^{\gamma-1}}{T_1} \right) \]

Which results in the dimensionless specific output work as a function of PR and TIT.
\[
\frac{w_{\text{net}}}{c_p T_1} = \frac{T_3}{T_1} \left(1 - \frac{1}{\Pi^{-1}}\right) + \left(1 - \Pi^{-1} \frac{\Pi^y}{y}\right)
\]

**B.2.2 Optimum Pressure Ratio**

The peak value of specific work output for a given temperature ratio (TR) \(\frac{T_3}{T_1}\) is called the optimum pressure ratio \(\Pi_{\text{optimum}}\). This pressure ratio (\(\Pi_{\text{opt}}\)) for peak specific work/power output can be calculated by differentiating equation (B-14) with respect to \(\Pi\) and setting it equal to 0, or we use equation (B-5), (B-6) and (B-7) and differentiate with respect to \(T_2\) as follows (\(c_{p,\text{gas}} = c_{p,\text{air}} = c_{p,\text{ideal}}\)):

\[
w_{g-4} = c_p(T_g - T_4) = w_t - w_c = c_p[(T_3 - T_4) - (T_2 - T_1)]
\]

We know that

\[
\Pi \frac{\Pi^{-1}}{y} = \frac{T_2}{T_1} = \frac{T_3}{T_4} \text{ and so } T_4 = \frac{T_3 T_1}{T_2}
\]

Equation can then be written

\[
w_{g-4} = c_p(T_g - T_4) = c_p \left[\left(T_3 - \frac{T_3 T_1}{T_2}\right) - (T_2 - T_1)\right]
\]

Partially differentiate \(T_2\) while holding other variables constant

\[
\frac{\delta}{\delta T_2} w_{g-4} = 0 \Rightarrow \left(\frac{T_3 T_1}{T_2^2} - 1\right) \Rightarrow T_2^2 = \frac{T_3 T_1}{T_2}
\]

We can see that for maximum gas power

\[
T_2 = \sqrt{T_3 T_1}
\]

Then \(\Pi_{\text{opt}}\) can be written as

\[
\Pi_{\text{opt}} = \left(\frac{T_2}{T_1}\right)^{\frac{y}{y-1}} = \left(\frac{\sqrt{T_3 T_1}}{T_1}\right)^{\frac{y}{y-1}} = \left(\frac{T_3}{T_1}\right)^{\frac{y}{2(y-1)}}
\]

The specific output power and thermodynamic efficiency for the optimum pressure ratio are respectively
\[
\frac{w_{\text{net}}}{c_p T_1}_{\text{opt}} = \frac{T_3}{T_1} \left( 1 - \frac{1}{\Pi_{\text{opt}}} \right) + \left( 1 - \frac{T_3}{T_1} \right)
\]

\[
\frac{w_{\text{net}}}{c_p T_1}_{\text{opt}} = \frac{T_3}{T_1} \left( 1 - \frac{1}{\sqrt{T_3/T_1}} \right) + \left( 1 - \sqrt{T_3/T_1} \right)
\]

\[
\frac{w_{\text{net}}}{c_p T_1}_{\Pi_{\text{opt}}} = \left( \sqrt{\frac{T_3}{T_1}} \right)^2 - 1 \tag{B-17}
\]

\[
(\eta_{\text{th,ideal}})_{\Pi_{\text{opt}}} = 1 - \sqrt{\frac{T_1}{T_3}} \tag{B-18}
\]

(Saravanamuttoo, Rogers et al. 1996)

A point worth noticing here is that the efficiency that corresponds to the optimum PR is solely a function of TR.

**B.2.3 Regeneration**

![Diagram of recuperated Brayton cycle](image)

Figure B-2 Recuperated Brayton Cycle (Boles 2006)
The simple gas turbine cycle generally has a turbine exit temperature \( T_4 \) that is higher than the exit temperature of the compressor \( T_2 \). There is an opportunity to reduce the amount of fuel required by the use of a regenerative\(^{20}\) process in which the turbines exhaust gases preheats the air between the compressor and the combustion chamber, (Boyce 2002). It is to be noted that if the pressure ratio of the simple cycle is large enough, then the compressor discharge temperature will be high and turbine exit temperature will be low, requiring no heat exchanger, (Shah 2005).

The net work output is not affected by the addition of a heat exchanger. The heat supplied to the system \( (Q_{in}) \) reduces from \( h_3 - h_2 \) to \( h_3 - h_5 \) and the heat emitted by the system \( (Q_{out}) \) reduces from \( h_4 - h_1 \) to \( h_6 - h_1 \).

In certain applications heat-exchangers have been phased out, because they no longer offer an advantage relative to simple cycles of high pressure ratio or, in the case of base-load applications, relative to combined cycle plant, (Saravanamuttoo, Rogers et al. 1996). However their use in automotive applications may still prove useful. The extent to which a recuperator approaches an ideal recuperator is called the **effectiveness** \( \varepsilon \) and is defined as: (Using the cold air assumptions) the effectiveness reduces solely to temperature differences.

\[
\varepsilon = \frac{q_{recup,act}}{q_{recup,max}} = \frac{h_5 - h_2}{h_4 - h_2} \approx \frac{T_5 - T_2}{T_4 - T_2} \tag{B-19}
\]

As the effectiveness of the regenerator is increased the heat transfer surface area grows, associated with this is; increases in cost, space requirements and a pressure drop.

The effectiveness is equal to the above Equation (B-19) when cold-air-assumptions are used and is assumed to be \( 100\% \) efficient in the ideal case. This is however impossible as one would need an infinite size of heat exchanger area. The thermodynamic efficiency of an ideal Brayton cycle with recuperation is

\[
\eta_{th,Recup} = \frac{h_3 - h_4 + h_1 - h_2}{h_3 - h_5}
\]

From equation (B-19)

---

\(^{20}\) ‘Regenerative’ is a term that simply means to employ the term heat exchange. It does not “generally” refer to a particular type of heat exchanger eg. regenerator or recuperator or in fact make relations to the type of flow whether mixed or separated. The recuperator will, however, be the typical component used to refer to the heat exchanger used within the cycles.
\[ h_5 = h_2 + \epsilon(h_4 - h_2) \]

\[ \eta_{\text{th,Regen}} = \frac{h_3 - h_4 + h_1 - h_2}{h_3 - h_2 - \epsilon(h_4 - h_2)} \]

\[ \epsilon = 100\% \text{ ideal} \]

\[ \eta_{\text{th,Regen}} = \frac{h_3 - h_4 + h_1 - h_2}{h_3 - h_2 - (h_4 - h_2)} \]

\[ \Delta h = c_p\Delta T \text{ for a gas in equilibrium} \]

\[ \eta_{\text{th,Regen}} = \frac{c_p(T_3 - T_4) + c_p(T_1 - T_2)}{c_p(T_3 - T_2) - c_p(T_4 - T_2)} \]

\[ \eta_{\text{th,Regen}} = \frac{(T_3 - T_4) + (T_1 - T_2)}{(T_3 - T_4)} \]

With the isentropic relations, \( T_2 = T_1(\Pi)^{\frac{1}{\gamma-1}} \) and \( T_4 = \frac{T_3}{(\Pi)^{\frac{1}{\gamma}}} \)

\[ \eta_{\text{th,Regen}} = \frac{(T_3 - \frac{T_3}{(\Pi)^{\frac{1}{\gamma}}} + (T_1 - T_1(\Pi)^{\frac{1}{\gamma}}))}{(T_3 - \frac{T_3}{(\Pi)^{\frac{1}{\gamma}}})} \]

\[ \eta_{\text{th,Regen}} = 1 - \frac{T_1}{T_3}(\Pi)^{\frac{1}{\gamma}} \]

It is noted that a high pressure ratio is not required for a heat exchange cycle and the increase in weight and cost of the total system because of the heat-exchanger is offset by the reduction in size of the compressor and increase in thermal efficiency, (Saravanamutto, Rogers et al. 1996). This was seen in an in-depth cycle performance analyse in Chapter 3. The effect of increasing pressure ratio for this cycle is opposite to that displayed in the simple cycle, (Boyce 2002). The current overall thermal efficiency of the microturbine with recuperator is 28-32\% - which means that for every joule of fuel input to the turbine 28-32\% of the fuel input will present as useful work, (Kolanowski 2004).
If the user wants to increase the work output and thermal efficiency of the recuperated cycle gas turbine, then intercooling and reheat should be employed. It is noted that, if only intercooling and reheat are added to a simple cycle then a decrease in thermal efficiency will occur.

**B.2.4 Co-generation**

Microturbines using recuperation technology are ideal for cogeneration systems and other industrial applications. Effective use of the thermal energy contained in the exhaust gas can improve microturbine system economics. Cogeneration is the process of using one fuel to produce two energy sources, and throughout this discussion the useable sources will be electricity and hot water, *(Kolanowski 2003)*. Utilizing the waste heat from the microturbine inevitably saves costs in the fuel. Just as the conventional cars radiator is used for cooling internal combustion engines, it also utilizes waste heat from the engine for space heating in the vehicle itself. A concept easily translated, for using the remaining heat energy that exists in the exhaust of a microturbine after recuperation. Almost all the energy that is not converted to shaft power in the final expansion stage is available in the exhaust gases for further potential use, *(Saravanamutto, Rogers et al. 1996)*. Cogenerators can use the waste heat to heat water, usually creating a low pressure steam. It is known that regular gas turbines are renowned for their use in combined cycles and cogeneration schemes.

In CHP operation, a second heat exchanger, the exhaust gas heat exchanger, transfers the remaining energy from the microturbine exhaust to a hot water system. Exhaust heat can be used for a number of different applications, including potable water heating, driving absorption cooling and desiccant dehumidification equipment, space heating and process heating. It is perfectly plausible that microturbine-based CHP applications do not have to use recuperators. With these microturbines, the temperature of the exhaust is higher and thus more heat is available for recovery. It would seem that CHP systems in the automotive sector would serve to satisfy space heating requirements, *(TechPro 2002)*.

The turbine expansion creates a temperature drop and so is directly related to the electrical efficiency; however microturbine CHP total system efficiency is a function of exhaust temperature alone. As noted before the recuperator effectiveness strongly influences the microturbine exhaust temperature. Variations in CHP efficiency and power output are mostly due
to the mechanical design and manufacturing cost of the recuperators. It is therefore pointed out that recuperators are very important components in current microturbine technology and should be given much more careful thought in their use for the application of HEVs.

**B.2.5 Microturbine component architecture**

The current microturbine architecture is represented in Figure B-2. This gives a general graphical representation of the cycle process of a current microturbine package. It is schematically similar to the recuperated cycle discussed above with the additional representation of a generator and further waste heat utilization to the user in the form of space heating.

![Figure B-3 Current recuperated microturbine configuration](image)

**B.2.6 Conclusion**

The different sectors the technology is utilized determines the exact configuration of all the components. We see in industry that high-pressure ratio simple cycles are used and in other areas low-pressure ratio with heat exchange. In the area of turbine technology, increased modifications don’t generally give much advantage to offset the substantially increased complexity and cost.
(same said for microturbine technology), (Saravanamuttoo, Rogers et al. 1996). Huge capital investment would be required in gas turbine manufacturing facilities. We see that designers try to raise the inlet temperatures of the gas turbine and respectively then increase the pressure ratio of the engine. Substantial improvements are seen but traded for engine component life. Creep-life and stress constraints of the materials used in microturbines are a particular issue needing attention and microturbine inlet temperatures in current designs vary between 840°C and 1000°C,(Watts 2000).

**B.3 Realistic Performance Parameters**

No turbine that operates on the Brayton Cycle is ideal due to the components within having efficiencies less than 100%. One needs to take into account irreversibilities and the pressure losses throughout the cycle resulting from friction, combustion losses, mechanical losses and heat transfer. A real measure of system performance is made through adapting the ideal equations to include these losses. There may be leakages in the system and can disrupt the mass flow throughout the cycle. The following is a performance analysis of a real gas generator simple/recuperated cycle. The cycle parameters are functions of **specific gas power** and **thermal efficiency**, and so these two relations will be used to analyze the performance. These relations will be used to optimize the cycle in preceding chapters. The cycle parameters such as mass flow, pressure ratio, component efficiencies and ambient conditions need to be known.

![Figure B-4 T-s diagram simple/recuperated cycle incorporating losses](Boles 2006)

Figure B-4 is a diagram that provides a realistic view of how parts of the cycle shift in reality. The diagram is basically the same as the simple cycle with deviations drawn to illustrate non-ideal behavior. All discussion and related equations hereafter will have incorporated the losses,
through various efficiency relations throughout the process, giving a more realistic outlook of the cycle.

The process 1-2a-3-g’ represents the process in the gas generator\textsuperscript{21} and the residual power, represented by g’-4a, is the specific \textit{gas power}. Gas power is the power that is extracted from the remaining expansion phase and can be used to rotate the generator or similar load.

The changes in the ideal assumptions have been listed below:

a. All processes are irreversible in nature because a finite amount of energy is always lost to the environment, which tends to increase, as components shrink in size.

b. Component losses need to be accounted for during the thermodynamic cycle. These will be accounted for through the compressor, turbine and mechanical efficiencies.

c. The properties of the working fluid change throughout the thermodynamic cycle, density and $c_{pgas}$, but \textit{will} be assumed constant during the actual cycle calculations for reasons of simplicity.

d. These values are expected to vary with temperature but will be taken as constant.

Dry air: $c_{p,air} = 1.005 \text{ kJ/kg.K}$, $\gamma_a = 1.40$

Combustion gases: $c_{p,gas} = 1.148 \text{ kJ/kg.K}$, $\gamma_g = 1.33$; and

e. Incomplete combustion of the fuel is takes place in the combustion chamber with an efficiency $\eta_b$.

The actual cycles will be recognized by the subscript ‘a’ and isentropic will be noted without one. Although not shown, it is also to be noted, that the actual compressor exit ($T_2$) and turbine exit temperatures ($T_4$) are higher than the isentropic.

\textsuperscript{21} The gas generator is a term used to separate the cycle process and physical arrangement of components. In regards to twin spool configurations where the cycle process and physical arrangement has a free turbine used to extract the ‘gas power’, then the gas generator can be easily seen as to encompasses only part of the cycle used to drive the compressor. This is seen in Figure B-5. In the case of single spool configurations the gas generator is physically the same as the free turbine. The gas generator is used both to power the compressor and supply the gas power. All gas turbine engines have the concept of the gas generator in common and it is only the way that they convert the gas power which is different.
The mechanical losses are usually quite low for microturbines due to low bearing friction.

The net input work required by the compressor is (with $\eta_m$ quite high)

$$w_{1-2a} = \frac{(h_{2a} - h_1)}{\eta_m}$$  \hspace{1cm} B-21

The actual turbine work output is

$$w_{3-g'} = (h_3 - h_{g'})$$  \hspace{1cm} B-22

The combustor (heat input rate):

$$w_{2a-3} = (h_3 - h_{2a})$$  \hspace{1cm} B-23

Gas Power:

$$w_{gg} = w_{g'-4a} = (h_{g'} - h_{4a})$$  \hspace{1cm} B-24

Waste Heat:

$$q_{4a-1} = (h_{4a} - h_1)$$  \hspace{1cm} B-25

**B.3.1 Actual Brayton Simple Cycle**

The net work (or specific gas work output) will be

$$w_{net,a} = w_{s,gg} = (h_3 - h_{4a}) + \frac{(h_1 - h_{2a})}{\eta_m}$$  \hspace{1cm} B-26
The heat supplied by the fuel usually comes from incomplete combustion in actual situations
\[ q_{in,a} = \frac{h_3 - h_{2a}}{\eta_b} \]

In the actual cycle \( \frac{P_2}{P_1} \neq \frac{P_3}{P_4} \) because of various pressure losses in the components and ducts. There is also a pressure loss in the combustion chamber (\( \Delta P_b \)). The aerodynamic resistance of flame stabilizing and mixing devices generate a loss in stagnation pressure (\( \Delta p_b \)). Therefore the turbine inlet pressure changes from \( P_2 = P_3 \) (ideal case) to \( P_{3a} = P_2 - \Delta p_b \). The turbine exit pressure remains the same \( P_{4a} = P_4 \).

The pressure ratio at the turbine expansion changes to
\[
\Pi_{\text{actual}}^{\text{expansion}} = \Pi_e = \frac{P_{3a}}{P_{4a}} = \frac{P_2}{P_4} \left( 1 - \frac{\Delta P_b}{P_2} \right)
\]

Since we assume that the pressure is expanded to atmospheric, \( P_4 = P_1 \)
\[ \frac{P_2}{P_1} = \Pi \]

Therefore
\[
\Pi_e^a = \frac{P_{3a}}{P_{4a}} = \frac{\Pi P_1}{P_1} \left( 1 - \frac{\Delta p_b}{\Pi P_1} \right)
\]
\[
\Pi_e^a = \frac{P_{3a}}{P_{4a}} = \Pi \left( 1 - \frac{\Delta p_b}{\Pi P_1} \right)
\]

(Boyce 2006), (Saravanamuttoo, Rogers et al. 1996)

The compressor efficiency\(^{22}\) \( \eta_C \) and turbine efficiency \( \eta_T \) are
\[ \eta_C = \frac{w_c}{w_{c,a}} = \frac{h_1 - h_2}{h_1 - h_{2a}} \approx \frac{c_{p,1-2,a,ir}(T_1 - T_2)}{c_{p,1-2a,air}(T_1 - T_{2a})} \]

\(^{22}\)The isentropic efficiencies will be used instead of polytropic, as this problem does not deal with multistage compression or expansion.
\[ \eta_r = \frac{w_{T,a}}{w_T} = \frac{h_3 - h_{4a}}{h_3 - h_4} \approx \frac{c_{p,3-4a,\text{gas}}(T_3 - T_{4a})}{c_{p,3-4a,\text{gas}}(T_3 - T_4)} \]

We will note here that there isn’t much error associated with assuming \( c_{p,\text{actual}} \approx c_p \) and \( \gamma = \gamma_{\text{air}} \). By substituting \( T_2 = T_1(\Pi)^{\frac{\gamma-1}{\gamma}} \) into component efficiency, the compressor exit temperature can be found.

\[ \eta_c = \frac{T_1 - T_1\Pi^{\frac{\gamma_{\text{air}}-1}{\gamma_{\text{air}}}}}{(T_1 - T_{2a})} \]

\[ (T_1 - T_{2a})\eta_c = T_1(1 - \Pi^{\frac{\gamma_{\text{air}}-1}{\gamma_{\text{air}}}}) \]

\[ T_{2a}\eta_c = T_1\eta_c - T_1(1 - \Pi^{\frac{\gamma_{\text{air}}-1}{\gamma_{\text{air}}}}) \]

\[ T_{2a} = T_1 \left[ 1 - \frac{1 - \Pi^{\frac{\gamma_{\text{air}}-1}{\gamma_{\text{air}}}}}{\eta_c} \right] \quad \text{B-29} \]

With the additional assumption that \( \gamma = \gamma_{\text{gas}} \) and \( \frac{T_4}{T_3} = \left( \frac{P_{3a}}{P_{4a}} \right)^{\frac{\gamma_{\text{gas}}-1}{\gamma_{\text{gas}}}} = \left( \frac{\Pi e}{\Pi} \right)^{\frac{\gamma_{\text{gas}}-1}{\gamma_{\text{gas}}}} \) the actual turbine temperature can be derived.

\[ T_{4a} = T_3 \left[ 1 - \eta_T + \frac{\eta_T}{\Pi \left( 1 - \frac{\Delta p_{b}}{\Pi P} \right) \left( \frac{\gamma_{\text{gas}}-1}{\gamma_{\text{gas}}} \right)} \right] \quad \text{B-30} \]

Taking into account the \( c_{p,g} \) and \( c_{p,a} \) and the combustion efficiency the heat input becomes

\[ q_{in,\text{act}} = \frac{c_{p,g} T_3 - c_{p,a} T_{2a}}{\eta_b} \]

It is known
\[ w_{\text{net,a}} = w_{s,gg} = c_{p,g}(T_3 - T_{4a}) + \frac{c_{p,a}(T_1 - T_{2a})}{\eta_m} \quad \text{B-31} \]

We can substitute the expressions developed for \( T_{2a} \) and \( T_{4a} \) into \( q_{\text{in,act}} \) and Equation B-31 above

\[
q_{\text{in,act}} = \frac{c_{p,g}T_3 - c_{p,a}T_1}{\eta_b} \left[ 1 - \left( 1 - \frac{\left( \frac{\gamma_{\text{air}} - 1}{\gamma_{\text{air}}} \right)}{\eta_C} \right) \right]
\]

\[
q_{\text{in,act}} = \frac{1}{\eta_b} \left[ c_{p,g}T_3 - c_{p,a}T_1 \left\{ 1 - \left( 1 - \frac{\left( \frac{\gamma_{\text{air}} - 1}{\gamma_{\text{air}}} \right)}{\eta_C} \right) \right\} \right]
\]

This can also be expressed as a heat flow rate by incorporating the gas and air mass flow rates within the equations.

\[
\hat{Q}_{\text{in,act}} = \frac{1}{\eta_b} \left[ c_{p,g}m_{\text{gas}}T_3 - \dot{m}_{\text{air}}c_{p,a}T_1 \left\{ 1 - \left( 1 - \frac{\left( \frac{\gamma_{\text{air}} - 1}{\gamma_{\text{air}}} \right)}{\eta_C} \right) \right\} \right] \quad \text{B-32}
\]

And now the specific work output

\[
w_{\text{net,a}} = w_{s,gg} = c_{p,g}(T_3 - T_{4a}) + \frac{c_{p,a}}{\eta_m} \left[ T_1 - T_1 \left\{ 1 - \left( 1 - \frac{\left( \frac{\gamma_{\text{air}} - 1}{\gamma_{\text{air}}} \right)}{\eta_C} \right) \right\} \right]
\]

\[
w_{s,gg} = c_{p,g} \left[ T_3\eta_T - T_3 \frac{\eta_T}{\left\{ \Pi \left( 1 - \frac{\Delta p}{\Pi p_1} \right) \right\}} \left( \frac{\gamma_{\text{gas}} - 1}{\gamma_{\text{gas}}} \right) \right] + \frac{c_{p,a}}{\eta_m} \left[ T_1 \left( 1 - \frac{\left( \frac{\gamma_{\text{air}} - 1}{\gamma_{\text{air}}} \right)}{\eta_C} \right) \right]
\]

\[
w_{s,gg} = \eta_T c_{p,g}T_3 \left[ 1 - \left\{ \Pi \left( 1 - \frac{\Delta p}{\Pi p_1} \right) \right\} \left( \frac{1 - \gamma_{\text{gas}}}{\gamma_{\text{gas}}} \right) \right] + \frac{c_{p,a}}{\eta_m} \left[ T_1 \left( 1 - \frac{\left( \frac{\gamma_{\text{air}} - 1}{\gamma_{\text{air}}} \right)}{\eta_C} \right) \right]
\]
Likewise the specific work output can be expressed as the power output of the shaft by the gas and air mass flow rates within the system.

\[
W_{s,gg} = \eta_T c_p g \dot{m}_{gas} T_3 \left[ 1 - \left\{ \Pi \left( 1 - \frac{\Delta p_b}{\Pi p_t} \right)^{\left( \frac{1 - \gamma_{gas}}{\gamma_{gas}} \right)} \right\} \right] + \frac{\dot{m}_{air} c_p a}{\eta_m} T_1 \left[ \frac{1 - \Pi^{\left( \frac{\gamma_{air} - 1}{\gamma_{air}} \right)}}{\eta_c} \right]
\]

B.3.1.1 Efficiency of turbine systems

The overall Turboshaft engine efficiency \( \eta_{thermal} \) (thermal efficiency) is a very important parameter for shaft power turbines, the ratio of net output energy to fuel input energy for the cycle. This efficiency is given by:

\[
\eta_{thermal} = \frac{\dot{W}_{s,gg}}{Q_{in,act}} = \frac{\dot{W}_{g' - 4}}{Q_{2a - 3}} = \frac{\dot{m} c_p (T_{g'} - T_{4a})}{\dot{m} c_p (T_3 - T_{2a})} \approx \frac{P_{shaft}}{m_{fuel} LHV_{fuel}}
\]

Where \( LHV_{fuel} \) is the lower heating value of the fuel.

The thermal efficiency is a direct measure for the energy conversion efficiency from fuel to work delivered (e.g. for power production). The exergetic efficiency would be a better indication for the amount of work which can be gained from the exhaust gas stream but requires additional data processing.

To improve the thermal efficiency, improvements in ceramic turbine technology, blade composition and cooling technology need to be further researched, but progress has been slow. If there are no losses or irreversibilities and the cycle is complete, the total energy added from the fuel sources will always be equal to the sum of the useful output energy and the wasted exhaust energy. A thermal efficiency of 30% means that for every 100 units of added energy 30 units will be available as useful output (in the form of power) while 70 units will leave the engine as wasted high-temperature exhaust or perhaps through various losses.

\[23\] The thermal efficiency is NOT the thermodynamic efficiency \( \eta_{th} \) introduced in the discussion of ideal cycles. In fact it is lower because this efficiency takes into account various types of losses.
Current microturbines have thermal efficiencies around 30% which is about 10% lower than their reciprocating counterparts. The low thermal efficiency is compensated by weight saving and the overall hybrid system efficiency\(^{24}\). The thermal efficiency can be increased by using waste heat absorption technology to power air conditioning or vehicle space heating. The Brayton cycle could also be combined with the Rankine cycle to generate more electrical energy but is not considered for HEV application for reasons of cost and complexity, (\textit{Limited 2000}). Increasing the pressure ratio also has the negative effect of raising stresses in the system, again reducing engine life.

If we substitute equation B-32 and B-33 into equation B-34 the actual cycle efficiency is

\[
\eta_{\text{ch,Bray,act}} = \frac{\eta_t c_p g m_{gas} T_3 \left[ 1 - \left( \Pi \left( 1 - \frac{\Delta p_b}{\Pi T_1} \right) \right)^{\left( \frac{1 - \gamma_{gas}}{\gamma_{gas}} \right)} \right] + \frac{\dot{m}_{air} c_p a}{\dot{m}} \left[ T_1 \left( 1 - \Pi \left( \frac{\gamma_{air} - 1}{\eta_{air}} \right) \right) \right]} {m_{gas} c_p g T_3 - m_{air} c_p a T_1 \left[ 1 - \left( \frac{1 - \Pi \left( \frac{\gamma_{air} - 1}{\eta_{air}} \right)}{\eta_a} \right) \right]} \]

\text{(Boyce 2006), (Saravanamutto, Rogers et al. 1996) and (Boles 2006)}

\(^{24}\) The electrification necessary in HEVs (various systems) increase the overall efficiency
B.3.1.2 Specific work output

Performance can be measured in terms of specific work/power output, defined in this case as output shaft work per kg.

\[ SFC_{shaft} = \frac{W_{g-4a}}{\dot{m}_g} = \frac{\dot{m}_g c_p (T_g - T_4a)}{\dot{m}_g} = \frac{P_{shaft}}{\dot{m}_g} \]

Where \( \dot{m}_g \) is the mass flow rate of the turbine. This includes the mass flow rate of the system with the fuel flow rate injected.

It is required that the above be put into dimensionless form

\[ \frac{P_{shaft}}{T_a c_{pgas} \dot{m}_g} \]

which is the ratio of output shaft power to the mass flow rate of the working fluid, flue gas specific heat and the ambient temperature. Specific gas power can be used as a measure for the compactness of the gas generator (i.e diameter) and so respectively, the dimensionless specific work output is a measure for the size of the apparatus; The length of the gas generator is determined by pressure ratio \( \Pi \), (Saravanamutto, Rogers et al. 1996).

The \( \dot{m}_g \) in the formula is the gas mass flow, \( c_{pshaft} \) is the heat capacity at constant pressure [J/kgk] and \( T_1 \) is the temperature of the environment [K]. The \( c_p \) in the formula above is the specific heat of the gas downstream of the combustor (flue gas). A distinction must be made from \( c_p \) of air because the temperature, pressure and gas composition has changed through the combustion process. The value of the variable, mass fuel flow rate is calculated through iterations in the program GSP for corresponding variations in temperature. The range of the temperature possible is already bounded by the choice of sizing the microturbine to a 30kW unit and specifying an inlet mass flow rate. The value of the ambient temperature is kept constant. Gas turbine engines have reasonable levels of efficiency and have high levels of specific power. They are especially useful for applications that need compact power, which is perfect for Hybrid Electric Vehicles.

\[ ^{25} \text{Flue gas or simply ‘gas’ is the gas downstream of the combustor} \]
If we take $c_{p,g}$ to be the value for the whole cycle ($c_{p,g} = c_p$) then we can make the power output dimensionless by dividing through by both $c_{p,g}$, a reference temperature $T_1$ and the gas mass flow rate $m_{\text{gas}}$. 

$$\frac{\dot{W}_{s,gg}}{m_{\text{gas}}T_1 c_{p,g}} = \eta_T \frac{T_3}{T_1} \left[ 1 - \left( \frac{1}{\Pi} \left( 1 - \frac{\Delta p_b}{\Pi p_1} \right) \right) \left( \frac{1 - \gamma_{\text{gas}}}{\gamma_{\text{gas}}} \right) \right] + \frac{c_{p,a}}{m_{\text{gas}} c_{p,g}} \left[ 1 - \frac{\Pi (\gamma_{\text{air}} - 1)}{\Pi T_1} \right]$$  

B-36 

(Boyce 2006), (Saravanamuttoo, Rogers et al. 1996) 

### B.3.2 Actual Brayton Recuperated Cycle

We will now add a realistic heat exchanger (recuperator) with a finite effectiveness. Exhaust heat will be transferred to increase the energy of the air entering the combustion chamber. It is pointed out that the specific net work output will only differ slightly due to the additional pressure losses in the heat exchanger. The efficiency will increase by a notable amount. The pressure losses that exist in our system are stagnation pressure losses $\Delta p_b$ in the combustion chamber and pressure losses in the heat exchanger, air- and gas-side $\Delta p_{ha}$ and $\Delta p_{hg}$ respectively.(Shah 2005).

Instead of the turbine inlet pressure being $p_3 = p_2$ the added losses due to the recuperator and combustor are

$$p_{3a,\text{recup}} = p_2 \left[ 1 - \frac{\Delta p_b}{p_2} - \frac{\Delta p_{ha}}{p_2} \right]$$

The pressure losses in the gas side of the recuperator change from $p_4 = p_1$ to $p_{4a,\text{recup}} = p_1 + \Delta p_{hg}$

$$p_{4a,\text{recup}} = p_1 + \Delta p_{hg}$$

Remembering that $\frac{p_2}{p_1} = \Pi$ then the pressure ratio for expansion in the actual recuperated cycles case is

$$\Pi_{e,\text{recup}} = \frac{p_{3a,\text{recup}}}{p_{4a,\text{recup}}} = \frac{p_{3a,\text{recup}}}{p_{4a,\text{recup}}} = \frac{\Pi p_1 \left[ 1 - \frac{\Delta p_b}{\Pi p_1} - \frac{\Delta p_{ha}}{\Pi p_1} \right]}{p_1 + \Delta p_{hg}}$$
$T_{2a}$ will not change as the additions to the equations will be nil. With $\frac{T_4}{T_3} = \left( \frac{P_{3a}}{P_{4a}} \right) \left( \frac{\gamma_{gas}^{-1}}{\gamma_{gas}} \right)$,

$\left( \Pi_{e,recup} \right) \left( \frac{\gamma_{gas}^{-1}}{\gamma_{gas}} \right)$, and equation $\eta_T$, $T_{4a}$ will change to

$$T_{4a,recup} = T_3 \left( 1 - \eta_T \right) + \frac{\eta_T}{\left( \Pi_{e,recup} \right) \left( \frac{\gamma_{gas}^{-1}}{\gamma_{gas}} \right)}$$  \hspace{1cm} (B-37)

From Figure B-2 we see that $q_{in}$ from the combustor is

$$q_{in,actal,recup} = \frac{h_3 - h_2'}{\eta_b}$$

We may find a relation for $h_2'$ via the effectiveness relation for the heat exchanger

$$\varepsilon = \frac{h_2' - h_{2a}}{h_{4a,recup} - h_{2a}}$$

$$h_2' = \varepsilon (h_{4a,recup} - h_{2a}) + h_{2a}$$

$$q_{in,actal,recup} = \frac{h_3 - \left[ \varepsilon (h_{4a,recup} - h_{2a}) + h_{2a} \right]}{\eta_b}$$

With $\Delta h = c_p \Delta T$

$$q_{in,actal,recup} = \frac{c_p_{gas} T_3 - \left[ \varepsilon \left( c_p_{gas} T_{4a,recup} - c_p_{air} T_{2a} \right) + c_p_{air} T_{2a} \right]}{\eta_b}$$

$$q_{in,actal,recup} = \frac{c_p_{gas} (T_3 - \varepsilon T_{4a,recup}) + c_p_{air} T_{2a} (\varepsilon - 1)}{\eta_b}$$  \hspace{1cm} (B-38)

Now we can substitute $T_{2a}$ and $T_{4a,recup}$ from equations B-29 and B-37 into equations B-38
\[ q_{in, a, recup} = \frac{T_3 c_{p, gas}}{\eta_b} \left[ (1 - \varepsilon) - \varepsilon \eta_T \left( -1 + \frac{1}{\left( \Pi_e^{a, recup} \frac{Y_{gas} - 1}{Y_{gas}} \right)} \right) \right] \]

\[ + \frac{c_{p, air} T_1}{\eta_b} \left[ 1 - \left( \frac{1 - \Pi \frac{Y_{air} - 1}{Y_{air}}}{\eta_c} \right) (\varepsilon - 1) \right] \]

And now the work output

\[ w_{s, gg} = \eta_T T_3 c_{p, g} \left[ 1 - \frac{1}{\left( \Pi_e^{a, recup} \frac{Y_{gas} - 1}{Y_{gas}} \right)} \right] - T_1 c_{p, a} \left[ \frac{\Pi \frac{Y_{air} - 1}{Y_{air}} - 1}{\eta_C} \right] \]

Corresponding power output

\[ \dot{W}_{s, gg} = m_{gas} \eta_T T_3 c_{p, g} \left[ 1 - \frac{1}{\left( \Pi_e^{a, recup} \frac{Y_{gas} - 1}{Y_{gas}} \right)} \right] - T_1 m_{air} c_{p, a} \left[ \frac{\Pi \frac{Y_{air} - 1}{Y_{air}} - 1}{\eta_C} \right] \]

The dimensionless form, power output while assuming that \( c_{p, g} \) is to be used for the complete cycle and \( m_{gas} \) then

\[ \frac{\dot{W}_{s, gg}}{m_g T_1 c_{p, g}} = \frac{T_3 \eta_T}{T_1} \left[ 1 - \left( \frac{\Pi_{p_1} \left[ 1 - \frac{\Delta p_b}{\Pi_{p_1}} - \frac{\Delta p_{ha}}{\Pi_{p_1}} \right]}{p_1 + \Delta p_{ng}} \right) \right] - \frac{c_{p, a}}{m_g} \left[ \frac{\Pi \frac{Y_{air} - 1}{Y_{air}} - 1}{\eta_m \eta_C} \right] \]

The efficiency is found by subbing equations B-39 and B-40 into B-34
Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application

\[ \eta_{th,recup} = \frac{m_{g,b} c_p T_3}{\eta_b} \left[ (1 - \varepsilon) - \varepsilon \eta_f \left( 1 - \frac{\Delta p_f}{\Delta p_{gb}} \right) \left( \frac{T_{gas}}{\gamma_{gas}} \right)^{1-\gamma_{gas}} \right] - \frac{T_m c_p (1 - \varepsilon)}{\eta_m} \left( 1 - \frac{T_{gas}}{\gamma_{gas}} \right) \]

(Saravanamuttoo, Rogers et al. 1996; Boyce 2006)

B.4 Other Performance Parameters

The power produced by providing a rotating shaft a given RPM, which then exerts a given amount of torque on a load and is related by the following.

Power transmitted by a shaft depends upon the torque and angular velocity.

\[ P = \tau(t) \ast \omega(t) \]

with
\[ \omega = \frac{2\pi N}{60} \]

\( \omega(t) \) is the angular velocity in radians per second, \( \tau(t) \) is the torque applied upon the center of mass of the body, changing with time and \( N \) is the angular velocity in revolutions per second. The efficiency of the microturbine engine (power producer versus fuel consumed) increases with rotational speed until it is 100 percent efficient.

The gas turbine produces shaft mechanical power. It can be connected to an electric generator in order to convert this shaft (mechanical) power into electric power. Some amount of this power is lost during this conversion (mechanical to electrical). The 30kW shaft mechanical power needs to be converted to electric power for it to be utilized by the HEV. As discussed before some finite amount of power loss will occur. The electric power may be calculated using

\[ P_M = \frac{P_E}{\eta_{\text{generator}}} \]

\( \eta_{\text{generator}} \) is the efficiency of the electric generator and usual values of around 92% are used.

The capstone C30 microturbine is rated to generate 27-28kW of electric power from a unit that generated 30kW of mechanical shaft power.

**B.5 Geometric Size, Weight and Cost**

The size of turbomachinery is not only related to component efficiency but driven by material selection and also other important performance parameters.

The type, geometry and dimensions depend primarily on:

- Specific speed, \( N_s \): determines the type & basic shape of the runner & other parts of the unit. \( N = N_s \times H^{(5/4)}/\sqrt{P} \) where \( N \) is synchronous speed in rpm, \( H \) is head and \( P \) is power

- Cost

**B.5.1 Specific Speed**

Similarity Relations and Design Criteria of Turbines:

Low specific speeds are associated with high heads and high specific speeds are associated with low heads. Moreover, there is a wide range of specific speeds which may be suitable for a given head. Higher specific speeds for a given head results in smaller turbine/generator dimensions and
higher speed generators. Since the turbine capital cost decreases with an increase in speed a balance must be found with the efficiency, (Benguedouar 1988).

Turbine Setting

The negative aspect of high specific speeds is they requiring a deeper setting to avoid cavitation and must also be included in the assessment.

The designer will initially use provided preliminary design data such as the head H, the volume flow rate Q and the rotational speed N during analysis of compressors/pumps. A turbine preliminary design requires the parameters shaft power $P \dot{W}_{shaft}$, the head at turbine entry H and the rotational speed N. Parameters that contain the rotational speed and rotor diameter are useful when describing the characteristics of turbomachines. These values are provided in the form of specific speed $N_s$ and specific diameter $D_s$.

Specific speed $N_s$ is a parameter used to facilitate the choice of the most appropriate machine required for a given duty. The specific speed is proportional to the net output power and since for a given turbine, the power and the rotational speed are normally specified. This parameter is used and is sometimes called the shape number, (Benguedouar 1988).

The other important parameter used in the selection of the type of machines required for a given duty is the specific diameter $D_s$. Using the $N_s$-$D_s$ diagrams another factor called specific diameter can be obtained and from which the size and the approximate weight of the turbine are calculated.

The specific diameter is defined as the diameter of a turbine which handles a volume flow Q of unity at exit and expands a head of unity.

$N_s$ and $D_s$ are obtained as follows:

The volume flow rate Q is proportional to the velocity and the area of passage. The area is proportional to $D^2$. The velocity is proportional to $\omega$ (angular velocity) and the diameter is proportional to N (rotational speed).
\[ Q \propto ND^3 \] \hspace{1cm} B-46

H is proportional to the square of the velocity which is proportional to ND.

\[ H \propto N^2D^2 \] \hspace{1cm} B-47

H: Is the head.
h: Is the enthalpy in this case not the specific
Q is the volume flow rate

Or from the non-dimensional groups

\[ \frac{Q}{ND^3} = \phi_1 = constant \] \hspace{1cm} B-48

\[ \frac{gh}{N^2D^2} = \psi_1 = constant \] \hspace{1cm} B-49

\[ \frac{P}{\rho N^3D^5} = \tilde{P}_1 = constant \] \hspace{1cm} B-50

It is a simple matter to combine any pair of these expressions in such a way as to eliminate the diameter. For a pump the customary way of eliminating D is to divide \( \phi_1^2 \) by \( \psi_1^{3/4} \).

If we solve for the above Equations B-46 to B-47 or B-48 to B-50. We get
\[ N_S = \frac{NQ^{1/2}}{(gH_{ad})^{3/4}} \]  

where,

\[ D_s = \frac{DH^{1/2}}{Q^{1/2}} \]

With

\[ \omega = \frac{\pi N}{30} \]

Ns is called the specific speed. Ns is directly proportional to N. In the case of a turbine the power specific speed Nsp is more useful and is defined by

\[ N_{ps} = \frac{\hat{\bar{Z}}_1}{\psi_1^{5/4}} = \frac{N(P/\rho)^{1/2}}{(gH_{ad})^{5/4}} \]

Both Ns equations are dimensionless. Calculate specific speed in one or other of these forms rather than dropping the factors g and r which would make the equations dimensional, (Dixon 1998). A design method relating specific speed, diameter and efficiency for the radial turbine and compressor is found in Appendix B.6.

Specific speed shows that when selecting a turbomachine for a given head H and capacity Q, the highest possible value of Ns should be chosen because of the resulting reduction in size, weight and cost. Keeping mind the constraints with efficiency a turbomachine could be made extremely small were it not for the corresponding increase in the fluid velocities. “For machines handling liquids the lower limit of size is dictated by the phenomenon of cavitation”, (Dixon 1998). Since a higher specific speed implies a smaller machine, for reasons of economy, it is desirable to select the highest possible specific speed consistent with good efficiency, (Dixon 1998).
B.5.2 Cavitation

Machines which are subjected to liquids the restriction on size is dictated by the phenomenon of cavitation. “Cavitation is the boiling of a liquid at normal temperature when the static pressure is made sufficiently low. It may occur at the entry to pumps or at the exit from hydraulic turbines in the vicinity of the moving blades. The dynamic action of the blades causes the static pressure to reduce locally in a region which is already normally below atmospheric pressure and cavitation can commence. The phenomenon is accentuated by the presence of dissolved gases which are released with a reduction in pressure”, (Dixon 1998).

B.6 Design Method for Specific speed, Efficiency and Tip diameter

Design Method Compressor

The method is based on specific speed, which is commonly used by industry for radial turbomachinery during the conceptual design stage. The method of estimating the performance and size of radial compressors and turbines is described by (Balje 1981). Two non-dimensional parameters are used to characterize the turbomachinery. These two parameters are referred to as specific speed (Ns) and specific diameter (Ds). The specific speed and specific diameter are defined as follows:

\[
Ns = \frac{N \sqrt{Q}}{(gH_{ad})^{\frac{3}{4}}} \tag{B-53}
\]

\[
Ds = \frac{D(gH_{ad})^{\frac{3}{4}}}{\sqrt{Q}} \tag{B-54}
\]

Where:

Q is the volume flow rate \([m^3/s]\), Had the adiabatic head \([m]\), N the rotating speed \([rad/s]\).
The exact physical meaning of $Ns$ is not obvious. One way to look at it is that it is a comparison between the non-dimensional flow rate, or flow coefficient ($\phi$) and the non-dimensional pressure rise ($\varphi$).

Non-dimensional flow coefficient can also be defined as

$$\phi = \frac{m}{\rho ND^2}$$

And non-dimensional pressure rise is also defined as

$$\varphi = \frac{\Delta P}{\rho ND^2}$$

where $N$ is the rotating speed, $D$ the impeller tip diameter, $\rho$ the density. Then $Ns$ can be written as (Dixon 1998)

$$Ns = \frac{\phi^\frac{1}{2}}{\varphi^{\frac{3}{4}}}$$

It is well established in the industry that radial turbomachinery efficiency can be correlated to the specific speed, (Whitfield and Baines 1990). For each given specific speed, there is a particular specific diameter, which will give that (optimal) efficiency. Figure B-7 and Figure B-8 are correlations for radial compressors. Figure B-9 is for a radial turbine.
The estimation procedure is as follows:

Figure B-7 Efficiency vs. Ns for Radial Compressors, Derived from Fig. 3.8 in Balje [1980-81]

Figure B-8 Ns vs. Ds for Radial Compressors, Derived from Fig. 3.8 in Balje [1980-81]
(1) For given cycle parameters, the exit density can be estimated. The density used by Equation B-53 and B-54 takes the mean value between the densities at the inlet and the exit.

(2) The specific speed can be calculated using density, rotating speed, and pressure rise.

(3) The efficiency can be estimated using the curve in Figure B-7 or Figure B-9.

(4) Specific diameter can be obtained using the curve in Figure B-8 or Figure B-9.

(5) From the definition of specific diameter (Ds), the impeller diameter can be computed.

The Design method for the turbine is similar. Use appropriate equations.

![Figure B-9 NsDs diagram for turbines calculated for minimum loss coefficients](image)

For preliminary design purpose (Balje 1981) presents NsDs diagrams similar to those in use for pumps.
Appendix C

This appendix provides the raw data from each scale in off-design analysis.

C.1 30kW Scale

The following three input parameters were the chosen operational conditions used to generate the tables below. The data sets are formed as follows. For every power code three temperatures are swept for each pressure. There are therefore 27 data points in each table.

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C.1.1 Max

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Appendix D

D.1 Engine Station Numbering and Nomenclature

This appendix provides the international standard for engine station numbering used throughout this report. Station number designation has been standardized to unambiguously define the station interfaces.

Aerospace Standard 755 by the SAE (Society of Automotive Engineers) for gas turbine engine station designation numbers.

Figure D-1Shaft power engine station numbering

The fundamental station numbers for the core stream of an engine are as below:
AMB  Ambient conditions
0  Ram conditions in free stream
1  Engine intake front flange, or leading edge
2  First compressor/fan front face
3  Last compressor exit face
4  Combustor exit plane
5  Last turbine exit face
6  Front face of mixer, afterburner etc.
7  Propelling nozzle inlet
8  Propelling nozzle throat
9  Propelling nozzle or exhaust diffuser exit plane

D.1.1 Intermediate station faces
Stations between the fundamental ones are numbered using a second digit suffixed to the upstream fundamental station number. Where more than ten intermediate stations are required a third digit is used.

D.1.2 Shaft power engines
For a simple cycle shaft power engine the key station numbers are as per above, however stations 6, 7 and 8 are normally redundant as there will only be an exhaust diffuser between stations 5 and 9. In the case of an industrial engine station 1 would be the engine inlet flange, and station 9 the engine exhaust flange. Station 0 would be used for the plant intake flange, and station 10 for the plant exhaust flange.
For the more complicated intercooled, recuperated shaft power cycle the following intermediate station numbers would typically be employed for these components:
21 First compressor exit face
23 Intercooler inlet face
25 Intercooler exit face
26 Second compressor inlet face
307 Recuperator air side inlet face
308 Recuperator air side exit face
31 Combustor inlet.
6 Recuperator gas side inlet
Appendix E

E.1 Scale Effects and Issues

Physics and mechanics influencing the design of the components change with scale. Therefore, the optimal detailed designs can be quite different from one another. This investigation targets the identification of the loss mechanisms that impellers suffer from when their size decreases.

Scaling is a common technique to define larger or smaller geometries with similar characteristics. However, a simple scaling of a high performance large gas turbine will not result in a good micro gas turbine. The main factors perturbing such a scaling are:

- The large change in Reynolds number. This is related to viscous forces in the fluid (which are larger at microscale),
- Massive heat transfer between the hot and cold components (negligible in large machines). Due to larger surface-area-to-volume ratios (increase at microscale). With decreasing turbine mass flow, heat leaks along the turbine housing and the rotor have an increasing effect in a negative manner on its efficiency.
- Geometrical restrictions related to material and manufacturing of miniaturized components. (Manufacturing tolerances, surface finish). Increased manufacturing constraints (limited mainly to two-dimensional planar geometries). Eg. Both the diameter and the blade height are reduced. Thus a very small geometry is to be machined with very high accuracy and smooth surfaces.
- Tip clearance effects. The clearance between wheel, its cover and its inlet geometry have to be decreased in order to keep clearance losses small. Blade tip and seal clearances are based on a number of factors: rotor sag, thermal distortion, bearing alignment, and rotor unbalance.
- General engine-to-engine variation; and
• Few effects from-- chemical reaction times (invariant), the electric-field strength that can be realized (higher at microscale),

All of which adversely affect efficiency, (Rodgers 2003). There are also other associated problems with shrinking geometry. As the turbine wheel diameter decreases, the turbine speed must be increased to maintain the required high tip speed. This promotes higher stresses in the materials.

While the above arguments seem particularly negative, other features, such as the low maintenance requirements, the exceptional reliability, low emissions and vibration levels of turbines make them quite attractive for use.

For fixed specific speed and cycle conditions, the scalar relationships can be applied.

\[ \text{Output power} \propto \dot{m}_a \alpha D^2 \] \hspace{1cm} \text{E-1}

It should also be pointed out that when scaling the component dimensions the associated power output and mass flow rates will not be the same as theoretical predictions. This is due to issues such as surface finish on airfoils and tolerances which do not scale. Recurring manufacturing cost is one of the other many factors that do not scale. Cost usually must be independently evaluated because it is difficult to predict. A usual trend is observed however that when scaling up the cost per unit, power should go down. It is simply more cost effective to use more expensive technologies for application in bigger engines first.

\textbf{E.1.1 Reynolds number effect (2nd order effect)}

Low Reynolds number and surface roughness are two very important limiting issues for reaching good efficiencies. Machining operations such as milling or casting impart a particular surface roughness and will increase with decreasing size of the impellor. The consequence is that the skin friction losses increase with decreasing impeller size for a given Reynolds Number. If manufacturing procedures can be optimized to allow for better surface finishes then skin friction
will be decreased. If the Reynolds number is sufficiently high, the efficiency essentially becomes independent of the Reynolds number.

**E.1.2 Tip Clearance**

The Tip clearance is the second limiting factor. A clearance gap must exist between the rotor vanes and the shroud. The tip clearance is defined as the radial gap between the rotor blades and casing. Increases in rms tip clearance decrease the efficiency and deteriorates the surge line. The turbomachinery geometry and hence the associated map is changed if tip clearance is varied. Small compressors are particularly susceptible because the clearance is a more significant percentage of blade height, (Walsh and Fletcher 2004).

There is a pressure difference between the pressure and the suction surface of a vane, this gives rise to leakage flow\(^26\) that is driven through the gap introducing a loss in efficiency of the turbine. The minimum clearance is usually a compromise between manufacturing difficulty and aerodynamic requirements. The manufacturing tolerances and the bearing technology severely impact relative tip clearance as impellers get smaller in size.

The hydrostatic oil bearings, as currently used in automotive turbochargers, present overall clearances in the order of 0.1–0.3 mm. In small turbomachinery the design of the system needs to minimize the tip clearance because it is a more significant percentage of blade height. It will be seen that it has quite an adverse effect on efficiency. The choice of the bearing technology as well as the assembly tolerances, directly affect the efficiency of the impeller. The MTT 3kW unit operates on oil bearings and so the expected losses can be expected to be more than a unit which would operate on air bearings, (Dixon 1998). The Tip clearance effects are listed below:

---

26 Leakage flow is largely unturned by the blades, so no work is done on this flow. The leakage flow exits the tip gap with a magnitude of velocity similar to that of the mainstream flow relative to the rotor, but in a direction almost perpendicular to the mainstream flow. The mixing of the mainstream and tip-leakage flows gives rise to the loss of efficiency, J. Schiffmann, D. F. (2010). "Design, experimental investigation and multi-objective optimization of a small-scale radial compressor for heat pump applications." Energy: 436-450.
• Leakage flow.
• Tip vortex; and
• Linear decline of power output and efficiency on gas turbines observed.

It may be then possible to linear interpolate the effects of tip clearance from two known microturbines if respective blade heights and tip clearances are known.

E.1.2.1 Alterations to reduce tip-clearance effects

The effect of tip clearance can be managed by mounting a shroud on the tip of the blade. If a shroud is fitted to a rotating blade it usually increases the aerodynamic efficiency of the rotor. It may also be able to actively control the tip gap, as the blade height varies with the expansion caused by temperature changes in the turbine. However, the mass of the shroud also requires the rotor to rotate at a lower speed.

E.2 Heat transfer (non-adiabatic effect)

As the impeller size decreases, the ratio between the available surface and the volume increases. The flow processes in the impeller are then not treatable as purely adiabatic. The higher the exhaust gas temperature or where hot and cold sources are in close contact (micro-turbine generators), the non-adiabatic effect may depreciate the overall efficiency considerably, (J. Schiffmann 2010). However if the microturbine is above the range of 9kW or tip diameters of 13cm then the heat transfer effect should have little influence on efficiency.

E.3 Conclusion

As a summary; regarding Reynolds number, tip clearance, minimum feature effects and non-adiabatic issues only the Reynolds number effects and the tip clearance are topics that will be given closer attention for scale effects, for this particular range extender application.

E.4 Radial Turbomachinery

The following will outline the major component design considerations and their connection with scale effects. A rotor assembly is shown in Figure E-1. An approximate method is needed for scaling to obtain a smaller engine, calling for minimal development while retaining the proven
durability of the existing engine. The following will discuss the effect of scaling on particular components of a microturbine and give empirical relations where necessary.

\[ E.4.1 \] Turbine

The turbine component is used to extract the kinetic energy from the working fluid by expanding the hot gases that leave the combustion chamber. The turbine converts this kinetic energy into shaft power which is used to drive the compressor and provide power to loads, (Gorla and Khan 2003). When the gases are expanded, a reduction in pressure is observed. The gas turbine is similar to the conventional gasoline and diesel engines in regards to its working medium and internal combustion. However the compression and expansion processes are both carried out by means of rotating machinery, rather than by positive displacement, as in reciprocating machinery. Turbines, like compressors, can be classified as axial, radial or a mixture according to the type of fluid flow being used. The turbine endures high rotational speeds and temperatures, and so its blades are subjected to a number of stresses. The bending and Centrifugal stresses, creep, mechanical and thermal fatigue can all lead to mechanical failure and so the Turbine Inlet Temperature (TIT) is restricted to the metallurgical limit of the materials, (Saravanamutto, Rogers et al. 1996). To achieve optimum overall efficiency the TIT is raised to the highest temperature that is possible for the materials to handle and the inlet airflow to the compressor as
low as possible. An additional free (power) turbine can be used to improve the overall efficiency of the system. However, the size, cost and simplicity of the microturbine engine would be compromised for a small gain in efficiency and as a result is not adopted, (Saravanamuttoo, Rogers et al. 1996).

Small turbomachinery will tend to primarily use inward flow radial and mixed flow turbines because of their simplicity, cost, relatively high performance and simple installation. Small gas turbines and turbochargers utilize these components and usually have flow in the range 0.05–2.0 kg/s. Figure E-2 shows turbine and compressor peak efficiency against rotor tip diameter for a nominal pressure ratio of 3. The attainable efficiencies for rotor tip diameters on the order of 70 mm are about 83% as shown on Figure E-2. The peak turbine component efficiencies of the Capstone C30 as measured on engine test were as high as 83%, and so for a 3 kw microturbine efficiencies as high as 78% should be attainable with 2010 gas turbine type technology (excluding significant heat transfer effects). Initial choice of specific speed would be important. Efficiencies for turbocharger radial turbines can be up to ten percentage points lower especially if including bearing losses and with no exhaust diffuser, (McDonald and Rodgers 2007).

![Figure E-2 Size effects on efficiency of radial flow compressor and turbine](adapted from Rodgers)
Automobile turbocharger rotor tip speeds rarely exceed 450 m/s. The turbine tip speed of 585 m/s would be higher than current state-of-the-art automobile ceramic turbines requiring progressive advancements in aerodynamic design to optimize the degree of reaction together with improved ceramic materials and brittle stress analysis techniques. The design of microturbines with single stage radial compressors and radial turbines is focused towards optimization of overall engine thermal efficiency, thus the selection of rotational speed becomes a compromise between both compressor and turbine aerothermodynamical design criteria.

To provide acceptable levels of power it is clear from the previous sections that a very high speed of rotation may be expected on the gas generator turbine to provide acceptable levels of power; this requirement is somehow relaxed if a power turbine is introduced. A two stage turbine design allows a different shaft speed and thus a lower and more acceptable angular velocity can be imposed on the power turbine, (Moore 2002).

**E.4.2 Compressor**

A compressor is a devise whose function is to pressurize a working fluid. The compressor which is primarily used in microturbine technology is of the radial flow types, which are low flow operating units, (Boyce 2002). Capstone incorporates a (centrifugal) Rotary Flow Compressor in its C-30 Model which compresses and slows the inlet working fluid to required combustion chamber conditions, (Kolanowski 2004). The compressor components are connected to the
turbine by a rotor shaft in order to allow the turbine to turn the compressor. A single shaft gas turbine has only one shaft connecting the compressor and turbine components.

It is well known that radial flow turbomachinery handles the small volumetric flows of air and combustion products with reasonably high component efficiency. Large-size axial flow turbines and compressors are typically more efficient than radial flow components. However, the size range of microturbines -- 0.2 to 2 kg/second of gas flow -- necessitates the use of radial flow components. These components offer minimum surface and end wall losses and provide the highest efficiency, (TechPro 2002). Radial components are usually more efficient at low pressure ratios due to their uncomplicated design, robustness and insensitivity to flaws. The ratio of the pressure at compressor exit to the pressure at compressor inlet (commonly atmospheric pressure) is known as the compression pressure ratio (Π) and is essential to the performance of the microturbine system, (Saravanamuttoo, Rogers et al. 1996).

A single stage centrifugal compressor is the choice for microturbines as a consequence of its cost, simplicity, compactness and performance characteristics such as wide surge margins with high inlet flow distortion tolerance. Typical efficiency characteristics with varying tip diameter have been shown in Figure E-2. The attainable efficiencies for small mass flow compressors are largely dependent on design choice of specific speed and Mach number, (McDonald and Rodgers 2007).

Specific speed is a function of rotational speed, volume flow, and adiabatic head. Mach number is a function of pressure ratio and is particularly critical for small compressor entry blading, necessitating very thin blades on the order of 0.25 mm.

Heat transfer between the turbine and compressor could either be a turbine cooling asset or a performance penalty. Its effect on engine performance can be minimized by designing the compressor near optimum specific speed (small surface area/mass flow), but it is aggravated by higher compressor exit to turbine inlet temperature differentials. It has been stated by Colin Rodgers that if the size of the turbine is maintained above 9kW then the microturbine will not suffer from any drastic heat transfer effects. The state of the art radial compressor technology continues to improve,(Miller 2002).
Pressure ratios of up to 4:1 are easily achievable in a single aluminum alloy stage readily obtained from turbo charger technology. If higher PRc in a single stage would be used more expensive materials such as stainless steel or titanium would be required. The latter is not yet suitable for mass production and would lose out on high-volume economics.

In general, once the mass-flow rate and pressure ratio have been specified, shaft speed can then be chosen. The choice of shaft speed dictates the cross sectional shape of the rotor which ultimately dictates rotor efficiency. A brief design process is outlined in Appendix B.6. Essentially there is an ideal rotor speed for any given specification and it is here that design difficulties become apparent. This speed can be estimated initially by means of specific speeds and in essence, the lower the power output of the MTG, the higher must be the shaft speed in order to obtain an efficient compressor, (Moore 2002).

E.5 Complete Similarity and the Moody’s Empirical scale effect correlation
It is important that we generalized the material with the concept of similarity. Similarity is a concept closely related to dimensional analysis. Theoretical analysis rarely solves practical problems and experimental results are frequently needed to complete the study. Results are frequently taken from tests under certain conditions and then applied to another set of conditions. This is made possible by the laws of similarity. The behavior of the fluid is modeled by these laws under different sets of circumstances. Comparisons are usually made between the prototype, and the model, that is, the full-size aircraft, ship, river, turbine, or other device apparatus. “Physical similarity” is a term that generalizes many different kinds of similarity. A comparison between a prototype and a model is said to be valid when the sets of conditions associated with each is physically similar.

There is geometric similarity:

- Concerns previously mentioned shape issues, surface finishes to be similar. The actual roughness of surfaces in small machines differ from that in larger machines; thus the relative roughness in the small machine is greater and the frictional losses are consequently more significant. The blades in the smaller machine may be relatively thicker. Clearances in the small machine cannot be reduced in the same proportion as other length measurements, and so leakage losses are relatively higher.

Exact geometric similarity cannot be achieved for the following reasons:

- the blades in the model will probably be relatively thicker than in the prototype;
- the relative surface roughness for the model blades will be greater;
- leakage losses around the blade tips of the model will be relatively greater as a result of increased relative tip clearances.

Kinematic similarity

- Concerns similarity of time intervals eg. velocities and accelerations of corresponding particles must be similar

Dynamic similarity

- Concerns similarity of forces. Eg. viscosity, differences in pressure, surface tension and so on.

There may be many different types of similarity and their associated components but all may not be required to achieve a state of compete similarity for the application at hand.
If the behaviour of the two systems, such as the prototype and a model, is governed by an equation in the form

$$\Phi(\pi_1, \pi_2, \pi_3 \ldots) = 0$$

The qualities measured during the testing of the model will depend on the values of the independent variables involved. The result will take the form:

$$\pi_1 = \Phi(\pi_2, \pi_3, \pi_4 \ldots) = 0$$

Where the $\pi$s are the dimensionless groups of variables. Tests are conducted for the model and prototype. The value of $\pi_1$ corresponds to particular values of $\pi_2, \pi_3, \pi_4$ and if each of the independent groups $\pi_2, \pi_3, \pi_4$ has the same value for the prototype as obtained for the model then the result is equally applicable to the prototype, (Massey 1979).

If the previous conditions are achieved the model and prototype are completely similar. As an example, if one of the independent groups concerns the Reynolds number then complete similarity requires:

$$\frac{u_m l_m \rho_m}{\mu_m} = \frac{u_p l_p \rho_p}{\mu_p}$$

(equality of Reynolds number)

The other common parameters such as the dimensionless mass flow rate, pressure ratio and speed present no particular problem but is often not possible to simultaneously satisfy these and the Reynolds number, (Whitfield and Baines 1990). Machines which are similar in these respects form a homologous series: members of such series are therefore simply enlargements or reductions of each other.

Significant deviation from complete similarity is known as a scale effect. If possible, corrections to compensate for these departures from complete similarity should be made.

Small turbomachines no matter how well designed always have lower efficiencies than their larger geometrically similar machines (members from the same homogenous series).

Experimentally developed correlations try and account for this scale effect. An empirical allowance is made for the difference in Reynolds number, (Massey 1979). However it is very important to make clear the definition of the Reynolds number, as many authors base their correlations on different forms. This makes it hard to directly compare the correlations (directly involving Re number) that have been published, (Whitfield and Baines 1990).
Various simple corrections have been devised to allow for the effects of size (or scale) on the efficiency. One of the simplest and best known is that due to (Moody, Zowski et al. 1969), also reported by (Addison 1954; Massey 1979), which, as applied to the efficiency of compressors and turbines is:

\[
\frac{1 - \eta_p}{1 - \eta_{m, ref}} = \left(\frac{D_{m, ref}}{D_p}\right)^n \tag{E-2}
\]

where the subscripts \(p\), \(m\) refer to prototype and model, and the index \(n\) is in the range from 0.2 to 0.5, which is dependent on the size and design of the turbomachinery. A comparison of field tests of large units with model tests, Moody and Zowski concluded that the best value for \(n\) was approximately 0.2 in contrary to other values used.

For rotor diameters smaller than 12.7cm (5 in), Colin Rodgers stated that the efficiency variation with size was best represented by the following expression:

\[
\frac{1 - \eta}{1 - \eta_{ref}} = \left(\frac{5.0}{D}\right)^n \tag{E-3}
\]

Where \(n=0.5\) for small compressors and turbines. The diameter is measured in inches and the efficiency as a percentage. The examination of the test performances studied in Colin’s paper reported that the effects due to size were moderate and best represented by Equation E-2. The more significant size effects were best represented by Equation E-3, (Rodgers 1969). It is noted that the reference turbine should be less than that of 12.7cm.

This important correlation takes into account the cumulative effect of the deviation from complete similarity and the various types of energy losses including hydraulic.

A more detailed and complicated form of the equations as reported by (Holeski and Futral 1967) and (Nusbaum 1967) proposed that the overall loss be divided into viscous friction and other losses which were assumed to be unaffected by Reynolds number. The expression was correlated:
\[
\frac{1 - \eta}{1 - \eta_{ref}} = K + (1 - K) \left( \frac{Re}{Re_{ref}} \right)^{-0.2}
\]

\( K \) is a constant which depends on the assumed split between the two types of loss. The values of \( K \) are between 0.3-0.4, (Whitfield and Baines 1990). The definition of the Reynolds number used is \( \frac{m}{\mu r_T} \), where \( \dot{m} \) is the mass flow rate, \( r_T \) is the tip radius and \( \mu \) is the dynamic viscosity at rotor inlet.

All the above equations do not take into account the losses due to the tip clearance nor due to high compressor ratio. The relations for these will be given below.

**E.5.1 Tip Clearance Empirical Relation**

Any tip clearance increase reduces efficiency, and for compressors it may significantly lower the surge line. For turbines the effect is simply on the efficiency. More sophisticated engines incorporate active tip clearance control, where the cooling air to the turbine castings is metered to control thermal growths and hence tip clearances, (Rodgers 1969). According to Rodgers, extensive development on small gas turbines has shown that it is difficult to maintain clearances less than about 0.4mm

\[
\Delta \eta_c = 0.20 * \left( \frac{C_c}{h_c} - 0.02 \right) \ll 0 \quad \text{E-4}
\]

\[
\Delta \eta_t = 0.1 \frac{C_t}{h_t} \quad \text{E-5}
\]

The losses resulting from clearance gaps are basically related to the ratio of the effective clearance gap to blade height. It can be seen that the turbine efficiency increases as the clearance gap is reduced. Manufacturing considerations and transient excursion show that it is difficult to maintain compressor and turbines clearances on radial components below 0.178 and 0.381 mm, respectively, (Rodgers 1969).
E.5.1.1 Cost

Major cost contributors are the controls, sensors, accessories, gearbox, or inverter for a high speed direct drive unit. Recuperation can easily double the powerhead cost depending upon the degree of recuperation. It may become obvious that converting a turbocharger for use as a microturbine may provide little cost advantage. If the costs of the auxiliary equipment can also be reduced then maybe this may become a viable option, (Rodgers 1997).

E.5.2 Mach number and Pressure Ratio Empirical Relation

Scale effects aren’t the only causes of component efficiency losses. There is also a loss due to high pressure ratios however increasing the cycle pressure ratio will result in an improvement in fuel economy. Compressor surge and engine matching do not always allow operation at peak compressor efficiency. It is important to note that engine design point compressor efficiency may be around 2% lower than peak efficiency however this depends on compressor flow range characteristics.

As a result of these factors the decrease of compressor efficiency with increasing pressure ratio can be approximated by the empirical relationship

\[
\Delta \eta_c = \frac{\text{Constant}}{\sqrt{\gamma c R}} \times \left( \frac{P_2}{P_1} - 2.0 \right)
\]

The value of the constant for current small compressor designs with zero inlet air pre-rotation is of the order 0.13. Small radial turbines do are not prone to significant Mach number and pressure ratio penalties providing pressure ratios are lower than approximately 5.0. R is the gas constant in ft lb/lb deg R and is 53.3 for air.

E.6 Empirical Correlations adapted to Experimental Data

E.6.1 Experimental Turbocharger Map Design points (Compressor and Turbine)

Copious amounts of turbocharger maps were collected and their maximum efficiency and corresponding mass flow rate and pressure ratio were noted and graphed. A tread line was plotted
to get an idea of the trends associated with efficiency and mass flow rate variation. The size correlations where then superimposed on top of the graph and corrected to the experimental data.

**E.6.2 Introduction**

**E.6.2.1 Assumptions**

The reader is reminded that the data sets the correlations are fitted to come from turbocharger based applications and that the spread is a representation of different groups of manufactures. Values in the empirical correlations were modified in a way to account for the technology level increase over the next five years. The scaling effect is assisted by further incorporating the effects due to tip clearance and PR effects. The following turbocharger data sets were plotted together to form one spread. Should the data be split into manufacturing types it would offer a more accurate representation of the trends. However there is not much error associated with the predefined procedure as the trends from all manufactures were similar.

**E.6.2.2 Extraction of scaling values**

The value used for n was estimated between 0.15-0.5. As you decrease the value of n the line flattens out with increasing mass flow (power). Which is indicative of no scale effect (more commonly experienced by larger gas turbines). The correlations and associated constants will be further used in the GSP simulations.
E.6.3 Turbine

E.6.3.1 Representation of data

Figure E-5 gives a general representation of the scaling effects encountered in small turbomachinery. The efficiency loss due to a decrease in mass flow rate for the turbine
component is shown in the above figure. Five microturbines of power outputs 30, 25, 10, 5 and 3kW are plotted together with turbocharger data. The Capstone C30 and MTT’s 3kW CHP unit were used as reference values for the empirical correlations. The power n values were modified to try and fit the gradient of the experimental data. The values were maintained between 0.15 and 0.5 which is in agreement with literature, (Rodgers 1969; Massey 1979). The gradient of the correlations should be modified in a way that they correspond closer with the microturbine application while trying to keep a general fit between MTTs unit and the C30.

The original graph was a simple representation of efficiency vs diameter. The graph was scaled with the scale factor $d^2$. The spread of the data is better represented with a log scale and made the selection of the appropriate value for n easier. The green triangle represents the effect of tip clearance on the MTT 3kW turbine and a point decrease of 0.7 was noted. The value of n was extracted and was determined to be 0.31.

The equation takes the form

$$\frac{1 - \eta_p}{1 - \eta_m} = \left(\frac{D_m}{D_p}\right)^{0.31}$$
E.6.4 Compressor

E.6.4.1 Representation of Data

![Radial Compressor Graph](image)

Figure E-6 Radial Turbine empirical/experimental comparison

E.6.4.2 Explanation
The graph has similar qualities as that for the turbine except for the compressor the tip clearance effect is greater. This can be seen from the triangle in the graph. The value for $n$ is

The equation takes the form

$$\frac{1 - \eta_p}{1 - \eta_m} = \left(\frac{D_m}{D_p}\right)^{0.15}$$

**E.6.4.3 Efficiency decrease due to increasing PR**

Figure E-7 Effect of PR on the compressor isentropic efficiency

**E.6.4.4 Explanation**
You see that at very high PR's efficiency often is less than 80% for the computer maps. However, with PR's lower than 3.5, max efficiencies are between 80 and 90% for the higher mass flow engines. For the turbocharger type the efficiency is 72-78% PR 2-3.5 and the value for the constant is 0.1. The rated PR determines efficiency. The Capstone C30 and the graph supports a reasonable assumption that we need to focus around a PR of 3.5. This will depend on the turbine due to the fact that it may not be possible to have efficient turbines for PR this high. It is expected that the GSP cycle analysis will tell the reader more.

The equation takes the form

\[ \Delta \eta_c = \frac{0.1}{\sqrt{\gamma_c R}} \left( \frac{P_2}{P_1} - 2.0 \right) \]

E.7 Generator

Figure 2-4 and Figure 2-5 show the location of the generator within the microturbine. The generator is the component that produces the electrical power needed for use in a HEV. The generator is either attached to a single shaft turbo-compressor or is on a separate shaft turning with a power turbine. The single shaft design used today incorporates a permanent magnet (typically Samarium-Cobalt) alternator and delivers high frequency AC output (about 1,600 Hz for a 30kW machine), this needs to be converted to 50-60Hz for general use. Air is drawn in through the generator cooling fins before it flows to the compressor inlet. This cools the machine as a result eliminates the need for any liquid cooling. Air from the generator then flows into the compressor and further processes in the cycle will be described in following sections. An efficiency loss of about 5-7% occurs due to the rectification and inverting process, (TechPro 2002). The gas turbine and the generator must be properly aligned and coupled, either directly or by a flexible coupling. It is critical that the engine and generator are properly matched. Improvements in magnetic materials have resulted in lighter and more efficient permanent magnet generators (PMG’s) than wound field generators. The field excitation is provided by permanent magnets that are capable of operating at temperatures up to 250°C. Power output is determined by the attainable generator tip speed, diameter and length-to-diameter ratio.
Figure E-8 gives an approximate generator sizing as limited by a rotor tip speed limit of 250 m/s. It is seen that generator tip speed, diameter, and length parameters may bracket rotating speeds between certain limits. Generator cooling and heat rejection is a major consideration and may incur additional parasitic power losses. Therefore the generator is usually air-cooled with ambient air, (McDonald and Rodgers 2007).

E.8 Bearings

Current automotive microturbines like the C-30 or C-65 Capstone units pay very close attention to bearing selection. Microturbines operate on either oil-lubricated or air bearings, which support the shaft(s). Oil bearings require an oil pump, oil filtering system, and liquid cooling system incorporated in the microturbine and so adds to the cost and maintenance. In addition, the exhaust from microturbines featuring oil-lubricated bearings may not be useable for direct space heating in cogeneration configurations due to the potential for contamination. Air bearings provide ease of operation without the cost, reliability concerns, maintenance requirements, or power drain of an oil supply and filtering system. However they do significantly lengthen microturbine startup time (one to two minutes), all Capstone’s microturbines for HEV applications for this reason use air bearings, (TechPro 2002).
Small gas turbines currently operate with rotational speeds from 60,000 to 150,000 rpm with both conventional antifriction and air bearings. Air bearings require no lubrication or associated lubrication cooling system, plus minimal parasitic drag during starting. Air bearings however possess low thrust bearing load capacity, and are sensitive to thermal gradients, and shock loading under high “g” accelerations. Air bearings do incur reduced power losses, especially the thrust bearing, which may be as large in diameter as the compressor impeller. It would be a strong advantage if future range extender microturbines incorporated air bearings.

**E.9 Combustor**

The fuel is injected into the combustor via an injector and mixes with the high-pressure air (air supplied by the compressor). Combustion takes place and the temperature of the gases is increased. (Saravanamuttoo, Rogers et al. 1996). A successful combustor design must satisfy many requirements. The basic design requirements can be classified as follows:

- High combustion efficiency at all operating conditions.
- Low levels of unburned hydrocarbons and carbon monoxide.
- High power and no visible smoke. (Minimized pollutants and emissions.)
- Low pressure drop. Three to four percent is common. (2% used in my model)
- Combustion must be stable under all operating conditions.
- Smooth combustion, with no pulsations or rough burning.
- A low temperature variation for good turbine life requirements.
- Useful life (thousands of hours), particularly for industrial and automotive use.
- Multi-fuel use. Characteristically natural gas and diesel/biodiesel fuel are used for automotive applications.
- Length and diameter compatible with engine envelope (outside dimensions).
- Designed for minimum cost, repair and maintenance; and
- Highly resistant to mechanical and thermal breakdown.

Highly efficient or *complete combustion* (stoichiometric) of the fuel reduces pollutants such as oxides of carbon (CO₂ and CO), nitrogen (NOₓ) and unburned hydrocarbons (UHC). A combustor consists of at least three basic parts: a *casing*, which encloses a relatively thin-walled
flame tube and a fuel injection system. Annular and cannular combustion chambers are designs of chambers that are appropriate for microturbine engines because of their compactness and simplicity of design, (Kolanowski 2004).

Turbomachinery aerodynamic design is guided by the classic scaling relationships for the effects of Mach and Reynolds numbers, plus the effects of surface finish and clearance gaps. Scaling techniques for the design of small combustors are less defined, due in part, to the effects of:

- Surface area / volume increase, with decreasing size.
- Increased effects of wall quenching.
- Low fuel flows necessitating minimum number of injectors and injector orifice sizing.
- Concentricity limitations between the inner and outer liners; and
- Increased effect of leakage gaps on pattern factor

As a consequence there is a reluctance to directly apply scaling from larger combustors and alternative design solutions have been considered, (Rodgers 2000).

Due to all these factors influencing the design and performance of combustion systems, it is not possible to specifically relate combustion efficiency with size. It is to be recognized that it is difficult and becomes more problematic to develop an adequate combustion system as size is diminished, (Rodgers 1969).

Despite the above issues with combustor sizing, Rodgers (1974) suggested that the key combustor sizing parameter is defined as the heat release rate (HRR):

\[
HRR = \text{Fuel flow} \times \frac{LHV}{(\text{Primary Volume} \times \text{Pressure Ratio})}
\]

“Typical HRRs for small single can combustors range from 6 to 10 million \(\frac{kJ}{m^3/\text{bar}}\) lower HRRs provide increased residence time and are conducive to reducing CO emissions. Relatively high HRRs can be obtained with catalytic combustors (CC), but they require the addition of some form of preburner (to 430°C) plus additional downstream volume for combustion completion. As a consequence, overall combustor volumes are similar to conventional fuel injection burners.
Ultra-lean burn CCs with compressor inlet injection require expensive catalysts such as platinum or palladium”, (Rodgers 1974; Rodgers, Watts et al. 2001)

E.10 Recuperator

The heat exchanger or more commonly known as the recuperator (microturbine applications) preheats the compressor discharge air by using waste heat from the engine exhaust. Transferring heat energy from the exhaust to the combustor inflow allows less fuel to be used to sustain the turbine operating temperature. The heat exchanger is the biggest physical component of the microturbine system and is subject to varying thermal stresses that reduces performance and respectively shorten its lifespan. The recuperated microturbine improves overall efficiency but influences the units durability, size and cost. There exists many different types of heat exchangers, and Capstone incorporates the basic counter flow design. The recuperator has four connections; to the compressor discharge, the expansion turbine discharge, the combustor inlet, and the system exhaust.

Incorporating a recuperator makes available more design parameters to performance-cost tradeoffs and so it is more difficult to analyse and size. The pressure ratio, effectiveness and corresponding pressure drop have to be carefully managed to reach a combination of high efficiency and optimum market opportunity (high power for low price). With increasing effectiveness, recuperation requires greater recuperator surface area, which both increases cost and incurs additional pressure drop. This increased internal pressure drop reduces net power production and consequently increases microturbine cost per kW, (TechPro 2002). It is also essential that a recuperator be compact, especially if space/weight restrictions, such as with automobiles are of a concern. Also a notable effect in regards to fuel, recuperators in microturbines running on ‘biomass based fuels’ are subjected to the formation of deposits of unburned particulate matter that distribute within the structure. This leads to a reduced effectiveness and accelerated corrosion of the recuperator material.(Shah 2005).

E.10.1 Specific size

The size of a prime surface metallic recuperator is impacted by several parameters, the recuperator matrix volume (and cost) can be considered approximately proportional to the
following group of parameters, that includes an engine power related function, heat exchanger parameters, and surface geometry characteristics:

\[
V\alpha \frac{m_a}{\sqrt{R}} * \left( \frac{\varepsilon}{1 - \varepsilon} \frac{1}{\Delta P} \right) * \left( \frac{1}{\sqrt{\Delta P}} \right) \]  

E-7

If the heat exchanger mass flow rates were kept constant, the impact of increasing the effectiveness from say 0.83 to 0.92 would result in an increase in recuperator size by a factor of about 2.3, (McDonald 2000).

It is seen that the Recuperator volume is increased by raising the effectiveness (to yield higher engine thermal efficiency), the size of the Recuperator is also affected by the reduction in airflow (the term \( m_a \) in the above equation. The selection of the actual surface geometry would be identified from detailed thermal analyses and sensitivity studies to establish an optimum recuperator design for a particular engine specification.

The Recuperator used for both the 30 and 65kW microturbines evolved from earlier work on a (around) 20kW microturbine that had been intended for application in Hybrid Vehicles. It was the automobile compact space requirements which encouraged the use of a small annular type recuperator. This configuration gives benefits of simplicity, minimal ducting and improved package integration. Capstone uses solar supplied recuperators and are the primary surface type, annular and fully welded assemblies. The resulting Recuperator weighs approx. 29.5 Kg and is 45.5cm in diameter. Recuperator cost can be a significant portion (20 to 30%) of the manufacturing cost of microturbine systems, (Treece, Vessa et al. 2002).

Assuming fixed surface geometry, these are approximately proportional to the following:

\[
\text{Matrix frontal area} \propto \bar{m} \sqrt{T_{avg}} * \sqrt{\left( \frac{\varepsilon}{1 - \varepsilon} \right)} \frac{1}{\sqrt{\Delta P}} \]

E-7
\[
\text{Matrix length} = \sqrt{\left(\frac{\varepsilon}{1 - \varepsilon}\right) \times \sqrt{\left(\frac{\Delta P}{P}\right)}} \times \frac{1}{T_{avg}}
\]

(McDonald 2000)

E.11 Power Electronics

Digital power controllers are essential on single-shaft microturbines. They are used to convert the high frequency AC power produced by the generator (due to the high rotational speeds) into usable electricity. The high frequency AC is rectified to DC, inverted back to 60 or 50 Hz AC, and then filtered to reduce harmonic distortion. An issue that is nonexistent in twin shaft models. Matching turbine output to the required load presents significant design challenges for the single shaft microturbine. The electronic components also direct all of the operating and startup functions. The mechanical characteristics of gas turbine engines are various and include very smooth operation and absence of vibration due to reciprocating action. Carefully and accurately balancing the rotor is essential to avoid damaging vibration. The high rotational speeds mean that the rotor parts are very sensitive to damages.

The components such as the electric generator, power conditioning systems, air bearings, nozzles, fuel filtration, metering and injection systems have not been analysed in this study. It is possible to add more components to the microturbine but depends on the application at hand.
Appendix F

Range extender innovations

The document below is the Range Extender Innovations and the thesis assignment is based one of the solutions for a specific part of the market.

- A virtually unlimited range, something what the consumer requires and is already accustomed to
- Less dependence of a new infrastructure with electric charging stations
- Very low emission values for NOx, CO2, and PM
- Low weight compared to batteries

Three Dutch companies develop 3 different Range Extender solutions for a specific part of the market:

- **Low-Power Range Extender (LP-RE) by Micro Turbine Technology BV (MTT):** a system that can produce <15 kW to be used to charge the batteries in an intelligent way by matching the battery load, electricity consumption, and the required distance. This RE is specifically aimed at smaller cars.
- **High-Power Range Extender (HP-RE) by Preev Power BV (PREEV):** a system that can produce 15-45 kW to be used to energize the drive train of the car and to charge the batteries at the same time.
- **Heavy-Duty Range Extender (HD-RE) by All Green Vehicles (AGV):** for heavier applications, such as vans and trucks, even higher power is required. In the range of 45-105 kW applications there is interest in the market to apply range extenders based on existing combustion engine technology.
F.1 Current “Range Extended” HEV and Companies

*Capstone Turbine Corporation* has recently partnered with *CalMotors*, manufacturer of traction drive systems. The two companies combined will be able to provide a complete line of microturbine-powered Hybrid electric drive solutions. Their produce will include small to midsized automobiles and heavy duty applications such as trucks and marine motive machines. The combined partnership will give Capstone an entry point into the growing hybrid electric vehicle market, *(Bristow 2010)*. The Capstone Drive Solution provides a great opportunity for vehicle manufacturers to integrate microturbines into a series hybrid electric drivetrains. Figure 3-4 outlines the product offering from both companies and includes inverter drives, traction motors and a vehicle power control module. All components will integrate with Capstone 30kW and 65kW microturbines.

**F.1.1 Transit Bus Applications**

Two bus companies have worked with Capstone since it went public in 2000 and have been able to penetrate the transport market using Capstone Microturbines coupled with their novel bus designs. They saw the opportunity to benefit from Capstone’s clean, low maintenance technology, and began integrating microturbines into their electric drive systems. Transit busses operate in densely populated areas and are required to operate as means of clean and quiet transportation.

*DesignLine*

A notable manufacturer called The Design Line Corporation, which is currently situated in the USA and New Zealand. DesignLine had the vision to develop a unique transit bus with more efficient operation than traditional buses. The buses are 10.5 meters long with gross vehicle weight rating of 17,200 kg, *(Capstone 2009)*.

*EcoPower Technology (EPT)*

EcoPower Technology is an Italian company that specializes in eco-friendly transportation, including transit buses. EPT buses are smaller than the DesignLine models described
above, with lengths of about 8 meters and currently operate in several cities in Italy, (Capstone 2009).

Capstone’s 30-kilowatt microturbines have since been installed in hybrid electric buses, trolleys and transit shuttles around the world. There are Hybrid buses operating today in U.S. cities like New York, Baltimore and Charlotte, and internationally in London, Tokyo, Paris, Rome and Auckland using Capstone technology.

**F.1.2 Heavy duty trucks**

![Intercity truck with Capstone and CalMotors Driveline technology](image)

*Figure F-1 Intercity truck with Capstone and CalMotors Driveline technology*

Major US truck manufacturers will utilize Capstone’s 65kW microturbine as a range extender in a series hybrid electric drive system. Class 5 to class 8 trucks will take advantage of the complete Capstone Drive Solution, (2010).
F.1.3 Some Unique HEV Applications

Private companies have taken advantage of the unique characteristics of the Capstone microturbine for specialty applications. There are two noted below.

Tomoe Mine Railcar
In the mining sector exhaust emissions can create health hazards in confined areas. There is a need for clean combustion and high energy density. Several 30kW Capstone Microturbines have been installed in mining applications by Tomoe (a Japanese company), and demonstrate the benefits of clean, reliable microturbine power.

“Spinner” Unmanned Military Vehicle
Another unique application that used Capstone microturbines was a prototype military vehicle that was to be an unmanned electric hybrid with the ability to completely flip upside down and still operate (thus the nickname the “Spinner”). Traditional reciprocating engines could not be used in this vehicle because their oil sump would not operate correctly if it was turned upside down. However, the Capstone C30 was perfect for this application, as its air bearings are not dependent on orientation.

F.1.4 Automotive Applications

Recently, Capstone has seen increased interest in using its microturbines for passenger vehicles and due to the recent partnership with CalMotors it seems likely that marketable units that will compete with conventional combustible vehicles will appear soon.

“Whisper” EcoLogic Hybrid SUV
Langford Performance Engineering in the UK received a government grant to develop a small hybrid electric passenger vehicle using the Capstone 30kW microturbine. The basic vehicle platform of choice was the Ford S-Max people carrier. It is a series Plug-in Hybrid Electric
Vehicle, the first Hybrid vehicle of its size to use Capstones C30 with a plug-in configuration. The Whisper

Figure F-2 Capstone CMT380 Microturbine Supercar Concept Vehicle

Eco-Logic vehicle is a plug in electric car with an on board microturbine to keep the batteries charged and extend the range of the car beyond that of a typical electric vehicle. It was able to be placed in the car without compromising space. The Lithium-iron batteries are sized for a 70km range before the microturbine is started to recharge them. Fuel economy has been measured at 34 kilometers per liter – and is much better than current production hybrids. This makes it different from marketable hybrids available, such as the Lexus and Toyota which use conventional 4 stroke engines to provide both vehicle drive and battery charging (Power-split driveline). C-30 liquid-fueled microturbine was successfully integrated into the “Whisper” and is registered and operating on public roads in the UK. Langford is now pursuing interest from major automobile manufacturers, (Capstone 2009).
One of Capstone’s customers also has integrated a Capstone C-30 into a high performance hybrid electric vehicle. This has challenged many as hybrid vehicles are generally associated with reduced emissions and high efficiency. The super performance CMT-380 microturbine Supercar has it all. This performance hybrid uses high energy density lithium-polymer batteries with an expected range of 130km. Energy from the batteries is supplemented by a 30kW microturbine operating on diesel or biodiesel fuel, with enough fuel storage to extend the vehicle range to an estimated 804km. This car is obviously not a worldwide marketable unit but has been successful in increasing people’s awareness and challenging expectations of this technology, (Capstone 2009).

Timing is important when introducing new technologies and HEVs are gaining interest. Many people are taking a serious look at microturbines as ideal electric vehicle battery chargers and it appears that it is the right time to supply the market with more advanced and developed models.
F.2 TNO Frame Work Simulations

**F.2.1 Vehicle weight**

The vehicle weight is to be evaluated at the following levels:

- 800 kg
- 1000 kg
- 1200 kg

**F.2.2 Battery size**

The battery size is to be evaluated at the following levels:

- 10 kWh
- 14 kWh
- 18 kWh
- 22 kWh

**F.2.3 Range extender power**

The range extender power is to be evaluated at the following levels:

- 9 kW
- 15 kW
- 22 kW
- 30 kW
- 36 kW

**F.2.4 Vehicle segments**

The low-power range extended electric vehicle is targeted at the vehicle segments listed in Table F-1. Descriptions of the vehicle segments are provided in (Wikipedia 2010).

<table>
<thead>
<tr>
<th>Segment</th>
<th>Typical drive power</th>
<th>Typical weight</th>
<th>Drive power @120 km/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>33-74</td>
<td>765-840</td>
<td>18.7</td>
</tr>
<tr>
<td>B</td>
<td>55-110</td>
<td>976-1113</td>
<td>21</td>
</tr>
</tbody>
</table>
Table F-1 Targeted vehicle segments and relevant characteristics

For range-extended electric vehicles, a large capacity battery is added to the vehicle increasing its weight substantially. The additional weight is estimated at 150 kg per 10 kWh battery capacity.